Third Edition

Fluid Mechanics

Fundamentals and Applications

INSTRUCTOR'S SOLUTIONS MANUAL

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Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

Third Edition

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CHAPTER 1

INTRODUCTION AND BASIC CONCEPTS

Introduction, Classification, and System

1-1C

Solution We are to define a fluid and how it differs between a solid and a gas.

Analysis **A substance in the liquid or gas phase is referred to as a fluid**. A fluid differs from a solid in that a solid can resist an applied shear stress by deforming, whereas **a fluid deforms continuously under the influence of shear stress**, no matter how small. A liquid takes the shape of the container it is in, and **a liquid forms a free surface in a larger container in a gravitational field**. A gas, on the other hand, expands until it encounters the walls of the container and fills the entire available space.

Discussion The subject of fluid mechanics deals with ball fluids, both gases and liquids.

1-2C

Solution We are to determine whether the flow of air over the wings of an aircraft and the flow of gases through a jet engine is internal or external.

Analysis The flow of air over the wings of an aircraft is **external** since this is an unbounded fluid flow over a surface. The flow of gases through a jet engine is **internal** flow since the fluid is completely bounded by the solid surfaces of the engine.

Discussion If we consider the entire airplane, the flow is both internal (through the jet engines) and external (over the body and wings).

1-3C

Solution We are to define incompressible and compressible flow, and discuss fluid compressibility.

Analysis A fluid flow during which the **density of the fluid remains nearly constant** is called *incompressible flow*. A flow in which **density varies significantly** is called *compressible flow*. A fluid whose density is practically independent of pressure (such as a liquid) is commonly referred to as an "incompressible fluid," although it is more proper to refer to incompressible *flow*. The flow of compressible fluid (such as air) does not necessarily need to be treated as compressible since the density of a compressible fluid may still remain nearly constant during flow – especially flow at low speeds.

Discussion It turns out that the Mach number is the critical parameter to determine whether the flow of a gas can be approximated as an incompressible flow. If Ma is less than about 0.3, the incompressible approximation yields results that are in error by less than a couple percent.

1-4C

Solution We are to define internal, external, and open-channel flows.

Analysis *External flow* is the **flow of an unbounded fluid over a surface** such as a plate, a wire, or a pipe. The flow in a pipe or duct is *internal flow* if the **fluid is completely bounded by solid surfaces**. The flow of liquids in a pipe is called *open-channel flow* if the pipe is **partially filled with the liquid and there is a free surface**, such as the flow of water in rivers and irrigation ditches.

Discussion As we shall see in later chapters, different approximations are used in the analysis of fluid flows based on their classification.

1-5C

Solution We are to define the Mach number of a flow and the meaning for a Mach number of 2.

Analysis The Mach number of a flow is defined as **the ratio of the speed of flow to the speed of sound** in the flowing fluid. **A Mach number of 2 indicate a flow speed that is twice the speed of sound in that fluid**.

Discussion Mach number is an example of a dimensionless (or nondimensional) parameter.

1-6C

Solution We are to discuss if the Mach number of a constant-speed airplane is constant.

Analysis No. The speed of sound, and thus the Mach number, changes with temperature which may change considerably from point to point in the atmosphere.

1-7C

Solution We are to determine if the flow of air with a Mach number of 0.12 should be approximated as incompressible.

Analysis Gas flows can often be approximated as incompressible if the density changes are under about 5 percent, which is usually the case when Ma < 0.3. Therefore, air flow with a Mach number of 0.12 **may be approximated as being incompressible**.

Discussion Air is of course a compressible fluid, but at low Mach numbers, compressibility effects are insignificant.

1-8C

Solution We are to define the no-slip condition and its cause.

Analysis A **fluid in direct contact with a solid surface sticks to the surface and there is no slip**. This is known as the *no-slip condition*, and it is due to the *viscosity* of the fluid.

Discussion There is no such thing as an inviscid fluid, since all fluids have viscosity.

1-9C

Solution We are to define forced flow and discuss the difference between forced and natural flow. We are also to discuss whether wind-driven flows are forced or natural.

Analysis In *forced flow*, the fluid is forced to flow over a surface or in a tube **by *external* means** such as a pump or a fan. In *natural flow*, any fluid motion is caused by natural means such as the buoyancy effect that manifests itself as the rise of the warmer fluid and the fall of the cooler fluid. **The flow caused by winds is natural flow for the earth, but it is forced flow for bodies subjected to the winds** since for the body it makes no difference whether the air motion is caused by a fan or by the winds.

Discussion As seen here, the classification of forced vs. natural flow may depend on your frame of reference.

1-10C

Solution We are to define a boundary layer, and discuss its cause.

Analysis The **region of flow** (usually near a wall) **in which the velocity gradients are significant and frictional effects are important** is called the *boundary layer*. When a fluid stream encounters a solid surface that is at rest, the fluid velocity assumes a value of zero at that surface. The velocity then varies from zero at the surface to some larger value sufficiently far from the surface. **The development of a boundary layer is caused by the *no-slip condition***.

Discussion As we shall see later, flow within a boundary layer is *rotational* (individual fluid particles rotate), while that outside the boundary layer is typically *irrotational* (individual fluid particles move, but do not rotate).

1-11C

Solution We are to discuss the differences between classical and statistical approaches.

Analysis The *classical approach* is **a *macroscopic* approach**, based on experiments or analysis of the gross behavior of a fluid, without knowledge of individual molecules, whereas the *statistical approach* is a ***microscopic* approach** based on the average behavior of large groups of individual molecules.

Discussion The classical approach is easier and much more common in fluid flow analysis.

1-12C

Solution We are to define a steady-flow process.

Analysis A process is said to be *steady* if it involves **no changes with time** anywhere within the system or at the system boundaries.

Discussion The opposite of steady flow is *unsteady flow*, which involves changes with time.

1-13C

Solution We are to define stress, normal stress, shear stress, and pressure.

Analysis *Stress* is defined as **force per unit area**, and is determined by dividing the force by the area upon which it acts. The **normal component of a force acting on a surface per unit area** is called the *normal stress*, and the **tangential component of a force acting on a surface per unit area** is called *shear stress*. In a fluid at rest, the normal stress is called *pressure*.

Discussion Fluids in motion may have both shear stresses and additional normal stresses besides pressure, but when a fluid is at rest, the only normal stress is the pressure, and there are no shear stresses.

1-14C

Solution We are to discuss how to select system when analyzing the acceleration of gases as they flow through a nozzle.

Analysis When analyzing the acceleration of gases as they flow through a nozzle, a wise choice for the system is **the volume within the nozzle**, bounded by the entire inner surface of the nozzle and the inlet and outlet cross-sections. This is a **control volume (or open system)** since mass crosses the boundary.

Discussion It would be much more difficult to follow a chunk of air as a closed system as it flows through the nozzle.

1-15C

Solution We are to discuss when a system is considered closed or open.

Analysis Systems may be considered to be *closed* or *open*, depending on whether a fixed mass or a volume in space is chosen for study. A *closed system* (also known as a *control mass* or simply a *system*) consists of a **fixed amount of mass, and no mass can cross its boundary**. An *open system***,** or a *control volume*, is a **selected region in space**. Mass may cross the boundary of a control volume or open system

Discussion In thermodynamics, it is more common to use the terms *open system* and *closed system*, but in fluid mechanics, it is more common to use the terms *system* and *control volume* to mean the same things, respectively.

1-16C

Solution We are to discuss how to select system for the operation of a reciprocating air compressor.

Analysis We would most likely take the system as **the air contained in the piston-cylinder device**. This system is a **closed or fixed mass system when it is compressing and no mass enters or leaves it**. However, it is an **open system during intake or exhaust**.

Discussion In this example, the system boundary is the same for either case – closed or open system.

1-17C

Solution We are to define system, surroundings, and boundary.

Analysis A *system* is defined as a **quantity of matter or a region in space chosen for study**. The mass or **region outside the system** is called the *surroundings***.** The real or imaginary **surface that separates the system from its surroundings** is called the *boundary*.

Discussion Some authors like to define *closed systems* and *open systems*, while others use the notation "system" to mean a closed system and "control volume" to mean an open system. This has been a source of confusion for students for many years. [See the next question for further discussion about this.]

Mass, Force, and Units

1-18C

Solution We are to explain why the light-year has the dimension of length.

Analysis In this unit, the word *light* refers to the speed of light. The light-year unit is then the product of a velocity and time. Hence, this product forms a distance dimension and unit.

1-19C

Solution We are to discuss the difference between kg-mass and kg-force.

Analysis The unit *kilogram* (kg) is the **mass unit in the SI system**, and it is sometimes called *kg-mass*, whereas *kg-force* (kgf) is a **force unit**. One kg-force is the force required to accelerate a 1-kg mass by 9.807 m/s^2. In other words, the weight of 1-kg mass at sea level on earth is 1 kg-force.

Discussion It is *not* proper to say that one kg-mass is equal to one kg-force since the two units have different dimensions.

1-20C

Solution We are to discuss the difference between pound-mass and pound-force.

Analysis *Pound-mass* lbm is the **mass unit in English system** whereas *pound-force* lbf is the **force unit in the English system**. One pound-force is the force required to accelerate a mass of 32.174 lbm by 1 ft/s^2. In other words, the weight of a 1-lbm mass at sea level on earth is 1 lbf.

Discussion It is *not* proper to say that one lbm is equal to one lbf since the two units have different dimensions.

1-21C

Solution We are to discuss the difference between pound-mass (lbm) and pound-force (lbf).

Analysis The "pound" mentioned here must be "**lbf**" since thrust is a force, and the lbf is the force unit in the English system.

Discussion You should get into the habit of *never* writing the unit "lb", but always use either "lbm" or "lbf" as appropriate since the two units have different dimensions.

1-22C

Solution We are to calculate the net force on a car cruising at constant velocity.

Analysis There is no acceleration (car moving at constant velocity), thus **the net force is zero in both cases**.

Discussion By Newton's second law, the force on an object is directly proportional to its acceleration. If there is zero acceleration, there must be zero net force.

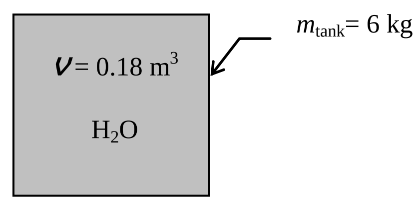
1-23

Solution A plastic tank is filled with water. The weight of the combined system is to be determined.

Assumptions The density of water is constant throughout.

Properties The density of water is given to be $\rho = 1000$ kg/m^3.

Analysis The mass of the water in the tank and the total mass are



$$m_w = \rho V = (1000 \text{ kg/m}^3)(0.18 \text{ m}^3) = 180 \text{ kg}$$

$$m_{\text{total}} = m_w + m_{tank} = 180 + 6 = 186 \text{ kg}$$

Thus,

$$W = mg = (186 \text{ kg})(9.81 \text{ m/s}^2)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{1825 N}$$

Discussion Note the unity conversion factor in the above equation.

1-24

Solution The mass of an object is given. Its weight is to be determined.

Analysis Applying Newton's second law, the weight is determined to be

$$W = mg = (200 \text{ kg})(9.6 \text{ m/s}^2) = \mathbf{1920 N}$$

1-25

Solution The mass of a substance is given. Its weight is to be determined in various units.

Analysis Applying Newton's second law, the weight is determined in various units to be

$$W = mg = (1 \text{ kg})(9.81 \text{ m/s}^2)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{9.81 N}$$

$$W = mg = (1 \text{ kg})(9.81 \text{ m/s}^2)\left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{0.00981 kN}$$

$$W = mg = (1 \text{ kg})(9.81 \text{ m/s}^2) = \mathbf{1 kg\cdot m/s^2}$$

$$W = mg = (1 \text{ kg})(9.81 \text{ m/s}^2)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kgf}}{9.81 \text{ N}}\right) = \mathbf{1 kgf}$$

$$W = mg = (1 \text{ kg})\left(\frac{2.205 \text{ lbm}}{1 \text{ kg}}\right)(32.2 \text{ ft/s}^2) = \mathbf{71 lbm\cdot ft/s^2}$$

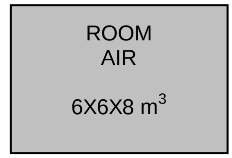
$$W = mg = (1 \text{ kg})\left(\frac{2.205 \text{ lbm}}{1 \text{ kg}}\right)(32.2 \text{ ft/s}^2)\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{2.21 lbf}$$

1-26

Solution The interior dimensions of a room are given. The mass and weight of the air in the room are to be determined.

Assumptions The density of air is constant throughout the room.

Properties The density of air is given to be $\rho = 1.16$ kg/m^3.



Analysis The mass of the air in the room is

$$m = \rho V = (1.16 \text{ kg/m}^3)(6 \times 6 \times 8 \text{ m}^3) = 334.1 \text{ kg} \cong \mathbf{334 \text{ kg}}$$

Thus,

$$W = mg = (334.1 \text{ kg})(9.81 \text{ m/s}^2)\left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2}\right) = 3277 \text{ N} \cong \mathbf{3280 \text{ N}}$$

Discussion Note that we round our final answers to three significant digits, but use extra digit(s) in intermediate calculations. Considering that the mass of an average man is about 70 to 90 kg, the mass of air in the room is probably larger than you might have expected.

1-27

Solution During an analysis, a relation with inconsistent units is obtained. A correction is to be found, and the probable cause of the error is to be determined.

Analysis The two terms on the right-hand side of the equation

E = 16 kJ + 7 kJ/kg

do not have the same units, and therefore they cannot be added to obtain the total energy. **Multiplying the last term by mass will eliminate the kilograms in the denominator, and the whole equation will become dimensionally homogeneous**; that is, every term in the equation will have the same unit.

Discussion Obviously this error was caused by forgetting to multiply the last term by mass at an earlier stage.

1-28E

Solution An astronaut takes his scales with him to the moon. It is to be determined how much he weighs on the spring and beam scales on the moon.

Analysis

(*a*) A spring scale measures weight, which is the local gravitational force applied on a body:

$$W = mg = (195 \text{ lbm})(5.48 \text{ ft/s}^2)\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm} \cdot \text{ft/s}^2}\right) = \mathbf{33.2 \text{ lbf}}$$

(*b*) A beam scale compares masses and thus is not affected by the variations in gravitational acceleration. The beam scale reads what it reads on earth,

$$W = \mathbf{195 \text{ lbf}}$$

Discussion The beam scale may be marked in units of weight (lbf), but it really compares mass, not weight. Which scale would you consider to be more accurate?

1-29

Solution The acceleration of an aircraft is given in g's. The net upward force acting on a man in the aircraft is to be determined.

Analysis From Newton's second law, the applied force is

$$F = ma = m(6\ \mathrm{g}) = (90\ \mathrm{kg})(6 \times 9.81\ \mathrm{m/s^2})\left(\frac{1\ \mathrm{N}}{1\ \mathrm{kg \cdot m/s^2}}\right) = 5297\ \mathrm{N} \cong \mathbf{5300\ N}$$

where we have rounded off the final answer to three significant digits.

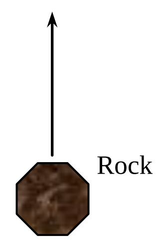
Discussion The man feels like he is six times heavier than normal. You get a similar feeling when riding an elevator to the top of a tall building, although to a much lesser extent.

1-30

Solution A rock is thrown upward with a specified force. The acceleration of the rock is to be determined.

Analysis The weight of the rock is

$$W = mg = (5\ \mathrm{kg})(9.79\ \mathrm{m/s^2})\left(\frac{1\ \mathrm{N}}{1\ \mathrm{kg \cdot m/s^2}}\right) = 48.95\ \mathrm{N} \cong \mathbf{49.0\ N}$$



Then the net force that acts on the rock is

$$F_{net} = F_{up} - F_{down} = 150 - 48.95 = 101.05\ \mathrm{N}$$

From Newton's second law, the acceleration of the rock becomes

$$a = \frac{F}{m} = \frac{101.05\ \mathrm{N}}{5\ \mathrm{kg}}\left(\frac{1\ \mathrm{kg \cdot m/s^2}}{1\ \mathrm{N}}\right) = \mathbf{20.2\ m/s^2}$$

Discussion This acceleration is more than twice the acceleration at which it would fall (due to gravity) if dropped.

1-31

Solution The previous problem is recalculated using EES. The entire EES solution is to be printed out, including the numerical results with proper units.

Analysis The EES *Equations* window is printed below, followed by the *Solution* window.

```
W=m*g "[N]"
m=5 [kg]
g=9.79 [m/s^2]
"The force balance on the rock yields the net force acting on the rock as"
F_net = F_up - F_down "[N]"
F_up=150 [N]
F_down=W "[N]"
"The acceleration of the rock is determined from Newton's second law."
F_net=a*m
"To Run the program, press F2 or click on the calculator icon from the Calculate menu"
```

SOLUTION

```
Variables in Main
a=20.21 [m/s^2]
F_down=48.95 [N]
F_net=101.1 [N]
F_up=150 [N]
g=9.79 [m/s^2]
m=5 [kg]
W=48.95 [N]
```

The final results are ***W*** **= 49.0 N** and ***a*** **= 20.2 m/s^2**, to three significant digits, which agree with the results of the previous problem.

Discussion Items in quotation marks in the EES Equation window are comments. Units are in square brackets.

1-32

Solution Gravitational acceleration g and thus the weight of bodies decreases with increasing elevation. The percent reduction in the weight of an airplane cruising at 13,000 m is to be determined.

Properties The gravitational acceleration g is 9.807 m/s^2 at sea level and 9.767 m/s^2 at an altitude of 13,000 m.

Analysis Weight is proportional to the gravitational acceleration g, and thus the percent reduction in weight is equivalent to the percent reduction in the gravitational acceleration, which is determined from

$$\%\text{ Reduction in weight} = \%\text{ Reduction in } g = \frac{\Delta g}{g} \times 100 = \frac{9.807 - 9.767}{9.807} \times 100 = \mathbf{0.41\%}$$

Therefore, **the airplane and the people in it will weigh 0.41% less at 13,000 m altitude**.

Discussion Note that the weight loss at cruising altitudes is negligible. Sorry, but flying in an airplane is not a good way to lose weight. The best way to lose weight is to carefully control your diet, and to exercise.

1-33

Solution The variation of gravitational acceleration above sea level is given as a function of altitude. The height at which the weight of a body decreases by 1% is to be determined.

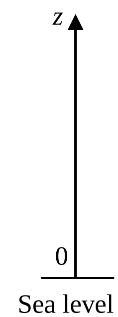
Analysis The weight of a body at the elevation z can be expressed as

$$W = mg = m(a - bz)$$

where $a = g_s = 9.807\ \text{m/s}^2$ is the value of gravitational acceleration at sea level and $b = 3.32\times10^{-6}\ \text{s}^{-2}$. In our case,

$$W = m(a - bz) = 0.99W_s = 0.99mg_s$$

We cancel out mass from both sides of the equation and solve for z, yielding



$$z = \frac{a - 0.99g_s}{b}$$

Substituting,

$$z = \frac{9.807\ \text{m/s}^2 - 0.99\left(9.807\ \text{m/s}^2\right)}{3.32\times10^{-6}\ \text{1/s}^2} = 29{,}539\ \text{m} \cong \mathbf{29{,}500\ m}$$

where we have rounded off the final answer to three significant digits.

Discussion This is more than three times higher than the altitude at which a typical commercial jet flies, which is about 30,000 ft (9140 m). So, flying in a jet is not a good way to lose weight – diet and exercise are always the best bet.

1-34

Solution A resistance heater is used to heat water to desired temperature. The amount of electric energy used in kWh and kJ are to be determined.

Analysis The resistance heater consumes electric energy at a rate of 4 kW or 4 kJ/s. Then the total amount of electric energy used in 2 hours becomes

Total energy = (Energy per unit time)(Time interval)
= (4 kW)(2 h)
= **8 kWh**

Noting that 1 kWh = (1 kJ/s)(3600 s) = 3600 kJ,

Total energy = (8 kWh)(3600 kJ/kWh)
= **28,800 kJ**

Discussion Note kW is a unit for power whereas kWh is a unit for energy.

1-35

Solution A gas tank is being filled with gasoline at a specified flow rate. Based on unit considerations alone, a relation is to be obtained for the filling time.

Assumptions Gasoline is an incompressible substance and the flow rate is constant.

Analysis The filling time depends on the volume of the tank and the discharge rate of gasoline. Also, we know that the unit of time is 'seconds'. Therefore, the independent quantities should be arranged such that we end up with the unit of seconds. Putting the given information into perspective, we have

$$t\ [\text{s}] \quad \leftrightarrow \quad V\ [\text{L}],\ \text{and}\ \dot{V}\ [\text{L/s}\}$$

It is obvious that the only way to end up with the unit "s" for time is to divide the tank volume by the discharge rate. Therefore, the desired relation is

$$\boxed{t = \frac{V}{\dot{V}}}$$

Discussion Note that this approach may not work for cases that involve dimensionless (and thus unitless) quantities.

1-36

Solution A pool is to be filled with water using a hose. Based on unit considerations, a relation is to be obtained for the volume of the pool.

Assumptions Water is an incompressible substance and the average flow velocity is constant.

Analysis The pool volume depends on the filling time, the cross-sectional area which depends on hose diameter, and flow velocity. Also, we know that the unit of volume is m^3. Therefore, the independent quantities should be arranged such that we end up with the unit of seconds. Putting the given information into perspective, we have

$$V\ [\text{m}^3] \quad \text{is a function of} \quad t\ [\text{s}],\ D\ [\text{m}],\ \text{and}\ V\ [\text{m/s}\}$$

It is obvious that the only way to end up with the unit "m^3" for volume is to multiply the quantities t and V with the square of D. Therefore, the desired relation is

$$\boldsymbol{V = CD^2Vt}$$

where the constant of proportionality is obtained for a round hose, namely, $C = \pi/4$ so that $\boldsymbol{V = (\pi D^2/4)Vt}$.

Discussion Note that the values of dimensionless constants of proportionality cannot be determined with this approach.

1-37

Solution It is to be shown that the power needed to accelerate a car is proportional to the mass and the square of the velocity of the car, and inversely proportional to the time interval.

Assumptions The car is initially at rest.

Analysis The power needed for acceleration depends on the mass, velocity change, and time interval. Also, the unit of power $\dot{W}$ is watt, W, which is equivalent to

$$\text{W} = \text{J/s} = \text{N·m/s} = (\text{kg·m/s}^2)\text{m/s} = \text{kg·m}^2/\text{s}^3$$

Therefore, the independent quantities should be arranged such that we end up with the unit kg·m^2/s^3 for power. Putting the given information into perspective, we have

$$\dot{W}\ [\text{ kg·m}^2/\text{s}^3] \text{ is a function of } m \text{ [kg]}, V \text{ [m/s], and } t \text{ [s]}$$

It is obvious that the only way to end up with the unit "kg·m^2/s^3" for power is to multiply mass with the square of the velocity and divide by time. Therefore, the desired relation is

$$\dot{W} \text{ is proportional to } mV^2/t$$

or,

$$\boxed{\dot{W} = CmV^2/t}$$

where C is the dimensionless constant of proportionality (whose value is ½ in this case).

Discussion Note that this approach cannot determine the numerical value of the dimensionless numbers involved.

1-38

Solution We are to calculate the useful power delivered by an airplane propeller.

Assumptions **1** The airplane flies at constant altitude and constant speed. **2** Wind is not a factor in the calculations.

Analysis At steady horizontal flight, the airplane's drag is balanced by the propeller's thrust. Energy is force times distance, and power is energy per unit time. Thus, by dimensional reasoning, the power supplied by the propeller must equal thrust times velocity,

$$\dot{W} = F_{\text{thrust}}V = (1500\text{ N})(70.0\text{ m/s})\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right) = \mathbf{105\,kW}\left(\frac{1.341\text{ hp}}{1\text{ kW}}\right) = \mathbf{141\,hp}$$

where we give our final answers to 3 significant digits.

Discussion We used two unity conversion ratios in the above calculation. The actual shaft power supplied by the airplane's engine will of course be larger than that calculated above due to inefficiencies in the propeller.

1-39

Solution We are to calculate lift produced by an airplane's wings.

Assumptions **1** The airplane flies at constant altitude and constant speed. **2** Wind is not a factor in the calculations.

Analysis At steady horizontal flight, the airplane's weight is balanced by the lift produced by the wings. Thus, the net lift force must equal the weight, or F_L = **1450 lbf**. We use unity conversion ratios to convert to newtons:

$$F_L = (1450\ \text{lbf})\left(\frac{1\ \text{N}}{0.22481\ \text{lbf}}\right) = \mathbf{6,450\ N}$$

where we give our final answers to 3 significant digits.

Discussion The answer is valid at any speed, since lift must balance weight in order to sustain straight, horizontal flight. As the fuel is consumed, the overall weight of the aircraft will decrease, and hence the lift requirement will also decrease. If the pilot does not adjust, the airplane will climb slowly in altitude.

1-40E

Solution We are to estimate the work required to lift a fireman, and estimate how long it takes.

Assumptions **1** The vertical speed of the fireman is constant.

Analysis

(*a*) Work *W* is a form of energy, and is equal to force times distance. Here, the force is the weight of the fireman (and equipment), and the vertical distance is Δz, where *z* is the elevation.

$$W = F\Delta z = (280\ \text{lbf})(40.0\ \text{ft})\left(\frac{1\ \text{Btu}}{778.169\ \text{ft}\cdot\text{lbf}}\right) = 14.393\ \text{Btu} \cong \mathbf{14.4\ Btu}$$

where we give our final answer to 3 significant digits, but retain 5 digits to avoid round-off error in part (*b*).

(*b*) Power is work (energy) per unit time. Assuming a constant speed,

$$\Delta t = \frac{W}{\dot{W}} = \frac{14.393\ \text{Btu}}{3.50\ \text{hp}}\left(\frac{1\ \text{hp}}{0.7068\ \text{Btu/s}}\right) = 5.8182\ \text{s} \cong \mathbf{5.82\ s}$$

Again we give our final answer to 3 significant digits.

Discussion The actual required power will be greater than calculated here, due to frictional losses and other inefficiencies in the boom's lifting system. One unity conversion ratio is used in each of the above calculations.

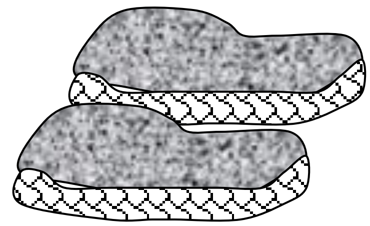
1-41

Solution A man is considering buying a 12-oz steak for \$3.15, or a 320-g steak for \$2.80. The steak that is a better buy is to be determined.

Assumptions The steaks are of identical quality.

Analysis To make a comparison possible, we need to express the cost of each steak on a common basis. We choose 1 kg as the basis for comparison. Using proper conversion factors, the unit cost of each steak is determined to be

12 ounce steak: $$\text{Unit Cost} = \left(\frac{\$3.15}{12\text{ oz}}\right)\left(\frac{16\text{ oz}}{1\text{ lbm}}\right)\left(\frac{1\text{ lbm}}{0.45359\text{ kg}}\right) = \mathbf{\$9.26/kg}$$



320 gram steak:

$$\text{Unit Cost} = \left(\frac{\$3.30}{320\text{ g}}\right)\left(\frac{1000\text{ g}}{1\text{ kg}}\right) = \mathbf{\$10.3/kg}$$

Therefore, **the steak at the traditional market is a better buy**.

Discussion Notice the unity conversion factors in the above equations.

1-42

Solution We are to calculate the volume flow rate and mass flow rate of water.

Assumptions **1** The volume flow rate, temperature, and density of water are constant over the measured time.

Properties The density of water at 20°C is $\rho = 998.0\text{ kg/m}^3$.

Analysis The volume flow rate is equal to the volume per unit time, i.e.,

$$\dot{V} = \frac{V}{\Delta t} = \frac{2.0\text{ L}}{2.85\text{ s}}\left(\frac{60\text{ s}}{1\text{ min}}\right) = 42.105\text{ L/min} \cong \mathbf{42.1\ Lpm}$$

where we give our final answer to 3 significant digits, but retain 5 digits to avoid round-off error in the second part of the problem. Since density is mass per unit volume, mass flow rate is equal to volume flow rate times density. Thus,

$$\dot{m} = \rho\dot{V} = \left(998.0\text{ kg/m}^3\right)\left(42.105\text{ L/min}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right)\left(\frac{1\text{ m}^3}{1000\text{ L}}\right) = \mathbf{0.700\ kg/s}$$

Discussion We used one unity conversion ratio in the first calculation, and two in the second. If we were interested only in the mass flow rate, we could have eliminated the intermediate calculation by solving for mass flow rate directly, i.e.,

$$\dot{m} = \rho\frac{V}{\Delta t} = \left(998.0\text{ kg/m}^3\right)\frac{2.0\text{ L}}{2.85\text{ s}}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right) = 0.700\text{ kg/s}.$$

1-43

Solution We are to estimate the work and power required to lift a crate.

Assumptions **1** The vertical speed of the crate is constant.

Properties The gravitational constant is taken as g = 9.807 m/s^2.

Analysis

(*a*) Work W is a form of energy, and is equal to force times distance. Here, the force is the weight of the crate, which is $F = mg$, and the vertical distance is Δz, where z is the elevation.

$$W = F\Delta z = mg\Delta z$$

$$= (90.5\text{ kg})(9.807\text{ m/s}^2)(1.80\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kJ}}{1000\text{ N}\cdot\text{m}}\right) = 1.5976\text{ kJ} \cong \mathbf{1.60\ kJ}$$

where we give our final answer to 3 significant digits, but retain 5 digits to avoid round-off error in part (*b*).

(*b*) Power is work (energy) per unit time. Assuming a constant speed,

$$\dot{W} = \frac{W}{\Delta t} = \frac{1.5976\text{ kJ}}{12.3\text{ s}}\left(\frac{1000\text{ W}}{1\text{ kJ/s}}\right) = 129.88\text{ W} \cong \mathbf{130\ W}$$

Again we give our final answer to 3 significant digits.

Discussion The actual required power will be greater than calculated here, due to frictional losses and other inefficiencies in the forklift system. Three unity conversion ratios are used in the above calculations.

Modeling and Solving Engineering Problems

1-44C

Solution We are to discuss choosing a model.

Analysis The right choice between a crude and complex model is usually **the *simplest* model that yields *adequate* results**. Preparing very accurate but complex models is not necessarily a better choice since such models are not much use to an analyst if they are very difficult and time consuming to solve. At a minimum, the model should reflect the essential features of the physical problem it represents. After obtaining preliminary results with the simpler model and optimizing the design, the complex, expensive model may be used for the final prediction.

Discussion Cost is always an issue in engineering design, and "adequate" is often determined by cost.

1-45C

Solution We are to discuss the difference between analytical and experimental approaches.

Analysis The *experimental approach* (**testing and taking measurements**) has the advantage of dealing with the actual physical system, and getting a physical value within the limits of experimental error. However, this approach is expensive, time consuming, and often impractical. The *analytical approach* (**analysis or calculations**) has the advantage that it is fast and inexpensive, but the results obtained are subject to the accuracy of the assumptions and idealizations made in the analysis.

Discussion Most engineering designs require both analytical and experimental components, and both are important. Nowadays, computational fluid dynamics (CFD) is often used in place of pencil-and-paper analysis and/or experiments.

1-46C

Solution We are to discuss the importance of modeling in engineering.

Analysis *Modeling* makes it possible to **predict the course of an event before it actually occurs, or to study various aspects of an event mathematically without actually running expensive and time-consuming experiments**. When preparing a mathematical model, all the variables that affect the phenomena are identified, reasonable assumptions and approximations are made, and the interdependence of these variables is studied. The relevant physical laws and principles are invoked, and the problem is formulated mathematically. Finally, the problem is solved using an appropriate approach, and the results are interpreted.

Discussion In most cases of actual engineering design, the results are verified by experiment – usually by building a prototype. CFD is also being used more and more in the design process.

1-47C

Solution We are to discuss the difference between accuracy and precision.

Analysis *Accuracy* refers to the **closeness of the measured or calculated value to the true value** whereas *precision* represents the **number of significant digits or the closeness of different measurements of the same quantity to each other**. **A measurement or calculation can be very precise without being very accurate, and vice-versa**. When measuring the boiling temperature of pure water at standard atmospheric conditions (100.00^{o}C), for example, a temperature measurement of 97.861°C is very precise, but not as accurate as the less precise measurement of 99.0°C.

Discussion Accuracy and precision are often confused; both are important for quality engineering measurements.

1-48C

Solution We are to discuss how differential equations arise in the study of a physical problem.

Analysis The description of most scientific problems involves equations that relate the *changes* in some key variables to each other, and the smaller the increment chosen in the changing variables, the more accurate the description. In **the limiting case of infinitesimal changes in variables**, we obtain *differential equations*, which provide precise mathematical formulations for the physical principles and laws by representing the rates of changes as *derivatives*.

Discussion As we shall see in later chapters, the differential equations of fluid mechanics are known, but very difficult to solve except for very simple geometries. Computers are extremely helpful in this area.

1-49C

Solution We are to discuss the value of engineering software packages.

Analysis *Software packages* **are of great value in engineering practice, and engineers today rely on software packages to solve large and complex problems quickly, and to perform optimization studies efficiently**. Despite the convenience and capability that engineering software packages offer, they are still just *tools*, and they cannot replace traditional engineering courses. They simply cause a shift in emphasis in the course material from mathematics to physics.

Discussion While software packages save us time by reducing the amount of number-crunching, we must be careful to understand how they work and what they are doing, or else incorrect results can occur.

1-50

Solution We are to solve a system of 3 equations with 3 unknowns using EES.

Analysis Using EES software, copy the following lines and paste on a blank EES screen to verify the solution:

```
2*x-y+z=9
3*x^2+2*y=z+2
x*y+2*z=14
```

Answers: ***x* = 1.556, *y* = 0.6254, *z* = 6.513**

Discussion To obtain the solution in EES, click on the icon that looks like a calculator, or Calculate-Solve.

1-51

Solution We are to solve a system of 2 equations and 2 unknowns using EES.

Analysis Using EES software, copy the following lines and paste on a blank EES screen to verify the solution:

```
x^3-y^2=10.5
3*x*y+y=4.6
```

Answers: $\boldsymbol{x = 2.215}$, $\boldsymbol{y = 0.6018}$

Discussion To obtain the solution in EES, click on the icon that looks like a calculator, or Calculate-Solve.

1-52

Solution We are to determine a positive real root of the following equation using EES: $3.5x^3 - 10x^{0.5} - 3x = -4$.

Analysis Using EES software, copy the following lines and paste on a blank EES screen to verify the solution:

```
3.5*x^3-10*x^0.5-3*x = -4
```

Answer: $\boldsymbol{x = 1.554}$

Discussion To obtain the solution in EES, click on the icon that looks like a calculator, or Calculate-Solve.

1-53

Solution We are to solve a system of 3 equations with 3 unknowns using EES.

Analysis Using EES software, copy the following lines and paste on a blank EES screen to verify the solution:

```
x^2*y-z=1.5
x-3*y^0.5+x*z=-2
x+y-z=4.2
```

Answers: $\boldsymbol{x = 0.9149}$, $\boldsymbol{y = 10.95}$, $\boldsymbol{z = 7.665}$

Discussion To obtain the solution in EES, click on the icon that looks like a calculator, or Calculate-Solve.

Review Problems

1-54

Solution The thrust developed by the jet engine of a Boeing 777 is given to be 85,000 pounds. This thrust is to be expressed in N and kgf.

Analysis Noting that 1 lbf = 4.448 N and 1 kgf = 9.81 N, the thrust developed is expressed in two other units as

Thrust in N:

$$\text{Thrust} = (85{,}000\ \text{lbf})\left(\frac{4.448\ \text{N}}{1\ \text{lbf}}\right) = \mathbf{3.78 \times 10^5\ N}$$

Thrust in kgf:

$$\text{Thrust} = (37.8 \times 10^5\ \text{N})\left(\frac{1\ \text{kgf}}{9.81\ \text{N}}\right) = \mathbf{3.85 \times 10^4\ kgf}$$

Discussion Because the gravitational acceleration on earth is close to 10 m/s^2, it turns out that the two force units N and kgf differ by nearly a factor of 10. This can lead to confusion, and we recommend that you do not use the unit kgf.

1-55

Solution The gravitational acceleration changes with altitude. Accounting for this variation, the weights of a body at different locations are to be determined.

Analysis The weight of an 80-kg man at various locations is obtained by substituting the altitude *z* (values in m) into the relation

$$W = mg = (80\ \text{kg})(9.807 - 3.32 \times 10^{-6} z)\left(\frac{1\ \text{N}}{1\ \text{kg} \cdot \text{m/s}^2}\right) \quad \left(\text{where } z \text{ is in units of m/s}^2\right)$$

Sea level: (z = 0 m): W = 80×(9.807-3.32x10^{-6}×0) = 80×9.807 = **784.6 N**

Denver: (z = 1610 m): W = 80×(9.807-3.32x10^{-6}×1610) = 80×9.802 = **784.2 N**

Mt. Ev.: (z = 8848 m): W = 80×(9.807-3.32x10^{-6}×8848) = 80×9.778 = **782.2 N**

Discussion We report 4 significant digits since the values are so close to each other. The percentage difference in weight from sea level to Mt. Everest is only about -0.3%, which is negligible for most engineering calculations.

1-56E

Solution We are to estimate the rate of heat transfer into a room and the cost of running an air conditioner for one hour.

Assumptions **1** The rate of heat transfer is constant. **2** The indoor and outdoor temperatures do not change significantly during the hour of operation.

Analysis

(*a*) In one hour, the air conditioner supplies 5,000 Btu of cooling, but runs only 60% of the time. Since the indoor and outdoor temperatures remain constant during the hour of operation, the average rate of heat transfer into the room is the same as the average rate of cooling supplied by the air conditioner. Thus,

$$\dot{Q} = \frac{0.60(5000\text{ Btu})}{1\text{ h}} = \mathbf{3,000\ Btu/h}\left(\frac{1\text{ kW}}{3412.14\text{ Btu/h}}\right) = \mathbf{0.879\ kW}$$

(*b*) Energy efficiency ratio is defined as the amount of heat removed from the cooled space in Btu for 1 Wh (watt-hour) of electricity consumed. Thus, for every Wh of electricity, this particular air conditioner removes 9.0 Btu from the room. To remove 3,000 Btu in one hour, the air conditioner therefore consumes 3,000/9.0 = 333.33 Wh = 0.33333 kWh of electricity. At a cost of 7.5 cents per kWh, it costs only **2.50 cents** to run the air conditioner for one hour.

Discussion Notice the unity conversion ratio in the above calculation. We also needed to use some common sense and dimensional reasoning to come up with the appropriate calculations. While this may seem very cheap, if this air conditioner is run at these conditions continuously for one month, the electricity will cost ($0.025/h) (24 h/day) (30 day/mo) = $18/mo.

1-57

Solution The constants appearing dynamic viscosity relation for methanol are to be determined using the data in Table A-7.

Analysis Using the data from Table A–7, we have

$$5.857 \times 10^{-4} = a10^{b/(293-c)}$$

$$4.460 \times 10^{-4} = a10^{b/(313-c)}$$

$$3.510 \times 10^{-4} = a10^{b/(333-c)}$$

which is a nonlinear system of three algebraic equations. Using EES or any other computer code, one finds

$$a = 8.493 \times 10^{-6}\text{Pa}\cdot\text{s} \qquad b = 534.5\text{ K} \qquad c = 2.27\text{ K}$$

Then the viscosity correlation for methanol becomes

$$\mu = (8.493 \times 10^{-6}) \times 10^{534.5/(T-2.27)}$$

For $T = 50\ °\text{C} = 323\text{ K}$ the correlation gives $\mu = 3.941 \times 10^{-4}\text{Pa}\cdot\text{s}$, which is nicely agreeing with the data in Table A–7.

1-58

Solution A relation for the terminal settling velocity of a solid particle is given. The dimension of a parameter in the relation is to be determined.

Analysis We have the dimensions for each term except F_L.

$$[g] = [LT^{-2}]$$
$$[D] = [L]$$
$$[V_L] = [LT^{-1}]$$

and

$$[LT^{-1}] = [F_L][2LT^{-2}]^{1/2}[L]^{1/2} = \sqrt{2}[F_L][LT^{-1}]$$

Therefore F_L is a dimensionless coefficient that is this equation is dimensionally homogeneous, and should hold for any unit system.

1-59

Solution The flow of air through a wind turbine is considered. Based on unit considerations, a proportionality relation is to be obtained for the mass flow rate of air through the blades.

Assumptions Wind approaches the turbine blades with a uniform velocity.

Analysis The mass flow rate depends on the air density, average wind velocity, and the cross-sectional area which depends on hose diameter. Also, the unit of mass flow rate $\dot{m}$ is kg/s. Therefore, the independent quantities should be arranged such that we end up with the proper unit. Putting the given information into perspective, we have

$\dot{m}$ [kg/s] is a function of ρ [kg/m^3], D [m], and V [m/s}

It is obvious that the only way to end up with the unit "kg/s" for mass flow rate is to multiply the quantities ρ and V with the square of D. Therefore, the desired proportionality relation is

$\dot{m}$ is proportional to $\rho D^2 V$

or,

$$\dot{m} = C\rho D^2 V$$

Where the constant of proportionality is $C = \pi/4$ so that $\dot{m} = \rho(\pi D^2/4)V$

Discussion Note that the dimensionless constants of proportionality cannot be determined with this approach.

1-60

Solution A relation for the air drag exerted on a car is to be obtained in terms of on the drag coefficient, the air density, the car velocity, and the frontal area of the car.

Analysis The drag force depends on a dimensionless drag coefficient, the air density, the car velocity, and the frontal area. Also, the unit of force F is newton N, which is equivalent to kg·m/s^2. Therefore, the independent quantities should be arranged such that we end up with the unit kg·m/s^2 for the drag force. Putting the given information into perspective, we have

$$F_D \text{ [kg·m/s}^2] \leftrightarrow C_{\text{Drag}} [], A_{\text{front}} [\text{m}^2], \rho \text{ [kg/m}^3], \text{ and } V \text{ [m/s]}$$

It is obvious that the only way to end up with the unit "kg·m/s^2" for drag force is to multiply mass with the square of the velocity and the fontal area, with the drag coefficient serving as the constant of proportionality. Therefore, the desired relation is

$$F_D = C_{\text{Drag}} \rho A_{\text{front}} V^2$$

Discussion Note that this approach is not sensitive to dimensionless quantities, and thus a strong reasoning is required.

Fundamentals of Engineering (FE) Exam Problems

1-61

The speed of an aircraft is given to be 260 m/s in air. If the speed of sound at that location is 330 m/s, the flight of aircraft is

(*a*) Sonic (*b*) Subsonic (*c*) Supersonic (*d*) Hypersonic

Answer (*b*) Subsonic

1-62

The speed of an aircraft is given to be 1250 km/h. If the speed of sound at that location is 315 m/s, the Mach number is

(*a*) 0.5 (*b*) 0.85 (*c*) 1.0 (*d*) 1.10 (*e*) 1.20

Answer (*d*) 1.10

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel=1250 [km/h]*Convert(km/h, m/s)
c=315 [m/s]
Ma=Vel/c
```

1-63

If mass, heat, and work are not allowed to cross the boundaries of a system, the system is called

(*a*) Isolated (*b*) Isothermal (*c*) Adiabatic (*d*) Control mass (*e*) Control volume

Answer (*a*) Isolated

1-64

The weight of a 10-kg mass at sea level is

(*a*) 9.81 N (*b*) 32.2 kgf (*c*) 98.1 N (*d*) 10 N (*e*) 100 N

Answer (*c*) 98.1 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=10 [kg]
g=9.81 [m/s^2]
W=m*g
```

1-65

The weight of a 1-lbm mass is

(*a*) 1 lbm·ft/s^2 (*b*) 9.81 lbf (*c*) 9.81 N (*d*) 32.2 lbf (*e*) 1 lbf

Answer (*e*) 1 lbf

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=1 [lbm]
g=32.2 [ft/s^2]
W=m*g*Convert(lbm-ft/s^2, lbf)
```

1-66

One kJ is NOT equal to

(*a*) 1 kPa·m^3 (*b*) 1 kN·m (*c*) 0.001 MJ (*d*) 1000 J (*e*) 1 kg·m^2/s^2

Answer (*e*) 1 kg·m^2/s^2

1-67

Which is a unit for the amount of energy?

(*a*) Btu/h (*b*) kWh (*c*) kcal/h (*d*) hp (*e*) kW

Answer (*b*) kWh

1-68

A hydroelectric power plant operates at its rated power of 7 MW. If the plant has produced 26 million kWh of electricity in a specified year, the number of hours the plant has operated that year is

(*a*) 1125 h (*b*) 2460 h (*c*) 2893 h (*d*) 3714 h (*e*) 8760 h

Answer (*d*) 3714 h

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

RatedPower=7000 [kW]
ElectricityProduced=26E6 [kWh]
Hours=ElectricityProduced/RatedPower

Design and Essay Problems

1-69 to 1-70

Solution Students' essays and designs should be unique and will differ from each other.

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

Third Edition

Yunus A. Çengel & John M. Cimbala

McGraw-Hill, 2013

CHAPTER 2
PROPERTIES OF FLUIDS

Density and Specific Gravity

2-1C

Solution We are to discuss the difference between mass and molar mass.

Analysis Mass m is the **actual mass in grams or kilograms**; molar mass M is the **mass per mole in grams/mol or kg/kmol**. These two are related to each other by $\boldsymbol{m = NM}$, where N is the number of moles.

Discussion Mass, number of moles, and molar mass are often confused. Molar mass is also called *molecular weight*.

2-2C

Solution We are to discuss the difference between intensive and extensive properties.

Analysis *Intensive properties* **do *not* depend on the size (extent) of the system** but *extensive properties* ***do* depend on the size (extent) of the system**.

Discussion An example of an intensive property is temperature. An example of an extensive property is mass.

2-3C

Solution We are to define specific gravity and discuss its relationship to density.

Analysis The *specific gravity***,** or *relative density***,** is defined as **the ratio of the density of a substance to the density of some standard substance at a specified temperature** (the standard is water at 4°C, for which ρ_{H2O} = 1000 kg/m^3). That is, $\boxed{SG = \rho / \rho_{H2O}}$. When specific gravity is known, density is determined from $\boxed{\rho = SG \times \rho_{H2O}}$.

Discussion Specific gravity is dimensionless and unitless [it is just a number without dimensions or units].

2-4C

Solution We are to decide if the specific weight is an extensive or intensive property.

Analysis The original specific weight is

$$\gamma_1 = \frac{W}{V}$$

If we were to divide the system into two halves, each half weighs W/2 and occupies a volume of V/2. The specific weight of one of these halves is

$$\gamma = \frac{W/2}{V/2} = \gamma_1$$

which is the same as the original specific weight. Hence, **specific weight is an *intensive property***.

Discussion If specific weight were an *extensive* property, its value for half of the system would be halved.

2-5C

Solution We are to define the state postulate.

Analysis The state postulate is expressed as: **The state of a simple compressible system is completely specified by two independent, intensive properties**.

Discussion An example of an intensive property is temperature.

2-6C

Solution We are to discuss the applicability of the ideal gas law.

Analysis A gas can be treated as an *ideal gas* when it is **at a *high temperature* and/or a *low pressure* relative to its critical temperature and pressure**.

Discussion Air and many other gases at room temperature and pressure can be approximated as ideal gases without any significant loss of accuracy.

2-7C

Solution We are to discuss the difference between R and R_u.

Analysis R_u is the ***universal gas constant*** that is the **same for all gases**, whereas R is the ***specific gas constant*** that is **different for different gases**. These two are related to each other by $\boxed{R = R_u / M}$, where M is the *molar mass* (also called the *molecular weight*) of the gas.

Discussion Since molar mass has dimensions of mass per mole, R and R_u do not have the same dimensions or units.

2-8

Solution The volume and the weight of a fluid are given. Its mass and density are to be determined.

Analysis Knowing the weight, the mass and the density of the fluid are determined to be

$$m = \frac{W}{g} = \frac{225\ \text{N}}{9.80\ \text{m/s}^2}\left(\frac{1\ \text{kg}\cdot\text{m/s}^3}{1\ \text{N}}\right) = \mathbf{23.0\ kg}$$

$$\rho = \frac{m}{V} = \frac{23.0\ \text{kg}}{24\ \text{L}} = \mathbf{0.957\ kg/L}$$

Discussion Note that mass is an intrinsic property, but weight is not.

2-9

Solution The pressure in a container that is filled with air is to be determined.

Assumptions At specified conditions, air behaves as an ideal gas.

Properties The gas constant of air is $R = 0.287 \frac{\text{kJ}}{\text{kg}\cdot\text{K}}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = 0.287 \frac{\text{kPa}\cdot\text{m}^3}{\text{kg}\cdot\text{K}}$ (see also Table A-1).

Analysis The definition of the specific volume gives

$$v = \frac{V}{m} = \frac{0.100\,\text{m}^3}{1\,\text{kg}} = 0.100\ \text{m}^3/\text{kg}$$

Using the ideal gas equation of state, the pressure is

$$Pv = RT \quad \rightarrow \quad P = \frac{RT}{v} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(27+273.15\ \text{K})}{0.100\ \text{m}^3/\text{kg}} = \mathbf{861\,kPa}$$

Discussion In ideal gas calculations, it saves time to convert the gas constant to appropriate units.

2-10E

Solution The volume of a tank that is filled with argon at a specified state is to be determined.

Assumptions At specified conditions, argon behaves as an ideal gas.

Properties The gas constant of argon is obtained from Table A-1E, R = 0.2686 psia·ft^3/lbm·R.

Analysis According to the ideal gas equation of state,

$$V = \frac{mRT}{P} = \frac{(1\,\text{lbm})(0.2686\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(100+460\,\text{R})}{200\,\text{psia}} = \mathbf{0.7521\,ft^3}$$

Discussion In ideal gas calculations, it saves time to write the gas constant in appropriate units.

2-11E

Solution The specific volume of oxygen at a specified state is to be determined.

Assumptions At specified conditions, oxygen behaves as an ideal gas.

Properties The gas constant of oxygen is obtained from Table A-1E, R = 0.3353 psia·ft^3/lbm·R.

Analysis According to the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.3353\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(80+460\,\text{R})}{40\,\text{psia}} = \mathbf{4.53\,ft^3/lbm}$$

Discussion In ideal gas calculations, it saves time to write the gas constant in appropriate units.

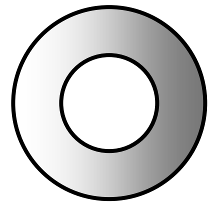
2-12E

Solution An automobile tire is under-inflated with air. The amount of air that needs to be added to the tire to raise its pressure to the recommended value is to be determined.

Assumptions **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tire remains constant.

Properties The gas constant of air is $R_u = 53.34 \dfrac{\text{ft}\cdot\text{lbf}}{\text{lbm}\cdot\text{R}}\left(\dfrac{1\text{ psia}}{144\text{ lbf/ft}^2}\right) = 0.3794 \dfrac{\text{psia}\cdot\text{ft}^3}{\text{lbm}\cdot\text{R}}$.

Analysis The initial and final absolute pressures in the tire are



$$P_1 = P_{g1} + P_{atm} = 20 + 14.6 = 34.6 \text{ psia}$$

$$P_2 = P_{g2} + P_{atm} = 30 + 14.6 = 44.6 \text{ psia}$$

Treating air as an ideal gas, the initial mass in the tire is

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(34.6\text{ psia})(2.60\text{ ft}^3)}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550\text{ R})} = 0.4416\text{ lbm}$$

Noting that the temperature and the volume of the tire remain constant, the final mass in the tire becomes

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(44.6\text{ psia})(2.60\text{ ft}^3)}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550\text{ R})} = 0.5692\text{ lbm}$$

Thus the amount of air that needs to be added is

$$\Delta m = m_2 - m_1 = 0.5692 - 0.4416 = \mathbf{0.128\,lbm}$$

Discussion Notice that *absolute* rather than gage pressure must be used in calculations with the ideal gas law.

2-13

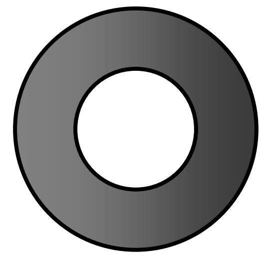
Solution An automobile tire is inflated with air. The pressure rise of air in the tire when the tire is heated and the amount of air that must be bled off to reduce the temperature to the original value are to be determined.

Assumptions **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tire remains constant.

Properties The gas constant of air is $R = 0.287 \dfrac{\text{kJ}}{\text{kg}\cdot\text{K}}\left(\dfrac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = 0.287 \dfrac{\text{kPa}\cdot\text{m}^3}{\text{kg}\cdot\text{K}}$.

Analysis Initially, the absolute pressure in the tire is

$$P_1 = P_g + P_{atm} = 210 + 100 = 310 \text{ kPa}$$



Treating air as an ideal gas and assuming the volume of the tire to remain constant, the final pressure in the tire is determined from

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{323\text{K}}{298\text{K}}(310\text{kPa}) = 336\text{kPa}$$

Thus the pressure rise is

$$\Delta P = P_2 - P_1 = 336 - 310 = \mathbf{26.0\ kPa}$$

The amount of air that needs to be bled off to restore pressure to its original value is

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(310\text{kPa})(0.025\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{K})} = 0.0906\text{kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(310\text{kPa})(0.025\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323\text{K})} = 0.0836\text{kg}$$

$$\Delta m = m_1 - m_2 = 0.0906 - 0.0836 = \mathbf{0.0070 kg}$$

Discussion Notice that *absolute* rather than gage pressure must be used in calculations with the ideal gas law.

2-14

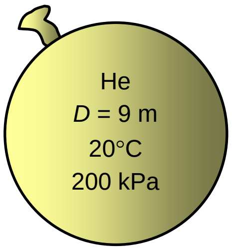
Solution A balloon is filled with helium gas. The number of moles and the mass of helium are to be determined.

Assumptions At specified conditions, helium behaves as an ideal gas.

Properties The molar mass of helium is 4.003 kg/kmol. The temperature of the helium gas is 20°C, which we must convert to absolute temperature for use in the equations: T = 20 + 273.15 = 293.15 K. The universal gas constant is $R_u = 8.31447 \frac{\text{kJ}}{\text{kmol}\cdot\text{K}}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = 8.31447 \frac{\text{kPa}\cdot\text{m}^3}{\text{kmol}\cdot\text{K}}$.

Analysis The volume of the sphere is

$$V = \frac{4}{3}\pi r^3 = \frac{4}{3}\pi(4.5\ \text{m})^3 = 381.704\ \text{m}^3$$



Assuming ideal gas behavior, the number of moles of He is determined from

$$N = \frac{PV}{R_u T} = \frac{(200\ \text{kPa})(381.704\ \text{m}^3)}{(8.31447\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293.15\ \text{K})} = 31.321\ \text{kmol} \cong \mathbf{31.3\ kmol}$$

Then the mass of He is determined from

$$m = NM = (31.321\ \text{kmol})(4.003\ \text{kg/kmol}) = 125.38\ \text{kg} \cong \mathbf{125\ kg}$$

Discussion Although the helium mass may seem large (about the mass of an adult football player!), it is much smaller than that of the air it displaces, and that is why helium balloons rise in the air.

2-15

Solution A balloon is filled with helium gas. The effect of the balloon diameter on the mass of helium is to be investigated, and the results are to be tabulated and plotted.

Properties The molar mass of helium is 4.003 kg/kmol. The temperature of the helium gas is 20°C, which we must convert to absolute temperature for use in the equations: T = 20 + 273.15 = 293.15 K. The universal gas constant is

$$R_u = 8.31447 \frac{\text{kJ}}{\text{kmol}\cdot\text{K}}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = 8.31447 \frac{\text{kPa}\cdot\text{m}^3}{\text{kmol}\cdot\text{K}}.$$

Analysis The EES *Equations* window is shown below, followed by the two parametric tables and the plot (we overlaid the two cases to get them to appear on the same plot).

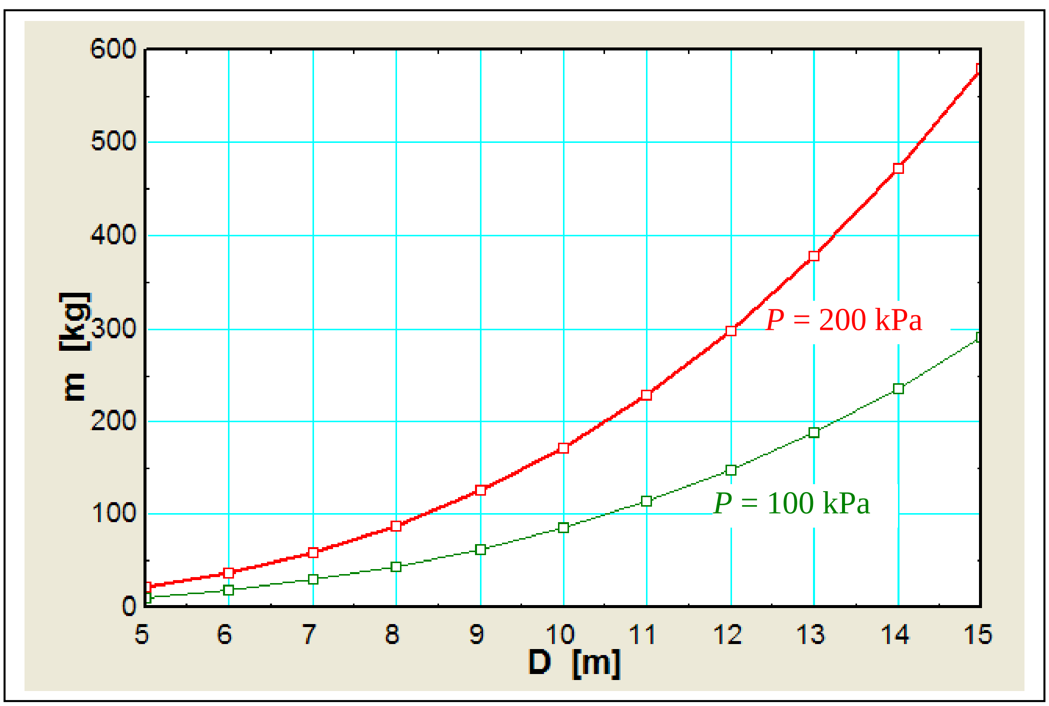
```
"Spherical balloon mass"

"Given information:"
"D      = 9.0 [m]"
T       = 20 + 273.15
P       = 100 [kPa]
R_u     = 8.31447 [kPa*m^3/(kmol*K)]
M_He  = 4.003 [kg/kmol]

"Equations:"
Vol     = pi*(D^3)/6
P*Vol = N*R_u*T
m       = N*M_He
```

P = 100 kPa:

D [m]	m [kg]
5.000	10.749
6.000	18.574
7.000	29.495
8.000	44.028
9.000	62.688
10.000	85.992
11.000	114.456
12.000	148.595
13.000	188.925
14.000	235.963
15.000	290.224

P = 200 kPa:

D [m]	m [kg]
5.000	21.498
6.000	37.149
7.000	58.991
8.000	88.056
9.000	125.377
10.000	171.985
11.000	228.912
12.000	297.190
13.000	377.850
14.000	471.926
15.000	580.448



Discussion Mass increases with diameter as expected, but not linearly since volume is proportional to D^3.

2-16

Solution A cylindrical tank contains methanol at a specified mass and volume. The methanol's weight, density, and specific gravity and the force needed to accelerate the tank at a specified rate are to be determined.

Assumptions **1** The volume of the tank remains constant.

Properties The density of water is 1000 kg/m^3.

Analysis The methanol's weight, density, and specific gravity are

$$W = mg = 40 \times 9.81 = 392.40 \text{ N}$$

$$\rho = \frac{m}{V} = \frac{40 \text{ kg}}{51 \text{ L} \times \dfrac{1 \text{ m}^3}{1000 \text{ L}}} = 784 \text{ kg/m}^3$$

$$\text{SG} = \frac{\rho}{\rho_{\text{H}_2\text{O}}} = \frac{784 \text{ kg/m}^3}{1000 \text{ kg/m}^3} = 0.784$$

The force needed to accelerate the tank at the given rate is

$$F = ma = (392.40 \text{ N}) \times \left(0.25 \frac{\text{m}}{\text{s}^2}\right) = 98.1 \text{ N}$$

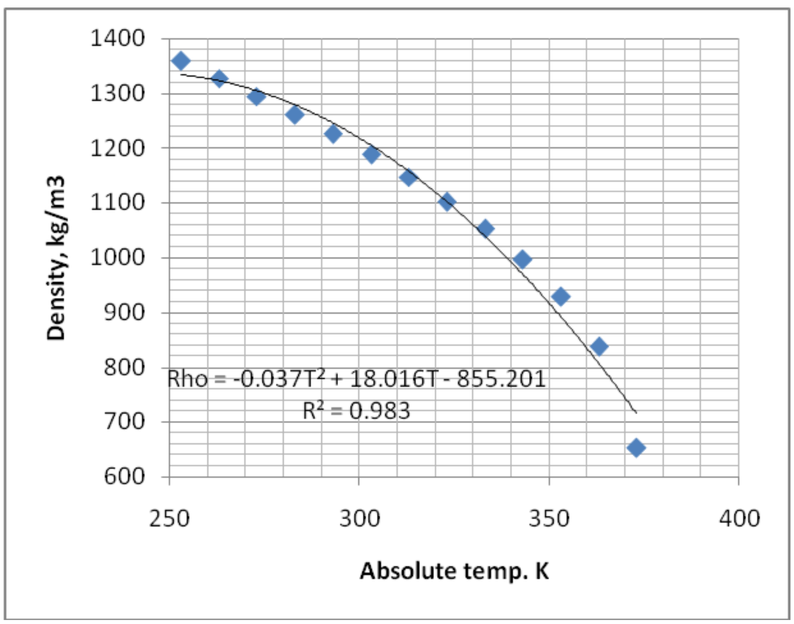
2-17

Solution Using the data for the density of R-134a in Table A-4, an expression for the density as a function of temperature in a specified form is to be obtained.

Analysis An Excel sheet gives the following results. Therefore we obtain

$$\rho\ (\mathrm{kg/m^3}) = -0.037T^2 + 18.016T - 855.201,\quad T\ (\mathrm{K})$$

Temp	Temp,K	Density	Rel. Error, %
-20	253	1359	-1.801766
-10	263	1327	-0.2446119
0	273	1295	0.8180695
10	283	1261	1.50943695
20	293	1226	1.71892333
30	303	1188	1.57525253
40	313	1147	1.04219704
50	323	1102	0.16279492
60	333	1053	-1.1173789
70	343	996.2	-2.502108
80	353	928.2	-3.693816
90	363	837.7	-3.4076638
100	373	651.7	10.0190272



The relative accuracy is quite reasonable except the last data point.

2-18E

Solution A rigid tank contains slightly pressurized air. The amount of air that needs to be added to the tank to raise its pressure and temperature to the recommended values is to be determined.

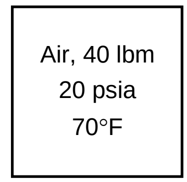
Assumptions **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tank remains constant.

Properties The gas constant of air is $R_u = 53.34\dfrac{\text{ft}\cdot\text{lbf}}{\text{lbm}\cdot\text{R}}\left(\dfrac{1\text{ psia}}{144\text{ lbf/ft}^2}\right) = 0.3794\dfrac{\text{psia}\cdot\text{ft}^3}{\text{lbm}\cdot\text{R}}$. The air temperature is $70^\circ\text{F} = 70 + 459.67 = 529.67$ R

Analysis Treating air as an ideal gas, the initial volume and the final mass in the tank are determined to be

$$V = \frac{m_1RT_1}{P_1} = \frac{(40\ \text{lbm})(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(529.67\ \text{R})}{20\ \text{psia}} = 392.380\ \text{ft}^3$$

$$m_2 = \frac{P_2V}{RT_2} = \frac{(35\ \text{psia})(392.380\ \text{ft}^3)}{(0.3704\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550\ \text{R})} = 67.413\ \text{lbm}$$



Thus the amount of air added is

$$\Delta m = m_2 - m_1 = 67.413 - 40.0 = 27.413\ \text{lbm} \cong \mathbf{27.4\ lbm}$$

Discussion As the temperature slowly decreases due to heat transfer, the pressure will also decrease.

2-19

Solution A relation for the variation of density with elevation is to be obtained, the density at 7 km elevation is to be calculated, and the mass of the atmosphere using the correlation is to be estimated.

Assumptions **1** Atmospheric air behaves as an ideal gas. **2** The earth is perfectly spherical with a radius of 6377 km at sea level, and the thickness of the atmosphere is 25 km.

Properties The density data are given in tabular form as a function of radius and elevation, where $r = z + 6377$ km:

r, km	z, km	ρ, kg/m^3
6377	0	1.225
6378	1	1.112
6379	2	1.007
6380	3	0.9093
6381	4	0.8194
6382	5	0.7364
6383	6	0.6601
6385	8	0.5258
6387	10	0.4135
6392	15	0.1948
6397	20	0.08891
6402	25	0.04008

Analysis Using EES, (1) Define a trivial function "rho= a+z" in the Equation window, (2) select new parametric table from Tables, and type the data in a two-column table, (3) select Plot and plot the data, and (4) select Plot and click on curve fit to get curve fit window. Then specify 2nd order polynomial and enter/edit equation. The results are:

$\rho(z) = a + bz + cz^2 = 1.20252 - 0.101674z + 0.0022375z^2$ for the unit of kg/m^3,
(or, $\rho(z) = (1.20252 - 0.101674z + 0.0022375z^2)\times 10^9$ for the unit of kg/km^3)

where z is the vertical distance from the earth surface at sea level. At $z = 7$ km, the equation gives $\boldsymbol{\rho}$ **= 0.600 kg/m^3**.

(b) The mass of atmosphere is evaluated by integration to be

$$m = \int_V \rho dV = \int_{z=0}^{h} (a + bz + cz^2)4\pi(r_0 + z)^2 dz = 4\pi\int_{z=0}^{h} (a + bz + cz^2)(r_0^2 + 2r_0 z + z^2)dz$$
$$= 4\pi\left[ar_0^2 h + r_0(2a + br_0)h^2/2 + (a + 2br_0 + cr_0^2)h^3/3 + (b + 2cr_0)h^4/4 + ch^5/5\right]$$

where r_0 = 6377 km is the radius of the earth, h = 25 km is the thickness of the atmosphere. Also, a = 1.20252, b = -0.101674, and c = 0.0022375 are the constants in the density function. Substituting and multiplying by the factor 10^9 to convert the density from units of kg/km^3 to kg/m^3, the mass of the atmosphere is determined to be approximately

m = **5.09×10^{18} kg**

EES Solution for final result:

```
a = 1.2025166
b = -0.10167
c = 0.0022375
r = 6377
h = 25
m = 4*pi*(a*r^2*h+r*(2*a+b*r)*h^2/2+(a+2*b*r+c*r^2)*h^3/3+(b+2*c*r)*h^4/4+c*h^5/5)*1E+9
```

Discussion At 7 km, the density of the air is approximately half of its value at sea level.

Vapor Pressure and Cavitation

2-20C

Solution We are to define and discuss cavitation.

Analysis In the flow of a liquid, *cavitation* is the **vaporization that may occur at locations where the pressure drops below the vapor pressure**. The vapor bubbles collapse as they are swept away from the low pressure regions, generating highly destructive, extremely high-pressure waves. This phenomenon is a common cause for drop in performance and even the erosion of impeller blades.

Discussion The word "cavitation" comes from the fact that a vapor bubble or "cavity" appears in the liquid. Not all cavitation is undesirable. It turns out that some underwater vehicles employ "super cavitation" on purpose to reduce drag.

2-21C

Solution We are to discuss whether the boiling temperature of water increases as pressure increases.

Analysis **Yes**. The saturation temperature of a pure substance depends on pressure; in fact, it *increases* with pressure. The higher the pressure, the higher the saturation or boiling temperature.

Discussion This fact is easily seen by looking at the saturated water property tables. Note that boiling temperature and saturation pressure at a given pressure are equivalent.

2-22C

Solution We are to determine if temperature increases or remains constant when the pressure of a boiling substance increases.

Analysis If the pressure of a substance increases during a boiling process, **the temperature also increases** since the boiling (or saturation) temperature of a pure substance depends on pressure and increases with it.

Discussion We are assuming that the liquid will continue to boil. If the pressure is increased fast enough, boiling may stop until the temperature has time to reach its new (higher) boiling temperature. A pressure cooker uses this principle.

2-23C

Solution We are to define vapor pressure and discuss its relationship to saturation pressure.

Analysis The *vapor pressure* P_v of a pure substance is defined as **the pressure exerted by a vapor in phase equilibrium with its liquid at a given temperature**. In general, the pressure of a vapor or gas, whether it exists alone or in a mixture with other gases, is called the *partial pressure*. During phase change processes between the liquid and vapor phases of a pure substance, the **saturation pressure and the vapor pressure are equivalent** since the vapor is pure.

Discussion Partial pressure is not necessarily equal to vapor pressure. For example, on a dry day (low relative humidity), the partial pressure of water vapor in the air is less than the vapor pressure of water. If, however, the relative humidity is 100%, the partial pressure and the vapor pressure are equal.

2-24E

Solution The minimum pressure in a pump is given. It is to be determined if there is a danger of cavitation.

Properties The vapor pressure of water at 70°F is 0.3632 psia.

Analysis To avoid cavitation, the pressure everywhere in the flow should remain above the vapor (or saturation) pressure at the given temperature, which is

$$P_v = P_{\text{sat @ 70°F}} = 0.3632 \text{ psia}$$

The minimum pressure in the pump is 0.1 psia, which is less than the vapor pressure. Therefore, **there is danger of cavitation in the pump**.

Discussion Note that the vapor pressure increases with increasing temperature, and the danger of cavitation increases at higher fluid temperatures.

2-25

Solution The minimum pressure in a pump to avoid cavitation is to be determined.

Properties The vapor pressure of water at 20°C is 2.339 kPa.

Analysis To avoid cavitation, the pressure anywhere in the system should not be allowed to drop below the vapor (or saturation) pressure at the given temperature. That is,

$$P_{\text{min}} = P_{\text{sat@20°C}} = \mathbf{2.339 kPa}$$

Therefore, **the lowest pressure that can exist in the pump is 2.339 kPa**.

Discussion Note that the vapor pressure increases with increasing temperature, and thus the risk of cavitation is greater at higher fluid temperatures.

2-26

Solution The minimum pressure in a piping system to avoid cavitation is to be determined.

Properties The vapor pressure of water at 30°C is 4.246 kPa.

Analysis To avoid cavitation, the pressure anywhere in the flow should not be allowed to drop below the vapor (or saturation) pressure at the given temperature. That is,

$$P_{\text{min}} = P_{\text{sat@30°C}} = \mathbf{4.246 kPa}$$

Therefore, the pressure should be maintained above 4.246 kPa everywhere in flow.

Discussion Note that the vapor pressure increases with increasing temperature, and thus the risk of cavitation is greater at higher fluid temperatures.

2-27

Solution The minimum pressure in a pump is given. It is to be determined if there is a danger of cavitation.

Properties The vapor pressure of water at 20°C is 2.339 kPa.

Analysis To avoid cavitation, the pressure everywhere in the flow should remain above the vapor (or saturation) pressure at the given temperature, which is

$$P_v = P_{\text{sat @ 20°C}} = 2.339 \text{ kPa}$$

The minimum pressure in the pump is 2 kPa, which is less than the vapor pressure. Therefore, a **there is danger of cavitation in the pump**.

Discussion Note that the vapor pressure increases with increasing temperature, and thus there is a greater danger of cavitation at higher fluid temperatures.

Energy and Specific Heats

2-28C

Solution We are to define and discuss flow energy.

Analysis *Flow energy* or *flow work* is the **energy needed to push a fluid into or out of a control volume**. Fluids at rest do not possess any flow energy.

Discussion Flow energy is not a fundamental quantity, like kinetic or potential energy. However, it is a useful concept in fluid mechanics since fluids are often forced into and out of control volumes in practice.

2-29C

Solution We are to compare the energies of flowing and non-flowing fluids.

Analysis **A flowing fluid possesses *flow energy*, which is the energy needed to push a fluid into or out of a control volume**, in addition to the forms of energy possessed by a non-flowing fluid. The total energy of a non-flowing fluid consists of internal and potential energies. If the fluid is moving as a rigid body, but not *flowing*, it may also have kinetic energy (e.g., gasoline in a tank truck moving down the highway at constant speed with no sloshing). The total energy of a flowing fluid consists of internal, kinetic, potential, *and* flow energies.

Discussion Flow energy is not to be confused with kinetic energy, even though both are zero when the fluid is at rest.

2-30C

Solution We are to discuss the difference between macroscopic and microscopic forms of energy.

Analysis The *macroscopic* forms of energy are those **a system possesses as a whole** with respect to some outside reference frame. The *microscopic* forms of energy, on the other hand, are those **related to the molecular structure of a system and the degree of the molecular activity**, and are independent of outside reference frames.

Discussion We mostly deal with macroscopic forms of energy in fluid mechanics.

2-31C

Solution We are to define total energy and identify its constituents.

Analysis The **sum of all forms of the energy a system possesses** is called *total energy*. In the absence of magnetic, electrical, and surface tension effects, **the total energy of a system consists of the kinetic, potential, and internal energies**.

Discussion All three constituents of total energy (kinetic, potential, and internal) need to be considered in an analysis of a general fluid flow.

2-32C

Solution We are to list the forms of energy that contribute to the internal energy of a system.

Analysis The *internal energy* of a system is made up of **sensible, latent, chemical, and nuclear energies**. The sensible internal energy is due to translational, rotational, and vibrational effects.

Discussion We deal with the flow of a single phase fluid in most problems in this textbook; therefore, latent, chemical, and nuclear energies do not need to be considered.

2-33C

Solution We are to discuss the relationship between heat, internal energy, and thermal energy.

Analysis *Thermal energy* is the **sensible and latent forms of internal energy**. It does not include chemical or nuclear forms of energy. In common terminology, thermal energy is referred to as *heat*. However, like work, heat is not a property, whereas thermal energy is a property.

Discussion Technically speaking, "heat" is defined only when there is *heat transfer*, whereas the energy state of a substance can always be defined, even if no heat transfer is taking place.

2-34C

Solution We are to explain how changes in internal energy can be determined.

Analysis Using specific heat values at the average temperature, the changes in the specific internal energy of ideal gases can be determined from $\boxed{\Delta u = c_{v,avg}\Delta T}$. For incompressible substances, $c_p \cong c_v \cong c$ and $\boxed{\Delta u = c_{avg}\Delta T}$.

Discussion If the fluid can be treated as neither incompressible nor an ideal gas, property tables must be used.

2-35C

Solution We are to explain how changes in enthalpy can be determined.

Analysis Using specific heat values at the average temperature, the changes in specific enthalpy of ideal gases can be determined from $\boxed{\Delta h = c_{p,avg}\Delta T}$. For incompressible substances, $c_p \cong c_v \cong c$ and $\boxed{\Delta h = \Delta u + v\Delta P \cong c_{avg}\Delta T + v\Delta P}$.

Discussion If the fluid can be treated as neither incompressible nor an ideal gas, property tables must be used.

2-36

Solution The total energy of saturated water vapor flowing in a pipe at a specified velocity and elevation is to be determined.

Analysis The total energy of a flowing fluid is given by (Eq. 2–8)

$$e = h + \frac{V^2}{2} + gz$$

The enthalpy of the vapor at the specified temperature can be found in any thermo text to be $2745.9\ \mathrm{kJ/kg}$. Then the total energy is determined as

$$e = 2745.9 \times 10^3\,\frac{\mathrm{J}}{\mathrm{kg}} + \frac{\left(50\,\frac{\mathrm{m}}{\mathrm{s}}\right)^2}{2} + \left(9.81\,\frac{\mathrm{m}}{\mathrm{s}^2}\right) \times (10\ \mathrm{m}) \cong 2.7472 \times 10^6\,\frac{\mathrm{J}}{\mathrm{kg}} = 2747.2\ \mathrm{kJ/kg}$$

Note that only 0.047% of the total energy comes from the combination of kinetic and potential energies, which explains why we usually neglect kinetic and potential energies in most flow systems.

Compressibility

2-37C

Solution We are to discuss the coefficient of compressibility and the isothermal compressibility.

Analysis The *coefficient of compressibility* represents the **variation of pressure of a fluid with volume or density at constant temperature**. *Isothermal compressibility* is the inverse of the coefficient of compressibility, and it represents the **fractional change in volume or density corresponding to a change in pressure**.

Discussion The coefficient of compressibility of an ideal gas is equal to its absolute pressure.

2-38C

Solution We are to define the coefficient of volume expansion.

Analysis The *coefficient of volume expansion* represents the **variation of the *density* of a fluid with temperature at constant pressure**. It differs from the coefficient of compressibility in that the latter represents the variation of *pressure* of a fluid with density at constant temperature.

Discussion The coefficient of volume expansion of an ideal gas is equal to the inverse of its absolute temperature.

2-39C

Solution We are to discuss the sign of the coefficient of compressibility and the coefficient of volume expansion.

Analysis The *coefficient of compressibility* of a fluid **cannot be negative**, but the *coefficient of volume expansion* **can be negative** (e.g., liquid water below 4°C).

Discussion This is the reason that ice floats on water.

2-40

Solution Water at a given temperature and pressure is heated to a higher temperature at constant pressure. The change in the density of water is to be determined.

Assumptions **1** The coefficient of volume expansion is constant in the given temperature range. **2** An approximate analysis is performed by replacing differential changes in quantities by finite changes.

Properties The density of water at 15°C and 1 atm pressure is ρ_1 = 999.1 kg/m^3. The coefficient of volume expansion at the average temperature of (15+95)/2 = 55°C is β = 0.484 × 10^{-3} K^{-1}.

Analysis When differential quantities are replaced by differences and the properties α and β are assumed to be constant, the change in density in terms of the changes in pressure and temperature is expressed approximately as

$$\Delta\rho = \alpha\rho\Delta P - \beta\rho\Delta T$$

The change in density due to the change of temperature from 15°C to 95°C at constant pressure is

$$\Delta\rho = -\beta\rho\Delta T = -(0.484\times10^{-3}\ \text{K}^{-1})(999.1\ \text{kg/m}^3)(95-15)\text{K} = \mathbf{-38.7\ kg/m^3}$$

Discussion Noting that $\Delta\rho = \rho_2 - \rho_1$, the density of water at 95°C and 1 atm is

$$\rho_2 = \rho_1 + \Delta\rho = 999.1 + (-38.7) = 960.4\ \text{kg/m}^3$$

which is very close to the listed value of 961.5 kg/m^3 at 95°C in water table in the Appendix. This is mostly due to β varying with temperature almost linearly. Note that the density of water decreases while being heated, as expected. This problem can be solved more accurately using differential analysis when functional forms of properties are available.

2-41

Solution The percent increase in the density of an ideal gas is given for a moderate pressure. The percent increase in density of the gas when compressed at a higher pressure is to be determined.

Assumptions The gas behaves an ideal gas.

Analysis For an ideal gas, $P = \rho RT$ and $(\partial P/\partial\rho)_T = RT = P/\rho$, and thus $\kappa_{\text{ideal gas}} = P$. Therefore, the coefficient of compressibility of an ideal gas is equal to its absolute pressure, and the coefficient of compressibility of the gas increases with increasing pressure.

Substituting $\kappa = P$ into the definition of the coefficient of compressibility $\kappa \cong -\dfrac{\Delta P}{\Delta v / v} \cong \dfrac{\Delta P}{\Delta\rho/\rho}$ and rearranging gives

$$\frac{\Delta\rho}{\rho} = \frac{\Delta P}{P}$$

Therefore, **the percent increase of density of an ideal gas during isothermal compression is equal to the percent increase in pressure**.

At 10 atm: $$\frac{\Delta\rho}{\rho} = \frac{\Delta P}{P} = \frac{11-10}{10} = 10\%$$

At 1000 atm: $$\frac{\Delta\rho}{\rho} = \frac{\Delta P}{P} = \frac{1001-1000}{1000} = 0.1\%$$

Therefore, a pressure change of 1 atm causes a density change of 10% at 10 atm and a density change of 1% at 100 atm.

Discussion If temperature were also allowed to change, the relationship would not be so simple.

2-42

Solution Using the definition of the coefficient of volume expansion and the expression $\beta_{\text{ideal gas}} = 1/T$, it is to be shown that the percent increase in the specific volume of an ideal gas during isobaric expansion is equal to the percent increase in absolute temperature.

Assumptions The gas behaves an ideal gas.

Analysis The coefficient of volume expansion β can be expressed as $\beta = \dfrac{1}{\boldsymbol{v}}\left(\dfrac{\partial \boldsymbol{v}}{\partial T}\right)_P \approx \dfrac{\Delta \boldsymbol{v}/\boldsymbol{v}}{\Delta T}$.

Noting that $\beta_{\text{ideal gas}} = 1/T$ for an ideal gas and rearranging give

$$\frac{\Delta \boldsymbol{v}}{\boldsymbol{v}} = \frac{\Delta T}{T}$$

Therefore, **the percent increase in the specific volume of an ideal gas during isobaric expansion is equal to the percent increase in absolute temperature**.

Discussion We must be careful to use *absolute* temperature (K or R), not relative temperature (^{o}C or ^{o}F).

2-43

Solution Water at a given temperature and pressure is compressed to a high pressure isothermally. The increase in the density of water is to be determined.

Assumptions **1** The isothermal compressibility is constant in the given pressure range. **2** An approximate analysis is performed by replacing differential changes by finite changes.

Properties The density of water at 20°C and 1 atm pressure is ρ_1 = 998 kg/m^3. The isothermal compressibility of water is given to be α = 4.80 × 10^{-5} atm^{-1}.

Analysis When differential quantities are replaced by differences and the properties α and β are assumed to be constant, the change in density in terms of the changes in pressure and temperature is expressed approximately as

$$\Delta\rho = \alpha\rho\Delta P - \beta\rho\Delta T$$

The change in density due to a change of pressure from 1 atm to 400 atm at constant temperature is

$$\Delta\rho = \alpha\rho\Delta P = (4.80\times10^{-5}\ \text{atm}^{-1})(998\ \text{kg/m}^3)(400-1)\text{atm} = \mathbf{19.2\ kg/m^3}$$

Discussion Note that the density of water increases from 998 to 1017.2 kg/m^3 while being compressed, as expected. This problem can be solved more accurately using differential analysis when functional forms of properties are available.

2-44

Solution The volume of an ideal gas is reduced by half at constant temperature. The change in pressure is to be determined.

Assumptions The process is isothermal and thus the temperature remains constant.

Analysis For an ideal gas of fixed mass undergoing an isothermal process, the ideal gas relation reduces to

$$\frac{P_2 \boldsymbol{V}_2}{T_2} = \frac{P_1 \boldsymbol{V}_1}{T_1} \quad \rightarrow \quad P_2 \boldsymbol{V}_2 = P_1 \boldsymbol{V}_1 \quad \rightarrow \quad P_2 = \frac{\boldsymbol{V}_1}{\boldsymbol{V}_2} P_1 = \frac{\boldsymbol{V}_1}{0.5\boldsymbol{V}_1} P_1 = 2P_1$$

Therefore, the change in pressure becomes

$$\Delta P = P_2 - P_1 = 2P_1 - P_1 = \boldsymbol{P_1}$$

Discussion Note that at constant temperature, pressure and volume of an ideal gas are inversely proportional.

2-45

Solution Saturated refrigerant-134a at a given temperature is cooled at constant pressure. The change in the density of the refrigerant is to be determined.

Assumptions **1** The coefficient of volume expansion is constant in the given temperature range. **2** An approximate analysis is performed by replacing differential changes in quantities by finite changes.

Properties The density of saturated liquid R-134a at 10°C is ρ_1 =1261 kg/m^3. The coefficient of volume expansion at the average temperature of (10+0)/2 = 5°C is β = 0.00269 K^{-1}.

Analysis When differential quantities are replaced by differences and the properties α and β are assumed to be constant, the change in density in terms of the changes in pressure and temperature is expressed approximately as

$$\Delta\rho = \alpha\rho\Delta P - \beta\rho\Delta T$$

The change in density due to the change of temperature from 10°C to 0°C at constant pressure is

$$\Delta\rho = -\beta\rho\Delta T = -(0.00269\,\text{K}^{-1})(1261\,\text{kg/m}^3)(0-10)\text{K} = \mathbf{33.9\ kg/m^3}$$

Discussion Noting that $\Delta\rho = \rho_2 - \rho_1$, the density of R-134a at 0°C is

$$\rho_2 = \rho_1 + \Delta\rho = 1261 + 33.9 = 1294.9\ \text{kg/m}^3$$

which is almost identical to the listed value of 1295 kg/m^3 at 0°C in R-134a table in the Appendix. This is mostly due to β varying with temperature almost linearly. Note that the density increases during cooling, as expected.

2-46

Solution A water tank completely filled with water can withstand tension caused by a volume expansion of 0.8%. The maximum temperature rise allowed in the tank without jeopardizing safety is to be determined.

Assumptions **1** The coefficient of volume expansion is constant. **2** An approximate analysis is performed by replacing differential changes in quantities by finite changes. **3** The effect of pressure is disregarded.

Properties The average volume expansion coefficient is given to be $\beta = 0.377 \times 10^{-3}\ \text{K}^{-1}$.

Analysis When differential quantities are replaced by differences and the properties α and β are assumed to be constant, the change in density in terms of the changes in pressure and temperature is expressed approximately as

$$\Delta\rho = \alpha\rho\Delta P - \beta\rho\Delta T$$

A volume increase of 0.8% corresponds to a density decrease of 0.8%, which can be expressed as $\Delta\rho = -0.008\rho$. Then the decrease in density due to a temperature rise of ΔT at constant pressure is

$$-0.008\rho = -\beta\rho\Delta T$$

Solving for ΔT and substituting, the maximum temperature rise is determined to be

$$\Delta T = \frac{0.008}{\beta} = \frac{0.008}{0.377 \times 10^{-3}\ \text{K}^{-1}} = \mathbf{21.2\ K} = \mathbf{21.2^\circ C}$$

Discussion This result is conservative since in reality the increasing pressure will tend to compress the water and increase its density.

2-47

Solution A water tank completely filled with water can withstand tension caused by a volume expansion of 1.5%. The maximum temperature rise allowed in the tank without jeopardizing safety is to be determined.

Assumptions **1** The coefficient of volume expansion is constant. **2** An approximate analysis is performed by replacing differential changes in quantities by finite changes. **3** The effect of pressure is disregarded.

Properties The average volume expansion coefficient is given to be $\beta = 0.377 \times 10^{-3}\ \text{K}^{-1}$.

Analysis When differential quantities are replaced by differences and the properties α and β are assumed to be constant, the change in density in terms of the changes in pressure and temperature is expressed approximately as

$$\Delta\rho = \alpha\rho\Delta P - \beta\rho\Delta T$$

A volume increase of 1.5% corresponds to a density decrease of 1.5%, which can be expressed as $\Delta\rho = -0.015\rho$. Then the decrease in density due to a temperature rise of ΔT at constant pressure is

$$-0.015\rho = -\beta\rho\Delta T$$

Solving for ΔT and substituting, the maximum temperature rise is determined to be

$$\Delta T = \frac{0.015}{\beta} = \frac{0.015}{0.377 \times 10^{-3}\ \text{K}^{-1}} = \mathbf{39.8\ K} = \mathbf{39.8^\circ C}$$

Discussion This result is conservative since in reality the increasing pressure will tend to compress the water and increase its density. The change in temperature is exactly half of that of the previous problem, as expected.

2-48

Solution The density of seawater at the free surface and the bulk modulus of elasticity are given. The density and pressure at a depth of 2500 m are to be determined.

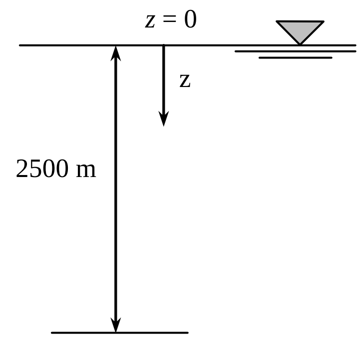
Assumptions **1** The temperature and the bulk modulus of elasticity of seawater is constant. **2** The gravitational acceleration remains constant.

Properties The density of seawater at free surface where the pressure is given to be 1030 kg/m^3, and the bulk modulus of elasticity of seawater is given to be 2.34×10^9 N/m^2.

Analysis The coefficient of compressibility or the bulk modulus of elasticity of fluids is expressed as

$$\kappa = \rho\left(\frac{\partial P}{\partial \rho}\right)_T \qquad \text{or} \qquad \kappa = \rho\frac{dP}{d\rho} \qquad \text{(at constant } T\text{)}$$

The differential pressure change across a differential fluid height of dz is given as



$$dP = \rho g dz$$

Combining the two relations above and rearranging,

$$\kappa = \rho\frac{\rho g dz}{d\rho} = g\rho^2\frac{dz}{d\rho} \qquad \rightarrow \qquad \frac{d\rho}{\rho^2} = \frac{gdz}{\kappa}$$

Integrating from $z = 0$ where $\rho = \rho_0 = 1030$ kg/m^3 to $z = z$ where $\rho = \rho$ gives

$$\int_{\rho_0}^{\rho}\frac{d\rho}{\rho^2} = \frac{g}{\kappa}\int_0^z dz \qquad \rightarrow \qquad \frac{1}{\rho_0} - \frac{1}{\rho} = \frac{gz}{\kappa}$$

Solving for ρ gives the variation of density with depth as

$$\rho = \frac{1}{(1/\rho_0) - (gz/\kappa)}$$

Substituting into the pressure change relation $dP = \rho g dz$ and integrating from $z = 0$ where $P = P_0 =$ 98 kPa to $z = z$ where $P = P$ gives

$$\int_{P_0}^{P} dP = \int_0^z \frac{gdz}{(1/\rho_0) - (gz/\kappa)} \qquad \rightarrow \qquad P = P_0 + \kappa\ln\left(\frac{1}{1 - (\rho_0 gz/\kappa)}\right)$$

which is the desired relation for the variation of pressure in seawater with depth. At z = 2500 m, the values of density and pressure are determined by substitution to be

$$\rho = \frac{1}{1/(1030\text{ kg/m}^3) - (9.81\text{ m/s}^2)(2500\text{ m})/(2.34\times10^9\text{ N/m}^2)} = \mathbf{1041\ kg/m^3}$$

$$P = (98{,}000\text{ Pa}) + (2.34\times10^9\text{ N/m}^2)\ln\left(\frac{1}{1-(1030\text{ kg/m}^3)(9.81\text{ m/s}^2)(2500\text{ m})/(2.34\times10^9\text{ N/m}^2)}\right)$$

$$= 2.550\times10^7\text{ Pa}$$

$$= \mathbf{25.50\ MPa}$$

since 1 Pa = 1 N/m^2 = 1 kg/m·s^2 and 1 kPa = 1000 Pa.

Discussion Note that if we assumed $\rho = \rho_o$ = constant at 1030 kg/m^3, the pressure at 2500 m would be $P = P_0 + \rho gz$ = 0.098 + 25.26 = 25.36 MPa. Then the density at 2500 m is estimated to be

$$\Delta\rho = \rho\alpha\Delta P = (1030)(2340\text{ MPa})^{-1}(25.26\text{ MPa}) = 11.1\text{ kg/m}^3 \text{ and thus } \rho = 1041\text{ kg/m}^3$$

2-49E

Solution The coefficient of compressibility of water is given. The pressure increases required to reduce the volume of water by 1 percent and then by 2 percent are to be determined.

Assumptions **1** The coefficient of compressibility is constant. **2** The temperature remains constant.

Properties The coefficient of compressibility of water is given to be 7×10^5 psia.

Analysis (*a*) A volume decrease of 1 percent can mathematically be expressed as

$$\frac{\Delta v}{v}=\frac{\Delta V}{V}=-0.01$$

The coefficient of compressibility is expressed as

$$\kappa=-v\left(\frac{\partial P}{\partial v}\right)_T\cong-\frac{\Delta P}{\Delta v/v}$$

Rearranging and substituting, the required pressure increase is determined to be

$$\Delta P=-\kappa\left(\frac{\Delta v}{v}\right)=-(7\times10^5\text{ psia})(-0.01)=7{,}000\text{ psia}$$

(b) Similarly, the required pressure increase for a volume reduction of 2 percent becomes

$$\Delta P=-\kappa\left(\frac{\Delta v}{v}\right)=-(7\times10^5\text{ psia})(-0.02)=14{,}000\text{ psia}$$

Discussion Note that at extremely high pressures are required to compress water to an appreciable amount.

2-50E

Solution We are to estimate the energy required to heat up the water in a hot-water tank.

Assumptions **1** There are no losses. **2** The pressure in the tank remains constant at 1 atm. **3** An approximate analysis is performed by replacing differential changes in quantities by finite changes.

Properties The specific heat of water is approximated as a constant, whose value is 0.999 Btu/lbm·R at the average temperature of (60 + 110)/2 = 85°F. In fact, *c* remains constant at 0.999 Btu/lbm·R (to three digits) from 60°F to 110°F. For this same temperature range, the density varies from 62.36 lbm/ft^3 at 60°F to 61.86 lbm/ft^3 at 110°F. We approximate the density as constant, whose value is 62.17 lbm/ft^3 at the average temperature of 85°F.

Analysis For a constant pressure process, $\Delta u\cong c_{\text{avg}}\Delta T$. Since this is energy per unit mass, we must multiply by the total mass of the water in the tank, i.e., $\Delta U\cong mc_{\text{avg}}\Delta T=\rho V c_{\text{avg}}\Delta T$. Thus,

$$\Delta U\cong\rho V c_{\text{avg}}\Delta T=(62.17\text{ lbm/ft}^3)(75\text{ gal})(0.999\text{ Btu/lbm}\cdot\text{R})[(110-60)\text{R}]\left(\frac{35.315\text{ ft}^3}{264.17\text{ gal}}\right)=31{,}135\text{ Btu}\cong\mathbf{31{,}100\ Btu}$$

where we note temperature *differences* are identical in °F and R.

Discussion We give the final answer to 3 significant digits. The actual energy required will be greater than this, due to heat transfer losses and other inefficiencies in the hot-water heating system.

2-51

Solution We are to prove that the coefficient of volume expansion for an ideal gas is equal to $1/T$.

Assumptions **1** Temperature and pressure are in the range that the gas can be approximated as an ideal gas.

Analysis The ideal gas law is $P = \rho RT$, which we re-write as $\rho = \frac{P}{RT}$. By definition, $\beta = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_P$. Thus, substitution and differentiation yields

$$\beta_{\text{ideal gas}} = -\frac{1}{\rho}\left(\frac{\partial\left(\frac{P}{RT}\right)}{\partial T}\right)_P = -\frac{1}{\rho}\left(-\frac{P}{RT^2}\right) = \frac{\rho}{\rho}\frac{1}{T} = \mathbf{1/\mathit{T}}$$

where both pressure and the gas constant R are treated as constants in the differentiation.

Discussion The coefficient of volume expansion of an ideal gas is not constant, but rather decreases with temperature. However, for small temperature differences, β is often approximated as a constant with little loss of accuracy.

2-52

Solution The coefficient of compressibility of nitrogen gas is to be estimated using Van der Waals equation of state. The result is to be compared to ideal gas and experimental values.

Assumptions **1** Nitrogen gas obeys the Van der Waals equation of state.

Analysis From the definition we have

$$\kappa = -v\left(\frac{\partial P}{\partial v}\right)_T = \frac{vRT}{(v-b)^2} - \frac{2a}{v^2}$$

since

$$\left(\frac{\partial P}{\partial v}\right)_T = \frac{2a}{v^3} - \frac{RT}{(v-b)^2}$$

The gas constant of nitrogen is $0.2968\ \text{kJ/kg}\cdot\text{K}$ (Table A–1). Substituting given data we obtain

$$\kappa = \frac{\left(0.00375\,\frac{\text{m}^3}{\text{kg}}\right)\times\left(0.2968\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}\right)\times(175\ \text{K})}{\left(0.00375\,\frac{\text{m}^3}{\text{kg}} - 0.00138\,\frac{\text{m}^3}{\text{kg}}\right)^2} - \frac{2\times 0.175\ \text{m}^6\cdot\frac{\text{kPa}}{\text{kg}^2}}{\left(0.00375\,\frac{\text{m}^3}{\text{kg}}\right)^2} \cong 9788\ \text{kPa}$$

For the ideal gas behavior, the coefficient of compressibility is equal to the pressure (Eq. 2–15). Therefore we get

$$\kappa = P = \frac{RT}{v} = \frac{\left(0.2968\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}\right)\times(175\ \text{K})}{0.00375\ \text{m}^3/\text{kg}} \cong 13851\ \text{kPa}$$

whichis in error by 38.5% compared to experimentally measured pressure.

2-53

Solution The water contained in a piston-cylinder device is compressed isothermally. The energy needed is to be determined.

Assumptions **1** The coefficient of compressibility of water remains unchanged during the compression.

Analysis We take the water in the cylinder as the system. The energy needed to compress water is equal to the work done on the system, and can be expressed as

$$W = -\int_1^2 PdV \qquad (1)$$

From the definition of coefficient of compressibility we have

$$\kappa = -\frac{dP}{dV/V}$$

Rearranging we obtain

$$\frac{dV}{V} = -\frac{dP}{\kappa}$$

which can be integrated from the initial state to any state as follows:

$$\int_{V_0}^{V} \frac{dV}{V} = -\int_{P_0}^{P} \frac{dP}{\kappa} \rightarrow \ln\frac{V}{V_0} = -\frac{P-P_0}{\kappa}$$

from which we obtain

$$P = P_0 - \kappa \ln\frac{V}{V_0}$$

Substituting in Eq. 1 we have

$$W = -\int_{V_0}^{V_1} PdV = -\int_{V_0}^{V_1}\left(P_0 - \kappa\ln\frac{V}{V_0}\right)dV = \left[\kappa V\ln\frac{V}{V_0} - (P_0+\kappa)V\right]_{V_0}^{V_1}$$

or

$$W = (P_0+\kappa)(V_0 - V_1) + \kappa V_0 \ln\frac{V_1}{V_0}$$

In terms of finite changes, the fractional change due to change in pressure can be expressed approximately as (Eq. 3–23)

$$\frac{V_1 - V_0}{V_0} \cong -\alpha(P_1 - P_0)$$

or

$$V_1 \cong V_0\big(1 - \alpha(P_1 - P_0)\big)$$

where α is the isothermal compressibility of water, which is $4.80 \times 10^{-5}\text{atm}^{-1}$ at 20 °C. Realizing that 10 kg water occupies initially a volume of $V_0 = 10 \times 10^{-3}\text{m}^3$, the final volume of water is determined to be

$$V_1 \cong (0.01\ \text{m}^3) \times [1 - (4.80 \times 10^{-5}\text{atm}^{-1}) \times (100\ \text{atm} - 1\ \text{atm})] = 9.952 \times 10^{-3}\text{m}^3$$

Then the work done on the water is

$$W = (1\ \text{atm} + 2100\ \text{atm}) \times (10 \times 10^{-3}\text{m}^3 - 9.952 \times 10^{-3}\text{m}^3)$$
$$+(2100\ \text{atm}) \times (10 \times 10^{-3}\text{m}^3)\ln\frac{9.952 \times 10^{-3}\text{m}^3}{10 \times 10^{-3}\text{m}^3}$$

from which we obtain

$$W = 2.903 \times 10^{-4}\text{atm}\cdot\text{m}^3 \cong 29.4\ \text{J}$$

since $1\ \text{atm} = 101325\ \text{Pa}$.

2-54

Solution The water contained in a piston-cylinder device is compressed isothermally and the pressure increases linearly. The energy needed is to be determined.

Assumptions **1** The pressure increases linearly.

Analysis We take the water in the cylinder as the system. The energy needed to compress water is equal to the work done on the system, and can be expressed as

$$W = -\int_1^2 P dV \qquad (1)$$

For a linear pressure increase we take

$$P = P_{\text{ave}} = \frac{P_1 + P_2}{2} = \frac{100 \text{ atm} + 1 \text{ atm}}{2} = 50.5 \text{ atm}$$

In terms of finite changes, the fractional change due to change in pressure can be expressed approximately as (Eq. 3−23)

$$\frac{V_1 - V_0}{V_0} \cong -\alpha(P_1 - P_0)$$

or

$$V_1 \cong V_0\big(1 - \alpha(P_1 - P_0)\big)$$

where α is the isothermal compressibility of water, which is $4.80 \times 10^{-5} \text{atm}^{-1}$ at 20 °C. Realizing that 10 kg water occupies initially a volume of $V_0 = 10 \times 10^{-3} \text{m}^3$, the final volume of water is determined to be

$$V_1 \cong (0.01 \text{ m}^3) \times [1 - (4.80 \times 10^{-5} \text{atm}^{-1}) \times (100 \text{ atm} - 1 \text{ atm})] = 9.952 \times 10^{-3} \text{m}^3$$

Therefore the work expression becomes

$$W = -\int_{V_0}^{V_1} P dV = -P_{\text{ave}}(V_1 - V_0) = -(50.5 \text{ atm}) \times (9.952 \times 10^{-3} \text{m}^3 - 10 \times 10^{-3} \text{m}^3)$$

or

$$W = 2.424 \times 10^{-3} \text{atm} \cdot \text{m}^3 = 245.6 \text{ J}$$

Thus, we conclude that linear pressure increase approximation does not work well since it gives almost ten times larger work.

Speed of Sound

2-55C

Solution We are to define and discuss sound and how it is generated and how it travels.

Analysis *Sound* is an **infinitesimally small pressure wave**. It is **generated by a small disturbance in a medium**. It **travels by wave propagation**. **Sound waves cannot travel in a vacuum**.

Discussion Electromagnetic waves, like light and radio waves, can travel in a vacuum, but sound cannot.

2-56C

Solution We are to discuss whether sound travels faster in warm or cool air.

Analysis Sound travels faster in **warm (higher temperature) air** since $c = \sqrt{kRT}$.

Discussion On the microscopic scale, we can imagine the air molecules moving around at higher speed in warmer air, leading to higher propagation of disturbances.

2-57C

Solution We are to compare the speed of sound in air, helium, and argon.

Analysis **Sound travels fastest in helium**, since $c = \sqrt{kRT}$ and helium has the highest kR value. It is about 0.40 for air, 0.35 for argon, and 3.46 for helium.

Discussion We are assuming, of course, that these gases behave as ideal gases – a good approximation at room temperature.

2-58C

Solution We are to compare the speed of sound in air at two different pressures, but the same temperature.

Analysis Air at specified conditions will behave like an ideal gas, and *the speed of sound in an ideal gas depends on temperature only*. Therefore, **the speed of sound is the same in both mediums**.

Discussion If the temperature were different, however, the speed of sound would be different.

2-59C

Solution We are to examine whether the Mach number remains constant in constant-velocity flow.

Analysis In general, **no**, because the Mach number also depends on the speed of sound in gas, which depends on the temperature of the gas. **The Mach number remains constant only if the temperature *and* the velocity are constant**.

Discussion It turns out that the speed of sound is not a strong function of pressure. In fact, it is not a function of pressure at all for an ideal gas.

2-60C

Solution We are to state whether the propagation of sound waves is an isentropic process.

Analysis **Yes, the propagation of sound waves is nearly isentropic**. Because the amplitude of an ordinary sound wave is very small, and it does not cause any significant change in temperature and pressure.

Discussion No process is truly isentropic, but the increase of entropy due to sound propagation is negligibly small.

2-61C

Solution We are to discuss sonic velocity – specifically, whether it is constant or it changes.

Analysis The *sonic speed* in a medium **depends on the properties of the medium, and it changes as the properties of the medium change**.

Discussion The most common example is the change in speed of sound due to temperature change.

2-62

Solution The Mach number of a passenger plane for specified limiting operating conditions is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is R = 0.287 kJ/kg·K. Its specific heat ratio at room temperature is k = 1.4.

Analysis From the speed of sound relation

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(\text{-}60 + 273\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 293\ \text{m/s}$$

Thus, the Mach number corresponding to the maximum cruising speed of the plane is

$$\text{Ma} = \frac{V_{\max}}{c} = \frac{(945/3.6)\ \text{m/s}}{293\ \text{m/s}} = \mathbf{0.897}$$

Discussion Note that this is a subsonic flight since Ma < 1. Also, using a k value at -60°C would give practically the same result.

2-63

Solution Carbon dioxide flows through a nozzle. The inlet temperature and velocity and the exit temperature of CO_2 are specified. The Mach number is to be determined at the inlet and exit of the nozzle.

Assumptions **1** CO_2 is an ideal gas with constant specific heats at room temperature. **2** This is a steady-flow process.

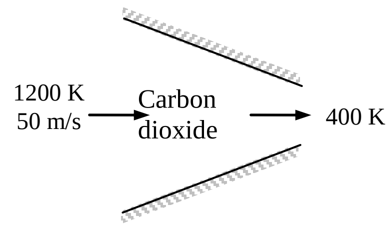
Properties The gas constant of carbon dioxide is R = 0.1889 kJ/kg·K. Its constant pressure specific heat and specific heat ratio at room temperature are c_p = 0.8439 kJ/kg·K and k = 1.288.

Analysis (*a*) At the inlet

$$c_1 = \sqrt{k_1 R T_1} = \sqrt{(1.288)(0.1889\text{ kJ/kg}\cdot\text{K})(1200\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 540.3\text{ m/s}$$

Thus,

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{50\text{ m/s}}{540.3\text{ m/s}} = \mathbf{0.0925}$$



(*b*) At the exit,

$$c_2 = \sqrt{k_2 R T_2} = \sqrt{(1.288)(0.1889\text{ kJ/kg}\cdot\text{K})(400\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 312.0\text{ m/s}$$

The nozzle exit velocity is determined from the steady-flow energy balance relation,

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \quad \rightarrow \quad 0 = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

$$0 = (0.8439\text{ kJ/kg}\cdot\text{K})(400 - 1200\text{ K}) + \frac{V_2^2 - (50\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \longrightarrow V_2 = 1163\text{ m/s}$$

Thus,

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{1163\text{ m/s}}{312\text{ m/s}} = \mathbf{3.73}$$

Discussion The specific heats and their ratio k change with temperature, and the accuracy of the results can be improved by accounting for this variation. Using EES (or another property database):

At 1200 K: c_p = 1.278 kJ/kg·K, k = 1.173 $\rightarrow$ c_1 = 516 m/s, V_1 = 50 m/s, Ma_1 = 0.0969

At 400 K: c_p = 0.9383 kJ/kg·K, k = 1.252 $\rightarrow$ c_2 = 308 m/s, V_2 = 1356 m/s, Ma_2 = 4.41

Therefore, the constant specific heat assumption results in an error of **4.5%** at the inlet and **15.5%** at the exit in the Mach number, which are significant.

2-64

Solution Nitrogen flows through a heat exchanger. The inlet temperature, pressure, and velocity and the exit pressure and velocity are specified. The Mach number is to be determined at the inlet and exit of the heat exchanger.

Assumptions **1** N_2 is an ideal gas. **2** This is a steady-flow process. **3** The potential energy change is negligible.

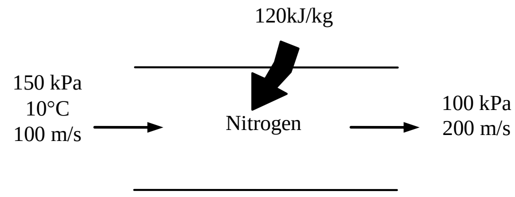
Properties The gas constant of N_2 is R = 0.2968 kJ/kg·K. Its constant pressure specific heat and specific heat ratio at room temperature are c_p = 1.040 kJ/kg·K and k = 1.4.

Analysis

$$c_1 = \sqrt{k_1 R T_1} = \sqrt{(1.400)(0.2968\text{ kJ/kg}\cdot\text{K})(283\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 342.9\text{ m/s}$$

Thus,

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{100\text{ m/s}}{342.9\text{ m/s}} = \mathbf{0.292}$$



From the energy balance on the heat exchanger,

$$q_{\text{in}} = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

$$120\text{ kJ/kg} = (1.040\text{ kJ/kg.°C})(T_2 - 10\text{°C}) + \frac{(200\text{ m/s})^2 - (100\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

It yields

$$T_2 = 111\text{°C} = 384\text{ K}$$

$$c_2 = \sqrt{k_2 R T_2} = \sqrt{(1.4)(0.2968\text{ kJ/kg}\cdot\text{K})(384\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 399\text{ m/s}$$

Thus,

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{200\text{ m/s}}{399\text{ m/s}} = \mathbf{0.501}$$

Discussion The specific heats and their ratio k change with temperature, and the accuracy of the results can be improved by accounting for this variation. Using EES (or another property database):

At 10°C : c_p = 1.038 kJ/kg·K, k = 1.400 → c_1 = 343 m/s, V_1 = 100 m/s, Ma_1 = 0.292

At 111°C c_p = 1.041 kJ/kg·K, k = 1.399 → c_2 = 399 m/s, V_2 = 200 m/s, Ma_2 = 0.501

Therefore, the constant specific heat assumption results in an error of **4.5%** at the inlet and **15.5%** at the exit in the Mach number, which are almost identical to the values obtained assuming constant specific heats.

2-65

Solution The speed of sound in refrigerant-134a at a specified state is to be determined.

Assumptions R-134a is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of R-134a is R = 0.08149 kJ/kg·K. Its specific heat ratio at room temperature is k = 1.108.

Analysis From the ideal-gas speed of sound relation,

$$c = \sqrt{kRT} = \sqrt{(1.108)(0.08149\ \text{kJ/kg}\cdot\text{K})(60+273\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{173\ m/s}$$

Discussion Note that the speed of sound is independent of pressure for ideal gases.

2-66

Solution The Mach number of an aircraft and the speed of sound in air are to be determined at two specified temperatures.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is R = 0.287 kJ/kg·K. Its specific heat ratio at room temperature is k = 1.4.

Analysis From the definitions of the speed of sound and the Mach number,

(a) At 300 K,

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{347 \text{ m/s}}$$

and $$\text{Ma} = \frac{V}{c} = \frac{330 \text{ m/s}}{347 \text{ m/s}} = \mathbf{0.951}$$

(b) At 800 K,

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(800 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{567 \text{ m/s}}$$

and $$\text{Ma} = \frac{V}{c} = \frac{330 \text{ m/s}}{567 \text{ m/s}} = \mathbf{0.582}$$

Discussion Note that a constant Mach number does not necessarily indicate constant speed. The Mach number of a rocket, for example, will be increasing even when it ascends at constant speed. Also, the specific heat ratio k changes with temperature.

2-67E

Solution Steam flows through a device at a specified state and velocity. The Mach number of steam is to be determined assuming ideal gas behavior.

Assumptions Steam is an ideal gas with constant specific heats.

Properties The gas constant of steam is R = 0.1102 Btu/lbm·R. Its specific heat ratio is given to be k = 1.3.

Analysis From the ideal-gas speed of sound relation,

$$c = \sqrt{kRT} = \sqrt{(1.3)(0.1102 \text{ Btu/lbm}\cdot\text{R})(1160 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = 2040 \text{ ft/s}$$

Thus,

$$\text{Ma} = \frac{V}{c} = \frac{900 \text{ ft/s}}{2040 \text{ ft/s}} = \mathbf{0.441}$$

Discussion Using property data from steam tables and not assuming ideal gas behavior, it can be shown that the Mach number in steam at the specified state is 0.446, which is sufficiently close to the ideal-gas value of 0.441. Therefore, the ideal gas approximation is a reasonable one in this case.

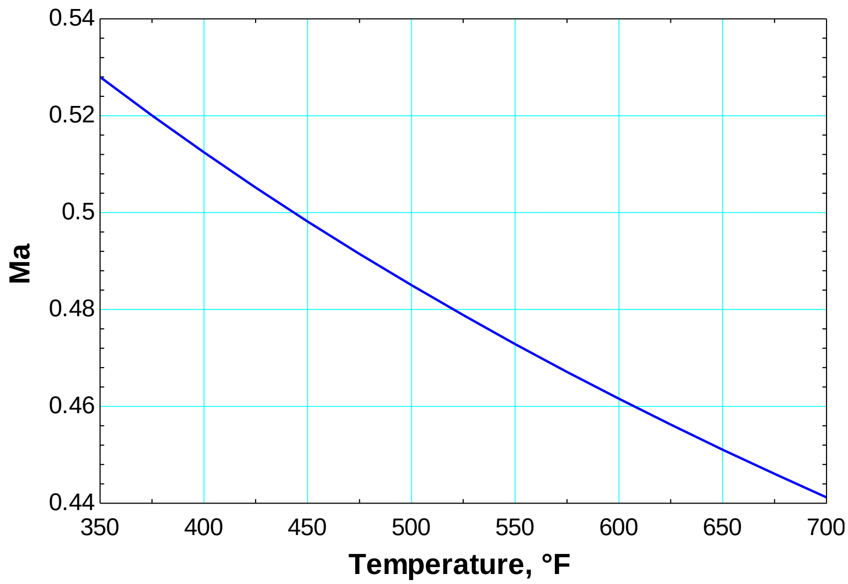
2-68E

Solution Problem 2-67e is reconsidered. The variation of Mach number with temperature as the temperature changes between 350° and 700°F is to be investigated, and the results are to be plotted.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
T=Temperature+460
R=0.1102
V=900
k=1.3
c=SQRT(k*R*T*25037)
Ma=V/c
```

Temperature, *T*, °F	Mach number Ma
350	0.528
375	0.520
400	0.512
425	0.505
450	0.498
475	0.491
500	0.485
525	0.479
550	0.473
575	0.467
600	0.462
625	0.456
650	0.451
675	0.446
700	0.441



Discussion Note that for a specified flow speed, the Mach number decreases with increasing temperature, as expected.

2-69E

Solution The inlet state and the exit pressure of air are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air are R = 0.06855 Btu/lbm·R and k = 1.4. The specific heat ratio k varies with temperature, but in our case this change is very small and can be disregarded.

Analysis The final temperature of air is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (659.7\text{ R})\left(\frac{60}{170}\right)^{(1.4-1)/1.4} = 489.9\text{ R}$$

Treating k as a constant, the ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1 R T_1}}{\sqrt{k_2 R T_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{659.7}}{\sqrt{489.9}} = \mathbf{1.16}$$

Discussion Note that the speed of sound is proportional to the square root of thermodynamic temperature.

2-70

Solution The inlet state and the exit pressure of air are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air are R = 0.287 kJ/kg·K and k = 1.4. The specific heat ratio k varies with temperature, but in our case this change is very small and can be disregarded.

Analysis The final temperature of air is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (350.2\ \text{K})\left(\frac{0.4\,\text{MPa}}{2.2\,\text{MPa}}\right)^{(1.4-1)/1.4} = 215.2\ \text{K}$$

Treating k as a constant, the ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1RT_1}}{\sqrt{k_2RT_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{350.2}}{\sqrt{215.2}} = \mathbf{1.28}$$

Discussion Note that the speed of sound is proportional to the square root of thermodynamic temperature.

2-71

Solution The inlet state and the exit pressure of helium are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

Assumptions Helium is an ideal gas with constant specific heats at room temperature.

Properties The properties of helium are R = 2.0769 kJ/kg·K and k = 1.667.

Analysis The final temperature of helium is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (350.2\ \text{K})\left(\frac{0.4}{2.2}\right)^{(1.667-1)/1.667} = 177.0\ \text{K}$$

The ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1RT_1}}{\sqrt{k_2RT_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{350.2}}{\sqrt{177.0}} = \mathbf{1.41}$$

Discussion Note that the speed of sound is proportional to the square root of thermodynamic temperature.

2-72

Solution The expression for the speed of sound for an ideal gas is to be obtained using the isentropic process equation and the definition of the speed of sound.

Analysis The isentropic relation $Pv^k = A$ where A is a constant can also be expressed as

$$P = A\left(\frac{1}{v}\right)^k = A\rho^k$$

Substituting it into the relation for the speed of sound,

$$c^2 = \left(\frac{\partial P}{\partial \rho}\right)_s = \left(\frac{\partial (A\rho)^k}{\partial \rho}\right)_s = kA\rho^{k-1} = k(A\rho^k)/\rho = k(P/\rho) = kRT$$

since for an ideal gas $P = \rho RT$ or $RT = P/\rho$. Therefore, $\boxed{c = \sqrt{kRT}}$, which is the desired relation.

Discussion Notice that pressure has dropped out; the speed of sound in an ideal gas is *not* a function of pressure.

Viscosity

2-73C

Solution We are to define and discuss viscosity.

Analysis *Viscosity* is **a measure of the "stickiness" or "resistance to deformation" of a fluid**. It is due to the internal frictional force that develops between different layers of fluids as they are forced to move relative to each other. Viscosity is caused by the cohesive forces between the molecules in liquids, and by the molecular collisions in gases. In general, **liquids have higher dynamic viscosities than gases**.

Discussion The ratio of viscosity μ to density ρ often appears in the equations of fluid mechanics, and is defined as the *kinematic viscosity*, $\nu = \mu/\rho$.

2-74C

Solution We are to discuss Newtonian fluids.

Analysis **Fluids whose shear stress is linearly proportional to the velocity gradient** (shear strain) are called *Newtonian fluids*. Most common fluids such as water, air, gasoline, and oils are Newtonian fluids.

Discussion In the differential analysis of fluid flow, only Newtonian fluids are considered in this textbook.

2-75C

Solution We are to discuss how kinematic viscosity varies with temperature in liquids and gases.

Analysis (*a*) **For *liquids*, the kinematic viscosity decreases with temperature**. (*b*) **For *gases*, the kinematic viscosity increases with temperature**.

Discussion You can easily verify this by looking at the appendices.

2-76C

Solution We are to discuss how dynamic viscosity varies with temperature in liquids and gases.

Analysis (*a*) The **dynamic viscosity of liquids decreases with temperature**. (*b*) The **dynamic viscosity of gases increases with temperature**.

Discussion A good way to remember this is that a car engine is much harder to start in the winter because the oil in the engine has a higher viscosity at low temperatures.

2-77C

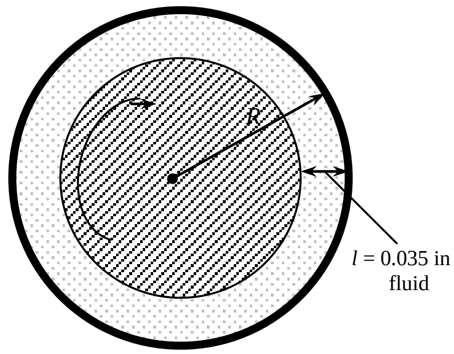
Solution We are to compare the settling speed of balls dropped in water and oil; namely, we are to determine which will reach the bottom of the container first.

Analysis When two identical small glass balls are dropped into two identical containers, one filled with water and the other with oil, **the ball dropped in water will reach the bottom of the container first** because of the much lower viscosity of water relative to oil.

Discussion Oil is very viscous, with typical values of viscosity approximately 800 times greater than that of water at room temperature.

2-78E

Solution The torque and the rpm of a double cylinder viscometer are given. The viscosity of the fluid is to be determined.



Assumptions **1** The inner cylinder is completely submerged in the fluid. **2** The viscous effects on the two ends of the inner cylinder are negligible. **3** The fluid is Newtonian.

Analysis Substituting the given values, the viscosity of the fluid is determined to be

$$\mu = \frac{\mathbf{T}\ell}{4\pi^2 R^3 \dot{n} L} = \frac{(1.2\ \text{lbf}\cdot\text{ft})(0.035/12\ \text{ft})}{4\pi^2 (3/12\ \text{ft})^3 (250/60\ \text{s}^{-1})(5\ \text{ft})} = \mathbf{2.72\times 10^{-4}\ lbf\cdot s/ft^2}$$

Discussion This is the viscosity value at temperature that existed during the experiment. Viscosity is a strong function of temperature, and the values can be significantly different at different temperatures.

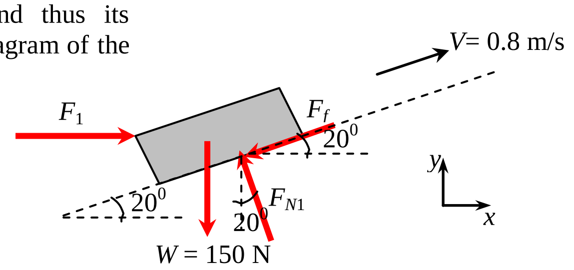
2-79

Solution A block is moved at constant velocity on an inclined surface. The force that needs to be applied in the horizontal direction when the block is dry, and the percent reduction in the required force when an oil film is applied on the surface are to be determined.

Assumptions **1** The inclined surface is plane (perfectly flat, although tilted). **2** The friction coefficient and the oil film thickness are uniform. **3** The weight of the oil layer is negligible.

Properties The absolute viscosity of oil is given to be $\mu = 0.012$ Pa·s = 0.012 N·s/m^2.

Analysis (*a*) The velocity of the block is constant, and thus its acceleration and the net force acting on it are zero. A free body diagram of the block is given. Then the force balance gives



$$\sum F_x = 0: \quad F_1 - F_f \cos 20° - F_{N1} \sin 20° = 0 \quad (1)$$

$$\sum F_y = 0: \quad F_{N1} \cos 20° - F_f \sin 20° - W = 0 \quad (2)$$

Friction force: $F_f = fF_{N1}$ (3)

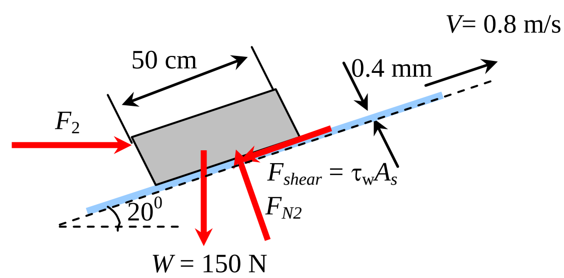
Substituting Eq. (3) into Eq. (2) and solving for F_{N1} gives

$$F_{N1} = \frac{W}{\cos 20° - f \sin 20°} = \frac{150 \text{ N}}{\cos 20° - 0.27 \sin 20°} = 177.0 \text{ N}$$

Then from Eq. (1):

$$F_1 = F_f \cos 20° + F_{N1} \sin 20° = (0.27 \times 177 \text{ N}) \cos 20° + (177 \text{ N}) \sin 20° = \mathbf{105.5 \text{ N}}$$

(*b*) In this case, the friction force is replaced by the shear force applied on the bottom surface of the block due to the oil. Because of the no-slip condition, the oil film sticks to the inclined surface at the bottom and the lower surface of the block at the top. Then the shear force is expressed as



$$\begin{aligned} F_{shear} &= \tau_w A_s \\ &= \mu A_s \frac{V}{h} \\ &= \left(0.012 \text{ N} \cdot \text{s/m}^2\right)\left(0.5 \times 0.2 \text{ m}^2\right) \frac{0.8 \text{ m/s}}{4 \times 10^{-4} \text{ m}} \\ &= 2.4 \text{ N} \end{aligned}$$

Replacing the friction force by the shear force in part (*a*),

$$\sum F_x = 0: \quad F_2 - F_{shear} \cos 20° - F_{N2} \sin 20° = 0 \quad (4)$$

$$\sum F_y = 0: \quad F_{N2} \cos 20° - F_{shear} \sin 20° - W = 0 \quad (5)$$

Eq. (5) gives $F_{N2} = (F_{shear} \sin 20° + W) / \cos 20° = [(2.4 \text{ N}) \sin 20° + (150 \text{ N})] / \cos 20° = 160.5 \text{ N}$

Substituting into Eq. (4), the required horizontal force is determined to be

$$F_2 = F_{shear} \cos 20° + F_{N2} \sin 20° = (2.4 \text{ N}) \cos 20° + (160.5 \text{ N}) \sin 20° = 57.2 \text{ N}$$

Then, our final result is expressed as

$$\text{Percentage reduction in required force} = \frac{F_1 - F_2}{F_1} \times 100\% = \frac{105.5 - 57.2}{105.5} \times 100\% = \mathbf{45.8\%}$$

Discussion Note that the force required to push the block on the inclined surface reduces significantly by oiling the surface.

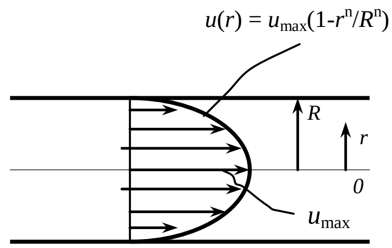
2-80

Solution The velocity profile of a fluid flowing though a circular pipe is given. The friction drag force exerted on the pipe by the fluid in the flow direction per unit length of the pipe is to be determined.

Assumptions The viscosity of the fluid is constant.

Analysis The wall shear stress is determined from its definition to be

$$\tau_w = -\mu \left.\frac{du}{dr}\right|_{r=R} = -\mu u_{\max} \frac{d}{dr}\left(1-\frac{r^n}{R^n}\right)_{r=R} = -\mu u_{\max} \left.\frac{-nr^{n-1}}{R^n}\right|_{r=R} = \frac{n\mu u_{\max}}{R}$$



Note that the quantity du/dr is negative in pipe flow, and the negative sign is added to the τ_w relation for pipes to make shear stress in the positive (flow) direction a positive quantity. (Or, $du/dr = - du/dy$ since $y = R - r$). Then the friction drag force exerted by the fluid on the inner surface of the pipe becomes

$$F = \tau_w A_w = \frac{n\mu u_{\max}}{R}(2\pi R)L = 2n\pi\mu u_{\max} L$$

Therefore, the drag force per unit length of the pipe is

$$\boxed{F/L = 2n\pi\mu u_{\max}}.$$

Discussion Note that the drag force acting on the pipe in this case is independent of the pipe diameter.

2-81

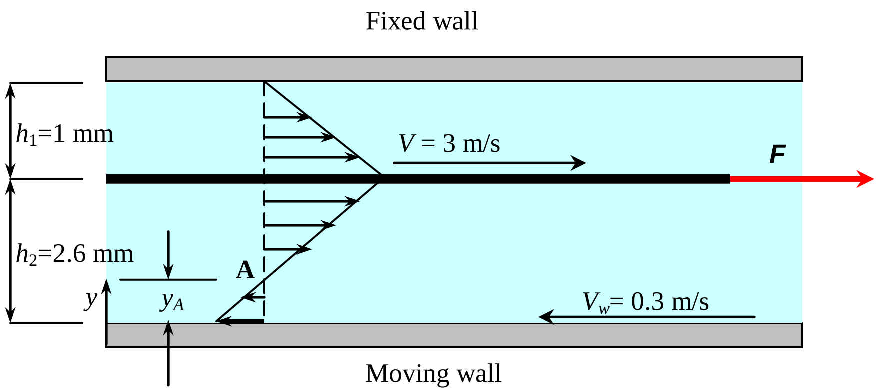
Solution A thin flat plate is pulled horizontally through an oil layer sandwiched between two plates, one stationary and the other moving at a constant velocity. The location in oil where the velocity is zero and the force that needs to be applied on the plate are to be determined.

Assumptions **1** The thickness of the plate is negligible. **2** The velocity profile in each oil layer is linear.

Properties The absolute viscosity of oil is given to be $\mu = 0.027$ Pa·s = 0.027 N·s/m^2.

Analysis (*a*) The velocity profile in each oil layer relative to the fixed wall is as shown in the figure below. The point of zero velocity is indicated by point *A*, and its distance from the lower plate is determined from geometric considerations (the similarity of the two triangles in the lower oil layer) to be

$$\frac{2.6 - y_A}{y_A} = \frac{3}{0.3} \quad \rightarrow \quad y_A = \mathbf{0.23636\ mm}$$



(*b*) The magnitudes of shear forces acting on the upper and lower surfaces of the plate are

$$F_{\text{shear, upper}} = \tau_{w,\text{upper}} A_s = \mu A_s \left|\frac{du}{dy}\right| = \mu A_s \frac{V-0}{h_1} = (0.027\ \text{N}\cdot\text{s/m}^2)(0.3\times 0.3\ \text{m}^2)\frac{3\ \text{m/s}}{1.0\times 10^{-3}\ \text{m}} = 7.29\ \text{N}$$

$$F_{\text{shear, lower}} = \tau_{w,\text{lower}} A_s = \mu A_s \left|\frac{du}{dy}\right| = \mu A_s \frac{V-V_w}{h_2} = (0.027\ \text{N}\cdot\text{s/m}^2)(0.3\times 0.3\ \text{m}^2)\frac{[3-(-0.3)]\ \text{m/s}}{2.6\times 10^{-3}\ \text{m}} = 3.08\ \text{N}$$

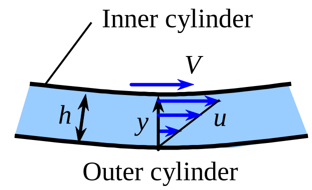
Noting that both shear forces are in the opposite direction of motion of the plate, the force *F* is determined from a force balance on the plate to be

$$F = F_{\text{shear, upper}} + F_{\text{shear, lower}} = 7.29 + 3.08 = \mathbf{10.4N}$$

Discussion Note that wall shear is a friction force between a solid and a liquid, and it acts in the opposite direction of motion.

2-82

Solution We are to determine the torque required to rotate the inner cylinder of two concentric cylinders, with the inner cylinder rotating and the outer cylinder stationary. We are also to explain what happens when the gap gets bigger.



Assumptions **1** The fluid is incompressible and Newtonian. **2** End effects (top and bottom) are negligible. **3** The gap is very small so that wall curvature effects are negligible. **4** The gap is so small that the velocity profile in the gap is linear.

Analysis (*a*) We assume a linear velocity profile between the two walls as sketched – the inner wall is moving at speed $V = \omega_i R_i$ and the outer wall is stationary. The thickness of the gap is h, and we let y be the distance from the outer wall into the fluid (towards the inner wall). Thus,

$$u = V\frac{y}{h} \quad \text{and} \quad \tau = \mu\frac{du}{dy} = \mu\frac{V}{h}$$

where

$$h = R_o - R_i \quad \text{and} \quad V = \omega_i R_i$$

Since shear stress τ has dimensions of force/area, the clockwise (mathematically negative) tangential force acting along the surface of the inner cylinder by the fluid is

$$F = -\tau A = -\mu\frac{V}{h}2\pi R_i L = -\frac{\mu\omega_i R_i}{R_o - R_i}2\pi R_i L$$

But the torque is the tangential force times the moment arm R_i. Also, we are asked for the torque required to turn the inner cylinder. This applied torque is counterclockwise (mathematically positive). Thus,

$$\boxed{\mathrm{T} = -FR_i = \frac{2\pi L\mu\omega_i R_i^3}{R_o - R_i} = \frac{2\pi L\mu\omega_i R_i^3}{h}}$$

(*b*) The above is only an **approximation because we assumed a linear velocity profile**. As long as the gap is very small, and therefore the wall curvature effects are negligible, this approximation should be very good. Another way to think about this is that when the gap is very small compared to the cylinder radii, a magnified view of the flow in the gap appears similar to flow between two infinite walls (Couette flow). However, as the gap increases, the curvature effects are no longer negligible, and the linear velocity profile is not expected to be a valid approximation. **We do not expect the velocity to remain linear as the gap increases**.

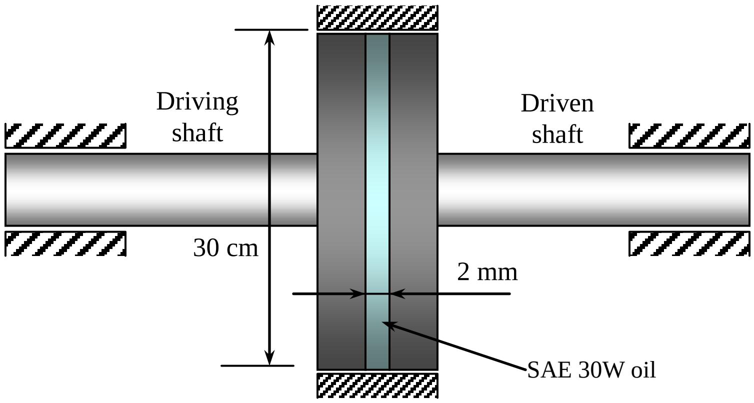
Discussion It is possible to solve for the exact velocity profile for this problem, and therefore the torque can be found analytically, but this has to wait until the differential analysis chapter.

2-83

Solution A clutch system is used to transmit torque through an oil film between two identical disks. For specified rotational speeds, the transmitted torque is to be determined.

Assumptions **1** The thickness of the oil film is uniform. **2** The rotational speeds of the disks remain constant.

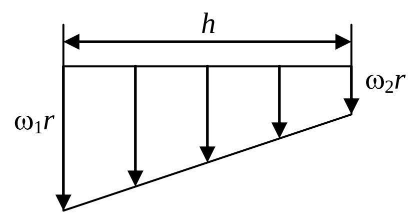
Properties The absolute viscosity of oil is given to be $\mu = 0.38$ N·s/m^2.



Analysis The disks are rotting in the same direction at different angular speeds of ω_1 and of ω_2. Therefore, we can assume one of the disks to be stationary and the other to be rotating at an angular speed of $\omega_1 - \omega_2$. The velocity gradient anywhere in the oil of film thickness h is V/h where $V = (\omega_1 - \omega_2)r$ is the tangential velocity. Then the wall shear stress anywhere on the surface of the faster disk at a distance r from the axis of rotation can be expressed as

$$\tau_w = \mu \frac{du}{dr} = \mu \frac{V}{h} = \mu \frac{(\omega_1 - \omega_2)r}{h}$$

Then the shear force acting on a differential area dA on the surface and the torque generation associated with it can be expressed as



$$dF = \tau_w dA = \mu \frac{(\omega_1 - \omega_2)r}{h}(2\pi r)dr$$

$$d\mathrm{T} = r dF = \mu \frac{(\omega_1 - \omega_2)r^2}{h}(2\pi r)dr = \frac{2\pi\mu(\omega_1 - \omega_2)}{h} r^3 dr$$

Integrating,

$$\mathrm{T} = \frac{2\pi\mu(\omega_1 - \omega_2)}{h}\int_{r=0}^{D/2} r^3 dr = \frac{2\pi\mu(\omega_1 - \omega_2)}{h}\left.\frac{r^4}{4}\right|_{r=0}^{D/2} = \frac{\pi\mu(\omega_1 - \omega_2)D^4}{32h}$$

Noting that $\omega = 2\pi\, \dot{n}$, the relative angular speed is

$$\omega_1 - \omega_2 = 2\pi\left(\dot{n}_1 - \dot{n}_2\right) = \left(2\pi \text{ rad/rev}\right)\left[\left(1450 - 1398\right) \text{ rev/min}\right]\left(\frac{1 \text{ min}}{60 \text{ s}}\right) = 5.445 \text{ rad/s},$$

Substituting, the torque transmitted is determined to be

$$\mathrm{T} = \frac{\pi(0.38\ \mathrm{N\cdot s/m^2})(5.445\ \mathrm{/s})(0.30\ \mathrm{m})^4}{32(0.002\ \mathrm{m})} = \mathbf{0.82\ N\cdot m}$$

Discussion Note that the torque transmitted is proportional to the fourth power of disk diameter, and is inversely proportional to the thickness of the oil film.

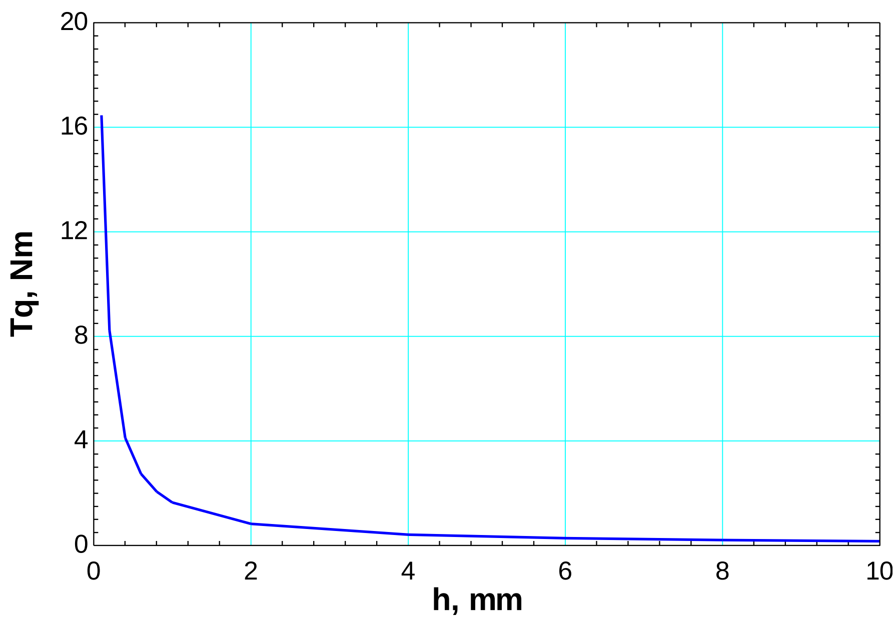
2-84

Solution We are to investigate the effect of oil film thickness on the transmitted torque.

Analysis The previous problem is reconsidered. Using EES software, the effect of oil film thickness on the torque transmitted is investigated. Film thickness varied from 0.1 mm to 10 mm, and the results are tabulated and plotted. The relation used is $\mathrm{T} = \dfrac{\pi\mu(\omega_1 - \omega_2)D^4}{32h}$. The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
mu=0.38
n1=1450 "rpm"
w1=2*pi*n1/60 "rad/s"
n2=1398 "rpm"
w2=2*pi*n2/60 "rad/s"
D=0.3 "m"
Tq=pi*mu*(w1-w2)*(D^4)/(32*h)
```

Film thickness *h*, mm	**Torque transmitted T, Nm**
0.1	16.46
0.2	8.23
0.4	4.11
0.6	2.74
0.8	2.06
1	1.65
2	0.82
4	0.41
6	0.27
8	0.21
10	0.16



Conclusion Torque transmitted is inversely proportional to oil film thickness, and the film thickness should be as small as possible to maximize the transmitted torque.

Discussion To obtain the solution in EES, we set up a parametric table, specify *h*, and let EES calculate T for each value of *h*.

2-85

Solution The viscosities of carbon dioxide at two temperatures are given. The constants of Sutherland correlation for carbon dioxide are to be determined and the viscosity of carbon dioxide at a specified temperature is to be predicted and compared to the value in table A-10.

Analysis Sutherland correlation is given by Eq. 2–32 as

$$\mu = \frac{a\sqrt{T}}{1 + b/T}$$

where T is the absolute temperature. Substituting the given values we have

$$\mu_1 = \frac{a\sqrt{T_1}}{1 + b/T_1} = \frac{a\sqrt{50 + 273.15}}{1 + \dfrac{b}{50 + 273.15}} \rightarrow 1.612 \times 10^{-5} = \frac{a\sqrt{323.15}}{1 + \dfrac{b}{323.15}}$$

$$\mu_2 = \frac{a\sqrt{T_2}}{1 + b/T_2} = \frac{a\sqrt{200 + 273.15}}{1 + \dfrac{b}{200 + 273.15}} \rightarrow 2.276 \times 10^{-5} = \frac{a\sqrt{473.15}}{1 + \dfrac{b}{473.15}}$$

which is a nonlinear system of two algebraic equations. Using EES or any other computer code, one finds the following result:

$$a = 1.633 \times 10^{-6} \mathrm{kg/(m \cdot s \cdot K^{1/2})} \qquad b = 265.5 \text{ K}$$

Using these values the Sutherland correlation becomes

$$\mu = \frac{1.633 \times 10^{-6}\sqrt{T}}{1 + 265.5/T}$$

Therefore the viscosity at 100 ℃ is found to be

$$\mu = \frac{1.633 \times 10^{-6}\sqrt{373.15}}{1 + 265.5/373.15} = 1.843 \times 10^{-5} \mathrm{Pa \cdot s}$$

The agreement is perfect and within approximately 0.1%.

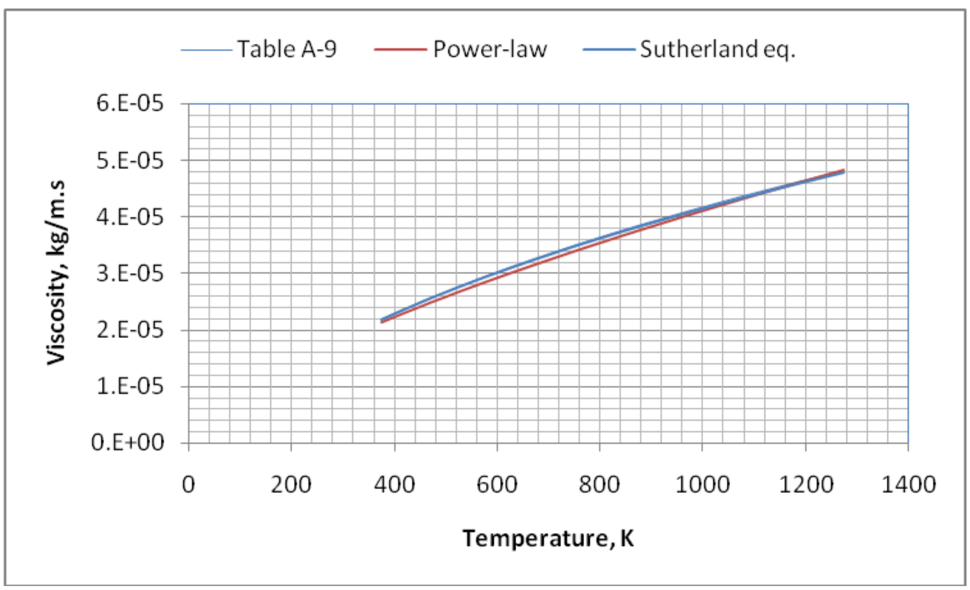
2-86

Solution The variation of air viscosity for a specified temperature range is to be evaluated using power and Sutherland laws and compared to values in Table A-9.

Analysis For the reference temperature we have $\mu_0 = 1.729 \times 10^{-5}\text{kg/m}\cdot\text{s}$ (Table A–9). Using an Excel sheet, we end up with the following calculations:

T (K)	Table A-9	Power-law	Sutherland	PL-Error %	Suth Error %
373	2.181E-05	2.12848E-05	2.17277E-05	2.41	0.38
393	2.264E-05	2.20382E-05	2.25649E-05	2.66	0.33
413	2.345E-05	2.27789E-05	2.33802E-05	2.86	0.30
433	0.0000242	2.35078E-05	2.41752E-05	2.86	0.10
453	2.504E-05	2.42255E-05	2.4951E-05	3.25	0.36
473	2.577E-05	2.49326E-05	2.57089E-05	3.25	0.24
523	0.0000276	2.66583E-05	2.75316E-05	3.41	0.25
573	2.934E-05	2.83297E-05	2.92627E-05	3.44	0.26
623	3.101E-05	2.9953E-05	3.09135E-05	3.41	0.31
673	3.261E-05	3.15332E-05	3.24935E-05	3.30	0.36
723	3.414E-05	3.30748E-05	3.40104E-05	3.12	0.38
773	3.563E-05	3.45811E-05	3.54707E-05	2.94	0.45
873	3.846E-05	3.74996E-05	3.82427E-05	2.50	0.56
973	4.111E-05	4.03082E-05	4.08449E-05	1.95	0.64
1073	4.362E-05	4.30219E-05	4.33038E-05	1.37	0.73
1173	0.000046	4.56524E-05	4.56397E-05	0.76	0.78
1273	4.826E-05	4.82088E-05	4.78688E-05	0.11	0.81

Following plot shows the accuracy of both model.



2-87

Solution For flow over a plate, the variation of velocity with distance is given. A relation for the wall shear stress is to be obtained.

Assumptions The fluid is Newtonian.

Analysis Noting that $u(y) = ay - by^2$, wall shear stress is determined from its definition to be

$$\tau_w = \mu \left.\frac{du}{dy}\right|_{y=0} = \mu \left.\frac{d(ay-by^2)}{dy}\right|_{y=0} = \mu(a-2by)\big|_{y=0} = \boldsymbol{a\mu}$$

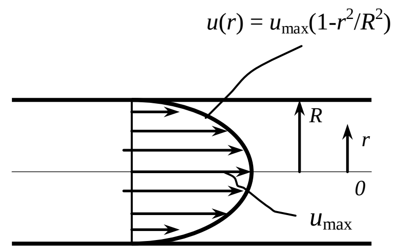
Discussion Note that shear stress varies with vertical distance in this case.

2-88

Solution The velocity profile for laminar one-dimensional flow through a circular pipe is given. A relation for friction drag force exerted on the pipe and its numerical value for water are to be determined.

Assumptions **1** The flow through the circular pipe is one-dimensional. **2** The fluid is Newtonian.

Properties The viscosity of water at 20°C is given to be 0.0010 kg/m⋅s.



Analysis (*a*) The velocity profile is given by $u(r) = u_{max}\left(1 - \frac{r^2}{R^2}\right)$

where R is the radius of the pipe, r is the radial distance from the center of the pipe, and u_{max} is the maximum flow velocity, which occurs at the center, $r = 0$. The shear stress at the pipe surface is expressed as

$$\tau_w = -\mu \left.\frac{du}{dr}\right|_{r=R} = -\mu u_{max} \frac{d}{dr}\left(1 - \frac{r^2}{R^2}\right)_{r=R} = -\mu u_{max} \left.\frac{-2r}{R^2}\right|_{r=R} = \frac{2\mu u_{max}}{R}$$

Note that the quantity du/dr is negative in pipe flow, and the negative sign is added to the τ_w relation for pipes to make shear stress in the positive (flow) direction a positive quantity. (Or, $du/dr = -du/dy$ since $y = R - r$). Then the friction drag force exerted by the fluid on the inner surface of the pipe becomes

$$F_D = \tau_w A_s = \frac{2\mu u_{max}}{R}(2\pi RL) = \boldsymbol{4\pi\mu L u_{max}}$$

(*b*) Substituting the values we get $F_D = 4\pi\mu L u_{max} = 4\pi(0.0010\text{ kg/m}\cdot\text{s})(30\text{ m})(3\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \textbf{1.13 N}$

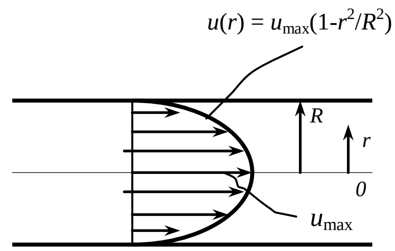
Discussion In the entrance region and during turbulent flow, the velocity gradient is greater near the wall, and thus the drag force in such cases will be greater.

2-89

Solution The velocity profile for laminar one-dimensional flow through a circular pipe is given. A relation for friction drag force exerted on the pipe and its numerical value for water are to be determined.

Assumptions **1** The flow through the circular pipe is one-dimensional. **2** The fluid is Newtonian.

Properties The viscosity of water at 20°C is given to be 0.0010 kg/m·s.



Analysis (*a*) The velocity profile is given by $u(r) = u_{max}\left(1 - \frac{r^2}{R^2}\right)$

where R is the radius of the pipe, r is the radial distance from the center of the pipe, and u_{max} is the maximum flow velocity, which occurs at the center, r = 0. The shear stress at the pipe surface can be expressed as

$$\tau_w = -\mu \left.\frac{du}{dr}\right|_{r=R} = -\mu u_{max} \frac{d}{dr}\left(1 - \frac{r^2}{R^2}\right)_{r=R} = -\mu u_{max} \left.\frac{-2r}{R^2}\right|_{r=R} = \frac{2\mu u_{max}}{R}$$

Note that the quantity *du/dr* is negative in pipe flow, and the negative sign is added to the τ_w relation for pipes to make shear stress in the positive (flow) direction a positive quantity. (Or, $du/dr = -du/dy$ since $y = R - r$). Then the friction drag force exerted by the fluid on the inner surface of the pipe becomes

$$F_D = \tau_w A_s = \frac{2\mu u_{max}}{R}(2\pi RL) = \boldsymbol{4\pi\mu L u_{max}}$$

(*b*) Substituting, we get $F_D = 4\pi\mu L u_{max} = 4\pi(0.0010\text{ kg/m}\cdot\text{s})(30\text{ m})(7\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{2.64\ N}$

Discussion In the entrance region and during turbulent flow, the velocity gradient is greater near the wall, and thus the drag force in such cases will be larger.

2-90

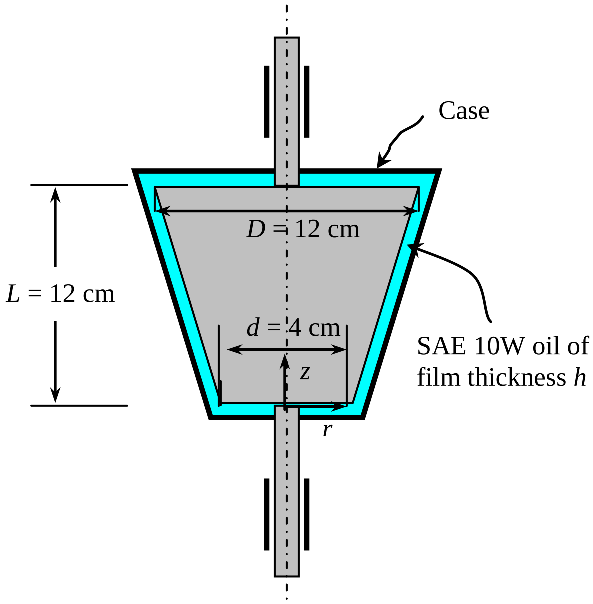
Solution A frustum shaped body is rotating at a constant angular speed in an oil container. The power required to maintain this motion and the reduction in the required power input when the oil temperature rises are to be determined.

Assumptions The thickness of the oil layer remains constant.

Properties The absolute viscosity of oil is given to be μ = 0.1 Pa⋅s = 0.1 N⋅s/m^2 at 20°C and 0.0078 Pa⋅s at 80°C.

Analysis The velocity gradient anywhere in the oil of film thickness h is V/h where $V = \omega r$ is the tangential velocity. Then the wall shear stress anywhere on the surface of the frustum at a distance r from the axis of rotation is

$$\tau_w = \mu \frac{du}{dr} = \mu \frac{V}{h} = \mu \frac{\omega r}{h}$$



The shear force acting on differential area dA on the surface, the torque it generates, and the shaft power associated with it are expressed as

$$dF = \tau_w dA = \mu \frac{\omega r}{h} dA \qquad d\mathrm{T} = r dF = \mu \frac{\omega r^2}{h} dA$$

$$\mathrm{T} = \frac{\mu\omega}{h}\int_A r^2 dA \qquad \dot{W}_{\mathrm{sh}} = \omega \mathrm{T} = \frac{\mu\omega^2}{h}\int_A r^2 dA$$

<u>Top surface</u>: For the top surface, $dA = 2\pi r dr$. Substituting and integrating,

$$\dot{W}_{\mathrm{sh,top}} = \frac{\mu\omega^2}{h}\int_{r=0}^{D/2} r^2 (2\pi r) dr = \frac{2\pi\mu\omega^2}{h}\int_{r=0}^{D/2} r^3 dr = \frac{2\pi\mu\omega^2}{h}\left.\frac{r^4}{4}\right|_{r=0}^{D/2} = \frac{\pi\mu\omega^2 D^4}{32h}$$

<u>Bottom surface</u>: A relation for the bottom surface is obtained by replacing D by d, $\dot{W}_{\mathrm{sh,bottom}} = \dfrac{\pi\mu\omega^2 d^4}{32h}$

<u>Side surface</u>: The differential area for the side surface can be expressed as $dA = 2\pi r dz$. From geometric considerations, the variation of radius with axial distance is expressed as $r = \dfrac{d}{2} + \dfrac{D-d}{2L} z$.

Differentiating gives $dr = \dfrac{D-d}{2L} dz$ or $dz = \dfrac{2L}{D-d} dr$. Therefore, $dA = 2\pi dz = \dfrac{4\pi L}{D-d} r dr$. Substituting and integrating,

$$\dot{W}_{\mathrm{sh,top}} = \frac{\mu\omega^2}{h}\int_{r=0}^{D/2} r^2 \frac{4\pi L}{D-d} r dr = \frac{4\pi\mu\omega^2 L}{h(D-d)}\int_{r=d/2}^{D/2} r^3 dr = \frac{4\pi\mu\omega^2 L}{h(D-d)}\left.\frac{r^4}{4}\right|_{r=d/2}^{D/2} = \frac{\pi\mu\omega^2 L(D^2 - d^2)}{16h(D-d)}$$

Then the total power required becomes

$$\dot{W}_{\mathrm{sh,total}} = \dot{W}_{\mathrm{sh,top}} + \dot{W}_{\mathrm{sh,bottom}} + \dot{W}_{\mathrm{sh,side}} = \frac{\pi\mu\omega^2 D^4}{32h}\left[1 + (d/D)^4 + \frac{2L[1-(d/D)^4)]}{D-d}\right],$$

where d/D = 4/12 = 1/3. Substituting,

$$\dot{W}_{\mathrm{sh,total}} = \frac{\pi(0.1\,\mathrm{N\cdot s/m^2})(200\,/\mathrm{s})^2(0.12\,\mathrm{m})^4}{32(0.0012\,\mathrm{m})}\left[1 + (1/3)^4 + \frac{2(0.12\,\mathrm{m})[1-(1/3)^4)]}{(0.12-0.04)\,\mathrm{m}}\right]\left(\frac{1\,\mathrm{W}}{1\,\mathrm{Nm/s}}\right) = \mathbf{270\,W}$$

Noting that power is proportional to viscosity, the power required at 80°C is

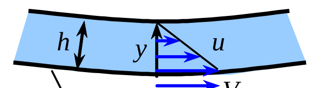
$$\dot{W}_{\mathrm{sh,total,80°C}} = \frac{\mu_{80°C}}{\mu_{20°C}}\dot{W}_{\mathrm{sh,total,20°C}} = \frac{0.0078\,\mathrm{N\cdot s/m^2}}{0.1\,\mathrm{N\cdot s/m^2}}(270\,\mathrm{W}) = 21.1\,\mathrm{W}$$

Therefore, the reduction in the requires power input at 80°C is $\text{Reduction} = \dot{W}_{\text{sh, total, 20°C}} - \dot{W}_{\text{sh, total, 80°C}} = 270 - 21.1 = \mathbf{249\,W}$, which is about 92%.

Discussion Note that the power required to overcome shear forces in a viscous fluid greatly depends on temperature.

2-91

Solution We are to determine the torque required to rotate the outer cylinder of two concentric cylinders, with the outer cylinder rotating and the inner cylinder stationary.



Assumptions **1** The fluid is incompressible and Newtonian. **2** End effects (top and bottom) are negligible. **3** The gap is very small so that wall curvature effects are negligible. **4** The gap is so small that the velocity profile in the gap is linear.

Analysis We assume a linear velocity profile between the two walls – the outer wall is moving at speed $V = \omega_o R_o$ and the inner wall is stationary. The thickness of the gap is h, and we let y be the distance from the outer wall into the fluid (towards the inner wall) as sketched. Thus,

$$u = V\frac{h-y}{h} \quad \text{and} \quad \tau = \mu\frac{du}{dy} = -\mu\frac{V}{h}$$

where

$$h = R_o - R_i \quad \text{and} \quad V = \omega_o R_o$$

Since shear stress τ has dimensions of force/area, the clockwise (mathematically negative) tangential force acting along the surface of the outer cylinder by the fluid is

$$F = -\tau A = -\mu\frac{V}{h}2\pi R_o L = -\frac{\mu\omega_o R_o}{R_o - R_i}2\pi R_o L$$

But the torque is the tangential force times the moment arm R_o. Also, we are asked for the torque required to turn the inner cylinder. This applied torque is counterclockwise (mathematically positive). Thus,

$$\boxed{\mathrm{T} = -FR_o = \frac{2\pi L\mu\omega_o R_o^{\;3}}{R_o - R_i} = \frac{2\pi L\mu\omega_o R_o^{\;3}}{h}}$$

Discussion The above is only an approximation because we assumed a linear velocity profile. As long as the gap is very small, and therefore the wall curvature effects are negligible, this approximation should be very good. It is possible to solve for the exact velocity profile for this problem, and therefore the torque can be found analytically, but this has to wait until the differential analysis chapter.

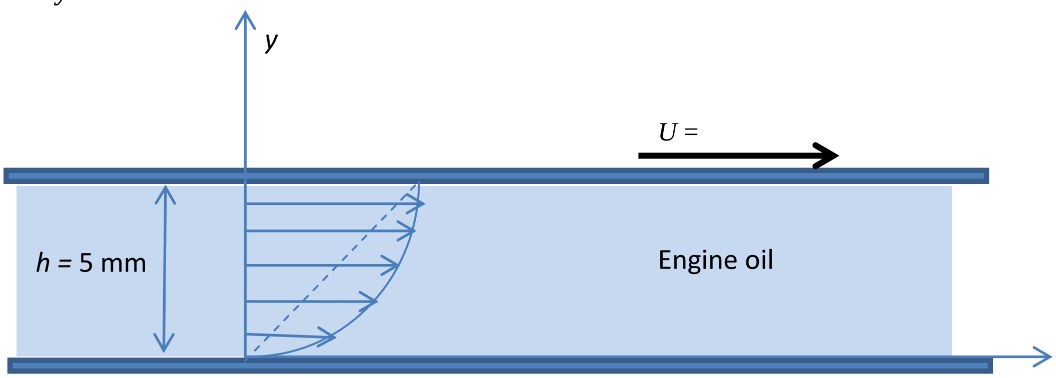
2-92

Solution A large plate is pulled at a constant speed over a fixed plate. The space between the plates is filled with engine oil. The shear stress developed on the upper plate and its direction are to be determined for parabolic and linear velocity profile cases.

Assumptions **1** The thickness of the plate is negligible.

Properties The viscosity of oil is $\mu = 0.8374$ Pa·s (Table A-7).

Analysis



Considering a parabolic profile we would have $V^2 = ky$, where k is a constant. Since $V = U = 4\ \mathrm{m/s}$ when $y = h = 5\ \mathrm{mm} = 5 \times 10^{-3}\mathrm{m}$, we write

$$\left(4\frac{\mathrm{m}}{\mathrm{s}}\right)^2 = k \times (5 \times 10^{-3}\ \mathrm{m}) \rightarrow k = 3200\ \mathrm{m^2/s}$$

Then the velocity profile becomes

$$V^2 = 3200y \rightarrow V = 56.568\sqrt{y}$$

Assuming Newtonian behavior, the shear stress on the upper wall is

$$\tau = \mu\frac{dV}{dy} = \mu\frac{d}{dy}\left(56.568\sqrt{y}\right) = 37.712\mu y^{3/2}\Big|\Big|_{y=0}^{y=5\times10^{-3}} = 37.712 \times (0.8374\ \mathrm{Pa\cdot s}) \times (0.0133)$$

or

$$\tau = 0.421\ \mathrm{N/m^2}$$

Since dynamic viscosity of oil is $0.8374\ \mathrm{Pa\cdot s}$ (see Table A–7). If we assume a linear profile we will have

$$\frac{dV}{dy} = \frac{U}{h} = \frac{4\ \mathrm{m/s}}{5 \times 10^{-3}\mathrm{m}} = 800\ \mathrm{s^{-1}}$$

Then the shear stress in this case would be

$$\tau = \mu\frac{dV}{dy} = \mu\frac{U}{h} = (0.8374\ \mathrm{Pa\cdot s}) \times (800) \approx 670\ \mathrm{N/m^2}$$

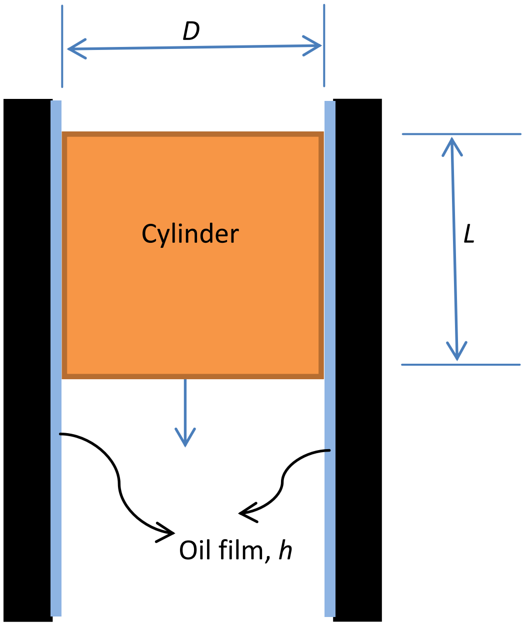
Therefore we conclude that the linear assumption is not realistic since it gives over prediction.

2-93

Solution A cylinder slides down from rest in a vertical tube whose inner surface is covered by oil. An expression for the velocity of the cylinder as a function of time is to be derived.

Assumptions **1** Velocity profile in the oil film is linear.

Analysis



Assuming a linear velocity profile in the oil film the drag force due to wall shear stress can be expressed as

$$F_D = \mu \frac{dV}{dy} A = \mu \frac{V}{h} \pi D L = kV$$

where V is the instantaneous velocity of the cylinder and

$$k = \mu \frac{\pi D L}{h}$$

Applying Newton's second law of motion for the cylinder, we write

$$mg - kV = m \frac{dV}{dt}$$

where t is the time. This is a first-order linear equation and can be expressed in standard form as follows:

$$\frac{dV}{dt} + \frac{k}{m} V = g \quad \text{with} \quad V(0) = 0$$

whose solution is obtained to be

$$V(t) = \frac{mg}{k}\left(1 - e^{-(k/m)t}\right)$$

As $t \to \infty$ the second term will vanish leaving us with

$$V(t) = \frac{mg}{k}$$

which is constant. This constant is referred to as "limit velocity, V_L". Rearranging for viscosity, we have

$$\mu = \frac{mgh}{\pi D L V_L}$$

Therefore this equation enables us to estimate dynamic viscosity of oil provided that the limit velocity of the cylinder is precisely measured.

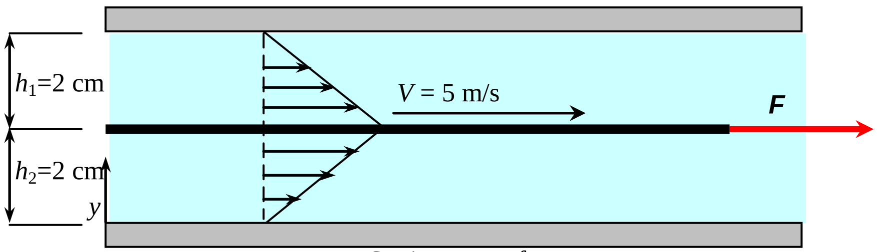
2-94

Solution A thin flat plate is pulled horizontally through the mid plane of an oil layer sandwiched between two stationary plates. The force that needs to be applied on the plate to maintain this motion is to be determined for this case and for the case when the plate .

Assumptions **1** The thickness of the plate is negligible. **2** The velocity profile in each oil layer is linear.

Properties The absolute viscosity of oil is given to be μ = 0.9 N·s/m^2.

Analysis The velocity profile in each oil layer relative to the fixed wall is as shown in the figure.



The magnitudes of shear forces acting on the upper and lower surfaces of the moving thin plate are

$$F_{\text{shear, upper}} = \tau_{w,\text{ upper}} A_s = \mu A_s \left|\frac{du}{dy}\right| = \mu A_s \frac{V-0}{h_1} = (0.9\ \text{N}\cdot\text{s/m}^2)(0.5\times 2\ \text{m}^2)\frac{5\ \text{m/s}}{0.02\ \text{m}} = 225\ \text{N}$$

$$F_{\text{shear, lower}} = \tau_{w,\text{ lower}} A_s = \mu A_s \left|\frac{du}{dy}\right| = \mu A_s \frac{V-V_w}{h_2} = (0.9\ \text{N}\cdot\text{s/m}^2)(0.5\times 2\ \text{m}^2)\frac{5\ \text{m/s}}{0.02\ \text{m}} = 225\ \text{N}$$

Noting that both shear forces are in the opposite direction of motion of the plate, the force F is determined from a force balance on the plate to be

$$F = F_{\text{shear, upper}} + F_{\text{shear, lower}} = 225 + 225 = \mathbf{450\ N}$$

When the plate is 1 cm from the bottom surface and 3 cm from the top surface, the force F becomes

$$F_{\text{shear, upper}} = \tau_{w,\text{ upper}} A_s = \mu A_s \left|\frac{du}{dy}\right| = \mu A_s \frac{V-0}{h_1} = (0.9\ \text{N}\cdot\text{s/m}^2)(0.5\times 2\ \text{m}^2)\frac{5\ \text{m/s}}{0.03\ \text{m}} = 150\ \text{N}$$

$$F_{\text{shear, lower}} = \tau_{w,\text{ lower}} A_s = \mu A_s \left|\frac{du}{dy}\right| = \mu A_s \frac{V-0}{h_2} = (0.9\ \text{N}\cdot\text{s/m}^2)(0.5\times 2\ \text{m}^2)\frac{5\ \text{m/s}}{0.01\ \text{m}} = 450\ \text{N}$$

Noting that both shear forces are in the opposite direction of motion of the plate, the force F is determined from a force balance on the plate to be

$$F = F_{\text{shear, upper}} + F_{\text{shear, lower}} = 150 + 450 = \mathbf{600\ N}$$

Discussion Note that the relative location of the thin plate affects the required force significantly.

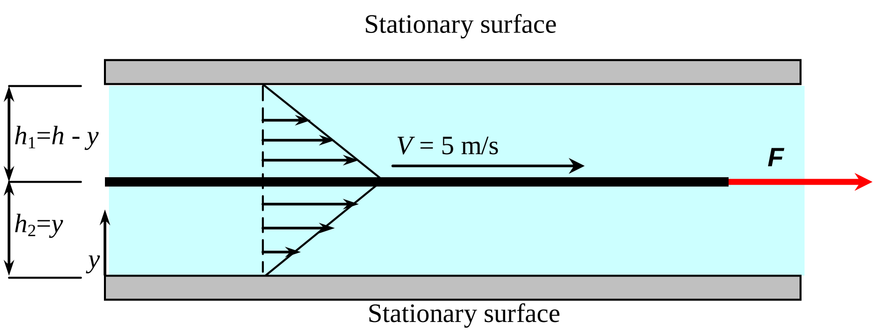
2-95

Solution A thin flat plate is pulled horizontally through the mid plane of an oil layer sandwiched between two stationary plates. The force that needs to be applied on the plate to maintain this motion is to be determined for this case and for the case when the plate .

Assumptions **1** The thickness of the plate is negligible. **2** The velocity profile in each oil layer is linear.

Properties The absolute viscosity of oil is μ = 0.9 N·s/m^2 in the lower part, and 4 times that in the upper part.

Analysis We measure vertical distance *y* from the lower plate. The total distance between the stationary plates is $h = h_1 + h_2 = 4\text{ cm}$, which is constant. Then the distance of the moving plate is *y* from the lower plate and *h* – *y* from the upper plate, where *y* is variable.



The shear forces acting on the upper and lower surfaces of the moving thin plate are

$$F_{\text{shear, upper}} = \tau_{w,\text{ upper}} A_s = \mu_{\text{upper}} A_s \left|\frac{du}{dy}\right| = \mu_{\text{upper}} A_s \frac{V}{h-y}$$

$$F_{\text{shear, lower}} = \tau_{w,\text{ lower}} A_s = \mu_{\text{lower}} A_s \left|\frac{du}{dy}\right| = \mu_{\text{lower}} A_s \frac{V}{y}$$

Then the total shear force acting on the plate becomes

$$F = F_{\text{shear, upper}} + F_{\text{shear, lower}} = \mu_{\text{upper}} A_s \frac{V}{h-y} + \mu_{\text{lower}} A_s \frac{V}{h-y} = A_s V\left(\frac{\mu_{\text{upper}}}{h-y} + \frac{\mu_{\text{lower}}}{y}\right)$$

The value of *y* that will minimize the force *F* is determined by setting $\dfrac{dF}{dy} = 0$:

$$\frac{\mu_{\text{upper}}}{(h-y)^2} - \frac{\mu_{\text{lower}}}{y^2} = 0 \quad \rightarrow \quad \frac{y}{h-y} = \sqrt{\frac{\mu_{\text{lower}}}{\mu_{\text{upper}}}}$$

Solving for *y* and substituting, the value of *y* that minimizes the shear force is determined to be

$$y = \frac{\sqrt{\mu_{\text{lower}} / \mu_{\text{upper}}}}{1 - \sqrt{\mu_{\text{lower}} / \mu_{\text{upper}}}} h = \frac{\sqrt{1/4}}{1 - \sqrt{1/4}} (4\text{ cm}) = \mathbf{1\ cm}$$

Discussion By showing that $\dfrac{d^2F}{dy^2} > 0$ at *y* = 1 cm, it can be verified that *F* is indeed a minimum at that location and not a maximum.

Surface Tension and Capillary Effect

2-96C

Solution We are to define and discuss surface tension.

Analysis The **magnitude of the pulling force at the surface of a liquid per unit length** is called *surface tension* σ_s. It is caused by the attractive forces between the molecules. The surface tension is also surface energy (per unit area) since it represents the stretching work that needs to be done to increase the surface area of the liquid by a unit amount.

Discussion Surface tension is the cause of some very interesting phenomena such as capillary rise and insects that can walk on water.

2-97C

Solution We are to determine whether the level of liquid in a tube will rise or fall due to the capillary effect.

Analysis The **liquid level in the tube will drop** since the contact angle is greater than 90°, and cos(110°) < 0.

Discussion This liquid must be a non-wetting liquid when in contact with the tube material. Mercury is an example of a non-wetting liquid with a contact angle (with glass) that is greater than 90^o.

2-98C

Solution We are to define and discuss the capillary effect.

Analysis The *capillary effect* is the **rise or fall of a liquid in a small-diameter tube inserted into the liquid**. It is caused by the net effect of the *cohesive forces* (the forces between like molecules, like water) and *adhesive forces* (the forces between unlike molecules, like water and glass). **The capillary effect is proportional to the cosine of the *contact angle***, which is the angle that the tangent to the liquid surface makes with the solid surface at the point of contact.

Discussion The contact angle determines whether the *meniscus* at the top of the column is concave or convex.

2-99C

Solution We are to analyze the pressure difference between inside and outside of a soap bubble.

Analysis The **pressure inside a soap bubble is greater than the pressure outside**, as evidenced by the stretch of the soap film.

Discussion You can make an analogy between the soap film and the skin of a balloon.

2-100C

Solution We are to compare the capillary rise in small and large diameter tubes.

Analysis The capillary rise is inversely proportional to the diameter of the tube, and thus **capillary rise is greater in the smaller-diameter tube**.

Discussion Note however, that if the tube diameter is large enough, there is no capillary rise (or fall) at all. Rather, the upward (or downward) rise of the liquid occurs only near the tube walls; the elevation of the middle portion of the liquid in the tube does not change for large diameter tubes.

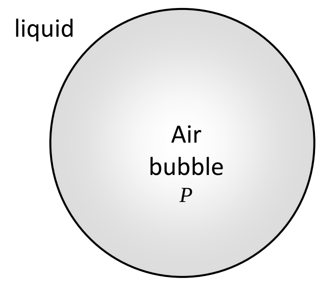
2-101

Solution An air bubble in a liquid is considered. The pressure difference between the inside and outside the bubble is to be determined.

Properties The surface tension σ_s is given for two cases to be 0.08 and 0.12 N/m.

Analysis Considering that an air bubble in a liquid has only one interface, he pressure difference between the inside and the outside of the bubble is determined from

$$\Delta P_{\text{bubble}} = P_i - P_0 = \frac{2\sigma_s}{R}$$



Substituting, the pressure difference is determined to be:

(a) σ_s = 0.08 N/m: $\Delta P_{\text{bubble}} = \dfrac{2(0.08\text{ N/m})}{0.00015/2\text{ m}} = 2133\text{ N/m}^2 = \mathbf{2.13\ kPa}$

(b) σ_s = 0.12 N/m: $\Delta P_{\text{bubble}} = \dfrac{2(0.12\text{ N/m})}{0.00015/2\text{ m}} = 3200\text{ N/m}^2 = \mathbf{3.20\ kPa}$

Discussion Note that a small gas bubble in a liquid is highly pressurized. The smaller the bubble diameter, the larger the pressure inside the bubble.

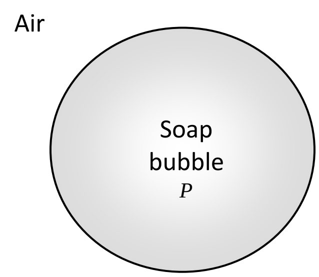
2-102E

Solution A soap bubble is enlarged by blowing air into it. The required work input is to be determined.

Properties The surface tension of solution is given to be σ_s = 0.0027 lbf/ft.

Analysis The work associated with the stretching of a film is the surface tension work, and is expressed in differential form as $\delta W_s = \sigma_s dA_s$. Noting that surface tension is constant, the surface tension work is simply surface tension multiplied by the change in surface area,

$$W_s = \sigma_s (A_2 - A_1) = 2\pi\sigma_s (D_2^2 - D_1^2)$$



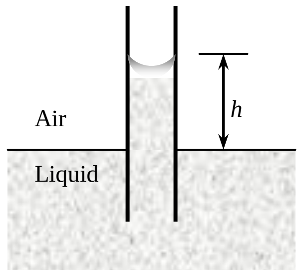
The factor 2 is due to having two surfaces in contact with air. Substituting, the required work input is determined to be

$$W_s = 2\pi(0.0027\text{ lbf/ft})\left((2.7/12\text{ ft})^2 - (2.4/12\text{ ft})^2\right)\left(\frac{1\text{ Btu}}{778.169\text{ lbf}\cdot\text{ft}}\right) = \mathbf{2.32\times10^{-7}\ Btu}$$

Discussion Note that when a bubble explodes, an equivalent amount of energy is released to the environment.

2-103

Solution A glass tube is inserted into a liquid, and the capillary rise is measured. The surface tension of the liquid is to be determined.



Assumptions **1** There are no impurities in the liquid, and no contamination on the surfaces of the glass tube. **2** The liquid is open to the atmospheric air.

Properties The density of the liquid is given to be 960 kg/m^3. The contact angle is given to be 15°.

Analysis Substituting the numerical values, the surface tension is determined from the capillary rise relation to be

$$\sigma_s = \frac{\rho g R h}{2\cos\phi} = \frac{(960\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.0012/2\text{ m})(0.005\text{ m})}{2(\cos 15°)}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{0.0146 N/m}$$

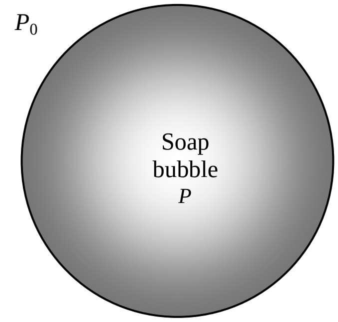
Discussion Since surface tension depends on temperature, the value determined is valid at the liquid's temperature.

2-104

Solution The diameter of a soap bubble is given. The gage pressure inside the bubble is to be determined.

Assumptions The soap bubble is in atmospheric air.

Properties The surface tension of soap water at 20°C is σ_s = 0.025 N/m.



Analysis The pressure difference between the inside and the outside of a bubble is given by

$$\Delta P_{\text{bubble}} = P_i - P_0 = \frac{4\sigma_s}{R}$$

In the open atmosphere P_0 = P_{atm}, and thus ΔP_{bubble} is equivalent to the gage pressure. Substituting,

$$D = 0.200\text{ cm: } P_{i,gage} = \Delta P_{\text{bubble}} = \frac{4(0.025\text{ N/m})}{(0.00200/2)\text{ m}} = 100\text{ N/m}^2 = 100\text{ Pa}$$

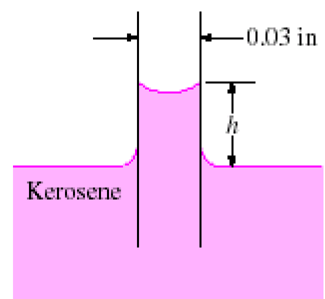
$$D = 5.00\text{ cm: } P_{i,gage} = \Delta P_{\text{bubble}} = \frac{4(0.025\text{ N/m})}{(0.0500/2)\text{ m}} = 4\text{ N/m}^2 = 4\text{ Pa}$$

Discussion Note that the gage pressure in a soap bubble is inversely proportional to the radius (or diameter). Therefore, the excess pressure is larger in smaller bubbles.

2-105E

Solution A slender glass tube is inserted into kerosene. The capillary rise of kerosene in the tube is to be determined.

Assumptions **1** There are no impurities in the kerosene, and no contamination on the surfaces of the glass tube. **2** The kerosene is open to the atmospheric air.



Properties The surface tension of kerosene-glass at 68°F (20°C) is σ_s = 0.028×0.06852 = 0.00192 lbf/ft. The density of kerosene at 68°F is ρ = 51.2 lbm/ft^3. The contact angle of kerosene with the glass surface is given to be 26°.

Analysis Substituting the numerical values, the capillary rise is determined to be

$$h = \frac{2\sigma_s \cos\phi}{\rho g R} = \frac{2(0.00192\text{ lbf/ft})(\cos 26°)}{(51.2\text{ lbm/ft}^3)(32.2\text{ ft/s}^2)(0.015/12\text{ ft})}\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)$$
$$= 0.0539\text{ ft} = \mathbf{0.650\ in}$$

Discussion The capillary rise in this case more than half of an inch, and thus it is clearly noticeable.

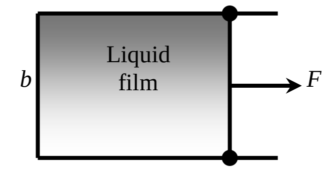
2-106

Solution The force acting on the movable wire of a liquid film suspended on a U-shaped wire frame is measured. The surface tension of the liquid in the air is to be determined.

Assumptions **1** There are no impurities in the liquid, and no contamination on the surfaces of the wire frame. **2** The liquid is open to the atmospheric air.

Analysis Substituting the numerical values, the surface tension is determined from the surface tension force relation to be

$$\sigma_s = \frac{F}{2b} = \frac{0.024\ \text{N}}{2(0.08\ \text{m})} = \mathbf{0.15\ N/m}$$



Discussion The surface tension depends on temperature. Therefore, the value determined is valid at the temperature of the liquid.

2-107

Solution A capillary tube is immersed vertically in water. The height of water rise in the tube is to be determined.

Assumptions **1** There are no impurities in water, and no contamination on the surfaces of the tube.. **2** Water is open to the atmospheric air.

Analysis The capillary rise is determined from Eq. 2–38 to be

$$h = \frac{2\sigma_s}{\rho g R}\cos\emptyset = \frac{2\times(1\ \text{N/m})\times\cos 6°}{(1000\ \text{kg/m}^3)\times(9.81\ \text{m/s}^2)\times(0.6\times10^{-3}\text{m})} = 0.338\ \text{m}$$

2-108

Solution A capillary tube is immersed vertically in water. The maximum capillary rise and tube diameter for the maximum rise case are to be determined.

Assumptions **1** There are no impurities in water, and no contamination on the surfaces of the tube. **2** Water is open to the atmospheric air.

Properties The surface tension is given to be σ_s = 1 N/m.

Analysis At the liquid side of the meniscus $P = 2\ \text{kPa}$. Therefore the capillary rise would be

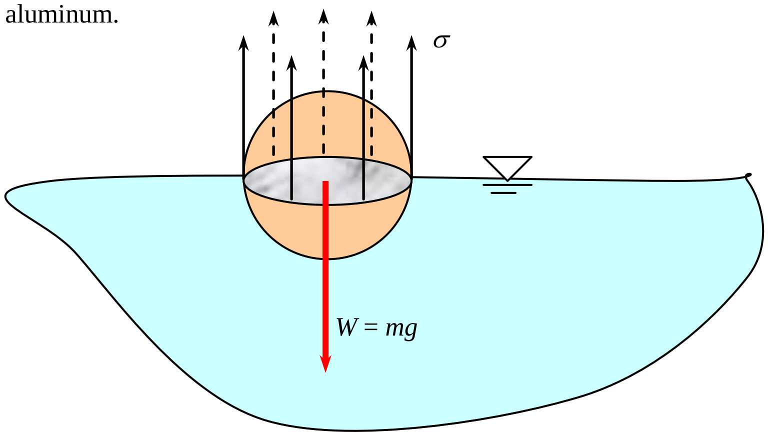
$$h = \frac{P_{\text{atm}} - P}{\rho g} = \frac{(101325 - 2000)\times10^3\text{Pa}}{(1000\ \text{kg/m}^3)\times(9.81\ \text{m/s}^2)} = 10.12\ \text{m}$$

Then the tube diameter needed for this capillary rise is, from Eq. 3–38,

$$R = \frac{2\sigma_s}{\rho g h}\cos\emptyset = \frac{2\times(1\ \text{N/m})\times\cos 6°}{(1000\ \text{kg/m}^3)\times(9.81\ \text{m/s}^2)\times(10.12\ \text{m})} \cong 2\times10^{-5}\text{m} = 20\ \mu\text{m}$$

2-109

Solution A steel ball floats on water due to the surface tension effect. The maximum diameter of the ball is to be determined, and the calculations are to be repeated for aluminum.



Assumptions **1** The water is pure, and its temperature is constant. **2** The ball is dropped on water slowly so that the inertial effects are negligible. **3** The contact angle is taken to be 0° for maximum diameter.

Properties The surface tension of water at 20°C is σ_s = 0.073 N/m. The contact angle is taken to be 0°. The densities of steel and aluminum are given to be ρ_{steel} = 7800 kg/m^3 and ρ_{Al} = 2700 kg/m^3.

Analysis The surface tension force and the weight of the ball can be expressed as

$$F_s = \pi D\sigma_s \quad \text{and} \quad W = mg = \rho g \boldsymbol{V} = \rho g \pi D^3/6$$

When the ball floats, the net force acting on the ball in the vertical direction is zero. Therefore, setting $F_s = W$ and solving for diameter D gives $D = \sqrt{\dfrac{6\sigma_s}{\rho g}}$. Substititing the known quantities, the maximum diameters for the steel and aluminum balls become

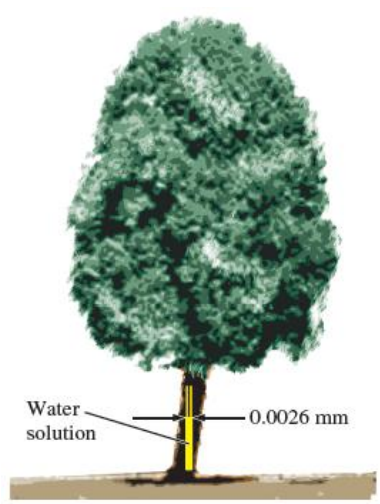
$$D_{steel} = \sqrt{\frac{6\sigma_s}{\rho g}} = \sqrt{\frac{6(0.073\text{ N/m})}{(7800\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)} = 2.4\times10^{-3}\text{ m} = \mathbf{2.4\,mm}$$

$$D_{Al} = \sqrt{\frac{6\sigma_s}{\rho g}} = \sqrt{\frac{6(0.073\text{ N/m})}{(2700\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)} = 4.1\times10^{-3}\text{ m} = \mathbf{4.1\,mm}$$

Discussion Note that the ball diameter is inversely proportional to the square root of density, and thus for a given material, the smaller balls are more likely to float.

2-110

Solution Nutrients dissolved in water are carried to upper parts of plants. The height to which the water solution rises in a tree as a result of the capillary effect is to be determined.



Assumptions **1** The solution can be treated as water with a contact angle of 15°. **2** The diameter of the tube is constant. **3** The temperature of the water solution is 20°C.

Properties The surface tension of water at 20°C is σ_s = 0.073 N/m. The density of water solution can be taken to be 1000 kg/m^3. The contact angle is given to be 15°.

Analysis Substituting the numerical values, the capillary rise is determined to be

$$h = \frac{2\sigma_s\cos\phi}{\rho g R} = \frac{2(0.073\text{ N/m})(\cos 15°)}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(1.3\times10^{-6}\text{ m})}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{11.1\,m}$$

Discussion Other effects such as the chemical potential difference also cause the fluid to rise in trees.

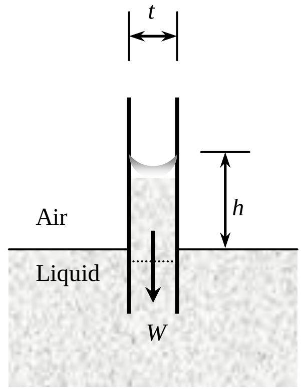
Review Problems

2-111

Solution A relation is to be derived for the capillary rise of a liquid between two large parallel plates a distance t apart inserted into a liquid vertically. The contact angle is given to be ϕ.

Assumptions There are no impurities in the liquid, and no contamination on the surfaces of the plates.

Analysis The magnitude of the capillary rise between two large parallel plates can be determined from a force balance on the rectangular liquid column of height h and width w between the plates. The bottom of the liquid column is at the same level as the free surface of the liquid reservoir, and thus the pressure there must be atmospheric pressure. This will balance the atmospheric pressure acting from the top surface, and thus these two effects will cancel each other. The weight of the liquid column is



$$W = mg = \rho g V = \rho g(w \times t \times h)$$

Equating the vertical component of the surface tension force to the weight gives

$$W = F_{surface} \quad \rightarrow \quad \rho g(w \times t \times h) = 2w\sigma_s \cos\phi$$

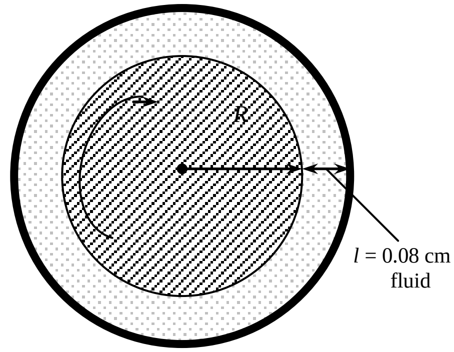
Canceling w and solving for h gives the capillary rise to be

Capillary rise: $$h = \frac{2\sigma_s \cos\phi}{\rho g t}$$

Discussion The relation above is also valid for non-wetting liquids (such as mercury in glass), and gives a capillary drop instead of a capillary rise.

2-112

Solution A journal bearing is lubricated with oil whose viscosity is known. The torques needed to overcome the bearing friction during start-up and steady operation are to be determined.



Assumptions **1** The gap is uniform, and is completely filled with oil. **2** The end effects on the sides of the bearing are negligible. **3** The fluid is Newtonian.

Properties The viscosity of oil is given to be 0.1 kg/m·s at 20°C, and 0.008 kg/m·s at 80°C.

Analysis The radius of the shaft is R = 0.04 m. Substituting the given values, the torque is determined to be

At start up at 20°C:

$$\mathbf{T} = \mu \frac{4\pi^2 R^3 \dot{n} L}{\ell} = (0.1\,\text{kg/m}\cdot\text{s}) \frac{4\pi^2 (0.04\,\text{m})^3 (1500/60\,\text{s}^{-1})(0.55\,\text{m})}{0.0008\,\text{m}} = \mathbf{4.34\,N\cdot m}$$

During steady operation at 80°C:

$$\mathbf{T} = \mu \frac{4\pi^2 R^3 \dot{n} L}{\ell} = (0.008\,\text{kg/m}\cdot\text{s}) \frac{4\pi^2 (0.04\,\text{m})^3 (1500/60\,\text{s}^{-1})(0.55\,\text{m})}{0.0008\,\text{m}} = \mathbf{0.347\,N\cdot m}$$

Discussion Note that the torque needed to overcome friction reduces considerably due to the decrease in the viscosity of oil at higher temperature.

2-113

Solution A U-tube with a large diameter arm contains water. The difference between the water levels of the two arms is to be determined.

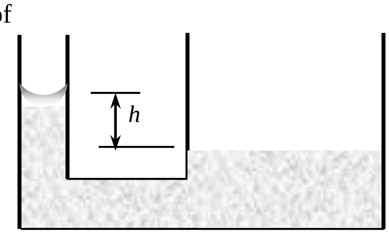
Assumptions **1** Both arms of the U-tube are open to the atmosphere. **2** Water is at room temperature. **3** The contact angle of water is zero, $\phi = 0$.

Properties The surface tension and density of water at 20°C are σ_s = 0.073 N/m and ρ = 1000 kg/m^3.

Analysis Any difference in water levels between the two arms is due to surface tension effects and thus capillary rise. Noting that capillary rise in a tube is inversely proportional to tube diameter there will be no capillary rise in the arm with a large diameter. Then the water level difference between the two arms is simply the capillary rise in the smaller diameter arm,

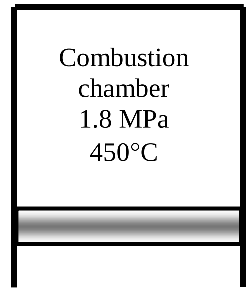
$$h = \frac{2\sigma_s \cos\phi}{\rho g R} = \frac{2(0.073\text{ N/m})(\cos 0°)}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.0025\text{ m})}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)\left(\frac{1000\text{ mm}}{1\text{ m}}\right) = \mathbf{5.95\,mm}$$

Discussion Note that this is a significant difference, and shows the importance of using a U-tube made of a uniform diameter tube.



2-114

Solution The cylinder conditions before the heat addition process is specified. The pressure after the heat addition process is to be determined.



Assumptions **1** The contents of cylinder are approximated by the air properties. **2** Air is an ideal gas.

Analysis The final pressure may be determined from the ideal gas relation

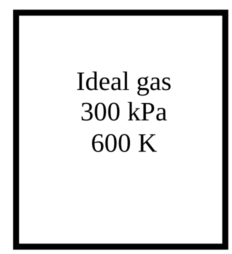
$$P_2 = \frac{T_2}{T_1} P_1 = \left(\frac{1300 + 273.15\,\text{K}}{450 + 273.15\,\text{K}} \right)(1800\,\text{kPa}) = \mathbf{3916\,kPa}$$

Discussion Note that some forms of the ideal gas equation are more convenient to use than the other forms.

2-115

Solution A rigid tank contains an ideal gas at a specified state. The final temperature when half the mass is withdrawn and final pressure when no mass is withdrawn are to be determined.

Analysis (*a*) The first case is a constant volume process. When half of the gas is withdrawn from the tank, the final temperature may be determined from the ideal gas relation as



$$T_2 = \frac{m_1}{m_2} \frac{P_2}{P_1} T_1 = (2)\left(\frac{100\,\text{kPa}}{300\,\text{kPa}} \right)(600\,\text{K}) = \mathbf{400\,K}$$

(*b*) The second case is a constant volume and constant mass process. The ideal gas relation for this case yields

$$P_2 = \frac{T_2}{T_1} P_1 = \left(\frac{400\,\text{K}}{600\,\text{K}} \right)(300\,\text{kPa}) = \mathbf{200\,kPa}$$

Discussion Note that some forms of the ideal gas equation are more convenient to use than the other forms.

2-116

Solution The pressure in an automobile tire increases during a trip while its volume remains constant. The percent increase in the absolute temperature of the air in the tire is to be determined.

Assumptions **1** The volume of the tire remains constant. **2** Air is an ideal gas.

Analysis Noting that air is an ideal gas and the volume is constant, the ratio of absolute temperatures after and before the trip are

$$\frac{P_1 \mathbf{V}_1}{T_1} = \frac{P_2 \mathbf{V}_2}{T_2} \rightarrow \frac{T_2}{T_1} = \frac{P_2}{P_1} = \frac{335\,\text{kPa}}{320\,\text{kPa}} = 1.047$$

Therefore, the absolute temperature of air in the tire will increase by **4.7%** during this trip.

Discussion This may not seem like a large temperature increase, but if the tire is originally at 20°C (293.15 K), the temperature increases to 1.047(293.15 K) = 306.92 K or about 33.8°C.

2-117E

Solution The minimum pressure on the suction side of a water pump is given. The maximum water temperature to avoid the danger of cavitation is to be determined.

Properties The saturation temperature of water at 0.95 psia is 100°F.

Analysis To avoid cavitation at a specified pressure, the fluid temperature everywhere in the flow should remain below the saturation temperature at the given pressure, which is

$$T_{\max} = T_{\text{sat @ 0.95 psia}} = \mathbf{100°F}$$

Therefore, ***T* must remain below 100°F to avoid the possibility of cavitation**.

Discussion Note that saturation temperature increases with pressure, and thus cavitation may occur at higher pressure at locations with higher fluid temperatures.

2-118

Solution Suspended solid particles in water are considered. A relation is to be developed for the specific gravity of the suspension in terms of the mass fraction $C_{s,\text{ mass}}$ and volume fraction $C_{s,\text{ vol}}$ of the particles.

Assumptions **1** The solid particles are distributed uniformly in water so that the solution is homogeneous. **2** The effect of dissimilar molecules on each other is negligible.

Analysis Consider solid particles of mass m_s and volume V_s dissolved in a fluid of mass m_f and volume V_m. The total volume of the suspension (or mixture) is

$$V_m = V_s + V_f$$

Dividing by V_m and using the definition $C_{\text{s, vol}} = V_s / V_m$ give

$$1 = C_{s,vol} + \frac{V_f}{V_m} \quad \rightarrow \quad \frac{V_f}{V_m} = 1 - C_{s,vol} \qquad (1)$$

The total mass of the suspension (or mixture) is

$$m_m = m_s + m_f$$

Dividing by m_m and using the definition $C_{\text{s, mass}} = m_s / m_m$ give

$$1 = C_{s,mass} + \frac{m_f}{m_m} = C_{s,mass} + \frac{\rho_f V_f}{\rho_m V_m} \quad \rightarrow \quad \frac{\rho_f}{\rho_m} = (1 - C_{s,mass})\frac{V_m}{V_f} \qquad (2)$$

Combining equations 1 and 2 gives

$$\frac{\rho_f}{\rho_m} = \frac{1 - C_{s,mass}}{1 - C_{s,vol}}$$

When the fluid is water, the ratio ρ_f / ρ_m is the inverse of the definition of specific gravity. Therefore, the desired relation for the specific gravity of the mixture is

$$\boxed{\text{SG}_m = \frac{\rho_m}{\rho_f} = \frac{1 - C_{s,vol}}{1 - C_{s,mass}}}$$

which is the desired result.

Discussion As a quick check, if there were no particles at all, $\text{SG}_m = 0$, as expected.

2-119

Solution The specific gravities of solid particles and carrier fluids of a slurry are given. The relation for the specific gravity of the slurry is to be obtained in terms of the mass fraction $C_{s,\text{ mass}}$ and the specific gravity SG_s of solid particles.

Assumptions **1** The solid particles are distributed uniformly in water so that the solution is homogeneous. **2** The effect of dissimilar molecules on each other is negligible.

Analysis Consider solid particles of mass m_s and volume V_s dissolved in a fluid of mass m_f and volume V_m. The total volume of the suspension (or mixture) is $V_m = V_s + V_f$.

Dividing by V_m gives

$$1 = \frac{V_s}{V_m} + \frac{V_f}{V_m} \quad \rightarrow \quad \frac{V_f}{V_m} = 1 - \frac{V_s}{V_m} = 1 - \frac{m_s/\rho_s}{m_m/\rho_m} = 1 - \frac{m_s}{m_m}\frac{\rho_m}{\rho_s} = 1 - C_{s,\text{mass}}\frac{\text{SG}_m}{\text{SG}_s} \qquad (1)$$

since ratio of densities is equal two the ratio of specific gravities, and $m_s / m_m = C_{\text{s, mass}}$. The total mass of the suspension (or mixture) is $m_m = m_s + m_f$. Dividing by m_m and using the definition $C_{\text{s, mass}} = m_s / m_m$ give

$$1 = C_{s,\text{mass}} + \frac{m_f}{m_m} = C_{s,\text{mass}} + \frac{\rho_f V_f}{\rho_m V_m} \qquad \rightarrow \qquad \frac{\rho_m}{\rho_f} = \frac{V_f}{(1 - C_{s,mass})V_m} \qquad (2)$$

Taking the fluid to be water so that $\rho_m / \rho_f = \text{SG}_m$ and combining equations 1 and 2 give

$$\text{SG}_m = \frac{1 - C_{s,\text{mass}}\text{SG}_m / \text{SG}_s}{1 - C_{s,\text{mass}}}$$

Solving for SG_m and rearranging gives

$$\boxed{\text{SG}_m = \frac{1}{1 + C_{\text{s, mass}}(1/\text{SG}_s - 1)}}$$

which is the desired result.

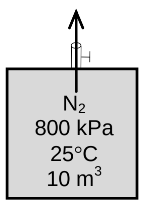
Discussion As a quick check, if there were no particles at all, $\text{SG}_m = 0$, as expected.

2-120

Solution A large tank contains nitrogen at a specified temperature and pressure. Now some nitrogen is allowed to escape, and the temperature and pressure of nitrogen drop to new values. The amount of nitrogen that has escaped is to be determined.

Assumptions The tank is insulated so that no heat is transferred.

Analysis Treating N_2 as an ideal gas, the initial and the final masses in the tank are determined to be



$$m_1 = \frac{P_1 V}{RT_1} = \frac{(800\text{ kPa})(10\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 90.45\text{ kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(600\text{ kPa})(10\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 69.00\text{ kg}$$

Thus the amount of N_2 that escaped is $\Delta m = m_1 - m_2 = 90.45 - 69.00 = \mathbf{21.5\text{ kg}}$

Discussion Gas expansion generally causes the temperature to drop. This principle is used in some types of refrigeration.

2-121

Solution Air in a partially filled closed water tank is evacuated. The absolute pressure in the evacuated space is to be determined.

Properties The saturation pressure of water at 60°C is 19.94 kPa.

Analysis When air is completely evacuated, the vacated space is filled with water vapor, and the tank contains a saturated water-vapor mixture at the given pressure. Since we have a two-phase mixture of a pure substance at a specified temperature, the vapor pressure must be the saturation pressure at this temperature. That is,

$$P_v = P_{\text{sat@60°C}} = 19.94\ \text{kPa} \cong \mathbf{19.9\ kPa}$$

Discussion If there is any air left in the container, the vapor pressure will be less. In that case the sum of the component pressures of vapor and air would equal 19.94 kPa.

2-122

Solution The variation of the dynamic viscosity of water with absolute temperature is given. Using tabular data, a relation is to be obtained for viscosity as a 4th-order polynomial. The result is to be compared to Andrade's equation in the form of $\mu = D \cdot e^{B/T}$.

Properties The viscosity data are given in tabular form as

T (K)	μ (Pa·s)
273.15	1.787×10^{-3}
278.15	1.519×10^{-3}
283.15	1.307×10^{-3}
293.15	1.002×10^{-3}
303.15	7.975×10^{-4}
313.15	6.529×10^{-4}
333.15	4.665×10^{-4}
353.15	3.547×10^{-4}
373.15	2.828×10^{-4}

Analysis Using EES, (1) Define a trivial function "a=mu+T" in the equation window, (2) select new parametric table from Tables, and type the data in a two-column table, (3) select Plot and plot the data, and (4) select plot and click on "curve fit" to get curve fit window. Then specify polynomial and enter/edit equation. The equations and plot are shown here.

μ = 0.489291758 - 0.00568904387T + 0.0000249152104T^2 - $4.86155745\times10^{-8}T^3$ + $3.56198079\times10^{-11}T^4$
μ = 0.000001475*EXP(1926.5/T) [used initial guess of a0=1.8×10^{-6} and a1=1800 in mu=a0*exp(a1/T)]

At T = 323.15 K, the polynomial and exponential curve fits give

Polynomial: μ(323.15 K) = 0.0005529 Pa·s (1.1% error, relative to 0.0005468 Pa·s)
Exponential: μ(323.15 K) = 0.0005726 Pa·s (4.7% error, relative to 0.0005468 Pa·s)

Discussion This problem can also be solved using an Excel worksheet, with the following results:
Polynomial: **A = 0.4893, B = -0.005689, C = 0.00002492, D = -0.000000048612, and E = 0.00000000003562**
Andrade's equation: $\boxed{\mu = 1.807952E-6 * e^{1864.06/T}}$

2-123

Solution A newly produced pipe is tested using pressurized water. The additional water that needs to be pumped to reach a specified pressure is to be determined.

Assumptions **1** There is no deformation in the pipe.

Properties The coefficient of compressibility is given to be 2.10×10^9 Pa.

Analysis From Eq. 2–13, we have

$$\kappa \cong \frac{\Delta P}{\frac{\Delta\rho}{\rho}} \rightarrow \frac{\Delta\rho}{\rho} = \frac{\Delta P}{\kappa} \text{ or } \frac{\rho_2 - \rho_1}{\rho_1} = \frac{\Delta P}{\kappa}$$

from which we write

$$\rho_2 = \rho_1\left(1 + \frac{\Delta P}{\kappa}\right) = (1000\ \text{kg/m}^3) \times \left(1 + \frac{10 \times 10^6 \text{Pa}}{2.10 \times 10^9 \text{Pa}}\right) = 1004.76\ \text{kg/m}^3$$

Then the amount of additional water is

$$m = V_{cyl}\Delta\rho = \frac{\pi D^2}{4} L\Delta\rho = \frac{\pi(2\ \text{m})^2}{4} \times (15) \times \left(1004.76\frac{\text{kg}}{\text{m}^3} - 1000\ \text{kg/m}^3\right) \cong 224.3\ \text{kg}$$

2-124

Solution The pressure is given at a certain depth of the ocean. An analytical relation between density and pressure is to be obtained and the density at a specified pressure is to be determined. The density is to be compared with that from Eq. 2-13.

Properties The coefficient of compressibility is given to be 2350 MPa. The liquid density at the free surface isgiven to be 1030 kg/m^3.

Analysis (a) From the definition, we have

$$\kappa = \frac{dP}{d\rho/\rho} \rightarrow \frac{d\rho}{\rho} = \frac{dP}{\kappa}$$

Integrating

$$\int_{\rho_0}^{\rho} \frac{d\rho}{\rho} = \int_0^P \frac{dP}{\kappa} \rightarrow \ln\frac{\rho}{\rho_0} = \frac{P}{\kappa} \rightarrow \rho = \rho_0 e^{P/\kappa}$$

With the given data we obtain

$$\rho = (1030\ \text{kg/m}^3) \times e^{100/2350} = 1074\ \ \text{kg/m}^3$$

(b) Eq. 2–13 can be rearranged to give

$$\Delta\rho \cong \rho\frac{\Delta P}{\kappa}$$

or

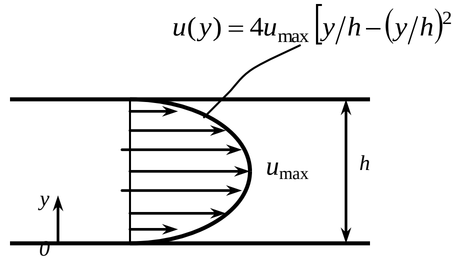
$$\rho - \rho_0 \cong \rho_0\frac{P - P_0}{\kappa} \rightarrow \rho \cong \rho_0 + \rho_0\frac{P - P_0}{\kappa} = 1030\frac{\text{kg}}{\text{m}^3} + 1030\frac{\text{kg}}{\text{m}^3} \times \frac{100\ \text{MPa}}{2350\ \text{MPa}} \approx 1074\ \text{kg/m}^3$$

which is identical with (a). Therefore we conclude that linear approximation (Eq. 2–13) is quite reasonable.

2-125

Solution The velocity profile for laminar one-dimensional flow between two parallel plates is given. A relation for friction drag force exerted on the plates per unit area of the plates is to be obtained.

Assumptions **1** The flow between the plates is one-dimensional. **2** The fluid is Newtonian.



Analysis The velocity profile is given by $u(y) = 4u_{\max}\left[y/h - (y/h)^2\right]$

where h is the distance between the two plates, y is the vertical distance from the bottom plate, and $u_{\max}$ is the maximum flow velocity that occurs at midplane. The shear stress at the bottom surface can be expressed as

$$\tau_w = \mu \left.\frac{du}{dy}\right|_{y=0} = 4\mu u_{\max} \frac{d}{dy}\left(\frac{y}{h} - \frac{y^2}{h^2}\right)_{y=0} = 4\mu u_{\max}\left.\left(\frac{1}{h} - \frac{2y}{h^2}\right)\right|_{y=0} = \frac{4\mu u_{\max}}{h}$$

Because of symmetry, the wall shear stress is identical at both bottom and top plates. Then the friction drag force exerted by the fluid on the inner surface of the plates becomes

$$F_D = 2\tau_w A_{plate} = \frac{8\mu u_{\max}}{h} A_{\text{plate}}$$

Therefore, the friction drag per unit plate area is

$$\boxed{F_D / A_{\text{plate}} = \frac{8\mu u_{\max}}{h}}$$

Discussion Note that the friction drag force acting on the plates is inversely proportional to the distance between plates.

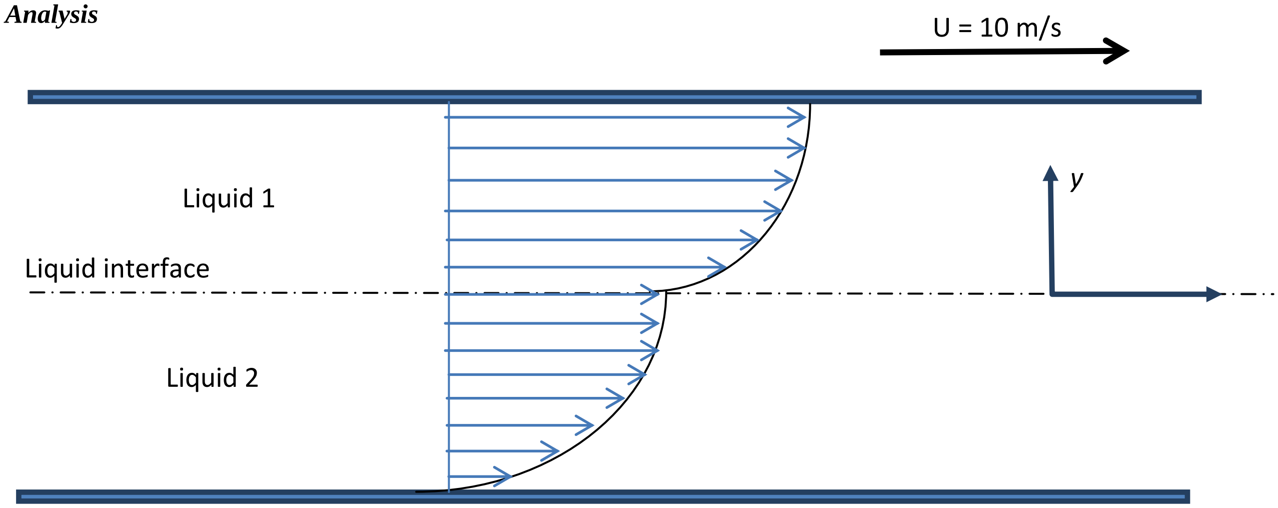
2-126

Solution Two immiscible Newtonian liquids flow steadily between two large parallel plates under the influence of an applied pressure gradient. The lower plate is fixed while the upper one is pulled with a constant velocity. The velocity profiles for each flow are given. The values of constants are to be determined. An expression for the viscosity ratio is to be developed. The forces and their directions exerted by liquids on both plates are to be determined.

Assumptions **1** The flow between the plates is one-dimensional. **2** The fluids are Newtonian.

Properties The viscosity of fluid one is given to be $\mu_1 = 10^{-3}\ \mathrm{Pa \cdot s}$.

Analysis



(a) The velocity profiles should satisfy the conditions $V_1(h) = 10,\ V_2(-h) = 0$ and $V_1(0) = V_2(0)$. It is clear that $V_1(0) = 6\ \mathrm{m/s}$.

$$V_1(h) = 10:\ \ 10 = 6 + a \times 0.5 - 3 \times (0.5)^2 \rightarrow a = 9.5$$

$$V_2(0) = 6 = b + c \times 0 - 9(0)^2 \rightarrow b = 6$$

Finally,

$$V_2(-h) = 0 \rightarrow 0 = 6 + c \times (-0.5) - 9(-0.5)^2 \rightarrow c = 7.5$$

Therefore we have the velocity profiles as follows:

$$V_1 = 6 + 9.5y - 3y^2, \qquad -0.5 \le y \le 0$$
$$V_2 = 6 + 7.5y - 9y^2, \qquad 0 \le y \le -0.5$$

(b) The shear stress at the interface is unique, and then we have

$$\mu_1 \left.\frac{dV_1}{dy}\right]_{y=0} = \mu_2 \left.\frac{dV_2}{dy}\right]_{y=0} \rightarrow \frac{\mu_1}{\mu_2} = \frac{\left.\frac{dV_2}{dy}\right]_{y=0}}{\left.\frac{dV_1}{dy}\right]_{y=0}} = \left.\frac{7.5 - 18y}{9.5 - 6y}\right]_{y=0} \cong 0.79$$

(c)

Lower plate:

$$F_L = \mu_2 \left.\frac{dV_2}{dy}\right]_{y=-h} A = \left(\frac{10^{-3}\mathrm{N \cdot s/m^2}}{0.79}\right) \times \underbrace{[7.5 - 18y]_{y=-0.5}}_{16.5} \times (4\ \mathrm{m^2}) = 0.0835\ \text{N to the right}$$

Upper plate:

$$F_U = \mu_1 \left.\frac{dV_1}{dy}\right]_{y=h} A = (10^{-3}\mathrm{N \cdot s/m^2}) \times \underbrace{[9.5 - 6y]_{y=-0.5}}_{12.5} \times (4\ \mathrm{m^2}) = 0.0500\ \text{N to the right}$$

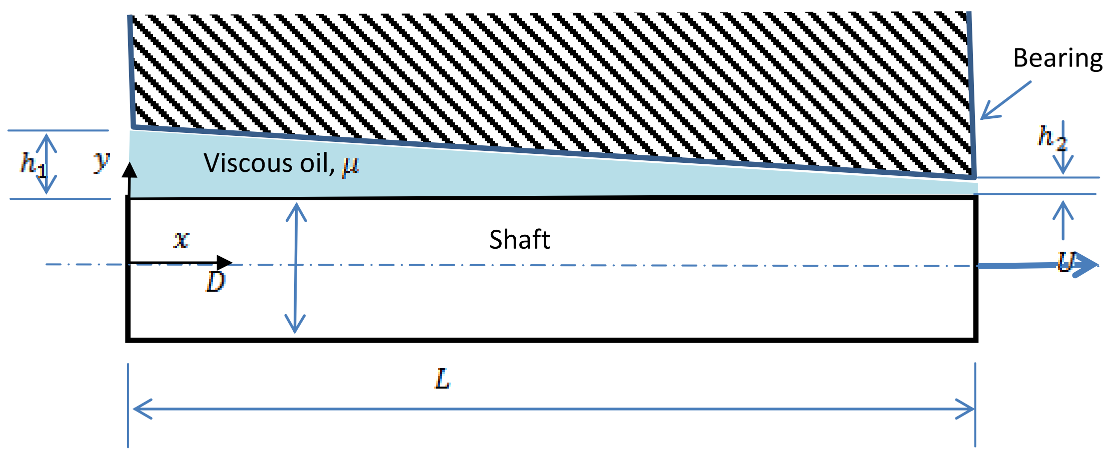
2-127

Solution A shaft is pulled with a constant velocity through a bearing. The space between the shaft and bearing is filled with a fluid. The force required to maintain the axial movement of the shaft is to be determined.

Assumptions **1** The fluid is Newtonian.

Properties The viscosity of the fluid is given to be 0.1 Pa·s.

Analysis



The varying clearance h can be expressed as a function of axial coordinate x (see figure). According to this sketch we obtain

$$h = h_1 - (h_1 - h_2)\frac{x}{L}$$

Assuming a linear velocity distribution in the gap, the viscous force acting on the differential strip element is

$$dF = \tau dA = \mu \frac{U}{h} \times \pi D dx = \frac{\mu U \pi D}{h_1 - (h_1 - h_2)\frac{x}{L}} dx$$

Integrating

$$F = \mu U \pi D \int_{x=0}^{x=L} \frac{dx}{h_1 - (h_1 - h_2)\frac{x}{L}} = -\mu U \pi D \frac{\ln\left(h_1 - (h_1 - h_2)\frac{x}{L}\right)}{\frac{(h_1 - h_2)}{L}}\Bigg]_{x=0}^{x=L} = \frac{\mu U \pi D L}{h_1 - h_2} \ln \frac{h_1}{h_2}$$

For the given data, we obtain

$$F = \frac{(0.1\ \text{Pa}\cdot\text{s})(5\ \text{m/s})\pi(80 \times 10^{-3}\text{mm})(400 \times 10^{-3}\text{mm})}{(1.2 - 0.4) \times 10^{-3}\text{mm}} \ln \frac{1.2}{0.4} \cong 69\ \text{N}$$

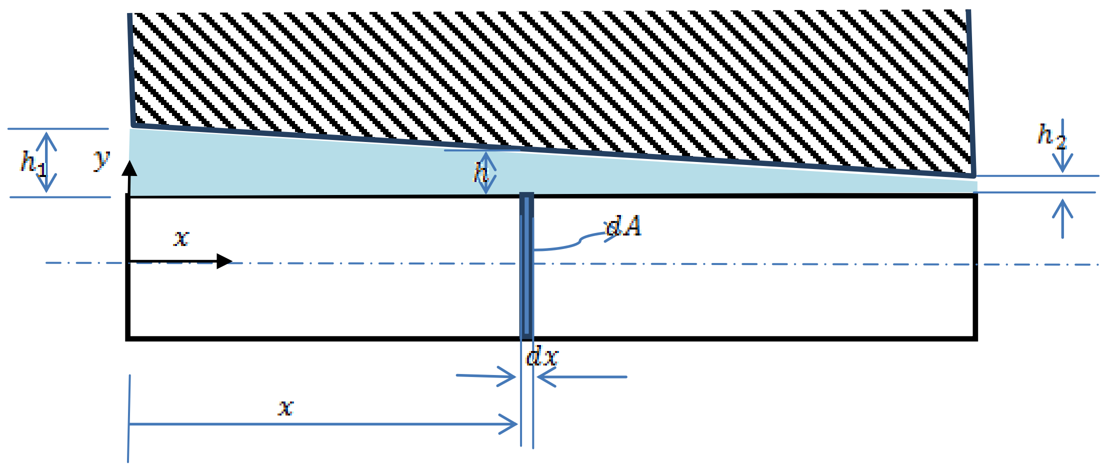
2-128

Solution A shaft rotates with a constant angual speed in a bearing. The space between the shaft and bearing is filled with a fluid. The torque required to maintain the motion is to be determined.

Assumptions **1** The fluid is Newtonian.

Properties The viscosity of the fluid is given to be 0.1 Pa·s.

Analysis The varying clearance h can be expressed as a function of axial coordinate x (see figure below).



According to this sketch we obtain

$$h = h_1 - (h_1 - h_2)\frac{x}{L}$$

Assuming a linear velocity distribution in the gap, the viscous force acting on the differential strip element is

$$dF = \tau dA = \mu\frac{U}{h} \times \pi D dx = \frac{\mu U \pi D}{h_1 - (h_1 - h_2)\frac{x}{L}} dx$$

where $U = 2n\pi/60$ in this case. Then the viscous torque developed on the shaft

$$dT = dF \times \frac{D}{2} = \frac{\mu\left(\frac{2n\pi}{60} \times \frac{D}{2}\right)\pi D \times \frac{D}{2}}{h_1 - (h_1 - h_2)\frac{x}{L}} dx = \frac{\mu n \pi^2 D^3}{120}\frac{dx}{h_1 - (h_1 - h_2)\frac{x}{L}}$$

Integrating

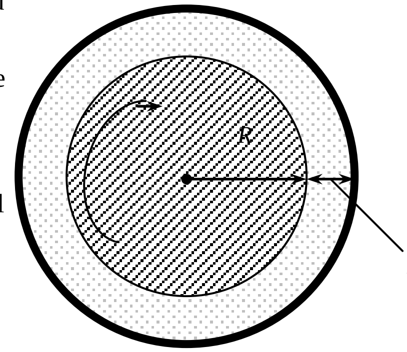
$$T = \frac{\mu n \pi^2 D^3}{120}\int_{x=0}^{x=L}\frac{dx}{h_1 - (h_1 - h_2)\frac{x}{L}} = -\frac{\mu n \pi^2 D^3}{120}\left.\frac{\ln\left(h_1 - (h_1 - h_2)\frac{x}{L}\right)}{\frac{(h_1 - h_2)}{L}}\right]_{x=0}^{x=L} = \frac{1}{120}\frac{\mu n \pi^2 D^3 L}{h_1 - h_2}\ln\frac{h_1}{h_2}$$

For the given data, we obtain

$$T = \frac{1}{120}\frac{(0.1\ \text{Pa}\cdot\text{s})(1450\ \text{rpm})\pi^2(80 \times 10^{-3}\text{mm})^3(400 \times 10^{-3}\text{mm})}{(1.2 - 0.4) \times 10^{-3}\text{mm}}\ln\frac{1.2}{0.4} \cong 3.354\ \text{N}\cdot\text{m}$$

2-129

Solution A cylindrical shaft rotates inside an oil bearing at a specified speed. The power required to overcome friction is to be determined.



Assumptions **1** The gap is uniform, and is completely filled with oil. **2** The end effects on the sides of the bearing are negligible. **3** The fluid is Newtonian.

Properties The viscosity of oil is given to be 0.300 N·s/m^2.

Analysis (*a*) The radius of the shaft is R = 0.05 m, and thickness of the oil layer is ℓ = (10.3 – 10)/2 = 0.15 cm. The power-torque relationship is

$$\dot{W} = \omega\mathbf{T} = 2\pi\dot{n}\mathbf{T} \quad \text{where, from Chap. 2,} \quad \mathbf{T} = \mu\frac{4\pi^2 R^3 \dot{n} L}{\ell}$$

Substituting, the required power to overcome friction is determined to be

$$\dot{W} = \mu\frac{6\pi^3 R^3 \dot{n}^2 L}{\ell} = (0.3\,\mathrm{N\cdot s/m^2})\frac{6\pi^3 (0.05\,\mathrm{m})^3 (600/60\,\mathrm{s^{-1}})^2 (0.40\,\mathrm{m})}{0.0015\,\mathrm{m}}\left(\frac{1\,\mathrm{W}}{1\,\mathrm{N\cdot m/s}}\right) = \mathbf{186\,W}$$

(*b*) For the case of $\dot{n} = 1200$ rpm :

$$\dot{W} = \mu\frac{6\pi^3 R^3 \dot{n}^2 L}{\ell} = (0.3\,\mathrm{N\cdot s/m^2})\frac{6\pi^3 (0.05\,\mathrm{m})^3 (1200/60\,\mathrm{s^{-1}})^2 (0.40\,\mathrm{m})}{0.0015\,\mathrm{m}}\left(\frac{1\,\mathrm{W}}{1\,\mathrm{N\cdot m/s}}\right) = \mathbf{744\,W}$$

Discussion Note the power dissipated in journal bearing is proportional to the cube of the shaft radius and to the square of the shaft speed, and is inversely proportional to the oil layer thickness.

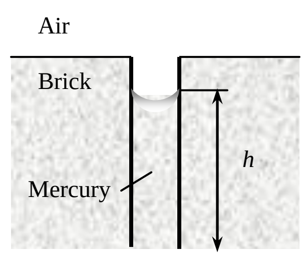
2-130

Solution Air spaces in certain bricks form air columns of a specified diameter. The height that water can rise in those tubes is to be determined.

Assumptions **1** The interconnected air pockets form a cylindrical air column. **2** The air columns are open to the atmospheric air. **3** The contact angle of water is zero, $\phi = 0$.

Properties The surface tension is given to be 0.085 N/m, and we take the water density to be 1000 kg/m^3.

Analysis Substituting the numerical values, the capillary rise is determined to be



$$h = \frac{2\sigma_s \cos\phi}{\rho g R} = \frac{2(0.085\,\mathrm{N/m})(\cos 0°)}{(1000\,\mathrm{kg/m^3})(9.81\,\mathrm{m/s^2})(3\times10^{-6}\,\mathrm{m})}\left(\frac{1\,\mathrm{kg\cdot m/s^2}}{1\,\mathrm{N}}\right) = \mathbf{5.78\,m}$$

Discussion The surface tension depends on temperature. Therefore, the value determined may change with temperature.

Fundamentals of Engineering (FE) Exam Problems

2-131

The specific gravity of a fluid is specified to be 0.82. The specific volume of this fluid is

(*a*) 0.001 m^3/kg (*b*) 0.00122 m^3/kg (*c*) 0.0082 m^3/kg (*d*) 82 m^3/kg
(*e*) 820 m^3/kg

Answer (*b*) 0.00122 m^3/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
SG=0.82
rho_water=1000 [kg/m^3]
rho_fluid=SG*rho_water
v=1/rho_fluid
```

2-132

The specific gravity of mercury is 13.6. The specific weight of mercury is

(*a*) 1.36 kN/m^3 (*b*) 9.81 kN/m^3 (*c*) 106 kN/m^3 (*d*) 133 kN/m^3 (*e*) 13,600 kN/m^3

Answer (*d*) 133 kN/m^3

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
SG=13.6
rho_water=1000 [kg/m^3]
rho=SG*rho_water
g=9.81 [m/s^2]
SW=rho*g
```

2-133

An ideal gas flows in a pipe at 20°C. The density of the gas is 1.9 kg/m^3 and its molar mass is 44 kg/kmol. The pressure of the gas is

(*a*) 7 kPa (*b*) 72 kPa (*c*) 105 kPa (*d*) 460 kPa (*e*) 4630 kPa

Answer (*c*) 105 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=(20+273) [K]
rho=1.9 [kg/m^3]
MM=44 [kg/kmol]
R_u=8.314 [kJ/kmol-K]
R=R_u/MM
P=rho*R*T
```

2-134

A gas mixture consists of 3 kmol oxygen, 2 kmol nitrogen, and 0.5 kmol water vapor. The total pressure of the gas mixture is 100 kPa. The partial pressure of water vapor in this gas mixture is

(*a*) 5 kPa (*b*) 9.1 kPa (*c*) 10 kPa (*d*) 22.7 kPa (*e*) 100 kPa

Answer (*b*) 9.1 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
N_O2=3 [kmol]
N_N2=2 [kmol]
N_vapor=0.5 [kmol]
P_total=100 [kPa]
N_total=N_O2+N_N2+N_vapor
y_vapor=N_vapor/N_total
P_partial=y_vapor*P_total
```

2-135

Liquid water vaporizes into water vapor as it flows in the piping of a boiler. If the temperature of water in the pipe is 180°C, the vapor pressure of water in the pipe is

(*a*) 1002 kPa (*b*) 180 kPa (*c*) 101.3 kPa (*d*) 18 kPa (*e*) 100 kPa

Answer (*a*) 1002 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=180 [C]
P_vapor=pressure(steam, T=T, x=1)
```

2-136

In a water distribution system, the pressure of water can be as low as 1.4 psia. The maximum temperature of water allowed in the piping to avoid cavitation is

(*a*) 50°F (*b*) 77°F (*c*) 100°F (*d*) 113°F (*e*) 140°F

Answer (*d*) 113°F

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

P=1.4 [psia]
T_max=temperature(steam, P=P, x=1)

2-137

The thermal energy of a system refers to

(*a*) Sensible energy (*b*) Latent energy (*c*) Sensible + latent energies

(*d*) Enthalpy (*e*) Internal energy

Answer (*c*) Sensible + latent energies

2-138

The difference between the energies of a flowing and stationary fluid per unit mass of the fluid is equal to

(*a*) Enthalpy (*b*) Flow energy (*c*) Sensible energy (*d*) Kinetic energy

(*e*) Internal energy

Answer (*b*) Flow energy

2-139

The pressure of water is increased from 100 kPa to 1200 kPa by a pump. The temperature of water also increases by 0.15°C. The density of water is 1 kg/L and its specific heat is c_p = 4.18 kJ/kg·°C. The enthalpy change of the water during this process is

(*a*) 1100 kJ/kg (*b*) 0.63 kJ/kg (*c*) 1.1 kJ/kg (*d*) 1.73 kJ/kg (*e*) 4.2 kJ/kg

Answer (*d*) 1.73 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 [kPa]
P2=1200 [kPa]
DELTAT=0.15 [C]
rho=1000 [kg/m^3]
c_p=4.18 [kJ/kg-C]
DELTAh=c_p*DELTAT+(P2-P1)/rho
```

2-140

The coefficient of compressibility of a truly incompressible substance is

(*a*) 0 (*b*) 0.5 (*c*) 1 (*d*) 100 (*e*) Infinity

Answer (*e*) Infinity

2-141

The pressure of water at atmospheric pressure must be raised to 210 atm to compress it by 1 percent. Then, the coefficient of compressibility value of water is

(*a*) 209 atm (*b*) 20,900 atm (*c*) 21 atm (*d*) 0.21 atm (*e*) 210,000 atm

Answer (*b*) 20,900 atm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1 [atm]
P2=210 [atm]
DELTArho\rho=0.01
DELTAP=P2-P1
CoeffComp=DELTAP/DELTArho\rho
```

2-142

When a liquid in a piping network encounters an abrupt flow restriction (such as a closing valve), it is locally compressed. The resulting acoustic waves that are produced strike the pipe surfaces, bends, and valves as they propagate and reflect along the pipe, causing the pipe to vibrate and produce the familiar sound. This is known as

(*a*) Condensation (*b*) Cavitation (*c*) Water hammer (*d*) Compression (*e*) Water arrest

Answer (*c*) Water hammer

2-143

The density of a fluid decreases by 5 percent at constant pressure when its temperature increases by 10°C. The coefficient of volume expansion of this fluid is

(*a*) 0.01 K^{-1} (*b*) 0.005 K^{-1} (*c*) 0.1 K^{-1} (*d*) 0.5 K^{-1} (*e*) 5 K^{-1}

Answer (*b*) 0.005 K^{-1}

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
DELTArho\rho=-0.05
DELTAT=10 [K]
beta=-DELTArho\rho/DELTAT
```

2-144

Water is compressed from 100 kPa to 5000 kPa at constant temperature. The initial density of water is 1000 kg/m^3 and the isothermal compressibility of water is $\alpha = 4.8\times10^{-5}$ atm^{-1}. The final density of the water is

(*a*) 1000 kg/m^3 (*b*) 1001.1 kg/m^3 (*c*) 1002.3 kg/m^3 (*d*) 1003.5 kg/m^3 (*e*) 997.4 kg/m^3

Answer (*c*) 1002.3 kg/m^3

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 [kPa]
P2=5000 [kPa]
rho_1=1000 [kg/m^3]
alpha=4.8E-5 [1/atm]
DELTAP=(P2-P1)*Convert(kPa, atm)
DELTArho=alpha*rho_1*DELTAP
DELTArho=rho_2-rho_1
```

2-145

The speed of a spacecraft is given to be 1250 km/h in atmospheric air at –40°C. The Mach number of this flow is

(*a*) 35.9 (*b*) 0.85 (*c*) 1.0 (*d*) 1.13 (*e*) 2.74

Answer (*d*) 1.13

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel=1250 [km/h]*Convert(km/h, m/s)
T=(-40+273.15) [K]
R=0.287 [kJ/kg-K]
k=1.4
c=sqrt(k*R*T*Convert(kJ/kg, m^2/s^2))
Ma=Vel/c
```

2-146

The dynamic viscosity of air at 20°C and 200 kPa is 1.83×10^{-5} kg/m·s. The kinematic viscosity of air at this state is

(*a*) 0.525×10^{-5} m^2/s (*b*) 0.77×10^{-5} m^2/s (*c*) 1.47×10^{-5} m^2/s (*d*) 1.83×10^{-5} m^2/s

(*e*) 0.380×10^{-5} m^2/s

Answer (*b*) 0.77×10^{-5} m^2/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=(20+273.15) [K]
P=200 [kPa]
mu=1.83E-5 [kg/m-s]
R=0.287 [kJ/kg-K]
rho=P/(R*T)
nu=mu/rho
```

2-147

A viscometer constructed of two 30-cm-long concentric cylinders is used to measure the viscosity of a fluid. The outer diameter of the inner cylinder is 9 cm, and the gap between the two cylinders is 0.18 cm.
The inner cylinder is rotated at 250 rpm, and the torque is measured to be 1.4 N· m. The viscosity of the fluid is

(*a*) 0.0084 N·s/m^2 (*b*) 0.017 N·s/m^2 (*c*) 0.062 N·s/m^2 (*d*) 0.0049 N·s/m^2

(*e*) 0.56 N·s/m^2

Answer (*e*) 0.56 N·s/m^2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
L=0.3 [m]
R=0.045 [m]
gap=0.0018 [m]
n_dot=(250/60) [1/s]
T=1.4 [N-m]
mu=(T*gap)/(4*pi^2*R^3*n_dot*L)
```

2-148

Which one is not a surface tension or surface energy (per unit area) unit?

(*a*) lbf/ft (*b*) N·m/m^2 (*c*) lbf/ft^2 (*d*) J/m^2 (*e*) Btu/ft^2

Answer (*c*) lbf/ft^2

2-149

The surface tension of soap water at 20°C is σ_s = 0.025 N/m. The gage pressure inside a soap bubble of diameter 2 cm at 20°C is

(*a*) 10 Pa (*b*) 5 Pa (*c*) 20 Pa (*d*) 40 Pa (*e*) 0.5 Pa

Answer (*a*) 10 Pa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
sigma_s=0.025 [N/m]
D=0.02 [m]
R=D/2
DELTAP=4*sigma_s/R
P_i_gage=DELTAP
```

2-150

A 0.4-mm-diameter glass tube is inserted into water at 20°C in a cup. The surface tension of water at 20°C is σ_s = 0.073 N/m. The contact angle can be taken as zero degrees. The capillary rise of water in the tube is

(*a*) 2.9 cm (*b*) 7.4 cm (*c*) 5.1 cm (*d*) 9.3 cm (*e*) 14.0 cm

Answer (*b*) 7.4 cm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.0004 [m]
R=D/2
sigma_s=0.073 [N/m]
phi=0 [degrees]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
h=(2*sigma_s*cos(phi))/(rho*g*R)
```

Design and Essay Problems

2-151, 2-152, 2-153

Solution Students' essays and designs should be unique and will differ from each other.

2-154

Solution We are to determine the inlet water speed at which cavitation is likely to occur in the throat of a converging-diverging tube or duct, and repeat for a higher temperature.

Assumptions **1** The fluid is incompressible and Newtonian. **2** Gravitational effects are negligible. **3** Irreversibilities are negligible. **4** The equations provided are valid for this flow.

Properties For water at 20°C, ρ = 998.0 kg/m^3 and P_{sat} = 2.339 kPa.

Analysis (*a*) Two equations are given for velocity, pressure, and cross-sectional area, namely,

$$V_1A_1 = V_2A_2 \quad \text{and} \quad P_1 + \rho\frac{V_1^2}{2} = P_2 + \rho\frac{V_2^2}{2}$$

Solving the first equation for V_2 gives

$$V_2 = V_1\frac{A_1}{A_2} \tag{1}$$

Substituting the above into the equation for pressure and solving for V_1 yields, after some algebra,

$$\boxed{V_1 = \sqrt{\frac{2(P_1 - P_2)}{\rho\left(\left(\frac{A_1}{A_2}\right)^2 - 1\right)}}}$$

But the pressure at which cavitation is likely to occur is the vapor (saturation) pressure of the water. We also know that throat diameter D_2 is 1/20 times the inlet diameter D_1, and since $A = \pi D^2/4$, $A_1/A_2 = (20)^2 = 400$. Thus,

$$V_1 = \sqrt{\frac{2(20.803 - 2.339)\text{ kPa}}{998.0\frac{\text{kg}}{\text{m}^3}(400^2 - 1)}\left(\frac{1000\text{ N/m}^2}{\text{kPa}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{\text{N}}\right)} = 0.015207\frac{\text{m}}{\text{s}}$$

So, the minimum inlet velocity at which cavitation is likely to occur is **0.0152 m/s** (to three significant digits). The velocity at the throat is much faster than this, of course. Using Eq. (1),

$$V_t = V_1\frac{A_1}{A_t} = V_1\frac{\pi D_1^2}{\pi D_t^2} = V_1\left(\frac{D_1}{D_t}\right)^2 = 0.015207\left(\frac{20}{1}\right)^2 = 6.0828\text{ m/s}$$

(*b*) If the water is warmer (50°C), the density reduces to 988.1 kg/m^3, and the vapor pressure increases to 12.35 kPa. At these conditions, $\boldsymbol{V_1}$ **= 0.0103 m/s**. As might be expected, **at higher temperature, a lower inlet velocity is required to generate cavitation, since the water is warmer and already closer to its boiling point**.

Discussion Cavitation is usually undesirable since it leads to noise, and the collapse of the bubbles can be destructive. It is therefore often wise to design piping systems and turbomachinery to avoid cavitation.

2-155

Solution We are to explain how objects like razor blades and paper clips can float on water, even though they are much denser than water.

Analysis Just as some insects like water striders can be supported on water by surface tension, **surface tension is the key to explaining this phenomenon**. If we think of surface tension like a skin on top of the water, somewhat like a stretched piece of balloon, we can understand how something heavier than water pushes down on the surface, but the **surface tension forces counteract the weight (to within limits) by providing an upward force**. Since soap decreases surface tension, we expect that **it would be harder to float objects like this on a soapy surface**; with a high enough soap concentration, in fact, we would expect that the razor blade or paper clip could not float at all.

Discussion If the razor blade or paper clip is fully submerged (breaking through the surface tension), it sinks.

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

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CHAPTER 3

PRESSURE AND FLUID STATICS

Pressure, Manometer, and Barometer

3-1C

Solution We are to examine a claim about absolute pressure.

Analysis **No, the absolute pressure in a liquid of constant density does not double when the depth is doubled**. It is the *gage pressure* that doubles when the depth is doubled.

Discussion This is analogous to temperature scales – when performing analysis using something like the ideal gas law, you *must* use absolute temperature (K), not relative temperature (^{o}C), or you will run into the same kind of problem.

3-2C

Solution We are to compare the pressure on the surfaces of a cube.

Analysis Since pressure increases with depth, **the pressure on the bottom face of the cube is higher than that on the top. The pressure varies linearly along the side faces**. However, if the lengths of the sides of the tiny cube suspended in water by a string are very small, the magnitudes of the pressures on all sides of the cube are nearly the same.

Discussion In the limit of an "infinitesimal cube", we have a fluid particle, with pressure P defined at a "point".

3-3C

Solution We are to define Pascal's law and give an example.

Analysis *Pascal's law* states that **the pressure applied to a confined fluid increases the pressure throughout by the same amount**. This is a consequence of the pressure in a fluid remaining constant in the horizontal direction. **An example of Pascal's principle is the operation of the hydraulic car jack**.

Discussion Students may have various answers to the last part of the question. The above discussion applies to fluids at rest (hydrostatics). When fluids are in motion, Pascal's principle does not necessarily apply. However, as we shall see in later chapters, the differential equations of incompressible fluid flow contain only pressure *gradients*, and thus an increase in pressure *in the whole system* does *not* affect fluid motion.

3-4C

Solution We are to compare the volume and mass flow rates of two fans at different elevations.

Analysis The density of air at sea level is higher than the density of air on top of a high mountain. Therefore, **the volume flow rates of the two fans running at identical speeds will be the same**, but **the mass flow rate of the fan at sea level will be higher**.

Discussion In reality, the fan blades on the high mountain would experience less frictional drag, and hence the fan motor would not have as much resistance – the rotational speed of the fan on the mountain may be slightly higher than that at sea level.

3-5C

Solution We are to discuss the difference between gage pressure and absolute pressure.

Analysis The **pressure relative to the atmospheric pressure** is called the *gage pressure*, and the **pressure relative to an absolute vacuum** is called *absolute pressure*.

Discussion Most pressure gages (like your bicycle tire gage) read relative to atmospheric pressure, and therefore read the gage pressure.

3-6C

Solution We are to explain nose bleeding and shortness of breath at high elevation.

Analysis Atmospheric air pressure which is the external pressure exerted on the skin decreases with increasing elevation. Therefore, **the pressure is lower at higher elevations. As a result, the difference between the blood pressure in the veins and the air pressure outside increases**. **This pressure imbalance may cause some thin-walled veins such as the ones in the nose to burst, causing bleeding**. The shortness of breath is caused by the lower air density at higher elevations, and thus lower amount of oxygen per unit volume.

Discussion People who climb high mountains like Mt. Everest suffer other physical problems due to the low pressure.

3-7

Solution A gas is contained in a vertical cylinder with a heavy piston. The pressure inside the cylinder and the effect of volume change on pressure are to be determined.

Assumptions Friction between the piston and the cylinder is negligible.

Analysis (*a*) The gas pressure in the piston–cylinder device depends on the atmospheric pressure and the weight of the piston. Drawing the free-body diagram of the piston as shown in Fig. 3–20 and balancing the vertical forces yield

$$PA = P_{\text{atm}} A + W$$

Solving for P and substituting,

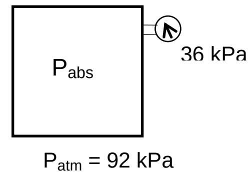
$$P = P_{\text{atm}} + \frac{mg}{A} = 95\,\text{kPa} + \frac{(40\,\text{kg})(9.81\,\text{m/s}^2)}{0.012\,\text{m}^2}\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1\ \text{kN/m}^2}\right) = \mathbf{128\,kPa}$$

(*b*) The volume change will have no effect on the free-body diagram drawn in part (*a*), and therefore **we do *not* expect the pressure inside the cylinder to change – it will remain the same**.

Discussion If the gas behaves as an ideal gas, the absolute temperature doubles when the volume is doubled at constant pressure.

3-8

Solution The pressure in a vacuum chamber is measured by a vacuum gage. The absolute pressure in the chamber is to be determined.



Analysis The absolute pressure in the chamber is determined from

$$P_{abs} = P_{atm} - P_{vac} = 92 - 36 = \mathbf{56\ kPa}$$

Discussion We must remember that "vacuum pressure" is the negative of gage pressure – hence the negative sign.

3-9E

Solution The pressure given in psia unit is to be converted to kPa.

Analysis Using the psia to kPa units conversion factor,

$$P = (150\ \text{psia})\left(\frac{6.895\ \text{kPa}}{1\ \text{psia}}\right) = \mathbf{1034 kPa}$$

3-10E

Solution The pressure in a tank in SI unit is given. The tank's pressure in various English units are to be determined.

Analysis Using appropriate conversion factors, we obtain

(*a*) $$P = (1500\ \text{kPa})\left(\frac{20.886\ \text{lbf/ft}^2}{1\ \text{kPa}}\right) = \mathbf{31{,}330 lbf/ft^2}$$

(*b*) $$P = (1500\ \text{kPa})\left(\frac{20.886\ \text{lbf/ft}^2}{1\ \text{kPa}}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right)\left(\frac{1\ \text{psia}}{1\ \text{lbf/in}^2}\right) = \mathbf{217.6 psia}$$

3-11E

Solution The pressure in a tank is measured with a manometer by measuring the differential height of the manometer fluid. The absolute pressure in the tank is to be determined for two cases: the manometer arm with the (*a*) higher and (*b*) lower fluid level being attached to the tank.

Assumptions The fluid in the manometer is incompressible.

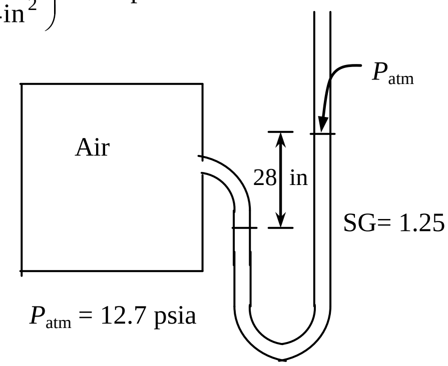
Properties The specific gravity of the fluid is given to be SG = 1.25. The density of water at 32°F is 62.4 lbm/ft^3.

Analysis The density of the fluid is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{H_2O} = (1.25)(62.4 \text{ lbm/ft}^3) = 78.0 \text{ lbm/ft}^3$$

The pressure difference corresponding to a differential height of 28 in between the two arms of the manometer is

$$\Delta P = \rho g h = (78 \text{ lbm/ft}^3)(32.174 \text{ ft/s}^2)(28/12 \text{ ft})\left(\frac{1 \text{ lbf}}{32.174 \text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right) = 1.26 \text{ psia}$$



Then the absolute pressures in the tank for the two cases become:

(*a*) The fluid level in the arm attached to the tank is higher (vacuum):

$$P_{abs} = P_{atm} - P_{vac} = 12.7 - 1.26 = 11.44 \text{ psia} \cong \mathbf{11.4 \text{ psia}}$$

(*b*) The fluid level in the arm attached to the tank is lower:

$$P_{abs} = P_{gage} + P_{atm} = 12.7 + 1.26 = 13.96 \text{ psia} \cong \mathbf{14.0 \text{ psia}}$$

Discussion The final results are reported to three significant digits. Note that we can determine whether the pressure in a tank is above or below atmospheric pressure by simply observing the side of the manometer arm with the higher fluid level.

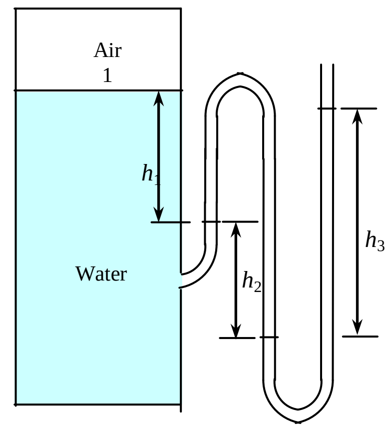
3-12

Solution The pressure in a pressurized water tank is measured by a multi-fluid manometer. The gage pressure of air in the tank is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus we can determine the pressure at the air-water interface.

Properties The densities of mercury, water, and oil are given to be 13,600, 1000, and 850 kg/m^3, respectively.

Analysis Starting with the pressure at point 1 at the air-water interface, and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach point 2, and setting the result equal to P_{atm} since the tube is open to the atmosphere gives



$$P_1 + \rho_{water} gh_1 + \rho_{oil} gh_2 - \rho_{mercury} gh_3 = P_{atm}$$

Solving for $P_{1,}$

$$P_1 = P_{atm} - \rho_{water} gh_1 - \rho_{oil} gh_2 + \rho_{mercury} gh_3$$

or,

$$P_1 - P_{atm} = g(\rho_{mercury} h_3 - \rho_{water} h_1 - \rho_{oil} h_2)$$

Noting that $P_{1,gage} = P_1 - P_{atm}$ and substituting,

$$P_{1,gage} = (9.81\text{ m/s}^2)[(13{,}600\text{ kg/m}^3)(0.8\text{ m}) - (1000\text{ kg/m}^3)(0.4\text{ m})$$

$$- (850\text{ kg/m}^3)(0.6\text{ m})]\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right)$$

$$= \mathbf{97.8\ kPa}$$

Discussion Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

3-13

Solution The barometric reading at a location is given in height of mercury column. The atmospheric pressure is to be determined.

Properties The density of mercury is given to be 13,600 kg/m^3.

Analysis The atmospheric pressure is determined directly from

$$P_{atm} = \rho gh$$

$$= (13{,}600\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.735\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right)$$

$$= \mathbf{98.1\ kPa}$$

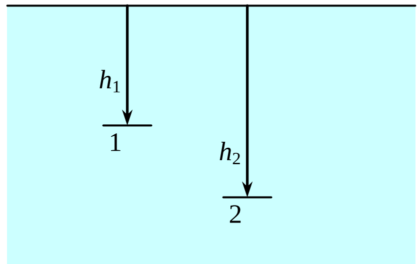
Discussion We round off the final answer to three significant digits. 100 kPa is a fairly typical value of atmospheric pressure on land slightly above sea level.

3-14

Solution The gage pressure in a liquid at a certain depth is given. The gage pressure in the same liquid at a different depth is to be determined.

Assumptions The variation of the density of the liquid with depth is negligible.

Analysis The gage pressure at two different depths of a liquid can be expressed as $P_1 = \rho g h_1$ and $P_2 = \rho g h_2$. Taking their ratio,



$$\frac{P_2}{P_1} = \frac{\rho g h_2}{\rho g h_1} = \frac{h_2}{h_1}$$

Solving for P_2 and substituting gives

$$P_2 = \frac{h_2}{h_1} P_1 = \frac{12\text{ m}}{3\text{ m}}(28\text{ kPa}) = \mathbf{112\ kPa}$$

Discussion Note that the gage pressure in a given fluid is proportional to depth.

3-15

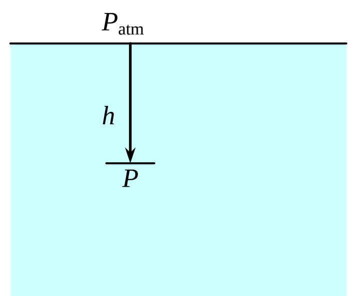
Solution The absolute pressure in water at a specified depth is given. The local atmospheric pressure and the absolute pressure at the same depth in a different liquid are to be determined.

Assumptions The liquid and water are incompressible.

Properties The specific gravity of the fluid is given to be SG = 0.78. We take the density of water to be 1000 kg/m^3. Then density of the liquid is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{H_2O} = (0.78)(1000\text{ kg/m}^3) = 780\ \text{kg/m}^3$$

Analysis (*a*) Knowing the absolute pressure, the atmospheric pressure can be determined from



$$\begin{aligned} P_{atm} &= P - \rho g h \\ &= (175\text{ kPa}) - (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(8\text{ m})\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right) \\ &= 96.52\text{ kPa} \cong \mathbf{96.5\,kPa} \end{aligned}$$

(*b*) The absolute pressure at a depth of 8 m in the other liquid is

$$\begin{aligned} P &= P_{atm} + \rho g h \\ &= (96.52\text{ kPa}) + (780\text{ kg/m}^3)(9.81\text{ m/s}^2)(8\text{ m})\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right) \\ &= 157.7\text{ kPa} \cong \mathbf{158\,kPa} \end{aligned}$$

Discussion Note that at a given depth, the pressure in the lighter fluid is lower, as expected.

3-16E

Solution It is to be shown that 1 kgf/cm^2 = 14.223 psi.

Analysis Noting that 1 kgf = 9.80665 N, 1 N = 0.22481 lbf, and 1 in = 2.54 cm, we have

$$1\text{ kgf} = 9.80665\text{N} = (9.80665\text{N})\left(\frac{0.22481\text{lbf}}{1\text{ N}}\right) = 2.20463\text{lbf}$$

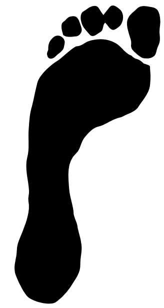
and $$1\text{ kgf/cm}^2 = 2.20463\text{ lbf/cm}^2 = (2.20463\text{ lbf/cm}^2)\left(\frac{2.54\text{ cm}}{1\text{ in}}\right)^2 = 14.223\text{ lbf/in}^2 = \mathbf{14.223psi}$$

Discussion This relationship may be used as a conversion factor.

3-17E

Solution The weight and the foot imprint area of a person are given. The pressures this man exerts on the ground when he stands on one and on both feet are to be determined.

Assumptions The weight of the person is distributed uniformly on foot imprint area.



Analysis The weight of the man is given to be 200 lbf. Noting that pressure is force per unit area, the pressure this man exerts on the ground is

(*a*) On one foot: $$P = \frac{W}{A} = \frac{200\ \text{lbf}}{36\ \text{in}^2} = 5.56\ \text{lbf/in}^2 = \mathbf{5.56\ psi}$$

(*a*) On both feet: $$P = \frac{W}{2A} = \frac{200\ \text{lbf}}{2\times 36\ \text{in}^2} = 2.78\ \text{lbf/in}^2 = \mathbf{2.78\ psi}$$

Discussion Note that the pressure exerted on the ground (and on the feet) is reduced by half when the person stands on both feet.

3-18

Solution The mass of a woman is given. The minimum imprint area per shoe needed to enable her to walk on the snow without sinking is to be determined.

Assumptions **1** The weight of the person is distributed uniformly on the imprint area of the shoes. **2** One foot carries the entire weight of a person during walking, and the shoe is sized for walking conditions (rather than standing). **3** The weight of the shoes is negligible.

Analysis The mass of the woman is given to be 55 kg. For a pressure of 0.5 kPa on the snow, the imprint area of one shoe must be

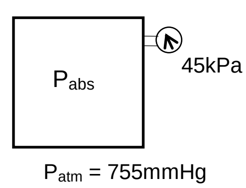
$$A = \frac{W}{P} = \frac{mg}{P} = \frac{(55\ \text{kg})(9.81\ \text{m/s}^2)}{0.5\ \text{kPa}}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) = \mathbf{1.08\ m^2}$$

Discussion This is a very large area for a shoe, and such shoes would be impractical to use. Therefore, some sinking of the snow should be allowed to have shoes of reasonable size.

3-19

Solution The vacuum pressure reading of a tank is given. The absolute pressure in the tank is to be determined.

Properties The density of mercury is given to be ρ = 13,590 kg/m^3.



Analysis The atmospheric (or barometric) pressure can be expressed as

$$\begin{aligned} P_{atm} &= \rho g h \\ &= (13{,}590\ \text{kg/m}^3)(9.807\ \text{m/s}^2)(0.755\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= 100.6\ \text{kPa} \end{aligned}$$

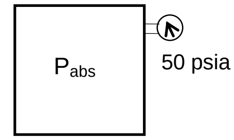
Then the absolute pressure in the tank becomes

$$P_{abs} = P_{atm} - P_{vac} = 100.6 - 45 = \mathbf{55.6\ kPa}$$

Discussion The gage pressure in the tank is the negative of the vacuum pressure, i.e., P_{gage} = –45 kPa.

3-20E

Solution A pressure gage connected to a tank reads 50 psi. The absolute pressure in the tank is to be determined.



Properties The density of mercury is given to be ρ = 848.4 lbm/ft^3.

Analysis The atmospheric (or barometric) pressure can be expressed as

$$\begin{aligned}P_{atm} &= \rho g h \\ &= (848.4\ \text{lbm/ft}^3)(32.174\ \text{ft/s}^2)(29.1/12\ \text{ft})\left(\frac{1\ \text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right) \\ &= 14.29\ \ \text{psia}\end{aligned}$$

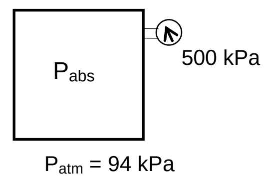
Then the absolute pressure in the tank is

$$P_{abs} = P_{gage} + P_{atm} = 50 + 14.29 = 64.29\ \text{psia} \cong \mathbf{64.3\ \ psia}$$

Discussion This pressure is more than four times as much as standard atmospheric pressure.

3-21

Solution A pressure gage connected to a tank reads 500 kPa. The absolute pressure in the tank is to be determined.



Analysis The absolute pressure in the tank is determined from

$$P_{\text{abs}} = P_{\text{gage}} + P_{\text{atm}} = 500 + 94 = \mathbf{594\ kPa}$$

Discussion This pressure is almost six times greater than standard atmospheric pressure.

3-22

Solution The pressure given in mm Hg unit is to be converted to psia.

Analysis Using the mm Hg to kPa and kPa to psia units conversion factors,

$$P = (1500\ \text{mm Hg})\left(\frac{0.1333\ \text{kPa}}{1\ \text{mm Hg}}\right)\left(\frac{1\ \text{psia}}{6.895\ \text{kPa}}\right) = \mathbf{29.0\ psia}$$

3-23

Solution The vacuum pressure given in kPa unit is to be converted to various units.

Analysis Using the definition of vacuum pressure,

$$P_{\text{gage}} = \text{not applicable for pressures below atmospheric pressure}$$

$$P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}} = 98 - 80 = \mathbf{18\ kPa}$$

Then using the conversion factors,

$$P_{\text{abs}} = (18\ \text{kPa})\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right) = \mathbf{18\ kN/m^2}$$

$$P_{\text{abs}} = (18\ \text{kPa})\left(\frac{1\ \text{lbf/in}^2}{6.895\ \text{kPa}}\right) = \mathbf{2.61\ lbf/in^2}$$

$$P_{\text{abs}} = (18\ \text{kPa})\left(\frac{1\ \text{psi}}{6.895\ \text{kPa}}\right) = \mathbf{2.61\ psi}$$

$$P_{\text{abs}} = (18\ \text{kPa})\left(\frac{1\ \text{mm Hg}}{0.1333\ \text{kPa}}\right) = \mathbf{135\ mm\ Hg}$$

3-24

Solution Water is raised from a reservoir through a vertical tube by the sucking action of a piston. The force needed to raise the water to a specified height is to be determined, and the pressure at the piston face is to be plotted against height.

Assumptions **1** Friction between the piston and the cylinder is negligible. **2** Accelerational effects are negligible.

Properties We take the density of water to be ρ = 1000 kg/m^3.

Analysis Noting that the pressure at the free surface is P_{atm} and hydrostatic pressure in a fluid decreases linearly with increasing height, the pressure at the piston face is

$$P = P_{\text{atm}} - \rho gh = 95\,\text{kPa} - (1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(1.5\,\text{m})\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1\ \text{kN/m}^2}\right) = 81.3\,\text{kPa}$$

Piston face area is

$$A = \pi D^2/4 = \pi(0.3\,\text{m})^2/4 = 0.07069\,\text{m}^2$$

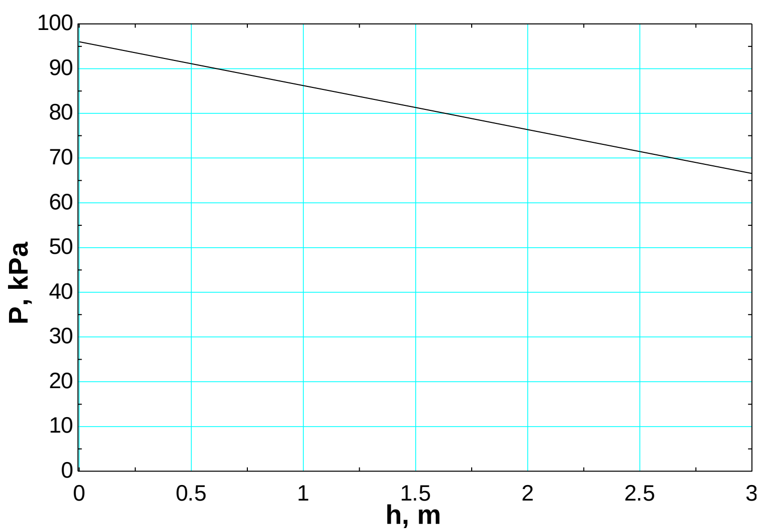
A force balance on the piston yields

$$F = (P_{\text{atm}} - P)A = (95 - 81.3\,\text{kPa})((0.07068\,\text{m}^2)\left(\frac{1\,\text{kN/m}^2}{1\,\text{kPa}}\right)\left(\frac{1\,\text{kPa}}{1\ \text{kN/m}^2}\right) = \mathbf{1.04\,kN}$$

Repeating calculations for h = 3 m gives P = 66.6 kPa and F = **2.08 kN**.

Using EES, the absolute pressure can be calculated from $P = P_{\text{atm}} - \rho gh$ for various values of h from 0 to 3 m, and the results can be plotted as shown below:

```
P_atm = 96 [kPa]
"h = 3 [m]"
D = 0.30 [m]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
P = P_atm - rho*g*h*CONVERT(Pa, kPa)
A = pi*D^2/4
F =(P_atm - P)*A
```

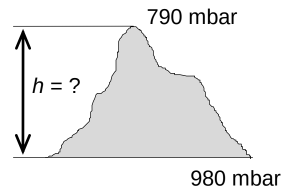


Discussion Note that the pressure at the piston face decreases, and the force needed to raise water increases linearly with increasing height of water column relative to the free surface.

3-25

Solution A mountain hiker records the barometric reading before and after a hiking trip. The vertical distance climbed is to be determined.

Assumptions The variation of air density and the gravitational acceleration with altitude is negligible.



Properties The density of air is given to be ρ = 1.20 kg/m^3.

Analysis Taking an air column between the top and the bottom of the mountain and writing a force balance per unit base area, we obtain

$$W_{\text{air}} / A = P_{\text{bottom}} - P_{\text{top}} \longrightarrow (\rho g h)_{air} = P_{\text{bottom}} - P_{\text{top}} \longrightarrow h = \frac{P_{\text{bottom}} - P_{\text{top}}}{\rho g}$$

$$h = \frac{(0.980 - 0.790)\text{ bar}}{(1.20\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{100{,}000\text{ N/m}^2}{1\text{ bar}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{1614\,m}$$

which is also the distance climbed.

Discussion A similar principle is used in some aircraft instruments to measure elevation.

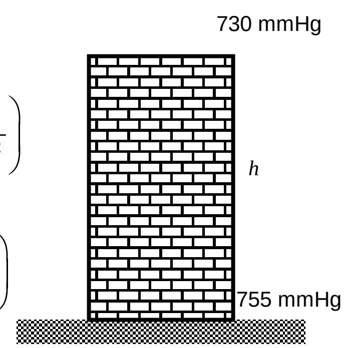
3-26

Solution A barometer is used to measure the height of a building by recording reading at the bottom and at the top of the building. The height of the building is to be determined.

Assumptions The variation of air density with altitude is negligible.

Properties The density of air is given to be ρ = 1.18 kg/m^3. The density of mercury is 13,600 kg/m^3.

Analysis Atmospheric pressures at the top and at the bottom of the building are



$$\begin{aligned} P_{\text{top}} &= (\rho g h)_{\text{top}} \\ &= (13{,}600\text{ kg/m}^3)(9.807\text{ m/s}^2)(0.730\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right) \\ &= 97.36\text{ kPa} \end{aligned}$$

$$\begin{aligned} P_{\text{bottom}} &= (\rho g h)_{\text{bottom}} \\ &= (13{,}600\text{ kg/m}^3)(9.807\text{ m/s}^2)(0.755\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right) \\ &= 100.70\text{ kPa} \end{aligned}$$

Taking an air column between the top and the bottom of the building, we write a force balance per unit base area,

$$W_{\text{air}} / A = P_{\text{bottom}} - P_{\text{top}} \quad \text{and} \quad (\rho g h)_{\text{air}} = P_{\text{bottom}} - P_{\text{top}}$$

$$(1.18\text{ kg/m}^3)(9.807\text{ m/s}^2)(h)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right) = (100.70 - 97.36)\text{ kPa}$$

which yields h = 288.6 m ≅ **289 m**, which is also the height of the building.

Discussion There are more accurate ways to measure the height of a building, but this method is quite simple.

3-27

Solution The previous problem is reconsidered. The EES solution is to be printed out, including proper units.

Analysis The EES *Equations* window is printed below, followed by the *Solution* window.

```
P_bottom=755"[mmHg]"
P_top=730"[mmHg]"
g=9.807    "[m/s^2]"     "local acceleration of gravity at sea level"
rho=1.18"[kg/m^3]"
DELTAP_abs=(P_bottom-P_top)*CONVERT('mmHg','kPa')"[kPa]"     "Delta P reading from the barometers, converted from mmHg to kPa."
DELTAP_h =rho*g*h/1000 "[kPa]"         "Equ. 1-16. Delta P due to the air fluid column height, h, between the top and bottom of the building."
"Instead of dividing by 1000 Pa/kPa we could have multiplied rho*g*h by the EES function, CONVERT('Pa','kPa')"
DELTAP_abs=DELTAP_h
```

SOLUTION

Variables in Main

DELTAP_abs=3.333 [kPa]	DELTAP_h=3.333 [kPa]
g=9.807 [m/s^2]	h=288 [m]
P_bottom=755 [mmHg]	P_top=730 [mmHg]
rho=1.18 [kg/m^3]	

Discussion To obtain the solution in EES, simply click on the icon that looks like a calculator, or Calculate-Solve.

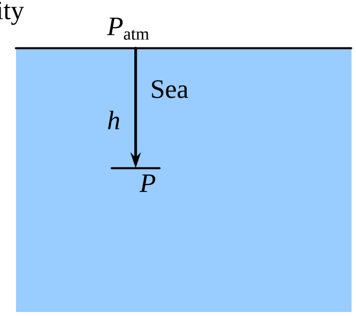
3-28

Solution A diver is moving at a specified depth from the water surface. The pressure exerted on the surface of the diver by the water is to be determined.

Assumptions The variation of the density of water with depth is negligible.

Properties The specific gravity of sea water is given to be SG = 1.03. We take the density of water to be 1000 kg/m^3.

Analysis The density of the sea water is obtained by multiplying its specific gravity by the density of water which is taken to be 1000 kg/m^3:



$$\rho = \text{SG} \times \rho_{H_2O} = (1.03)(1000\ \text{kg/m}^3) = 1030\ \text{kg/m}^3$$

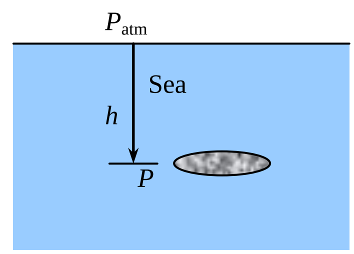
The pressure exerted on a diver at 20 m below the free surface of the sea is the absolute pressure at that location:

$$\begin{aligned} P &= P_{atm} + \rho g h \\ &= (101\ \text{kPa}) + (1030\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(20\ \text{m})\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= \mathbf{303\ kPa} \end{aligned}$$

Discussion This is about 3 times the normal sea level value of atmospheric pressure.

3-29E

Solution A submarine is cruising at a specified depth from the water surface. The pressure exerted on the surface of the submarine by water is to be determined.



Assumptions The variation of the density of water with depth is negligible.

Properties The specific gravity of sea water is given to be SG = 1.03. The density of water at 32°F is 62.4 lbm/ft^3.

Analysis The density of the seawater is obtained by multiplying its specific gravity by the density of water,

$$\rho = \mathrm{SG} \times \rho_{H_2O} = (1.03)(62.4\ \mathrm{lbm/ft^3}) = 64.27\ \mathrm{lbm/ft^3}$$

The pressure exerted on the surface of the submarine cruising 300 ft below the free surface of the sea is the absolute pressure at that location:

$$\begin{aligned} P &= P_{atm} + \rho g h \\ &= (14.7\ \mathrm{psia}) + (64.27\ \mathrm{lbm/ft^3})(32.174\ \mathrm{ft/s^2})(225\ \mathrm{ft})\left(\frac{1\ \mathrm{lbf}}{32.174\ \mathrm{lbm \cdot ft/s^2}}\right)\left(\frac{1\ \mathrm{ft^2}}{144\ \mathrm{in^2}}\right) \\ &= \mathbf{115\ psia} \end{aligned}$$

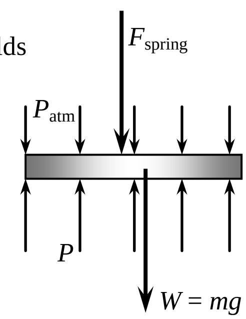
where we have rounded the final answer to three significant digits.

Discussion This is about 8 times the value of atmospheric pressure at sea level.

3-30

Solution A gas contained in a vertical piston-cylinder device is pressurized by a spring and by the weight of the piston. The pressure of the gas is to be determined.

Analysis Drawing the free body diagram of the piston and balancing the vertical forces yields



$$PA = P_{atm}A + W + F_{spring}$$

Thus,

$$\begin{aligned} P &= P_{atm} + \frac{mg + F_{spring}}{A} \\ &= (95\ \mathrm{kPa}) + \frac{(4\ \mathrm{kg})(9.807\ \mathrm{m/s^2}) + 60\ \mathrm{N}}{35 \times 10^{-4}\ \mathrm{m^2}}\left(\frac{1\ \mathrm{kPa}}{1000\ \mathrm{N/m^2}}\right) = 123.4\ \mathrm{kPa} \cong \mathbf{123\ kPa} \end{aligned}$$

Discussion This setup represents a crude but functional way to control the pressure in a tank.

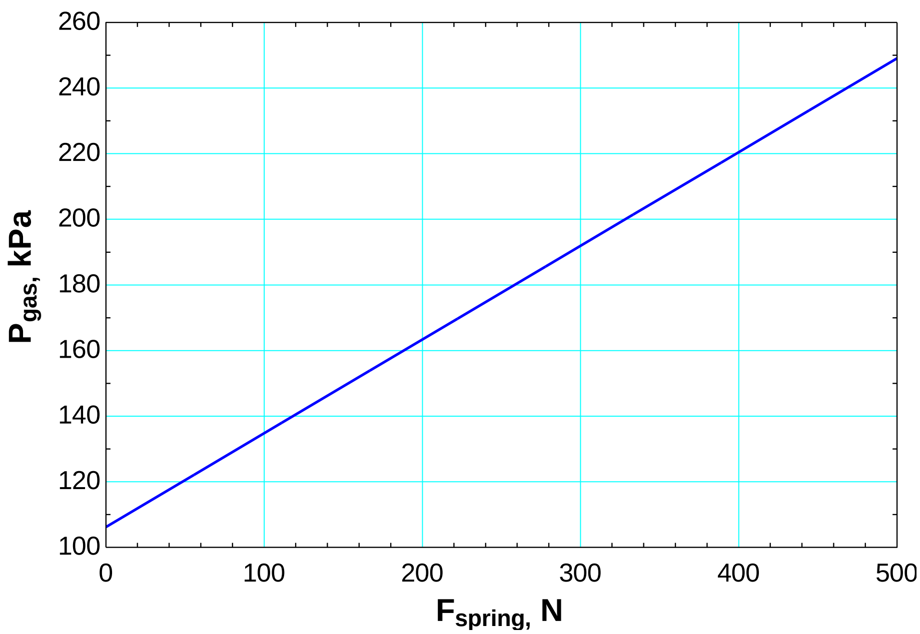
3-31

Solution The previous problem is reconsidered. The effect of the spring force in the range of 0 to 500 N on the pressure inside the cylinder is to be investigated. The pressure against the spring force is to be plotted, and results are to be discussed.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.807"[m/s^2]"
P_atm= 95"[kPa]"
m_piston=4"[kg]"
{F_spring=60"[N]"}
A=35*CONVERT('cm^2','m^2')"[m^2]"
W_piston=m_piston*g"[N]"
F_atm=P_atm*A*CONVERT('kPa','N/m^2')"[N]"
"From the free body diagram of the piston, the balancing vertical forces yield:"
F_gas= F_atm+F_spring+W_piston"[N]"
P_gas=F_gas/A*CONVERT('N/m^2','kPa')"[kPa]"
```

F_{spring} [N]	P_{gas} [kPa]
0	106.2
55.56	122.1
111.1	138
166.7	153.8
222.2	169.7
277.8	185.6
333.3	201.4
388.9	217.3
444.4	233.2
500	249.1



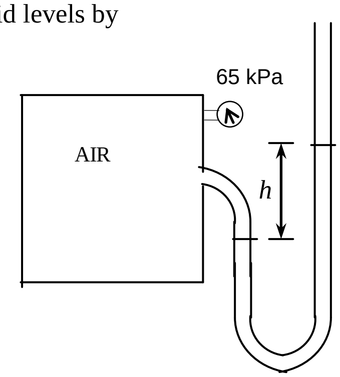
Discussion The relationship is linear, as expected.

3-32

Solution Both a pressure gage and a manometer are attached to a tank of gas to measure its pressure. For a specified reading of gage pressure, the difference between the fluid levels of the two arms of the manometer is to be determined for mercury and water.

Properties The densities of water and mercury are given to be ρ_{water} = 1000 kg/m^3 and be ρ_{Hg} = 13,600 kg/m^3.

Analysis The gage pressure is related to the vertical distance *h* between the two fluid levels by



$$P_{gage} = \rho g h \quad \longrightarrow \quad h = \frac{P_{gage}}{\rho g}$$

(*a*) For mercury,

$$h = \frac{P_{gage}}{\rho_{Hg} g} = \frac{65 \text{ kPa}}{(13600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right)\left(\frac{1000 \text{ kg/m}\cdot\text{s}^2}{1 \text{ kN}}\right) = \mathbf{0.49\ m}$$

(*b*) For water,

$$h = \frac{P_{gage}}{\rho_{H_2O} g} = \frac{65 \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{m/s}^2)}\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right)\left(\frac{1000 \text{ kg/m}\cdot\text{s}^2}{1 \text{ kN}}\right) = \mathbf{6.63\ m}$$

Discussion The manometer with water is more precise since the column height is bigger (better resolution). However, a column of water more than 8 meters high would be impractical, so mercury is the better choice of manometer fluid here. *Note*: Mercury vapors are hazardous, and the use of mercury is no longer encouraged.

3-33

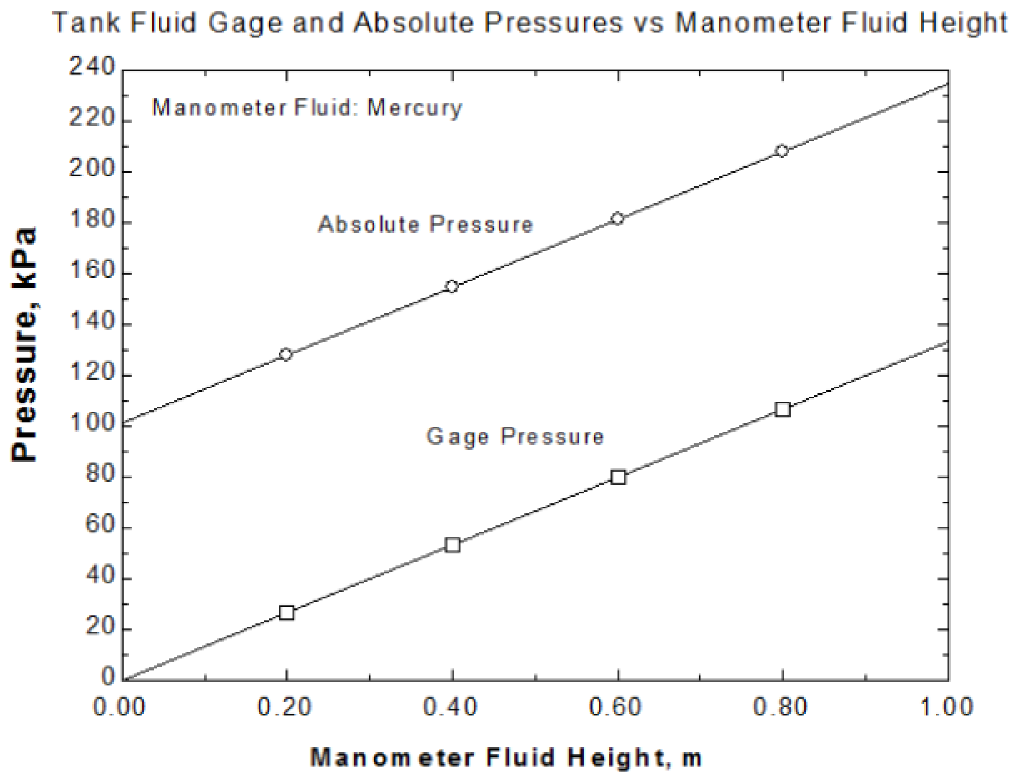
Solution The previous problem is reconsidered. The effect of the manometer fluid density in the range of 800 to 13,000 kg/m^3 on the differential fluid height of the manometer is to be investigated. Differential fluid height is to be plotted as a function of the density, and the results are to be discussed.

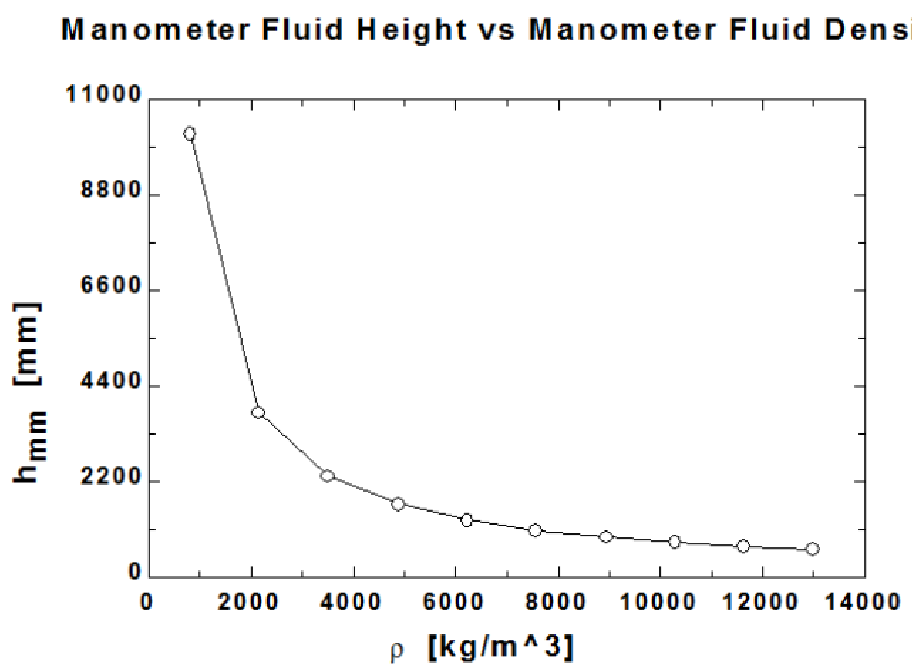
Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
Function fluid_density(Fluid$)
   If fluid$='Mercury' then  fluid_density=13600 else fluid_density=1000
end

{Input from the diagram window.  If the diagram window is hidden, then all of the input must come from the
equations window.  Also note that brackets can also denote comments - but these comments do not appear in the
formatted equations window.}

{Fluid$='Mercury'
P_atm = 101.325                    "kpa"
DELTAP=80                "kPa  Note how DELTAP is displayed on the Formatted Equations Window."}

g=9.807                              "m/s2, local acceleration of gravity at sea level"
rho=Fluid_density(Fluid$)       "Get the fluid density, either Hg or H2O, from the function"
"To plot fluid height against density place {} around the above equation.  Then set up the parametric table and solve."
DELTAP = RHO*g*h/1000
"Instead of dividing by 1000 Pa/kPa we could have multiplied by the EES function, CONVERT('Pa','kPa')"
h_mm=h*convert('m','mm')       "The fluid height in mm is found using the built-in CONVERT function."
P_abs= P_atm + DELTAP

"To make the graph, hide the diagram window and remove the {}brackets from Fluid$ and from P_atm.  Select New
Parametric Table from the Tables menu.  Choose P_abs, DELTAP and h to be in the table.  Choose Alter Values from
the Tables menu.  Set values of h to range from 0 to 1 in steps of 0.2.  Choose Solve Table (or press F3) from the
Calculate menu.  Choose New Plot Window from the Plot menu.  Choose to plot P_abs vs h and then choose Overlay
Plot from the Plot menu and plot DELTAP on the same scale."
```

Results:

h_{mm} [mm]	ρ [kg/m^3]
10197	800
3784	2156
2323	3511
1676	4867
1311	6222
1076	7578
913.1	8933
792.8	10289
700.5	11644
627.5	13000





Discussion Many comments are provided in the Equation window above to help you learn some of the features of EES.

3-34

Solution A relation for the variation of pressure in a gas with density is given. A relation for the variation of pressure with elevation is to be obtained.

Analysis Since $p = C\rho^n$, we write $\dfrac{p}{\rho^n} = \dfrac{p_o}{\rho_o^n} = C$ (1)

The pressure field in a fluid is given by,

$$dp = -\rho g dz \tag{2}$$

Combining Eqs. 1 and 2 yields

$$\int_{p_o}^{p} \frac{dp}{\dfrac{\rho_o}{p_o^{1/n}} p^{1/n}} = -g\int_0^z dz$$

$$\frac{p_o^{1/n}}{\rho_o}\int_{p_o}^{p} p^{-1/n} dp = \frac{p_o^{1/n}}{\rho_o}\left.\frac{p^{1-1/n}}{1-1/n}\right|_{p_o}^{p} = \frac{n}{n-1}\frac{p_o^{1/n}}{\rho_o}\left. p^{1-1/n}\right|_{p_o}^{p} = -gz$$

$$\frac{n}{n-1}\frac{p_o^{1/n}}{\rho_o}\left(p^{1-1/n} - p_o^{1-1/n}\right) = -gz$$

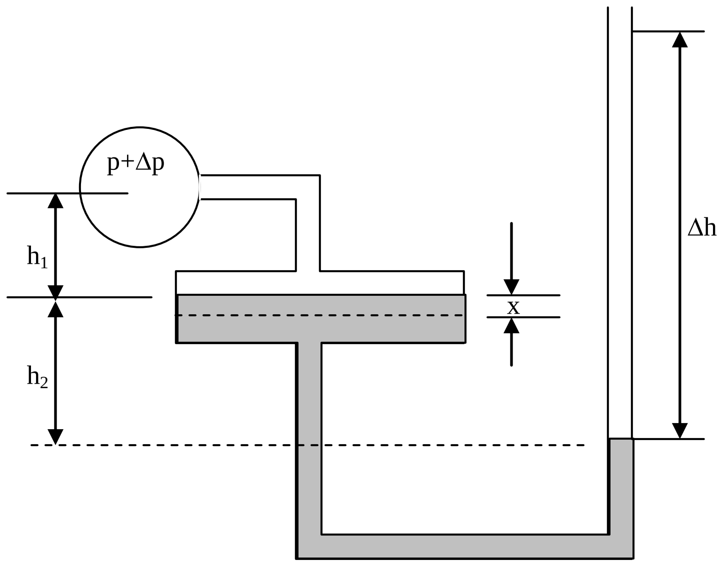
$$p = p(z) = \left(p_o^{\frac{n-1}{n}} - \frac{n-1}{n}\frac{\rho_o g}{p_o^{1/n}} z\right)^{\frac{n}{n-1}} \tag{3}$$

After having calculated the pressure at any elevation, using Eq. 1, the density at that point can also be determined.

3-35

Solution The change in pipe pressure is to be determined for a given system.

Analysis



Initially,

$$p + \gamma_w h_1 + \gamma_{gly} h_2 = 0$$

After the pressure is applied

$$p + \Delta p + \gamma_w (h_1 + x) + \gamma_{gly}(h_2 - x) - \gamma_{gly}\Delta h = 0$$

On the other hand, from the continuity;

$$\frac{\pi D^2}{4} x = \frac{\pi d^2}{4}\Delta h \text{ , or } \quad x = \left(\frac{d}{D}\right)^2 \Delta h = \left(\frac{3}{30}\right)^2 \Delta h = 0.01\Delta h$$

From the first equation,

$$P = -\gamma_w h_1 - \gamma_{gly} h_2 \, .$$

Substituting into second equation, and solving for Δp will give

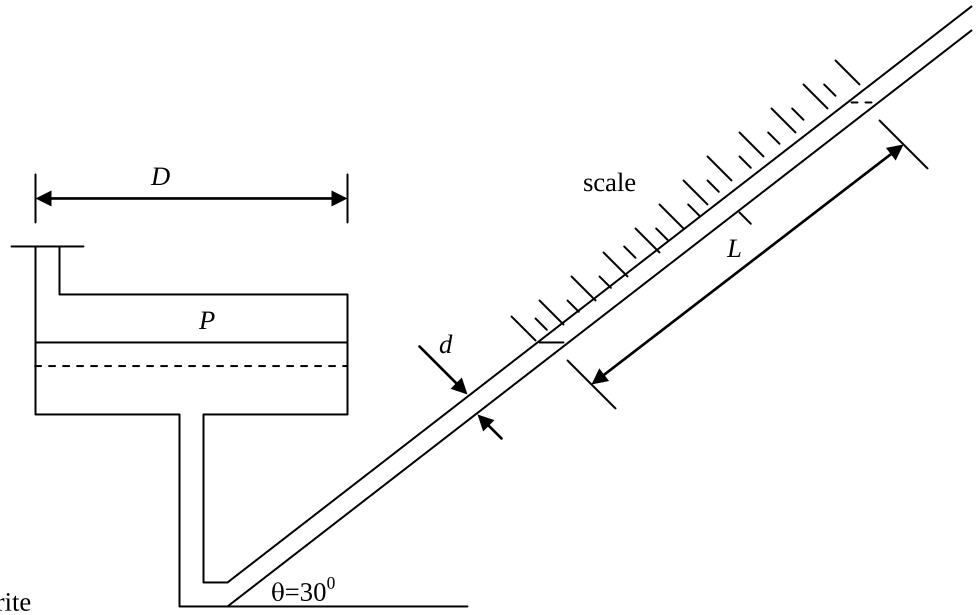
$$\Delta p = \gamma_{gly}\Delta h(1 + 0.01) - \gamma_w \Delta h(0.01)$$
$$= SG\gamma_w \Delta h(1.01) - \gamma_w \Delta h(0.01)$$
$$= \gamma_w \Delta h(1.01SG - 0.01) = 9810 \times 70 \times 10^{-3}(1.01 \times 1.26 - 0.01)$$

$\Delta P =$ **867 Pa**

3-36

Solution A manometer is designed to measure pressures. A certain geometric ratio in the manımeter for keeping the error under a specified value is to be determined.

Analysis



Since $P_A = P_B$ we write

$$\frac{P}{\gamma} = x + LSin\theta,$$

On the other hand

$$x\frac{\pi D^2}{4} = L\frac{\pi d^2}{4}, \text{ or } x = \left(\frac{d}{D}\right)^2 L$$

Therefore

$$\frac{P}{\gamma} = \left(\frac{d}{D}\right)^2 L + LSin\theta, \text{ or } P = \gamma L\left[\left(\frac{d}{D}\right)^2 + Sin\theta\right]$$

In order to find the error due to the reading error in "*L*", we differentiate *P* wrt *L* as below:

$$dP = \gamma dL\left[\left(\frac{d}{D}\right)^2 + Sin\theta\right]$$

From the definition of error, we obtain

$$error = \frac{dP}{P} = \left[\left(\frac{d}{D}\right)^2 + Sin\theta\right]\frac{dL}{P/\gamma}$$

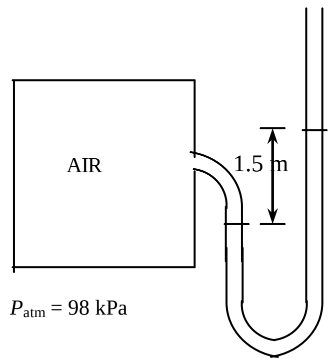
From the given data $dP/P = 0.025$, $Sin\theta = 0.5$, $P/\gamma = 100/9810 = 0.010194 = 10.194\ mm$ and $dL = 0.5\ mm$

$$0.025 = \left[\left(\frac{d}{D}\right)^2 + 0.50\right]\frac{0.5}{10.194},$$

$$d/D = 0.0985 \approx 0.10$$

3-37

Solution The air pressure in a tank is measured by an oil manometer. For a given oil-level difference between the two columns, the absolute pressure in the tank is to be determined.



Properties The density of oil is given to be ρ = 850 kg/m^3.

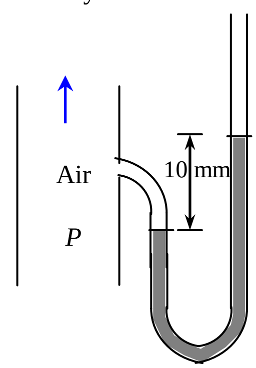
Analysis The absolute pressure in the tank is determined from

$$P = P_{atm} + \rho gh$$
$$= (98\text{ kPa}) + (850\text{ kg/m}^3)(9.81\text{ m/s}^2)(1.50\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right)$$
$$= \mathbf{111\,kPa}$$

Discussion If a heavier liquid, such as water, were used for the manometer fluid, the column height would be smaller, and thus the reading would be less precise (lower resolution).

3-38

Solution The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.



Properties The density of mercury is given to be ρ = 13,600 kg/m^3.

Analysis (*a*) The pressure in the duct is above atmospheric pressure since the fluid column on the duct side is at a lower level.

(b) The absolute pressure in the duct is determined from

(c)
$$P = P_{atm} + \rho gh$$
$$= (100\text{ kPa}) + (13{,}600\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.010\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right)$$
$$= \mathbf{101.3\ kPa}$$

Discussion When measuring pressures in a fluid flow, the *difference* between two pressures is usually desired. In this case, the difference is between the measurement point and atmospheric pressure.

3-39

Solution The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.

Properties The density of mercury is given to be ρ = 13,600 kg/m^3.

Analysis (*a*) The pressure in the duct is above atmospheric pressure since the fluid column on the duct side is at a lower level.

(*b*) The absolute pressure in the duct is determined from

$$P = P_{atm} + \rho gh$$
$$= (100\ \text{kPa}) + (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.030\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right)$$
$$= 104.00\text{ kPa} \cong \mathbf{104\ \ kPa}$$

Discussion The final result is given to three significant digits.

3-40

Solution The systolic and diastolic pressures of a healthy person are given in mm of Hg. These pressures are to be expressed in kPa, psi, and meters of water column.

Assumptions Both mercury and water are incompressible substances.

Properties We take the densities of water and mercury to be 1000 kg/m^3 and 13,600 kg/m^3, respectively.

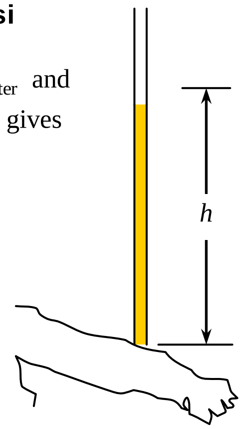
Analysis Using the relation $P = \rho g h$ for gage pressure, the high and low pressures are expressed as

$$P_{\text{high}} = \rho g h_{\text{high}} = (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.12\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) = \mathbf{16.0\ kPa}$$

$$P_{\text{low}} = \rho g h_{\text{low}} = (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.08\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) = \mathbf{10.7\ kPa}$$

Noting that 1 psi = 6.895 kPa,

$$P_{\text{high}} = (16.0\ \text{kPa})\left(\frac{1\ \text{psi}}{6.895\text{kPa}}\right) = \mathbf{2.32\ psi} \quad \text{and} \quad P_{\text{low}} = (10.7\ \text{kPa})\left(\frac{1\ \text{psi}}{6.895\text{kPa}}\right) = \mathbf{1.55\ psi}$$



For a given pressure, the relation $P = \rho g h$ is expressed for mercury and water as $P = \rho_{\text{water}} g h_{\text{water}}$ and $P = \rho_{\text{mercury}} g h_{\text{mercury}}$. Setting these two relations equal to each other and solving for water height gives

$$P = \rho_{\text{water}} g h_{\text{water}} = \rho_{\text{mercury}} g h_{\text{mercury}} \quad \rightarrow \quad h_{\text{water}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury}}$$

Therefore,

$$h_{\text{water, high}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, high}} = \frac{13{,}600\ \text{kg/m}^3}{1000\ \text{kg/m}^3}(0.12\ \text{m}) = \mathbf{1.63\ m}$$

$$h_{\text{water, low}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, low}} = \frac{13{,}600\ \text{kg/m}^3}{1000\ \text{kg/m}^3}(0.08\ \text{m}) = \mathbf{1.09\ m}$$

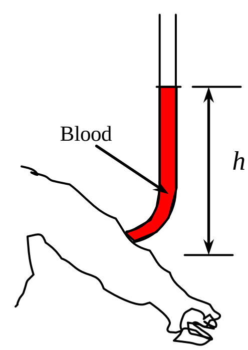
Discussion Note that measuring blood pressure with a water monometer would involve water column heights higher than the person's height, and thus it is impractical. This problem shows why mercury is a suitable fluid for blood pressure measurement devices.

3-41

Solution A vertical tube open to the atmosphere is connected to the vein in the arm of a person. The height that the blood rises in the tube is to be determined.

Assumptions **1** The density of blood is constant. **2** The gage pressure of blood is 120 mmHg.

Properties The density of blood is given to be ρ = 1040 kg/m^3.



Analysis For a given gage pressure, the relation $P = \rho g h$ can be expressed for mercury and blood as $P = \rho_{\text{blood}} g h_{\text{blood}}$ and $P = \rho_{\text{mercury}} g h_{\text{mercury}}$. Setting these two relations equal to each other we get

$$P = \rho_{\text{blood}} g h_{\text{blood}} = \rho_{\text{mercury}} g h_{\text{mercury}}$$

Solving for blood height and substituting gives

$$h_{\text{blood}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{blood}}} h_{\text{mercury}} = \frac{13{,}600\ \text{kg/m}^3}{1040\ \text{kg/m}^3}(0.12\ \text{m}) = \mathbf{1.57 m}$$

Discussion Note that the blood can rise about one and a half meters in a tube connected to the vein. This explains why IV tubes must be placed high to force a fluid into the vein of a patient.

3-42

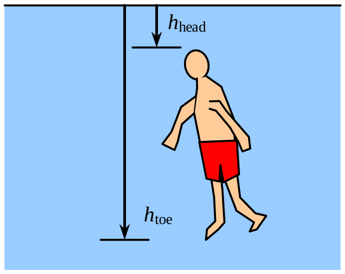
Solution A man is standing in water vertically while being completely submerged. The difference between the pressure acting on his head and the pressure acting on his toes is to be determined.

Assumptions Water is an incompressible substance, and thus the density does not change with depth.

Properties We take the density of water to be ρ =1000 kg/m^3.

Analysis The pressures at the head and toes of the person can be expressed as

$$P_{\text{head}} = P_{\text{atm}} + \rho g h_{\text{head}} \quad \text{and} \quad P_{\text{toe}} = P_{\text{atm}} + \rho g h_{\text{toe}}$$



where h is the vertical distance of the location in water from the free surface. The pressure difference between the toes and the head is determined by subtracting the first relation above from the second,

$$P_{\text{toe}} - P_{\text{head}} = \rho g h_{\text{toe}} - \rho g h_{\text{head}} = \rho g (h_{\text{toe}} - h_{\text{head}})$$

Substituting,

$$P_{\text{toe}} - P_{\text{head}} = (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(1.73 \text{ m - 0})\left(\frac{1\text{N}}{1\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{kPa}}{1000\,\text{N/m}^2}\right) = \mathbf{17.0\ kPa}$$

Discussion This problem can also be solved by noting that the atmospheric pressure (1 atm = 101.325 kPa) is equivalent to 10.3-m of water height, and finding the pressure that corresponds to a water height of 1.73 m.

3-43

Solution Water is poured into the U-tube from one arm and oil from the other arm. The water column height in one arm and the ratio of the heights of the two fluids in the other arm are given. The height of each fluid in that arm is to be determined.

Assumptions Both water and oil are incompressible substances.

Properties The density of oil is given to be ρ_{oil} = 790 kg/m^3. We take the density of water to be ρ_{w} =1000 kg/m^3.

Analysis The height of water column in the left arm of the manometer is given to be h_{w1} = 0.70 m. We let the height of water and oil in the right arm to be h_{w2} and h_{a}, respectively. Then, h_{a} = 6h_{w2}. Noting that both arms are open to the atmosphere, the pressure at the bottom of the U-tube can be expressed as

$$P_{\text{bottom}} = P_{\text{atm}} + \rho_{\text{w}} g h_{\text{w1}} \quad \text{and} \quad P_{\text{bottom}} = P_{\text{atm}} + \rho_{\text{w}} g h_{\text{w2}} + \rho_{\text{a}} g h_{\text{a}}$$

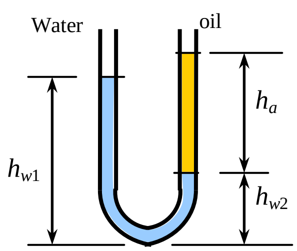
Setting them equal to each other and simplifying,

$$\rho_{\text{w}} g h_{\text{w1}} = \rho_{\text{w}} g h_{\text{w2}} + \rho_{\text{a}} g h_{\text{a}} \quad \rightarrow \quad \rho_{\text{w}} h_{\text{w1}} = \rho_{\text{w}} h_{\text{w2}} + \rho_{\text{a}} h_{\text{a}} \quad \rightarrow \quad h_{\text{w1}} = h_{\text{w2}} + (\rho_{\text{a}} / \rho_w) h_{\text{a}}$$

Noting that h_{a} = 6h_{w2} and we take ρ_{a} =ρ_{oil}, the water and oil column heights in the second arm are determined to be

$$0.7\,\text{m} = h_{w2} + (790/1000)6h_{w2} \quad \rightarrow \quad h_{w2} = \mathbf{0.122m}$$

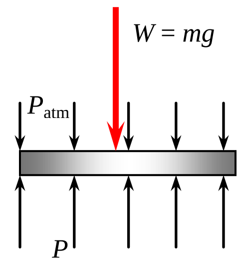
$$0.7\,\text{m} = 0.122\,\text{m} + (790/1000)h_a \quad \rightarrow \quad h_a = \mathbf{0.732m}$$



Discussion Note that the fluid height in the arm that contains oil is higher. This is expected since oil is lighter than water.

3-44

Solution The hydraulic lift in a car repair shop is to lift cars. The fluid gage pressure that must be maintained in the reservoir is to be determined.



Assumptions The weight of the piston of the lift is negligible.

Analysis Pressure is force per unit area, and thus the gage pressure required is simply the ratio of the weight of the car to the area of the lift,

$$P_{gage} = \frac{W}{A} = \frac{mg}{\pi D^2/4} = \frac{(1800\text{ kg})(9.81\text{ m/s}^2)}{\pi(0.40\text{ m})^2/4}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 141\text{ kN/m}^2 = \mathbf{141\ kPa}$$

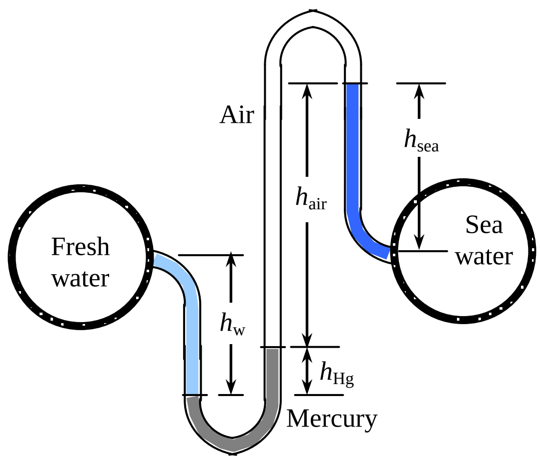
Discussion Note that the pressure level in the reservoir can be reduced by using a piston with a larger area.

3-45

Solution Fresh and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer. The pressure difference between the two pipelines is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible.

Properties The densities of seawater and mercury are given to be ρ_{sea} = 1035 kg/m^3 and ρ_{Hg} = 13,600 kg/m^3. We take the density of water to be ρ_w =1000 kg/m^3.



Analysis Starting with the pressure in the fresh water pipe (point 1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the sea water pipe (point 2), and setting the result equal to P_2 gives

$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{air} g h_{air} + \rho_{sea} g h_{sea} = P_2$$

Rearranging and neglecting the effect of air column on pressure,

$$P_1 - P_2 = -\rho_w g h_w + \rho_{Hg} g h_{Hg} - \rho_{sea} g h_{sea} = g(\rho_{Hg} h_{Hg} - \rho_w h_w - \rho_{sea} h_{sea})$$

Substituting,

$$\begin{aligned} P_1 - P_2 &= (9.81\text{ m/s}^2)[(13{,}600\text{ kg/m}^3)(0.1\text{ m}) \\ &\quad -(1000\text{ kg/m}^3)(0.5\text{ m}) - (1035\text{ kg/m}^3)(0.3\text{ m})]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) \\ &= 5.39\text{ kN/m}^2 = \mathbf{5.39\ kPa} \end{aligned}$$

Therefore, the pressure in the fresh water pipe is 5.39 kPa higher than the pressure in the sea water pipe.

Discussion A 0.70-m high air column with a density of 1.2 kg/m^3 corresponds to a pressure difference of 0.008 kPa. Therefore, its effect on the pressure difference between the two pipes is negligible.

3-46

Solution Fresh and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer. The pressure difference between the two pipelines is to be determined.

Assumptions All the liquids are incompressible.

Properties The densities of seawater and mercury are given to be ρ_{sea} = 1035 kg/m^3 and ρ_{Hg} = 13,600 kg/m^3. We take the density of water to be ρ_w =1000 kg/m^3. The specific gravity of oil is given to be 0.72, and thus its density is 720 kg/m^3.

Analysis Starting with the pressure in the fresh water pipe (point 1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the sea water pipe (point 2), and setting the result equal to P_2 gives

$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} + \rho_{sea} g h_{sea} = P_2$$

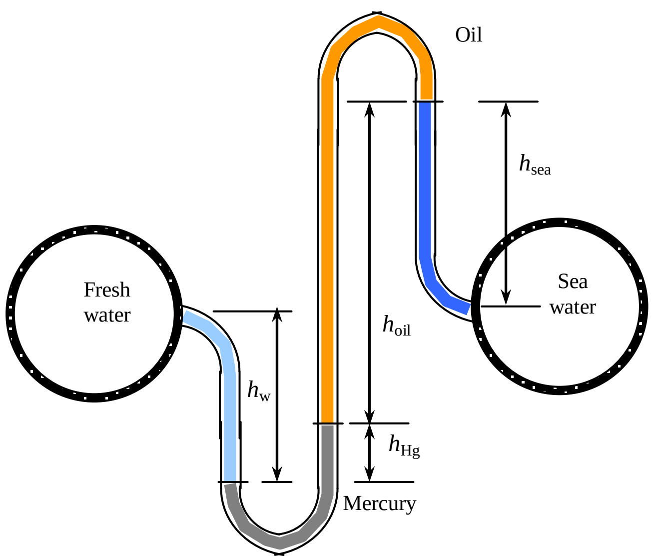
Rearranging,

$$\begin{aligned} P_1 - P_2 &= -\rho_w g h_w + \rho_{Hg} g h_{Hg} + \rho_{oil} g h_{oil} - \rho_{sea} g h_{sea} \\ &= g(\rho_{Hg} h_{Hg} + \rho_{oil} h_{oil} - \rho_w h_w - \rho_{sea} h_{sea}) \end{aligned}$$

Substituting,

$$\begin{aligned} P_1 - P_2 &= (9.81\,\text{m/s}^2)[(13{,}600\,\text{kg/m}^3)(0.1\,\text{m}) + (720\,\text{kg/m}^3)(0.7\,\text{m}) - (1000\,\text{kg/m}^3)(0.5\,\text{m}) \\ &\quad - (1035\,\text{kg/m}^3)(0.3\,\text{m})]\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right) \\ &= 10.3\,\text{kN/m}^2 = \mathbf{10.3\,kPa} \end{aligned}$$

Therefore, the pressure in the fresh water pipe is 10.3 kPa higher than the pressure in the sea water pipe.



Discussion The result is greater than that of the previous problem since the oil is heavier than the air.

3-47E

Solution The pressure in a natural gas pipeline is measured by a double U-tube manometer with one of the arms open to the atmosphere. The absolute pressure in the pipeline is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible. **3** The pressure throughout the natural gas (including the tube) is uniform since its density is low.

Properties We take the density of water to be $\rho_w = 62.4$ lbm/ft^3. The specific gravity of mercury is given to be 13.6, and thus its density is $\rho_{Hg} = 13.6 \times 62.4 = 848.6$ lbm/ft^3.

Analysis Starting with the pressure at point 1 in the natural gas pipeline, and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives

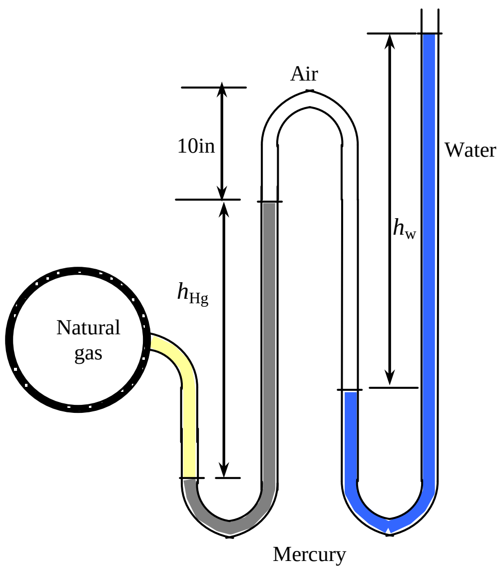
$$P_1 - \rho_{Hg} g h_{Hg} - \rho_{water} g h_{water} = P_{atm}$$

Solving for $P_{1,}$

$$P_1 = P_{atm} + \rho_{Hg} g h_{Hg} + \rho_{water} g h_1$$

Substituting,

$$P = 14.2 \text{ psia} + (32.2 \text{ ft/s}^2)[(848.6 \text{ lbm/ft}^3)(6/12 \text{ ft}) + (62.4 \text{lbm/ft}^3)(24/12 \text{ ft})]\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right)$$
$$= \mathbf{18.0 \text{ psia}}$$



Discussion Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly. Also, it can be shown that the 15-in high air column with a density of 0.075 lbm/ft^3 corresponds to a pressure difference of 0.00065 psi. Therefore, its effect on the pressure difference between the two pipes is negligible.

3-48E

Solution The pressure in a natural gas pipeline is measured by a double U-tube manometer with one of the arms open to the atmosphere. The absolute pressure in the pipeline is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The pressure throughout the natural gas (including the tube) is uniform since its density is low.

Properties We take the density of water to be ρ_w = 62.4 lbm/ft^3. The specific gravity of mercury is given to be 13.6, and thus its density is ρ_{Hg} = 13.6×62.4 = 848.6 lbm/ft^3. The specific gravity of oil is given to be 0.69, and thus its density is ρ_{oil} = 0.69×62.4 = 43.1 lbm/ft^3.

Analysis Starting with the pressure at point 1 in the natural gas pipeline, and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives

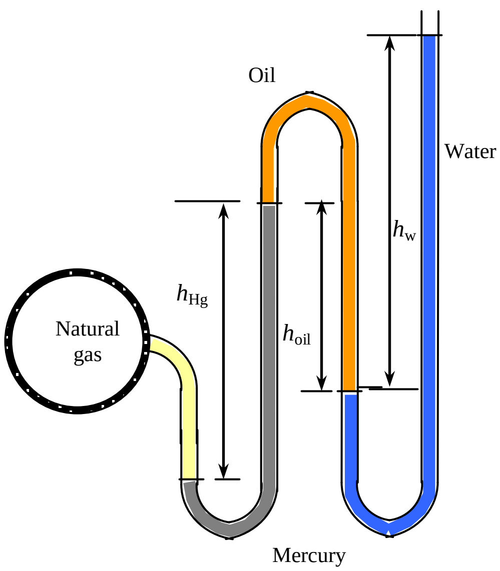
$$P_1 - \rho_{Hg} g h_{Hg} + \rho_{oil} g h_{oil} - \rho_{water} g h_{water} = P_{atm}$$

Solving for P_1,

$$P_1 = P_{atm} + \rho_{Hg} g h_{Hg} + \rho_{water} g h_1 - \rho_{oil} g h_{oil}$$

Substituting,

$$P_1 = 14.2 \text{psia} + (32.2 \text{ ft/s}^2)[(848.6 \text{lbm/ft}^3)(6/12 \text{ ft}) + (62.4 \text{lbm/ft}^3)(27/12 \text{ ft})$$
$$- (43.1 \text{lbm/ft}^3)(15/12 \text{ ft})]\left(\frac{1 \text{lbf}}{32.2 \text{lbm} \cdot \text{ft/s}^2}\right)\left(\frac{1 \text{ft}^2}{144 \text{in}^2}\right)$$
$$= \mathbf{17.7 psia}$$



Discussion Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

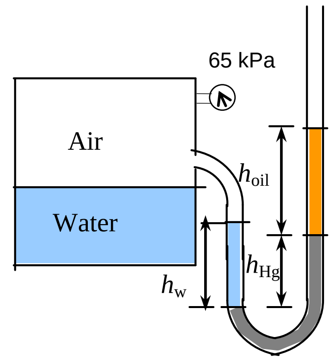
3-49

Solution The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

Properties We take the density of water to be ρ_w =1000 kg/m^3. The specific gravities of oil and mercury are given to be 0.72 and 13.6, respectively.

Analysis Starting with the pressure of air in the tank (point 1), and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho g h$ terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives



$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} = P_{atm}$$

Rearranging,

$$P_1 - P_{atm} = \rho_{oil} g h_{oil} + \rho_{Hg} g h_{Hg} - \rho_w g h_w$$

or,

$$\frac{P_{1,gage}}{\rho_w g} = \rho_{s,oil} h_{oil} + \rho_{s,Hg} h_{Hg} - h_w$$

Substituting,

$$\left(\frac{65\text{ kPa}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)}\right)\left(\frac{1000\ \text{ kg}\cdot\text{m/s}^2}{1\text{ kPa.}\cdot\text{m}^2}\right) = 0.72\times(0.75\text{ m}) + 13.6\times h_{Hg} - 0.3\text{ m}$$

Solving for h_{Hg} gives h_{Hg} = **0.47 m**. Therefore, the differential height of the mercury column must be 47 cm.

Discussion Double instrumentation like this allows one to verify the measurement of one of the instruments by the measurement of another instrument.

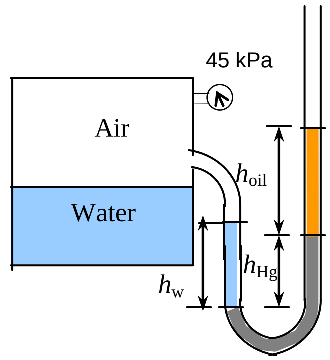
3-50

Solution The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

Assumptions The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

Properties We take the density of water to be ρ_w =1000 kg/m^3. The specific gravities of oil and mercury are given to be 0.72 and 13.6, respectively.

Analysis Starting with the pressure of air in the tank (point 1), and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives



$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} = P_{atm}$$

Rearranging,

$$P_1 - P_{atm} = \rho_{oil} g h_{oil} + \rho_{Hg} g h_{Hg} - \rho_w g h_w$$

or,

$$\frac{P_{1,gage}}{\rho_w g} = SG_{oil} h_{oil} + SG_{Hg} h_{Hg} - h_w$$

Substituting,

$$\frac{45\,\text{kPa}}{(1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\,\text{kPa.}\cdot\text{m}^2}\right) = 0.72\times(0.75\,\text{m}) + 13.6\times h_{Hg} - 0.3\,\text{m}$$

Solving for h_{Hg} gives h_{Hg} = **0.32 m**. Therefore, **the differential height of the mercury column must be 32 cm**.

Discussion Double instrumentation like this allows one to verify the measurement of one of the instruments by the measurement of another instrument.

3-51

Solution A load on a hydraulic lift is to be raised by pouring oil from a thin tube. The height of oil in the tube required in order to raise that weight is to be determined.

Assumptions **1** The cylinders of the lift are vertical. **2** There are no leaks. **3** Atmospheric pressure act on both sides, and thus it can be disregarded.

Properties The density of oil is given to be ρ =780 kg/m^3.

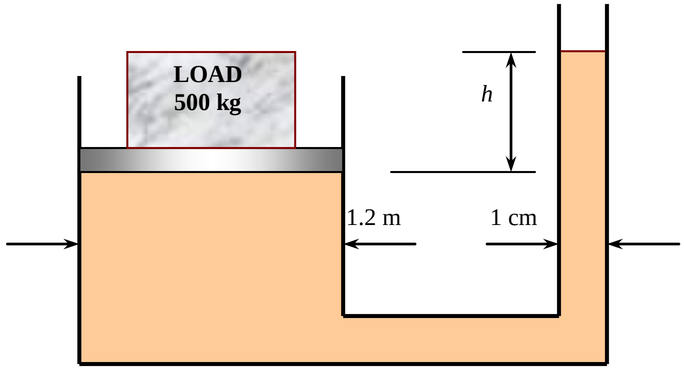
Analysis Noting that pressure is force per unit area, the gage pressure in the fluid under the load is simply the ratio of the weight to the area of the lift,

$$P_{\text{gage}} = \frac{W}{A} = \frac{mg}{\pi D^2/4} = \frac{(500\text{ kg})(9.81\text{ m/s}^2)}{\pi(1.20\text{ m})^2/4}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 4.34\text{ kN/m}^2 = 4.34\text{ kPa}$$

The required oil height that will cause 4.34 kPa of pressure rise is

$$P_{\text{gage}} = \rho g h \quad \rightarrow \quad h = \frac{P_{\text{gage}}}{\rho g} = \frac{4.34\text{ kN/m}^2}{(780\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{kN/m}^2}\right) = \mathbf{0.567m}$$

Therefore, a 500 kg load can be raised by this hydraulic lift by simply raising the oil level in the tube by 56.7 cm.

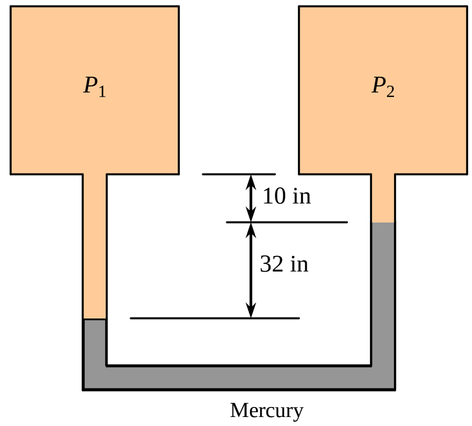


Discussion Note that large weights can be raised by little effort in hydraulic lift by making use of Pascal’s principle.

3-52E

Solution Two oil tanks are connected to each other through a mercury manometer. For a given differential height, the pressure difference between the two tanks is to be determined.

Assumptions **1** Both the oil and mercury are incompressible fluids. **2** The oils in both tanks have the same density.



Properties The densities of oil and mercury are given to be ρ_{oil} = 45 lbm/ft^3 and ρ_{Hg} = 848 lbm/ft^3.

Analysis Starting with the pressure at the bottom of tank 1 (where pressure is P_1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the bottom of tank 2 (where pressure is P_2) gives

$$P_1 + \rho_{\text{oil}} g(h_1 + h_2) - \rho_{\text{Hg}} g h_2 - \rho_{\text{oil}} g h_1 = P_2$$

where h_1 = 10 in and h_2 = 32 in. Rearranging and simplifying,

$$P_1 - P_2 = \rho_{\text{Hg}} g h_2 - \rho_{\text{oil}} g h_2 = (\rho_{\text{Hg}} - \rho_{\text{oil}}) g h_2$$

Substituting,

$$\Delta P = P_1 - P_2 = (848\text{-}\ 45\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(32/12\text{ft})\left(\frac{1\text{lbf}}{32.2\text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\text{ft}^2}{144\text{in}^2}\right) = \mathbf{14.9\ psia}$$

Therefore, the pressure in the left oil tank is 14.9 psia higher than the pressure in the right oil tank.

Discussion Note that large pressure differences can be measured conveniently by mercury manometers. If a water manometer were used in this case, the differential height would be over 30 ft.

3-53

Solution The standard atmospheric pressure is expressed in terms of mercury, water, and glycerin columns.

Assumptions The densities of fluids are constant.

Properties The specific gravities are given to be SG = 13.6 for mercury, SG = 1.0 for water, and SG = 1.26 for glycerin. The standard density of water is 1000 kg/m^3, and the standard atmospheric pressure is 101,325 Pa.

Analysis The atmospheric pressure is expressed in terms of a fluid column height as

$$P_{atm} = \rho gh = SG\rho_w gh \quad \rightarrow \quad h = \frac{P_{atm}}{SG\rho_w g}$$

Substituting,

(*a*) Mercury: $$h = \frac{P_{\text{atm}}}{\text{SG}\rho_w g} = \frac{101{,}325\ \text{N/m}^2}{13.6(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N/m}^2}\right) = \mathbf{0.759\ m}$$

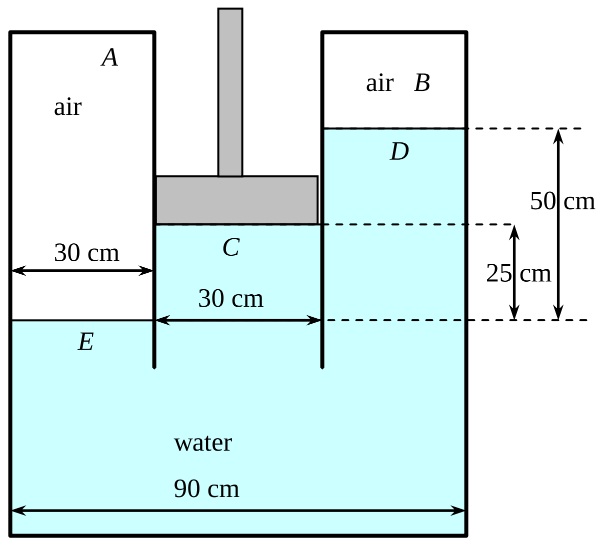
(*b*) Water: $$h = \frac{P_{\text{atm}}}{\text{SG}\rho_w g} = \frac{101{,}325\ \text{N/m}^2}{1(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N/m}^2}\right) = \mathbf{10.3\ m}$$

(*c*) Glycerin: $$h = \frac{P_{\text{atm}}}{\text{SG}\rho_w g} = \frac{101{,}325\ \text{N/m}^2}{1.26(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N/m}^2}\right) = \mathbf{8.20\ m}$$

Discussion Using water or glycerin to measure atmospheric pressure requires very long vertical tubes (over 10 m for water), which is not practical. This explains why mercury is used instead of water or a light fluid.

3-54

Solution Two chambers with the same fluid at their base are separated by a piston. The gage pressure in each air chamber is to be determined.



Assumptions **1** Water is an incompressible substance. **2** The variation of pressure with elevation in each air chamber is negligible because of the low density of air.

Properties We take the density of water to be ρ =1000 kg/m^3.

Analysis The piston is in equilibrium, and thus the net force acting on the piston must be zero. A vertical force balance on the piston involves the pressure force exerted by water on the piston face, the atmospheric pressure force, and the piston weight, and yields

$$P_C A_{\text{piston}} = P_{\text{atm}} A_{\text{piston}} + W_{\text{piston}} \quad \rightarrow \quad P_C = P_{\text{atm}} + \frac{W_{\text{piston}}}{A_{\text{piston}}}$$

The pressure at the bottom of each air chamber is determined from the hydrostatic pressure relation to be

$$P_{\text{air A}} = P_E = P_C + \rho g \overline{CE} = P_{\text{atm}} + \frac{W_{\text{piston}}}{A_{\text{piston}}} + \rho g \overline{CE} \quad \rightarrow \quad P_{\text{air A, gage}} = \frac{W_{\text{piston}}}{A_{\text{piston}}} + \rho g \overline{CE}$$

$$P_{\text{air B}} = P_D = P_C - \rho g \overline{CD} = P_{\text{atm}} + \frac{W_{\text{piston}}}{A_{\text{piston}}} - \rho g \overline{CD} \quad \rightarrow \quad P_{\text{air B, gage}} = \frac{W_{\text{piston}}}{A_{\text{piston}}} - \rho g \overline{CD}$$

Substituting,

$$P_{\text{air A, gage}} = \frac{25\text{ N}}{\pi(0.3\text{ m})^2/4} + (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.25\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 2806\text{ N/m}^2 = \mathbf{2.81\ kPa}$$

$$P_{\text{air B, gage}} = \frac{25\text{ N}}{\pi(0.3\text{ m})^2/4} - (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.25\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = -2099\text{ N/m}^2 = \mathbf{-2.10\ kPa}$$

Discussion Note that there is a vacuum of about 2 kPa in tank B which pulls the water up.

3-55

Solution A double-fluid manometer attached to an air pipe is considered. The specific gravity of one fluid is known, and the specific gravity of the other fluid is to be determined.

Assumptions **1** Densities of liquids are constant. **2** The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

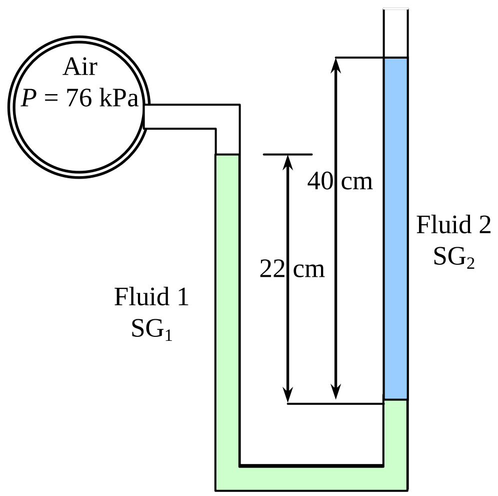
Properties The specific gravity of one fluid is given to be 13.55. We take the standard density of water to be 1000 kg/m^3.

Analysis Starting with the pressure of air in the tank, and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} give

$$P_{air} + \rho_1 gh_1 - \rho_2 gh_2 = P_{atm} \quad \rightarrow \quad P_{air} - P_{atm} = SG_2 \rho_w gh_2 - SG_1 \rho_w gh_1$$

Rearranging and solving for SG_2,

$$SG_2 = SG_1 \frac{h_1}{h_2} + \frac{P_{air} - P_{atm}}{\rho_w gh_2} = 13.55 \frac{0.22\text{ m}}{0.40\text{ m}} + \left(\frac{(76-100)\text{ kPa}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.40\text{ m})} \right) \left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kPa}\cdot\text{m}^2} \right) = 1.3363 \cong \mathbf{1.34}$$



Discussion Note that the right fluid column is higher than the left, and this would imply above atmospheric pressure in the pipe for a single-fluid manometer.

3-56

Solution The pressure difference between two pipes is measured by a double-fluid manometer. For given fluid heights and specific gravities, the pressure difference between the pipes is to be calculated.

Assumptions All the liquids are incompressible.

Properties The specific gravities are given to be 13.5 for mercury, 1.26 for glycerin, and 0.88 for oil. We take the standard density of water to be ρ_w=1000 kg/m^3.

Analysis Starting with the pressure in the water pipe (point A) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the oil pipe (point B), and setting the result equal to P_B give

$$P_A + \rho_{\mathrm{w}} gh_w + \rho_{\mathrm{Hg}} gh_{\mathrm{Hg}} - \rho_{\mathrm{gly}} gh_{\mathrm{gly}} + \rho_{\mathrm{pil}} gh_{\mathrm{oil}} = P_B$$

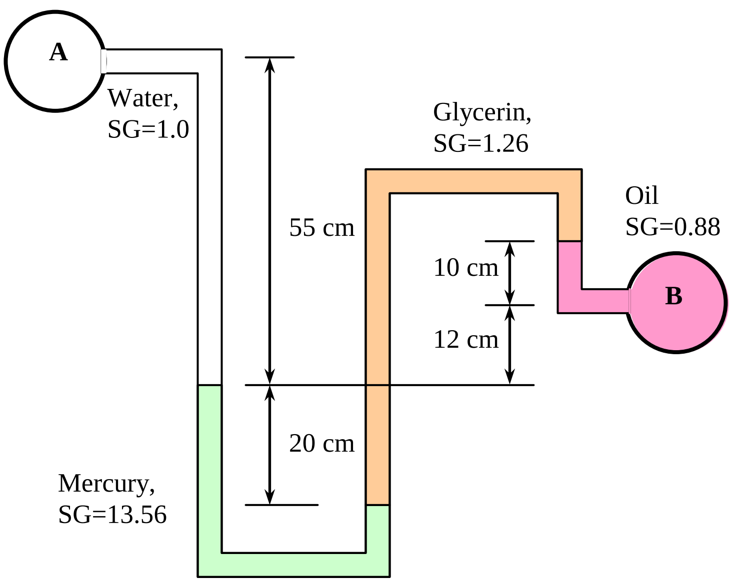
Rearranging and using the definition of specific gravity,

$$\begin{aligned} P_B - P_A &= \mathrm{SG}_w \rho_w gh_w + \mathrm{SG}_{Hg} \rho_w gh_{\mathrm{Hg}} - \mathrm{SG}_{gly} \rho_w gh_{\mathrm{gly}} + \mathrm{SG}_{oil} \rho_w gh_{\mathrm{oil}} \\ &= g\rho_w \left(\mathrm{SG}_w h_w + \mathrm{SG}_{Hg} h_{\mathrm{Hg}} - \mathrm{SG}_{gly} h_{\mathrm{gly}} + \mathrm{SG}_{oil} h_{\mathrm{oil}} \right) \end{aligned}$$

Substituting,

$$\begin{aligned} P_B - P_A &= (9.81\,\mathrm{m/s^2})(1000\,\mathrm{kg/m^3})[1(0.55\,\mathrm{m}) + 13.5(0.2\,\mathrm{m}) - 1.26(0.42\,\mathrm{m}) + 0.88(0.1\,\mathrm{m})]\left(\frac{1\,\mathrm{kN}}{1000\,\mathrm{kg \cdot m/s^2}} \right) \\ &= 27.6\,\mathrm{kN/m^2} = \mathbf{27.6\,kPa} \end{aligned}$$

Therefore, the pressure in the oil pipe is 27.6 kPa higher than the pressure in the water pipe.



Discussion Using a manometer between two pipes is not recommended unless the pressures in the two pipes are relatively constant. Otherwise, an over-rise of pressure in one pipe can push the manometer fluid into the other pipe, creating a short circuit.

3-57

Solution The fluid levels in a multi-fluid U-tube manometer change as a result of a pressure drop in the trapped air space. For a given pressure drop and brine level change, the area ratio is to be determined.

Assumptions **1** All the liquids are incompressible. **2** Pressure in the brine pipe remains constant. **3** The variation of pressure in the trapped air space is negligible.

Properties The specific gravities are given to be 13.56 for mercury and 1.1 for brine. We take the standard density of water to be ρ_w =1000 kg/m^3.

Analysis It is clear from the problem statement and the figure that the brine pressure is much higher than the air pressure, and when the air pressure drops by 0.9 kPa, the pressure difference between the brine and the air space also increases by the same amount. Starting with the air pressure (point A) and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the brine pipe (point B), and setting the result equal to P_B before and after the pressure change of air give

Before: $P_{A1} + \rho_{\mathrm{w}} g h_w + \rho_{\mathrm{Hg}} g h_{\mathrm{Hg},1} - \rho_{\mathrm{br}} g h_{\mathrm{br},1} = P_B$

After: $P_{A2} + \rho_{\mathrm{w}} g h_w + \rho_{\mathrm{Hg}} g h_{\mathrm{Hg},2} - \rho_{\mathrm{br}} g h_{\mathrm{br},2} = P_B$

Subtracting,

$$P_{A2} - P_{A1} + \rho_{\mathrm{Hg}} g \Delta h_{\mathrm{Hg}} - \rho_{\mathrm{br}} g \Delta h_{\mathrm{br}} = 0 \quad \rightarrow \quad \frac{P_{A1} - P_{A2}}{\rho_w g} = \mathrm{SG}_{\mathrm{Hg}} \Delta h_{\mathrm{Hg}} - \mathrm{SG}_{\mathrm{br}} \Delta h_{\mathrm{br}} = 0 \qquad (1)$$

where Δh_{Hg} and Δh_{br} are the changes in the differential mercury and brine column heights, respectively, due to the drop in air pressure. Both of these are positive quantities since as the mercury-brine interface drops, the differential fluid heights for both mercury and brine increase. Noting also that the volume of mercury is constant, we have $A_1 \Delta h_{\mathrm{Hg,left}} = A_2 \Delta h_{\mathrm{Hg,right}}$ and

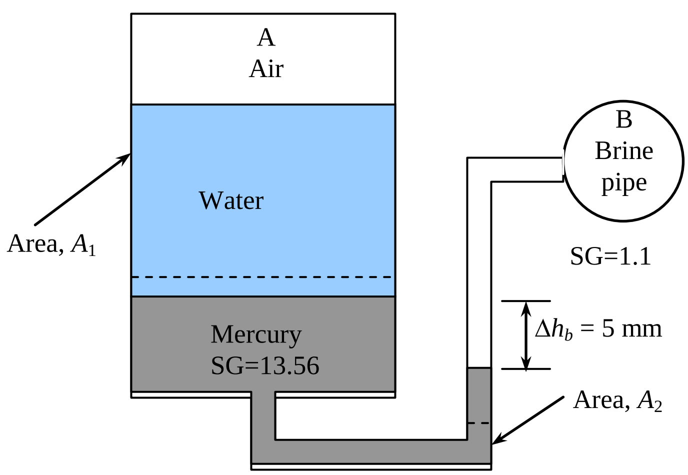
$$P_{A2} - P_{A1} = -0.9\ \mathrm{kPa} = -900\ \mathrm{N/m^2} = -900\ \mathrm{kg/m \cdot s^2}$$
$$\Delta h_{\mathrm{br}} = 0.005\ \mathrm{m}$$
$$\Delta h_{\mathrm{Hg}} = \Delta h_{\mathrm{Hg,right}} + \Delta h_{\mathrm{Hg,left}} = \Delta h_{\mathrm{br}} + \Delta h_{\mathrm{br}} A_2/\mathrm{A}_1 = \Delta h_{\mathrm{br}}(1 + A_2/\mathrm{A}_1)$$

Substituting,

$$\frac{900\ \mathrm{kg/m \cdot s^2}}{(1000\ \mathrm{kg/m^3})(9.81\ \mathrm{m/s^2})} = [13.56 \times 0.005(1 + A_2/A_1) - 1.1 \times 0.005]\ \mathrm{m}$$

It gives

A_2/A_1 = **0.434**



Discussion In addition to the equations of hydrostatics, we also utilize conservation of mass in this problem.

3-58

Solution Two water tanks are connected to each other through a mercury manometer with inclined tubes. For a given pressure difference between the two tanks, the parameters a and θ are to be determined.

Assumptions Both water and mercury are incompressible liquids.

Properties The specific gravity of mercury is given to be 13.6. We take the standard density of water to be ρ_w =1000 kg/m^3.

Analysis Starting with the pressure in the tank A and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach tank B, and setting the result equal to P_B give

$$P_A + \rho_w ga + \rho_{Hg} g2a - \rho_w ga = P_B \quad \rightarrow \quad 2\rho_{Hg} ga = P_B - P_A$$

Rearranging and substituting the known values,

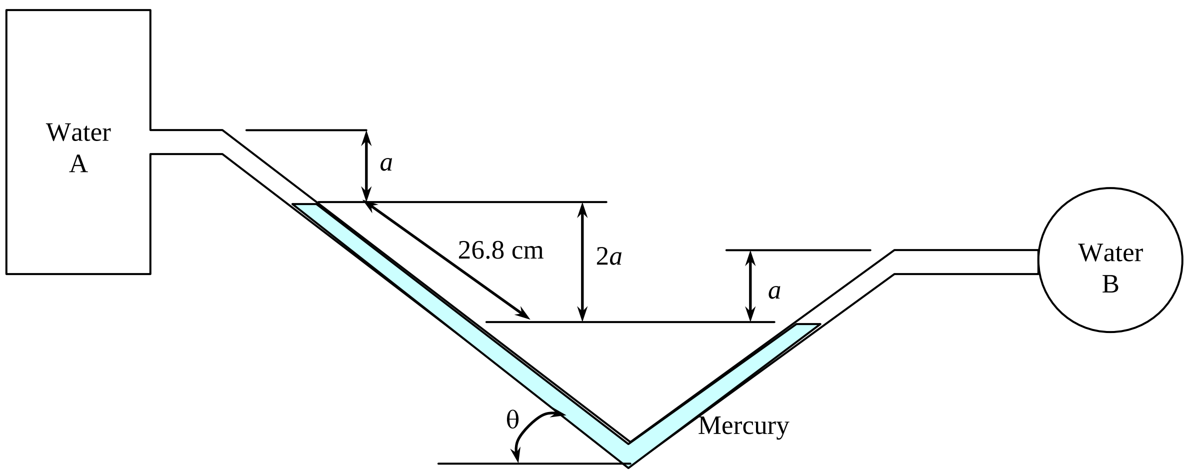
$$a = \frac{P_B - P_A}{2\rho_{Hg} g} = \frac{20 \text{ kN/m}^2}{2(13.6)(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1000 \text{ kg}\cdot\text{m/s}^2}{1 \text{ kN}}\right) = 0.0750 \text{ m} = \mathbf{7.50 \text{ cm}}$$

From geometric considerations,

$$26.8 \sin\theta = 2a \quad \text{(cm)}$$

Therefore,

$$\sin\theta = \frac{2a}{26.8} = \frac{2 \times 7.50}{26.8} = 0.560 \quad \rightarrow \quad \theta = \mathbf{34.0°}$$



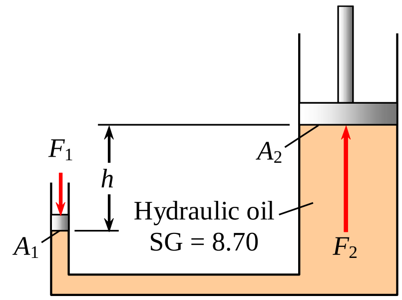
Discussion Note that vertical distances are used in manometer analysis. Horizontal distances are of no consequence.

3-59

Solution We are to determine the force required to lift a car with a hydraulic jack at two different elevations.

Assumptions **1** The oil is incompressible. **2** The system is at rest during the analysis (hydrostatics).

Analysis (*a*) When h = 0, the pressure at the bottom of each piston must be the same. Thus,



$$P_1 = \frac{F_1}{A_1} = P_2 = \frac{F_2}{A_2} \longrightarrow F_1 = F_2\frac{A_1}{A_2} = (13{,}000\text{ N})\frac{0.8\,\text{cm}^2}{0.0400\,\text{m}^2}\left(\frac{1\text{ m}}{100\text{ cm}}\right)^2 = \mathbf{26.0 N}$$

At the beginning, when h = 0, the required force is thus F_1 = **26.0 N**.

(*b*) When $h \neq 0$, the hydrostatic pressure due to the elevation difference must be taken into account, namely,

$$P_1 = \frac{F_1}{A_1} = P_2 + \rho g h = \frac{F_2}{A_2} + \rho g h$$

$$F_1 = F_2\frac{A_1}{A_2} + \rho g h A_1$$

$$= (13{,}000\text{ N})\frac{0.00008\,\text{m}^2}{0.04\,\text{m}^2} + (870\text{ kg/m}^3)(9.807\text{ m/s}^2)(2.00\text{ m})(0.00008\,\text{m}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= \mathbf{27.4 N}$$

Thus, after the car has been raised 2 meters, the required force is 27.4 N.

Comparing the two results, **it takes more force to keep the car elevated** than it does to hold it at h = 0. This makes sense physically because **the elevation difference generates a higher pressure (and thus a higher required force) at the lower piston due to hydrostatics**.

Discussion When h = 0, the specific gravity (or density) of the hydraulic fluid does not enter the calculation – the problem simplifies to setting the two pressure equal. However, when $h \neq 0$, there is a hydrostatic head and therefore the density of the fluid enters the calculation.

Fluid Statics: Hydrostatic Forces on Plane and Curved Surfaces

3-60C

Solution We are to define resultant force and center of pressure.

Analysis The *resultant hydrostatic force* acting on a submerged surface is the **resultant of the pressure forces acting on the surface**. The **point of application of this resultant force** is called the *center of pressure*.

Discussion The center of pressure is generally not at the center of the body, due to hydrostatic pressure variation.

3-61C

Solution We are to examine a claim about hydrostatic force.

Analysis **Yes**, because the magnitude of the resultant force acting on a plane surface of a completely submerged body in a homogeneous fluid is equal to the product of the pressure P_C at the centroid of the surface and the area A of the surface. The pressure at the centroid of the surface is $P_C = P_0 + \rho g h_C$ where h_C is the vertical distance of the centroid from the free surface of the liquid.

Discussion We have assumed that we also know the pressure at the liquid surface.

3-62C

Solution We are to consider the effect of plate rotation on the hydrostatic force on the plate surface.

Analysis There will be **no change** on the hydrostatic force acting on the top surface of this submerged horizontal flat plate as a result of this rotation since the magnitude of the resultant force acting on a plane surface of a completely submerged body in a homogeneous fluid is equal to the product of the pressure P_C at the centroid of the surface and the area A of the surface.

Discussion If the rotation were not around the centroid, there *would* be a change in the force.

3-63C

Solution We are to explain why dams are bigger at the bottom than at the top.

Analysis Dams are built much thicker at the bottom because **the pressure force increases with depth, and the bottom part of dams are subjected to largest forces**.

Discussion Dam construction requires an enormous amount of concrete, so tapering the dam in this way saves a lot of concrete, and therefore a lot of money.

3-64C

Solution We are to explain how to determine the horizontal component of hydrostatic force on a curved surface.

Analysis The horizontal component of the hydrostatic force acting on a curved surface is equal (in both magnitude and the line of action) to **the hydrostatic force acting on the vertical projection of the curved surface**.

Discussion We could also integrate pressure along the surface, but the method discussed here is much simpler and yields the same answer.

3-65C

Solution We are to explain how to determine the vertical component of hydrostatic force on a curved surface.

Analysis The vertical component of the hydrostatic force acting on a curved surface is equal to **the hydrostatic force acting on the horizontal projection of the curved surface**, **plus** (minus, if acting in the opposite direction) **the weight of the fluid block**.

Discussion We could also integrate pressure along the surface, but the method discussed here is much simpler and yields the same answer.

3-66C

Solution We are to explain how to determine the line of action on a circular surface.

Analysis The resultant hydrostatic force acting on a circular surface always passes through **the center of the circle** since the pressure forces are normal to the surface, and all lines normal to the surface of a circle pass through the center of the circle. Thus the pressure forces form a concurrent force system at the center, which can be reduced to a single equivalent force at that point. If the magnitudes of the horizontal and vertical components of the resultant hydrostatic force are known, the tangent of the angle the resultant hydrostatic force makes with the horizontal is $\boxed{\tan\alpha = F_V / F_H}$.

Discussion This fact makes analysis of circular-shaped surfaces simple. There is no corresponding simplification for shapes other than circular, unfortunately.

3-67

Solution A car is submerged in water. The hydrostatic force on the door and its line of action are to be determined for the cases of the car containing atmospheric air and the car is filled with water.

Assumptions **1** The bottom surface of the lake is horizontal. **2** The door can be approximated as a vertical rectangular plate. **3** The pressure in the car remains at atmospheric value since there is no water leaking in, and thus no compression of the air inside. Therefore, we can ignore the atmospheric pressure in calculations since it acts on both sides of the door.

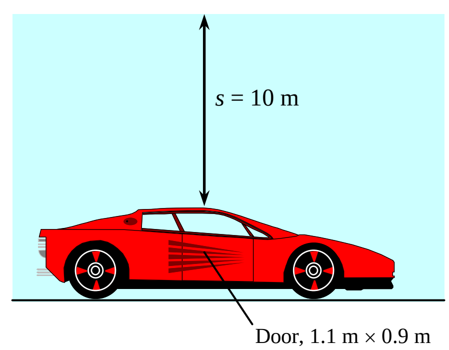
Properties We take the density of lake water to be 1000 kg/m^3 throughout.

Analysis (*a*) When the car is well-sealed and thus the pressure inside the car is the atmospheric pressure, the average pressure on the outer surface of the door is the pressure at the centroid (midpoint) of the surface, and is determined to be

$$P_{ave} = P_C = \rho g h_C = \rho g (s + b/2)$$

$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(10 + 1.1/2\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 103.5\ \text{kN/m}^2$$



Then the resultant hydrostatic force on the door becomes

$$F_R = P_{ave}A = (103.5\ \text{kN/m}^2)(0.9\ \text{m} \times 1.1\ \text{m}) = \mathbf{102.5\ kN}$$

The pressure center is directly under the midpoint of the plate, and its distance from the surface of the lake is determined to be

$$y_P = s + \frac{b}{2} + \frac{b^2}{12(s + b/2)} = 10 + \frac{1.1}{2} + \frac{1.1^2}{12(10 + 1.1/2)} = \mathbf{10.56\ m}$$

(*b*) When the car is filled with water, the net force normal to the surface of the door is **zero** since the pressure on both sides of the door will be the same.

Discussion Note that it is impossible for a person to open the door of the car when it is filled with atmospheric air. But it takes little effort to open the door when car is filled with water, because then the pressure on each side of the door is the same.

3-68E

Solution The height of a water reservoir is controlled by a cylindrical gate hinged to the reservoir. The hydrostatic force on the cylinder and the weight of the cylinder per ft length are to be determined.

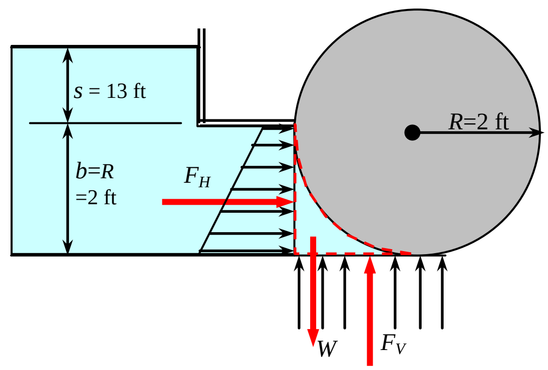
Assumptions **1** The hinge is frictionless. **2** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 62.4 lbm/ft^3 throughout.

Analysis (*a*) We consider the free body diagram of the liquid block enclosed by the circular surface of the cylinder and its vertical and horizontal projections. The hydrostatic forces acting on the vertical and horizontal plane surfaces as well as the weight of the liquid block per ft length of the cylinder are:

Horizontal force on vertical surface:

$$F_H = F_x = P_{ave}A = \rho g h_C A = \rho g(s + R/2)A$$
$$= (62.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(13 + 2/2\ \text{ft})(2\ \text{ft} \times 1\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)$$
$$= 1747\ \text{lbf}$$



Vertical force on horizontal surface (*upward*):

$$F_y = P_{avg}A = \rho g h_C A = \rho g h_{bottom} A$$
$$= \left(62.4\ \text{lbm/ft}^3\right)\left(32.2\ \text{ft/s}^2\right)\left(15\ \text{ft}\right)\left(2\ \text{ft} \times 1\ \text{ft}\right)\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)$$
$$= 1872\ \text{lbf}$$

Weight of fluid block per ft length *(downward):*

$$W = mg = \rho g \boldsymbol{V} = \rho g(R^2 - \pi R^2/4)(1\ \text{ft}) = \rho g R^2(1 - \pi/4)(1\ \text{ft})$$
$$= (62.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(2\ \text{ft})^2(1 - \pi/4)(1\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)$$
$$= 54\ \text{lbf}$$

Therefore, the net upward vertical force is

$$F_V = F_y - W = 1872 - 54 = 1818\ \text{lbf}$$

Then the magnitude and direction of the hydrostatic force acting on the cylindrical surface become

$$F_R = \sqrt{F_H^2 + F_V^2} = \sqrt{1747^2 + 1818^2} = 2521\ \text{lbf} \cong \mathbf{2520\ lbf}$$

$$\tan\theta = \frac{F_V}{F_H} = \frac{1818\ \text{lbf}}{1747\ \text{lbf}} = 1.041 \quad \rightarrow \quad \theta = 46.1°$$

Therefore, the magnitude of the hydrostatic force acting on the cylinder is 2521 lbf per ft length of the cylinder, and its line of action passes through the center of the cylinder making an angle 46.1° upwards from the horizontal.

(*b*) When the water level is 15-ft high, the gate opens and the reaction force at the bottom of the cylinder becomes zero. Then the forces other than those at the hinge acting on the cylinder are its weight, acting through the center, and the hydrostatic force exerted by water. Taking a moment about the point *A* where the hinge is and equating it to zero gives

$$F_R R\sin\theta - W_{cyl}R = 0 \quad \rightarrow \quad W_{cyl} = F_R\sin\theta = (2521\ \text{lbf})\sin 46.1° = 1817\ \text{lbf} \cong \mathbf{1820\ lbf} \quad \text{(per ft)}$$

Discussion The weight of the cylinder per ft length is determined to be 1820 lbf, which corresponds to a mass of 1820 lbm, and to a density of 145 lbm/ft^3 for the material of the cylinder.

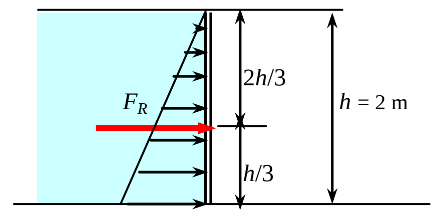
3-69

Solution An above the ground swimming pool is filled with water. The hydrostatic force on each wall and the distance of the line of action from the ground are to be determined, and the effect of doubling the wall height on the hydrostatic force is to be assessed.

Assumptions Atmospheric pressure acts on both sides of the wall of the pool, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and is determined to be



$$P_{avg} = P_C = \rho g h_C = \rho g (h/2)$$
$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(2/2\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)$$
$$= 9810\ \text{N/m}^2$$

Then the resultant hydrostatic force on each wall becomes

$$F_R = P_{avg}A = (9810\ \text{N/m}^2)(8\ \text{m}\times 2\ \text{m}) = 156{,}960\ \text{N} = \mathbf{157\ kN}$$

The line of action of the force passes through the pressure center, which is $2h/3$ from the free surface and $h/3$ from the bottom of the pool. Therefore, the distance of the line of action from the ground is

$$y_P = \frac{h}{3} = \frac{2}{3} = \mathbf{0.667\ m} \quad \text{(from the bottom)}$$

If the height of the walls of the pool is doubled, the hydrostatic force **quadruples** since

$$F_R = \rho g h_C A = \rho g (h/2)(h\times w) = \rho g w h^2/2$$

and thus the hydrostatic force is proportional to the square of the wall height, h^2.

Discussion This is one reason why above-ground swimming pools are not very deep, whereas in-ground swimming pools can be quite deep.

3-70E

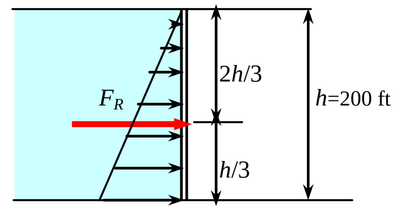
Solution A dam is filled to capacity. The total hydrostatic force on the dam, and the pressures at the top and the bottom are to be determined.

Assumptions Atmospheric pressure acts on both sides of the dam, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 62.4 lbm/ft^3 throughout.

Analysis The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and is determined to be

$$P_{\text{avg}} = \rho g h_C = \rho g (h/2)$$
$$= (62.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(200/2\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)$$
$$= 6240\ \text{lbf/ft}^2$$



Then the resultant hydrostatic force acting on the dam becomes

$$F_R = P_{\text{ave}} A = (6240\ \text{lbf/ft}^2)(200\ \text{ft} \times 1200\ \text{ft}) = \mathbf{1.50 \times 10^9\ lbf}$$

Resultant force per unit area is pressure, and its value at the top and the bottom of the dam becomes

$$P_{\text{top}} = \rho g h_{\text{top}} = \mathbf{0\ lbf/ft^2}$$

$$P_{\text{bottom}} = \rho g h_{\text{bottom}} = (62.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(200\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = 12{,}480\ \text{lbf/ft}^2 \cong \mathbf{12{,}500\ lbf/ft^2}$$

Discussion The values above are gave pressures, of course. The gage pressure at the bottom of the dam is about 86.6 psig, or 101.4 psia, which is almost seven times greater than standard atmospheric pressure.

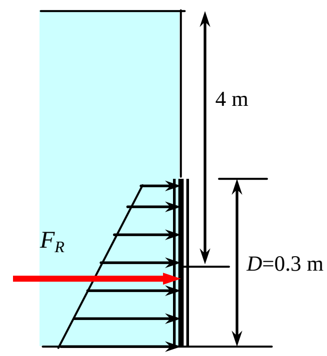
3-71

Solution A room in the lower level of a cruise ship is considered. The hydrostatic force acting on the window and the pressure center are to be determined.

Assumptions Atmospheric pressure acts on both sides of the window, and thus it can be ignored in calculations for convenience.

Properties The specific gravity of sea water is given to be 1.025, and thus its density is 1025 kg/m^3.

Analysis The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and is determined to be



$$P_{ave} = P_C = \rho g h_C = (1025\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(4\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = 40{,}221\ \text{N/m}^2$$

Then the resultant hydrostatic force on each wall becomes

$$F_R = P_{ave}A = P_{ave}[\pi D^2/4]$$
$$= (40{,}221\ \text{N/m}^2)[\pi(0.3\ \text{m})^2/4] = 2843\ \text{N} \cong \mathbf{2840} \quad \text{(three significant digit)}$$

The line of action of the force passes through the pressure center, whose vertical distance from the free surface is determined from

$$y_P = y_C + \frac{I_{xx,C}}{y_C A} = y_C + \frac{\pi R^4/4}{y_C \pi R^2} = y_C + \frac{R^2}{4y_C} = 4 + \frac{(0.15\ \text{m})^2}{4(5\ \text{m})} = \mathbf{4.001\ m}$$

Discussion For small surfaces deep in a liquid, the pressure center nearly coincides with the centroid of the surface. Here, in fact, to three significant digits in the final answer, the center of pressure and centroid are coincident. We give the answer to four significant digits to show that the center of pressure and the centroid are *not* coincident.

3-72

Solution The cross-section of a dam is a quarter-circle. The hydrostatic force on the dam and its line of action are to be determined.

Assumptions Atmospheric pressure acts on both sides of the dam, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.

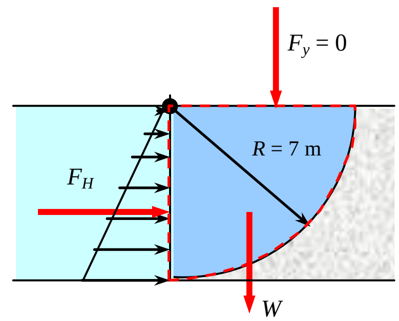
Analysis We consider the free body diagram of the liquid block enclosed by the circular surface of the dam and its vertical and horizontal projections. The hydrostatic forces acting on the vertical and horizontal plane surfaces as well as the weight of the liquid block are:

Horizontal force on vertical surface:

$$F_H = F_x = P_{avg}A = \rho g h_C A = \rho g(R/2)A$$

$$= (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(7/2\text{ m})(7\text{ m}\times 70\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 1.682\times 10^7\text{ N}$$



Vertical force on horizontal surface is zero since it coincides with the free surface of water. The weight of fluid block per m length is

$$F_V = W = \rho g \boldsymbol{V} = \rho g[w\times \pi R^2/4]$$

$$= (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)[(70\text{ m})\pi(7\text{ m})^2/4]\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 2.643\times 10^7\text{ N}$$

Then the magnitude and direction of the hydrostatic force acting on the surface of the dam become

$$F_R = \sqrt{F_H^2 + F_V^2} = \sqrt{(1.682\times 10^7\text{ N})^2 + (2.643\times 10^7\text{ N})^2} = \mathbf{3.13\times 10^7\ N}$$

$$\tan\theta = \frac{F_V}{F_H} = \frac{2.643\times 10^7\text{ N}}{1.682\times 10^7\text{ N}} = 1.571 \longrightarrow \theta = \mathbf{57.5°}$$

Therefore, the line of action of the hydrostatic force passes through the center of the curvature of the dam, making 57.5° downwards from the horizontal.

Discussion If the shape were not circular, it would be more difficult to determine the line of action.

3-73

Solution The force required to hold a gate at its location is to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties Specific gravities are given in the figure.

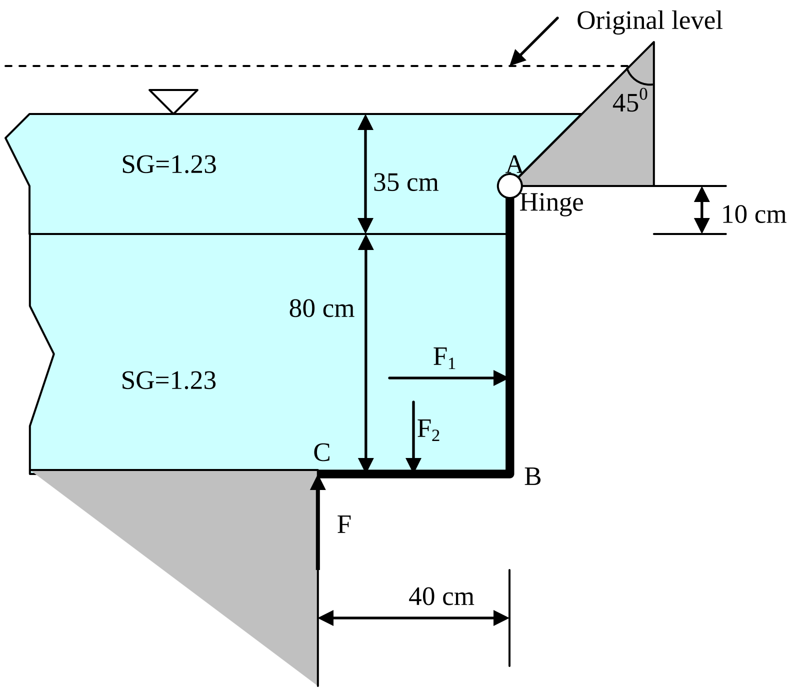
Analysis Since there are two different fluid layers it would be useful to convert one of them to another one to make the problem easier. The pressure at the interface is

$$p = 0.86 \times 9810 \times 0.5 = 4218.3 \ Pa$$

Now, the question is how much fluid from the second one can make the same pressure.

$$h_{SG=1.23} = \frac{p}{1.23 \times 9810} = \frac{4218.3}{12066.3} = 0.35 \ m = 35 \ cm$$

Therefore the system can be simplified as shown:



$$F_1 = \gamma h_{cg} A = 1.23 \times 9810 \times \left(\frac{0.8 + .010}{2} + 0.25 \right) \times (0.9 \times 2) = 15204 \ N$$

$$y_{cp} = y_{cg} + \frac{I_{xc}}{y_{cg} A} = 0.55 + \frac{2 \times 0.9^3 / 12}{0.55 \times (0.9 \times 2)} = 0.1227 \ m$$

$$F_2 = \gamma h A = 1.23 \times 9810 \times (0.80 + 0.35) \times (0.4 \times 2) = 11101 \ N$$

Take moment about hinge will give

$$F \times 0.4 - 15204 \times (1.15 - 0.1227) - 11101 \times \frac{0.4}{2} = 0$$

F = **17.8 kN**

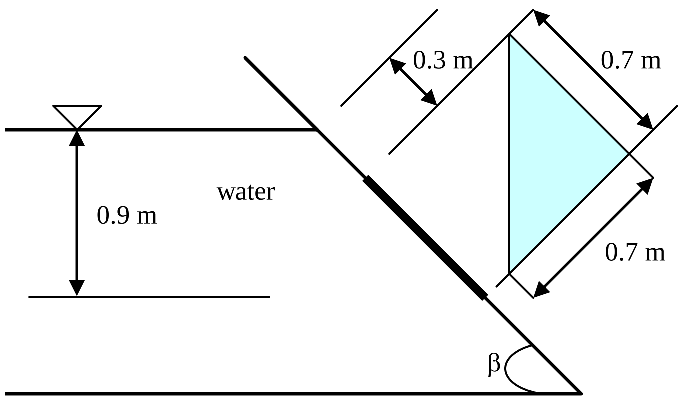
3-74

Solution The resulting force acting on a triangular gate and its line of action are to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis



We first determine the angle;

$$Sin(\beta) = \frac{0.9}{1} = 0.9, \ \ \beta = 64.16^o$$

$$F_R = \gamma h_{cg} A = 9810 \times \left[\left(0.3 + \frac{2}{3}0.7\right)\right] \times Sin(64.16) \times \frac{1}{2} 0.7 \times 0.7 = \mathbf{1658\,N}$$

In order to locate F_R on the gate x_{cp}, and y_{cp} must be found.

$$x_{cp} = x_{cg} + \frac{I_{xyc}}{y_{cg} A}$$

For simplicity, we can consider x axis to be passing through center of gravity of the gate, so that x_{cg}=0.

$$I_{xyc} = \frac{0.7 \times (0.7 - 2 \times 0) \times 0.7^2}{72} = 3.334 \times 10^{-3}\ m^4$$

$$y_{cg} = 0.3 + \frac{2}{3} 0.7 = 0.766\ m$$

$$A = 0.5 \times 0.7^2 = 0.245\ m^2$$

$$x_{cp} = 0 + \frac{3.334 \times 10^{-3}}{0.766 \times 0.245} = 1.776 \times 10^{-2}\ m = 1.77\ cm$$

$$I_{xc} = \frac{0.7 \times 0.7^3}{36} = 6.67 \times 10^{-3}\ m^4$$

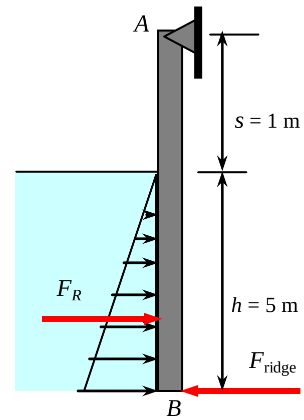
$$y_{cp} = 0.766 + \frac{6.67 \times 10^{-3}}{0.766 \times 0.245} = \mathbf{0.801 m}$$

3-75

Solution A rectangular plate hinged about a horizontal axis along its upper edge blocks a fresh water channel. The plate is restrained from opening by a fixed ridge at a point *B*. The force exerted to the plate by the ridge is to be determined.

Assumptions Atmospheric pressure acts on both sides of the plate, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.



Analysis The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and is determined to be

$$P_{ave} = P_C = \rho g h_C = \rho g(h/2)$$

$$= (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(5/2\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 24.53\text{ kN/m}^2$$

Then the resultant hydrostatic force on each wall becomes

$$F_R = P_{ave}A = (24.53\text{ kN/m}^2)(6\text{ m}\times 5\text{ m}) = 735.9\text{ m}$$

The line of action of the force passes through the pressure center, which is $2h/3$ from the free surface,

$$y_P = \frac{2h}{3} = \frac{2\times(5\text{ m})}{3} = 3.333\text{ m}$$

Taking the moment about point *A* and setting it equal to zero gives

$$\sum M_A = 0 \quad \rightarrow \quad F_R(s+y_P) = F_{\text{ridge}}\overline{AB}$$

Solving for F_{ridge} and substituting, the reaction force is determined to be

$$F_{\text{ridge}} = \frac{s+y_P}{\overline{AB}}F_R = \frac{(1+3.333)\text{ m}}{5\text{ m}}(735.9\text{ kN}) = \mathbf{638\text{ kN}}$$

Discussion The difference between F_R and F_{ridge} is the force acting on the hinge at point *A*.

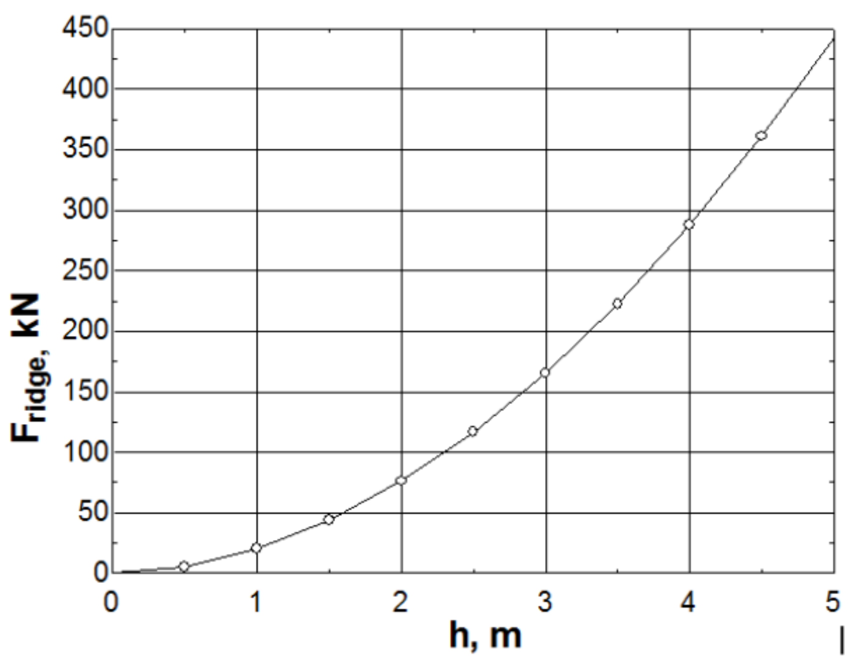
3-76

Solution The previous problem is reconsidered. The effect of water depth on the force exerted on the plate by the ridge as the water depth varies from 0 to 5 m in increments of 0.5 m is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
s=1"m"

w=5 "m"
A=w*h
P_ave=rho*g*h/2000 "kPa"
F_R=P_ave*A "kN"
y_p=2*h/3
F_ridge=(s+y_p)*F_R/(s+h)
```

Dept h, m	P_{ave}, kPa	F_R kN	y_p m	F_{ridge} kN
0.0	0	0.0	0.00	0
0.5	2.453	6.1	0.33	5
1.0	4.905	24.5	0.67	20
1.5	7.358	55.2	1.00	44
2.0	9.81	98.1	1.33	76
2.5	12.26	153.3	1.67	117
3.0	14.72	220.7	2.00	166
3.5	17.17	300.4	2.33	223
4.0	19.62	392.4	2.67	288
4.5	22.07	496.6	3.00	361
5.0	24.53	613.1	3.33	443



Discussion The force on the ridge does not increase linearly, as we may have suspected.

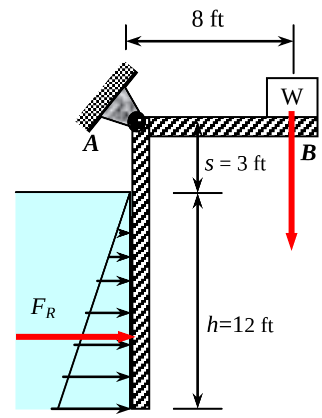
3-77E

Solution The flow of water from a reservoir is controlled by an L-shaped gate hinged at a point *A*. The required weight *W* for the gate to open at a specified water height is to be determined.

Assumptions **1** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience. **2** The weight of the gate is negligible.

Properties We take the density of water to be 62.4 lbm/ft^3 throughout.

Analysis The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and is determined to be



$$P_{\text{avg}} = \rho g h_C = \rho g (h/2)$$
$$= (62.4 \text{ lbm/ft}^3)(32.2 \text{ ft/s}^2)(12/2 \text{ ft})\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)$$
$$= 374.4 \text{ lbf/ft}^2$$

Then the resultant hydrostatic force acting on the dam becomes

$$F_R = P_{\text{avg}} A = (374.4 \text{ lbf/ft}^2)(12 \text{ ft} \times 5 \text{ ft}) = 22{,}464 \text{ lbf}$$

The line of action of the force passes through the pressure center, which is $2h/3$ from the free surface,

$$y_P = \frac{2h}{3} = \frac{2 \times (12 \text{ ft})}{3} = 8 \text{ ft}$$

Taking the moment about point *A* and setting it equal to zero gives

$$\sum M_A = 0 \quad \rightarrow \quad F_R(s + y_P) = W\overline{AB}$$

Solving for *W* and substituting, the required weight is determined to be

$$W = \frac{s + y_P}{\overline{AB}} F_R = \frac{(3+8) \text{ ft}}{8 \text{ ft}}(22{,}464 \text{ lbf}) = \mathbf{30{,}900 \text{ lbf}}$$

The corresponding mass is thus **30,900 lbm**.

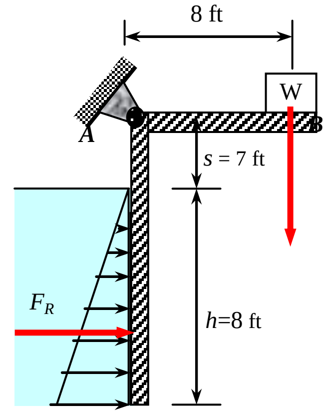
Discussion Note that the required weight is inversely proportional to the distance of the weight from the hinge.

3-78E

Solution The flow of water from a reservoir is controlled by an L-shaped gate hinged at a point *A*. The required weight *W* for the gate to open at a specified water height is to be determined.

Assumptions **1** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience. **2** The weight of the gate is negligible.

Properties We take the density of water to be 62.4 lbm/ft^3 throughout.



Analysis The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and is determined to be

$$P_{\text{avg}} = \rho g h_C = \rho g (h/2)$$
$$= (62.4 \text{ lbm/ft}^3)(32.2 \text{ ft/s}^2)(8/2 \text{ ft})\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)$$
$$= 249.6 \text{ lbf/ft}^2$$

Then the resultant hydrostatic force acting on the dam becomes

$$F_R = P_{\text{avg}} A = (249.6 \text{ lbf/ft}^2)(8 \text{ ft} \times 5 \text{ ft}) = 9984 \text{ lbf}$$

The line of action of the force passes through the pressure center, which is $2h/3$ from the free surface,

$$y_P = \frac{2h}{3} = \frac{2 \times (8 \text{ ft})}{3} = 5.333 \text{ ft}$$

Taking the moment about point *A* and setting it equal to zero gives

$$\sum M_A = 0 \quad \rightarrow \quad F_R(s + y_P) = W\overline{AB}$$

Solving for *W* and substituting, the required weight is determined to be

$$W = \frac{s + y_P}{\overline{AB}} F_R = \frac{(7 + 5.333) \text{ ft}}{8 \text{ ft}}(9984 \text{ lbf}) = 15{,}390 \text{ lbf} \cong \mathbf{15{,}400 \text{ lbf}}$$

Discussion Note that the required weight is inversely proportional to the distance of the weight from the hinge.

3-79

Solution Two parts of a water trough of semi-circular cross-section are held together by cables placed along the length of the trough. The tension T in each cable when the trough is full is to be determined.

Assumptions **1** Atmospheric pressure acts on both sides of the trough wall, and thus it can be ignored in calculations for convenience. **2** The weight of the trough is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.

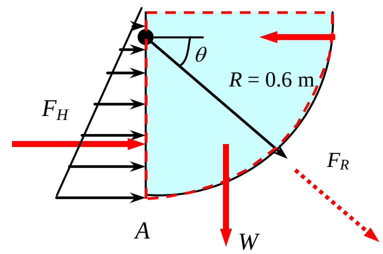
Analysis To expose the cable tension, we consider half of the trough whose cross-section is quarter-circle. The hydrostatic forces acting on the vertical and horizontal plane surfaces as well as the weight of the liquid block are:

Horizontal force on vertical surface:

$$F_H = F_x = P_{ave}A = \rho g h_C A = \rho g (R/2)A$$

$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.6/2\ \text{m})(0.6\ \text{m} \times 3\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 5297\ \text{N}$$



The vertical force on the horizontal surface is zero, since it coincides with the free surface of water. The weight of fluid block per 3-m length is

$$F_V = W = \rho g V = \rho g[w \times \pi R^2/4]$$

$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)[(3\ \text{m})\pi(0.6\ \text{m})^2/4]\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 8321\ \text{N}$$

Then the magnitude and direction of the hydrostatic force acting on the surface of the 3-m long section of the trough become

$$F_R = \sqrt{F_H^2 + F_V^2} = \sqrt{(5297\ \text{N})^2 + (8321\ \text{N})^2} = 9864\ \text{N}$$

$$\tan\theta = \frac{F_V}{F_H} = \frac{8321\ \text{N}}{5297\ \text{N}} = 1.571 \quad \rightarrow \quad \theta = 57.52°$$

Therefore, the line of action passes through the center of the curvature of the trough, making 57.52° downwards from the horizontal. Taking the moment about point A where the two parts are hinged and setting it equal to zero gives

$$\sum M_A = 0 \quad \rightarrow \quad F_R R \sin(90 - 57.52)° = \mathbf{T}R$$

Solving for T and substituting, the tension in the cable is determined to be

$$\mathbf{T} = F_R \sin(90 - 57.52)° = (9864\ \text{N})\sin(90 - 57.52)° = \mathbf{5297\ N}$$

Discussion This problem can also be solved without finding F_R by finding the lines of action of the horizontal hydrostatic force and the weight.

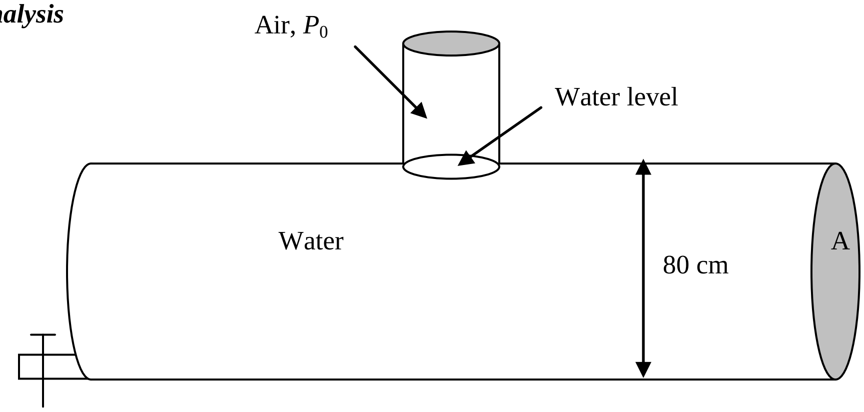
3-80

Solution A cylindrical tank is fully filled by water. The hydrostatic force on the surface A is to be determined for three different pressures on the water surface.

Assumptions Atmospheric pressure acts on both sides of the cylinder, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis



p=0 bar.

$$F_R = 9810 \times 0.4 \times \frac{\pi 0.8^2}{4} = \mathbf{1972\,N \cong 1.97\,kN}$$

$$y_{cp} = y_{cg} + \frac{I_{xc}}{y_{cg} A} = 0.4 + \frac{\pi 0.8^4 / 64}{0.4 \times \pi 0.8^2 / 4} = 0.5\ m$$

p=3 bar.

Additional imaginary water column

$$h = \frac{p_{air}}{\gamma_{water}} = \frac{3 \times 10^5\ Pa}{9810} = 30.58\,m$$

Therefore we can imagine the water level as if it were 30.58 m higher than its original level. In this case,

$$y_{cg} = h_{cg} = 0.4 + 30.58 = 30.98\ m$$

$$F_R = 9810 \times 30.58 \times \frac{\pi 0.8^2}{4} = \mathbf{150{,}791N \cong 151kN}$$

$$y_{cp} = y_{cg} + \frac{I_{xc}}{y_{cg} A} = 30.98 + \frac{\pi 0.8^4 / 64}{30.98 \times \pi 0.8^2 / 4} = 30.981\ m$$

p=10 bar.

Additional imaginary water column

$$h = \frac{p_{air}}{\gamma_{water}} = \frac{10 \times 10^5\ Pa}{9810} = 101.94\ m$$

$$F_R = 9810 \times 101.94 \times \frac{\pi 0.8^2}{4} = \mathbf{502671N \cong 503kN}$$

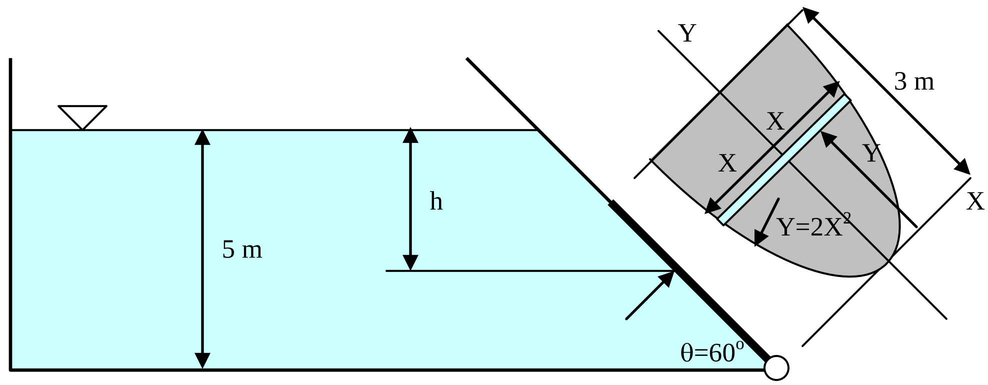
3-81

Solution An open settling tank contains liquid suspension. The resultant force acting on the gate and its line of action are to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis



$$\rho_{ave} = \frac{800+900}{2} = 850\ kg/m^3$$

$$dF_R = pdA,\ \text{or}\quad F_R = \int_A pdA = \int_A \rho_{ave} ghdA = \rho_{ave} g\int_A hdA \qquad (1)$$

Based on the figure below, $dA = 2XdY$, and $h = 5 - YSin(\theta)$, $X = \sqrt{Y/2}$.

Therefore

$$F_R = 2\rho_{ave} g\int_{Y=0}^{Y=3}(5-YSin\theta)XdY = 2\rho_{ave} g\int_{Y=0}^{Y=3}(5-YSin\theta)\sqrt{Y/2}\ dY = \sqrt{2}\rho_{ave} g\int_{Y=0}^{Y=3}(5-YSin\theta)\sqrt{Y}\ dY$$

$$F_R = \sqrt{2}\times 850\times 9.8\times\left(\frac{10}{3}Y^{3/2} - \frac{2}{5}Sin\theta Y^{5/2}\right)_{Y=0}^{Y=3} = \mathbf{140{,}428\,N \cong 140\,kN}$$

To locate F_R, we would use Eq. 1 again.

$$F_R y_{cp} = \left(\sqrt{2}\rho_{ave} g\int_{Y=0}^{Y=3}(5-YSin\theta)\sqrt{Y}\ dY\right)\times Y = \sqrt{2}\rho_{ave} g\int_{Y=0}^{Y=3}(5Y - Y^2 Sin\theta)\sqrt{Y}\ dY$$

$$140{,}428\times y_{cp} = \sqrt{2}\rho_{ave} g\left(2Y^{5/2} - 0.247435Y^{7/2}\right)_{Y=0}^{Y=3} = 230{,}961$$

$$y_{cp} = \frac{230{,}961}{140{,}428} = \mathbf{1.64\ m}\ \text{(from bottom), and}\ x_{cp} = 0\ \text{obviously.}$$

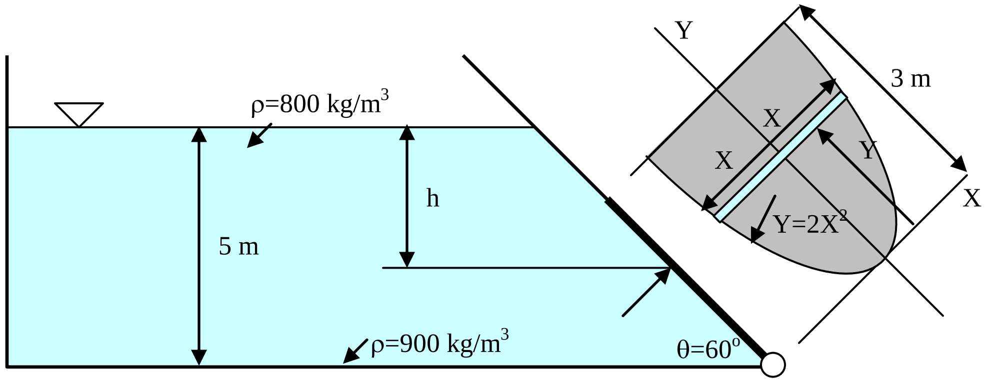
3-82

Solution An open settling tank contains liquid suspension. The density of the suspension depends on liquid depth linearly. The resultant force acting on the gate and its line of action are to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis



$$dF_R = pdA\text{, or }\ F_R = \int_A pdA = \int_A \rho ghdA \qquad (1)$$

Since the density of suspension is linearly changing with ***h*** we would propose

$$\rho = 800 + 20h\ [kg/m^3]$$

Based on the figure below, $dA = 2XdY$, and $h = 5 - YSin(\theta)$, $X = \sqrt{Y/2}$. Therefore

$$F_R = g\int_{Y=0}^{Y=3}(800+20h)\times h\times 2XdY = \sqrt{2}g\int_{Y=0}^{Y=3}(800+20(5-YSin\theta))\times(5-YSin\theta)\times\sqrt{Y}\,dY$$

$$F_R = g\sqrt{2}\left(\frac{40}{7}Sin^2\theta Y^{7/2} - 400Sin\theta Y^{5/2} + 3000Y^{3/2}\right)_{Y=0}^{Y=3} = \mathbf{143{,}983\,N \cong 144\,kN}$$

To locate F$_R$, we would use Eq. 1 again.

$$F_R y_{cp} = \left(g\int_{Y=0}^{Y=3}(800+20h)\times h\times 2XdY\right)\times Y = \sqrt{2}g\int_{Y=0}^{Y=3}(800+20(5-YSin\theta))\times(5-YSin\theta)\times Y^{3/2}dY$$

$$143{,}983y_{cp} = g\sqrt{2}\left(\frac{40}{9}Sin^2\theta Y^{9/2} - \frac{2000}{7}Sin\theta Y^{7/2} + 1800Y^{5/2}\right)_{Y=0}^{Y=3} = 234{,}991$$

$$y_{cp} = \frac{234{,}991}{143{,}983} = \mathbf{1.63\ m}$$ (from bottom), and $x_{cp} = 0$ obviously.

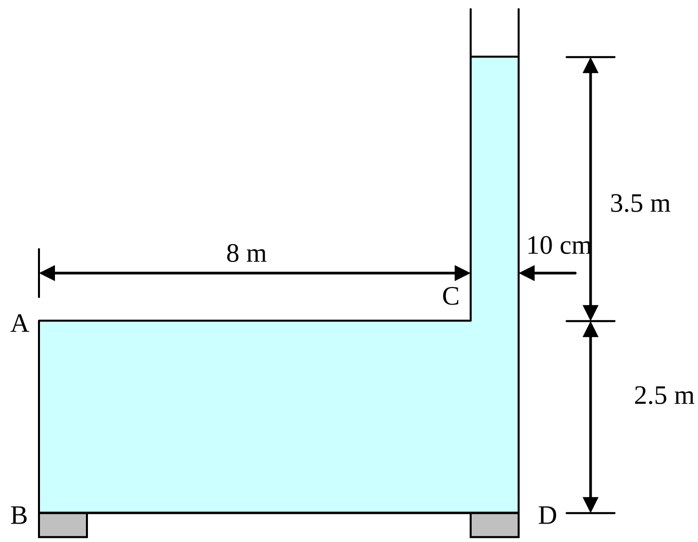
3-83

Solution A tank is filled by oil. The magnitude and the location of the line of action of the resultant force acting on the surface and the pressure force acting on the surface are to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties The specific gravity of oil is given.

Analysis



(a)

$$F_{R-AB} = \gamma g h_{cg} A = 0.88 \times 9810 \times (3.5 + 2.5/2) \times (2.5 \times 6) = 615{,}087\ N \cong \mathbf{615\,kN}$$

$$y_{cp} = y_{cg} + \frac{I_{xc}}{y_{cg} A} = 4.75 + \frac{6 \times 2.5^3/12}{4.75 \times (2.5 \times 6)} = \mathbf{4.873\,m}$$

(b)

$$p_{BD} = 0.88 \times 9810 \times (3.5 + 2.5) = 51797\ Pa$$

$$F_{BD} = p_{BD} A = 51797 \times (8 + 0.1) \times 6 = 2.517 \times 10^6\ N = \mathbf{2.52\,MN}$$

The weight of the oil is

$$W = \forall \gamma = (6 \times 8.1 \times 2.5 + 0.1 \times 0.1 \times 3.5) \times 0.88 \times 9810 = 1.049 \times 10^6\ N = 1.049\ MN$$

It is interesting that the weight of the oil is pretty less than the pressure force acting on the bottom surface of the tank. However, if we calculate the force acting on top surface,

$$F_{AC} = p_{AC} A = 0.88 \times 9810 \times 3.5 \times (8 + 0.1) \times 6 = 1.468 \times 10^6\ N = 1.468\ MN \quad \text{(upward)}$$

The difference between F_{BD} and F_{AC} would give the weight of the oil

$$W = 2.517 - 1.468 = 1.049\ MN$$

3-84

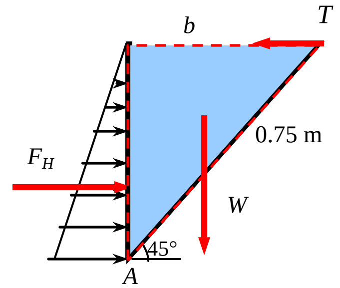
Solution Two parts of a water trough of triangular cross-section are held together by cables placed along the length of the trough. The tension T in each cable when the trough is filled to the rim is to be determined.

Assumptions **1** Atmospheric pressure acts on both sides of the trough wall, and thus it can be ignored in calculations for convenience. **2** The weight of the trough is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis To expose the cable tension, we consider half of the trough whose cross-section is triangular. The water height h at the midsection of the trough and width of the free surface are

$$h = L\sin\theta = (0.75\text{ m})\sin 45° = 0.530\text{ m}$$
$$b = L\cos\theta = (0.75\text{ m})\cos 45° = 0.530\text{ m}$$



The hydrostatic forces acting on the vertical and horizontal plane surfaces as well as the weight of the liquid block are determined as follows:

Horizontal force on vertical surface:

$$F_H = F_x = P_{\text{avg}}A = \rho g h_C A = \rho g\left(h/2\right)A$$
$$= \left(1000\text{ kg/m}^3\right)\left(9.81\text{ m/s}^2\right)\left(0.530/2\text{ m}\right)(0.530\text{ m}\times 6\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$
$$= 8267\text{ N}$$

The vertical force on the horizontal surface is zero since it coincides with the free surface of water. The weight of fluid block per 6-m length is

$$F_V = W = \rho g V = \rho g[w\times bh/2]$$
$$= (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)[(6\text{ m})(0.530\text{ m})(0.530\text{ m})/2]\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$
$$= 8267\text{ N}$$

The distance of the centroid of a triangle from a side is 1/3 of the height of the triangle for that side. Taking the moment about point A where the two parts are hinged and setting it equal to zero gives

$$\sum M_A = 0 \qquad \rightarrow \qquad W\frac{b}{3} + F_H\frac{h}{3} = Th$$

Solving for T and substituting, and noting that $h = b$, the tension in the cable is determined to be

$$T = \frac{F_H + W}{3} = \frac{(8267 + 8267)\text{ N}}{3} = 5511\text{ N} \cong \mathbf{5510\text{ N}}$$

Discussion The analysis is simplified because of the symmetry of the trough.

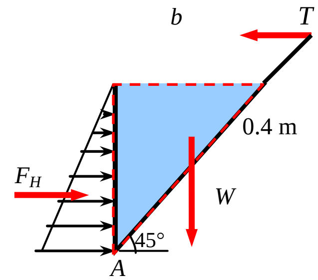
3-85

Solution Two parts of a water trough of triangular cross-section are held together by cables placed along the length of the trough. The tension T in each cable when the trough is filled to the rim is to be determined.

Assumptions **1** Atmospheric pressure acts on both sides of the trough wall, and thus it can be ignored in calculations for convenience. **2** The weight of the trough is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis To expose the cable tension, we consider half of the trough whose cross-section is triangular. The water height is given to be h = 0.4 m at the midsection of the trough, which is equivalent to the width of the free surface b since tan 45° = b/h = 1. The hydrostatic forces acting on the vertical and horizontal plane surfaces as well as the weight of the liquid block are determined as follows:



Horizontal force on vertical surface:

$$F_H = F_x = P_{\text{avg}} A = \rho g h_C A = \rho g \left(h/2\right) A$$

$$= \left(1000 \text{ kg/m}^3\right)\left(9.81 \text{ m/s}^2\right)\left(0.4/2 \text{ m}\right)\left(0.4 \text{ m} \times 3 \text{ m}\right)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 2354 \text{ N}$$

The vertical force on the horizontal surface is zero since it coincides with the free surface of water. The weight of fluid block per 3-m length is

$$F_V = W = \rho g \mathbf{V} = \rho g[w \times bh/2]$$

$$= (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)[(3 \text{ m})(0.4 \text{ m})(0.4 \text{ m})/2]\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 2354 \text{ N}$$

The distance of the centroid of a triangle from a side is 1/3 of the height of the triangle for that side. Taking the moment about point A where the two parts are hinged and setting it equal to zero gives

$$\sum M_A = 0 \qquad \rightarrow \qquad W\frac{b}{3} + F_H\frac{h}{3} = Th$$

Solving for T and substituting, and noting that $h = b$, the tension in the cable is determined to be

$$T = \frac{F_H + W}{3} = \frac{\left(2354 + 2354\right) \text{ N}}{3} = 1569 \text{ N} \cong \mathbf{1570 \text{ N}}$$

Discussion The tension force here is a factor of about 3.5 smaller than that of the previous problem, even though the trough is more than half full.

3-86

Solution A retaining wall against mud slide is to be constructed by rectangular concrete blocks. The mud height at which the blocks will start sliding, and the blocks will tip over are to be determined.

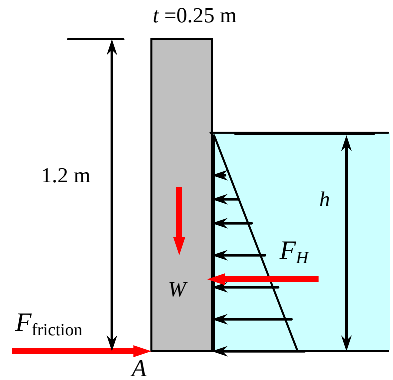
Assumptions Atmospheric pressure acts on both sides of the wall, and thus it can be ignored in calculations for convenience.

Properties The density is given to be 1400 kg/m^3 for the mud, and 2700 kg/m^3 for concrete blocks.

Analysis (*a*) The weight of the concrete wall per unit length (L = 1 m) and the friction force between the wall and the ground are

$$W_{\text{block}} = \rho g V = (2700\ \text{kg/m}^3)(9.81\ \text{m/s}^2)[0.25\times 1.2\times 1\ \text{m}^3]\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = 7946\ \text{N}$$

$$F_{\text{friction}} = \mu W_{\text{block}} = 0.4(7946\ \text{N}) = 3178\ \text{N}$$



The hydrostatic force exerted by the mud to the wall is

$$F_H = F_x = P_{avg}A = \rho g h_C A = \rho g (h/2)A$$

$$= (1400\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(h/2)(1\times h)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 6867h^2\ \text{N}$$

Setting the hydrostatic and friction forces equal to each other gives

$$F_H = F_{\text{friction}} \quad\rightarrow\quad 6867\,h^2 = 3178 \quad\rightarrow\quad h = \mathbf{0.680m}$$

(*b*) The line of action of the hydrostatic force passes through the pressure center, which is 2*h*/3 from the free surface. The line of action of the weight of the wall passes through the midplane of the wall. Taking the moment about point *A* and setting it equal to zero gives

$$\sum M_A = 0 \quad\rightarrow\quad W_{\text{block}}(t/2) = F_H(h/3) \quad\rightarrow\quad W_{\text{block}}(t/2) = 6867h^3/3$$

Solving for *h* and substituting, the mud height for tip over is determined to be

$$h = \left(\frac{3W_{\text{block}}t}{2\times 8829}\right)^{1/3} = \left(\frac{3\times 7946\times 0.25}{2\times 6867}\right)^{1/3} = \mathbf{0.757m}$$

Discussion The concrete wall will slide before tipping. Therefore, sliding is more critical than tipping in this case.

3-87

Solution A retaining wall against mud slide is to be constructed by rectangular concrete blocks. The mud height at which the blocks will start sliding, and the blocks will tip over are to be determined.

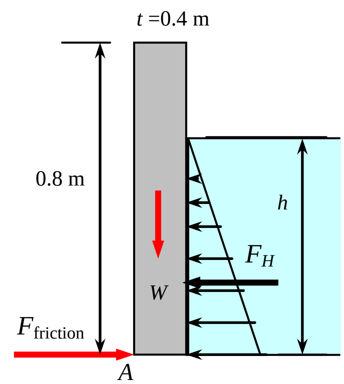
Assumptions Atmospheric pressure acts on both sides of the wall, and thus it can be ignored in calculations for convenience.

Properties The density is given to be 1400 kg/m^3 for the mud, and 2700 kg/m^3 for concrete blocks.

Analysis (*a*) The weight of the concrete wall per unit length (L = 1 m) and the friction force between the wall and the ground are

$$W_{\text{block}} = \rho g V = (2700\text{ kg/m}^3)(9.81\text{ m/s}^2)[0.4 \times 0.8 \times 1\text{ m}^3]\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 8476\text{ N}$$

$$F_{\text{friction}} = \mu W_{\text{block}} = 0.4(8476\text{ N}) = 3390\text{ N}$$



The hydrostatic force exerted by the mud to the wall is

$$F_H = F_x = P_{avg}A = \rho g h_C A = \rho g (h/2) A$$

$$= (1400\text{ kg/m}^3)(9.81\text{ m/s}^2)(h/2)(1 \times h)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 6867h^2\text{ N}$$

Setting the hydrostatic and friction forces equal to each other gives

$$F_H = F_{\text{friction}} \quad \rightarrow \quad 6867h^2 = 3390 \quad \rightarrow \quad h = \mathbf{0.703\,m}$$

(*b*) The line of action of the hydrostatic force passes through the pressure center, which is 2*h*/3 from the free surface. The line of action of the weight of the wall passes through the midplane of the wall. Taking the moment about point *A* and setting it equal to zero gives

$$\sum M_A = 0 \quad \rightarrow \quad W_{\text{block}}(t/2) = F_H(h/3) \quad \rightarrow \quad W_{\text{block}}(t/2) = 6867h^3/3$$

Solving for *h* and substituting, the mud height for tip over is determined to be

$$h = \left(\frac{3W_{\text{block}}t}{2 \times 6867}\right)^{1/3} = \left(\frac{3 \times 8476 \times 0.4}{2 \times 6867}\right)^{1/3} = \mathbf{0.905\,m}$$

Discussion Note that the concrete wall will slide before tipping. Therefore, sliding is more critical than tipping in this case.

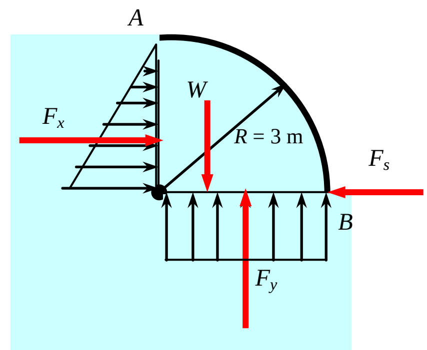
3-88

Solution A quarter-circular gate hinged about its upper edge controls the flow of water over the ledge at *B* where the gate is pressed by a spring. The minimum spring force required to keep the gate closed when the water level rises to *A* at the upper edge of the gate is to be determined.

Assumptions **1** The hinge is frictionless. **2** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience. **3** The weight of the gate is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis We consider the free body diagram of the liquid block enclosed by the circular surface of the gate and its vertical and horizontal projections. The hydrostatic forces acting on the vertical and horizontal plane surfaces as well as the weight of the liquid block are determined as follows:



Horizontal force on vertical surface:

$$F_H = F_x = P_{ave}A = \rho g h_C A = \rho g (R/2)A$$
$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(3/2\ \text{m})(4\ \text{m}\times 3\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$
$$= 176.6\ \text{kN}$$

Vertical force on horizontal surface (upward):

$$F_y = P_{\text{avg}}A = \rho g h_C A = \rho g h_{\text{bottom}} A$$
$$= \left(1000\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)\left(3\ \text{m}\right)\left(4\ \text{m}\times 3\ \text{m}\right)\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 353.2\ \text{kN}$$

The weight of fluid block per 4-m length (downwards):

$$W = \rho g V = \rho g\left[w \times \pi R^2/4\right]$$
$$= \left(1000\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)\left[\left(4\ \text{m}\right)\pi(3\ \text{m})^2/4\right]\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 277.4\ \text{kN}$$

Therefore, the net upward vertical force is

$$F_V = F_y - W = 353.2 - 277.4 = 75.8\ \text{kN}$$

Then the magnitude and direction of the hydrostatic force acting on the surface of the 4-m long quarter-circular section of the gate become

$$F_R = \sqrt{F_H^2 + F_V^2} = \sqrt{(176.6\ \text{kN})^2 + (75.8\ \text{kN})^2} = 192.2\ \text{kN}$$
$$\tan\theta = \frac{F_V}{F_H} = \frac{75.8\ \text{kN}}{176.6\ \text{kN}} = 0.429 \quad \to \quad \theta = 23.2°$$

Therefore, the magnitude of the hydrostatic force acting on the gate is 192.2 kN, and its line of action passes through the center of the quarter-circular gate making an angle 23.2° upwards from the horizontal.

The minimum spring force needed is determined by taking a moment about the point *A* where the hinge is, and setting it equal to zero,

$$\sum M_A = 0 \quad \to \quad F_R R\sin(90-\theta) - F_{\text{spring}}R = 0$$

Solving for F_{spring} and substituting, the spring force is determined to be

$$F_{\text{spring}} = F_R\sin(90-\theta) = (192.2\ \text{kN})\sin(90° - 23.2°) = \mathbf{177\ kN}$$

Discussion Several variations of this design are possible. Can you think of some of them?

3-89

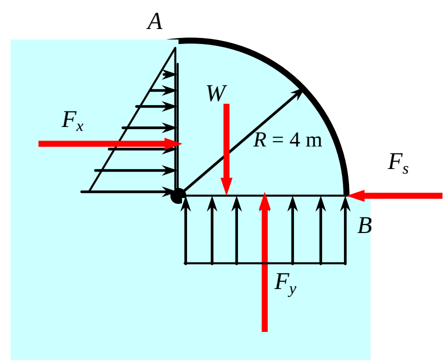
Solution A quarter-circular gate hinged about its upper edge controls the flow of water over the ledge at *B* where the gate is pressed by a spring. The minimum spring force required to keep the gate closed when the water level rises to *A* at the upper edge of the gate is to be determined.

Assumptions **1** The hinge is frictionless. **2** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience. **3** The weight of the gate is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis We consider the free body diagram of the liquid block enclosed by the circular surface of the gate and its vertical and horizontal projections. The hydrostatic forces acting on the vertical and horizontal plane surfaces as well as the weight of the liquid block are determined as follows:

Horizontal force on vertical surface:



$$F_H = F_x = P_{ave}A = \rho g h_C A = \rho g(R/2)A$$

$$= (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(4/2\text{ m})(4\text{ m}\times 4\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 313.9\text{ kN}$$

Vertical force on horizontal surface (upward):

$$F_y = P_{ave}A = \rho g h_C A = \rho g h_{\text{bottom}}A$$

$$= (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(4\text{ m})(4\text{ m}\times 4\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 627.8\text{ kN}$$

The weight of fluid block per 4-m length (downwards):

$$W = \rho g \mathbf{V} = \rho g[w\times \pi R^2/4]$$

$$= (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)[(4\text{ m})\pi(4\text{ m})^2/4]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 493.1\text{ kN}$$

Therefore, the net upward vertical force is

$$F_V = F_y - W = 627.8 - 493.1 = 134.7\text{ kN}$$

Then the magnitude and direction of the hydrostatic force acting on the surface of the 4-m long quarter-circular section of the gate become

$$F_R = \sqrt{F_H^2 + F_V^2} = \sqrt{(313.9\text{ kN})^2 + (134.7\text{ kN})^2} = 341.6\text{ kN}$$

$$\tan\theta = \frac{F_V}{F_H} = \frac{134.7\text{kN}}{313.9\text{ kN}} = 0.429 \quad\rightarrow\quad \theta = 23.2°$$

Therefore, the magnitude of the hydrostatic force acting on the gate is 341.6 kN, and its line of action passes through the center of the quarter-circular gate making an angle 23.2° upwards from the horizontal.

The minimum spring force needed is determined by taking a moment about the point *A* where the hinge is, and setting it equal to zero,

$$\sum M_A = 0 \quad\rightarrow\quad F_R R\sin(90-\theta) - F_{\text{spring}}R = 0$$

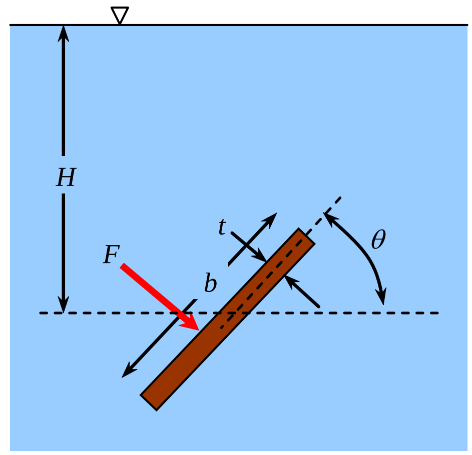
Solving for F_{spring} and substituting, the spring force is determined to be

$$F_{\text{spring}} = F_R\sin(90-\theta) = (341.6\text{ kN})\sin(90° - 23.2°) = \mathbf{314\ kN}$$

Discussion If the previous problem is solved using a program like EES, it is simple to repeat with different values.

3-90

Solution We are to determine the force on the upper face of a submerged flat plate.



Assumptions **1** The water is incompressible. **2** The system is at rest during the analysis (hydrostatics). **3** Atmospheric pressure is ignored since it acts on both sides of the plate.

Analysis (*a*) At first, and as a good approximation as plate thickness *t* approaches zero, the pressure force is simply $F = \rho gHA = \rho gHbw$, since the centroid of the plate is at its center regardless of the tilt angle. However, the plate thickness must be taken into account since we are concerned with the *upper* face of the plate. Some trig yields that the depth of water from the surface to the centroid of the upper plate is $H - (t/2)\cos\theta$, i.e., somewhat smaller than *H* itself since the upper face of the plate is above the center of the plate when it is tilted ($\theta > 0$). Thus,

$$\boxed{F = \rho g\left(H - \frac{t}{2}\cos\theta\right)bw}$$

(*b*) For the given values,

$$F = \rho g\left(H - \frac{t}{2}\cos\theta\right)bw$$
$$= \left(998.3\frac{\text{kg}}{\text{m}^3}\right)\left(9.807\frac{\text{m}}{\text{s}^2}\right)\left(1.25\text{ m} - \frac{0.200\text{ m}}{2}\cos 30^\circ\right)(1.00\text{ m})(1.00\text{ m})\left(\frac{\text{N}}{\text{kg}\cdot\text{m/s}^2}\right)$$
$$= 11390\text{ N}$$

Thus, the "gage" force (ignoring atmospheric pressure) on the upper plate surface is $\boxed{F = 11{,}400\text{ N}}$ (to three digits).

Discussion If we ignore plate thickness (set $t = 0$), the result becomes 12,200 N, which represents an error of around 7%, since the plate here is fairly thick.

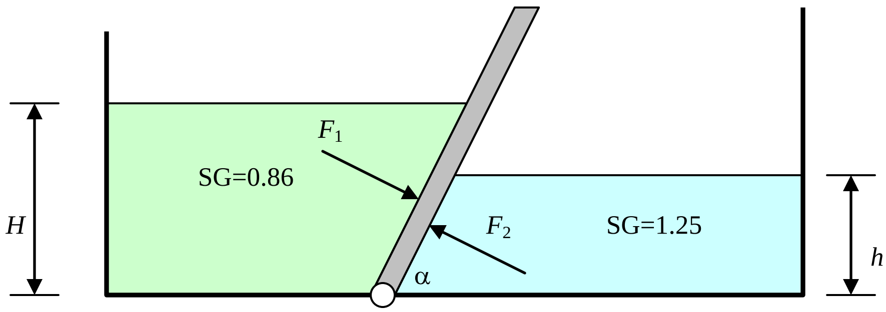
3-91

Solution Two fluids are separated by a gate. The height ratio of the two fluids is to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties The specific gravities of two fluids are given in the figure.

Analysis



$$F_1 = \gamma h_{cg} A = \gamma \frac{H}{2} \frac{H}{Sin(\alpha)} b = \frac{\gamma H^2 b}{2Sin(\alpha)}$$

$$F_2 = \gamma h_{cg} A = \gamma \frac{h}{2} \frac{h}{Sin(\alpha)} b = \frac{\gamma h^2 b}{2Sin(\alpha)}$$

$$\frac{F_1}{F_2} = \frac{\gamma_1}{\gamma_2}\left(\frac{H}{h}\right)^2,$$

or we get

$$\frac{H}{h} = \sqrt{\frac{\gamma_2}{\gamma_1}\frac{F_1}{F_2}} = \sqrt{\frac{1.25}{0.86}\times 1.70} = \mathbf{1.57}$$

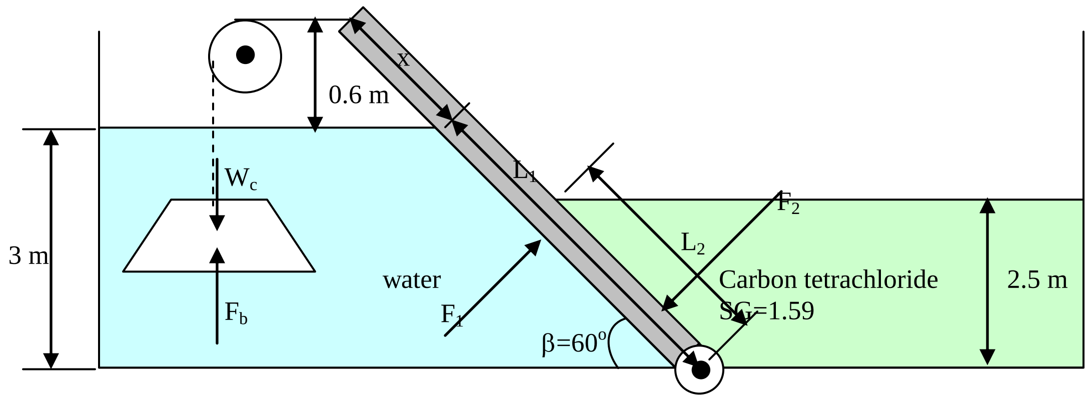
3-92

Solution An inclined gate separates water from another fluid. The volume of the concrete block to keep the gate at the given position is to be determined.

Assumptions **1** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience. **2** The weight of the gate is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout. The specific gravities of concerete and carbon tetrachloride are 2.4 and 1.59, respectively.

Analysis



The force applied by water:

$$F_1 = \gamma h_{cg1} A_1 = 9810 \times \frac{3}{2} \times \left(1 \times \frac{3}{Sin\beta}\right) = 50974\ N$$

$$y_{cp1} = \frac{3}{2Sin60} + \frac{1 \times \left(\frac{3}{Sin60}\right)^3 \Big/ 12}{\frac{3}{2Sin60} \times \left(1 \times \frac{3}{Sin60}\right)} = 2.31\ m$$

The force applied by carbon tetrachloride:

$$F_2 = \gamma h_{cg2} A_2 = 1.59 \times 9810 \times \frac{2.5}{2} \times \left(1 \times \frac{2.5}{Sin60}\right) = 56284\ N$$

$$y_{cp2} = \frac{2.5}{2Sin60} + \frac{1 \times \left(\frac{2.5}{Sin60}\right)^3 \Big/ 12}{\frac{2.5}{2Sin60} \times \left(1 \times \frac{2.5}{Sin60}\right)} = 1.924\ m \qquad x = \frac{0.6}{Sin60} = 0.693\ m$$

Moment about hinge would give

$$(W_c - F_b) \times Sin\beta \times (x + L_1) + F_2 \times (L_2 - y_{cp2}) - F_1 \times (L_1 - y_{cp1}) = 0$$

Since $W_c - F_b = \forall_c(\gamma_c - \gamma_w)$, we obtain

$$\forall_c = \frac{F_1 \times (L_1 - y_{cp1}) - F_2 \times (L_2 - y_{cp2})}{(\gamma_c - \gamma_w) \times Sin\beta \times (L_1 + x)} = \frac{50974(3.464 - 2.41) - 56284(2.886 - 1.924)}{9810(2.4 - 1)Sin60(3.464 + 0.693)} = \mathbf{0.0946\,m^3}$$

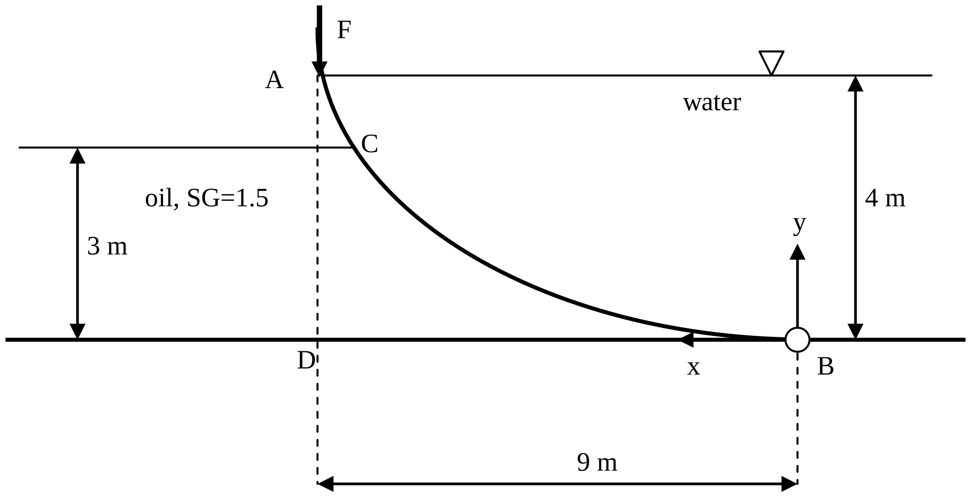
3-93

Solution A parabolic shaped gate is hinged. The force needed to keep the gate stationary is to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout. The specific gravity of oil is 1.5.

Analysis



We should need to determine the function describing the curve of the gate. Generally a parabolic shape is defined by $y = C_1x^2 + C_2x + C_3$. Since the parabola is passing through the origin, C_2=C_3=0. On the other hand,

x=9 m when y=4 m, and we write $\quad 4 = C_1 9^2$

Solving for C_1 yields $C_1 = \dfrac{4}{81}$

Therefore we obtain $y = \dfrac{4}{81}x^2$.

Force applied by oil:

$$F_{Ho} = \int_{y_1}^{y_2} pbdy = \int_{y_1}^{y_2}(\gamma h)bdy = b\gamma\int_{y_1}^{y_2} hdy \qquad (1)$$

It is clear that $\mathrm{h} + \mathrm{y} = 3\ \mathrm{m}$, and we may write $\mathrm{h} = 3 - \mathrm{y}$. Therefore Eq. 1 would take the form of

$$F_{Ho} = b\gamma\int_{y_1}^{y_2}(3-y)dy = 2\times1.5\times9810\int_0^3(3-y)dy = 2\times1.5\times9810\left(3y-0.5y^2\right)_0^3$$
$$= 2\times1.5\times9810\times(4.5) = 132435\ N$$

To locate F_{Ho}, we write

$$F_{Ho}y_{c-o} = \left(b\gamma\int_{y_1}^{y_2}(3-y)dy\right)y = 2\times1.5\times9810\int_0^3(3y-y^2)dy = 2\times1.5\times9810\left(\frac{3}{2}y^2-\frac{1}{3}y^3\right)_0^3$$
$$= 2\times1.5\times9810\times(4.5) = 132435\ Nm$$

$$y_{c-o} = \frac{132435}{132435} = 1\ m$$

or we could find y_{cp} from $y_{c-o} = \frac{1}{3} \times 3 = 1\ m$

Vertical component of the force

$$dF_{Vo} = pdA_x \text{, where } dA_x = bdx \qquad (1)$$

$$F_{Vo} = \int_{x_1}^{x_2} pbdx = \int_{x_1}^{x_2} (\gamma h)bdx = b\gamma \int_{x_1}^{x_2} hdx \qquad (2)$$

Since $h = 3 - y$ .and , and $y = \frac{4}{81}x^2$, we get $h = 3 - \frac{4}{81}x^2$

$$F_{Vo} = b\gamma \int_{x_1}^{x_2} (3 - \frac{4}{81}x^2)dx = 2 \times 1.5 \times 9810 \int_0^{7.794} (3 - \frac{4}{81}x^2)dx$$

$$= 2 \times 1.5 \times 9810 \times \left(3x - \frac{4}{243}x^3\right)_0^{7.794} = 2 \times 1.5 \times 9810 \times (15.5884) = 458767\ N$$

To locate F_{Vo}, we write the moment equation about origin:

$$F_{Vo} x_{c-o} = \left(b\gamma \int_{x_1}^{x_2} (3 - \frac{4}{81}x^2)dx \right) x$$

$$= b\gamma \int_0^{7.794} (3x - \frac{4}{81}x^3)dx = 2 \times 1.5 \times 9810 \times \left(\frac{3}{2}x^2 - \frac{1}{81}x^3\right)_0^{7.794} = 2 \times 1.5 \times 9810 \times (45.5625)$$

$$= 1340904.375$$

$$x_{c-o} = \frac{1340904.375}{458767} = 2.923\ m$$

Force applied by water

Horizontal component (we use an alternative method for this part)

$$F_{Hw} = \gamma (h_{cg} A)_{projected} = 9810 \times \left(\frac{4}{2} \times 4 \times 2\right) = 156960\ N$$

$$y_{c-w} = \frac{1}{3} \times 4 = 1.333\ m$$

Vertical component

$$F_{Vw} = b\gamma \int_{x_1}^{x_2} (4 - \frac{4}{81}x^2)dx = 2 \times 9810 \int_0^9 (4 - \frac{4}{81}x^2)dx$$

$$= 2 \times 9810 \times \left(4x - \frac{4}{243}x^3\right)_0^9 = 2 \times 9810 \times (24) = 470880\ N$$

$$F_{Vw} x_{c-w} = \left(b\gamma \int_{x_1}^{x_2} (4 - \frac{4}{81} x^2) dx \right) x$$

$$= b\gamma \int_0^9 (4x - \frac{4}{81} x^3) dx = 2 \times 9810 \times \left(2x^2 - \frac{1}{81} x^4 \right)_0^9 = 2 \times 9810 \times (81) = 1589220$$

$$x_{c-w} = \frac{1589220}{470880} = 3.375\ m$$

Moment equation about hinge would yield

$$F \times BD - F_{Ho} \times y_{c-o} - F_{Vo} x_{c-o} + F_{Hw} y_{c-w} + F_{Vw} x_{c-w} = 0$$

$$F = \frac{132435 \times 1 + 458767 \times 2.923 - 156960 \times 1.333 - 470880 \times 3.375}{9} = \frac{-614059.95}{9} = -68229\ N$$

$$F = \mathbf{68{,}229\,N} \uparrow \cong \mathbf{68{,}3\,kN} \uparrow$$

Buoyancy

3-94C

Solution We are to define and discuss the buoyant force.

Analysis The **upward force a fluid exerts on an immersed body** is called the *buoyant force*. The buoyant force is **caused by the increase of pressure in a fluid with depth**. The *magnitude* of the buoyant force acting on a submerged body whose volume is V is expressed as $\boxed{F_B = \rho_f g V}$. The *direction* of the buoyant force is **upwards**, and its *line of action* **passes through the centroid of the displaced volume**.

Discussion If the buoyant force is greater than the body's weight, it floats.

3-95C

Solution We are to compare the buoyant force on two spheres.

Analysis The magnitude of the buoyant force acting on a submerged body whose volume is V is expressed as $F_B = \rho_f g V$, which is independent of depth. Therefore, **the buoyant forces acting on two identical spherical balls submerged in water at different depths is the same**.

Discussion Buoyant force depends only on the volume of the object, not its density.

3-96C

Solution We are to compare the buoyant force on two spheres.

Analysis The magnitude of the buoyant force acting on a submerged body whose volume is V is expressed as $F_B = \rho_f g V$, which is independent of the density of the body (ρ_f is the fluid density). Therefore, **the buoyant forces acting on the 5-cm diameter aluminum and iron balls submerged in water is the same**.

Discussion Buoyant force depends only on the volume of the object, not its density.

3-97C

Solution We are to compare the buoyant forces on a cube and a sphere.

Analysis The magnitude of the buoyant force acting on a submerged body whose volume is V is expressed as $F_B = \rho_f g V$, which is independent of the shape of the body. Therefore, **the buoyant forces acting on the cube and sphere made of copper submerged in water are the same since they have the same volume**.

Discussion The two objects have the same volume because they have the same mass *and* density.

3-98C

Solution We are to discuss the stability of a submerged and a floating body.

Analysis A *submerged body* whose **center of gravity *G* is above the center of buoyancy *B*, which is the centroid of the displaced volume, is *unstable***. But a *floating body* **may still be stable when *G* is above *B* since the centroid of the displaced volume shifts to the side to a point *B'* during a rotational disturbance while the center of gravity *G* of the body remains unchanged**. If the point *B'* is sufficiently far, these two forces create a restoring moment, and return the body to the original position.

Discussion Stability analysis like this is critical in the design of ship hulls, so that they are least likely to capsize.

3-99

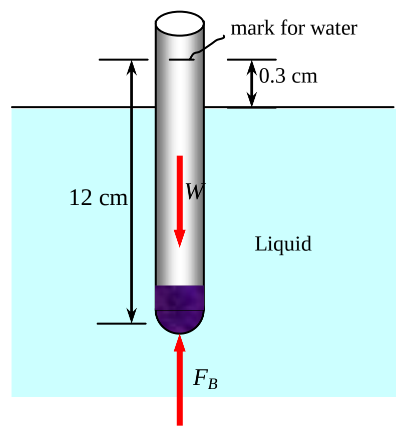
Solution The density of a liquid is to be determined by a hydrometer by establishing division marks in water and in the liquid, and measuring the distance between these marks.

Properties We take the density of pure water to be 1000 kg/m^3.

Analysis A hydrometer floating in water is in static equilibrium, and the buoyant force F_B exerted by the liquid must always be equal to the weight W of the hydrometer, $F_B = W$.

$$F_B = \rho g V_{\text{sub}} = \rho g h A_c$$

where h is the height of the submerged portion of the hydrometer and A_c is the cross-sectional area which is constant.



In pure water: $W = \rho_w g h_w A_c$

In the liquid: $W = \rho_{\text{liquid}} g h_{\text{liquid}} A_c$

Setting the relations above equal to each other (since both equal the weight of the hydrometer) gives

$$\rho_w g h_w A_c = \rho_{\text{liquid}} g h_{\text{liquid}} A_c$$

Solving for the liquid density and substituting,

$$\rho_{\text{liquid}} = \frac{h_{\text{water}}}{h_{\text{liquid}}} \rho_{\text{water}} = \frac{12\text{ cm}}{(12-0.3)\text{ cm}} (1000\text{ kg/m}^3) = \mathbf{1026\ kg/m^3}$$

Discussion Note that for a given cylindrical hydrometer, the product of the fluid density and the height of the submerged portion of the hydrometer is constant in any fluid.

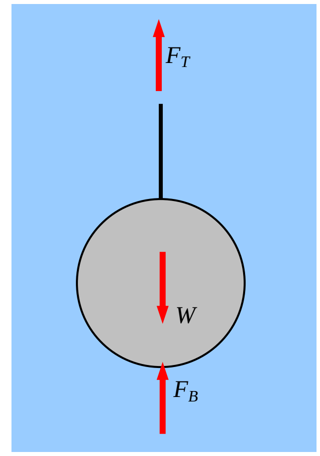
3-100E

Solution A concrete block is lowered into the sea. The tension in the rope is to be determined before and after the block is immersed in water.

Assumptions **1** The buoyancy force in air is negligible. **2** The weight of the rope is negligible.

Properties The density of steel block is given to be 494 lbm/ft^3.

Analysis (*a*) The forces acting on the concrete block in air are its downward weight and the upward pull action (tension) by the rope. These two forces must balance each other, and thus the tension in the rope must be equal to the weight of the block:



$$V = 4\pi R^3/3 = 4\pi(1.5\text{ ft})^3/3 = 14.137\text{ ft}^3$$

$$F_T = W = \rho_{\text{concrete}} gV$$

$$= (494\text{ lbm/ft}^3)(32.2\text{ ft/s}^2)(14.137\text{ ft}^3)\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 6984\text{ lbf} \cong \mathbf{6980\ lbf}$$

(*b*) When the block is immersed in water, there is the additional force of buoyancy acting upwards. The force balance in this case gives

$$F_B = \rho_f gV = (62.4\text{ lbm/ft}^3)(32.2\text{ ft/s}^2)(14.137\text{ ft}^3)\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 882\text{ lbf}$$

$$F_{\text{T,water}} = W - F_B = 6984 - 882 = 6102\text{ lbf} \cong \mathbf{6100\ lbf}$$

Discussion Note that the weight of the concrete block and thus the tension of the rope decreases by (6984 – 6102)/6984 = 12.6% in water.

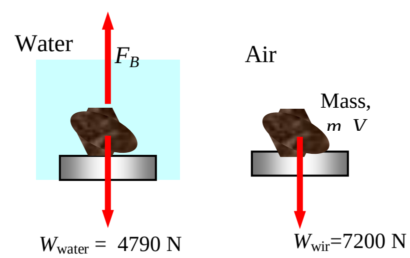
3-101

Solution An irregularly shaped body is weighed in air and then in water with a spring scale. The volume and the average density of the body are to be determined.

Properties We take the density of water to be 1000 kg/m^3.

Assumptions **1** The buoyancy force in air is negligible. **2** The body is completely submerged in water.

Analysis The mass of the body is



$$m = \frac{W_{\text{air}}}{g} = \frac{7200\text{ N}}{9.81\text{ m/s}^2}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = 733.9\text{ kg}$$

The difference between the weights in air and in water is due to the buoyancy force in water,

$$F_B = W_{\text{air}} - W_{\text{water}} = 7200 - 4790 = 2410\text{ N}$$

Noting that $F_B = \rho_{\text{water}} g \boldsymbol{V}$, the volume of the body is determined to be

$$V = \frac{F_B}{\rho_{\text{water}} g} = \frac{2410\text{ N}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)} = 0.2457\text{ m}^3 \cong \mathbf{0.246\ m^3}$$

Then the density of the body becomes

$$\rho = \frac{m}{V} = \frac{733.9\text{ kg}}{0.2457\text{ m}^3} = 2987\text{ kg/m}^3 \cong \mathbf{2990\ kg/m^3}$$

Discussion The volume of the body can also be measured by observing the change in the volume of the container when the body is dropped in it (assuming the body is not porous).

3-102

Solution The height of the portion of a cubic ice block that extends above the water surface is measured. The height of the ice block below the surface is to be determined.

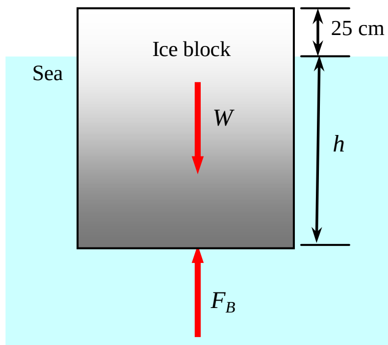
Assumptions **1** The buoyancy force in air is negligible. **2** The top surface of the ice block is parallel to the surface of the sea.

Properties The specific gravities of ice and seawater are given to be 0.92 and 1.025, respectively, and thus the corresponding densities are 920 kg/m^3 and 1025 kg/m^3.

Analysis The weight of a body floating in a fluid is equal to the buoyant force acting on it (a consequence of vertical force balance from static equilibrium). Therefore, in this case the average density of the body must be equal to the density of the fluid since

$$W = F_B \quad \rightarrow \quad \rho_{\text{body}} g V_{\text{total}} = \rho_{\text{fluid}} g V_{\text{submerged}}$$

$$\frac{V_{\text{submerged}}}{V_{\text{total}}} = \frac{\rho_{\text{body}}}{\rho_{\text{fluid}}}$$



The cross-sectional of a cube is constant, and thus the "volume ratio" can be replaced by "height ratio". Then,

$$\frac{h_{\text{submerged}}}{h_{\text{total}}} = \frac{\rho_{\text{body}}}{\rho_{\text{fluid}}} \quad \rightarrow \quad \frac{h}{h + 0.25} = \frac{\rho_{\text{ice}}}{\rho_{\text{water}}} \quad \rightarrow \quad \frac{h}{h + 0.25} = \frac{0.92}{1.025}$$

where h is the height of the ice block below the surface. Solving for h gives

$$h = \frac{(0.92)(0.25)}{1.025 - 0.92} = \mathbf{2.19\ m}$$

Discussion Note that 0.92/1.025 = 0.89756, so approximately 90% of the volume of an ice block remains under water. For symmetrical ice blocks this also represents the fraction of height that remains under water.

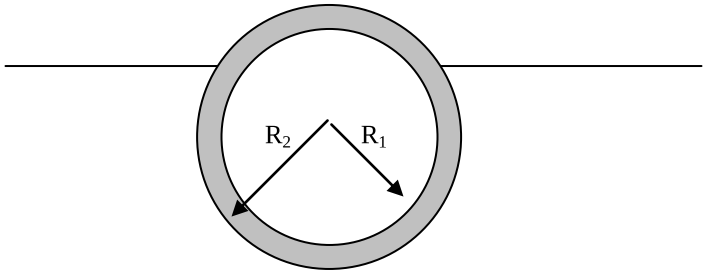
3-103

Solution A spherical shell is placed in water. The percentage of the shell's total volume that would be submerged is to be determined.

Assumptions The buoyancy force in air is negligible.

Properties The density of shell is given to be 1600 kg/m^3 and that for water is 1000 kg/m^3.

Analysis



The weight of the shell:

$$W_S = mg = \rho \frac{4\pi}{3}\left(R_2^3 - R_1^3\right)g = 1600 \times \frac{4\pi}{3}\left(0.06^3 - 0.05^3\right) \times 9.81$$

$$W_S = 5.98\ N$$

The buoyancy force:

$$F_b = \gamma_w \forall_{submerged} = 9810 \times \forall_{submerged}$$

Since $W_S = F_b$,

$$5.98 = 9810 \times \forall_{submerged}, \quad \forall_{submerged} = 5.096 \times 10^{-4}\ m^3$$

$$\frac{\forall_{submerged}}{\forall} = \frac{5.096 \times 10^{-4}}{\frac{4\pi}{3} 0.06^3} \times 100 = \mathbf{67.4\%}$$

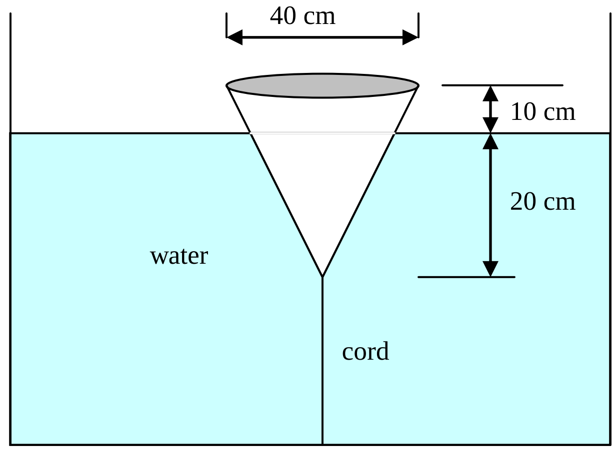
3-104

Solution An inverted cone is placed in a water tank. The tensile in the cord connecting the cone to the bottom of the tank is to be determined.

Assumptions The buoyancy force in air is negligible.

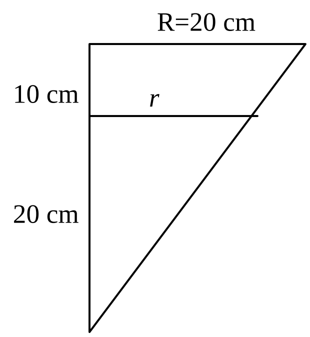
Properties The density of water is 1000 kg/m^3.

Analysis



From the figure below,

$$\frac{R}{30}=\frac{r}{20} \quad \text{and} \quad r=\frac{2R}{3}=\frac{40}{3}=13.33\ cm$$



The displaced volume of water is

$$\forall=\frac{1}{3}\pi r^2 h=\frac{1}{3}\pi\times 0.1333^2\times 0.2=3.72\times 10^{-3}\ m^3$$

Therefore the buoyancy force acting on the cone is

$$F_b=\gamma\forall=9810\times 3.72\times 10^{-3}=36.5\ N$$

For the static equilibrium, we write

$$F+W_c=F_b$$
$$F+16.5=36.5$$
$$F=36.5-16.5=20\ N$$

3-105

Solution The percentage error associated with the neglecting of air buoyancy in the weight of a body is to be determined.

Properties The density of body is 7800 kg/m^3 and that for air is 1.2 kg/m^3.

Analysis If we neglect the buoyancy force, the weight will be

$$W = \gamma \frac{\pi D^3}{6} = 9.81 \times 7800 \frac{\pi 0.2^3}{6} = 320.518\ N$$

If we consider F_b,

$$W' = W - F_b = 320.518 - 1.2 \times 9.81 \times \frac{\pi 0.2^3}{6} = 320.468\ N$$

The percentage error is then

$$e = \frac{W' - W}{W'} \times 100 = \frac{320.468 - 320.518}{320.468} \times 100 = -\mathbf{0.0156\%}$$

It is therefore concluded that the air buoyancy effect can be neglected.

3-106

Solution A man dives into a lake and tries to lift a large rock. The force that the man needs to apply to lift it from the bottom of the lake is to be determined.

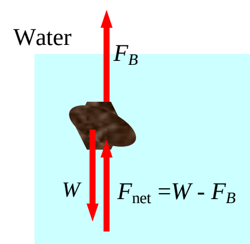
Assumptions **1** The rock is c completely submerged in water. **2** The buoyancy force in air is negligible.

Properties The density of granite rock is given to be 2700 kg/m^3. We take the density of water to be 1000 kg/m^3.

Analysis The weight and volume of the rock are

$$W = mg = (170\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 1668\text{ N}$$

$$V = \frac{m}{\rho} = \frac{170\text{ kg}}{2700\text{ kg/m}^3} = 0.06296\text{ m}^3$$



The buoyancy force acting on the rock is

$$F_B = \rho_{\text{water}} g V = \left(1000\text{ kg/m}^3\right)\left(9.81\text{ m/s}^2\right)\left(0.06296\text{ m}^3\right)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 618\text{ N}$$

The weight of a body submerged in water is equal to the weigh of the body in air minus the buoyancy force,

$$W_{\text{in water}} = W_{\text{in air}} - F_B = 1668 - 618 = \mathbf{1050\ N}$$

Discussion This force corresponds to a mass of $m = \frac{W_{\text{in water}}}{g} = \frac{1050\text{ N}}{9.81\text{ m/s}^2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 107\text{ kg}$. Therefore, a person who can lift 107 kg on earth can lift this rock in water.

3-107

Solution An irregularly shaped crown is weighed in air and then in water with a spring scale. It is to be determined if the crown is made of pure gold.

Assumptions **1** The buoyancy force in air is negligible. **2** The crown is completely submerged in water.

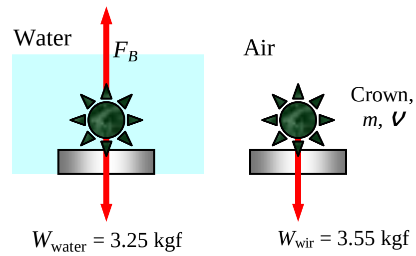
Properties We take the density of water to be 1000 kg/m^3. The density of gold is given to be 19,300 kg/m^3.

Analysis The mass of the crown is

$$m = \frac{W_{\text{air}}}{g} = \frac{34.8\ \text{N}}{9.81\ \text{m/s}^2}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 3.55\ \text{kg}$$

The difference between the weights in air and in water is due to the buoyancy force in water, and thus

$$F_B = W_{\text{air}} - W_{\text{water}} = 34.8 - 31.9 = 2.9\ \text{N}$$



Noting that $F_B = \rho_{\text{water}} g V$, the volume of the crown is determined to be

$$V = \frac{F_B}{\rho_{\text{water}} g} = \frac{2.9\ \text{N}}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)} = 2.96\times10^{-4}\ \text{m}^3$$

Then the density of the crown becomes

$$\rho = \frac{m}{V} = \frac{3.55\ \text{kg}}{2.96\times10^{-4}\ \text{m}^3} = 12{,}000\ \text{kg/m}^3$$

which is considerably less than the density of gold. Therefore, **the crown is NOT made of pure gold**.

Discussion This problem can also be solved without doing any under-water weighing as follows: We would weigh a bucket half-filled with water, and drop the crown into it. After marking the new water level, we would take the crown out, and add water to the bucket until the water level rises to the mark. We would weigh the bucket again. Dividing the weight difference by the density of water and g will give the volume of the crown. Knowing both the weight and the volume of the crown, the density can easily be determined.

3-108

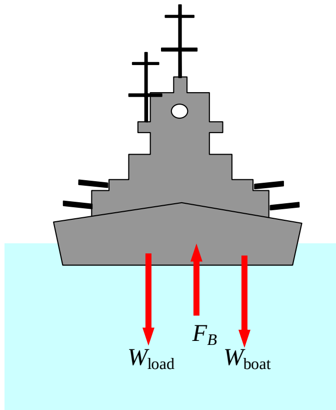
Solution The volume of the hull of a boat is given. The amounts of load the boat can carry in a lake and in the sea are to be determined.

Assumptions **1** The dynamic effects of the waves are disregarded. **2** The buoyancy force in air is negligible.

Properties The density of sea water is given to be 1.03×1000 = 1030 kg/m^3. We take the density of water to be 1000 kg/m^3.

Analysis The weight of the unloaded boat is

$$W_{\text{boat}} = mg = (8560\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 84.0\text{ kN}$$



The buoyancy force becomes a maximum when the entire hull of the boat is submerged in water, and is determined to be

$$F_{B,\text{lake}} = \rho_{\text{lake}} g V = (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(180\text{ m}^3)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 1766\text{ kN}$$

$$F_{B,\text{sea}} = \rho_{\text{sea}} g V = (1030\text{ kg/m}^3)(9.81\text{ m/s}^2)(180\text{ m}^3)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 1819\text{ kN Th}$$

e total weight of a floating boat (load + boat itself) is equal to the buoyancy force. Therefore, the weight of the maximum load is

$$W_{\text{load,lake}} = F_B\text{, lake} - W_{\text{boat}} = 1766 - 84 = 1682\text{ kN}$$
$$W_{\text{load,sea}} = F_{B,\text{sea}} - W_{\text{boat}} = 1819 - 84 = 1735\text{ kN}$$

The corresponding masses of load are

$$m_{\text{load,lake}} = \frac{W_{\text{load,lake}}}{g} = \frac{1682\text{ kN}}{9.81\text{ m/s}^2}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kN}}\right) = 171{,}500\text{ kg}$$

$$m_{\text{load,sea}} = \frac{W_{\text{load,sea}}}{g} = \frac{1735\text{ kN}}{9.81\text{ m/s}^2}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kN}}\right) = 176{,}900\text{ kg}$$

Discussion Note that this boat can carry nearly 5400 kg more load in the sea than it can in fresh water. Fully-loaded boats in sea water should expect to sink into water deeper when they enter fresh water, such as a river where the port may be.

Fluids in Rigid-Body Motion

3-109C

Solution We are to discuss when a fluid can be treated as a rigid body.

Analysis A moving body of fluid can be treated as a rigid body **when there are no shear stresses (i.e., no motion between fluid layers relative to each other) in the fluid body**.

Discussion When there is no relative motion between fluid particles, there are no viscous stresses, and pressure (normal stress) is the only stress.

3-110C

Solution We are to compare the pressure at the bottom of a glass of water moving at various velocities.

Analysis The water pressure at the bottom surface is **the same for all cases** since the acceleration for all four cases is zero.

Discussion When any body, fluid or solid, moves at constant velocity, there is no acceleration, regardless of the direction of the movement.

3-111C

Solution We are to compare the pressure in a glass of water for stationary and accelerating conditions.

Analysis The pressure at the bottom surface is constant when the glass is stationary. For a glass moving on a horizontal plane with constant acceleration, water will collect at the back but the water depth will remain constant at the center. Therefore, the pressure at the midpoint will be the same for both glasses. But **the bottom pressure will be low at the front relative to the stationary glass, and high at the back** (again relative to the stationary glass). Note that the pressure in all cases is the hydrostatic pressure, which is directly proportional to the fluid height.

Discussion We ignore any sloshing of the water.

3-112C

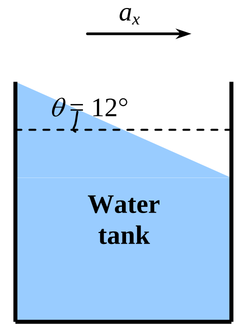
Solution We are to analyze the pressure in a glass of water that is rotating.

Analysis When a vertical cylindrical container partially filled with water is rotated about its axis and rigid body motion is established, the fluid level will drop at the center and rise towards the edges. Noting that hydrostatic pressure is proportional to fluid depth, **the pressure at the mid point will drop and the pressure at the edges of the bottom surface will rise due to the rotation**.

Discussion The highest pressure occurs at the bottom corners of the container.

3-113

Solution A water tank is being towed by a truck on a level road, and the angle the free surface makes with the horizontal is measured. The acceleration of the truck is to be determined.



Assumptions **1** The road is horizontal so that acceleration has no vertical component ($a_z = 0$). **2** Effects of splashing, breaking, driving over bumps, and climbing hills are assumed to be secondary, and are not considered. **3** The acceleration remains constant.

Analysis We take the *x*-axis to be the direction of motion, the *z*-axis to be the upward vertical direction. The tangent of the angle the free surface makes with the horizontal is

$$\tan\theta = \frac{a_x}{g + a_z}$$

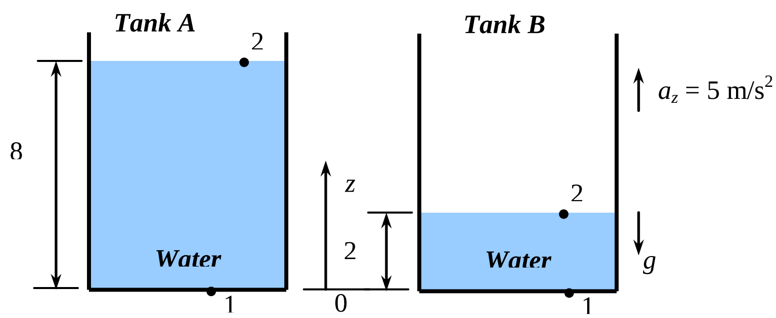
Solving for a_x and substituting,

$$a_x = (g + a_z)\tan\theta = (9.81\,\text{m/s}^2 + 0)\tan 12° = \mathbf{2.09\,m/s^2}$$

Discussion Note that the analysis is valid for any fluid with constant density since we used no information that pertains to fluid properties in the solution.

3-114

Solution Two water tanks filled with water, one stationary and the other moving upwards at constant acceleration. The tank with the higher pressure at the bottom is to be determined.



Assumptions **1** The acceleration remains constant. **2** Water is an incompressible substance.

Properties We take the density of water to be 1000 kg/m^3.

Analysis The pressure difference between two points 1 and 2 in an incompressible fluid is given by

$$P_2 - P_1 = -\rho a_x(x_2 - x_1) - \rho(g + a_z)(z_2 - z_1) \qquad \text{or} \qquad P_1 - P_2 = \rho(g + a_z)(z_2 - z_1)$$

since $a_x = 0$. Taking point 2 at the free surface and point 1 at the tank bottom, we have $P_2 = P_{atm}$ and $z_2 - z_1 = h$ and thus

$$P_{1,\,\text{gage}} = P_{\text{bottom}} = \rho(g + a_z)h$$

Tank A: We have $a_z = 0$, and thus the pressure at the bottom is

$$P_{A,\ \text{bottom}} = \rho g h_A = (1000\ \text{kg/m}^3)(9.81\,\text{m/s}^2)(8\,\text{m})\left(\frac{1\,\text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 78.5\ \text{kN/m}^2$$

Tank B: We have $a_z = +5$ m/s^2, and thus the pressure at the bottom is

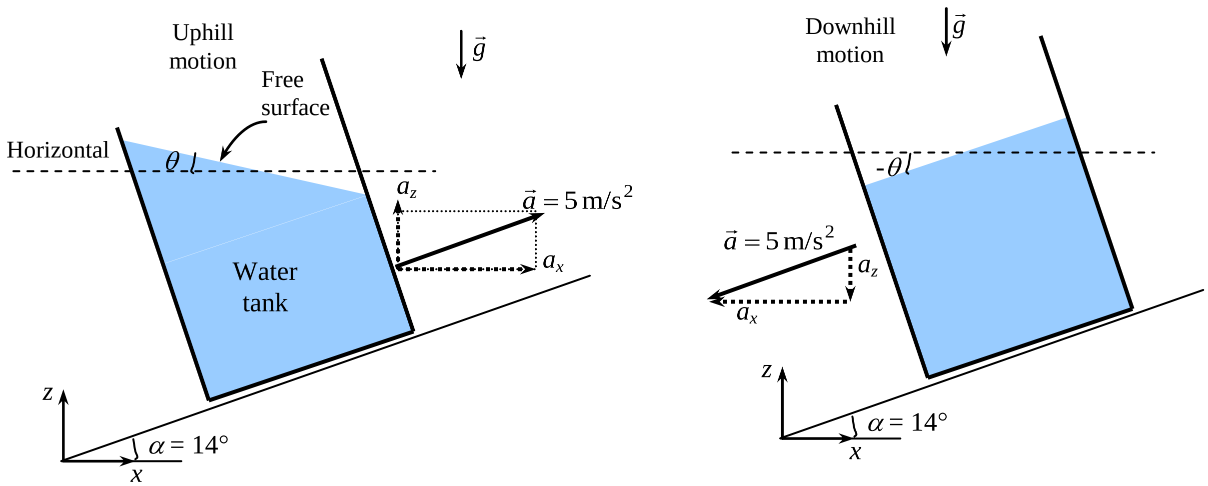
$$P_{B,\ \text{bottom}} = \rho(g + a_z)h_B = (1000\ \text{kg/m}^3)(9.81 + 5\,\text{m/s}^2)(2\,\text{m})\left(\frac{1\,\text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 29.6\ \text{kN/m}^2$$

Therefore, **tank A has a higher pressure at the bottom**.

Discussion We can also solve this problem quickly by examining the relation $P_{\text{bottom}} = \rho(g + a_z)h$. Acceleration for tank B is about 1.5 times that of Tank A (14.81 vs 9.81 m/s^2), but the fluid depth for tank A is 4 times that of tank B (8 m vs 2 m). Therefore, the tank with the larger acceleration-fluid height product (tank A in this case) will have a higher pressure at the bottom.

3-115

Solution A water tank is being towed on an uphill road at constant acceleration. The angle the free surface of water makes with the horizontal is to be determined, and the solution is to be repeated for the downhill motion case.



Assumptions **1** Effects of splashing, breaking, driving over bumps, and climbing hills are assumed to be secondary, and are not considered. **2** The acceleration remains constant.

Analysis We take the *x*- and *z*-axes as shown in the figure. From geometrical considerations, the horizontal and vertical components of acceleration are

$$a_x = a\cos\alpha$$
$$a_z = a\sin\alpha$$

The tangent of the angle the free surface makes with the horizontal is

$$\tan\theta = \frac{a_x}{g + a_z} = \frac{a\cos\alpha}{g + a\sin\alpha} = \frac{(3.5\ \text{m/s}^2)\cos 14°}{9.81\ \text{m/s}^2 + (3.5\ \text{m/s}^2)\sin 14°} = 0.3187 \quad \rightarrow \quad \theta = \mathbf{17.7°}$$

When the direction of motion is reversed, both a_x and a_z are in negative *x*- and *z*-direction, respectively, and thus become negative quantities,

$$a_x = -a\cos\alpha$$
$$a_z = -a\sin\alpha$$

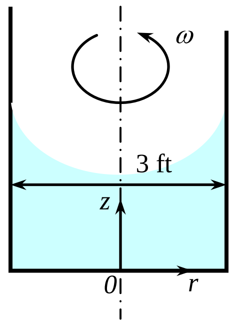
Then the tangent of the angle the free surface makes with the horizontal becomes

$$\tan\theta = \frac{a_x}{g + a_z} = \frac{a\cos\alpha}{g + a\sin\alpha} = \frac{-(3.5\ \text{m/s}^2)\cos 14°}{9.81\ \text{m/s}^2 - (3.5\ \text{m/s}^2)\sin 14°} = -0.3789 \quad \rightarrow \quad \theta = \mathbf{-\ 20.8°}$$

Discussion Note that the analysis is valid for any fluid with constant density, not just water, since we used no information that pertains to water in the solution.

3-116E

Solution A vertical cylindrical tank open to the atmosphere is rotated about the centerline. The angular velocity at which the bottom of the tank will first be exposed, and the maximum water height at this moment are to be determined.



Assumptions **1** The increase in the rotational speed is very slow so that the liquid in the container always acts as a rigid body. **2** Water is an incompressible fluid.

Analysis Taking the center of the bottom surface of the rotating vertical cylinder as the origin (r = 0, z = 0), the equation for the free surface of the liquid is given as

$$z_s(r) = h_0 - \frac{\omega^2}{4g}(R^2 - 2r^2)$$

where h_0 = 1 ft is the original height of the liquid before rotation. Just before dry spot appear at the center of bottom surface, the height of the liquid at the center equals zero, and thus $z_s(0)$ = 0. Solving the equation above for ω and substituting,

$$\omega = \sqrt{\frac{4gh_0]}{R^2}} = \sqrt{\frac{4(32.2\text{ ft/s}^2)(1\text{ ft})}{(1.5\text{ ft})^2}} = 7.566\text{ rad/s} \cong \mathbf{7.57 rad/s}$$

Noting that one complete revolution corresponds to 2π radians, the rotational speed of the container can also be expressed in terms of revolutions per minute (rpm) as

$$\dot{n} = \frac{\omega}{2\pi} = \frac{7.566\text{ rad/s}}{2\pi\text{ rad/rev}}\left(\frac{60\text{ s}}{1\text{ min}}\right) = \mathbf{72.3 rpm}$$

Therefore, the rotational speed of this container should be limited to 72.3 rpm to avoid any dry spots at the bottom surface of the tank.

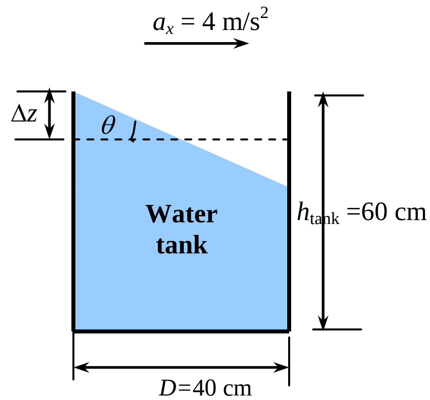
The maximum vertical height of the liquid occurs a the edges of the tank (r = R = 1 ft), and it is

$$z_s(R) = h_0 + \frac{\omega^2 R^2}{4g} = (1\text{ ft}) + \frac{(7.566\text{ rad/s})^2(1.5\text{ ft})^2}{4(32.2\text{ ft/s}^2)} = \mathbf{2.00\text{ ft}}$$

Discussion Note that the analysis is valid for any liquid since the result is independent of density or any other fluid property.

3-117

Solution A cylindrical tank is being transported on a level road at constant acceleration. The allowable water height to avoid spill of water during acceleration is to be determined.



Assumptions **1** The road is horizontal during acceleration so that acceleration has no vertical component (a_z = 0). **2** Effects of splashing, breaking, driving over bumps, and climbing hills are assumed to be secondary, and are not considered. **3** The acceleration remains constant.

Analysis We take the *x*-axis to be the direction of motion, the *z*-axis to be the upward vertical direction, and the origin to be the midpoint of the tank bottom. The tangent of the angle the free surface makes with the horizontal is

$$\tan\theta = \frac{a_x}{g + a_z} = \frac{4}{9.81 + 0} = 0.4077 \quad \text{(and thus } \theta = 22.2°\text{)}$$

The maximum vertical rise of the free surface occurs at the back of the tank, and the vertical midplane experiences no rise or drop during acceleration. Then the maximum vertical rise at the back of the tank relative to the midplane is

$$\Delta z_{max} = (D/2)\tan\theta = [(0.40 \text{ m})/2] \times 0.4077 = 0.082 \text{ m} = 8.2 \text{ cm}$$

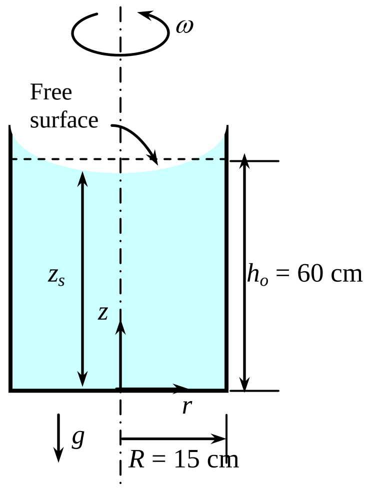
Therefore, the maximum initial water height in the tank to avoid spilling is

$$h_{max} = h_{tank} - \Delta z_{max} = 60 - 8.2 = \mathbf{51.8 \text{ cm}}$$

Discussion Note that the analysis is valid for any fluid with constant density, not just water, since we used no information that pertains to water in the solution.

3-118

Solution A vertical cylindrical container partially filled with a liquid is rotated at constant speed. The drop in the liquid level at the center of the cylinder is to be determined.



Assumptions **1** The increase in the rotational speed is very slow so that the liquid in the container always acts as a rigid body. **2** The bottom surface of the container remains covered with liquid during rotation (no dry spots).

Analysis Taking the center of the bottom surface of the rotating vertical cylinder as the origin (r = 0, z = 0), the equation for the free surface of the liquid is given as

$$z_s(r) = h_0 - \frac{\omega^2}{4g}(R^2 - 2r^2)$$

where h_0 = 0.6 m is the original height of the liquid before rotation, and

$$\omega = 2\pi\dot{n} = 2\pi(180\text{ rev/min})\left(\frac{1\text{ min}}{60\text{ s}}\right) = 18.85\text{ rad/s}$$

Then the vertical height of the liquid at the center of the container where r = 0 becomes

$$z_s(0) = h_0 - \frac{\omega^2 R^2}{4g} = (0.60\text{ m}) - \frac{(18.85\text{ rad/s})^2(0.15\text{ m})^2}{4(9.81\text{ m/s}^2)} = 0.204\text{ m}$$

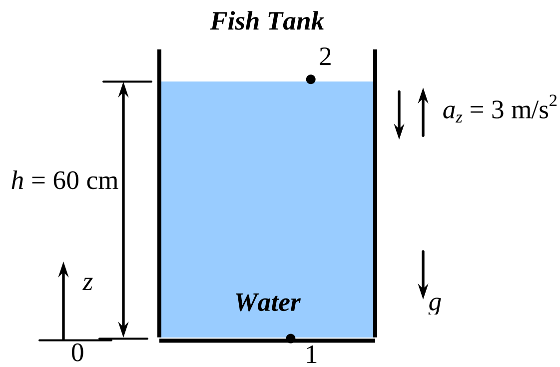
Therefore, the drop in the liquid level at the center of the cylinder is

$$\Delta h_{\text{drop,center}} = h_0 - z_s(0) = 0.60 - 0.204 = \mathbf{0.396\,m}$$

Discussion Note that the analysis is valid for any liquid since the result is independent of density or any other fluid property. Also, our assumption of no dry spots is validated since $z_0(0)$ is positive.

3-119

Solution The motion of a fish tank in the cabin of an elevator is considered. The pressure at the bottom of the tank when the elevator is stationary, moving up with a specified acceleration, and moving down with a specified acceleration is to be determined.



Assumptions **1** The acceleration remains constant. **2** Water is an incompressible substance.

Properties We take the density of water to be 1000 kg/m^3.

Analysis The pressure difference between two points 1 and 2 in an incompressible fluid is given by

$$P_2 - P_1 = -\rho a_x (x_2 - x_1) - \rho(g + a_z)(z_2 - z_1) \quad \text{or} \quad P_1 - P_2 = \rho(g + a_z)(z_2 - z_1)$$

since $a_x = 0$. Taking point 2 at the free surface and point 1 at the tank bottom, we have $P_2 = P_{atm}$ and $z_2 - z_1 = h$ and thus

$$P_{1,\text{gage}} = P_{\text{bottom}} = \rho(g + a_z)h$$

(***a***) **Tank stationary**: We have $a_z = 0$, and thus the gage pressure at the tank bottom is

$$P_{\text{bottom}} = \rho g h = (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.6\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 5.89\ \text{kN/m}^2 = \mathbf{5.89\ kPa}$$

(***b***) **Tank moving up**: We have a_z = +3 m/s^2, and thus the gage pressure at the tank bottom is

$$P_{\text{bottom}} = \rho(g + a_z)h_B = (1000\ \text{kg/m}^3)(9.81 + 3\ \text{m/s}^2)(0.6\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 7.69\ \text{kN/m}^2 = \mathbf{7.69\ kPa}$$

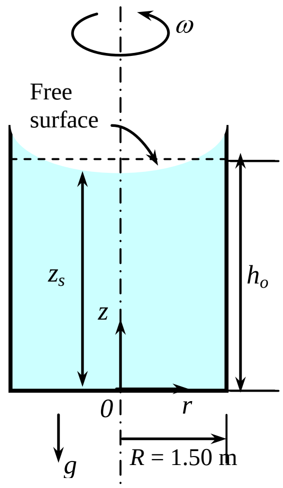
(***c***) **Tank moving down**: We have a_z = -3 m/s^2, and thus the gage pressure at the tank bottom is

$$P_{\text{bottom}} = \rho(g + a_z)h_B = (1000\ \text{kg/m}^3)(9.81 - 3\ \text{m/s}^2)(0.6\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 4.09\ \text{kN/m}^2 = \mathbf{4.09\ kPa}$$

Discussion Note that the pressure at the tank bottom while moving up in an elevator is almost twice that while moving down, and thus the tank is under much greater stress during upward acceleration.

3-120

Solution A vertical cylindrical milk tank is rotated at constant speed, and the pressure at the center of the bottom surface is measured. The pressure at the edge of the bottom surface is to be determined.



Assumptions **1** The increase in the rotational speed is very slow so that the liquid in the container always acts as a rigid body. **2** Milk is an incompressible substance.

Properties The density of the milk is given to be 1030 kg/m^3.

Analysis Taking the center of the bottom surface of the rotating vertical cylinder as the origin (r = 0, z = 0), the equation for the free surface of the liquid is given as

$$z_s(r) = h_0 - \frac{\omega^2}{4g}(R^2 - 2r^2)$$

where R = 1.5 m is the radius, and

$$\omega = 2\pi\dot{n} = 2\pi(12 \text{ rev/min})\left(\frac{1 \text{ min}}{60 \text{ s}}\right) = 1.2566 \text{ rad/s}$$

The fluid rise at the edge relative to the center of the tank is

$$\Delta h = z_s(R) - z_s(0) = \left(h_0 + \frac{\omega^2 R^2}{4g}\right) - \left(h_0 - \frac{\omega^2 R^2}{4g}\right) = \frac{\omega^2 R^2}{2g} = \frac{(1.2566 \text{ rad/s})^2 (1.50 \text{ m})^2}{2(9.81 \text{ m/s}^2)} = 1.1811 \text{ m}$$

The pressure difference corresponding to this fluid height difference is

$$\Delta P_{\text{bottom}} = \rho g \Delta h = (1030 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(1.1811 \text{ m})\left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2}\right) = 1.83 \text{ kN/m}^2 = 1.83 \text{ kPa}$$

Then the pressure at the edge of the bottom surface becomes

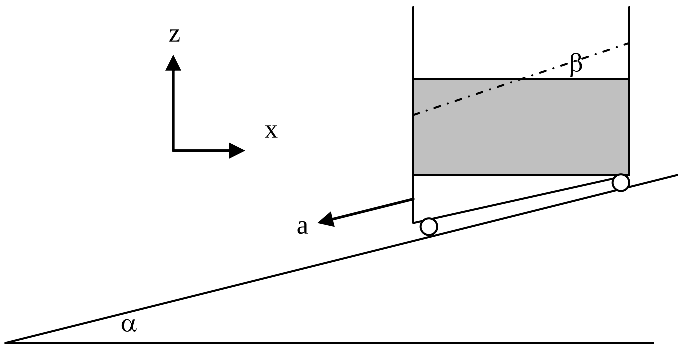
$$P_{\text{bottom, edge}} = P_{\text{bottom, center}} + \Delta P_{\text{bottom}} = 130 + 1.83 = 131.83 \text{ kPa} \cong \mathbf{132\ kPa}$$

Discussion Note that the pressure is 1.4% higher at the edge relative to the center of the tank, and there is a fluid level difference of 1.18 m between the edge and center of the tank, and these differences should be considered when designing rotating fluid tanks.

3-121

Solution A tank of rectangular cross-section partially filled with a liquid placed on an inclined surface is considered. It is to be shown that the slope of the liquid surface will be the same as the slope of the inclined surface when the tank is released.

Analysis



$$\tan\beta = \frac{-a_y}{g + a_z} = -\frac{-a\cos\alpha}{g - a\sin\alpha}$$

Since $a = g\sin\alpha$, we get

$$\tan\beta = \frac{g\sin\alpha\cos\alpha}{g - g\sin\alpha\sin\alpha} = \frac{\sin\alpha\cos\alpha}{1-\sin^2\alpha} = \frac{\sin\alpha\cos\alpha}{\cos^2\alpha} = \tan\alpha$$

Therefore $\alpha = \beta$

If the surface were rough, $a' = g\sin\alpha - \lambda g\cos\alpha < a$, where λ is the surface friction coefficient. Therefore we may conclude that $\beta < \alpha$

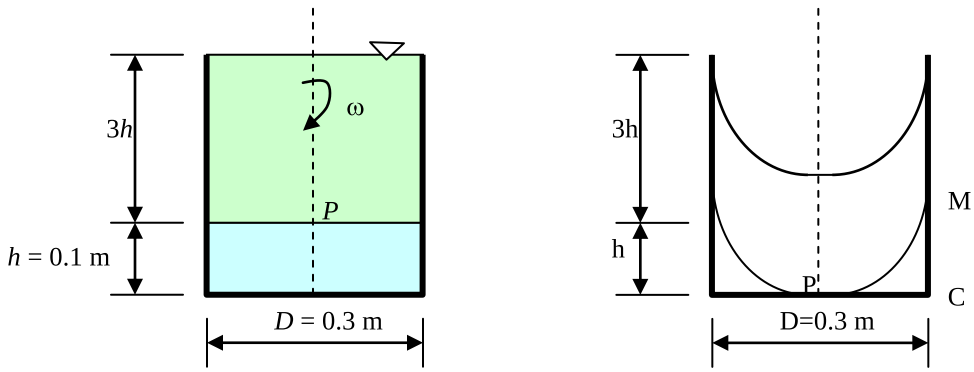
3-122

Solution The bottom quarter of a vertical cylindrical tank is filled with oil and the rest with water. The tank is now rotated about its vertical axis at a constant angular speed. The value of the angular speed when the point *P* on the axis at the oil-water interface touches the bottom of the tank and the amount of water that would be spilled out at this angular speed are to be determined.

Assumptions **1** The acceleration remains constant. **2** Water is an incompressible substance.

Properties We take the density of water to be 1000 kg/m^3.

Analysis



When the steady-state conditions are achieved, the shape of the isobaric surface will be as below:

The volume of oil does not change, and we write

$$\pi\frac{D^2}{4}h = \frac{1}{2}\pi\frac{D^2}{4}(CM)$$, from which we get *CM=2h*=0.20 m

Two surfaces will be parallel to each other since the fluid interface is an isobar surface. Therefore the amount of water that spilled from the tank will be half of volume of CM paraboloid, that is

$$\forall = \frac{1}{2}\pi\frac{D^2}{4}2h = \pi\frac{D^2}{4}h = \pi\frac{0.3^2}{4}0.2 = \mathbf{0.0141 m^3}$$

The pressure difference between point P and C can be expressed as

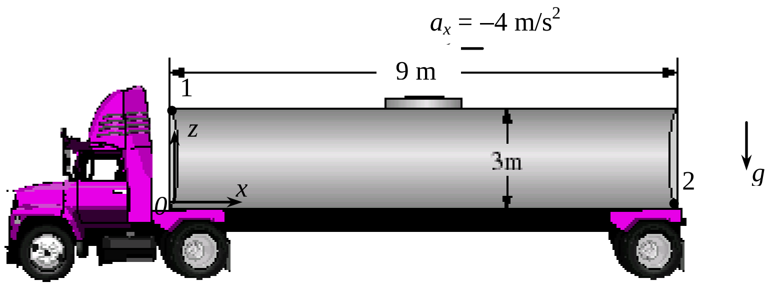
$$\frac{p_c - p_p}{\rho g} = \frac{\omega^2}{2g}\left(r_c^2 - r_p^2\right) = 2h$$

Solving for angular velocity gives

$$\omega = \sqrt{\frac{4gh}{r_c^2}} = \sqrt{\frac{4\times 9.81\times 0.1}{0.15^2}} = \mathbf{13.2\ rad/s}$$

3-123

Solution Milk is transported in a completely filled horizontal cylindrical tank accelerating at a specified rate. The maximum pressure difference in the tanker is to be determined.



Assumptions **1** The acceleration remains constant. **2** Milk is an incompressible substance.

Properties The density of the milk is given to be 1020 kg/m^3.

Analysis We take the *x*- and *z*- axes as shown. The horizontal acceleration is in the negative *x* direction, and thus a_x is negative. Also, there is no acceleration in the vertical direction, and thus $a_z = 0$. The pressure difference between two points 1 and 2 in an incompressible fluid in linear rigid body motion is given by

$$P_2 - P_1 = -\rho a_x (x_2 - x_1) - \rho(g + a_z)(z_2 - z_1) \quad \rightarrow \quad P_2 - P_1 = -\rho a_x (x_2 - x_1) - \rho g(z_2 - z_1)$$

The first term is due to acceleration in the horizontal direction and the resulting compression effect towards the back of the tanker, while the second term is simply the hydrostatic pressure that increases with depth. Therefore, we reason that the lowest pressure in the tank will occur at point 1 (upper front corner), and the higher pressure at point 2 (the lower rear corner). Therefore, the maximum pressure difference in the tank is

$$\Delta P_{\max} = P_2 - P_1 = -\rho a_x (x_2 - x_1) - \rho g(z_2 - z_1) = -[a_x (x_2 - x_1) + g(z_2 - z_1)]$$

$$= -(1020\ \text{kg/m}^3)\left[(-4\ \text{m/s}^2)(9\ \text{m}) + (9.81\ \text{m/s}^2)(-3\ \text{m})\right]\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= (36.72 + 30.0)\ \text{kN/m}^2 = \mathbf{66.7\ kPa}$$

since $x_1 = 0$, $x_2 = 9$ m, $z_1 = 3$ m, and $z_2 = 0$.

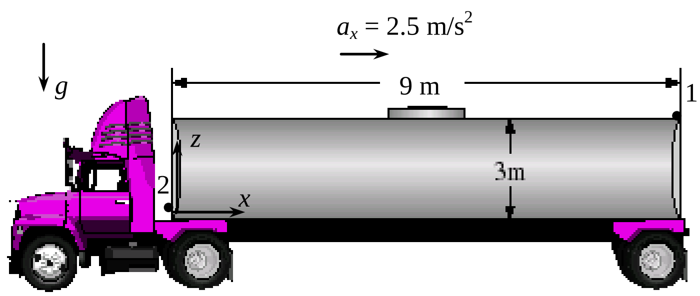
Discussion Note that the variation of pressure along a horizontal line is due to acceleration in the horizontal direction while the variation of pressure in the vertical direction is due to the effects of gravity and acceleration in the vertical direction (which is zero in this case).

3-124

Solution Milk is transported in a completely filled horizontal cylindrical tank decelerating at a specified rate. The maximum pressure difference in the tanker is to be determined.

Assumptions **1** The acceleration remains constant. **2** Milk is an incompressible substance.

Properties The density of the milk is given to be 1020 kg/m^3.



Analysis We take the x- and z- axes as shown. The horizontal deceleration is in the x direction, and thus a_x is positive. Also, there is no acceleration in the vertical direction, and thus a_z = 0. The pressure difference between two points 1 and 2 in an incompressible fluid in linear rigid body motion is given by

$$P_2 - P_1 = -\rho a_x(x_2 - x_1) - \rho(g + a_z)(z_2 - z_1) \quad \rightarrow \quad P_2 - P_1 = -\rho a_x(x_2 - x_1) - \rho g(z_2 - z_1)$$

The first term is due to deceleration in the horizontal direction and the resulting compression effect towards the front of the tanker, while the second term is simply the hydrostatic pressure that increases with depth. Therefore, we reason that the lowest pressure in the tank will occur at point 1 (upper front corner), and the higher pressure at point 2 (the lower rear corner). Therefore, the maximum pressure difference in the tank is

$$\Delta P_{\max} = P_2 - P_1 = -\rho a_x(x_2 - x_1) - \rho g(z_2 - z_1) = -[a_x(x_2 - x_1) + g(z_2 - z_1)]$$

$$= -(1020\text{ kg/m}^3)\left[(2.5\text{ m/s}^2)(-9\text{ m}) + (9.81\text{ m/s}^2)(-3\text{ m})\right]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= (22.95 + 30.0)\text{ kN/m}^2 = \mathbf{53.0\,kPa}$$

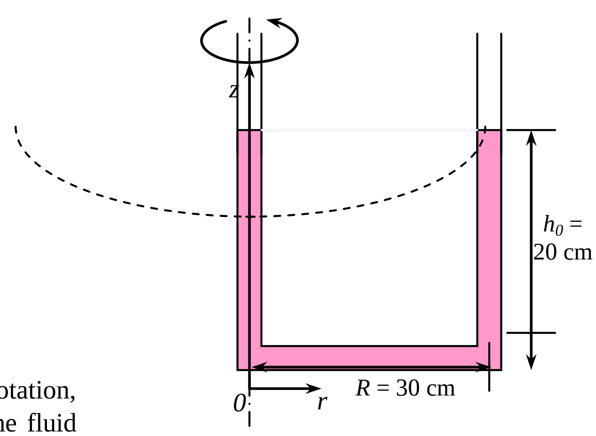
since x_1 = 9 m, x_2 = 0, z_1 = 3 m, and z_2 = 0.

Discussion Note that the variation of pressure along a horizontal line is due to acceleration in the horizontal direction while the variation of pressure in the vertical direction is due to the effects of gravity and acceleration in the vertical direction (which is zero in this case).

3-125

Solution A vertical U-tube partially filled with alcohol is rotated at a specified rate about one of its arms. The elevation difference between the fluid levels in the two arms is to be determined.

Assumptions **1** Alcohol is an incompressible fluid.



Analysis Taking the base of the left arm of the U-tube as the origin (r = 0, z = 0), the equation for the free surface of the liquid is given as

$$z_s(r) = h_0 - \frac{\omega^2}{4g}(R^2 - 2r^2)$$

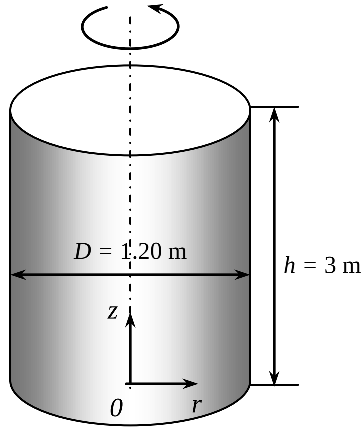
where h_0 = 0.20 m is the original height of the liquid before rotation, and ω = 4.2 rad/s. The fluid rise at the right arm relative to the fluid level in the left arm (the center of rotation) is

$$\Delta h = z_s(R) - z_s(0) = \left(h_0 + \frac{\omega^2 R^2}{4g}\right) - \left(h_0 - \frac{\omega^2 R^2}{4g}\right) = \frac{\omega^2 R^2}{2g} = \frac{(4.2\text{ rad/s})^2(0.30\text{ m})^2}{2(9.81\text{ m/s}^2)} = \mathbf{0.081\,m}$$

Discussion The analysis is valid for any liquid since the result is independent of density or any other fluid property.

3-126

Solution A vertical cylindrical tank is completely filled with gasoline, and the tank is rotated about its vertical axis at a specified rate. The pressures difference between the centers of the bottom and top surfaces, and the pressures difference between the center and the edge of the bottom surface are to be determined.



Assumptions **1** The increase in the rotational speed is very slow so that the liquid in the container always acts as a rigid body. **2** Gasoline is an incompressible substance.

Properties The density of the gasoline is given to be 740 kg/m^3.

Analysis The pressure difference between two points 1 and 2 in an incompressible fluid rotating in rigid body motion is given by

$$P_2 - P_1 = \frac{\rho\omega^2}{2}(r_2^2 - r_1^2) - \rho g(z_2 - z_1)$$

where R = 0.60 m is the radius, and

$$\omega = 2\pi\dot{n} = 2\pi(70\text{ rev/min})\left(\frac{1\text{ min}}{60\text{ s}}\right) = 7.330\text{ rad/s}$$

(*a*) Taking points 1 and 2 to be the centers of the bottom and top surfaces, respectively, we have $r_1 = r_2 = 0$ and $z_2 - z_1 = h = 3\text{ m}$. Then,

$$P_{\text{center, top}} - P_{\text{center, bottom}} = 0 - \rho g(z_2 - z_1) = -\rho g h$$

$$= -(740\text{ kg/m}^3)(9.81\text{ m/s}^2)(3\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 21.8\text{ kN/m}^2 = \mathbf{21.8\ kPa}$$

(*b*) Taking points 1 and 2 to be the center and edge of the bottom surface, respectively, we have $r_1 = 0$, $r_2 = R$, and $z_2 = z_1 = 0$. Then,

$$P_{\text{edge,bottom}} - P_{\text{center, bottom}} = \frac{\rho\omega^2}{2}(R_2^2 - 0) - 0 = \frac{\rho\omega^2 R^2}{2}$$

$$= \frac{(740\text{ kg/m}^3)(7.33\text{ rad/s})^2(0.60\text{ m})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 7.16\text{ kN/m}^2 = \mathbf{7.16\ kPa}$$

Discussion Note that the rotation of the tank does not affect the pressure difference along the axis of the tank. But the pressure difference between the edge and the center of the bottom surface (or any other horizontal plane) is due entirely to the rotation of the tank.

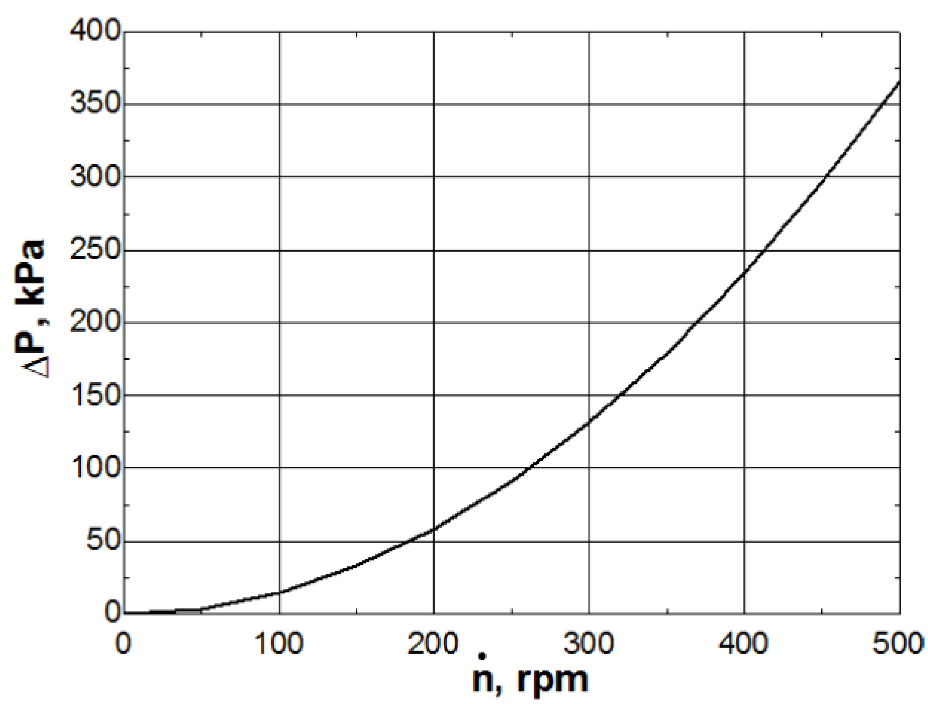
3-127

Solution The previous problem is reconsidered. The effect of rotational speed on the pressure difference between the center and the edge of the bottom surface of the cylinder as the rotational speed varies from 0 to 500 rpm in increments of 50 rpm is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=740 "kg/m3"
R=0.6 "m"
h=3 "m"
omega=2*pi*n_dot/60 "rad/s"
DeltaP_axis=rho*g*h/1000 "kPa"
DeltaP_bottom=rho*omega^2*R^2/2000 "kPa"
```

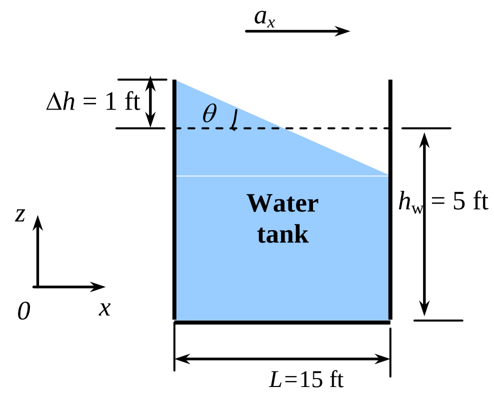
Rotation rate $\dot{n}$, rpm	Angular speed ω, rad/s	$\Delta P_{\text{center-edge}}$ kPa
0	0.0	0.0
50	5.2	3.7
100	10.5	14.6
150	15.7	32.9
200	20.9	58.4
250	26.2	91.3
300	31.4	131.5
350	36.7	178.9
400	41.9	233.7
450	47.1	295.8
500	52.4	365.2



Discussion The pressure rise with rotation rate is not linear, but rather quadratic.

3-128E

Solution A water tank partially filled with water is being towed by a truck on a level road. The maximum acceleration (or deceleration) of the truck to avoid spilling is to be determined.



Assumptions **1** The road is horizontal so that acceleration has no vertical component (a_z = 0). **2** Effects of splashing, breaking, driving over bumps, and climbing hills are assumed to be secondary, and are not considered. **3** The acceleration remains constant.

Analysis We take the x-axis to be the direction of motion, the z-axis to be the upward vertical direction. The shape of the free surface just before spilling is shown in figure. The tangent of the angle the free surface makes with the horizontal is given by

$$\tan\theta = \frac{a_x}{g + a_z} \quad \rightarrow \quad a_x = g\tan\theta$$

where a_z = 0 and, from geometric considerations, $\tan\theta$ is $\tan\theta = \dfrac{\Delta h}{L/2}$. Substituting, we get

$$a_x = g\tan\theta = g\frac{\Delta h}{L/2} = (32.2\text{ ft/s}^2)\frac{1\text{ ft}}{(15\text{ ft})/2} = \mathbf{4.29\ ft/s^2}$$

The solution can be repeated for deceleration by replacing a_x by $-a_x$. We obtain a_x = **–4.29 m/s²**.

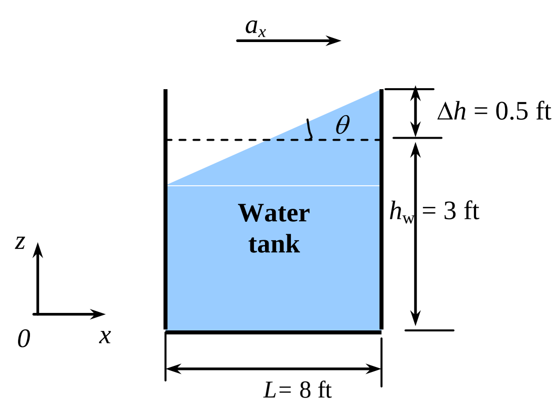
Discussion Note that the analysis is valid for any fluid with constant density since we used no information that pertains to fluid properties in the solution.

3-129E

Solution A water tank partially filled with water is being towed by a truck on a level road. The maximum acceleration (or deceleration) of the truck to avoid spilling is to be determined.

Assumptions **1** The road is horizontal so that deceleration has no vertical component (a_z = 0). **2** Effects of splashing and driving over bumps are assumed to be secondary, and are not considered. **3** The deceleration remains constant.

Analysis We take the x-axis to be the direction of motion, the z-axis to be the upward vertical direction. The shape of the free surface just before spilling is shown in figure. The tangent of the angle the free surface makes with the horizontal is given by



$$\tan\theta = \frac{-a_x}{g + a_z} \quad \rightarrow \quad a_x = -g\tan\theta$$

where a_z = 0 and, from geometric considerations, $\tan\theta$ is

$$\tan\theta = \frac{\Delta h}{L/2}$$

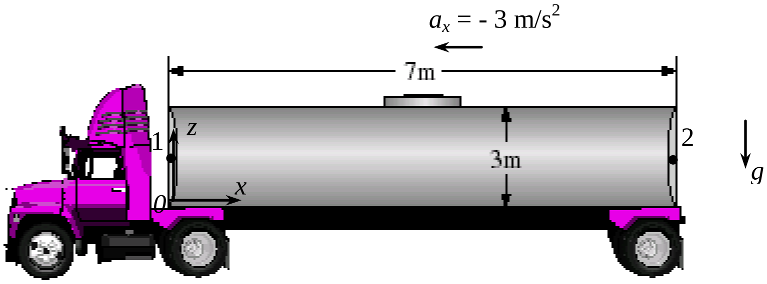
Substituting,

$$a_x = -g\tan\theta = -g\frac{\Delta h}{L/2} = -(32.2\text{ ft/s}^2)\frac{0.5\text{ ft}}{(8\text{ ft})/2} = -4.025\text{ ft/s}^2 \cong \mathbf{-4.03\ ft/s^2}$$

Discussion Note that the analysis is valid for any fluid with constant density since we used no information that pertains to fluid properties in the solution.

3-130

Solution Water is transported in a completely filled horizontal cylindrical tanker accelerating at a specified rate. The pressure difference between the front and back ends of the tank along a horizontal line when the truck accelerates and decelerates at specified rates.



Assumptions **1** The acceleration remains constant. **2** Water is an incompressible substance.

Properties We take the density of the water to be 1000 kg/m^3.

Analysis (*a*) We take the *x*- and *z*- axes as shown. The horizontal acceleration is in the negative *x* direction, and thus a_x is negative. Also, there is no acceleration in the vertical direction, and thus a_z = 0. The pressure difference between two points 1 and 2 in an incompressible fluid in linear rigid body motion is given by

$$P_2 - P_1 = -\rho a_x (x_2 - x_1) - \rho(g + a_z)(z_2 - z_1) \quad \rightarrow \quad P_2 - P_1 = -\rho a_x (x_2 - x_1)$$

since z_2 - z_1 = 0 along a horizontal line. Therefore, the pressure difference between the front and back of the tank is due to acceleration in the horizontal direction and the resulting compression effect towards the back of the tank. Then the pressure difference along a horizontal line becomes

$$\Delta P = P_2 - P_1 = -\rho a_x (x_2 - x_1) = -(1000 \text{ kg/m}^3)(-3 \text{ m/s}^2)(7 \text{ m})\left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2}\right) = 21 \text{ kN/m}^2 = \mathbf{21\ kPa}$$

since x_1 = 0 and x_2 = 7 m.

(*b*) The pressure difference during deceleration is determined the way, but a_x = 4 m/s^2 in this case,

$$\Delta P = P_2 - P_1 = -\rho a_x (x_2 - x_1) = -(1000 \text{ kg/m}^3)(4 \text{ m/s}^2)(7 \text{ m})\left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2}\right) = -28 \text{ kN/m}^2 = \mathbf{-28\ kPa}$$

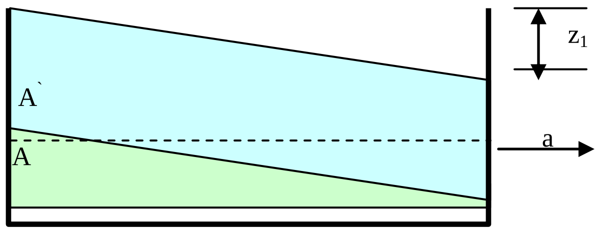
Discussion Note that the pressure is higher at the back end of the tank during acceleration, but at the front end during deceleration (during breaking, for example) as expected.

3-131

Solution A rectangular tank is filled with heavy oil at the bottom and water at the top. The tank is now moved to the right horizontally with a constant acceleration and some water is spilled out as a result from the back. The height of the point A at the back of the tank on the oil-water interface that will rise under this acceleration is to be determined.

Assumptions **1** The acceleration remains constant. **2** Water and oil are incompressible substances.

Analysis



Before the acceleration the water volume for unit width was $1\text{m} \times \text{L}$. Therefore ¼ of this volume must be equal to the emptied volume in the tank, which is $1/2 \times \text{L} \times \text{z}_1$. Equating two equations we get $\text{z}_1 = 0.5 \text{ m}$

The slope of the free surface is

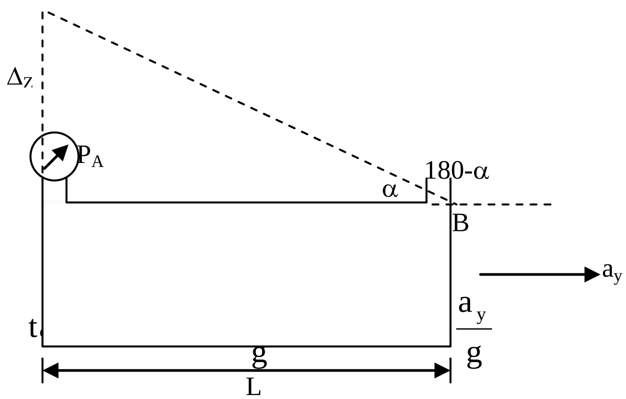
$$\tan\alpha = \frac{0.5}{L} = \frac{AA'}{L/2}, \text{ and } AA'=0.25 \text{ m.}$$

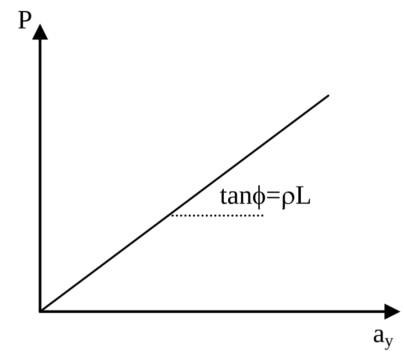
3-132

Solution A sealed box filled with a liquid is considered. A relation between the pressure P_A and the acceleration a is to be obtained.

Assumptions **1** The acceleration remains constant. **2** The liquid is an incompressible substance.

Analysis





$$\frac{\Delta_Z}{L} = \tan\alpha$$

$$P_A = \gamma.\Delta_Z$$

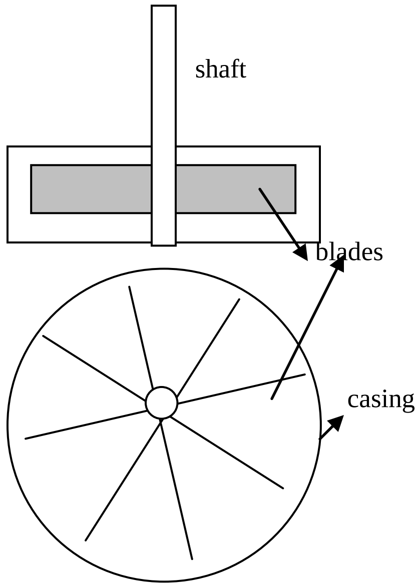
$$P_A = \gamma.L.\tan\alpha$$

$$P_A = \gamma.L.\frac{|a_y|}{g} = \rho.L.a_y$$

3-133

Solution The shaft of a centrifugal pump is rotated at a constant speed. The theoretical pum head due to this rotation is to be determined.

Analysis



$$n\text{=2400 rpm},\quad \omega = \frac{n\pi}{30} = \frac{2400\pi}{30} = 251.3\ rad/s$$

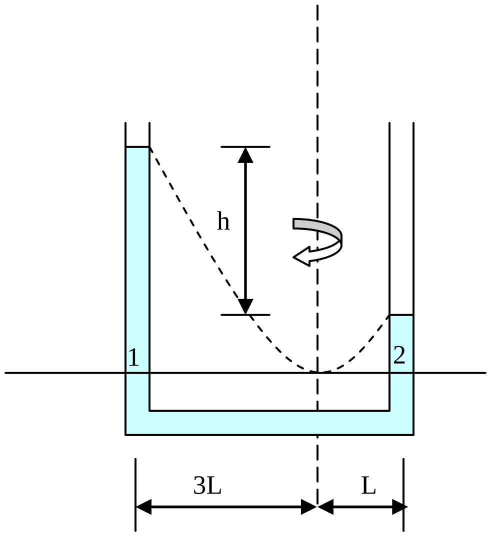
Pump head is then

$$H = \frac{\omega^2 R^2}{2g} = \frac{251.3^2 \times (0.35/2)^2}{19.62} = \mathbf{98.5\ m}$$

3-134

Solution A U-tube is rotating at a constant angular velocity of ω. A relation for ω in terms of g, h, and L is to be obtained.

Analysis



$$z_1 - z_2 = \frac{\omega^2}{2g}\left(r_1^2 - r_2^2\right) = \frac{p_1 - p_2}{\rho g} = h$$

$$\frac{\omega^2}{2g}\left(9L^2 - L^2\right) = h\,,$$

$$\omega = \frac{\sqrt{gh}}{2L}$$

Review Problems

3-135

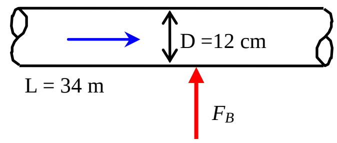
Solution One section of the duct of an air-conditioning system is laid underwater. The upward force the water exerts on the duct is to be determined.

Assumptions **1** The diameter given is the outer diameter of the duct (or, the thickness of the duct material is negligible). **2** The weight of the duct and the air in is negligible.

Properties The density of air is given to be $\rho = 1.30$ kg/m^3. We take the density of water to be 1000 kg/m^3.

Analysis Noting that the weight of the duct and the air in it is negligible, the net upward force acting on the duct is the buoyancy force exerted by water. The volume of the underground section of the duct is

$$V = AL = (\pi D^2/4)L = [\pi(0.12\text{ m})^2/4](34\text{ m}) = 0.3845\text{ m}^3$$



Then the buoyancy force becomes

$$F_B = \rho g V = (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.3845\text{ m}^3)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{3.77\text{ kN}}$$

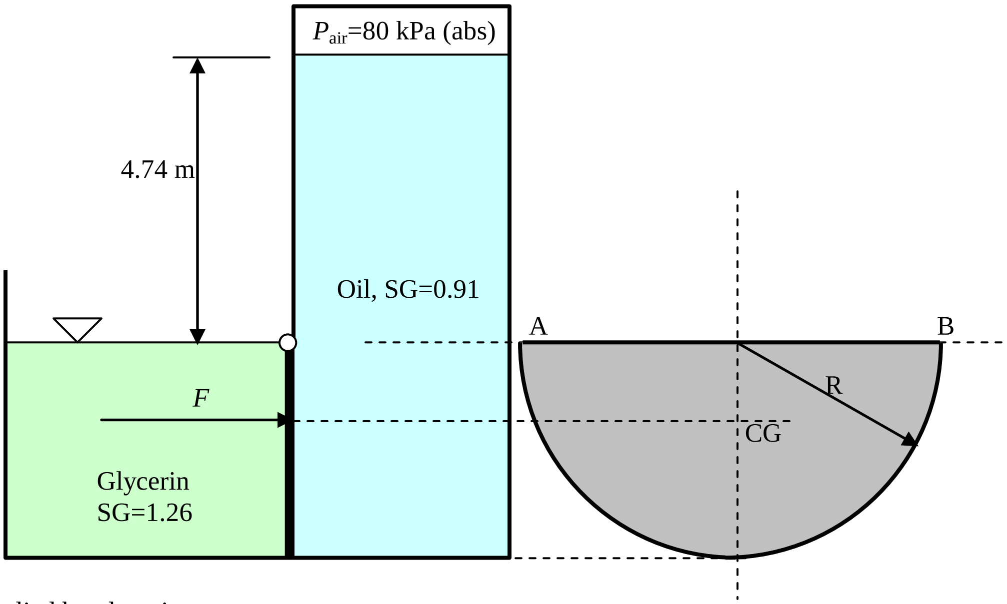
Discussion The upward force exerted by water on the duct is 3.77 kN, which is equivalent to the weight of a mass of 354 kg. Therefore, this force must be treated seriously.

3-136

Solution A semi-circular gate is hinged. The required force at the center of gravity to keep the gate closed is to be determined.

Properties The specific gravities of oil and glycerin are given in the figure.

Analysis



Force applied by glycerin

$$F_{Rg} = \gamma h_{cg} A = 1.26 \times 9810 \times \frac{4 \times 0.5}{3 \times \pi} \times \frac{\pi \times 0.5^2}{2} = 1030\ N$$

The gage pressure of air entrapped on the top of the oil surface

$$p = 80 - 100 = -20\ kPa\ (gage)$$

This negative pressure would result in an imaginary reduction in the oil level by

$$h = \frac{20000}{0.91 \times 9810} = 2.24\ m$$

therefore the imaginary oil level would be $H = 4.74 - 2.24 = 2.50\ m$ from glycerin surface. The force applied by oil is then

$$F_{Ro} = \gamma h_{cg} A = 0.91 \times 9810 \times \left(2.5 + \frac{4 \times 0.5}{3 \times \pi}\right) \times \frac{\pi \times 0.5^2}{2} = 9508\ N$$

Locations of F_{Rg} and F_{Ro}:

$$A = \frac{1}{2}\pi 0.5^2 = 0.39267\ m^2$$

$$I_{xc} = 0.1098 R^4 = 0.1098 \times 0.5^4 = 0.0068625\ m^4$$

$$y_{cp-g} = y_{cg-g} + \frac{I_{xc}}{y_{cg-g} A} = 0.2122 + \frac{0.0068625}{0.2122 \times 0.39267} = 0.2945\ m$$

$$y_{cg-g} = \frac{4 \times 0.5}{3 \times \pi} = 0.2122\ m$$

$$y_{cg-o} = 2.5 + \frac{4 \times 0.5}{3 \times \pi} = 2.712 \ m$$

$$y_{cp-o} = y_{cg-o} + \frac{I_{xc}}{y_{cg-o} A} = 2.712 + \frac{0.0068625}{2.712 \times 0.39267} = 2.784 \ m$$

Moment about hinge would give

$$F \times y_{cg-g} + F_{Rg} \times y_{cp-g} - F_{Ro} \left(y_{cp-o} - 2.5 \right) = 0$$

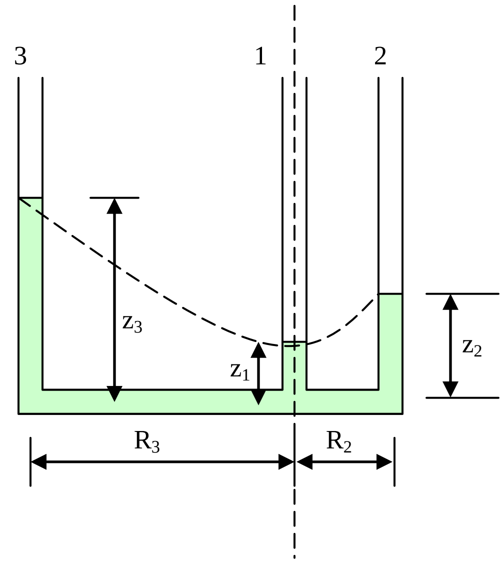
$$F \times 0.2122 + 1030 \times 0.2945 - 9508 (2.784 - 2.5) = 0$$

$$F = 11296 \ N \cong \mathbf{11.3 kN}$$

3-137

Solution The water height in each tube leg of a 3-tube system and the rotational speed at which the middle tube is empty are to be determined.

Analysis



The equation describing the water surface is

$$z = \frac{\omega^2 r^2}{2g} + C$$

Since z=z_1 when r=0, C=z_1. Therefore we can write the following expressions:

$$z_2 = z_1 + \frac{\omega^2 R_2^2}{2g} \tag{1}$$

$$z_3 = z_1 + \frac{\omega^2 R_3^2}{2g} \tag{2}$$

There are 3 unknowns (z_1, z_2, z_3). The 3rd equation will be obtained from continuity such as

$$3h = z_1 + z_2 + z_3 \tag{3}$$

Substituting Eqs. 1 and 2 into Eq. 3 we have

$$3h = 3z_1 + \frac{\omega^2}{2g}\left(R_2^2 + R_3^2\right)$$

For the given data we obtain $z_1 = \mathbf{0.065\ m}$, $z_2 = \mathbf{0.116\ m}$, $z_3 = \mathbf{0.269\ m}$

For this case z_1=0. Therefore, from Eq. 3,

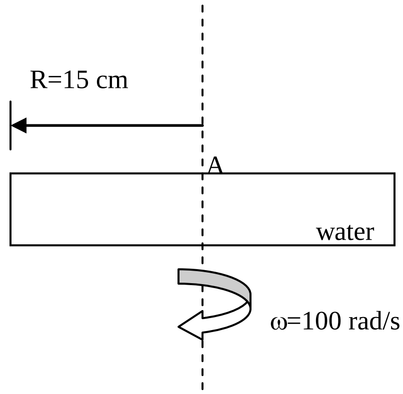
$$3h = z_2 + z_3 = \frac{\omega^2}{2g}\left(R_2^2 + R_3^2\right) = \frac{\omega^2}{2g}\left(0.20^2 + 0.10^2\right)$$

$$\omega = \mathbf{13.3\ rad/s}$$

3-138

Solution A vertical cylindrical vessel is rotated at a constant angular velocity. The total upward force acting upon the entire top surface inside the cylinder is to be determined.

Analysis



Since z=constant along the top surface, we may write

$$P = \rho \frac{\omega^2 r^2}{2} + C$$

If we take the point A to be reference, then C=0.

$$P = 100\,\frac{100^2 r^2}{2} = 5\times 10^6 r^2$$

$$dF = PdA = P2\pi r dr, \text{ or } \quad F = 10^7 \pi \int_0^R r^3 dr = 3976 \ N$$

3-139

Solution A helium balloon tied to the ground carries 2 people. The acceleration of the balloon when it is first released is to be determined.

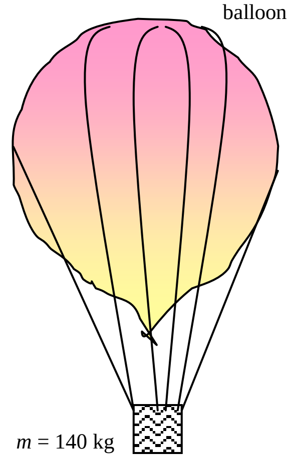
Assumptions The weight of the cage and the ropes of the balloon is negligible.

Properties The density of air is given to be $\rho = 1.16$ kg/m^3. The density of helium gas is 1/7th of this.

Analysis The buoyancy force acting on the balloon is

$$V_{balloon} = 4\pi r^3/3 = 4\pi(6\text{ m})^3/3 = 904.8\text{ m}^3$$

$$F_B = \rho_{air} g V_{balloon}$$
$$= (1.16\text{ kg/m}^3)(9.81\text{ m/s}^2)(904.8\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 10{,}296\text{ N}$$



The total mass is

$$m_{He} = \rho_{He} V = \left(\frac{1.16}{7}\text{kg/m}^3\right)(904.8\text{ m}^3) = 149.9\text{ kg}$$

$$m_{total} = m_{He} + m_{people} = 149.9 + 2\times 70 = 289.9\text{ kg}$$

The total weight is

$$W = m_{total} g = (289.9\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 2844\text{ N}$$

Thus the net force acting on the balloon is

$$F_{net} = F_B - W = 10{,}296 - 2844 = 7452\text{ N}$$

Then the acceleration becomes

$$a = \frac{F_{net}}{m_{total}} = \frac{7452\text{ N}}{289.9\text{ kg}}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{25.7\ m/s^2}$$

Discussion This is almost three times the acceleration of gravity – aerodynamic drag on the balloon acts quickly to slow down the acceleration.

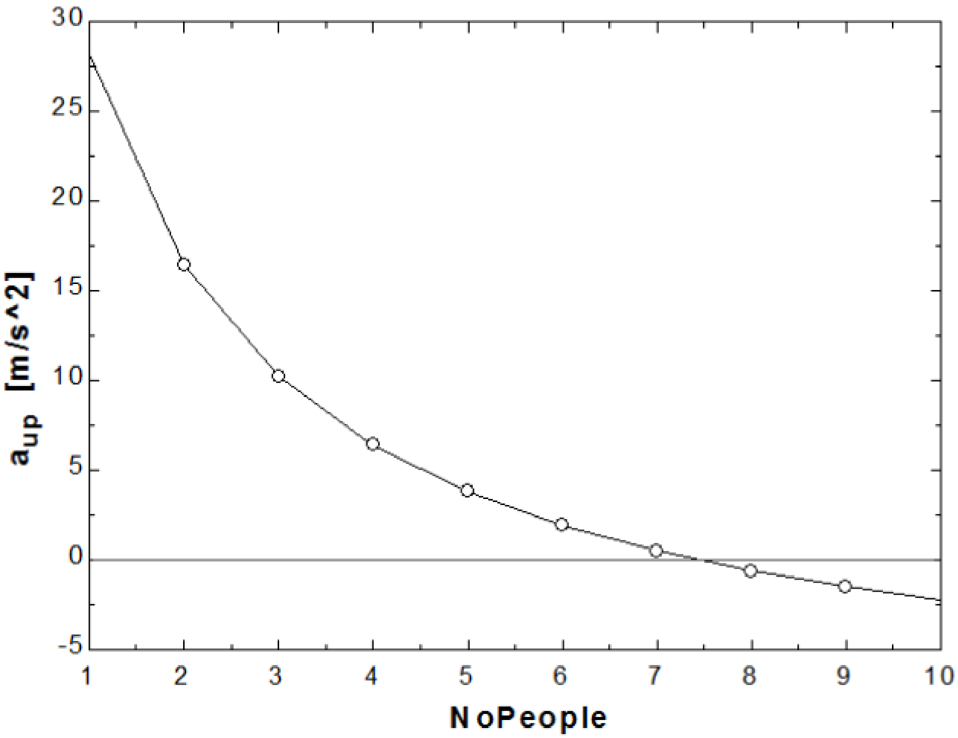
3-140

Solution The previous problem is reconsidered. The effect of the number of people carried in the balloon on acceleration is to be investigated. Acceleration is to be plotted against the number of people, and the results are to be discussed.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
"Given Data:"
rho_air=1.16"[kg/m^3]" "density of air"
g=9.807"[m/s^2]"
d_balloon=10"[m]"
m_1person=70"[kg]"
{NoPeople = 2} "Data suppied in Parametric Table"

"Calculated values:"
rho_He=rho_air/7"[kg/m^3]" "density of helium"
r_balloon=d_balloon/2"[m]"
V_balloon=4*pi*r_balloon^3/3"[m^3]"
m_people=NoPeople*m_1person"[kg]"
m_He=rho_He*V_balloon"[kg]"
m_total=m_He+m_people"[kg]"
"The total weight of balloon and people is:"
W_total=m_total*g"[N]"
"The buoyancy force acting on the balloon, F_b, is equal to the weight of the air displaced by the balloon."
F_b=rho_air*V_balloon*g"[N]"
"From the free body diagram of the balloon, the balancing vertical forces must equal the product of the total mass and the vertical acceleration:"
F_b- W_total=m_total*a_up
```

A_{up} [m/s^2]	No. People
28.19	1
16.46	2
10.26	3
6.434	4
3.831	5
1.947	6
0.5204	7
-0.5973	8
-1.497	9
-2.236	10



Discussion As expected, the more people, the slower the acceleration. In fact, if more than 7 people are on board, the balloon does not rise at all.

3-141

Solution A balloon is filled with helium gas. The maximum amount of load the balloon can carry is to be determined.

Assumptions The weight of the cage and the ropes of the balloon is negligible.

Properties The density of air is given to be ρ = 1.16 kg/m^3. The density of helium gas is 1/7th of this.

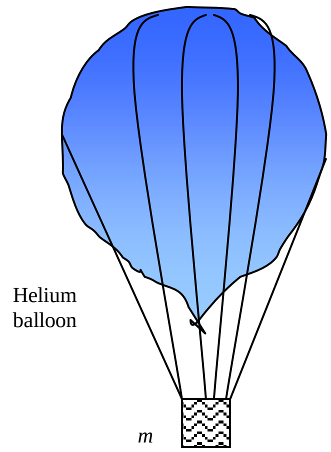
Analysis In the limiting case, the net force acting on the balloon will be zero. That is, the buoyancy force and the weight will balance each other:

$$W = mg = F_B$$

$$m_{total} = \frac{F_B}{g} = \frac{5958.4 \text{ N}}{9.81 \text{ m/s}^2} = 607.4 \text{ kg}$$

Thus,

$$m_{\text{people}} = m_{\text{total}} - m_{\text{He}} = 607.4 - 86.8 = 520.6 \text{ kg} \cong \mathbf{521\ kg}$$



Discussion When the net weight of the balloon and its cargo exceeds the weight of the air it displaces, the balloon/cargo is no longer "lighter than air", and therefore cannot rise.

3-142E

Solution The pressure in a steam boiler is given in kgf/cm^2. It is to be expressed in psi, kPa, atm, and bars.

Analysis We note that 1 atm = 1.03323 kgf/cm^2, 1 atm = 14.696 psi, 1 atm = 101.325 kPa, and 1 atm = 1.01325 bar (inner cover page of text). Then the desired conversions become:

In atm: $$P = (90 \text{ kgf/cm}^2)\left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2}\right) = \mathbf{87.1 atm}$$

In psi: $$P = (90 \text{ kgf/cm}^2)\left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2}\right)\left(\frac{14.696 \text{ psi}}{1 \text{ atm}}\right) = \mathbf{1280\ psi}$$

In kPa: $$P = (90 \text{ kgf/cm}^2)\left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2}\right)\left(\frac{101.325 \text{ kPa}}{1 \text{ atm}}\right) = \mathbf{8826\ kPa}$$

In bars: $$P = (90 \text{ kgf/cm}^2)\left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2}\right)\left(\frac{1.01325 \text{ bar}}{1 \text{ atm}}\right) = \mathbf{88.3 bar}$$

Discussion Note that the units atm, kgf/cm^2, and bar are almost identical to each other. The final results are given to three or four significant digits, but conversion ratios are typically precise to at least five significant digits.

3-143

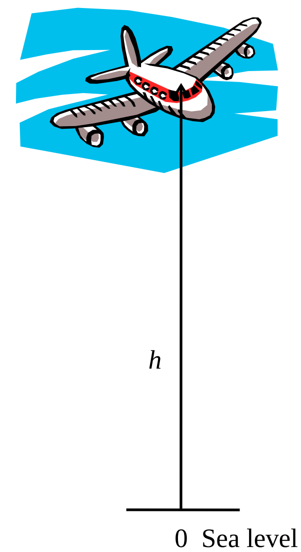
Solution A barometer is used to measure the altitude of a plane relative to the ground. The barometric readings at the ground and in the plane are given. The altitude of the plane is to be determined.

Assumptions The variation of air density with altitude is negligible.

Properties The densities of air and mercury are given to be ρ_{air} = 1.20 kg/m^3 and $\rho_{mercury}$ = 13,600 kg/m^3.

Analysis Atmospheric pressures at the location of the plane and the ground level are

$$\begin{aligned} P_{\text{plane}} &= (\rho g h)_{\text{plane}} \\ &= (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.420\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= 56.03\ \text{kPa} \end{aligned}$$



$$\begin{aligned} P_{\text{ground}} &= (\rho g h)_{\text{ground}} \\ &= (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.760\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= 101.40\ \text{kPa} \end{aligned}$$

Taking an air column between the airplane and the ground and writing a force balance per unit base area, we obtain

$$W_{\text{air}} / A = P_{\text{ground}} - P_{\text{plane}}$$

$$(\rho g h)_{\text{air}} = P_{\text{ground}} - P_{\text{plane}}$$

$$(1.20\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(h)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) = (101.40 - 56.03)\ \text{kPa}$$

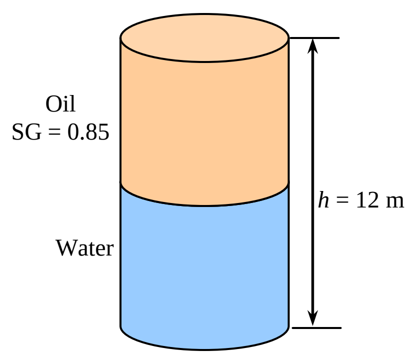
It yields h = **3853 m,** which is also the altitude of the airplane.

Discussion Obviously, a mercury barometer is not practical on an airplane – an electronic barometer is used instead.

3-144

Solution A 12-m high cylindrical container is filled with equal volumes of water and oil. The pressure difference between the top and the bottom of the container is to be determined.

Properties The density of water is given to be ρ = 1000 kg/m^3. The specific gravity of oil is given to be 0.85.



Analysis The density of the oil is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{H_2O} = (0.85)(1000\ \text{kg/m}^3) = 850\ \text{kg/m}^3$$

The pressure difference between the top and the bottom of the cylinder is the sum of the pressure differences across the two fluids,

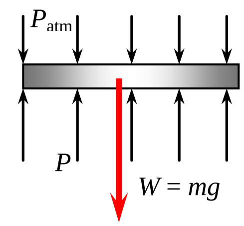
$$\begin{aligned} \Delta P_{total} &= \Delta P_{oil} + \Delta P_{water} = (\rho g h)_{oil} + (\rho g h)_{water} \\ &= \left[(850\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(6\ \text{m}) + (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(6\ \text{m})\right]\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= \mathbf{109\ kPa} \end{aligned}$$

Discussion The pressure at the interface must be the same in the oil and the water. Therefore, we can use the rules for hydrostatics across the two fluids, since they are at rest and there are no appreciable surface tension effects.

3-145

Solution The pressure of a gas contained in a vertical piston-cylinder device is measured to be 500 kPa. The mass of the piston is to be determined.

Assumptions There is no friction between the piston and the cylinder.



Analysis Drawing the free body diagram of the piston and balancing the vertical forces yield

$$W = PA - P_{atm}A$$

$$mg = (P - P_{atm})A$$

$$(m)(9.81 \text{ m/s}^2) = (500 - 100 \text{ kPa})(30 \times 10^{-4} \text{ m}^2)\left(\frac{1000 \text{ kg/m}\cdot\text{s}^2}{1 \text{ kPa}}\right)$$

Solution of the above equation yields m = **122 kg.**

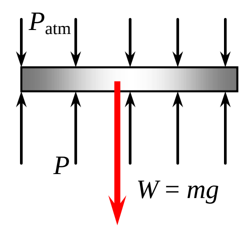
Discussion The gas cannot distinguish between pressure due to the piston weight and atmospheric pressure – both "feel" like a higher pressure acting on the top of the gas in the cylinder.

3-146

Solution The gage pressure in a pressure cooker is maintained constant at 120 kPa by a petcock. The mass of the petcock is to be determined.

Assumptions There is no blockage of the pressure release valve.

Analysis Atmospheric pressure is acting on all surfaces of the petcock, which balances itself out. Therefore, it can be disregarded in calculations if we use the gage pressure as the cooker pressure. A force balance on the petcock ($\Sigma F_y = 0$) yields



$$W = P_{gage}A$$

$$m = \frac{P_{gage}A}{g} = \frac{(120 \text{ kPa})(3\times 10^{-6} \text{ m}^2)}{9.81 \text{ m/s}^2}\left(\frac{1000 \text{ kg}\cdot\text{m/s}^2}{1 \text{ kPa}}\right)$$

$$= \mathbf{0.0367 kg = 36.7 g}$$

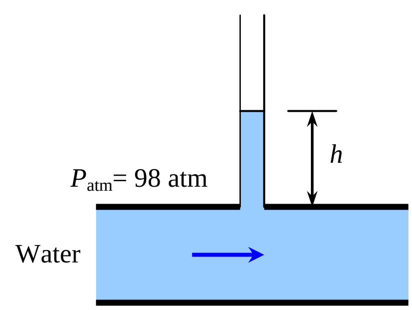
Discussion The higher pressure causes water in the cooker to boil at a higher temperature.

3-147

Solution A glass tube open to the atmosphere is attached to a water pipe, and the pressure at the bottom of the tube is measured. It is to be determined how high the water will rise in the tube.

Properties The density of water is given to be ρ = 1000 kg/m^3.

Analysis The pressure at the bottom of the tube can be expressed as



$$P = P_{atm} + (\rho g h)_{tube}$$

Solving for h,

$$h = \frac{P - P_{atm}}{\rho g}$$

$$= \frac{(115 - 98) \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ N}}\right)\left(\frac{1000 \text{ N/m}^2}{1 \text{ kPa}}\right)$$

$$= \mathbf{1.73 m}$$

Discussion Even though the water is flowing, the water in the tube itself is at rest. If the pressure at the tube bottom had been given in terms of gage pressure, we would not have had to take into account the atmospheric pressure term.

3-148

Solution The average atmospheric pressure is given as $P_{\text{atm}} = 101.325(1 - 0.02256\, z)^{5.256}$ where z is the altitude in km. The atmospheric pressures at various locations are to be determined.

Analysis Atmospheric pressure at various locations is obtained by substituting the altitude z values in km into the relation $P_{atm} = 101.325(1 - 0.02256 z)^{5.256}$. The results are tabulated below.

Atlanta:	(z = 0.306 km): P_{atm} = 101.325(1 - 0.02256×0.306)$^{5.256}$ = **97.7 kPa**
Denver:	(z = 1.610 km): P_{atm} = 101.325(1 - 0.02256×1.610)$^{5.256}$ = **83.4 kPa**
M. City:	(z = 2.309 km): P_{atm} = 101.325(1 - 0.02256×2.309)$^{5.256}$ = **76.5 kPa**
Mt. Ev.:	(z = 8.848 km): P_{atm} = 101.325(1 - 0.02256×8.848)$^{5.256}$ = **31.4 kPa**

Discussion It may be surprising, but the atmospheric pressure on Mt. Everest is less than 1/3 that at sea level!

3-149

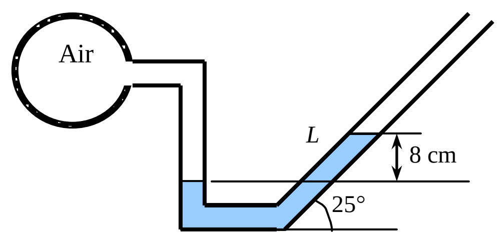
Solution The air pressure in a duct is measured by an inclined manometer. For a given vertical level difference, the gage pressure in the duct and the length of the differential fluid column are to be determined.

Assumptions The manometer fluid is an incompressible substance.

Properties The density of the liquid is given to be ρ = 0.81 kg/L = 810 kg/m^3.

Analysis The gage pressure in the duct is determined from

$$P_{\text{gage}} = P_{\text{abs}} - P_{\text{atm}} = \rho g h$$
$$= (810\,\text{kg/m}^3)(9.81\ \text{m/s}^2)(0.08\,\text{m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ Pa}}{1\text{N/m}^2}\right)$$
$$= \mathbf{636\ \ Pa}$$



The length of the differential fluid column is

$$L = \frac{h}{\sin\theta} = \frac{8\,\text{cm}}{\sin 25°} = \mathbf{18.9\,cm}$$

Discussion Note that the length of the differential fluid column is extended considerably by inclining the manometer arm for better readability (and therefore higher *precision*).

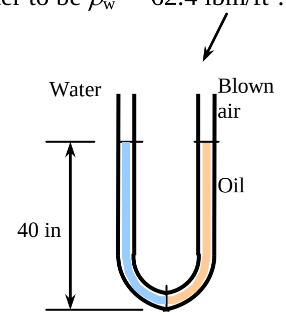
3-150E

Solution Equal volumes of water and oil are poured into a U-tube from different arms, and the oil side is pressurized until the contact surface of the two fluids moves to the bottom and the liquid levels in both arms become the same. The excess pressure applied on the oil side is to be determined.

Assumptions **1** Both water and oil are incompressible substances. **2** Oil does not mix with water. **3** The cross-sectional area of the U-tube is constant.

Properties The density of oil is given to be ρ_{oil} = 49.3 lbm/ft^3. We take the density of water to be ρ_w = 62.4 lbm/ft^3.

Analysis Noting that the pressure of both the water and the oil is the same at the contact surface, the pressure at this surface can be expressed as



$$P_{\text{contact}} = P_{\text{blow}} + \rho_{\text{a}} g h_{\text{a}} = P_{\text{atm}} + \rho_{\text{w}} g h_{\text{w}}$$

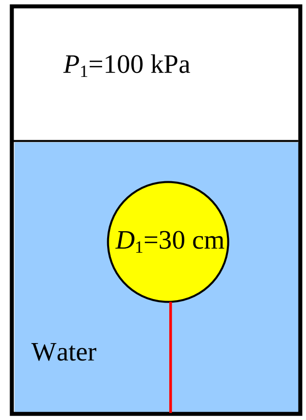
Noting that $h_a = h_w$ and rearranging,

$$\begin{aligned} P_{\text{gage,blow}} &= P_{\text{blow}} - P_{\text{atm}} = (\rho_w - \rho_{oil})gh \\ &= (62.4 - 49.3\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(40/12\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right) \\ &= \mathbf{0.303\ psi} \end{aligned}$$

Discussion When the person stops blowing, the oil rises and some water flows into the right arm. It can be shown that when the curvature effects of the tube are disregarded, the differential height of water is 23.7 in to balance 30-in of oil.

3-151

Solution An elastic air balloon submerged in water is attached to the base of the tank. The change in the tension force of the cable is to be determined when the tank pressure is increased and the balloon diameter is decreased in accordance with the relation $P = CD^{-2}$.



Assumptions **1** Atmospheric pressure acts on all surfaces, and thus it can be ignored in calculations for convenience. **2** Water is an incompressible fluid. **3** The weight of the balloon and the air in it is negligible.

Properties We take the density of water to be 1000 kg/m^3.

Analysis The tension force on the cable holding the balloon is determined from a force balance on the balloon to be

$$F_{cable} = F_B - W_{balloon} \cong F_B$$

The buoyancy force acting on the balloon initially is

$$F_{B,1} = \rho_w g V_{balloon,1} = \rho_w g \frac{\pi D_1^3}{6} = (1000\ \text{kg/m}^3)(9.81\,\text{m/s}^2)\frac{\pi(0.30\,\text{m})^3}{6}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = 138.7\,\text{N}$$

The variation of pressure with diameter is given as $P = CD^{-2}$, which is equivalent to $D = \sqrt{C/P}$. Then the final diameter of the ball becomes

$$\frac{D_2}{D_1} = \frac{\sqrt{C/P_2}}{\sqrt{C/P_1}} = \sqrt{\frac{P_1}{P_2}} \quad \rightarrow \quad D_2 = D_1\sqrt{\frac{P_1}{P_2}} = (0.30\,\text{m})\sqrt{\frac{0.1\,\text{MPa}}{1.6\,\text{MPa}}} = 0.075\,\text{m}$$

The buoyancy force acting on the balloon in this case is

$$F_{B,2} = \rho_w g V_{balloon,2} = \rho_w g \frac{\pi D_2^3}{6} = (1000\ \text{kg/m}^3)(9.81\,\text{m/s}^2)\frac{\pi(0.075\,\text{m})^3}{6}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = 2.2\,\text{N}$$

Then the percent change in the cable for becomes

$$Change\% = \frac{F_{cable,1} - F_{cable,2}}{F_{cable,1}} * 100 = \frac{F_{B,1} - F_{B,2}}{F_{B,1}} * 100 = \frac{138.7 - 2.2}{138.7} * 100 = \mathbf{98.4\%}.$$

Therefore, increasing the tank pressure in this case results in 98.4% reduction in cable tension.

Discussion We can obtain a relation for the change in cable tension as follows:

$$Change\% = \frac{F_{B,1} - F_{B,2}}{F_{B,1}} * 100 = \frac{\rho_w g V_{\text{balloon},1} - \rho_w g V_{\text{balloon},2}}{\rho_w g V_{\text{balloon},1}} * 100$$

$$= 100\left(1 - \frac{V_{\text{balloon},2}}{V_{\text{balloon},1}}\right) = 100\left(1 - \frac{D_2^3}{D_1^3}\right) = 100\left(1 - \left(\frac{P_1}{P_2}\right)^{3/2}\right)$$

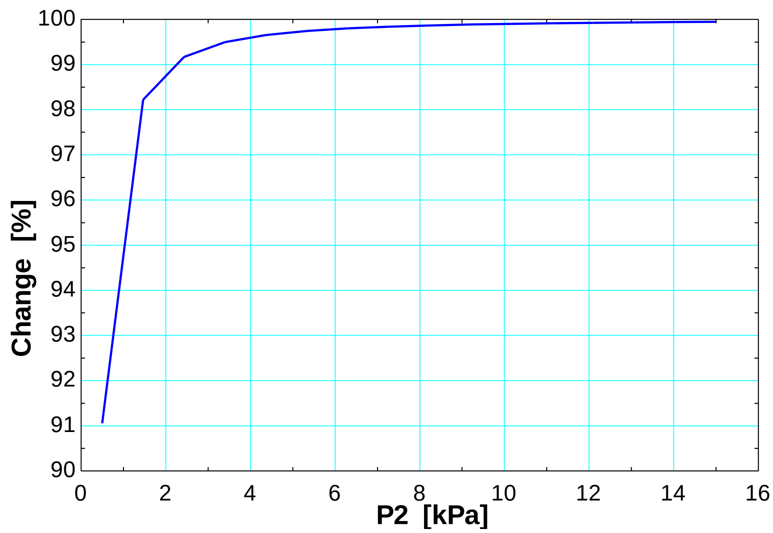
3-152

Solution The previous problem is reconsidered. The effect of the air pressure above the water on the cable force as the pressure varies from 0.1 MPa to 10 MPa is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
P1=0.1 "MPa"
Change=100*(1-(P1/P2)^1.5)
```

Tank pressure P_2, MPa	%Change in cable tension
0.5	91.06
1.467	98.22
2.433	99.17
3.4	99.5
4.367	99.65
5.333	99.74
6.3	99.8
7.267	99.84
8.233	99.87
9.2	99.89
10.17	99.9
11.13	99.91
12.1	99.92
13.07	99.93
14.03	99.94
15	99.95



Discussion The change in cable tension is at first very rapid, but levels off as the balloon shrinks to nearly zero diameter at high pressure.

3-153

Solution A gasoline line is connected to a pressure gage through a double-U manometer. For a given reading of the pressure gage, the gage pressure of the gasoline line is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible.

Properties The specific gravities of oil, mercury, and gasoline are given to be 0.79, 13.6, and 0.70, respectively. We take the density of water to be ρ_w = 1000 kg/m^3.

Analysis Starting with the pressure indicated by the pressure gage and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the gasoline pipe, and setting the result equal to P_{gasoline} gives

$$P_{\text{gage}} - \rho_{\text{w}} g h_w + \rho_{\text{oil}} g h_{\text{oil}} - \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{gasoline}} g h_{\text{gasoline}} = P_{\text{gasoline}}$$

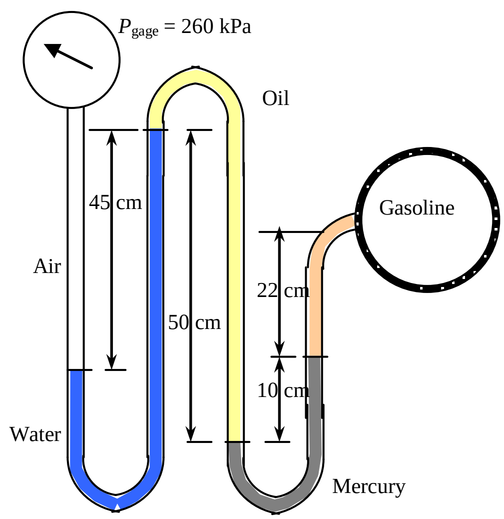
Rearranging,

$$P_{\text{gasoline}} = P_{\text{gage}} - \rho_{\text{w}} g(h_w - SG_{\text{oil}} h_{\text{oil}} + SG_{\text{Hg}} h_{\text{Hg}} + SG_{\text{gasoline}} h_{\text{gasoline}})$$

Substituting,

$$\begin{aligned} P_{\text{gasoline}} &= 260\text{ kPa} - (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)[(0.45\text{ m}) - (0.79(0.5\text{ m}) + 13.6(0.1\text{m}) + 0.70(0.22\text{ m})] \\ &\quad \times \left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) \\ &= \mathbf{245\,kPa} \end{aligned}$$

Therefore, the pressure in the gasoline pipe is 15 kPa lower than the pressure reading of the pressure gage.



Discussion Note that sometimes the use of specific gravity offers great convenience in the solution of problems that involve several fluids.

3-154

Solution A gasoline line is connected to a pressure gage through a double-U manometer. For a given reading of the pressure gage, the gage pressure of the gasoline line is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible.

Properties The specific gravities of oil, mercury, and gasoline are given to be 0.79, 13.6, and 0.70, respectively. We take the density of water to be ρ_w = 1000 kg/m^3.

Analysis Starting with the pressure indicated by the pressure gage and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the gasoline pipe, and setting the result equal to P_{gasoline} gives

$$P_{gage} - \rho_{\text{w}} gh_w + \rho_{\text{alcohol}} gh_{\text{alcohol}} - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{gasoline}} gh_{\text{gasoline}} = P_{\text{gasoline}}$$

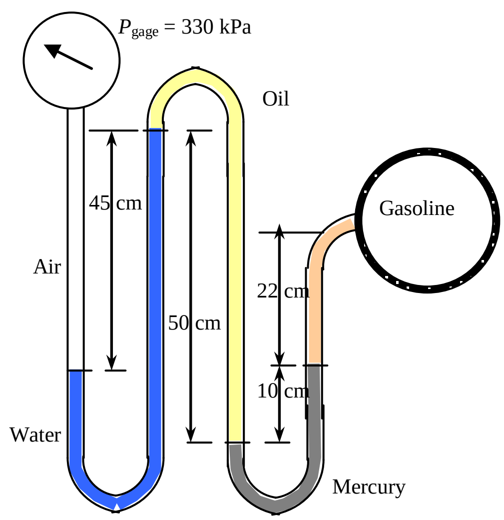
Rearranging,

$$P_{\text{gasoline}} = P_{\text{gage}} - \rho_{\text{w}} g(h_w - SG_{\text{alcohol}} h_{\text{s,alcohol}} + SG_{\text{Hg}} h_{\text{Hg}} + SG_{\text{gasoline}} h_{\text{s,gasoline}})$$

Substituting,

$$P_{\text{gasoline}} = 330\text{ kPa} - (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)[(0.45\text{ m}) - 0.79(0.5\text{ m}) + 13.6(0.1\text{ m}) + 0.70(0.22\text{ m})]$$
$$\times \left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)$$
$$= \mathbf{315\ kPa}$$

Therefore, the pressure in the gasoline pipe is 15 kPa lower than the pressure reading of the pressure gage.



Discussion Note that sometimes the use of specific gravity offers great convenience in the solution of problems that involve several fluids.

3-155E

Solution A water pipe is connected to a double-U manometer whose free arm is open to the atmosphere. The absolute pressure at the center of the pipe is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The solubility of the liquids in each other is negligible.

Properties The specific gravities of mercury and oil are given to be 13.6 and 0.80, respectively. We take the density of water to be ρ_w = 62.4 lbm/ft^3.

Analysis Starting with the pressure at the center of the water pipe, and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} gives

$$P_{\text{water pipe}} - \rho_{\text{water}} gh_{\text{water}} + \rho_{\text{alcohol}} gh_{\text{alcohol}} - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{oil}} gh_{\text{oil}} = P_{atm}$$

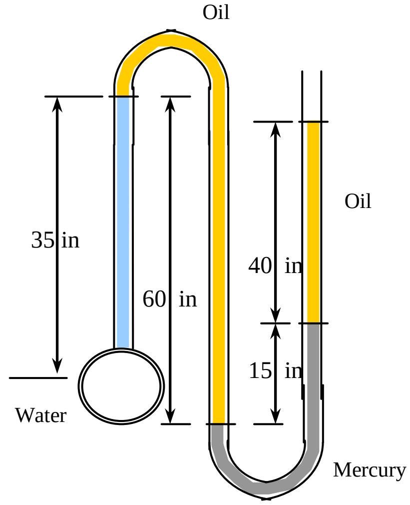
Solving for $P_{\text{water pipe,}}$

$$P_{\text{water pipe}} = P_{atm} + \rho_{\text{water}} g(h_{\text{water}} - SG_{\text{oil}} h_{\text{alcohol}} + SG_{\text{Hg}} h_{\text{Hg}} + SG_{\text{oil}} h_{\text{oil}})$$

Substituting,

$$\begin{aligned} P_{\text{water pipe}} &= 14.2\text{psia} + (62.4\text{lbm/ft}^3)(32.2\text{ ft/s}^2)[(35/12\text{ ft}) - 0.80(60/12\text{ ft}) + 13.6(15/12\text{ ft}) \\ &\quad + 0.8(40/12\text{ ft})] \times \left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\text{ ft}^2}{144\text{ in}^2}\right) \\ &= \mathbf{22.3\ psia} \end{aligned}$$

Therefore, the absolute pressure in the water pipe is 22.3 psia.



Discussion Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

3-156

Solution The pressure of water flowing through a pipe is measured by an arrangement that involves both a pressure gage and a manometer. For the values given, the pressure in the pipe is to be determined.

Assumptions **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible.

Properties The specific gravity of gage fluid is given to be 2.4. We take the standard density of water to be ρ_w = 1000 kg/m^3.

Analysis Starting with the pressure indicated by the pressure gage and moving along the tube by adding (as we go down) or subtracting (as we go up) the ρgh terms until we reach the water pipe, and setting the result equal to P_{water} give

$$P_{\text{gage}} + \rho_{\text{w}} g h_{\text{w1}} - \rho_{\text{gage}} g h_{\text{gage}} - \rho_{\text{w}} g h_{\text{w2}} = P_{\text{water}}$$

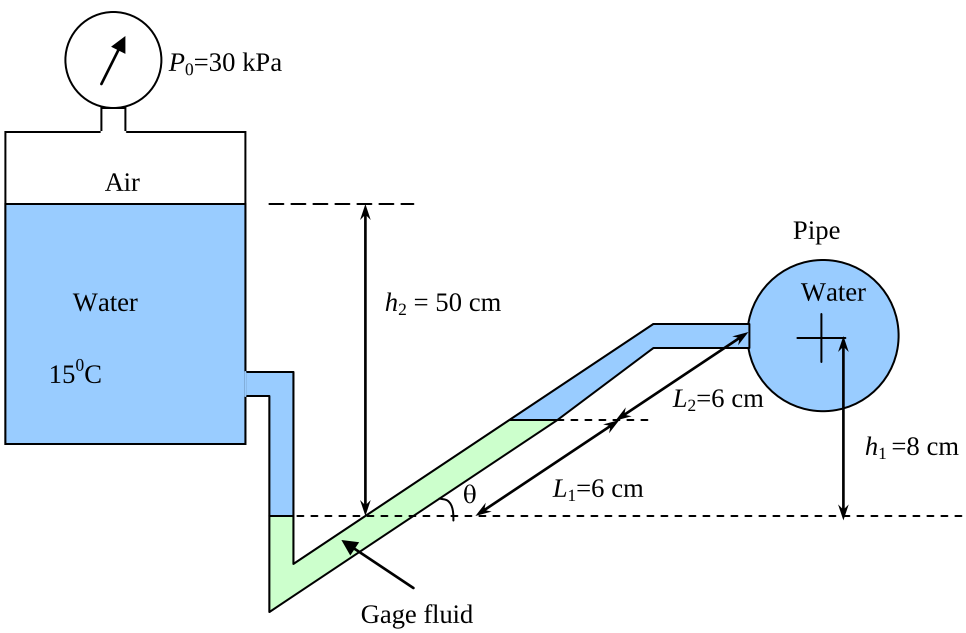
Rearranging,

$$P_{\text{water}} = P_{\text{gage}} + \rho_{\text{w}} g\left(h_{\text{w1}} - \text{SG}_{\text{gage}} h_{\text{gage}} - h_{\text{w2}}\right) = P_{\text{gage}} + \rho_{\text{w}} g\left(h_2 - \text{SG}_{\text{gage}} L_1 \sin\theta - L_2 \sin\theta\right)$$

Noting that $\sin\theta = 8/12 = 0.6667$ and substituting,

$$\begin{aligned} P_{\text{water}} &= 30 \text{ kPa} + (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)[(0.50 \text{ m}) - 2.4(0.06 \text{ m})0.6667 - (0.06 \text{ m})0.6667] \\ &\times \left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2}\right) \\ &= \mathbf{33.6 \text{ kPa}} \end{aligned}$$

Therefore, the pressure in the gasoline pipe is 3.6 kPa over the reading of the pressure gage.



Discussion Note that even without a manometer, the reading of a pressure gage can be in error if it is not placed at the same level as the pipe when the fluid is a liquid.

3-157

Solution A U-tube filled with mercury except the 12-cm high portion at the top. Oil is poured into the left arm, forcing some mercury from the left arm into the right one. The maximum amount of oil that can be added into the left arm is to be determined.

Assumptions **1** Both liquids are incompressible. **2** The U-tube is perfectly vertical.

Properties The specific gravities are given to be 2.72 for oil and 13.6 for mercury.

Analysis Initially, the mercury levels in both tubes are the same. When oil is poured into the left arm, it will push the mercury in the left down, which will cause the mercury level in the right arm to rise. Noting that the volume of mercury is constant, the decrease in the mercury volume in left column must be equal to the increase in the mercury volume in the right arm. Therefore, if the drop in mercury level in the left arm is x, the rise in the mercury level in the right arm h corresponding to a drop of x in the left arm is

$$V_{\text{left}} = V_{\text{right}} \quad \rightarrow \quad \pi(2d)^2 x = \pi d^2 h \quad \rightarrow \quad h = 4x$$

The pressures at points A and B are equal $P_A = P_B$ and thus

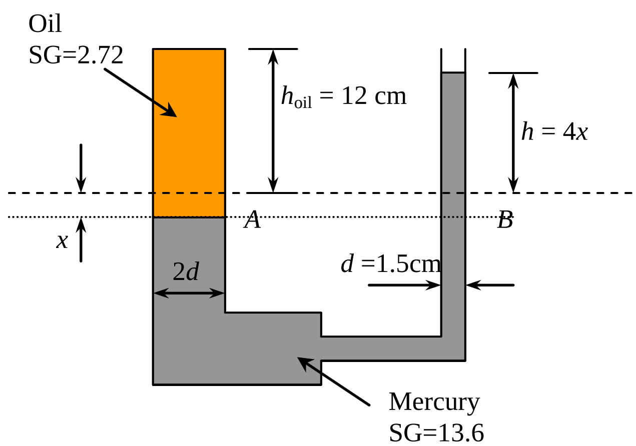
$$P_{atm} + \rho_{\text{oil}} g(h_{oil} + x) = P_{atm} + \rho_{Hg} g h_{Hg} \quad \rightarrow \quad \text{SG}_{\text{oil}}\rho_w g\left(h_{\text{oil}} + x\right) = \text{SG}_{\text{Hg}}\rho_w g\left(5x\right)$$

Solving for x and substituting,

$$x = \frac{SG_{\text{oil}} h_{oil}}{5SG_{\text{Hg}} - SG_{\text{oil}}} = \frac{2.72(12\text{ cm})}{5 \times 13.6 - 2.72} = 0.5\text{ cm}$$

Therefore, the maximum amount of oil that can be added into the left arm is

$$V_{oil} = \pi(2d/2)^2(h_{oil} + x) = \pi(1.5\text{ cm})^2(12\text{ cm} + 0.5\text{ cm}) = 88.4\text{ cm}^3 = \mathbf{0.0884 L}$$



Discussion Note that the fluid levels in the two arms of a U-tube can be different when two different fluids are involved.

3-158

Solution The temperature of the atmosphere varies with altitude z as $T = T_0 - \beta z$, while the gravitational acceleration varies by $g(z) = g_0 /(1 + z / 6{,}370{,}320)^2$. Relations for the variation of pressure in atmosphere are to be obtained (*a*) by ignoring and (*b*) by considering the variation of g with altitude.

Assumptions The air in the troposphere behaves as an ideal gas.

Analysis (*a*) Pressure change across a differential fluid layer of thickness dz in the vertical z direction is

$$dP = -\rho g dz$$

From the ideal gas relation, the air density can be expressed as $\rho = \dfrac{P}{RT} = \dfrac{P}{R(T_0 - \beta z)}$. Then,

$$dP = -\frac{P}{R(T_0 - \beta z)} g dz$$

Separating variables and integrating from $z = 0$ where $P = P_0$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} \frac{dP}{P} = -\int_0^z \frac{g dz}{R(T_0 - \beta z)}$$

Performing the integrations.

$$\ln \frac{P}{P_0} = \frac{g}{R\beta} \ln \frac{T_0 - \beta z}{T_0}$$

Rearranging, the desired relation for atmospheric pressure for the case of constant g becomes

$$\boxed{P = P_0 \left(1 - \frac{\beta z}{T_0}\right)^{\frac{g}{\beta R}}}$$

(*b*) When the variation of g with altitude is considered, the procedure remains the same but the expressions become more complicated,

$$dP = -\frac{P}{R(T_0 - \beta z)} \frac{g_0}{(1 + z / 6{,}370{,}320)^2} dz$$

Separating variables and integrating from $z = 0$ where $P = P_0$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} \frac{dP}{P} = -\int_0^z \frac{g_0 dz}{R(T_0 - \beta z)(1 + z / 6{,}370{,}320)^2}$$

Performing the integrations,

$$\ln P \Big|_{P_0}^{P} = \frac{g_0}{R\beta} \left| \frac{1}{(1 + kT_0 / \beta)(1 + kz)} - \frac{1}{(1 + kT_0 / \beta)^2} \ln \frac{1 + kz}{T_0 - \beta z} \right|_0^z$$

where R = 287 J/kg·K = 287 m^2/s^2·K is the gas constant of air. After some manipulations, we obtain

$$\boxed{P = P_0 \exp\left[-\frac{g_0}{R(\beta + kT_0)} \left(\frac{1}{1 + 1/kz} + \frac{1}{1 + kT_0 / \beta} \ln \frac{1 + kz}{1 - \beta z / T_0} \right) \right]}$$

where T_0 = 288.15 K, β = 0.0065 K/m, g_0 = 9.807 m/s^2, k = 1/6,370,320 m^{-1}, and z is the elevation in m..

Discussion When performing the integration in part (*b*), the following expression from integral tables is used, together with a transformation of variable $x = T_0 - \beta z$,

$$\int \frac{dx}{x(a + bx)^2} = \frac{1}{a(a + bx)} - \frac{1}{a^2} \ln \frac{a + bx}{x}$$

Also, for z = 11,000 m, for example, the relations in (*a*) and (*b*) give 22.62 and 22.69 kPa, respectively.

3-159

Solution The variation of pressure with density in a thick gas layer is given. A relation is to be obtained for pressure as a function of elevation *z*.

Assumptions The property relation $P = C\rho^n$ is valid over the entire region considered.

Analysis The pressure change across a differential fluid layer of thickness *dz* in the vertical *z* direction is given as,

$$dP = -\rho g dz$$

Also, the relation $P = C\rho^n$ can be expressed as $C = P/\rho^n = P_0/\rho_0^n$, and thus

$$\rho = \rho_0 (P/P_0)^{1/n}$$

Substituting,

$$dP = -g\rho_0 (P/P_0)^{1/n} dz$$

Separating variables and integrating from $z = 0$ where $P = P_0 = C\rho_0^n$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} (P/P_0)^{-1/n} dP = -\rho_0 g \int_0^z dz$$

Performing the integrations.

$$P_0 \frac{(P/P_0)^{-1/n+1}}{-1/n+1}\Bigg|_{P_0}^{P} = -\rho_0 gz \quad \rightarrow \quad \left(\frac{P}{P_0}\right)^{(n-1)/n} - 1 = -\frac{n-1}{n}\frac{\rho_0 gz}{P_0}$$

Solving for *P*,

$$\boxed{P = P_0 \left(1 - \frac{n-1}{n}\frac{\rho_0 gz}{P_0}\right)^{n/(n-1)}}$$

which is the desired relation.

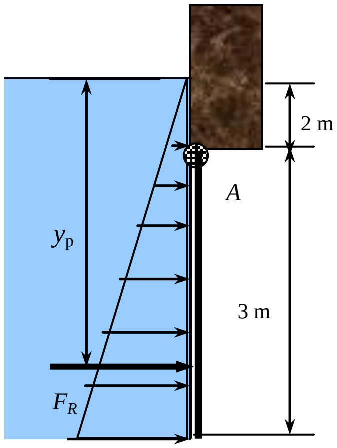
Discussion The final result could be expressed in various forms. The form given is very convenient for calculations as it facilitates unit cancellations and reduces the chance of error.

3-160

Solution A rectangular gate hinged about a horizontal axis along its upper edge is restrained by a fixed ridge at point *B*. The force exerted to the plate by the ridge is to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.



Analysis The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and multiplying it by the plate area gives the resultant hydrostatic force on the gate,

$$F_R = P_{avg}A = \rho g h_C A$$

$$= \left(1000\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)\left(3.5\ \text{m}\right)\left(3\times 6\ \text{m}^2\right)\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= \mathbf{618\ kN}$$

The vertical distance of the pressure center from the free surface of water is

$$y_P = s + \frac{b}{2} + \frac{b^2}{12(s+b/2)} = 2 + \frac{3}{2} + \frac{3^2}{12(2+3/2)} = \mathbf{3.71\ m}$$

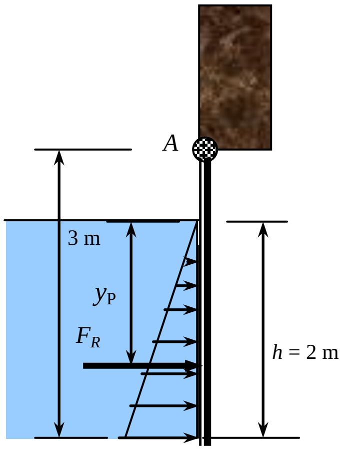
Discussion You can calculate the force at point *B* required to hold back the gate by setting the net moment around hinge point *A* to zero.

3-161

Solution A rectangular gate hinged about a horizontal axis along its upper edge is restrained by a fixed ridge at point *B*. The force exerted to the plate by the ridge is to be determined.

Assumptions Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 1000 kg/m^3 throughout.



Analysis The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and multiplying it by the wetted plate area gives the resultant hydrostatic force on the gate,

$$F_R = P_{ave}A = \rho g h_C A$$

$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(1\ \text{m})[2\times 6\ \text{m}^2]\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= \mathbf{118\ kN}$$

The vertical distance of the pressure center from the free surface of water is

$$y_P = \frac{2h}{3} = \frac{2(2\ \text{m})}{3} = \mathbf{1.33\ m}$$

Discussion Compared to the previous problem (with higher water depth), the force is much smaller, as expected. Also, the center of pressure on the gate is much lower (closer to the ground) for the case with the lower water depth.

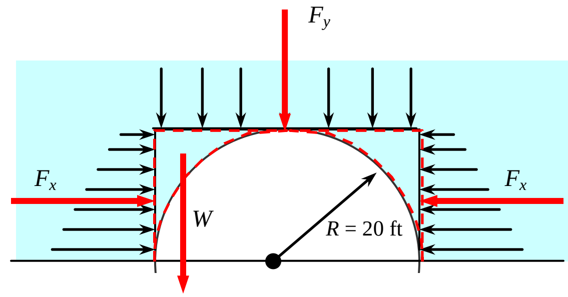
3-162E

Solution A semicircular tunnel is to be built under a lake. The total hydrostatic force acting on the roof of the tunnel is to be determined.

Assumptions Atmospheric pressure acts on both sides of the tunnel, and thus it can be ignored in calculations for convenience.

Properties We take the density of water to be 62.4 lbm/ft^3 throughout.

Analysis We consider the free body diagram of the liquid block enclosed by the circular surface of the tunnel and its vertical (on both sides) and horizontal projections. The hydrostatic forces acting on the vertical and horizontal plane surfaces as well as the weight of the liquid block are determined as follows:



Horizontal force on vertical surface (each side):

$$F_H = F_x = P_{ave}A = \rho g h_C A = \rho g(s + R/2)A$$

$$= (62.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(130 + 20/2\ \text{ft})(20\ \text{ft} \times 800\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)$$

$$= 1.398 \times 10^8\ \text{lbf}\ \ \text{(on each side of the tunnel)}$$

Vertical force on horizontal surface (*downward*):

$$F_y = P_{ave}A = \rho g h_C A = \rho g h_{\text{top}} A$$

$$= (62.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(130\ \text{ft})(40\ \text{ft} \times 800\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)$$

$$= 2.596 \times 10^8\ \text{lbf}$$

Weight of fluid block on each side within the control volume *(downward):*

$$W = mg = \rho g V = \rho g(R^2 - \pi R^2/4)(2000\ \text{ft})$$

$$= (62.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(20\ \text{ft})^2(1 - \pi/4)(800\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)$$

$$= 4.285 \times 10^6\ \text{lbf}\ \ \text{(on each side)}$$

Therefore, the net downward vertical force is

$$F_V = F_y + 2W = 2.596 \times 10^8 + 2 \times 0.04285 \times 10^8 = \mathbf{2.64 \times 10^8\ lbf}$$

This is also the **net force** acting on the tunnel since the horizontal forces acting on the right and left side of the tunnel cancel each other since they are equal and opposite.

Discussion The weight of the two water bocks on the sides represents only about 3.3% of the total vertical force on the tunnel. Therefore, to obtain a reasonable first approximation for deep tunnels, these volumes can be neglected, yielding F_V = 2.596 × 10^8 lbf. A more conservative approximation would be to estimate the force on the *bottom* of the lake if the tunnel were not there. This yields F_V = 2.995 × 10^8 lbf. The actual force is between these two estimates, as expected.

3-163

Solution A hemispherical dome on a level surface filled with water is to be lifted by attaching a long tube to the top and filling it with water. The required height of water in the tube to lift the dome is to be determined.

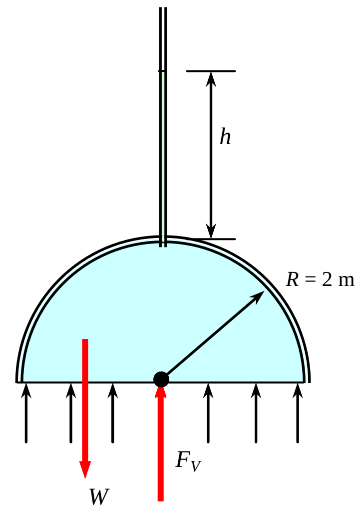
Assumptions **1** Atmospheric pressure acts on both sides of the dome, and thus it can be ignored in calculations for convenience. **2** The weight of the tube and the water in it is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis We take the dome and the water in it as the system. When the dome is about to rise, the reaction force between the dome and the ground becomes zero. Then the free body diagram of this system involves the weights of the dome and the water, balanced by the hydrostatic pressure force from below. Setting these forces equal to each other gives

$$\sum F_y = 0: \quad F_V = W_{dome} + W_{water}$$

$$\rho g(h+R)\pi R^2 = m_{dome} g + m_{water} g$$



Solving for h gives

$$h = \frac{m_{dome} + m_{water}}{\rho \pi R^2} - R = \frac{m_{dome} + \rho[4\pi R^3 / 6]}{\rho \pi R^2} - R$$

Substituting,

$$h = \frac{(30{,}000\text{ kg}) + 4\pi(1000\text{ kg/m}^3)(2\text{ m})^3/6}{(1000\text{ kg/m}^3)\pi(2\text{ m})^2} - (2\text{ m}) = \mathbf{1.72\,m}$$

Therefore, this dome can be lifted by attaching a tube which is about 1.72 m long.

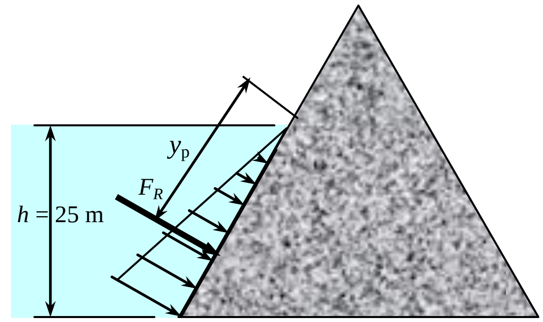
Discussion Note that the water pressure in the dome can be changed greatly by a small amount of water in the vertical tube. Two significant digits in the answer is sufficient for this problem.

3-164

Solution The water in a reservoir is restrained by a triangular wall. The total force (hydrostatic + atmospheric) acting on the inner surface of the wall and the horizontal component of this force are to be determined.

Assumptions **1** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience. **2** Friction at the hinge is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.



Analysis The length of the wall surface underwater is

$$b = \frac{25\text{ m}}{\sin 60°} = 28.87\text{ m}$$

The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and multiplying it by the plate area gives the resultant hydrostatic force on the surface,

$$\begin{aligned} F_R &= P_{\text{avg}}A = \left(P_{\text{atm}} + \rho g h_C\right)A \\ &= \left[100{,}000\text{ N/m}^2 + \left(1000\text{ kg/m}^3\right)\left(9.81\text{ m/s}^2\right)\left(12.5\text{ m}\right)\right]\left(150\times 28.87\text{ m}^2\right)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) \\ &= \mathbf{9.64\times 10^8\ N} \end{aligned}$$

Noting that

$$\frac{P_0}{\rho g \sin 60°} = \frac{100{,}000\text{ N/m}^2}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)\sin 60°}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = 11.77\text{ m}$$

the distance of the pressure center from the free surface of water along the wall surface is

$$y_p = s + \frac{b}{2} + \frac{b^2}{12\left(s + \dfrac{b}{2} + \dfrac{P_0}{\rho g \sin\theta}\right)} = 0 + \frac{28.87\text{ m}}{2} + \frac{(28.87\text{ m})^2}{12\left(0 + \dfrac{28.87\text{ m}}{2} + 11.77\text{ m}\right)} = \mathbf{17.1 m}$$

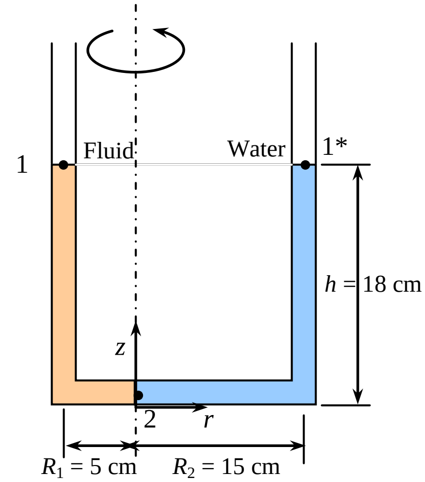
The magnitude of the horizontal component of the hydrostatic force is simply $F_R \sin\theta$,

$$F_H = F_R \sin\theta = (9.64\times 10^8\ \text{N})\sin 60° = \mathbf{8.35\times 10^8\ N}$$

Discussion Atmospheric pressure is usually ignored in the analysis for convenience since it acts on both sides of the walls.

3-165

Solution A U-tube that contains water in its right arm and another liquid in its left arm is rotated about an axis closer to the left arm. For a known rotation rate at which the liquid levels in both arms are the same, the density of the fluid in the left arm is to be determined.



Assumptions **1** Both the fluid and the water are incompressible fluids. **2** The two fluids meet at the axis of rotation, and thus there is only water to the right of the axis of rotation.

Properties We take the density of water to be 1000 kg/m^3.

Analysis The pressure difference between two points 1 and 2 in an incompressible fluid rotating in rigid body motion (the *same* fluid) is given by

$$P_2 - P_1 = \frac{\rho\omega^2}{2}(r_2^2 - r_1^2) - \rho g(z_2 - z_1)$$

where

$$\omega = 2\pi\dot{n} = 2\pi(50 \text{ rev/min})\left(\frac{1 \text{ min}}{60 \text{ s}}\right) = 5.236 \text{ rad/s}$$

(for both arms of the U-tube).

The pressure at point 2 is the same for both fluids, so are the pressures at points 1 and 1* ($P_1 = P_1^* = P_{atm}$). Therefore, $P_2 - P_1$ is the same for both fluids. Noting that $z_2 - z_1 = -h$ for both fluids and expressing $P_2 - P_1$ for each fluid,

$$\textit{Water}: \quad P_2 - P_1^* = \frac{\rho_w\omega^2}{2}(0 - R_2^2) - \rho_w g(-h) = \rho_w(-\omega^2 R_2^2/2 + gh)$$

$$\textit{Fluid}: \quad P_2 - P_1 = \frac{\rho_f\omega^2}{2}(0 - R_1^2) - \rho_f g(-h) = \rho_f(-\omega^2 R_1^2/2 + gh)$$

Setting them equal to each other and solving for ρ_f gives

$$\rho_f = \frac{-\omega^2 R_2^2/2 + gh}{-\omega^2 R_1^2/2 + gh}\rho_w = \frac{-(5.236 \text{ rad/s})^2(0.15 \text{ m})^2 + (9.81 \text{ m/s}^2)(0.18 \text{ m})}{-(5.236 \text{ rad/s})^2(0.05 \text{ m})^2 + (9.81 \text{ m/s}^2)(0.18 \text{ m})}(1000 \text{ kg/m}^3) = \mathbf{677 \text{ kg/m}^3}$$

Discussion Note that this device can be used to determine relative densities, though it wouldn't be very practical.

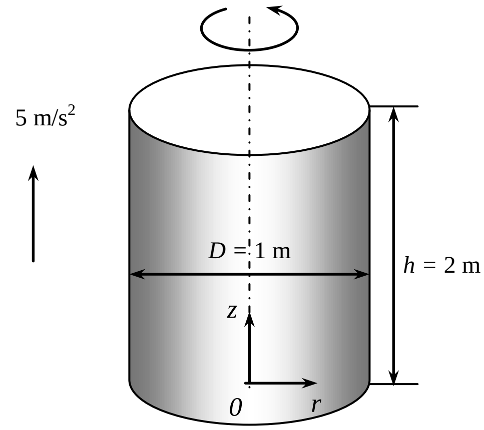
3-166

Solution A vertical cylindrical tank is completely filled with gasoline, and the tank is rotated about its vertical axis at a specified rate while being accelerated upward. The pressures difference between the centers of the bottom and top surfaces, and the pressures difference between the center and the edge of the bottom surface are to be determined.

Assumptions **1** The increase in the rotational speed is very slow so that the liquid in the container always acts as a rigid body. **2** Gasoline is an incompressible substance.

Properties The density of the gasoline is given to be 740 kg/m^3.

Analysis The pressure difference between two points 1 and 2 in an incompressible fluid rotating in rigid body motion is given by $P_2 - P_1 = \frac{\rho\omega^2}{2}(r_2^2 - r_1^2) - \rho g(z_2 - z_1)$. The effect of linear acceleration in the vertical direction is accounted for by replacing g by $g + a_z$. Then,



$$P_2 - P_1 = \frac{\rho\omega^2}{2}(r_2^2 - r_1^2) - \rho(g + a_z)(z_2 - z_1)$$

where R = 0.50 m is the radius, and

$$\omega = 2\pi\dot{n} = 2\pi(130 \text{ rev/min})\left(\frac{1 \text{ min}}{60 \text{ s}}\right) = 13.61 \text{ rad/s}$$

(*a*) Taking points 1 and 2 to be the centers of the bottom and top surfaces, respectively, we have $r_1 = r_2 = 0$ and $z_2 - z_1 = h = 3 \text{ m}$. Then,

$$P_{\text{center, top}} - P_{\text{center, bottom}} = 0 - \rho(g + a_z)(z_2 - z_1) = -\rho(g + a_z)h$$

$$= -(740 \text{ kg/m}^3)(9.81 \text{ m/s}^2 + 5)(2 \text{ m})\left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2}\right) = 21.8 \text{ kN/m}^2 = \mathbf{21.9\ kPa}$$

(*b*) Taking points 1 and 2 to be the center and edge of the bottom surface, respectively, we have $r_1 = 0$, $r_2 = R$, and $z_2 = z_1 = 0$. Then,

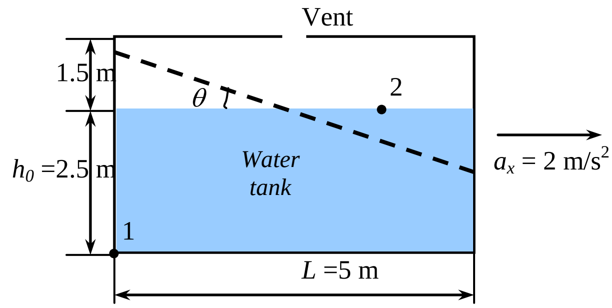
$$P_{\text{edge,bottom}} - P_{\text{center, bottom}} = \frac{\rho\omega^2}{2}(R_2^2 - 0) - 0 = \frac{\rho\omega^2 R^2}{2}$$

$$= \frac{(740 \text{ kg/m}^3)(13.61 \text{ rad/s})^2(0.50 \text{ m})^2}{2}\left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2}\right) = 17.13 \text{ kN/m}^2 = \mathbf{17.1\ kPa}$$

Discussion Note that the rotation of the tank does not affect the pressure difference along the axis of the tank. Likewise, the vertical acceleration does not affect the pressure difference between the edge and the center of the bottom surface (or any other horizontal plane).

3-167

Solution A rectangular water tank open to the atmosphere is accelerated to the right on a level surface at a specified rate. The maximum pressure in the tank above the atmospheric level is to be determined.



Assumptions **1** The road is horizontal during acceleration so that acceleration has no vertical component (a_z = 0). **2** Effects of splashing, breaking and driving over bumps are assumed to be secondary, and are not considered. **3** The vent is never blocked, and thus the minimum pressure is the atmospheric pressure.

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take the *x*-axis to be the direction of motion, the *z*-axis to be the upward vertical direction. The tangent of the angle the free surface makes with the horizontal is

$$\tan\theta = \frac{a_x}{g+a_z} = \frac{2}{9.81+0} = 0.2039 \quad \text{(and thus } \theta = 11.5°\text{)}$$

The maximum vertical rise of the free surface occurs at the back of the tank, and the vertical midsection experiences no rise or drop during acceleration. Then the maximum vertical rise at the back of the tank relative to the neutral midplane is

$$\Delta z_{\max} = (L/2)\tan\theta = [(5\text{ m})/2]\times 0.2039 = 0.510\text{ m}$$

which is less than 1.5 m high air space. Therefore, water never reaches the ceiling, and the maximum water height and the corresponding maximum pressure are

$$h_{\max} = h_0 + \Delta z_{\max} = 2.50 + 0.510 = 3.01\text{ m}$$

$$P_{\max} = P_1 = \rho g h_{\max} = (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(3.01\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 29.5\text{ kN/m}^2 = \mathbf{29.5\ kPa}$$

Discussion It can be shown that the gage pressure at the bottom of the tank varies from 29.5 kPa at the back of the tank to 24.5 kPa at the midsection and 19.5 kPa at the front of the tank.

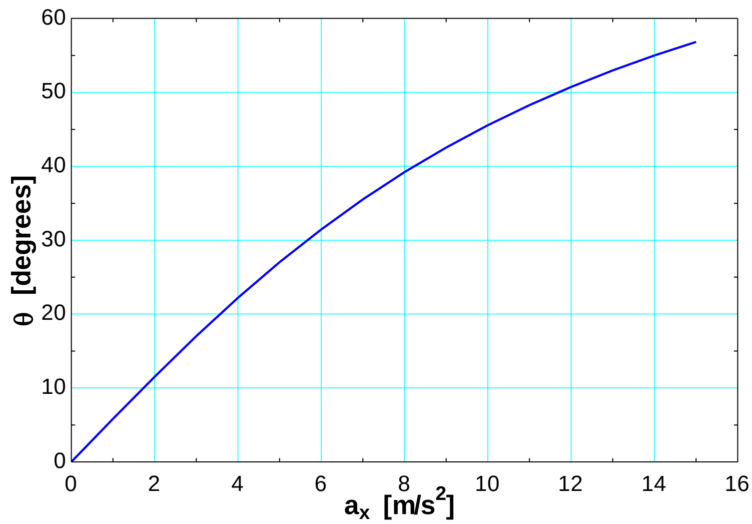
3-168

Solution The previous problem is reconsidered. The effect of acceleration on the slope of the free surface of water in the tank as the acceleration varies from 0 to 5 m/s^2 in increments of 0.5 m/s^2 is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
"a_x=5 [m/s^2]"
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
L=5 [m]
h0=2.5 [m]
a_z=0 [m/s^2]
tan(theta)=a_x/(g+a_z)
h_max=h0+(L/2)*tan(theta)
P_max=rho*g*h_max*Convert(Pa, kPa)
```

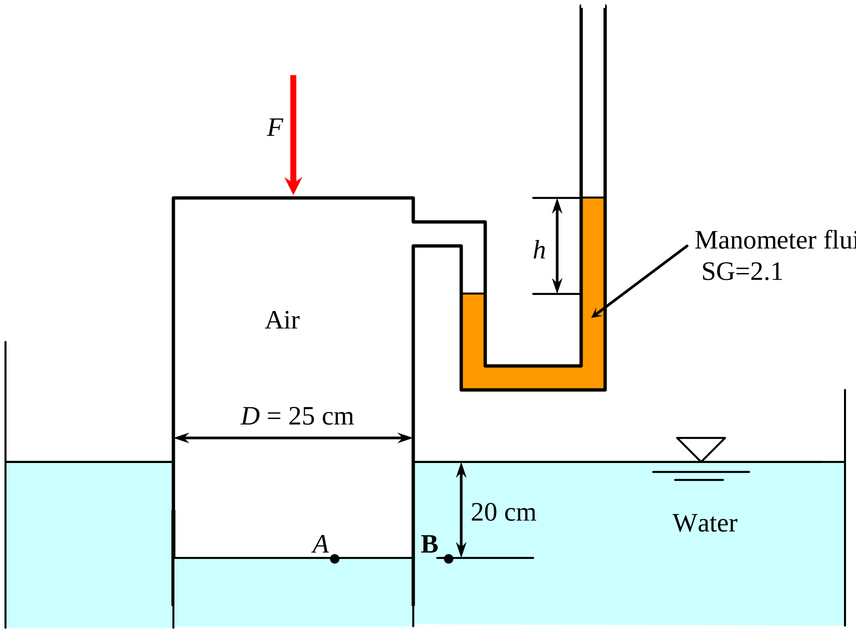
Acceleration a_x, m/s^2	Free surface angle, $\theta°$	Maximum height h_{max}, m	Maximum pressure, P_{max}, kPa
0	0	2.5	24.53
1	5.82	2.755	27.02
2	11.52	3.01	29.52
3	17	3.265	32.02
4	22.18	3.519	34.52
5	27.01	3.774	37.02
6	31.45	4.029	39.52
7	35.51	4.284	42.02
8	39.2	4.539	44.52
9	42.53	4.794	47.02
10	45.55	5.048	49.52
11	48.27	5.303	52.02
12	50.73	5.558	54.52
13	52.96	5.813	57.02
14	54.98	6.068	59.52
15	56.82	6.323	62.02



Discussion Note that water never reaches the ceiling, and a full free surface is formed in the tank.

3-169

Solution A cylindrical container equipped with a manometer is inverted and pressed into water. The differential height of the manometer and the force needed to hold the container in place are to be determined.



Assumptions **1** Atmospheric pressure acts on all surfaces, and thus it can be ignored in calculations for convenience. **2** The variation of air pressure inside cylinder is negligible.

Properties We take the density of water to be 1000 kg/m^3. The density of the manometer fluid is

$$\rho_{\text{mano}} = \text{SG} \times \rho_w = 2.1\left(1000 \text{ kg/m}^3\right) = 2100 \text{ kg/m}^3$$

Analysis The pressures at point *A* and *B* must be the same since they are on the same horizontal line in the same fluid. Then the gage pressure in the cylinder becomes

$$P_{\text{air, gage}} = \rho_{\text{w}} g h_{\text{w}} \quad = (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.20 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = 1962 \text{ N/m}^2 = 1962 \text{ Pa}$$

The manometer also indicates the gage pressure in the cylinder. Therefore,

$$P_{\text{air, gage}} = (\rho g h)_{\text{mano}} \quad \rightarrow \quad h = \frac{P_{\text{air, gage}}}{\rho_{\text{mano}} g} = \frac{1962 \text{ N/m}^2}{(2100 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ kN/m}^2}\right) = 0.0950 \text{ m} = \mathbf{9.50 \text{ cm}}$$

A force balance on the cylinder in the vertical direction yields

$$F + W = P_{\text{air, gage}} A_c$$

Solving for *F* and substituting,

$$F = P_{aie,gage}\frac{\pi D^2}{4} - W = (1962 \text{ N/m}^2)\frac{\pi(0.25 \text{ m})^2}{4} - 65 \text{ N} = \mathbf{31.3N}$$

Discussion We could also solve this problem by considering the atmospheric pressure, but we would obtain the same result since atmospheric pressure would cancel out.

3-170

Solution An iceberg floating in seawater is considered. The volume fraction of the iceberg submerged in seawater is to be determined, and the reason for their turnover is to be explained.

Assumptions **1** The buoyancy force in air is negligible. **2** The density of iceberg and seawater are uniform.

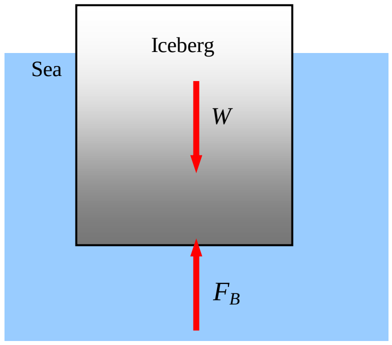
Properties The densities of iceberg and seawater are given to be 917 kg/m^3 and 1042 kg/m^3, respectively.

Analysis (*a*) The weight of a body floating in a fluid is equal to the buoyant force acting on it (a consequence of vertical force balance from static equilibrium). Therefore,

$$W = F_B$$

$$\rho_{\text{body}} g V_{\text{total}} = \rho_{\text{fluid}} g V_{\text{submerged}}$$

$$\frac{V_{\text{submerged}}}{V_{\text{total}}} = \frac{\rho_{\text{body}}}{\rho_{\text{fluid}}} = \frac{\rho_{\text{iceberg}}}{\rho_{\text{seawater}}} = \frac{917}{1042} = 0.880 \quad \text{or} \quad \mathbf{88\%}$$



Therefore, 88% of the volume of the iceberg is submerged in this case.

(*b*) Heat transfer to the iceberg due to the temperature difference between the seawater and an iceberg causes uneven melting of the irregularly shaped iceberg. The resulting **shift in the center of mass causes the iceberg to turn over**.

Discussion The submerged fraction depends on the density of seawater, and this fraction can differ in different seas.

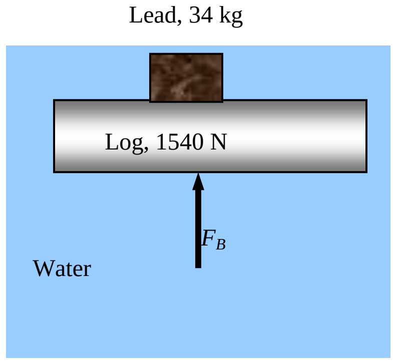
3-171

Solution The density of a wood log is to be measured by tying lead weights to it until both the log and the weights are completely submerged, and then weighing them separately in air. The average density of a given log is to be determined by this approach.

Properties The density of lead weights is given to be 11,300 kg/m^3. We take the density of water to be 1000 kg/m^3.

Analysis The weight of a body is equal to the buoyant force when the body is floating in a fluid while being completely submerged in it (a consequence of vertical force balance from static equilibrium). In this case the average density of the body must be equal to the density of the fluid since

$$W = F_B \quad \rightarrow \quad \rho_{\text{body}} g V = \rho_{\text{fluid}} g V \quad \rightarrow \quad \rho_{\text{body}} = \rho_{\text{fluid}}$$



Therefore,

$$\rho_{ave} = \frac{m_{\text{total}}}{V_{\text{total}}} = \frac{m_{\text{lead}} + m_{\text{log}}}{V_{\text{lead}} + V_{\text{log}}} = \rho_{\text{water}} \quad \rightarrow \quad V_{\text{log}} = \frac{m_{\text{lead}} + m_{\text{log}}}{\rho_{\text{water}}} - V_{\text{lead}}$$

where

$$V_{\text{lead}} = \frac{m_{\text{lead}}}{\rho_{\text{lead}}} = \frac{34 \text{ kg}}{11,300 \text{ kg/m}^3} = 3.0089 \times 10^{-3} \text{ m}^3$$

$$m_{\text{log}} = \frac{W_{\text{log}}}{g} = \frac{1540 \text{ N}}{9.807 \text{ m/s}^2}\left(\frac{1 \text{ kg} \cdot \text{m/s}^2}{1 \text{ N}}\right) = 157.031 \text{ kg}$$

Substituting, the volume and density of the log are determined to be

$$V_{\text{log}} = \frac{m_{\text{lead}} + m_{\text{log}}}{\rho_{\text{water}}} - V_{\text{lead}} = \frac{(34 + 157.031) \text{ kg}}{1000 \text{ kg/m}^3} - 3.0089 \times 10^{-3} \text{ m}^3 = 0.18802 \text{ m}^3$$

$$\rho_{\text{log}} = \frac{m_{\text{log}}}{V_{\text{log}}} = \frac{157.031 \text{ kg}}{0.18802 \text{ m}^3} = 835.174 \text{ kg/m}^3 \cong \mathbf{835 \text{ kg/m}^3}$$

Discussion Note that the log must be completely submerged for this analysis to be valid. Ideally, the lead weights must also be completely submerged, but this is not very critical because of the small volume of the lead weights.

3-172

Solution A rectangular gate that leans against the floor with an angle of 45° with the horizontal is to be opened from its lower edge by applying a normal force at its center. The minimum force F required to open the water gate is to be determined.

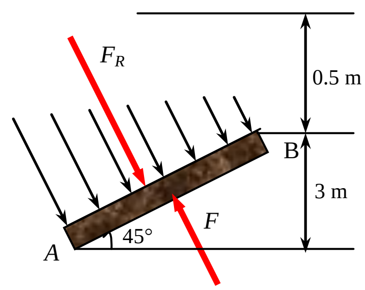
Assumptions **1** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience. **2** Friction at the hinge is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis The length of the gate and the distance of the upper edge of the gate (point B) from the free surface in the plane of the gate are

$$b = \frac{3\,\text{m}}{\sin 45°} = 4.243\,\text{m} \quad \text{and} \quad s = \frac{0.5\,\text{m}}{\sin 45°} = 0.7071\,\text{m}$$

$$h_C = \frac{h}{2} + 0.5 = \frac{3\,\text{m}}{2} + 0.5\,\text{m} = 2\,\text{m}$$



The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and multiplying it by the plate area gives the resultant hydrostatic on the surface,

$$F_R = P_{ave}A = \rho g h_C A$$

$$= (1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(2\,\text{m})[6 \times 4.243\,\text{m}^2]\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)$$

$$= 499.5\,\text{kN}$$

The distance of the pressure center from the free surface of water along the plane of the gate is

$$y_P = s + \frac{b}{2} + \frac{b^2}{12(s+b/2)} = 0.7071 + \frac{4.243}{2} + \frac{4.243^2}{12(0.7071 + 4.243/2)} = 3.359\,\text{m}$$

The distance of the pressure center from the hinge at point B is

$$L_P = y_P - s = 3.359 - 0.7071 = 2.652\,\text{m}$$

Taking the moment about point B and setting it equal to zero gives

$$\sum M_B = 0 \quad \rightarrow \quad F_R L_P = Fb/2$$

Solving for F and substituting, the required force to overcome the pressure is

$$F = \frac{2F_R L_P}{b} = \frac{2(499.5\,\text{kN})(2.652\,\text{m})}{4.243\,\text{m}} = 624.4\,\text{kN}$$

In addition to this, there is the weight of the gate itself, which must be added. In the 45° direction,

$$F_{gate} = W\cos(45°) = mg\cos(45°) = (280\,\text{kg})(9.81\,\text{m/s}^2)\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)\cos(45°) = 1.942\,\text{kN}$$

Thus, the total force required in the 45° direction is the sum of these two values,

$$F_{total} = 624.4 + 1.942 = 626.3\,\text{kN} \cong \mathbf{626\,kN} \text{ in the } 45° \text{ direction}$$

Discussion The applied force is inversely proportional to the distance of the point of application from the hinge, and the required force can be reduced by applying the force at a lower point on the gate. The weight of the gate is nearly negligible compared to the pressure force in this example; in reality, a heavier gate would probably be required.

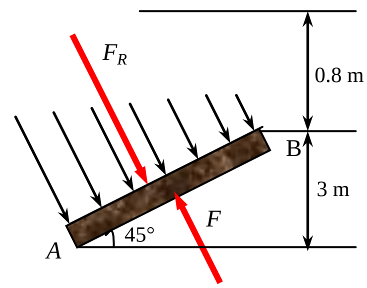
3-173

Solution A rectangular gate that leans against the floor with an angle of 45° with the horizontal is to be opened from its lower edge by applying a normal force at its center. The minimum force *F* required to open the water gate is to be determined.

Assumptions **1** Atmospheric pressure acts on both sides of the gate, and thus it can be ignored in calculations for convenience. **2** Friction at the hinge is negligible.

Properties We take the density of water to be 1000 kg/m^3 throughout.

Analysis The length of the gate and the distance of the upper edge of the gate (point *B*) from the free surface in the plane of the gate are



$$b = \frac{3\text{ m}}{\sin 45°} = 4.243\text{ m} \quad \text{and} \quad s = \frac{0.8\text{ m}}{\sin 45°} = 1.131\text{ m}$$

$$h_C = \frac{h}{2} + 0.5 = \frac{3\text{ m}}{2} + 0.8\text{ m} = 2.3\text{ m}$$

The average pressure on a surface is the pressure at the centroid (midpoint) of the surface, and multiplying it by the plate area gives the resultant hydrostatic on the surface,

$$F_R = P_{ave}A = \rho g h_C A$$

$$= (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(2.3\text{ m})[6 \times 4.243\text{ m}^2]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 574.4\text{ kN}$$

The distance of the pressure center from the free surface of water along the plane of the gate is

$$y_P = s + \frac{b}{2} + \frac{b^2}{12(s+b/2)} = 1.131 + \frac{4.243}{2} + \frac{4.243^2}{12(1.131+4.243/2)} = 3.714\text{ m}$$

The distance of the pressure center from the hinge at point *B* is

$$L_P = y_P - s = 3.714 - 1.131 = 2.583\text{ m}$$

Taking the moment about point *B* and setting it equal to zero gives

$$\sum M_B = 0 \quad \rightarrow \quad F_R L_P = Fb/2$$

Solving for *F* and substituting, the required force to overcome the pressure is

$$F = \frac{2F_R L_P}{b} = \frac{2(574.4\text{ kN})(2.583\text{ m})}{4.243\text{ m}} = 699.4\text{ kN}$$

In addition to this, there is the weight of the gate itself, which must be added. In the 45° direction,

$$F_{gate} = W\cos(45°) = mg\cos(45°) = (280\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\cos(45°) = 1.942\text{ kN}$$

Thus, the total force required in the 45° direction is the sum of these two values,

$$F_{total} = 699.4 + 1.942 = 701.3\text{ kN} \cong \mathbf{701\ kN} \text{ in the } 45° \text{ direction}$$

Discussion The applied force is inversely proportional to the distance of the point of application from the hinge, and the required force can be reduced by applying the force at a lower point on the gate. The weight of the gate is nearly negligible compared to the pressure force in this example; in reality, a heavier gate would probably be required.

Fundamentals of Engineering (FE) Exam Problems

3-174

The absolute pressure in a tank is measured to be 35 kPa. If the atmospheric pressure is 100 kPa, the vacuum pressure in the tank is

(*a*) 35 kPa (*b*) 100 kPa (*c*) 135 psi (*d*) 0 kPa (*e*) 65 kPa

Answer (*e*) 65 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P_abs=35 [kPa]
P_atm=100 [kPa]
P_vacuum=P_atm-P_abs
```

3-175

The pressure difference between the top and bottom of a water body with a depth of 10 m is (Take the density of water to be 1000 kg/m^3.)

(*a*) 98,100 kPa (*b*) 98.1 kPa (*c*) 100 kPa (*d*) 10 kPa (*e*) 1.9 kPa

Answer (*b*) 98.1 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h=10 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
DELTAP=rho*g*h*Convert(Pa, kPa)
```

3-176

The gage pressure in a pipe is measured by a manometer containing mercury (ρ = 13,600 kg/m^3). The top of the mercury is open to the atmosphere and the atmospheric pressure is 100 kPa. If the mercury column height is 24 cm, the gage pressure in the pipe is

(*a*) 32 kPa (*b*) 24 kPa (*c*) 76 kPa (*d*) 124 kPa (*e*) 68 kPa

Answer (*a*) 32 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h=0.24 [m]
P_atm=100 [kPa]
rho=13600 [kg/m^3]
g=9.81 [m/s^2]
P_gage=rho*g*h*Convert(Pa, kPa)
```

3-177

Consider a hydraulic car jack with a piston diameter ratio of 9. A person can lift a 2000-kg car by applying a force of

(*a*) 2000 N (*b*) 200 N (*c*) 19,620 N (*d*) 19.6 N (*e*) 18,000 N

Answer (*c*) 19,620 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D2\D1=9
m_car=2000 [kg]
g=9.81 [m/s^2]
F_car=m_car*g
A2\A1=D2\D1^2
F_car/F_person=A2\A1
```

3-178

The atmospheric pressure in a location is measured by a mercury (ρ = 13,600 kg/m^3) barometer. If the height of mercury column is 715 mm, the atmospheric pressure at that location is

(*a*) 85.6 kPa (*b*) 93.7 kPa (*c*) 95.4 kPa (*d*) 100 kPa (*e*) 101 kPa

Answer (*c*) 95.4 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h=0.715 [m]
rho=13600 [kg/m^3]
g=9.81 [m/s^2]
P_atm=rho*g*h*Convert(Pa, kPa)
```

3-179

A manometer is used to measure the pressure of a gas in a tank. The manometer fluid is water (ρ = 1000 kg/m^3) and the manometer column height is 1.8 m. If the local atmospheric pressure is 100 kPa, the absolute pressure within the tank is

(*a*) 17,760 kPa (*b*) 100 kPa (*c*) 180 kPa (*d*) 101 kPa (*e*) 118 kPa

Answer (*e*) 118 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1000 [kg/m^3]
h=1.8 [m]
P_atm=100 [kPa]
g=9.81 [m/s^2]
P=P_atm+rho*g*h*Convert(Pa, kPa)
```

3-180

Consider the vertical rectangular wall of a water tank with a width of 5 m and a height of 8 m. The other side of the wall is open to the atmosphere. The resultant hydrostatic force on this wall is

(*a*) 1570 kN (*b*) 2380 kN (*c*) 2505 kN (*d*) 1410 kN (*e*) 404 kPa

Answer (*a*) 1570 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
a=5 [m]
b=8 [m]
P_atm=101 [kPa]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
Area=a*b
P_C=(rho*g*b)/2*Convert(Pa, kPa)
F_R=P_C*Area
```

3-181

A vertical rectangular wall with a width of 20 m and a height of 12 m is holding a 7-m-deep water body. The resultant hydrostatic force acting on this wall is

(*a*) 1370 kN (*b*) 4807 kN (*c*) 8240 kN (*d*) 9740 kN (*e*) 11,670 kN

Answer (*b*) 4807 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
a=20 [m]
h=12 [m]
b=7 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
Area=a*b
P_C=rho*g*b/2*Convert(Pa, kPa)
F_R=P_C*Area
```

3-182

A vertical rectangular wall with a width of 20 m and a height of 12 m is holding a 7-m-deep water body. The line of action y_p for the resultant hydrostatic force on this wall is (disregard the atmospheric pressure)

(*a*) 5 m (*b*) 4.0 m (*c*) 4.67 m (*d*) 9.67 m (*e*) 2.33 m

Answer (*c*) 4.67 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
a=20 [m]
h=12 [m]
b=7 [m]
y_p=2*b/3
```

3-183

A rectangular plate with a width of 16 m and a height of 12 m is located 4 m below a water surface. The plate is tilted and makes a 35° angle with the horizontal. The resultant hydrostatic force acting on the top surface of this plate is

(*a*) 10,800 kN (*b*) 9745 kN (*c*) 8470 kN (*d*) 6400 kN (*e*) 5190 kN

Answer (*a*) 10,800 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
a=16 [m]
b=12 [m]
s=4 [m]
theta=35 [degree]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
Area=a*b
P_C=rho*g*(s+b/2)*Sin(theta)*Convert(Pa, kPa)
F_R=P_C*Area
```

3-184

A 2-m-long and 3-m-wide horizontal rectangular plate is submerged in water. The distance of the top surface from the free surface is 5 m. The atmospheric pressure is 95 kPa. Considering atmospheric pressure, the hydrostatic force acting on the top surface of this plate is

(*a*) 307 kN (*b*) 688 kN (*c*) 747 kN (*d*) 864 kN (*e*) 2950 kN

Answer (*d*) 864 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
a=2 [m]
b=3 [m]
h=5 [m]
P_atm=95 [kPa]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
Area=a*b
P_C=(P_atm+rho*g*h*Convert(Pa, kPa))
F_R=P_C*Area
```

3-185

A 1.8-m-diameter and 3.6-m-long cylindrical container contains a fluid with a specific gravity of 0.73. The container is positioned vertically and is full of the fluid. Disregarding atmospheric pressure, the hydrostatic force acting on the top and bottom surfaces of this container, respectively, are

(*a*) 0 kN, 65.6 kN (*b*) 65.6 kN, 0 kN (*c*) 65.6 kN, 65.6 kN (*d*) 25.5 kN, 0 kN

(*e*) 0 kN, 25.5 kN

Answer (*a*) 0 kN, 65.6 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=1.8 [m]
h=3.6 [m]
SG=0.73
rho_w=1000 [kg/m^3]
rho=SG*rho_w
g=9.81 [m/s^2]
A=pi*D^2/4
P_top=0
F_R_top=P_top*A*Convert(N, kN)
P_bottom=rho*g*h
F_R_bottom=P_bottom*A*Convert(N, kN)
```

3-186

Consider a 6-m-diameter spherical gate holding a body of water whose height is equal to the diameter of the gate. Atmospheric pressure acts on both sides of the gate. The horizontal component of the hydrostatic force acting on this curved surface is

(*a*) 709 kN (*b*) 832 kN (*c*) 848 kN (*d*) 972 kN (*e*) 1124 kN

Answer (*b*) 832 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=6 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
R=D/2
A=pi*R^2
P_C=rho*g*R*Convert(Pa, kPa)
F_x=P_C*A
```

3-187

Consider a 6-m-diameter spherical gate holding a body of water whose height is equal to the diameter of the gate. Atmospheric pressure acts on both sides of the gate. The vertical component of the hydrostatic force acting on this curved surface is

(*a*) 89 kN (*b*) 270 kN (*c*) 327 kN (*d*) 416 kN (*e*) 505 kN

Answer (*e*) 505 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=6 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
R=D/2
V=D^3/2-4/3*pi*R^3/2
m=rho*V
W=m*g*Convert(N, kN)
A=pi*R^2/2
h=0 [m]
P_C=rho*g*h
F_y=P_C*A*Convert(N, kN)
F_v=F_y-W
```

3-188

A 0.75-cm-diameter spherical object is completely submerged in water. The buoyant force acting on this object is

(*a*) 13,000 N (*b*) 9835 N (*c*) 5460 N (*d*) 2167 N (*e*) 1267 N

Answer (*d*) 2167 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.75 [m]
rho_f=1000 [kg/m^3]
g=9.81 [m/s^2]
V=pi*D^3/6
F_B=rho_f*g*V
```

3-189

A 3-kg object with a density of 7500 kg/m^3 is placed in water. The weight of this object in water is

(*a*) 29.4 N (*b*) 25.5 N (*c*) 14.7 N (*d*) 30 N (*e*) 3 N

Answer (*b*) 25.5 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_body=3 [kg]
rho_body=7500 [kg/m^3]
rho_f=1000 [kg/m^3]
g=9.81 [m/s^2]
V_body=m_body/rho_body
F_B=rho_f*g*V_body
W_inair=m_body*g
W_inwater=W_inair-F_B
```

3-190

A 7-m-diameter hot air balloon is neither rising nor falling. The density of atmospheric air is 1.3 kg/m^3. The total mass of the balloon including the people on board is

(*a*) 234 kg (*b*) 207 kg (*c*) 180 kg (*d*) 163 kg (*e*) 134 kg

Answer (*a*) 234 kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=7 [m]
rho_f=1.3 [kg/m^3]
V=pi*D^3/6
g=9.81 [m/s^2]
W=m*g
F_B=rho_f*g*V
W=F_B
```

3-191

A 10-kg object with a density of 900 kg/m^3 is placed in a fluid with a density of 1100 kg/m^3. The fraction of the volume of the object submerged in water is

(*a*) 0.637 (*b*) 0.716 (*c*) 0.818 (*d*) 0.90 (*e*) 1

Answer (*c*) 0.818

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_object=10 [kg]
rho_object=900 [kg/m^3]
rho_f=1100 [kg/m^3]
g=9.81 [m/s^2]
V_object=m_object/rho_object
W=m_object*g
F_B=rho_f*g*V_submerged
W=F_B
Fraction=V_submerged/V_object
```

3-192

Consider a cubical water tank with a side length of 3 m. The tank is half filled with water, and is open to the atmosphere with a pressure of 100 kPa. Now, a truck carrying this tank is accelerated at a rate of 5 m/s^2. The maximum pressure in the water is

(*a*) 115 kPa (*b*) 122 kPa (*c*) 129 kPa (*d*) 137 kPa (*e*) 153 kPa

Answer (*b*) 122 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
s=3 [m]
a_x=5 [m/s^2]
P_atm=100 [kPa]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
a_z=0 [m/s^2]
tan(theta)=a_x/(g+a_z)
DELTAz_max=s/2*tan(theta)
h=s/2+DELTAz_max
P=P_atm+rho*g*h*Convert(Pa, kPa)
```

3-193

A 15-cm-diameter, 40-cm-high vertical cylindrical container is partially filled with 25-cm-high water. Now the cylinder is rotated at a constant speed of 20 rad/s. The maximum height difference between the edge and the center of the free surface is

(*a*) 15 cm (*b*) 7.2 cm (*c*) 5.4 cm (*d*) 9.5 cm (*e*) 11.5 cm

Answer (*e*) 11.5 cm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.15 [m]
H=0.4 [m]
h_0=0.25 [m]
Omega=20 [rad/s]
R=D/2
g=9.81 [m/s^2]
DELTAz_max=Omega^2/(2*g)*R^2
```

3-194

A 20-cm-diameter, 40-cm-high vertical cylindrical container is partially filled with 25-cm-high water. Now the cylinder is rotated at a constant speed of 15 rad/s. The height of water at the center of the cylinder is

(*a*) 25 cm (*b*) 19.5 cm (*c*) 22.7 cm (*d*) 17.7 cm (*e*) 15 cm

Answer (*b*) 19.5 cm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.20 [m]
H=0.4 [m]
h_0=0.25 [m]
Omega=15 [rad/s]
R=D/2
g=9.81 [m/s^2]
rr=0 [m]
z_s=h_0-omega^2/(4*g)*(R^2-2*rr^2)
```

3-195

A 15-cm-diameter, 50-cm-high vertical cylindrical container is partially filled with 30-cm-high water. Now the cylinder is rotated at a constant speed of 20 rad/s. The pressure difference between the center and edge of the container at the base surface is

(*a*) 7327 Pa (*b*) 8750 Pa (*c*) 9930 Pa (*d*) 1045 Pa (*e*) 1125 Pa

Answer (*e*) 1125 Pa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.15 [m]
H=0.5 [m]
h_0=0.30 [m]
Omega=20 [rad/s]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
R=D/2
DELTAP=rho*omega^2*R^2/2
```

Design and Essay Problems

3-196

Solution We are to discuss the design of shoes that enable people to walk on water.

Discussion Students' discussions should be unique and will differ from each other.

3-197

Solution We are to discuss how to measure the volume of a rock without using any volume measurement devices.

Analysis The volume of a rock can be determined without using any volume measurement devices as follows: We weigh the rock in the air and then in the water. The difference between the two weights is due to the buoyancy force, which is equal to $F_B = \rho_{\text{water}} g V_{\text{body}}$. Solving this relation for V_{body} gives the volume of the rock.

Discussion Since this is an open-ended design problem, students may come up with different, but equally valid techniques.

3-198

Solution The maximum total weight and mass of a razor blade floating on water along with additional weights on the razor blade is to be estimated.

Assumptions **1** Surface tension acts only on the outer edges of the blade. **2** The blade is approximated as a rectangle for simplicity – three-dimensional corner effects are neglected. **3** In the limiting case, the water surface is vertical at the junction with the razor blade – as soon as the water starts to move over the razor blade surface, the razor blade would sink.

Properties The surface tension of water at 20°C is 0.073 N/m, and its density is 998.0 kg/m^3

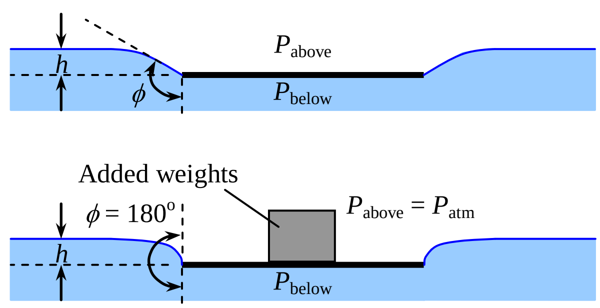
Analysis (*a*) Considering surface tension alone, the total upward force due to surface tension is the perimeter of the razor blade times the surface tension acting at contact angle ϕ. But here, the limiting case is when ϕ = 180°. This must balance the weight *W*,

$$W = -2\sigma_s\left(L+w\right)\cos\phi = -2\left(0.073\frac{\text{N}}{\text{m}}\right)\left(0.043+0.022\right)\ \text{m}\cdot\cos\left(180^{\text{o}}\right) = 0.00949\ \text{N}$$

which we convert to mass by dividing by the gravitational constant, namely,

$$m = \frac{W}{g} = \frac{0.00949\ \text{N}}{9.807\ \text{m/s}^2}\left(\frac{\text{kg}\cdot\text{m/s}^2}{\text{N}}\right)\left(\frac{1000\ \text{g}}{\text{kg}}\right) = 0.96768\ \text{g}$$

The values and properties are give to only two significant digits, so our final results are ***W* = 0.0095 N** and ***m* = 0.97 g**.



(*b*) Since the razor blade pushes down on the water, the pressure at the bottom of the blade is larger than that at the top of the blade due to hydrostatic effects as sketched. Thus, more weight can be supported due to the difference in pressure. Since $P_{\text{below}} = P_{\text{atm}} + \rho gh$, we write

$$W = -2\sigma_s\left(L+w\right)\cos\phi\ + \rho ghLw$$

However, from the hint, we know also that the maximum possible depth is $h = \sqrt{\dfrac{2\sigma_s}{\rho g}}$. When we set ϕ = 180° and substitute this expression for *h*, we can solve for *W*,

$$\begin{aligned}W &= -2\sigma_s\left(L+w\right)\cos\phi + \sqrt{2\rho g\sigma_s}Lw \\ &= -2\left(0.073\frac{\text{N}}{\text{m}}\right)\left(0.043+0.022\right)\ \text{m}\cdot\cos\left(180^{\text{o}}\right) \\ &\quad + \sqrt{2\left(998.0\frac{\text{kg}}{\text{m}^3}\right)\left(9.807\frac{\text{m}}{\text{s}^2}\right)\left(0.073\frac{\text{N}}{\text{m}}\right)\left(\frac{\text{N}}{\text{kg}\cdot\text{m/s}^2}\right)}\left(0.043\ \text{m}\right)\left(0.022\ \text{m}\right) \\ &= 0.045250\ \text{N}\end{aligned}$$

Again, since the values given to only two significant digits, our final results are ***W* = 0.0495 N** and ***m* = 4.6 g**.

Discussion The hydrostatic pressure component has greatly increased the amount of weight that can be supported, by a factor of almost 5.

Solutions Manual for

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CHAPTER 4
FLUID KINEMATICS

Introductory Problems

4-1C

Solution We are to define and explain kinematics and fluid kinematics.

Analysis *Kinematics* means **the study of *motion***. *Fluid kinematics* is the **study of how fluids flow and how to describe fluid motion**. Fluid kinematics deals with describing the motion of fluids without considering (or even understanding) the forces and moments that *cause* the motion.

Discussion Fluid kinematics deals with such things as describing how a fluid particle translates, distorts, and rotates, and how to visualize flow fields.

4-2C

Solution We are to discuss the difference between derivative operators d and ∂.

Analysis Derivative operator d is a **total derivative**, and implies that **the dependent variable is a function of only one independent variable**. On the other hand, derivative operator ∂ is a **partial derivative**, and implies that **the dependent variable is a function of more than one independent variable**. When $\partial u/\partial x$ appears in an equation, we immediately know that **u is a function of x and at least one other independent variable**.

Discussion In our study of fluid mechanics, velocity is usually a function of more than one variable, although for some simple problems, we approximate it as a function of only one variable so that the problem can be solved analytically.

4-3

Solution We are to write an equation for centerline speed through a nozzle, given that the flow speed increases parabolically.

Assumptions **1** The flow is steady. **2** The flow is axisymmetric. **3** The water is incompressible.

Analysis A general equation for a parabola in the x direction is

General parabolic equation:
$$u = a + b(x - c)^2 \qquad (1)$$

We have two boundary conditions, namely at $x = 0$, $u = u_{\text{entrance}}$ and at $x = L$, $u = u_{\text{exit}}$. By inspection, Eq. 1 is satisfied by setting $c = 0$, $a = u_{\text{entrance}}$ and $b = (u_{\text{exit}} - u_{\text{entrance}})/L^2$. Thus, Eq. 1 becomes

Parabolic speed:
$$\boxed{u = u_{\text{entrance}} + \frac{(u_{\text{exit}} - u_{\text{entrance}})}{L^2}x^2} \qquad (2)$$

Discussion You can verify Eq. 2 by plugging in $x = 0$ and $x = L$.

4-4

Solution For a given velocity field we are to find out if there is a stagnation point. If so, we are to calculate its location.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is

$$\vec{V} = (u, v) = \left(a^2 - (b - cx)^2\right)\vec{i} + \left(-2cby + 2c^2xy\right)\vec{j} \qquad (1)$$

At a stagnation point, both *u* and *v* must equal zero. At any point (*x*,*y*) in the flow field, the velocity components *u* and *v* are obtained from Eq. 1,

Velocity components:
$$u = a^2 - (b - cx)^2 \qquad v = -2cby + 2c^2xy \qquad (2)$$

Setting these to zero and solving simultaneously yields

Stagnation point:
$$\begin{aligned} 0 &= a^2 - (b - cx)^2 \qquad & x &= \frac{b-a}{c} \\ v &= -2cby + 2c^2xy \qquad & y &= 0 \end{aligned} \qquad (3)$$

So, **yes there is a stagnation point**; its location is ***x* = (*b* – *a*)/*c*, *y* = 0**.

Discussion If the flow were three-dimensional, we would have to set *w* = 0 as well to determine the location of the stagnation point. In some flow fields there is more than one stagnation point.

4-5

Solution For a given velocity field we are to find out if there is a stagnation point. If so, we are to calculate its location.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is

$$\vec{V} = (u, v) = (-0.781 - 4.67x)\vec{i} + (-3.54 + 4.67y)\vec{j} \qquad (1)$$

At a stagnation point, both *u* and *v* must equal zero. At any point (*x*,*y*) in the flow field, the velocity components *u* and *v* are obtained from Eq. 1,

Velocity components:
$$u = -0.781 - 4.67x \qquad v = -3.54 + 4.67y \qquad (2)$$

Setting these to zero yields

Stagnation point:
$$\begin{aligned} 0 &= -0.781 - 4.67x \qquad & x &= -0.16724 \\ 0 &= -3.54 + 4.67y \qquad & y &= 0.75803 \end{aligned} \qquad (3)$$

So, **yes there is a stagnation point**; its location is ***x* = -0.167, *y* = 0.758** (to 3 digits).

Discussion If the flow were three-dimensional, we would have to set *w* = 0 as well to determine the location of the stagnation point. In some flow fields there is more than one stagnation point.

4-6

Solution For a given velocity field we are to find out if there is a stagnation point. If so, we are to calculate its location.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is

$$\vec{V} = (u, v) = (0.66 + 2.1x)\vec{i} + (-2.7 - 2.1y)\vec{j} \qquad (1)$$

At a stagnation point, both *u* and *v* must equal zero. At any point (*x*,*y*) in the flow field, the velocity components *u* and *v* are obtained from Eq. 1,

Velocity components: $u = 0.66 + 2.1x \qquad v = -2.7 - 2.1y \qquad (2)$

Stagnation point:

$$\begin{aligned} 0 &= 0.66 + 2.1x & x &= -0.314 \\ 0 &= -2.7 - 2.1y & y &= -1.29 \end{aligned} \qquad (3)$$

So, **yes there is a stagnation point**; its location is ***x* = -0.314, *y* = -1.29** (to 3 digits).

Discussion If the flow were three-dimensional, we would have to set *w* = 0 as well to determine the location of the stagnation point. In some flow fields there is more than one stagnation point.

Lagrangian and Eulerian Descriptions

4-7C

Solution We are to define the Eulerian description of fluid motion, and explain how it differs from the Lagrangian description.

Analysis In the *Eulerian description* of fluid motion, we are concerned with ***field variables*, such as velocity, pressure, temperature, etc., as functions of space and time within a flow domain or control volume**. In contrast to the Lagrangian method, fluid flows into and out of the Eulerian flow domain, and we do not keep track of the motion of particular identifiable fluid particles.

Discussion The Eulerian method of studying fluid motion is not as "natural" as the Lagrangian method since the fundamental conservation laws apply to moving particles, not to fields.

4-8C

Solution We are to compare the Lagrangian method to the study of systems and control volumes and determine to which of these it is most similar.

Analysis **The Lagrangian method is more similar to system analysis** (i.e., *closed* system analysis). In both cases, we follow a mass of fixed identity as it moves in a flow. In a control volume analysis, on the other hand, mass moves into and out of the control volume, and we don't follow any particular chunk of fluid. Instead we analyze whatever fluid happens to be inside the control volume at the time.

Discussion In fact, the Lagrangian analysis is the same as a system analysis in the limit as the size of the system shrinks to a point.

4-9C

Solution We are to define the Lagrangian description of fluid motion.

Analysis In the *Lagrangian description* of fluid motion, **individual fluid particles** (fluid elements composed of a fixed, identifiable mass of fluid) **are followed**.

Discussion The Lagrangian method of studying fluid motion is similar to that of studying billiard balls and other solid objects in physics.

4-10C

Solution We are to determine whether a measurement is Lagrangian or Eulerian.

Analysis Since the probe is fixed in space and the fluid flows around it, we are *not* following individual fluid particles as they move. Instead, we are measuring a field variable at a particular location in space. Thus this is an **Eulerian measurement**.

Discussion If a neutrally buoyant probe were to move with the flow, its results would be Lagrangian measurements – following fluid particles.

4-11C

Solution We are to determine whether a measurement is Lagrangian or Eulerian.

Analysis Since the probe moves with the flow and is neutrally buoyant, we are following individual fluid particles as they move through the pump. Thus this is a **Lagrangian measurement**.

Discussion If the probe were instead fixed at one location in the flow, its results would be Eulerian measurements.

4-12C

Solution We are to define a steady flow field in the Eulerian description, and discuss particle acceleration in such a flow.

Analysis A flow field is defined as *steady* in the Eulerian frame of reference when **properties at any point in the flow field do not change with respect to time**. In such a flow field, individual fluid particles may still experience non-zero acceleration – the answer to the question is **yes**.

Discussion Although velocity is not a function of time in a steady flow field, its total derivative with respect to time $\left(\vec{a} = d\vec{V}/dt\right)$ is not necessarily zero since the acceleration is composed of a local (unsteady) part which is zero and an advective part which is not necessarily zero.

4-13C

Solution We are to list three alternate names for material derivative.

Analysis The material derivative is also called **total derivative**, **particle derivative**, **Eulerian derivative**, **Lagrangian derivative,** and **substantial derivative**. "Total" is appropriate because the material derivative includes both local (unsteady) and convective parts. "Particle" is appropriate because it stresses that the material derivative is one following fluid particles as they move about in the flow field. "Eulerian" is appropriate since the material derivative is used to transform from Lagrangian to Eulerian reference frames. "Lagrangian" is appropriate since the material derivative is used to transform from Lagrangian to Eulerian reference frames. Finally, "substantial" is not as clear of a term for the material derivative, and we are not sure of its origin.

Discussion All of these names emphasize that we are following a fluid particle as it moves through a flow field.

4-14C

Solution We are to determine whether a measurement is Lagrangian or Eulerian.

Analysis Since the weather balloon moves with the air and is neutrally buoyant, we are following individual "fluid particles" as they move through the atmosphere. Thus this is a **Lagrangian measurement**. Note that in this case the "fluid particle" is huge, and can follow gross features of the flow – the balloon obviously cannot follow small scale turbulent fluctuations in the atmosphere.

Discussion When weather monitoring instruments are mounted on the roof of a building, the results are Eulerian measurements.

4-15C

Solution We are to determine whether a measurement is Lagrangian or Eulerian.

Analysis Relative to the airplane, the probe is fixed and the air flows around it. We are *not* following individual fluid particles as they move. Instead, we are measuring a field variable at a particular location in space relative to the moving airplane. Thus this is an **Eulerian measurement**.

Discussion The airplane is moving, but it is not moving with the flow.

4-16C

Solution We are to compare the Eulerian method to the study of systems and control volumes and determine to which of these it is most similar.

Analysis **The Eulerian method is more similar to control volume analysis**. In both cases, mass moves into and out of the flow domain or control volume, and we don't follow any particular chunk of fluid. Instead we analyze whatever fluid happens to be inside the control volume at the time.

Discussion In fact, the Eulerian analysis is the same as a control volume analysis except that Eulerian analysis is usually applied to infinitesimal volumes and differential equations of fluid flow, whereas control volume analysis usually refers to finite volumes and integral equations of fluid flow.

4-17

Solution We are to calculate the material acceleration for a given velocity field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is

$$\vec{V} = (u, v) = (U_0 + bx)\vec{i} - by\vec{j} \qquad (1)$$

The acceleration field components are obtained from its definition (the material acceleration) in Cartesian coordinates,

$$\begin{aligned} a_x &= \frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = 0 + (U_0 + bx)b + (-by)0 + 0 \\ a_y &= \frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} = 0 + (U_0 + bx)0 + (-by)(-b) + 0 \end{aligned} \qquad (2)$$

where the unsteady terms are zero since this is a steady flow, and the terms with *w* are zero since the flow is two-dimensional. Eq. 2 simplifies to

Material acceleration components: $$\boxed{a_x = b(U_0 + bx) \qquad a_y = b^2 y} \qquad (3)$$

In terms of a vector,

Material acceleration vector: $$\boxed{\vec{a} = b(U_0 + bx)\vec{i} + b^2 y\vec{j}} \qquad (4)$$

Discussion For positive *x* and *b*, fluid particles accelerate in the positive *x* direction. Even though this flow is steady, there is still a non-zero acceleration field.

4-18

Solution For a given pressure and velocity field, we are to calculate the rate of change of pressure following a fluid particle.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The pressure field is

Pressure field: $$P = P_0 - \frac{\rho}{2}\left[2U_0 bx + b^2\left(x^2 + y^2\right)\right] \qquad (1)$$

By definition, the material derivative, when applied to pressure, produces the rate of change of pressure following a fluid particle. Using Eq. 1 and the velocity components from the previous problem,

$$\begin{aligned} \frac{DP}{Dt} &= \underbrace{\cancel{\frac{\partial P}{\partial t}}}_{\text{Steady}} + u\frac{\partial P}{\partial x} + v\frac{\partial P}{\partial y} + \underbrace{\cancel{w\frac{\partial P}{\partial z}}}_{\text{Two-dimensional}} \\ &= (U_0 + bx)\left(-\rho U_0 b - \rho b^2 x\right) + (-by)\left(-\rho b^2 y\right) \end{aligned} \qquad (2)$$

where the unsteady term is zero since this is a steady flow, and the term with *w* is zero since the flow is two-dimensional. Eq. 2 simplifies to the following rate of change of pressure following a fluid particle:

$$\boxed{\frac{DP}{Dt} = \rho\left[-U_0^{\,2} b - 2U_0 b^2 x + b^3\left(y^2 - x^2\right)\right]} \qquad (3)$$

Discussion The material derivative can be applied to any flow property, scalar or vector. Here we apply it to the pressure, a scalar quantity.

4-19

Solution For a given velocity field we are to calculate the acceleration.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity components are

Velocity components: $u = 1.85 + 2.33x + 0.656y \qquad v = 0.754 - 2.18x - 2.33y$ (1)

The acceleration field components are obtained from its definition (the material acceleration) in Cartesian coordinates,

$$\begin{aligned} a_x &= \frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = 0 + \left(1.85 + 2.33x + 0.656y\right)\left(2.33\right) + \left(0.754 - 2.18x - 2.33y\right)\left(0.656\right) + 0 \\ a_y &= \frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} = 0 + \left(1.85 + 2.33x + 0.656y\right)\left(-2.18\right) + \left(0.754 - 2.18x - 2.33y\right)\left(-2.33\right) + 0 \end{aligned} \qquad (2)$$

where the unsteady terms are zero since this is a steady flow, and the terms with *w* are zero since the flow is two-dimensional. Eq. 2 simplifies to

Acceleration components: $\boxed{a_x = 4.8051 + 3.9988x \qquad a_y = -5.7898 + 3.9988y}$ (3)

At the point (*x*,*y*) = (-1,2), the acceleration components of Eq. 3 are

Acceleration components at (-1,2): $a_x = 0.80628 \cong \mathbf{0.806} \qquad a_y = 2.2078 \cong \mathbf{2.21}$

Discussion The final answers are given to three significant digits. No units are given in either the problem statement or the answers. We assume that the coefficients have appropriate units.

4-20

Solution For a given velocity field we are to calculate the acceleration.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity components are

Velocity components: $u = 0.205 + 0.97x + 0.851y \qquad v = -0.509 + 0.953x - 0.97y$ (1)

The acceleration field components are obtained from its definition (the material acceleration) in Cartesian coordinates,

$$\begin{aligned} a_x &= \frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = 0 + (0.205 + 0.97x + 0.851y)(0.97) + (-0.509 + 0.953x - 0.97y)(0.851) + 0 \\ a_y &= \frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} = 0 + (0.205 + 0.97x + 0.851y)(0.953) + (-0.509 + 0.953x - 0.97y)(-0.97) + 0 \end{aligned} \tag{2}$$

where the unsteady terms are zero since this is a steady flow, and the terms with *w* are zero since the flow is two-dimensional. Eq. 2 simplifies to

Acceleration components:

$$\begin{aligned} a_x &= -0.234309 + 1.751903x \\ a_y &= 0.689095 + 1.751903y \end{aligned} \tag{3}$$

At the point (*x*,*y*) = (2, 1.5), the acceleration components of Eq. 3 are

Acceleration components at (2, 1.5):

$$\begin{aligned} a_x &= 3.269497 \cong \mathbf{3.27} \\ a_y &= 3.31699 \cong \mathbf{3.32} \end{aligned}$$

Discussion The final answers are given to three significant digits. No units are given in either the problem statement or the answers. We assume that the coefficients have appropriate units.

4-21

Solution For a given velocity field we are to calculate the streamline that will pass through a given point.

Assumptions **1** The flow is steady. **2** The flow is three-dimensional in the *x*-*y*-*z* plane.

Analysis

$$\frac{dx}{u}=\frac{dy}{v}=\frac{dz}{w}$$

$$\frac{dx}{3x}=\frac{dy}{-2y}=\frac{dz}{2z}$$

For the first two pairs we have

$$\frac{\mathrm{dx}}{3\mathrm{x}}=\frac{dy}{-2y} \text{ or } \frac{1}{3}\ln x=-\frac{1}{2}\ln y+\ln c_1$$

$$x^{\frac{1}{3}}y^{\frac{-1}{2}}=c_1 \text{ or,}$$

For the point given $x=1$, y = 1, z = 0

$$1^{\frac{1}{3}}.1^{\frac{-1}{2}}=c_1 \Rightarrow c_1=1 \text{ or } \sqrt[3]{x}=\sqrt{y},\ y=x^{2/3}$$

on the other hand,

$$\frac{dx}{2z}=\frac{dx}{3x} \text{ or } \frac{1}{2}\ln z-\frac{1}{3}\ln x=\ln c$$

$$\sqrt{z}/x^{1/3}=c \text{ or } \frac{\mathrm{z}}{\mathrm{x}^{2/3}}=c \Rightarrow z=c.x^{2/3}$$

A(1,1,0),

$$0=c.1^{2/3},\ c=0 \text{ or } \mathrm{z}=0$$

Therefore the streamline is given by,

$$y=x^{2/3},\ z=0$$

4-22

Solution We are to write an equation for centerline speed through a diffuser, given that the flow speed decreases parabolically.

Assumptions **1** The flow is steady. **2** The flow is axisymmetric.

Analysis A general equation for a parabola in *x* is

General parabolic equation: $$u=a+b\left(x-c\right)^2 \quad (1)$$

We have two boundary conditions, namely at $x = 0$, $u = u_{\text{entrance}}$ and at $x = L$, $u = u_{\text{exit}}$. By inspection, Eq. 1 is satisfied by setting $c = 0$, $a = u_{\text{entrance}}$ and $b = (u_{\text{exit}} - u_{\text{entrance}})/L^2$. Thus, Eq. 1 becomes

Parabolic speed: $$\boxed{u=u_{\text{entrance}}+\frac{\left(u_{\text{exit}}-u_{\text{entrance}}\right)}{L^2}x^2} \quad (2)$$

Discussion You can verify Eq. 2 by plugging in $x = 0$ and $x = L$.

4-23

Solution We are to generate an expression for the fluid acceleration for a given velocity, and then calculate its value at two x locations.

Assumptions **1** The flow is steady. **2** The flow is axisymmetric.

Analysis In the previous problem, we found that along the centerline,

Speed along centerline of diffuser:
$$u = u_{\text{entrance}} + \frac{\left(u_{\text{exit}} - u_{\text{entrance}}\right)}{L^2} x^2 \tag{1}$$

To find the acceleration in the x-direction, we use the material acceleration,

Acceleration along centerline of diffuser:
$$a_x = \cancel{\frac{\partial u}{\partial t}} + u\frac{\partial u}{\partial x} + \cancel{v\frac{\partial u}{\partial y}} + \cancel{w\frac{\partial u}{\partial z}} \tag{2}$$

The first term in Eq. 2 is zero because the flow is steady. The last two terms are zero because the flow is axisymmetric, which means that along the centerline there can be no v or w velocity component. We substitute Eq. 1 for u to obtain

Acceleration along centerline of diffuser:
$$a_x = u\frac{\partial u}{\partial x} = \left(u_{\text{entrance}} + \frac{\left(u_{\text{exit}} - u_{\text{entrance}}\right)}{L^2} x^2\right)(2)\frac{\left(u_{\text{exit}} - u_{\text{entrance}}\right)}{L^2} x$$

or

$$\boxed{a_x = 2u_{\text{entrance}}\frac{\left(u_{\text{exit}} - u_{\text{entrance}}\right)}{L^2} x + 2\frac{\left(u_{\text{exit}} - u_{\text{entrance}}\right)^2}{L^4} x^3} \tag{3}$$

At the given locations, we substitute the given values. At $x = 0$,

Acceleration along centerline of diffuser at x = 0:
$$a_x\left(x = 0\right) = \mathbf{0} \tag{4}$$

At $x = 1.0$ m,

Acceleration along centerline of diffuser at x = 1.0 m:
$$a_x\left(x = 1.0\text{ m}\right) = 2\left(24.3\text{ m/s}\right)\frac{\left(-7.5\text{ m/s}\right)}{\left(1.56\text{ m}\right)^2}\left(1.0\text{ m}\right) + 2\frac{\left(-7.5\text{ m/s}\right)^2}{\left(1.56\text{ m}\right)^4}\left(1.0\text{ m}\right)^3 \tag{5}$$
$$= -130.782\text{ m/s}^2 \cong \mathbf{-131\text{ m/s}^2}$$

Discussion a_x is negative implying that fluid particles are *decelerated* along the centerline of the diffuser, even though the flow is steady. Because of the parabolic nature of the velocity field, the acceleration is zero at the entrance of the diffuser, but its magnitude increases rapidly downstream.

4-24

Solution For a given velocity field we are to calculate the acceleration.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity components are

Velocity components: $$u = 0.523 - 1.88x + 3.94y \qquad v = -2.44 + 1.26x + 1.88y \tag{1}$$

The acceleration field components are obtained from its definition (the material acceleration) in Cartesian coordinates,

$$\begin{aligned} a_x &= \frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = 0 + (0.523 - 1.88x + 3.94y)(-1.88) + (-2.44 + 1.26x + 1.88y)(3.94) + 0 \\ a_y &= \frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} = 0 + (0.523 - 1.88x + 3.94y)(1.26) + (-2.44 + 1.26x + 1.88y)(1.88) + 0 \end{aligned} \tag{2}$$

where the unsteady terms are zero since this is a steady flow, and the terms with *w* are zero since the flow is two-dimensional. Eq. 2 simplifies to

Acceleration components: $$\boxed{a_x = -10.59684 + 8.4988x \qquad a_y = -3.92822 + 8.4988y} \tag{3}$$

At the point (*x*,*y*) = (-1.55, 2.07), the acceleration components of Eq. 3 are

Acceleration components at (-1.55, 2.07): $a_x = -23.76998 \cong \mathbf{-23.8}$ $\qquad a_y = 13.6643 \cong \mathbf{13.7}$

Discussion The final answers are given to three significant digits. No units are given in either the problem statement or the answers. We assume that the coefficients have appropriate units.

4-25

Solution We are to generate an expression for the fluid acceleration for a given velocity.

Assumptions **1** The flow is steady. **2** The flow is axisymmetric. **3** The water is incompressible.

Analysis In Problem 4-2 we found that along the centerline,

Speed along centerline of nozzle: $$u = u_{\text{entrance}} + \frac{(u_{\text{exit}} - u_{\text{entrance}})}{L^2}x^2 \tag{1}$$

To find the acceleration in the *x*-direction, we use the material acceleration,

Acceleration along centerline of nozzle: $$a_x = \cancel{\frac{\partial u}{\partial t}} + u\frac{\partial u}{\partial x} + \cancel{v\frac{\partial u}{\partial y}} + \cancel{w\frac{\partial u}{\partial z}} \tag{2}$$

The first term in Eq. 2 is zero because the flow is steady. The last two terms are zero because the flow is axisymmetric, which means that along the centerline there can be no *v* or *w* velocity component. We substitute Eq. 1 for *u* to obtain

Acceleration along centerline of nozzle: $$a_x = u\frac{\partial u}{\partial x} = \left(u_{\text{entrance}} + \frac{(u_{\text{exit}} - u_{\text{entrance}})}{L^2}x^2\right)(2)\frac{(u_{\text{exit}} - u_{\text{entrance}})}{L^2}x \tag{3}$$

or

$$\boxed{a_x = 2u_{\text{entrance}}\frac{(u_{\text{exit}} - u_{\text{entrance}})}{L^2}x + 2\frac{(u_{\text{exit}} - u_{\text{entrance}})^2}{L^4}x^3} \tag{4}$$

Discussion Fluid particles are accelerated along the centerline of the nozzle, even though the flow is steady.

Flow Patterns and Flow Visualization

4-26C

Solution We are to define pathline and discuss what pathlines indicate.

Analysis A *pathline* is the **actual path traveled by an individual fluid particle over some time period**. It indicates the exact route along which a fluid particle travels from its starting point to its ending point. Unlike streamlines, pathlines are not instantaneous, but involve a finite time period.

Discussion If a flow field is steady, streamlines, pathlines, and streaklines are identical.

4-27C

Solution We are to determine what kind of flow visualization is seen in a photograph.

Analysis Since the picture is a snapshot of dye streaks in water, each streak shows the time history of dye that was introduced earlier from a port in the body. Thus these are **streaklines**. Since the flow appears to be steady, these streaklines are the same as pathlines and streamlines.

Discussion It is assumed that the dye follows the flow of the water. If the dye is of nearly the same density as the water, this is a reasonable assumption.

4-28C

Solution We are to define streamline and discuss what streamlines indicate.

Analysis A *streamline* is a **curve that is everywhere tangent to the instantaneous local velocity vector**. It indicates the instantaneous direction of fluid motion throughout the flow field.

Discussion If a flow field is steady, streamlines, pathlines, and streaklines are identical.

4-29C

Solution We are to define streakline and discuss the difference between streaklines and streamlines.

Analysis A *streakline* is the **locus of fluid particles that have passed sequentially through a prescribed point in the flow**. Streaklines are very different than streamlines. Streamlines are instantaneous curves, everywhere tangent to the local velocity, while streaklines are produced over a finite time period. In an unsteady flow, streaklines distort and then retain features of that distorted shape even as the flow field changes, whereas streamlines change instantaneously with the flow field.

Discussion If a flow field is steady, streamlines and streaklines are identical.

4-30C

Solution We are to determine what kind of flow visualization is seen in a photograph.

Analysis Since the picture is a snapshot of dye streaks in water, each streak shows the time history of dye that was introduced earlier from a port in the body. Thus these are **streaklines**. Since the flow appears to be unsteady, these streaklines are *not* the same as pathlines or streamlines.

Discussion It is assumed that the dye follows the flow of the water. If the dye is of nearly the same density as the water, this is a reasonable assumption.

4-31C

Solution We are to determine what kind of flow visualization is seen in a photograph.

Analysis Since the picture is a snapshot of smoke streaks in air, each streak shows the time history of smoke that was introduced earlier from the smoke wire. Thus these are **streaklines**. Since the flow appears to be unsteady, these streaklines are *not* the same as pathlines or streamlines.

Discussion It is assumed that the smoke follows the flow of the air. If the smoke is neutrally buoyant, this is a reasonable assumption. In actuality, the smoke rises a bit since it is hot; however, the air speeds are high enough that this effect is negligible.

4-32C

Solution We are to determine what kind of flow visualization is seen in a photograph.

Analysis Since the picture is a time exposure of air bubbles in water, each white streak shows the path of an individual air bubble. Thus these are **pathlines**. Since the outer flow (top and bottom portions of the photograph) appears to be steady, these pathlines are the same as streaklines and streamlines.

Discussion It is assumed that the air bubbles follow the flow of the water. If the bubbles are small enough, this is a reasonable assumption.

4-33C

Solution We are to define timeline and discuss how timelines can be produced in a water channel. We are also to describe an application where timelines are more useful than streaklines.

Analysis A *timeline* is a **set of adjacent fluid particles that were marked at the same instant of time**. Timelines can be produced in a water flow by using a hydrogen bubble wire. There are also techniques in which a chemical reaction is initiated by applying current to the wire, changing the fluid color along the wire. Timelines are more useful than streaklines when the uniformity of a flow is to be visualized. Another application is to visualize the velocity profile of a boundary layer or a channel flow.

Discussion Timelines differ from streamlines, streaklines, and pathlines even if the flow is steady.

4-34C

Solution For each case we are to decide whether a vector plot or contour plot is most appropriate, and we are to explain our choice.

Analysis In general, contour plots are most appropriate for scalars, while vector plots are necessary when vectors are to be visualized.

(*a*) A **contour plot** of speed is most appropriate since fluid speed is a scalar.
(*b*) A **vector plot** of velocity vectors would clearly show where the flow separates. Alternatively, a vorticity contour plot of vorticity normal to the plane would also show the separation region clearly.
(*c*) A **contour plot** of temperature is most appropriate since temperature is a scalar.
(*d*) A **contour plot** of this component of vorticity is most appropriate since one component of a vector is a scalar.

Discussion There are other options for case (b) – temperature contours can also sometimes be used to identify a separation zone.

4-35

Solution For a given velocity field we are to generate an equation for the streamlines.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The steady, two-dimensional velocity field of Problem 4-17 is

Velocity field:
$$\vec{V} = (u, v) = (U_0 + bx)\vec{i} - by\vec{j} \qquad (1)$$

For two-dimensional flow in the *x*-*y* plane, streamlines are given by

Streamlines in the x-y plane:
$$\left.\frac{dy}{dx}\right)_{\text{along a streamline}} = \frac{v}{u} \qquad (2)$$

We substitute the *u* and *v* components of Eq. 1 into Eq. 2 and rearrange to get

$$\frac{dy}{dx} = \frac{-by}{U_0 + bx}$$

We solve the above differential equation by separation of variables:

$$-\int \frac{dy}{by} = \int \frac{dx}{U_0 + bx}$$

Integration yields

$$-\frac{1}{b}\ln(by) = \frac{1}{b}\ln(U_0 + bx) + \frac{1}{b}\ln C_1 \qquad (3)$$

where we have set the constant of integration as the natural logarithm of some constant C_1, with a constant in front in order to simplify the algebra (notice that the factor of 1/*b* can be removed from each term in Eq. 3). When we recall that ln(*ab*) = ln*a* + ln*b*, and that –ln*a* = ln(1/*a*), Eq. 3 simplifies to

Equation for streamlines:
$$\boxed{y = \frac{C}{(U_0 + bx)}} \qquad (4)$$

The new constant *C* is related to C_1, and is introduced for simplicity.

Discussion Each value of constant *C* yields a unique streamline of the flow.

4-36

Solution For a given velocity field we are to calculate the pathline of a particle at a given location.

Analysis

u= 4x

v= 5y+3 are the velocity components.

w= $3t^2$

From the definition,

$$u = \frac{dx}{dt} = 4x$$

$$v = \frac{dy}{dt} = 5y + 3$$

$$w = \frac{dz}{dt} = 3t^2$$

For t=1 sec, the location of the particle is (x,y,z)=(1,2,4) that is, x=1m, y=2m, z=4m. Integrating given functions,

$$\frac{dx}{x} = 4dt \rightarrow \ln x\Big|_1^x = 4\,t\Big|_1^t \rightarrow \quad \text{lnx-ln1= 4(t-1)}$$

lnx= 4(t-1)

$$\frac{dy}{5y+3} = dt \rightarrow \frac{1}{5}\ln(5y+3)\Big|_2^y = t\Big|_1^t$$

ln(5y+3) - ln(13)= 5(t-1)

ln(5y+3)= ln(13) + 5(t-1)

$$\int dz = \int 3t^2 dt \rightarrow z\Big|_4^z = t^3\Big|_1^t \rightarrow \text{z-4=}\ t^3 \text{ -1}$$

$$z = t^3 + 3$$

We have 3 equations. If we eliminate the time (t) we get path line function in terms of x, y, z only.

lnx= 4t - 4 .... (1)

$$\ln\left(\frac{5y+3}{13}\right) = 5t - 5 \text{ ....(2)}$$

$$z = t^3 + 3 \text{ .... (3)}$$

Adding (1) and (2) would yield,

$$\ln x + \ln\left(\frac{5y+3}{13}\right) = 9t - 9,$$

Solve for t

$$\ln\left(\frac{5xy+3x}{13}\right) + 9 = 9t$$

or

$$t = 1 + \ln\left(\frac{5xy+3x}{13}\right),$$

Substituting this t into Eq. 3 leads to

$$z = f(x,y) = \left[1 + \ln\left(\frac{5xy+3x}{13}\right)\right]^3 + 3$$

4-37

Solution For a given velocity field we are to generate an equation for the streamlines and sketch several streamlines in the first quadrant.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is given by

$$\vec{V} = (u, v) = (4.35 + 0.656x)\vec{i} + (-1.22 - 0.656y)\vec{j} \tag{1}$$

For two-dimensional flow in the *x*-*y* plane, streamlines are given by

Streamlines in the x-y plane:

$$\left.\frac{dy}{dx}\right)_{\text{along a streamline}} = \frac{v}{u} \tag{2}$$

We substitute the *u* and *v* components of Eq. 1 into Eq. 2 and rearrange to get

$$\frac{dy}{dx} = \frac{-1.22 - 0.656y}{4.35 + 0.656x}$$

We solve the above differential equation by separation of variables:

$$\frac{dy}{-1.22 - 0.656y} = \frac{dx}{4.35 + 0.656x} \quad \rightarrow \quad \int \frac{dy}{-1.22 - 0.656y} = \int \frac{dx}{4.35 + 0.656x}$$

Integration yields

$$-\frac{1}{0.656}\ln(-1.22 - 0.656y) = \frac{1}{0.656}\ln(4.35 + 0.656x) - \frac{1}{0.656}\ln C_1 \tag{3}$$

where we have set the constant of integration as the natural logarithm of some constant C_1, with a constant in front in order to simplify the algebra. When we recall that ln(*ab*) = ln*a* + ln*b*, and that –ln*a* = ln(1/*a*), Eq. 3 simplifies to

Equation for streamlines: $$\boxed{y = \frac{C}{0.656(4.35 + 0.656x)} - 1.85976}$$

The new constant *C* is related to C_1, and is introduced for simplicity. *C* can be set to various values in order to plot the streamlines. Several streamlines in the upper right quadrant of the given flow field are shown in **Fig. 1**.

The direction of the flow is found by calculating *u* and *v* at some point in the flow field. We choose *x* = 3, *y* = 3. At this point *u* is positive and *v* is negative. The direction of the velocity at this point is obviously to the lower right. This sets the direction of all the streamlines. The arrows in Fig. 1 indicate the direction of flow.

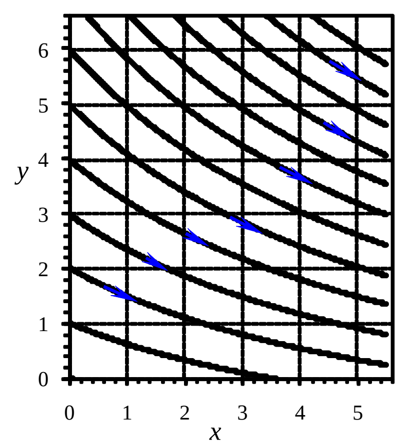


FIGURE 1
Streamlines (solid black curves) for the given velocity field.

Discussion The flow appears to be a counterclockwise turning flow in the upper right quadrant.

4-38

Solution For a given velocity field we are to generate a velocity vector plot in the first quadrant.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is given by

$$\vec{V} = (u, v) = (4.35 + 0.656x)\vec{i} + (-1.22 - 0.656y)\vec{j} \qquad (1)$$

At any point (*x*,*y*) in the flow field, the velocity components *u* and *v* are obtained from Eq. 1,

Velocity components: $u = 4.35 + 0.656x \qquad v = -1.22 - 0.656y \qquad (2)$

To plot velocity vectors, we simply pick an (*x*,*y*) point, calculate *u* and *v* from Eq. 2, and plot an arrow with its tail at (*x*,*y*), and its tip at (*x*+*Su*, *y*+*Sv*) where *S* is some scale factor for the vector plot. For the vector plot shown in **Fig. 1**, we chose *S* = 0.13, and plot velocity vectors at several locations in the first quadrant.

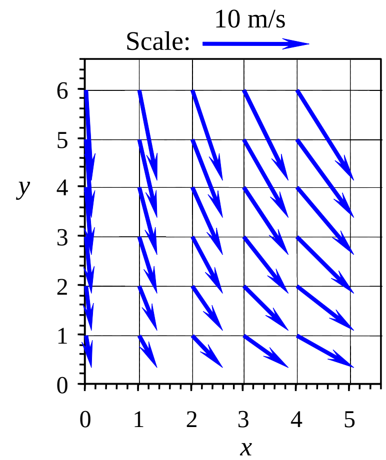


FIGURE 1
Velocity vectors for the given velocity field. The scale is shown by the top arrow.

Discussion The flow agrees with the previous problem – a counterclockwise turning flow in the upper right quadrant.

4-39

Solution For a given velocity field we are to generate an acceleration vector plot in the first quadrant.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is given by

$$\vec{V} = (u, v) = (4.35 + 0.656x)\vec{i} + (-1.22 - 0.656y)\vec{j} \qquad (1)$$

At any point (*x*,*y*) in the flow field, the velocity components *u* and *v* are obtained from Eq. 1,

Velocity components: $u = 4.35 + 0.656x \qquad v = -1.22 - 0.656y \qquad (2)$

The acceleration field is obtained from its definition (the material acceleration),

Acceleration components:

$$a_x = \frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = 0 + (4.35 + 0.656x)(0.656) + 0 + 0$$
$$a_y = \frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} = 0 + 0 + (-1.22 - 0.656y)(-0.656) + 0 \qquad (3)$$

where the unsteady terms are zero since this is a steady flow, and the terms with *w* are zero since the flow is two-dimensional. Eq. 3 simplifies to

Acceleration components:

$$a_x = 2.8536 + 0.43034x \qquad a_y = 0.80032 + 0.43034y \qquad (4)$$

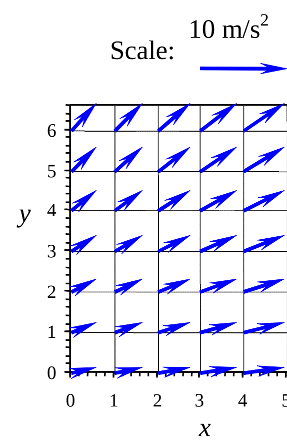


FIGURE 1
Acceleration vectors for the velocity field. The scale is shown by the top arrow.

To plot the acceleration vectors, we simply pick an (*x*,*y*) point, calculate a_x and a_y from Eq. 4, and plot an arrow with its tail at (*x*,*y*), and its tip at ($x+Sa_x$, $y+Sa_y$) where *S* is some scale factor for the vector plot. For the vector plot shown in **Fig. 1**, we chose *S* = 0.20, and plot acceleration vectors at several locations in the first quadrant.

Discussion Since the flow is a counterclockwise turning flow in the upper right quadrant, the acceleration vectors point to the upper right (centripetal acceleration).

4-40

Solution For the given velocity field, the location(s) of stagnation point(s) are to be determined. Several velocity vectors are to be sketched and the velocity field is to be described.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional, implying no z-component of velocity and no variation of u or v with z.

Analysis (a) The velocity field is

$$\vec{V} = (u, v) = (1 + 2.5x + y)\vec{i} + (-0.5 - 3x - 2.5y)\vec{j} \qquad (1)$$

Since $\vec{V}$ is a vector, all its components must equal zero in order for $\vec{V}$ itself to be zero. Setting each component of Eq. 1 to zero,

Simultaneous equations:
$$u = 1 + 2.5x + y = 0$$
$$v = -0.5 - 3x - 2.5y = 0$$

We can easily solve this set of two equations and two unknowns simultaneously. Yes, there is one stagnation point, and it is located at

Stagnation point: $x = \mathbf{-0.615\ m}$ $\quad y = \mathbf{0.538\ m}$

(b) The x and y components of velocity are calculated from Eq. 1 for several (x,y) locations in the specified range. For example, at the point (x = 2 m, y = 3 m), u = 9.00 m/s and v = -14.0 m/s. The magnitude of velocity (the *speed*) at that point is 16.64 m/s. At this and at an array of other locations, the velocity vector is constructed from its two components, the results of which are shown in Fig. 1. The flow can be described as a turning slightly counterclockwise, accelerating flow from the upper left to the lower right. The stagnation point of Part (a) does not lie in the upper right quadrant, and therefore does not appear on the sketch.

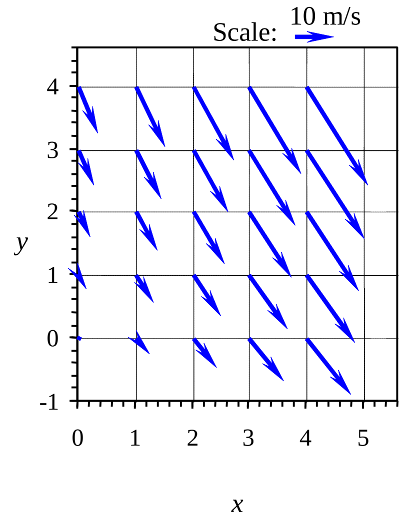


FIGURE 1
Velocity vectors in the upper right quadrant for the given velocity field.

Discussion The stagnation point location is given to three significant digits. It will be verified in Chap. 9 that this flow field is physically valid because it satisfies the differential equation for conservation of mass.

4-41

Solution For the given velocity field, the material acceleration is to be calculated at a particular point and plotted at several locations in the upper right quadrant.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional, implying no *z*-component of velocity and no variation of *u* or *v* with *z*.

Analysis (*a*) The velocity field is

$$\vec{V} = (u, v) = (1 + 2.5x + y)\vec{i} + (-0.5 - 3x - 2.5y)\vec{j} \qquad (1)$$

Using the velocity field of Eq. 1 and the equation for material acceleration in Cartesian coordinates, we write expressions for the two non-zero components of the acceleration vector:

$$a_x = \frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}$$
$$= 0 + (1 + 2.5x + y)(2.5) + (-0.5 - 3x - 2.5y)(1) + 0$$

and

$$a_y = \frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z}$$
$$= 0 + (1 + 2.5x + y)(-3) + (-0.5 - 3x - 2.5y)(-2.5) + 0$$

At (x = 2 m, y = 3 m), a_x = **8.50 m/s²** and a_y = **8.00 m/s²**.

(*b*) The above equations are applied to an array of *x* and *y* values in the upper right quadrant, and the acceleration vectors are plotted in Fig. 1.

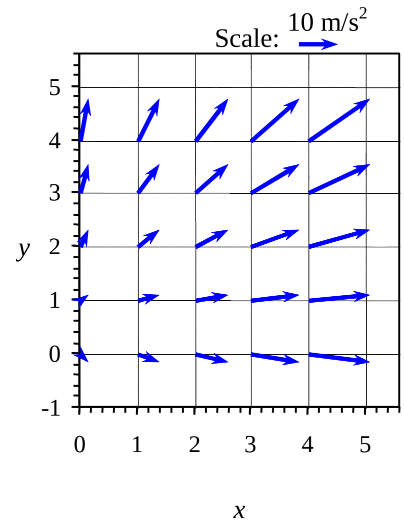


FIGURE 1
Acceleration vectors in the upper right quadrant for the given velocity field.

Discussion The acceleration vectors plotted in Fig. 1 point to the upper right, increasing in magnitude away from the origin. This agrees qualitatively with the velocity vectors of Fig. 1 of the previous problem; namely, fluid particles are accelerated to the right and are turned in the counterclockwise direction due to centripetal acceleration towards the upper right. Note that the acceleration field is non-zero, even though the flow is steady.

4-42

Solution For a given velocity field we are to plot a velocity magnitude contour plot at five given values of speed.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis Since $u_r = 0$, and since ω is positive, the speed is equal to the magnitude of the θ-component of velocity,

Speed:
$$V = \sqrt{\underbrace{\cancel{u_r^2}}_{0} + u_\theta^2} = |u_\theta| = \omega r$$

Thus, contour lines of constant speed are simply circles of constant radius given by

Contour line of constant speed:
$$r = \frac{V}{\omega}$$

For example, at V = 2.0 m/s, the corresponding contour line is a circle of radius 1.3333… m,

Contour line at constant speed V = 2.0 m/s:
$$r = \frac{2.0\text{ m/s}}{1.5\text{ 1/s}} = 1.33333...\text{ m}$$

We plot a circle at this radius and repeat this simple calculation for the four other values of V. We plot the contours in **Fig. 1**. The speed increases linearly from the center of rotation (the origin).

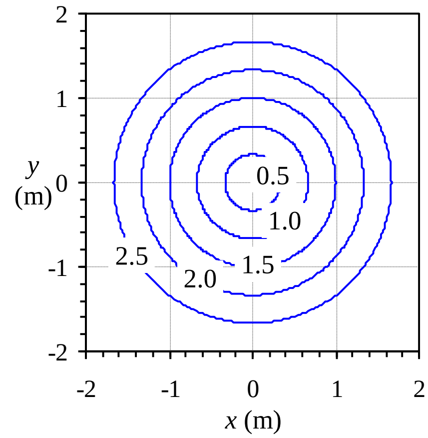


FIGURE 1
Contour plot of velocity magnitude for solid body rotation. Values of speed are labeled in units of m/s.

Discussion The contours are equidistant apart because of the linear nature of the velocity field.

4-43

Solution For a given velocity field we are to plot a velocity magnitude contour plot at five given values of speed.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis Since $u_r = 0$, and since K is positive, the speed is equal to the magnitude of the θ-component of velocity,

Speed:
$$V = \sqrt{\underbrace{\cancel{u_r^2}}_{0} + u_\theta^2} = |u_\theta| = \frac{K}{r}$$

Thus, contour lines of constant speed are simply circles of constant radius given by

Contour line of constant speed:
$$r = \frac{K}{V}$$

For example, at V = 2.0 m/s, the corresponding contour line is a circle of radius 0.75 m,

Contour line at constant speed V = 2.0 m/s:
$$r = \frac{1.5\text{ m}^2\text{/s}}{2.0\text{ m/s}} = 0.75\text{ m}$$

We plot a circle at this radius and repeat this simple calculation for the four other values of V. We plot the contours in **Fig. 1**. The speed near the center is faster than that further away from the center.

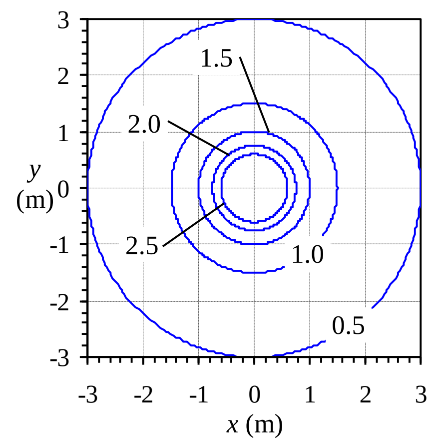


FIGURE 1
Contour plot of velocity magnitude for a line vortex. Values of speed are labeled in units of m/s.

Discussion The contours are *not* equidistant apart because of the nonlinear nature of the velocity field.

4-44

Solution For a given velocity field we are to plot a velocity magnitude contour plot at five given values of speed.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis The velocity field is

Line source:
$$u_r = \frac{m}{2\pi r} \qquad u_\theta = 0 \tag{1}$$

Since $u_\theta = 0$, and since m is positive, the speed is equal to the magnitude of the r-component of velocity,

Speed:
$$V = \sqrt{u_r^2 + \underbrace{\cancel{u_\theta^2}}_{0}} = |u_r| = \frac{m}{2\pi r} \tag{2}$$

Thus, contour lines of constant speed are simply circles of constant radius given by

Contour line of constant speed:
$$r = \frac{m}{2\pi V} = \frac{\left(\dfrac{m}{2\pi}\right)}{V} \tag{3}$$

For example, at V = 2.0 m/s, the corresponding contour line is a circle of radius 0.75 m,

Contour line at speed V = 2.0 m/s:
$$r = \frac{1.5 \text{ m}^2\text{/s}}{2.0 \text{ m/s}} = 0.75 \text{ m} \tag{4}$$

We plot a circle at this radius and repeat this simple calculation for the four other values of V. We plot the contours in **Fig. 1**. The flow slows down as it travels further from the origin.

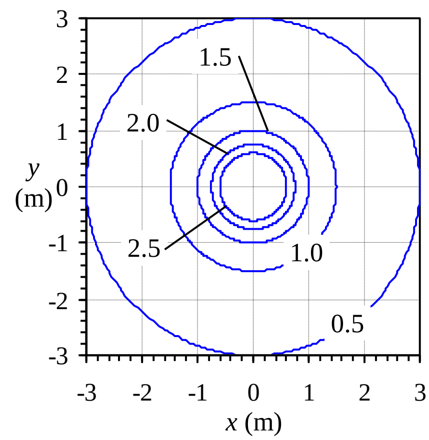


FIGURE 1
Contour plot of velocity magnitude for a line source. Values of speed are labeled in units of m/s.

Discussion The contours are *not* equidistant apart because of the nonlinear nature of the velocity field.

4-45

Solution We are to generate an expression for the tangential velocity of a liquid confined between two concentric cylinders, and we are to estimate the torque exerted by the fluid on the cylinders.

Assumptions **1** The flow is incompressible and two-dimensional, and thus the end effects (front and back of the cylinder) are negligible. **2** The flow has been running for a long time so that it is steady.

Analysis Since both cylinders are rotating at the same rate, after a long enough time, the fluid will also rotate at the same rate. The entire system will behave as solid body rotation. So, the tangential velocity will be $u_\theta = r\omega$, where $\omega = \omega_I = \omega_o$ = constant. Thus,

$$\boxed{u_\theta = \omega r}$$

and u_θ is not a function of any of the other variables.

There is no shear stress on the walls since everything is rotating like a solid body. Thus, we expect that the torque on either cylinder wall is zero.

Discussion The equation for u_θ applies to both the solid cylinders and the fluid, since everything in the system is rotating as solid body rotation.

4-46

Solution We are to discuss the type of flow that is approximated by two concentric cylinders with the inner cylinder spinning very fast while its radius goes towards zero, while the outer cylinder is large (far away) and stationary.

Assumptions **1** The flow is incompressible and two-dimensional, and thus the end effects (front and back of the cylinder) are negligible. **2** The flow has been running for a long time so that it is steady.

Analysis Since the inner cylinder is rotating but the outer cylinder is not, after a long enough time, the fluid behaves **like a line vortex**, but with a missing core region. [This is good, actually, since the tangential velocity of a line vortex at the origin is infinite!] Thus, we expect $u_\theta = \frac{\text{constant}}{r}$. Note that u_θ is not a function of any of the fluid properties. We calculate the constant by specifying u_θ at the inner cylinder surface, where $u_\theta = \omega_i R_i$ and $r = R_i$. The constant becomes $\omega_i R_i^2$, and therefore

$$\boxed{u_\theta = \frac{\omega_i R_i^2}{r}}$$

There is no shear stress on the walls since everything is rotating like a solid body. Thus, we expect that the torque on either cylinder wall is zero.

Discussion The equation for u_θ is valid in the fluid only, and we expect some error in our approximate analysis as the radius approaches the outer cylinder radius, which is not infinitely far away in a real-life situation.

4-47E

Solution For a given velocity field we are to plot several streamlines for a given range of x and y values.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis From the solution to the previous problem, an equation for the streamlines is

Streamlines in the x-y plane: $$y = \frac{C}{(U_0 + bx)} \qquad (1)$$

Constant C is set to various values in order to plot the streamlines. Several streamlines in the given range of x and y are plotted in **Fig. 1**.

The direction of the flow is found by calculating u and v at some point in the flow field. We choose $x = 1$ ft, $y = 1$ ft. At this point u is positive and v is negative. The direction of the velocity at this point is obviously to the lower right. This sets the direction of all the streamlines. The arrows in Fig. 1 indicate the direction of flow.

Discussion The flow is type of converging channel flow.

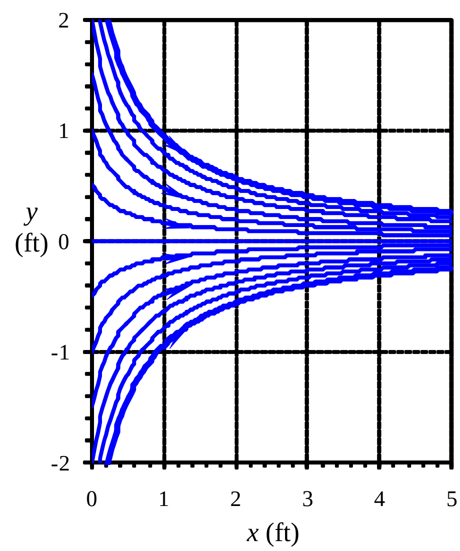


FIGURE 1
Streamlines (solid blue curves) for the given velocity field; x and y are in units of ft.

Motion and Deformation of Fluid Elements; Vorticity and Rotationality

4-48C

Solution We are to explain the relationship between vorticity and rotationality.

Analysis Vorticity is a measure of the rotationality of a fluid particle. If a particle rotates, its vorticity is non-zero. Mathematically, **the vorticity vector is twice the angular velocity vector**.

Discussion If the vorticity is zero, the flow is called irrotational.

4-49C

Solution We are to name and describe the four fundamental types of motion or deformation of fluid particles.

Analysis

1. **Translation** – a fluid particle **moves from one location to another**.
2. **Rotation** – a fluid particle **rotates about an axis** drawn through the particle.
3. **Linear strain or extensional strain** – a fluid particle **stretches in a direction such that a line segment in that direction is elongated at some later time**.
4. **Shear strain** – a fluid particle distorts in such a way that **two lines through the fluid particle that are initially perpendicular are *not* perpendicular at some later time**.

Discussion In a complex fluid flow, all four of these occur simultaneously.

4-50

Solution For a given velocity field, we are to determine whether the flow is rotational or irrotational.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is

$$\vec{V} = (u, v) = (U_0 + bx)\vec{i} - by\vec{j} \qquad (1)$$

By definition, the flow is rotational if the vorticity is non-zero. So, we calculate the vorticity. In a 2-D flow in the *x*-*y* plane, the only non-zero component of vorticity is in the *z* direction, i.e. ζ_z,

Vorticity component in the z direction:

$$\zeta_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} = 0 - 0 = 0 \qquad (1)$$

Since the vorticity is zero, this flow is **irrotational**.

Discussion We shall see in Chap. 10 that the fluid very close to the walls is rotational due to important viscous effects near the wall (a *boundary layer*). However, in the majority of the flow field, the irrotational approximation is reasonable.

4-51

Solution For a given velocity field we are to generate an equation for the x location of a fluid particle along the x-axis as a function of time.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis The velocity field is

Velocity field: $$\vec{V} = (u, v) = (U_0 + bx)\vec{i} - by\vec{j} \qquad (1)$$

We start with the definition of u following a fluid particle,

x-component of velocity of a fluid particle: $$\frac{dx_{\text{particle}}}{dt} = u = U_0 + bx_{\text{particle}} \qquad (2)$$

where we have substituted u from Eq. 1. We rearrange and separate variables, dropping the "particle" subscript for convenience,

$$\frac{dx}{U_0 + bx} = dt \qquad (3)$$

Integration yields

$$\frac{1}{b}\ln(U_0 + bx) = t - \frac{1}{b}\ln C_1 \qquad (4)$$

where we have set the constant of integration as the natural logarithm of some constant C_1, with a constant in front in order to simplify the algebra. When we recall that $\ln(ab) = \ln a + \ln b$, Eq. 4 simplifies to

$$\ln\left(C_1\left(U_0 + bx\right)\right) = t$$

from which

$$U_0 + bx = C_2 e^{bt} \qquad (5)$$

where C_2 is a new constant defined for convenience. We now plug in the known initial condition that at $t = 0$, $x = x_A$ to find constant C_2 in Eq. 5. After some algebra,

Fluid particle's x location at time t: $$\boxed{x = x_{A'} = \frac{1}{b}\left[\left(U_0 + bx_A\right)e^{bt} - U_0\right]} \qquad (6)$$

Discussion We verify that at $t = 0$, $x = x_A$ in Eq. 6.

4-52

Solution For a given velocity field we are to generate an equation for the change in length of a line segment moving with the flow along the x-axis.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis Using the results of the previous problem,

Location of particle A at time t: $$x_{A'} = \frac{1}{b}\left[\left(U_0 + bx_B\right)e^{bt} - U_0\right] \qquad (1)$$

and

Location of particle B at time t: $$x_{B'} = \frac{1}{b}\left[\left(U_0 + bx_B\right)e^{bt} - U_0\right] \qquad (2)$$

Since length $\xi = x_B - x_A$ and length $\xi + \Delta\xi = x_{B'} - x_{A'}$, we write an expression for $\Delta\xi$,

Change in length of the line segment:

$$\begin{aligned}\Delta\xi &= \left(x_{B'} - x_{A'}\right) - \left(x_B - x_A\right) \\ &= \frac{1}{b}\left[\left(U_0 + bx_B\right)e^{bt} - U_0\right] - \frac{1}{b}\left[\left(U_0 + bx_A\right)e^{bt} - U_0\right] - \left(x_B - x_A\right) \qquad (3) \\ &= x_B e^{bt} - x_A e^{bt} - x_B + x_A\end{aligned}$$

Eq. 3 simplifies to

Change in length of the line segment: $$\boxed{\Delta\xi = \left(x_B - x_A\right)\left(e^{bt} - 1\right)} \qquad (4)$$

Discussion We verify from Eq. 4 that when $t = 0$, $\Delta\xi = 0$.

4-53

Solution By examining the increase in length of a line segment along the axis of a converging duct, we are to generate an equation for linear strain rate in the x direction and compare to the exact equation given in this chapter.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis From the previous problem, we have an expression for the change in length of the line segment AB,

Change in length of the line segment:
$$\Delta\xi = \left(x_{\mathrm{B}} - x_{\mathrm{A}}\right)\left(e^{bt} - 1\right) \qquad (1)$$

The fundamental definition of linear strain rate is the rate of increase in length of a line segment per unit length of the line segment. For the case at hand,

Linear strain rate in x direction:
$$\varepsilon_{xx} = \frac{d}{dt}\frac{\left(\xi + \Delta\xi\right) - \xi}{\xi} = \frac{d}{dt}\frac{\Delta\xi}{\xi} = \frac{d}{dt}\frac{\Delta\xi}{x_{\mathrm{B}} - x_{\mathrm{A}}} \qquad (2)$$

We substitute Eq. 1 into Eq. 2 to obtain

Linear strain rate in x direction:
$$\varepsilon_{xx} = \frac{d}{dt}\frac{\left(x_{\mathrm{B}} - x_{\mathrm{A}}\right)\left(e^{bt} - 1\right)}{x_{\mathrm{B}} - x_{\mathrm{A}}} = \frac{d}{dt}\left(e^{bt} - 1\right) \qquad (3)$$

In the limit as $t \to 0$, we apply the first two terms of the series expansion for e^{bt},

Series expansion for e^{bt}:
$$e^{bt} = 1 + bt + \frac{\left(bt\right)^2}{2!} + ... \approx 1 + bt \qquad (4)$$

Finally, for small t we approximate the time derivative as $1/t$, yielding

Linear strain rate in x direction:
$$\boxed{\varepsilon_{xx} \to \frac{1}{t}\left(1 + bt - 1\right) = b} \qquad (5)$$

Comparing to the equation for ε_{xx},

Linear strain rate in x direction:
$$\boxed{\varepsilon_{xx} = \frac{\partial u}{\partial x} = b} \qquad (6)$$

Equations 5 and 6 agree, verifying our algebra.

Discussion Although we considered a line segment on the x-axis, it turns out that $\varepsilon_{xx} = b$ everywhere in the flow, as seen from Eq. 6. We could also have taken the analytical time derivative of Eq. 3, yielding $\varepsilon_{xx} = be^{bt}$. Then, as $t \to 0$, $\varepsilon_{xx} \to b$.

4-54

Solution For a given velocity field we are to generate an equation for the *y* location of a fluid particle as a function of time.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is

Velocity field:
$$\vec{V} = (u, v) = (U_0 + bx)\vec{i} - by\vec{j} \qquad (1)$$

We start with the definition of *v* following a fluid particle,

y-component of velocity of a fluid particle:
$$\frac{dy_{\text{particle}}}{dt} = v = -by_{\text{particle}} \qquad (2)$$

where we have substituted *v* from Eq. 1. We and rearrange and separate variables, dropping the "particle" subscript for convenience,

$$\frac{dy}{y} = -bdt \qquad (3)$$

Integration yields

$$\ln(y) = -bt - \ln C_1 \qquad (4)$$

where we have set the constant of integration as the natural logarithm of some constant C_1, with a constant in front in order to simplify the algebra. When we recall that ln(*ab*) = ln*a* + ln*b*, Eq. 4 simplifies to

$$\ln(C_1 y) = -t$$

from which

$$y = C_2 e^{-bt} \qquad (5)$$

where C_2 is a new constant defined for convenience. We now plug in the known initial condition that at $t = 0$, $y = y_A$ to find constant C_2 in Eq. 5. After some algebra,

Fluid particle's y location at time t:
$$\boxed{y = y_{A'} = y_A e^{-bt}} \qquad (6)$$

Discussion The fluid particle approaches the *x*-axis exponentially with time. The fluid particle also moves downstream in the *x* direction during this time period. However, in this particular problem *v* is not a function of *x*, so the streamwise movement is irrelevant (*u* and *v* act independently of each other).

4-55

Solution For a given velocity field we are to generate an equation for the change in length of a line segment in the y direction.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis Using the results of the previous problem,

Location of particle A at time t: $$y_{A'} = y_A e^{-bt} \quad (1)$$

and

Location of particle B at time t: $$y_{B'} = y_B e^{-bt} \quad (2)$$

Since length $\eta = y_B - y_A$ and length $\eta + \Delta\eta = y_{B'} - y_{A'}$, we write an expression for $\Delta\eta$,

Change in length of the line segment:

$$\Delta\eta = \left(y_{B'} - y_{A'}\right) - \left(y_B - y_A\right) = y_B e^{-bt} - y_A e^{-bt} - \left(y_B - y_A\right) = y_B e^{-bt} - y_A e^{-bt} - y_B + y_A$$

which simplifies to

Change in length of the line segment: $$\boxed{\Delta\eta = \left(y_B - y_A\right)\left(e^{-bt} - 1\right)} \quad (3)$$

Discussion We verify from Eq. 3 that when $t = 0$, $\Delta\eta = 0$.

4-56

Solution By examining the increase in length of a line segment as it moves down a converging duct, we are to generate an equation for linear strain rate in the *y* direction and compare to the exact equation given in this chapter.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis From the previous problem we have an expression for the change in length of the line segment AB,

Change in length of the line segment:
$$\Delta\eta = \left(y_{\mathrm{B}} - y_{\mathrm{A}}\right)\left(e^{-bt} - 1\right) \qquad (1)$$

The fundamental definition of linear strain rate is the rate of increase in length of a line segment per unit length of the line segment. For the case at hand,

Linear strain rate in y direction:
$$\varepsilon_{yy} = \frac{d}{dt}\frac{\left(\eta + \Delta\eta\right) - \eta}{\eta} = \frac{d}{dt}\frac{\Delta\eta}{\eta} = \frac{d}{dt}\frac{\Delta\eta}{y_{\mathrm{B}} - y_{\mathrm{A}}} \qquad (2)$$

We substitute Eq. 1 into Eq. 2 to obtain

Linear strain rate in y direction:
$$\varepsilon_{yy} = \frac{d}{dt}\frac{\left(y_{\mathrm{B}} - y_{\mathrm{A}}\right)\left(e^{-bt} - 1\right)}{y_{\mathrm{B}} - y_{\mathrm{A}}} = \frac{d}{dt}\left(e^{-bt} - 1\right) \qquad (3)$$

In the limit as $t \to 0$, we apply the first two terms of the series expansion for e^{-bt},

Series expansion for e^{-bt}:
$$e^{-bt} = 1 + \left(-bt\right) + \frac{\left(-bt\right)^2}{2!} + \ldots \approx 1 - bt \qquad (4)$$

Finally, for small *t* we approximate the time derivative as $1/t$, yielding

Linear strain rate in y direction:
$$\boxed{\varepsilon_{yy} \to \frac{1}{t}\left(1 - bt - 1\right) = -b} \qquad (5)$$

Comparing to the equation for ε,

Linear strain rate in y direction:
$$\boxed{\varepsilon_{yy} = \frac{\partial v}{\partial y} = -b} \qquad (6)$$

Equations 5 and 6 agree, verifying our algebra.

Discussion Since *v* does not depend on *x* location in this particular problem, the algebra is simple. In a more general case, both *u* and *v* depend on both *x* and *y*, and a numerical integration scheme is required. We could also have taken the analytical time derivative of Eq. 3, yielding $\varepsilon_{yy} = -be^{-bt}$. Then, as $t \to 0$, $\varepsilon_{xx} \to -b$.

4-57

Solution For a given velocity field we are to use volumetric strain rate to verify that the flow field is incompressible..

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is

Velocity field:
$$\vec{V} = (u,v) = (U_0 + bx)\vec{i} - by\vec{j} \qquad (1)$$

We use the equation for volumetric strain rate in Cartesian coordinates, and apply Eq. 1,

Volumetric strain rate:
$$\frac{1}{V}\frac{DV}{Dt} = \varepsilon_{xx} + \varepsilon_{yy} + \varepsilon_{zz} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = b + (-b) + 0 = 0 \qquad (2)$$

Where ε_{zz} = 0 since the flow is two-dimensional. Since the volumetric strain rate is zero everywhere, **the flow is incompressible**.

Discussion The fluid particle stretches in the horizontal direction and shrinks in the vertical direction, but the net volume of the fluid particle does not change.

4-58

Solution For a given steady two-dimensional velocity field, we are to calculate the *x* and *y* components of the acceleration field.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field is

$$\vec{V} = (u,v) = (U + a_1 x + b_1 y)\vec{i} + (V + a_2 x + b_2 y)\vec{j} \qquad (1)$$

The acceleration field is obtained from its definition (the material acceleration). The *x*-component is

x-component of material acceleration:
$$\boxed{a_x = \underbrace{\cancel{\frac{\partial u}{\partial t}}}_{\text{Steady}} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + \underbrace{\cancel{w\frac{\partial u}{\partial z}}}_{\text{Two-D}} = (U + a_1 x + b_1 y)a_1 + (V + a_2 x + b_2 y)b_1} \qquad (2)$$

The *y*-component is

y-component of material acceleration:
$$\boxed{a_y = \underbrace{\cancel{\frac{\partial v}{\partial t}}}_{\text{Steady}} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + \underbrace{\cancel{w\frac{\partial v}{\partial z}}}_{\text{Two-D}} = (U + a_1 x + b_1 y)a_2 + (V + a_2 x + b_2 y)b_2} \qquad (3)$$

Discussion If there were a *z*-component, it would be treated in the same fashion.

4-59

Solution We are to find a relationship among the coefficients that causes the flow field to be incompressible.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis We use the equation for volumetric strain rate in Cartesian coordinates, and apply Eq. 1 of the previous problem,

Volumetric strain rate:
$$\frac{1}{V}\frac{DV}{Dt} = \varepsilon_{xx} + \varepsilon_{yy} + \varepsilon_{zz} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{Two-D}} = a_1 + b_2 \qquad (1)$$

We recognize that when the volumetric strain rate is zero everywhere, the flow is incompressible. Thus, the desired relationship is

Relationship to ensure incompressibility:
$$\boxed{a_1 + b_2 = 0} \qquad (2)$$

Discussion If Eq. 2 is satisfied, the flow is incompressible, regardless of the values of the other coefficients.

4-60

Solution For a given velocity field we are to calculate the linear strain rates in the *x* and *y* directions.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis We use the equations for linear strain rates in Cartesian coordinates, and apply Eq. 1 of Problem 4-58,

Linear strain rates:
$$\boxed{\varepsilon_{xx} = \frac{\partial u}{\partial x} = a_1 \qquad \varepsilon_{yy} = \frac{\partial v}{\partial y} = b_2} \qquad (1)$$

Discussion In general, since coefficients a_1 and b_2 are non-zero, fluid particles stretch (or shrink) in the *x* and *y* directions.

4-61

Solution For a given velocity field we are to calculate the shear strain rate in the *x*-*y* plane.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis We use the equation for shear strain rate ε_{xy} in Cartesian coordinates, and apply Eq. 1 of Problem 4-58,

Shear strain rate in x-y plane:
$$\boxed{\varepsilon_{xy} = \varepsilon_{yx} = \frac{1}{2}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) = \frac{1}{2}\left(b_1 + a_2\right)} \qquad (1)$$

Note that by symmetry $\varepsilon_{yx} = \varepsilon_{xy}$.

Discussion In general, since coefficients b_1 and a_2 are non-zero, fluid particles distort via shear strain in the *x* and *y* directions.

4-62

Solution For a given velocity field we are to form the 2-D strain rate tensor and determine the conditions necessary for the *x* and *y* axes to be principal axes.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The two-dimensional form of the strain rate tensor is

2-D strain rate tensor:
$$\varepsilon_{ij} = \begin{pmatrix} \varepsilon_{xx} & \varepsilon_{xy} \\ \varepsilon_{yx} & \varepsilon_{yy} \end{pmatrix} \qquad (1)$$

We use the linear strain rates and the shear strain rate from the previous two problems to generate the tensor,

2-D strain rate tensor:
$$\boxed{\varepsilon_{ij} = \begin{pmatrix} \varepsilon_{xx} & \varepsilon_{xy} \\ \varepsilon_{yx} & \varepsilon_{yy} \end{pmatrix} = \begin{pmatrix} a_1 & \frac{1}{2}\left(b_1 + a_2\right) \\ \frac{1}{2}\left(b_1 + a_2\right) & b_2 \end{pmatrix}} \qquad (2)$$

If the *x* and *y* axes were principal axes, the diagonals of ε_{ij} would be non-zero, and the off-diagonals would be zero. Here the off-diagonals go to zero when

Condition for x and y axes to be principal axes:
$$\boxed{b_1 + a_2 = 0} \qquad (3)$$

Discussion For the more general case in which Eq. 3 is not satisfied, the principal axes can be calculated using tensor algebra.

4-63

Solution For a given velocity field we are to calculate the vorticity vector and discuss its orientation.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis We use the equation for vorticity vector $\vec{\zeta}$ in Cartesian coordinates, and apply Eq. 1 of Problem 4-52,

Vorticity vector:
$$\boxed{\vec{\zeta} = \left(\underbrace{\cancel{\frac{\partial w}{\partial y}}}_{\text{Two-D}} - \underbrace{\cancel{\frac{\partial v}{\partial z}}}_{\text{Two-D}}\right)\vec{i} + \left(\underbrace{\cancel{\frac{\partial u}{\partial z}}}_{\text{Two-D}} - \underbrace{\cancel{\frac{\partial w}{\partial x}}}_{\text{Two-D}}\right)\vec{j} + \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right)\vec{k} = \left(a_2 - b_1\right)\vec{k}} \qquad (1)$$

The only non-zero component of vorticity is in the **z (or –z) direction**.

Discussion For any two-dimensional flow in the *x*-*y* plane, the vorticity vector *must* point in the *z* (or –*z*) direction. The sign of the *z*-component of vorticity in Eq. 1 obviously depends on the sign of $a_2 - b_1$.

4-64

Solution For the given velocity field we are to calculate the two-dimensional linear strain rates from fundamental principles and compare with the given equation.

Assumptions **1** The flow is incompressible. **2** The flow is steady. **3** The flow is two-dimensional.

Analysis First, for convenience, we number the equations in the problem statement:

Velocity field: $$\vec{V}=(u,v)=(a+by)\vec{i}+0\vec{j} \qquad (1)$$

Lower left corner at t + dt: $$\left(x+(a+by)dt,y\right) \qquad (2)$$

Linear strain rate in Cartesian coordinates: $$\varepsilon_{xx}=\frac{\partial u}{\partial x} \qquad \varepsilon_{yy}=\frac{\partial v}{\partial y} \qquad (3)$$

(*a*) The lower right corner of the fluid particle moves the same amount as the lower left corner since *u* does not depend on *y* position. Thus,

Lower right corner at t + dt: $$\left(x+dx+(a+by)dt,y\right) \qquad (4)$$

Similarly, the top two corners of the fluid particle move to the right at speed $a + b(y+dy)dt$. Thus,

Upper left corner at t + dt: $$\left(x+\left(a+b\left(y+dy\right)\right)dt,y+dy\right) \qquad (5)$$

and

Upper right corner at t + dt: $$\left(x+dx+\left(a+b\left(y+dy\right)\right)dt,y+dy\right) \qquad (6)$$

(*b*) From the fundamental definition of linear strain rate in the *x*-direction, we consider the lower edge of the fluid particle. Its rate of increase in length divided by its original length is found by using Eqs. 2 and 4,

ε_{xx}: $$\varepsilon_{xx}=\frac{1}{dt}\left[\frac{\overbrace{x+dx+(a+by)dt-\left(x+(a+by)dt\right)}^{\text{Length of lower edge at } t+dt}-\overbrace{dx}^{\text{Length of lower edge at } t}}{dx}\right]=0 \qquad (6)$$

We get the same result by considering the *upper* edge of the fluid particle. Similarly, using the left edge of the fluid particle and Eqs. 2 and 5 we get

ε_{yy}: $$\varepsilon_{yy}=\frac{1}{dt}\left[\frac{\overbrace{y+dy-y}^{\text{Length of left edge at } t+dt}-\overbrace{dy}^{\text{Length of left edge at } t}}{dy}\right]=0 \qquad (7)$$

We get the same result by considering the *right* edge of the fluid particle. Thus both the *x*- and *y*-components of linear strain rate are zero for this flow field.

(*c*) From Eq. 3 we calculate

Linear strain rates: $$\boxed{\varepsilon_{xx}=\frac{\partial u}{\partial x}=0 \qquad \varepsilon_{yy}=\frac{\partial v}{\partial y}=0} \qquad (8)$$

Discussion Although the algebra in this problem is rather straight-forward, it is good practice for the more general case (a later problem).

4-65

Solution We are to verify that the given flow field is incompressible using two different methods.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional.

Analysis

(*a*) The volume of the fluid particle at time *t* is.

Volume at time t: $$V(t) = dxdydz \quad (1)$$

where *dz* is the length of the fluid particle in the *z* direction. At time *t* + *dt*, we assume that the fluid particle's dimension *dz* remains fixed since the flow is two-dimensional. Thus its volume is *dz* times the area of the rhombus shown in Fig. P4-58, as illustrated in Fig. 1,

Volume at time t + dt: $$V(t+dt) = dxdydz \quad (2)$$

Since Eqs. 1 and 2 are equal, the volume of the fluid particle has not changed, and **the flow is therefore incompressible**.

(*b*) We use the equation for volumetric strain rate in Cartesian coordinates, and apply the results of the previous problem,

Volumetric strain rate: $$\frac{1}{V}\frac{DV}{Dt} = \varepsilon_{xx} + \varepsilon_{yy} + \varepsilon_{zz} = 0+0+0=0 \quad (3)$$

Where ε_{zz} = 0 since the flow is two-dimensional. Since the volumetric strain rate is zero everywhere, **the flow is incompressible**.

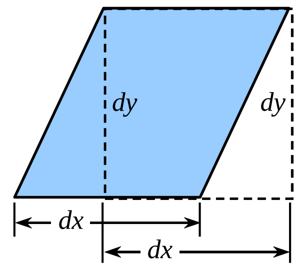


FIGURE 1
The area of a rhombus is equal to its base times its height, which here is *dxdy*.

Discussion Although the fluid particle deforms with time, its height, its depth, and the length of its horizontal edges remain constant.

4-66

Solution For the given velocity field we are to calculate the two-dimensional shear strain rate in the *x*-*y* plane from fundamental principles and compare with the given equation.

Assumptions **1** The flow is incompressible. **2** The flow is steady. **3** The flow is two-dimensional.

Analysis

(*a*) The shear strain rate is

Shear strain rate in Cartesian coordinates: $$\varepsilon_{xy} = \frac{1}{2}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) \qquad (1)$$

From the fundamental definition of shear strain rate in the *x*-*y* plane, we consider the bottom edge and the left edge of the fluid particle, which intersect at 90° at the lower left corner at time *t*. We define angle α between the lower edge and the left edge of the fluid particle, and angle β, the complement of α (Fig. 1). The rate of decrease of angle α over time interval *dt* is obtained from application of trigonometry. First, we calculate angle β,

Angle β at time t + dt: $$\beta = \arctan\left(\frac{bdydt}{dy}\right) = \arctan\left(bdt\right) \approx bdt \qquad (2)$$

The approximation is valid for very small angles. As the time interval $dt \to 0$, Eq. 2 is correct. At time $t + dt$, angle α is

Angle α at time t + dt: $$\alpha = \frac{\pi}{2} - \beta \approx \frac{\pi}{2} - bdt \qquad (3)$$

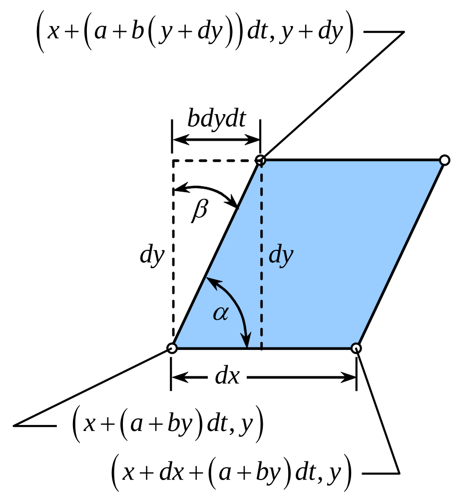


FIGURE 1
A magnified view of the deformed fluid particle at time $t + dt$, with the location of three corners indicated, and angles α and β defined.

During this time interval, α changes from 90° (π/2 radians) to the expression given by Eq. 2. Thus the rate of change of α is

Rate of change of angle α: $$\frac{d\alpha}{dt} = \frac{1}{dt}\left[\underbrace{\left(\frac{\pi}{2} - bdt\right)}_{\alpha \text{ at } t+dt} - \underbrace{\frac{\pi}{2}}_{\alpha \text{ at } t}\right] = -b \qquad (4)$$

Finally, since shear strain rate is defined as *half* of the rate of *decrease* of angle α,

Shear strain rate: $$\varepsilon_{xy} = -\frac{1}{2}\frac{d\alpha}{dt} = \boxed{\frac{b}{2}} \qquad (5)$$

(*b*) From Eq. 1 we calculate

Shear strain rate: $$\varepsilon_{xy} = \frac{1}{2}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) = \frac{1}{2}\left(b+0\right) = \boxed{\frac{b}{2}} \qquad (6)$$

Both methods for obtaining the shear strain rate agree (Eq. 5 and Eq. 6).

Discussion Although the algebra in this problem is rather straight-forward, it is good practice for the more general case (a later problem).

4-67

Solution For the given velocity field we are to calculate the two-dimensional rate of rotation in the *x*-*y* plane from fundamental principles and compare with the given equation.

Assumptions **1** The flow is incompressible. **2** The flow is steady. **3** The flow is two-dimensional.

Analysis

(*a*) The rate of rotation in Cartesian coordinates is

Rate of rotation in Cartesian coordinates: $$\omega_z = \frac{1}{2}\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) \qquad (1)$$

From the fundamental definition of rate of rotation in the *x*-*y* plane, we consider the bottom edge and the left edge of the fluid particle, which intersect at 90° at the lower left corner at time *t*. We define angle β in Fig. 1, where β is the negative of the angle of rotation of the left edge of the fluid particle (negative because rotation is mathematically positive in the *counterclockwise* direction). We calculate angle β using trigonometry,

Angle β at time t + dt: $$\beta = \arctan\left(\frac{bdydt}{dy}\right) = \arctan(bdt) \approx bdt \qquad (2)$$

The approximation is valid for very small angles. As the time interval $dt \to 0$, Eq. 2 is correct. Meanwhile, the bottom edge of the fluid particle has not rotated at all. Thus, the average angle of rotation of the two line segments (lower and left edges) at time *t* + *dt* is

$$AVG = \frac{1}{2}(0-\beta) \approx -\frac{b}{2}dt \qquad (3)$$

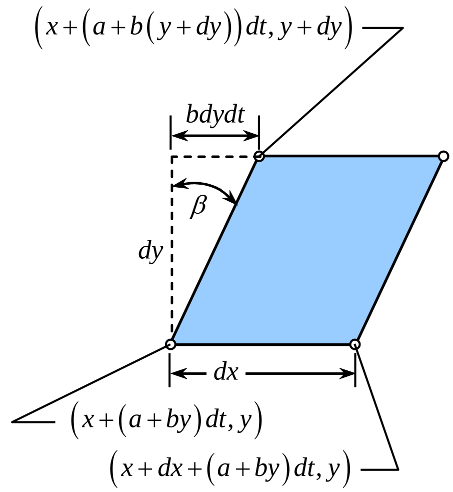


FIGURE 1
A magnified view of the deformed fluid particle at time *t* + *dt*, with the location of three corners indicated, and angle β defined.

Thus the average rotation *rate* during time interval *dt* is

Rate of rotation in x-y plane: $$\omega_z = \frac{d(AVG)}{dt} = \frac{1}{dt}\left(-\frac{b}{2}dt\right) = \boxed{-\frac{b}{2}} \qquad (4)$$

(*b*) From Eq. 1 we calculate

Rate of rotation: $$\omega_z = \frac{1}{2}\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) = \frac{1}{2}(0-b) = \boxed{-\frac{b}{2}} \qquad (5)$$

Both methods for obtaining the rate of rotation agree (Eq. 4 and Eq. 5).

Discussion The rotation rate is negative, indicating *clockwise* rotation about the *z*-axis. This agrees with our intuition as we follow the fluid particle.

4-68

Solution We are to determine whether the shear flow of Problem 4-22 is rotational or irrotational, and we are to calculate the vorticity in the *z* direction.

Analysis

(*a*) Since the rate of rotation is non-zero, it means that **the flow is *rotational***.

(*b*) Vorticity is defined as twice the rate of rotation, or twice the angular velocity. In the *z* direction,

Vorticity component: $$\boxed{\zeta_z = 2\omega_z = 2\left(-\frac{b}{2}\right) = -b} \qquad (1)$$

Discussion Vorticity is negative, indicating *clockwise* rotation about the *z*-axis.

4-69

Solution We are to prove the given expression for flow in the *xy*-plane.

Assumptions **1** The flow is incompressible and two-dimensional.

Analysis For flow in the *xy*-plane, we are to show that:

Rate of rotation:
$$\omega = \omega_z = \frac{1}{2}\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) \qquad (1)$$

By definition, the rate of rotation (angular velocity) at a point is the average rotation rate of two initially perpendicular lines that intersect at the point. In this particular problem, Line a (PA) and Line b (PB) are initially perpendicular, and intersect at point P. Line a rotates by angle α_a, and Line b rotates by angle α_b, Thus, the average angle of rotation is

Average angle of rotation:
$$\frac{\alpha_a + \alpha_b}{2} \qquad (2)$$

During time increment *dt*, point P moves a distance *udt* to the right and *vdt* up (to first order, assuming *dt* is very small). Similarly, point A moves a distance $\left(u + \frac{\partial u}{\partial x}dx\right)dt$ to the right and $\left(v + \frac{\partial v}{\partial x}dx\right)dt$ up, and point B moves a distance $\left(u + \frac{\partial u}{\partial y}dy\right)dt$ to the right and $\left(v + \frac{\partial v}{\partial y}dy\right)dt$ up. Since point A is initially at distance *dx* to the right of point P, the horizontal distance from point P′ to point A′ at the later time t_2 is

$$dx + \frac{\partial u}{\partial x}dxdt$$

On the other hand, point A is at the same *vertical* level as point P at time t_1. Thus, the vertical distance from point P′ to point A′ at time t_2 is

$$\frac{\partial v}{\partial x}dxdt \qquad (3)$$

Similarly, point B is located at distance *dy* vertically above point P at time t_1, and thus the horizontal distance from point P′ to point B′ at time t_2 is

$$-\frac{\partial u}{\partial y}dydt \qquad (4)$$

and

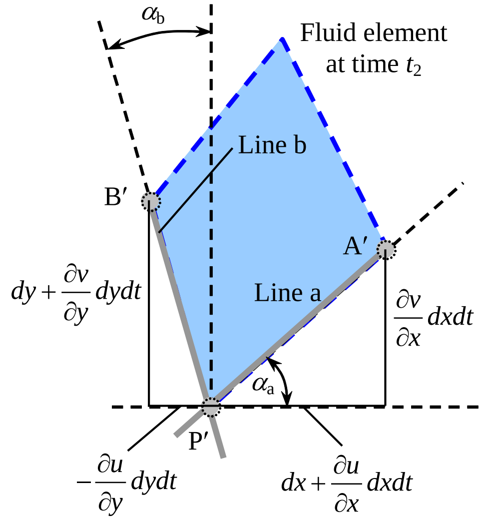


FIGURE 1
A close-up view of the distorted fluid element at time t_2.

Vertical distance from point P′ to point B′ at time t_2:
$$dy + \frac{\partial v}{\partial y}dydt \qquad (5)$$

We mark the horizontal and vertical distances between point A′ and point P′ and between point B′ and point P′ at time t_2 in Fig. 1. From the figure we see that

Angle α_a in terms of velocity components:
$$\alpha_a = \tan^{-1}\left(\frac{\frac{\partial v}{\partial x}dxdt}{dx + \frac{\partial u}{\partial x}dxdt}\right) \approx \tan^{-1}\left(\frac{\frac{\partial v}{\partial x}dxdt}{dx}\right) = \tan^{-1}\left(\frac{\partial v}{\partial x}dt\right) \approx \frac{\partial v}{\partial x}dt \qquad (6)$$

The first approximation in Eq. 6 is due to the fact that as the size of the fluid element shrinks to a point, $dx \to 0$, and at the same time $dt \to 0$. Thus, the second term in the denominator is second-order compared to the first-order term *dx* and can be neglected. The second approximation in Eq. 6 is because as $dt \to 0$ angle α_a is very small, and $\tan\alpha_a \to \alpha_a$. Similarly, *angle* α_b is written in terms of velocity components as

$$\alpha_{\mathrm{b}} = \tan^{-1}\left(\frac{-\dfrac{\partial u}{\partial y}dydt}{dy+\dfrac{\partial v}{\partial y}dydt}\right) \approx \tan^{-1}\left(\frac{-\dfrac{\partial u}{\partial y}dydt}{dy}\right) = \tan^{-1}\left(-\frac{\partial u}{\partial y}dt\right) \approx -\frac{\partial u}{\partial y}dt \qquad (7)$$

Finally then, the average rotation angle (Eq. 2) becomes

Average angle of rotation:
$$\frac{\alpha_{\mathrm{a}}+\alpha_{\mathrm{b}}}{2} = \frac{1}{2}\left(\frac{\partial v}{\partial x}dt - \frac{\partial u}{\partial y}dt\right) = \frac{dt}{2}\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) \qquad (8)$$

and the average *rate* of rotation (angular velocity) of the fluid element about point P in the *x*-*y* plane becomes

$$\boxed{\omega = \omega_z = \frac{d}{dt}\left(\frac{\alpha_{\mathrm{a}}+\alpha_{\mathrm{b}}}{2}\right) = \frac{1}{2}\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right)} \qquad (9)$$

Discussion Eq. 9 can be extended to three dimensions by performing a similar analysis in the *x*-*z* and *y*-*z* planes.

4-70

Solution We are to prove the given expression.

Assumptions **1** The flow is incompressible and two-dimensional.

Analysis We are to prove the following:

Linear strain rate in x-direction: $\varepsilon_{xx} = \dfrac{\partial u}{\partial x}$ (1)

By definition, the rate of linear strain is the rate of increase in length of a line segment in a given direction divided by the original length of the line segment in that direction. During time increment *dt*, point P moves a distance *udt* to the right and *vdt* up (to first order, assuming *dt* is very small). Similarly, point A moves a distance $\left(u + \dfrac{\partial u}{\partial x}dx\right)dt$ to the right and $\left(v + \dfrac{\partial v}{\partial x}dx\right)dt$ up. Since point A is initially at distance *dx* to the right of point P, its position to the right of point P′ at the later time t_2 is

$$dx + \frac{\partial u}{\partial x}dxdt \qquad (2)$$

On the other hand, point A is at the same *vertical* level as point P at time t_1. Thus, the vertical distance from point P′ to point A′ at time t_2 is

Vertical distance from point P′ to point A′ at time t_2: $\dfrac{\partial v}{\partial x}dxdt$ (3)

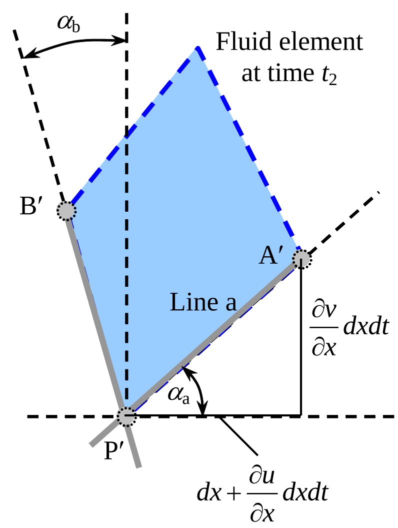


FIGURE 1
A close-up view of the distorted fluid element at time t_2.

We mark the horizontal and vertical distances between point A′ and point P′ at time t_2 in Fig. 1. From the figure we see that

Linear strain rate in the x direction as line PA changes to P′A′:

$$\varepsilon_{xx} = \frac{d}{dt}\left(\frac{\overbrace{dx + \frac{\partial u}{\partial x}dxdt}^{\text{Length of P}'\text{A}'\text{ in } x \text{ direction}} - \overbrace{dx}^{\text{Length of PA in } x \text{ direction}}}{\underbrace{dx}_{\text{Length of PA in } x \text{ direction}}}\right) = \frac{d}{dt}\left(\frac{\partial u}{\partial x}dt\right) = \frac{\partial u}{\partial x} \qquad (4)$$

Thus **Eq. 1 is verified**.

Discussion The distortion of the fluid element is exaggerated in Fig. 1. As time increment *dt* and fluid element length *dx* approach zero, the first-order approximations become exact.

4-71

Solution We are to prove the given expression.

Assumptions **1** The flow is incompressible and two-dimensional.

Analysis We are to prove the following:

Shear strain rate in xy-plane: $$\varepsilon_{xy} = \frac{1}{2}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) \qquad (1)$$

By definition, the shear strain rate at a point is half of the rate of decrease of the angle between two initially perpendicular lines that intersect at the point. In Fig. P4-63, Line a (PA) and Line b (PB) are initially perpendicular, and intersect at point P. Line a rotates by angle α_a, and Line b rotates by angle α_b. The angle between these two lines changes from $\pi/2$ at time t_1 to $\alpha_{a\text{-}b}$ at time t_2 as sketched in Fig. 1. The shear strain rate at point P for initially perpendicular lines in the *x* and *y* directions is thus

$$\varepsilon_{xy} = -\frac{1}{2}\frac{d}{dt}\alpha_{a\text{-}b} \qquad (2)$$

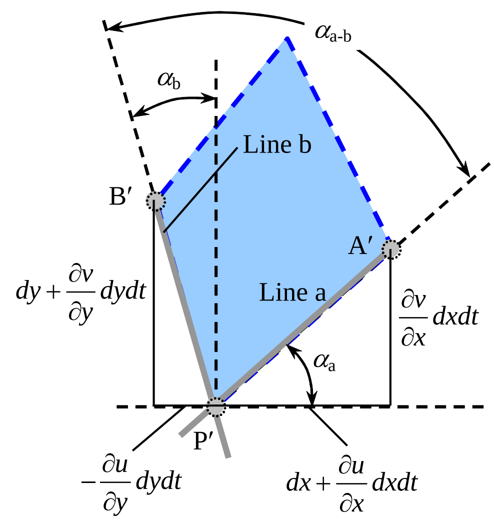


FIGURE 1
A close-up view of the distorted fluid element at time t_2.

During time increment *dt*, point P moves a distance *udt* to the right and *vdt* up (to first order, assuming *dt* is very small). Similarly, point A moves a distance $\left(u + \frac{\partial u}{\partial x}dx\right)dt$ to the right and $\left(v + \frac{\partial v}{\partial x}dx\right)dt$ up, and point B moves a distance $\left(u + \frac{\partial u}{\partial y}dy\right)dt$ to the right and $\left(v + \frac{\partial v}{\partial y}dy\right)dt$ up. Since point A is initially at distance *dx* to the right of point P, its position to the right of point P′ at the later time t_2 is

Horizontal distance from point P′ to point A′ at time t_2: $$dx + \frac{\partial u}{\partial x}dxdt \qquad (3)$$

On the other hand, point A is at the same *vertical* level as point P at time t_1. Thus, the vertical distance from point P′ to point A′ at time t_2 is

Vertical distance from point P′ to point A′ at time t_2: $$\frac{\partial v}{\partial x}dxdt \qquad (3)$$

Similarly, point B is located at distance *dy* vertically above point P at time t_1, and thus we write

Horizontal distance from point P′ to point B′ at time t_2: $$-\frac{\partial u}{\partial y}dydt \qquad (4)$$

and

Vertical distance from point P′ to point B′ at time t_2: $$dy + \frac{\partial v}{\partial y}dydt \qquad (5)$$

We mark the horizontal and vertical distances between point A′ and point P′ and between point B′ and point P′ at time t_2 in Fig. 1. From the figure we see that

Angle α_a in terms of velocity components:

$$\alpha_a = \tan^{-1}\left(\frac{\frac{\partial v}{\partial x}dxdt}{dx + \frac{\partial u}{\partial x}dxdt}\right) \approx \tan^{-1}\left(\frac{\frac{\partial v}{\partial x}dxdt}{dx}\right) = \tan^{-1}\left(\frac{\partial v}{\partial x}dt\right) \approx \frac{\partial v}{\partial x}dt \qquad (6)$$

The first approximation in Eq. 6 is due to the fact that as the size of the fluid element shrinks to a point, $dx \to 0$, and at the same time $dt \to 0$. Thus, the second term in the denominator is second-order compared to the first-order term dx and can be neglected. The second approximation in Eq. 6 is because as $dt \to 0$ angle α_{a} is very small, and $\tan\alpha_{\mathrm{a}} \to \alpha_{\mathrm{a}}$. Similarly,

Angle α_{b} in terms of velocity components:

$$\alpha_{\mathrm{b}} = \tan^{-1}\left(\frac{-\frac{\partial u}{\partial y}dydt}{dy + \frac{\partial v}{\partial y}dydt}\right) \approx \tan^{-1}\left(\frac{-\frac{\partial u}{\partial y}dydt}{dy}\right) = \tan^{-1}\left(-\frac{\partial u}{\partial y}dt\right) \approx -\frac{\partial u}{\partial y}dt \qquad (7)$$

Angle $\alpha_{\mathrm{a\text{-}b}}$ at time t_2 is calculated from Fig. 1 as

Angle $\alpha_{\mathrm{a\text{-}b}}$ at time t_2 in terms of velocity components:

$$\alpha_{\mathrm{a\text{-}b}} = \frac{\pi}{2} + \alpha_{\mathrm{b}} - \alpha_{\mathrm{a}} = \frac{\pi}{2} - \frac{\partial u}{\partial y}dt - \frac{\partial v}{\partial x}dt \qquad (8)$$

where we have used Eqs. 6 and 7. Finally then, the shear strain rate (Eq. 2) becomes

Shear strain rate, initially perpendicular lines in the x and y directions:

$$\varepsilon_{xy} = -\frac{1}{2}\frac{d}{dt}\alpha_{\mathrm{a\text{-}b}} \approx -\frac{1}{2}\frac{1}{dt}\left(\overbrace{\frac{\pi}{2} - \frac{\partial u}{\partial y}dt - \frac{\partial v}{\partial x}dt}^{\alpha_{\mathrm{a\text{-}b}}\text{ at } t_2} - \overbrace{\frac{\pi}{2}}^{\alpha_{\mathrm{a\text{-}b}}\text{ at } t_1}\right) = \frac{1}{2}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) \qquad (9)$$

which agrees with Eq. 1. Thus, **Eq. 1 is proven**.

Discussion Eq. 9 can be easily extended to three dimensions by performing a similar analysis in the *x-z* plane and in the *y-z* plane.

4-72

Solution For a given linear strain rate in the *x*-direction, we are to calculate the linear strain rate in the *y*-direction.

Analysis Since the flow is incompressible, the volumetric strain rate must be zero. In two dimensions,

Volumetric strain rate in the x-y plane: $$\frac{1}{V}\frac{DV}{Dt} = \varepsilon_{xx} + \varepsilon_{yy} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad (1)$$

Thus, the linear strain rate in the *y*-direction is the negative of that in the *x*-direction,

Linear strain rate in y-direction: $$\varepsilon_{yy} = \frac{\partial v}{\partial y} = -\frac{\partial u}{\partial x} = \mathbf{-2.5\ 1/s} \qquad (2)$$

Discussion The fluid element *stretches* in the *x*-direction since ε_{xx} is positive. Because the flow is incompressible, the fluid element must *shrink* in the *y*-direction, yielding a value of ε_{yy} that is negative.

4-73

Solution We are to calculate the vorticity of fluid particles in a tank rotating in solid body rotation about its vertical axis.

Assumptions **1** The flow is steady. **2** The *z*-axis is in the vertical direction.

Analysis Vorticity $\vec{\zeta}$ is twice the angular velocity $\vec{\omega}$. Here,

Angular velocity: $$\vec{\omega} = 175\frac{rot}{\text{min}}\left(\frac{1\ \text{min}}{60\ \text{s}}\right)\left(\frac{2\pi\ \text{rad}}{\text{rot}}\right)\vec{k} = 18.326\vec{k}\ \text{rad/s} \qquad (1)$$

where $\vec{k}$ is the unit vector in the vertical (*z*) direction. The vorticity is thus

Vorticity: $$\vec{\zeta} = 2\vec{\omega} = 2\times 18.326\vec{k}\ \text{rad/s} = 36.652\vec{k}\ \text{rad/s} \cong \mathbf{36.7\vec{k}\ rad/s} \qquad (2)$$

Discussion Because the water rotates as a solid body, the vorticity is constant throughout the tank, and points vertically upward.

4-74

Solution We are to calculate the angular speed of a tank rotating about its vertical axis.

Assumptions **1** The flow is steady. **2** The *z*-axis is in the vertical direction.

Analysis Vorticity $\vec{\zeta}$ is twice the angular velocity $\vec{\omega}$. Thus,

Angular velocity: $$\vec{\omega} = \frac{\vec{\zeta}}{2} = \frac{-45.4\vec{k}\ \text{rad/s}}{2} = -22.7\vec{k}\ \text{rad/s} \qquad (1)$$

where $\vec{k}$ is the unit vector in the vertical (*z*) direction. The angular velocity is negative, which by definition is in the clockwise direction about the vertical axis. We express the rate of rotation in units of rpm,

Rate of rotation: $$\dot{n} = -22.7\frac{\text{rad}}{\text{s}}\left(\frac{60\ \text{s}}{1\ \text{min}}\right)\left(\frac{\text{rot}}{2\pi\ \text{rad}}\right) = -216.769\frac{\text{rot}}{\text{min}} \cong \mathbf{-217\ rpm} \qquad (2)$$

Discussion Because the vorticity is constant throughout the tank, the water rotates as a solid body.

4-75

Solution For a tank of given rim radius and speed, we are to calculate the magnitude of the component of vorticity in the vertical direction.

Assumptions **1** The flow is steady. **2** The z-axis is in the vertical direction.

Analysis The linear speed at the rim is equal to $r_{\text{rim}}\omega_z$. Thus,

Component of angular velocity in z-direction:
$$\omega_z = \frac{V_{\text{rim}}}{r_{\text{rim}}} = \frac{3.61 \text{ m/s}}{0.354 \text{ m}} = 10.19774 \text{ rad/s} \tag{1}$$

Vorticity $\vec{\zeta}$ is twice the angular velocity $\vec{\omega}$. Thus,

z-component of vorticity:
$$\zeta_z = 2\omega_z = 2\left(10.19774 \text{ rad/s}\right) = 20.39548 \text{ rad/s} \cong \mathbf{20.4 \text{ rad/s}} \tag{2}$$

Discussion Radian is a non-dimensional unit, so we can insert it into Eq. 1. The final answer is given to three significant digits for consistency with the given information.

4-76

Solution For a given deformation of a fluid particle in one direction, we are to calculate its deformation in the other direction.

Assumptions **1** The flow is incompressible. **2** The flow is two-dimensional in the x-y plane.

Analysis Since the flow is incompressible and two-dimensional, the area of the fluid element must remain constant (volumetric strain rate must be zero in an incompressible flow). The area of the original fluid particle is a^2. Hence, **the vertical dimension of the fluid particle at the later time must be $a^2/2a = a/2$**.

Discussion Since the particle stretches by a factor of two in the x-direction, it shrinks by a factor of two in the y-direction.

4-77

Solution We are to calculate the percentage change in fluid density for a fluid particle undergoing two-dimensional deformation.

Assumptions **1** The flow is two-dimensional in the x-y plane.

Analysis The area of the original fluid particle is a^2. Assuming that the mass of the fluid particle is m and its dimension in the z-direction is also a, the initial density is $\rho = m/V = m/a^3$. As the particle moves and deforms, its mass must remain constant. If its dimension in the z-direction remains equal to a, the density at the later time is

Density at the later time:
$$\rho = \frac{m}{V} = \frac{m}{(1.08a)(0.903a)(a)} = 1.025\frac{m}{a^3} \tag{1}$$

Compared to the original density, **the density has increased by about 2.5%**.

Discussion The fluid particle has stretched in the x-direction and shrunk in the y-direction, but there is nevertheless a net decrease in volume, corresponding to a net increase in density.

4-78

Solution For a given velocity field we are to calculate the vorticity.

Analysis The velocity field is

$$\vec{V}=(u,v,w)=(3.0+2.0x-y)\vec{i}+(2.0x-2.0y)\vec{j}+(0.5xy)\vec{k} \tag{1}$$

In Cartesian coordinates, the vorticity vector is

Vorticity vector in Cartesian coordinates:

$$\vec{\zeta}=\left(\frac{\partial w}{\partial y}-\frac{\partial v}{\partial z}\right)\vec{i}+\left(\frac{\partial u}{\partial z}-\frac{\partial w}{\partial x}\right)\vec{j}+\left(\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}\right)\vec{k} \tag{2}$$

We substitute the velocity components u = 3.0 + 2.0x – y, v = 2.0x – 2.0y, and w = 0.5xy from Eq. 1 into Eq. 2 to obtain

Vorticity vector:

$$\boxed{\vec{\zeta}=(0.5x-0)\vec{i}+(0-0.5y)\vec{j}+(2.0-(-1))\vec{k}=(0.5x)\vec{i}-(0.5y)\vec{j}+(3.0)\vec{k}} \tag{3}$$

Discussion The vorticity is non-zero implying that this flow field is *rotational*.

4-79

Solution We are to determine if the flow is rotational, and if so calculate the z-component of vorticity.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the x-y plane.

Analysis The velocity field is given by

Velocity field, Couette flow:

$$\vec{V}=(u,v)=\left(V\frac{y}{h}\right)\vec{i}+0\vec{j} \tag{1}$$

If the vorticity is non-zero, the flow is rotational. So, we calculate the z-component of vorticity,

z-component of vorticity:

$$\boxed{\zeta_z=\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}=0-\frac{V}{h}=-\frac{V}{h}} \tag{2}$$

Since vorticity is non-zero, **yes this flow is rotational**. Furthermore, the vorticity is negative, implying that **particles rotate in the clockwise direction**.

Discussion The vorticity is constant at every location in this flow.

4-80

Solution For the given velocity field for Couette flow, we are to calculate the two-dimensional linear strain rates and the shear strain rate.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The linear strain rates in the *x* direction and in the *y* direction are

Linear strain rates:
$$\varepsilon_{xx} = \frac{\partial u}{\partial x} = \mathbf{0} \qquad \varepsilon_{yy} = \frac{\partial v}{\partial y} = \mathbf{0} \tag{1}$$

The shear strain rate in the *x*-*y* plane is

Shear strain rate:
$$\boxed{\varepsilon_{xy} = \frac{1}{2}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) = \frac{1}{2}\left(\frac{V}{h} + 0\right) = \frac{V}{2h}} \tag{2}$$

Fluid particles in this flow have non-zero shear strain rate.

Discussion Since the linear strain rates are zero, fluid particles deform (shear), but do not *stretch* in either the horizontal or vertical directions.

4-81

Solution For the Couette flow velocity field we are to form the 2-D strain rate tensor and determine if the *x* and *y* axes are principal axes.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The two-dimensional strain rate tensor, ε_{ij}, is

2-D strain rate tensor:
$$\varepsilon_{ij} = \begin{pmatrix} \varepsilon_{xx} & \varepsilon_{xy} \\ \varepsilon_{yx} & \varepsilon_{yy} \end{pmatrix} \tag{1}$$

We use the linear strain rates and the shear strain rate from the previous problem to generate the tensor,

2-D strain rate tensor:
$$\boxed{\varepsilon_{ij} = \begin{pmatrix} \varepsilon_{xx} & \varepsilon_{xy} \\ \varepsilon_{yx} & \varepsilon_{yy} \end{pmatrix} = \begin{pmatrix} 0 & \dfrac{V}{2h} \\ \dfrac{V}{2h} & 0 \end{pmatrix}} \tag{2}$$

Note that by symmetry $\varepsilon_{yx} = \varepsilon_{xy}$. If the *x* and *y* axes were principal axes, the diagonals of ε_{ij} would be non-zero, and the off-diagonals would be zero. Here we have the opposite case, so **the x and y axes are not principal axes**.

Discussion The principal axes can be calculated using tensor algebra.

4-82

Solution For a given velocity field we are to calculate the vorticity.

Analysis The velocity field is

$$\vec{V} = (u, v, w) = (2.49 + 1.36x - 0.867y)\vec{i} + (1.95x - 1.36y)\vec{j} + (-0.458xy)\vec{k} \qquad (1)$$

In Cartesian coordinates, the vorticity vector is

Vorticity vector in Cartesian coordinates:

$$\vec{\zeta} = \left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right)\vec{i} + \left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right)\vec{j} + \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right)\vec{k} \qquad (2)$$

We substitute velocity components $u = 2.49 + 1.36x - 0.867y$, $v = 1.95x - 1.36y$, and $w = -0.458xy$ to obtain

Vorticity vector:

$$\boxed{\vec{\zeta} = (-0.458x - 0)\vec{i} + (0 - (-0.458y))\vec{j} + (1.95 - (-0.867))\vec{k} = (-0.458x)\vec{i} + (0.458y)\vec{j} + (2.817)\vec{k}} \qquad (3)$$

Discussion The vorticity is non-zero implying that this flow field is *rotational*.

4-83

Solution For a given velocity field we are to calculate the constant *c* such that the flow field is irrotational.

Analysis The velocity field is

$$\vec{V} = (u, v) = (2.85 + 1.26x - 0.896y)\vec{i} + (3.45 + cx - 1.26y)\vec{j} \qquad (1)$$

In Cartesian coordinates, the vorticity vector is

Vorticity vector in Cartesian coordinates:

$$\vec{\zeta} = \left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right)\vec{i} + \left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right)\vec{j} + \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right)\vec{k} \qquad (2)$$

We substitute velocity components $u = 2.85 + 1.26x - 0.896y$, $v = 3.45 + cx - 1.26y$, and $w = 0$ to obtain

Vorticity vector:

$$\vec{\zeta} = (0)\vec{i} + (0)\vec{j} + (c - (-0.896))\vec{k} = (c + 0.896)\vec{k}$$

For irrotational flow, the vorticity is set to zero, yielding ***c* = -0.896**.

Discussion For any other value of *c* the vorticity would be non-zero implying that the flow field would be *rotational*.

4-84

Solution For a given velocity field we are to calculate the constants *b* and *c* such that the flow field is irrotational.

Analysis The velocity field is

$$\vec{V} = (1.35 + 2.78x + 0.754y + 4.21z)\vec{i} + (3.45 + cx - 2.78y + bz)\vec{j} + (-4.21x - 1.89y)\vec{j} \qquad (1)$$

In Cartesian coordinates, the vorticity vector is

Vorticity vector in Cartesian coordinates:

$$\vec{\zeta} = \left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right)\vec{i} + \left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right)\vec{j} + \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right)\vec{k} \qquad (2)$$

We substitute velocity components $u = 1.35 + 2.78x + 0.754y + 4.21z$, $v = 3.45 + cx - 2.78y + bz$, and $w = -4.21x - 1.89y$ from Eq. 1 into Eq. 2 to obtain

Vorticity vector:

$$\vec{\zeta} = (-1.89 - b)\vec{i} + (4.21 - (-4.21))\vec{j} + (c - 0.754)\vec{k} = (-1.89 - b)\vec{i} + (0)\vec{j} + (c - 0.754)\vec{k}$$

For irrotational flow, each component of vorticity must be zero, yielding ***b* = -1.89** and ***c* = 0.754**.

Discussion Any other values of *b* and/or *c* would make the vorticity non-zero implying a rotational flow field.

4-85

Solution For a given velocity field we are to calculate the constants *a*, *b*, and *c* such that the flow field is irrotational.

Analysis The velocity field is

$$\vec{V} = \left(0.657 + 1.73x + 0.948y + az\right)\vec{i} + \left(2.61 + cx + 1.91y + bz\right)\vec{j} + \left(-2.73x - 3.66y - 3.64z\right)\vec{j} \qquad (1)$$

In Cartesian coordinates, the vorticity vector is

Vorticity vector in Cartesian coordinates:

$$\vec{\zeta} = \left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right)\vec{i} + \left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right)\vec{j} + \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right)\vec{k} \qquad (2)$$

We substitute components $u = 0.657 + 1.73x + 0.948y + az$, $v = 2.61 + cx + 1.91y + bz$, and $w = -2.73x - 3.66y - 3.64z$ from Eq. 1 into Eq. 2 to obtain

Vorticity vector:

$$\vec{\zeta} = \left(-3.66 - b\right)\vec{i} + \left(a - \left(-2.73\right)\right)\vec{j} + \left(c - 0.948\right)\vec{k}$$

For irrotational flow, each component of vorticity must be zero, yielding ***a* = -2.73**, ***b* = -3.66** and ***c* = 0.948**.

Discussion Any other values of *b* and/or *c* would make the vorticity non-zero implying a rotational flow field.

4-86E

Solution For a given velocity field and an initially square fluid particle, we are to calculate and plot its location and shape after a given time period.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis Using the results of Problems 4-51 and 4-54, we can calculate the location of any point on the fluid particle after the elapsed time. We pick 6 points along each edge of the fluid particle, and plot their *x* and *y* locations at *t* = 0 and at *t* = 0.2 s. For example, the point at the lower left corner of the particle is initially at *x* = 0.25 ft and *y* = 0.75 ft at *t* = 0. At *t* = 0.2 s,

x-location of lower left corner of the fluid particle at time t = 0.2 s:

$$x = \frac{1}{4.6\ 1/\text{s}}\left[\left(5.0\ \text{ft/s} + \left(4.6\ 1/\text{s}\right)\left(0.25\ \text{ft}\right)\right)e^{(4.6\ 1/\text{s})(0.2\ \text{s})} - 5.0\ \text{ft/s}\right] = \mathbf{2.268\ ft}$$

and

y-location of lower left corner of the fluid particle at time t = 0.2 s:

$$y = \left(0.75\ \text{ft}\right)e^{-(4.6\ 1/\text{s})(0.2\ \text{s})} = \mathbf{0.2989\ ft}$$

We repeat the above calculations at all the points along the edges of the fluid particle, and plot both their initial and final positions in **Fig. 1** as dots. Finally, we connect the dots to draw the fluid particle shape. It is clear from the results that the fluid particle shrinks in the *y* direction and stretches in the *x* direction. However, it does not shear or rotate.

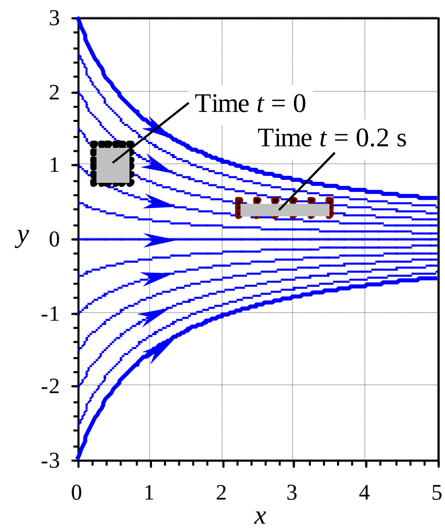


FIGURE 1
Movement and distortion of an initially square fluid particle in a converging duct; *x* and *y* are in units of ft. Streamlines (solid blue curves) are also shown for reference.

Discussion The flow is irrotational since fluid particles do not rotate.

4-87E

Solution By analyzing the shape of a fluid particle, we are to verify that the given flow field is incompressible.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis Since the flow is two-dimensional, we assume unit depth (1 ft) in the *z* direction (into the page in the figure). In the previous problem, we calculated the initial and final locations of several points on the perimeter of an initially square fluid particle. At t = 0, the particle volume is

Fluid particle volume at t = 0 s: $$V = (0.50\text{ ft})(0.50\text{ ft})(1.0\text{ ft}) = 0.25\text{ ft}^3 \qquad (1)$$

At t = 0.2 s, the lower left corner of the fluid particle has moved to x = 2.2679 ft, y = 0.29889 ft, and the upper right corner has moved to x = 3.5225 ft, y = 0.49815 ft. Since the fluid particle remains rectangular, we can calculate the fluid particle volume from these two corner locations,

Fluid particle volume at t = 0.2 s:

$$V = (3.5225\text{ ft} - 2.2679\text{ ft})(0.49815\text{ ft} - 0.29889\text{ ft})(1.0\text{ ft}) = 0.2500\text{ ft}^3 \qquad (2)$$

Thus, to at least four significant digits, the fluid particle volume has not changed, and **the flow is therefore incompressible**.

Discussion The fluid particle stretches in the horizontal direction and shrinks in the vertical direction, but the net volume of the fluid particle does not change.

Reynolds Transport Theorem

4-88C

Solution We are to explain the similarities and differences between the material derivative and the RTT.

Analysis The main similarity is that both of them **transform from a Lagrangian or system viewpoint to an Eulerian or control volume viewpoint**. Other similarities include that both the material derivative and the RTT contain two terms on the right side – an unsteady term that is nonzero only when the flow is changing in time, and an advective part, which accounts for the fluid particle or system moving to a new part of the flow field. The main difference between the two is that the material derivative applies to infinitesimal fluid particles, while the RTT applies to finite systems and control volumes.

Discussion It turns out that if we let the system shrink to a point, the RTT reduces directly to the material derivative.

4-89C

Solution We are to explain the purpose of the Reynolds transport theorem, and write the RTT for intensive property *b* as a "word equation."

Analysis The purpose of the RTT is **to convert conservation equations from their fundamental form for a system (closed system) to a form that can be applied to a control volume (open system)**. In other words, the RTT provides a **link between the system approach and the control volume approach** to a fluid flow problem. We can also explain the RTT as a **transformation from the Lagrangian to the Eulerian frame of reference**. The RTT (Eq. 4-41) is

$$\frac{dB_{\text{sys}}}{dt} = \frac{d}{dt}\int_{\text{CV}} \rho b dV + \int_{\text{CS}} \rho b \vec{V} \cdot \vec{n} dA \qquad (1)$$

In word form, Eq. 1 may be stated something like this: **The time rate of change of property B of the system is equal to the time rate of change of *B* of the control volume due to unsteadiness plus the net flux of *B* across the control surface due to fluid flow**.

Discussion Students should write the RTT in their own words.

4-90C

Solution

(*a*) **False**: The statement is backwards, since the conservation laws are naturally occurring in the system form.
(*b*) **False**: The RTT can be applied to any control volume, fixed, moving, or deforming.
(*c*) **True**: The RTT has an unsteady term and can be applied to unsteady problems.
(d) **True**: The extensive property *B* (or its intensive form *b*) in the RTT can be any property of the fluid – scalar, vector, or even tensor.

4-91

Solution We are to solve an integral two ways – straightforward and using the Leibniz theorem.

Analysis (*a*) We integrate first and then take the time derivative,

$$\frac{d}{dt}\int_t^{2t} x^{-2}dx = \frac{d}{dt}\left\{\left[-x^{-1}\right]_t^{2t}\right\} = \frac{d}{dt}\left[-\frac{1}{2t}+\frac{1}{t}\right] = \frac{d}{dt}\left[\frac{1}{2t}\right] = -\frac{1}{2t^2} \qquad (1)$$

(*b*) We use the 1-D Leibniz theorem,

$$\frac{d}{dt}\int_{x=a(t)}^{x=b(t)} G(x,t)\,dx = \int_a^b \frac{\partial G}{\partial t}dx + \frac{db}{dt}G(b,t) - \frac{da}{dt}G(a,t) \qquad (2)$$

Here, $G = x^{-2}$, $a = t$, $b = 2t$, $\partial G/\partial t = 0$, $db/dt = 2$, and $da/dt = 1$. Thus, Eq. 2 becomes

$$\frac{d}{dt}\int_t^{2t} x^{-2}dx = 0 + 2(2t)^{-2} - 1t^{-2} = -\frac{1}{2t^2} \qquad (3)$$

Thus, the integral reduces to $\mathbf{-2t^{-2}}$, and we get the same answer using either technique.

Discussion In this problem, we could integrate before taking the time derivative, but the real usefulness of Leibniz theorem is in situations where this cannot be done.

4-92

Solution We are to solve an integral.

Analysis There does not appear to be a simple straightforward solution, so we use the 1-D Leibniz theorem,

$$\frac{d}{dt}\int_{x=a(t)}^{x=b(t)} G(x,t)\,dx = \int_a^b \frac{\partial G}{\partial t}dx + \frac{db}{dt}G(b,t) - \frac{da}{dt}G(a,t) \qquad (1)$$

Here, $G = x^x$, $a = t$, $b = 2t$, $\partial G/\partial t = 0$, $db/dt = 2$, and $da/dt = 1$. Thus, the integral becomes

$$\frac{d}{dt}\int_t^{2t} x^x dx = 0 + 2(2t)^{2t} - 1t^t = 2(2t)^{2t} - t^t \qquad (3)$$

Thus, the integral reduces to $\boxed{2(2t)^{2t} - t^t}$.

Discussion The present author does not know how to solve this integral without using Leibniz theorem.

4-93

Solution For the case in which B_{sys} is the mass m of a system, we are to use the RTT to derive the equation of conservation of mass for a control volume.

Analysis The general form of the Reynolds transport theorem is given by

General form of the RTT: $$\frac{dB_{\text{sys}}}{dt} = \frac{d}{dt}\int_{\text{CV}} \rho b dV + \int_{\text{CS}} \rho b \vec{V}_r \cdot \vec{n} dA \qquad (1)$$

Setting $B_{\text{sys}} = m$ means that $b = m/m = 1$. Plugging these and $dm/dt = 0$ into Eq. 1 yields

Conservation of mass for a CV: $$\boxed{0 = \frac{d}{dt}\int_{\text{CV}} \rho dV + \int_{\text{CS}} \rho \vec{V}_r \cdot \vec{n} dA} \qquad (2)$$

Discussion Eq. 2 is general and applies to any control volume – fixed, moving, or even deforming.

4-94

Solution For the case in which B_{sys} is the linear momentum $m\vec{V}$ of a system, we are to use the RTT to derive the equation of conservation of linear momentum for a control volume.

Analysis Newton's second law is

Newton's second law for a system: $$\sum \vec{F} = m\vec{a} = m\frac{d\vec{V}}{dt} = \frac{d}{dt}\left(m\vec{V}\right)_{sys} \qquad (1)$$

Setting B_{sys} = $m\vec{V}$ means that b = $m\vec{V}/m = \vec{V}$. Plugging these and Eq. 1 into the equation of the previous problem yields

$$\sum \vec{F} = \frac{d}{dt}\left(m\vec{V}\right)_{sys} = \frac{d}{dt}\int_{CV} \rho\vec{V}dV + \int_{CS} \rho\vec{V}\left(\vec{V}_r \cdot \vec{n}\right)dA$$

or simply

Conservation of linear momentum for a CV:

$$\boxed{\sum \vec{F} = \frac{d}{dt}\int_{CV} \rho\vec{V}dV + \int_{CS} \rho\vec{V}\left(\vec{V}_r \cdot \vec{n}\right)dA} \qquad (2)$$

Discussion Eq. 2 is general and applies to any control volume – fixed, moving, or even deforming.

4-95

Solution For the case in which B_{sys} is the angular momentum $\vec{H}$ of a system, we are to use the RTT to derive the equation of conservation of angular momentum for a control volume.

Analysis The conservation of angular momentum is expressed as

Conservation of angular momentum for a system: $$\sum \vec{M} = \frac{d}{dt}\vec{H}_{sys} \qquad (1)$$

Setting B_{sys} = $\vec{H}$ means that b = $\left(\vec{r} \times m\vec{V}\right)/m = \vec{r} \times \vec{V}$, noting that m = constant for a system. Plugging these and Eq. 1 into the equation of Problem 4-78 yields

$$\sum \vec{M} = \frac{d}{dt}\vec{H}_{sys} = \frac{d}{dt}\int_{CV} \rho\left(\vec{r} \times \vec{V}\right)dV + \int_{CS} \rho\left(\vec{r} \times \vec{V}\right)\left(\vec{V}_r \cdot \vec{n}\right)dA$$

or simply

Conservation of angular momentum for a CV:

$$\boxed{\sum \vec{M} = \frac{d}{dt}\int_{CV} \rho\left(\vec{r} \times \vec{V}\right)dV + \int_{CS} \rho\left(\vec{r} \times \vec{V}\right)\left(\vec{V}_r \cdot \vec{n}\right)dA} \qquad (2)$$

Discussion Eq. 2 is general and applies to any control volume – fixed, moving, or even deforming.

4-96

Solution $F(t)$ is to be evaluated from the given expression.

Analysis The integral is

$$F(t) = \frac{d}{dt}\int_{x=At}^{x=Bt} e^{-2x^2}\,dx \qquad (1)$$

We could try integrating first, and then differentiating, but we can instead use the 1-D Leibnitz theorem. Here, $G(x,t) = e^{-2x^2}$ (G is not a function of time in this simple example). The limits of integration are $a(t) = At$ and $b(t) = Bt$. Thus,

$$\begin{aligned} F(t) &= \int_a^b \frac{\partial G}{\partial t}dx + \frac{db}{dt}G(b,t) - \frac{da}{dt}G(a,t) \\ &= \quad 0 \quad + Be^{-2b^2} \quad - Ae^{-2a^2} \end{aligned} \qquad (2)$$

or

$$\boxed{F(t) = Be^{-B^2t^2} - Ae^{-A^2t^2}} \qquad (3)$$

Discussion You are welcome to try to obtain the same solution without using the Leibnitz theorem.

Review Problems

4-97

Solution For a given expression for u, we are to find an expression for v such that the flow field is incompressible.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis The x-component of velocity is given as

x-component of velocity:
$$u = a + b(x-c)^2 \qquad (1)$$

In order for the flow field to be incompressible, the volumetric strain rate must be zero,

Volumetric strain rate:
$$\frac{1}{V}\frac{DV}{Dt} = \varepsilon_{xx} + \varepsilon_{yy} + \varepsilon_{zz} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{Two-D}} = 0 \qquad (2)$$

This gives us a necessary condition for v,

Necessary condition for v:
$$\frac{\partial v}{\partial y} = -\frac{\partial u}{\partial x} \qquad (3)$$

We substitute Eq. 1 into Eq. 3 and integrate to solve for v,

Expression for v:
$$\frac{\partial v}{\partial y} = -\frac{\partial u}{\partial x} = -2b(x-c)$$
$$v = \int \frac{\partial v}{\partial y} dy = \int \left(-2b(x-c)\right) dy + f(x)$$

Note that we must add an arbitrary function of x rather than a simple constant of integration since this is a partial integration with respect to y. v is a function of both x and y. The result of the integration is

Expression for v:
$$\boxed{v = -2b(x-c)y + f(x)} \qquad (4)$$

Discussion We verify by plugging Eqs. 1 and 4 into Eq. 2,

Volumetric strain rate:
$$\frac{1}{V}\frac{DV}{Dt} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 2b(x-c) - 2b(x-c) = 0 \qquad (5)$$

Since the volumetric strain rate is zero for any function $f(x)$, Eqs. 1 and 4 represent an incompressible flow field.

4-98

Solution For a given expression for u, we are to find an expression for v such that the flow field is incompressible.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis The x-component of velocity is given as

x-component of velocity:
$$u = ax + by + cx^2 \qquad (1)$$

In order for the flow field to be incompressible, the volumetric strain rate must be zero,

Volumetric strain rate:
$$\frac{1}{V}\frac{DV}{Dt} = \varepsilon_{xx} + \varepsilon_{yy} + \varepsilon_{zz} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{Two-D}} = 0 \qquad (2)$$

This gives us a necessary condition for v,

Necessary condition for v:
$$\frac{\partial v}{\partial y} = -\frac{\partial u}{\partial x} \qquad (3)$$

We substitute Eq. 1 into Eq. 3 and integrate to solve for v,

Expression for v:
$$\frac{\partial v}{\partial y} = -\frac{\partial u}{\partial x} = -\left(a + 2cx\right)$$
$$v = \int \frac{\partial v}{\partial y} dy = -\int a dy - \int 2cx dy + f(x)$$

Note that we must add an arbitrary function of x rather than a simple constant of integration since this is a partial integration with respect to y. v is a function of both x and y. The result of the integration is

Expression for v:
$$\boxed{v = -ay - 2cxy + f(x)} \qquad (4)$$

Discussion We verify by plugging Eqs. 1 and 4 into Eq. 2,

Volumetric strain rate:
$$\frac{1}{V}\frac{DV}{Dt} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = a + 2cx - a - 2cx = 0 \qquad (5)$$

Since the volumetric strain rate is zero for any function $f(x)$, Eqs. 1 and 4 represent an incompressible flow field.

4-99

Solution We are to determine if the flow is rotational, and if so calculate the *z*-component of vorticity.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity components are given by

Velocity components, 2-D Poiseuille flow: $$u = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right) \qquad v = 0 \qquad (1)$$

If the vorticity is non-zero, the flow is rotational. So, we calculate the *z*-component of vorticity,

z-component of vorticity:
$$\boxed{\zeta_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} = 0 - \frac{1}{2\mu}\frac{dP}{dx}\left(2y - h\right) = -\frac{1}{2\mu}\frac{dP}{dx}\left(2y - h\right)} \qquad (2)$$

Since vorticity is non-zero, **this flow is rotational**. Furthermore, in the lower half of the flow ($y < h/2$) the vorticity is negative (note that dP/dx is negative). Thus, **particles rotate in the clockwise direction in the lower half of the flow**. Similarly, **particles rotate in the counterclockwise direction in the upper half of the flow**.

Discussion The vorticity varies linearly across the channel.

4-100

Solution For the given velocity field for 2-D Poiseuille flow, we are to calculate the two-dimensional linear strain rates and the shear strain rate.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The linear strain rates in the *x* direction and in the *y* direction are

Linear strain rates: $$\varepsilon_{xx} = \frac{\partial u}{\partial x} = \mathbf{0} \qquad \varepsilon_{yy} = \frac{\partial v}{\partial y} = \mathbf{0} \qquad (1)$$

The shear strain rate in the *x*-*y* plane is

Shear strain rate:
$$\boxed{\varepsilon_{xy} = \frac{1}{2}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) = \frac{1}{2}\left(\frac{1}{2\mu}\frac{dP}{dx}\left(2y - h\right) + 0\right) = \frac{1}{4\mu}\frac{dP}{dx}\left(2y - h\right)} \qquad (2)$$

Fluid particles in this flow have non-zero shear strain rate.

Discussion Since the linear strain rates are zero, fluid particles deform (shear), but do not *stretch* in either the horizontal or vertical directions.

4-101

Solution For the 2-D Poiseuille flow velocity field we are to form the 2-D strain rate tensor and determine if the x and y axes are principal axes.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the x-y plane.

Analysis The two-dimensional strain rate tensor, ε_{ij}, in the x-y plane,

2-D strain rate tensor:

$$\varepsilon_{ij} = \begin{pmatrix} \varepsilon_{xx} & \varepsilon_{xy} \\ \varepsilon_{yx} & \varepsilon_{yy} \end{pmatrix} \qquad (1)$$

We use the linear strain rates and the shear strain rate from the previous problem to generate the tensor,

$$\boxed{\varepsilon_{ij} = \begin{pmatrix} \varepsilon_{xx} & \varepsilon_{xy} \\ \varepsilon_{yx} & \varepsilon_{yy} \end{pmatrix} = \begin{pmatrix} 0 & \dfrac{1}{4\mu}\dfrac{dP}{dx}\left(2y-h\right) \\ \dfrac{1}{4\mu}\dfrac{dP}{dx}\left(2y-h\right) & 0 \end{pmatrix}} \qquad (2)$$

Note that by symmetry $\varepsilon_{yx} = \varepsilon_{xy}$. If the x and y axes were principal axes, the diagonals of ε_{ij} would be non-zero, and the off-diagonals would be zero. Here we have the opposite case, so **the x and y axes are not principal axes**.

Discussion The principal axes can be calculated using tensor algebra.

4-102

Solution For a given velocity field we are to plot several pathlines for fluid particles released from various locations and over a specified time period.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the x-y plane.

Properties For water at 40°C, $\mu = 6.53\times10^{-4}$ kg/m·s.

Analysis Since the flow is steady, pathlines, streamlines, and streaklines are all straight horizontal lines. We simply need to integrate velocity component u with respect to time over the specified time period. The horizontal velocity component is

$$u = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right) \qquad (1)$$

We integrate as follows:

$$x = x_{\text{start}} + \int_{t_{\text{start}}}^{t_{\text{end}}} u\,dt = 0 + \int_0^{10\text{ s}} \left(\frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)\right)dt$$
$$x = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)\left(10\text{ s}\right) \qquad (2)$$

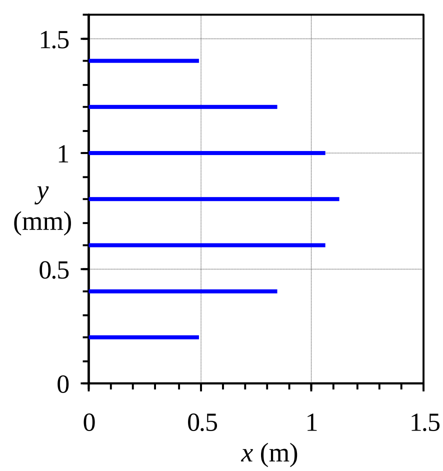


FIGURE 1
Pathlines for the given velocity field at t = 12 s. Note that the vertical scale is greatly expanded for clarity (x is in m, but y is in mm).

We substitute the given values of y and the values of μ and dP/dx into Eq. 2 to calculate the ending x position of each pathline. We plot the pathlines in **Fig. 1**.

Discussion Streaklines introduced at the same locations and developed over the same time period would look identical to the pathlines of Fig. 1.

4-103

Solution For a given velocity field we are to plot several streaklines at a given time for dye released from various locations over a specified time period.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Properties For water at 40°C, $\mu = 6.53\times10^{-4}$ kg/m·s.

Analysis Since the flow is steady, pathlines, streamlines, and streaklines are all straight horizontal lines. We simply need to integrate velocity component u with respect to time over the specified time period. The horizontal velocity component is

$$u = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right) \qquad (1)$$

We integrate as follows to obtain the final x location of the first dye particle released:

$$x = x_{\text{start}} + \int_{t_{\text{start}}}^{t_{\text{end}}} u\,dt = 0 + \int_0^{10\text{ s}}\left(\frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)\right)dt$$
$$x = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)\times(10\text{ s}) \qquad (2)$$

We substitute the given values of y and the values of μ and dP/dx into Eq. 2 to calculate the ending x position of the first released dye particle of each streakline. The *last* released dye particle is at $x = x_{\text{start}} = 0$, because it hasn't had a chance to go anywhere. We connect the beginning and ending points to plot the streaklines (**Fig. 1**).

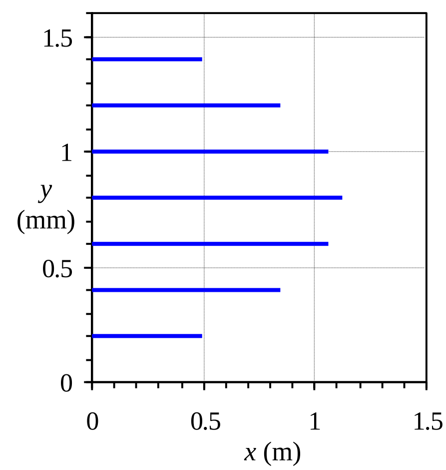


FIGURE 1
Streaklines for the given velocity field at t = 10 s. Note that the vertical scale is greatly expanded for clarity (x is in m, but y is in mm).

Discussion These streaklines are introduced at the same locations and are developed over the same time period as the pathlines of the previous problem. They are identical since the flow is steady.

4-104

Solution For a given velocity field we are to plot several streaklines at a given time for dye released from various locations over a specified time period.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Properties For water at 40°C, $\mu = 6.53\times10^{-4}$ kg/m·s.

Analysis Since the flow is steady, pathlines, streamlines, and streaklines are all straight horizontal lines. The horizontal velocity component is

$$u = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right) \qquad (1)$$

In the previous problem we generated streaklines at t = 10 s. Imagine the dye at the source being suddenly cut off at that time, but the streaklines are observed 2 seconds later, at t = 12 s. The dye streaks will not stretch any further, but will simply move at the same horizontal speed for 2 more seconds. At each y location, the x locations of the first and last dye particle are thus

first dye particle of streakline: $$x = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)\left(12\text{ s}\right) \qquad (2)$$

and

last dye particle of streakline: $$x = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)\left(2\text{ s}\right) \qquad (3)$$

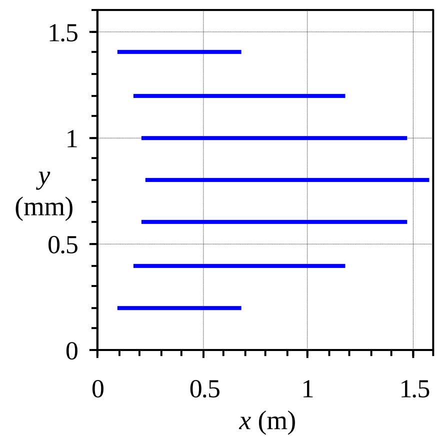


FIGURE 1
Streaklines for the given velocity field at t = 12 s. Note that the vertical scale is greatly expanded for clarity (x is in m, but y is in mm).

We substitute the given values of y and the values of μ and dP/dx into Eqs. 2 and 3 to calculate the ending and beginning x positions of the first released dye particle and the last released dye particle of each streakline. We connect the beginning and ending points to plot the streaklines (**Fig. 1**).

Discussion Both the left and right ends of each dye streak have moved by the same amount compared to those of the previous problem.

4-105

Solution For a given velocity field we are to compare streaklines at two different times and comment about linear strain rate in the *x* direction.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Properties For water at 40°C, $\mu = 6.53\times10^{-4}$ kg/m·s.

Analysis Comparing the results of the previous two problems we see that the streaklines have not stretched at all – they have simply convected downstream. Thus, based on the fundamental definition of linear strain rate, **it is zero**:

Linear strain rate in the x direction: $$\boxed{\varepsilon_{xx} = 0} \qquad (1)$$

Discussion Our result agrees with that of Problem 4-83.

4-106

Solution For a given velocity field we are to plot several timelines at a specified time. The timelines are created by hydrogen bubbles released from a vertical wire at $x = 0$.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x-y* plane.

Properties For water at 40°C, $\mu = 6.53\times10^{-4}$ kg/m·s.

Analysis Since the flow is steady, pathlines, streamlines, and streaklines are all straight horizontal lines, but *timelines* are completely different from any of the others. To simulate a timeline, we integrate velocity component u with respect to time over the specified time period from $t = 0$ to $t = t_{end}$. We introduce the bubbles at $x = 0$ and at many values of y (we used 50 in our simulation). By connecting these x locations with a line, we simulate a timeline. The horizontal velocity component is

x-velocity component: $$u = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right) \qquad (1)$$

We integrate as follows to find the x position on the timeline at t_{end}:

$$x = x_{start} + \int_{t_{start}}^{t_{end}} u\,dt = 0 + \int_0^{t_{end}}\left(\frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)\right)dt$$

$$\rightarrow \qquad x = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)t_{end}$$

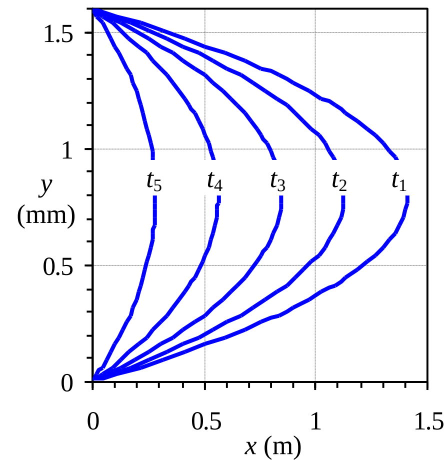


FIGURE 1
Timelines for the given velocity field at t = 12.5 s, generated by a simulated hydrogen bubble wire at $x = 0$. Timelines created at t_5 = 10.0 s, t_4 = 7.5 s, t_3 = 5.0 s, t_2 = 2.5 s, and t_1 = 0 s. Note that the vertical scale is greatly expanded for clarity (x is in m, but y is in mm).

We substitute the values of y and the values of μ and dP/dx into the above equation to calculate the ending x position of each point in the timeline. We repeat for the five values of t_{end}. We plot the timelines in **Fig. 1**.

Discussion Each timeline has the exact shape of the velocity profile.

4-107

Solution For a given velocity field we are to calculate the normal acceleration of a particle.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis The streamlines for a two-dimensional flow are governed by $\frac{dy}{dx}=\frac{v}{u}$. Therefore

$$\frac{dy}{dx}=-\frac{2kxy}{k(x^2-y^2)}=-\frac{2xy}{x^2-y^2}$$

or

$$2xydx+(x^2-y^2)dy=0\,.$$

This is a 2nd order homogenous differential equation. To solve this ODE we set y=px, where p=p(x). Differentiating we get

$$\frac{dy}{dx}=\frac{dp}{dx}x+p$$

The differential equation is then

$$2xy+(x^2-y^2)\frac{dy}{dx}=0$$

or

$$2xpx+(x^2-p^2x^2)\left(\frac{dp}{dx}x+p\right)=0$$

or

$$\frac{2p}{1-p^2}+\frac{dp}{dx}x+p=0\quad,\qquad x\frac{dp}{dx}+\frac{3p-p^3}{1-p^2}=0$$

Rearranging the DE we get

$$\frac{1-p^2}{3p-p^3}dp+\frac{dx}{x}=0$$

Since $\frac{1-p^2}{3p-p^3}=\frac{p^2-1}{p(p^2-3)}=\frac{A}{p}+\frac{Bp+C}{p^2-3}$

$$p^2-1=Ap^2-3A+Bp^2+Cp\,,$$

or

$$A+B=1,\quad C=0$$

$$A=\frac{1}{3}\qquad,\qquad B=\frac{2}{3}$$

Therefore the differential equation becomes,

$$\left(\frac{1}{3p}+\frac{1}{3}\frac{2p}{p^2-3}\right)dp+\frac{dx}{x}=0$$

Integrating both sides of the equation, we get

$$\frac{1}{3}\ln p+\frac{1}{3}\ln(p^2-3)+\ln x=C_1$$

or

$$\ln p+\ln(p^2-3)+\ln x^3=\ln C_2$$

$$\ln(p(p^2-3)x^3)=\ln C_2\ ,$$

Recalling that

$$y=px \quad \text{or} \quad p=\frac{y}{x}$$

$$y^3-3x^2y=C_2$$

is the streamline function. For the givenpoint (x,y)=(1,2)

$$C_2=2^3-3\times 1^2\times 2=8-6=2$$

Therefore the streamline passing through position (1,2) is

$$y^3-3x^2y=2$$

$$3y^2y'+3(2xy+x^2y')=0 \qquad \text{(x,y)=(1,2)}$$

$$3\times 2^2y'+3(2\times 1\times 2+1^2y')=0$$

$$12y'+12+3y'=0\ , \quad 15y'=-12$$

$$y'=-\frac{12}{15}=-\frac{4}{5}=-0.8$$

Differentiating one more time, we get

$$3(2yy'y'+y^2y'')+3(2y+2xy'+2xy'+x^2y'')=0$$

For the given point (x,y)=(1,2)

$$2\times 2\left(-\frac{4}{5}\right)^2+2^2y''+2\times 2+2\times 1\left(-\frac{4}{5}\right)+2\times 1\left(-\frac{4}{5}\right)+1^2y''=0$$

$$2.56+4y''+4-1\times 6\times 2+y''=0$$

$$5y''=-3.36 \qquad , \qquad y''=-0.672$$

Since

$$R=\frac{(1+y'^2)^{3/2}}{\left|y''\right|}=\frac{\left[1+(-0.8)^2\right]^{3/2}}{\left|-0.672\right|}=3.125$$

At the given position

$$u=k(x^2-y^2)=k(1^2-2^2)=-3k$$

$$v=-2kxy=-2k(1\times 2)=-4k$$

The absolute velocity of the particle at that point

$$V^2=u^2+v^2=(-3k)^2+(-4k)^2=25k^2$$

Normal acceleration is then

$$a_n=\frac{V^2}{R}=\frac{25k^2}{3.125}=8k^2$$

4-108

Solution For a given velocity field we are to determine whether the flow is steady and calculate the velocity and acceleration of a particle.

Assumptions **1** The flow is incompressible. **2** The flow is three-dimensional in the *x*-*y*-*z* plane.

Analysis The components of the velocity field are

$$u = 5x^2,\ v = -20xy,\ w = 100t$$

For the steady flow of an incompressible fluid;

$$\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y} + \frac{\partial w}{\partial z} = 0$$

$$\frac{\partial u}{\partial x} = 10x\ ,\ \frac{\partial u}{\partial y} = -20x\ ,\ \frac{\partial u}{\partial z} = 0\ ,$$

Therefore

$$10x - 20x + 0 \neq 0$$

and the flow is unsteady flow. For point $P(\overset{x,}{1},\overset{y,}{2},\overset{z}{3})$ the velocity components are

$$u = 5 - 1^2 = 5, v = -20(2\,x1) = -40$$

$$w = 100\,x\,0.2 = 20$$

and therefore

$$\vec{V}_{P(1,2,3)} = 5\vec{i} - 40\vec{j} + 20\vec{k}$$

The components of the acceleration

$$a_x = u\,\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} + \frac{\partial u}{\partial t}$$

$$= 5x^2 10x + 20xy(-20x) + 100t\,x\,0 + 0$$

$a_x = 50x^3 + 400x^2 y$**,** at point P,

$$a_x = 50\,x1^3 + 400\,x1^2\,x2$$

$$P(1,2,3) \rightarrow a_x = 850$$

$$a_y = u\,\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} + \frac{\partial v}{\partial t}$$

$$u = 5x^2, v = -20xy, w = 100t$$

$$a_y = 5x^2(-20y) - (20xy)\,x(-20x) + 100t\,x\,0 + 0$$

$$= -100x^2 y + 400x^2 y = 300x^2 y$$

at point P(1,2,3),

$$a_y = 300\,x1^2 = 600$$

$$a_z = u\,\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z} + \frac{\partial w}{\partial z}$$

$$= 5x^2\,x\,0 + (-20xy)\,x\,0 + 100t\,x\,0 + 100 = 100$$

Therefore the acceleration at P (1, 2, 3) when t = 0.2 s

$$\vec{a} = 850\vec{i} + 600\vec{j} + 60\vec{k}$$

4-109

Solution We are to determine if the flow is rotational, and if so calculate the θ-component of vorticity.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric about the *x* axis.

Analysis The velocity components are given by

$$u = \frac{1}{4\mu}\frac{dP}{dx}\left(r^2 - R^2\right) \qquad u_r = 0 \qquad u_\theta = 0 \qquad (1)$$

If the vorticity is non-zero, the flow is rotational. So, we calculate the θ-component of vorticity,

θ-component of vorticity:

$$\boxed{\zeta_\theta = \frac{\partial u_r}{\partial z} - \frac{\partial u}{\partial r} = 0 - \frac{1}{4\mu}\frac{dP}{dx}2r = -\frac{r}{2\mu}\frac{dP}{dx}} \qquad (2)$$

Since the vorticity is non-zero, **this flow is rotational**. The vorticity is positive since dP/dx is negative. In this coordinate system, positive vorticity is counterclockwise with respect to the positive θ direction. This agrees with our intuition since in the top half of the flow, θ points out of the page, and the rotation is counterclockwise. Similarly, in the bottom half of the flow, θ points into the page, and the rotation is clockwise.

Discussion The vorticity varies linearly across the pipe from zero at the centerline to a maximum at the pipe wall.

4-110

Solution For the given velocity field for axisymmetric Poiseuille flow, we are to calculate the linear strain rates and the shear strain rate.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric about the *x* axis.

Analysis The linear strain rates in the *x* direction and in the *r* direction are

Linear strain rates:

$$\varepsilon_{xx} = \frac{\partial u}{\partial x} = \mathbf{0} \qquad \varepsilon_{rr} = \frac{\partial u_r}{\partial r} = \mathbf{0} \qquad (1)$$

Thus there is no linear strain rate in either the *x* or the *r* direction. The shear strain rate in the *x*-*r* plane is

Shear strain rate:

$$\boxed{\varepsilon_{xr} = \frac{1}{2}\left(\frac{\partial u_r}{\partial x} + \frac{\partial u}{\partial r}\right) = \frac{1}{2}\left(0 + \frac{1}{4\mu}\frac{dP}{dx}2r\right) = \frac{r}{4\mu}\frac{dP}{dx}} \qquad (2)$$

Fluid particles in this flow have non-zero shear strain rate.

Discussion Since the linear strain rates are zero, fluid particles deform (shear), but do not *stretch* in either the horizontal or radial directions.

4-111

Solution For the axisymmetric Poiseuille flow velocity field we are to form the axisymmetric strain rate tensor and determine if the x and r axes are principal axes.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric about the x axis.

Analysis The axisymmetric strain rate tensor, ε_{ij}, is

Axisymmetric strain rate tensor:
$$\varepsilon_{ij} = \begin{pmatrix} \varepsilon_{rr} & \varepsilon_{rx} \\ \varepsilon_{xr} & \varepsilon_{xx} \end{pmatrix} \qquad (1)$$

We use the linear strain rates and the shear strain rate from the previous problem to generate the tensor,

Axisymmetric strain rate tensor:
$$\boxed{\varepsilon_{ij} = \begin{pmatrix} \varepsilon_{rr} & \varepsilon_{rx} \\ \varepsilon_{xr} & \varepsilon_{xx} \end{pmatrix} = \begin{pmatrix} 0 & \dfrac{r}{4\mu}\dfrac{dP}{dx} \\ \dfrac{r}{4\mu}\dfrac{dP}{dx} & 0 \end{pmatrix}} \qquad (2)$$

Note that by symmetry $\varepsilon_{rx} = \varepsilon_{xr}$. If the x and r axes were principal axes, the diagonals of ε_{ij} would be non-zero, and the off-diagonals would be zero. Here we have the opposite case, so **the *x* and *r* axes are not principal axes**.

Discussion The principal axes can be calculated using tensor algebra.

4-112

Solution We are to determine the location of stagnation point(s) in a given velocity field.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis The velocity components are

x-component of velocity:
$$u = \frac{-\dot{V}x}{\pi L}\frac{x^2+y^2+b^2}{x^4+2x^2y^2+2x^2b^2+y^4-2y^2b^2+b^4} \qquad (1)$$

and

y-component of velocity:
$$v = \frac{-\dot{V}y}{\pi L}\frac{x^2+y^2-b^2}{x^4+2x^2y^2+2x^2b^2+y^4-2y^2b^2+b^4} \qquad (2)$$

Both u and v must be zero at a stagnation point. From Eq. 1, u can be zero only when $x = 0$. From Eq. 2, v can be zero either when $y = 0$ or when $x^2 + y^2 - b^2 = 0$. Combining the former with the result from Eq. 1, we see that **there is a stagnation point at (x,y) = (0,0), i.e. at the origin**,

Stagnation point:
$$\boxed{u = 0 \text{ and } v = 0 \text{ at } (x,y) = (0,0)} \qquad (3)$$

Combining the latter with the result from Eq. 1, there appears to be another stagnation point at (x,y) = (0,b). However, at that location, Eq. 2 becomes

y-component of velocity:
$$v = \frac{-\dot{V}b}{\pi L}\frac{0}{b^4-2b^2b^2+b^4} = \frac{0}{0} \qquad (4)$$

This point turns out to be a **singularity point** in the flow. Thus, **the location (0,*b*) is *not* a stagnation point** after all.

Discussion There is only one stagnation point in this flow, and it is at the origin.

4-113

Solution We are to draw a velocity vector plot for a given velocity field.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis We generate an array of *x* and *y* values in the given range and calculate *u* and *v* from Eqs. 1 and 2 respectively at each location. We choose an appropriate scale factor for the vectors and then draw arrows to form the velocity vector plot (**Fig. 1**).

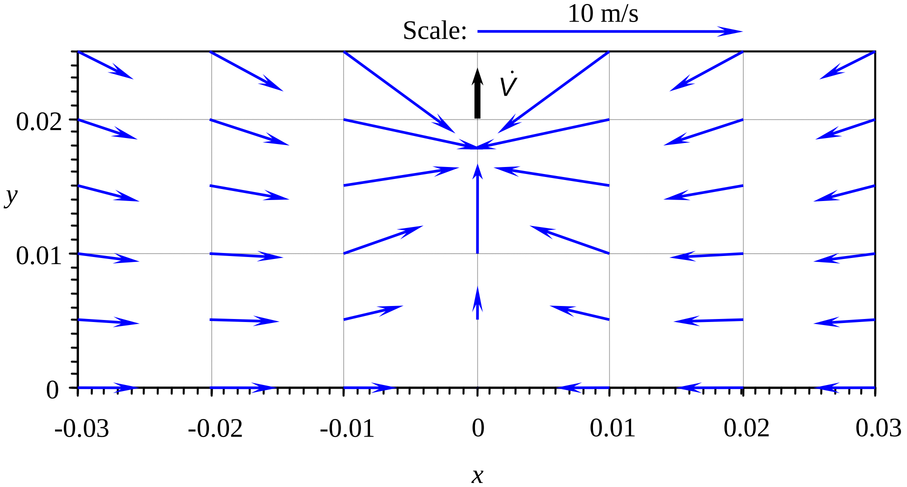


FIGURE 1
Velocity vector plot for the vacuum cleaner; the scale factor for the velocity vectors is shown on the legend. *x* and *y* values are in meters. The vacuum cleaner inlet is at the point *x* = 0, *y* = 0.02 m.

It is clear from the velocity vector plot how the air gets sucked into the vacuum cleaner from all directions. We also see that there is no flow through the floor.

Discussion We discuss this problem in more detail in Chap. 10.

4-114

Solution We are to calculate the speed of air along the floor due to a vacuum cleaner, and find the location of maximum speed.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis At the floor, *y* = 0. Setting *y* = 0 in Eq. 2 of Problem 4-93 shows that *v* = 0, as expected – no flow through the floor. Setting *y* = 0 in Eq. 1 of Problem 4-93 results in the speed along the floor,

Speed on the floor:

$$u = \frac{-\dot{V}x}{\pi L}\frac{x^2+b^2}{x^4+2x^2b^2+b^4} = \frac{-\dot{V}x}{\pi L}\frac{x^2+b^2}{\left(x^2+b^2\right)^2} = \frac{-\dot{V}x}{\pi L\left(x^2+b^2\right)} \qquad (1)$$

We find the maximum speed be differentiating Eq. 1 and setting the result to zero,

Maximum speed on the floor:

$$\frac{du}{dx} = \frac{-\dot{V}}{\pi L}\left[\frac{-2x^2}{\left(x^2+b^2\right)^2} + \frac{1}{x^2+b^2}\right] = 0 \qquad (2)$$

After some algebraic manipulation, we find that Eq. 2 has solutions at *x* = *b* and *x* = -*b*. **It is at *x* = *b* and *x* = -*b* where we expect the best performance**. At the origin, directly below the vacuum cleaner inlet, the flow is stagnant. Thus, despite our intuition, **the vacuum cleaner will work poorly directly below the inlet**.

Discussion Try some experiments at home to verify these results!

4-115

Solution For a given expression for u, we are to find an expression for v such that the flow field is incompressible.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the x-y plane.

Analysis The x-component of velocity is given as

x-component of velocity:
$$u = ax + by + cx^2 - dxy \qquad (1)$$

In order for the flow field to be incompressible, the volumetric strain rate must be zero,

Volumetric strain rate:
$$\frac{1}{V}\frac{DV}{Dt} = \varepsilon_{xx} + \varepsilon_{yy} + \varepsilon_{zz} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{Two-D}} = 0 \qquad (2)$$

This gives us a necessary condition for v,

Necessary condition for v:
$$\frac{\partial v}{\partial y} = -\frac{\partial u}{\partial x} \qquad (3)$$

We substitute Eq. 1 into Eq. 3 and integrate to solve for v,

Expression for v:
$$\frac{\partial v}{\partial y} = -\frac{\partial u}{\partial x} = -\left(a + 2cx - dy\right)$$
$$v = \int \frac{\partial v}{\partial y} dy = -\int ady - \int 2cxdy + \int dydy + f(x)$$

Note that we must add an arbitrary function of x rather than a simple constant of integration since this is a partial integration with respect to y. v is a function of both x and y. The result of the integration is

Expression for v:
$$\boxed{v = -ay - 2cxy + d\frac{y^2}{2} + f(x)} \qquad (4)$$

Discussion We verify by plugging Eqs. 1 and 4 into Eq. 2,

Volumetric strain rate:
$$\frac{1}{V}\frac{DV}{Dt} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = a + 2cx - dy - a - 2cx + dy = 0 \qquad (5)$$

Since the volumetric strain rate is zero for any function $f(x)$, Eqs. 1 and 4 represent an incompressible flow field.

4-116

Solution For a given velocity field we are to determine if the flow is rotational or irrotational.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis The velocity components for flow over a circular cylinder of radius r are

$$u_r = V\cos\theta\left(1-\frac{a^2}{r^2}\right) \qquad u_\theta = -V\sin\theta\left(1+\frac{a^2}{r^2}\right) \tag{1}$$

Since the flow is assumed to be two-dimensional in the r-θ plane, the only non-zero component of vorticity is in the z direction. In cylindrical coordinates,

Vorticity component in the z direction:

$$\zeta_z = \frac{1}{r}\left(\frac{\partial(ru_\theta)}{\partial r}-\frac{\partial u_r}{\partial\theta}\right) \tag{2}$$

We plug in the velocity components of Eq. 1 into Eq. 2 to solve for ζ_z,

$$\zeta_z = \frac{1}{r}\left[\frac{\partial}{\partial r}\left(-V\sin\theta\left(r+\frac{a^2}{r}\right)\right)+V\sin\theta\left(1-\frac{a^2}{r^2}\right)\right] = \frac{1}{r}\left[-V\sin\theta+V\frac{a^2}{r^2}\sin\theta+V\sin\theta-V\frac{a^2}{r^2}\sin\theta\right]=0 \tag{3}$$

Hence, since the vorticity is everywhere zero, **this flow is irrotational**.

Discussion Fluid particles distort as they flow around the cylinder, but their net rotation is zero.

4-117

Solution For a given velocity field we are to find the location of the stagnation point.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis The stagnation point occurs when both components of velocity are zero. We set u_r = 0 and u_θ = 0 in Eq. 1 of the previous problem,

$$u_r = V\cos\theta\left(1-\frac{a^2}{r^2}\right)=0 \qquad \text{Either } \cos\theta=0 \text{ or } r^2=a^2$$
$$u_\theta = -V\sin\theta\left(1+\frac{a^2}{r^2}\right)=0 \qquad \text{Either } \sin\theta=0 \text{ or } r^2=-a^2 \tag{1}$$

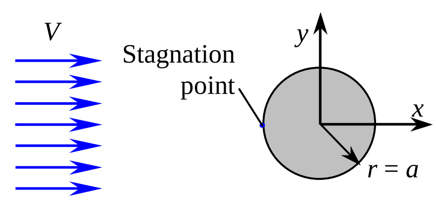


FIGURE 1
The stagnation point on the upstream half of the flow field is located at the nose of the cylinder at $r = a$ and $\theta = 180^\circ$.

The second part of the u_θ condition in Eq. 1 is obviously impossible since cylinder radius a is a real number. Thus $\sin\theta = 0$, which means that $\theta = 0^\circ$ or 180°. We are restricted to the left half of the flow ($x < 0$); therefore we choose $\theta = 180^\circ$. Now we look at the u_r condition in Eq. 1. At $\theta = 180^\circ$, $\cos\theta = -1$, and thus we conclude that r must equal a. Summarizing,

Stagnation point:

$$\boxed{r=a \qquad \theta=-180^\circ} \tag{2}$$

Or, in Cartesian coordinates,

Stagnation point:

$$\boxed{x=-a \qquad y=0} \tag{3}$$

The stagnation point is located at the nose of the cylinder (Fig. 1).

Discussion This result agrees with our intuition, since the fluid must divert around the cylinder at the nose.

4-118

Solution For a given stream function we are to generate an equation for streamlines, and then plot several streamlines in the upstream half of the flow field.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis

(a) The stream function is

$$\psi = V\sin\theta\left(r - \frac{a^2}{r}\right) \qquad (1)$$

First we multiply both sides of Eq. 1 by r, and then solve the quadratic equation for r using the quadratic rule. This gives us an equation for r as a function of θ, with ψ, a, and V as parameters,

Equation for a streamline:

$$\boxed{r = \frac{\psi \pm \sqrt{\psi^2 + 4a^2V^2\sin^2\theta}}{2V\sin\theta}} \qquad (2)$$

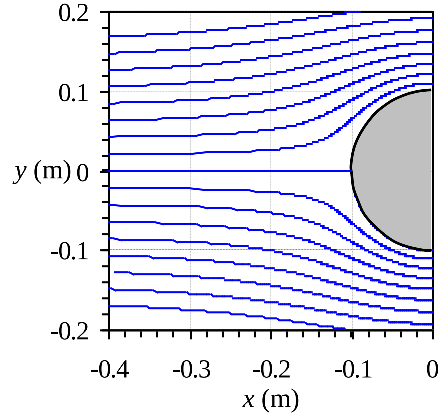


FIGURE 1
Streamlines corresponding to flow over a circular cylinder. Only the upstream half of the flow field is plotted.

(b) For the particular case in which V = 1.00 m/s and cylinder radius a = 10.0 cm, we choose various values of ψ in Eq. 2, and plot streamlines in the upstream half of the flow (**Fig. 1**). Each value of ψ corresponds to a unique streamline.

Discussion The stream function is discussed in greater detail in Chap. 9.

4-119

Solution For a given velocity field we are to calculate the linear strain rates ε_{rr} and $\varepsilon_{\theta\theta}$ in the r-θ plane.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis We substitute the equation of Problem 4-97 into that of Problem 4-91,

Linear strain rate in r direction:

$$\boxed{\varepsilon_{rr} = \frac{\partial u_r}{\partial r} = 2V\cos\theta\frac{a^2}{r^3}} \qquad (1)$$

and

Linear strain rate in θ direction:

$$\boxed{\varepsilon_{\theta\theta} = \frac{1}{r}\left[\frac{\partial u_\theta}{\partial \theta} + u_r\right] = \frac{1}{r}\left[-V\cos\theta\left(1+\frac{a^2}{r^2}\right) + V\cos\theta\left(1-\frac{a^2}{r^2}\right)\right] = -2V\cos\theta\frac{a^2}{r^3}} \qquad (2)$$

The linear strain rates are non-zero, implying that fluid line segments *do* stretch (or shrink) as they move about in the flow field.

Discussion The linear strain rates decrease rapidly with distance from the cylinder.

4-120

Solution We are to discuss whether the flow field of the previous problem is incompressible or compressible.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis For two-dimensional flow we know that a flow is incompressible if its volumetric strain rate is zero. In that case,

Volumetric strain rate, incompressible 2-D flow in the x-y plane:
$$\frac{1}{V}\frac{DV}{Dt}=\varepsilon_{xx}+\varepsilon_{yy}=\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad (1)$$

We can extend Eq. 1 to cylindrical coordinates by writing

Volumetric strain rate, incompressible 2-D flow in the r-θ plane:
$$\frac{1}{V}\frac{DV}{Dt}=\varepsilon_{rr}+\varepsilon_{r\theta}=\frac{\partial u_r}{\partial r}+\frac{1}{r}\left[\frac{\partial u_\theta}{\partial \theta}+u_r\right]=0 \qquad (2)$$

Plugging in the results of the previous problem we see that

Volumetric strain rate for flow over a circular cylinder:
$$\frac{1}{V}\frac{DV}{Dt}=2V\cos\theta\frac{a^2}{r^3}-2V\cos\theta\frac{a^2}{r^3}=0 \qquad (3)$$

Since the volumetric strain rate is zero everywhere, **the flow is incompressible**.

Discussion In Chap. 9 we show that Eq. 2 can be obtained from the differential equation for conservation of mass.

4-121

Solution For a given velocity field we are to calculate the shear strain rate $\varepsilon_{r\theta}$.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis We substitute the equation of Problem 4-97 into that of Problem 4-91,

Shear strain rate in r-θ plane:
$$\begin{aligned}\varepsilon_{r\theta}&=\frac{1}{2}\left[r\frac{\partial}{\partial r}\left(\frac{u_\theta}{r}\right)+\frac{1}{r}\frac{\partial u_r}{\partial \theta}\right]\\&=\frac{1}{2}\left[r\frac{\partial}{\partial r}\left(-\frac{V\sin\theta}{r}-V\sin\theta\frac{a^2}{r^3}\right)+\frac{1}{r}\left(-V\sin\theta\left(1-\frac{a^2}{r^2}\right)\right)\right] \qquad (1)\\&=\frac{1}{2}V\sin\theta\left[\frac{1}{r}+3\frac{a^2}{r^3}-\frac{1}{r}+\frac{a^2}{r^3}\right]=2V\sin\theta\frac{a^2}{r^3}\end{aligned}$$

which reduces to

Shear strain rate in r-θ plane:
$$\boxed{\varepsilon_{r\theta}=2V\sin\theta\frac{a^2}{r^3}} \qquad (2)$$

The shear strain rate is non-zero, implying that fluid line segments *do* deform with shear as they move about in the flow field.

Discussion The shear strain rate decreases rapidly (as r^{-3}) with distance from the cylinder.

Fundamentals of Engineering (FE) Exam Problems

4-122

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u, v) = (2.5 - 1.6x)\vec{i} + (0.7 + 1.6y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The values of *x* and *y* at the stagnation point, respectively, are

(*a*) 0.9375 m, 0.375 m (*b*) 1.563 m, –0.4375 m (*c*) 2.5 m, 0.7 m (*d*) 0.731 m, 1.236 m

(*e*) –1.6 m, 0.8 m

Answer (*b*) 1.563 m, –0.4375 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
u=2.5-1.6*x
v=0.7+1.6*y
u=0
v=0
```

4-123

Water is flowing in a 3-cm-diameter garden hose at a rate of 30 L/min. A 20-cm nozzle is attached to the hose which decreases the diameter to 1.2 cm. The magnitude of the acceleration of a fluid particle moving down the centerline of the nozzle is

(*a*) 9.81 m/s^2 (*b*) 14.5 m/s^2 (*c*) 25.4 m/s^2 (*d*) 39.1 m/s^2 (*e*) 47.6 m/s^2

Answer (*e*) 47.6 m/s^2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D1=0.03 [m]
V_dot=30 [L/min]*Convert(L/min, m^3/s)
DELTAx=0.20 [m]
D2=0.012 [m]
A1=pi*D1^2/4
A2=pi*D2^2/4
u_inlet=V_dot/A1
u_outlet=V_dot/A2
a_x=(u_outlet^2-u_inlet^2)/(2*DELTAx)
```

4-124

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u, v) = (2.5 - 1.6x)\vec{i} + (0.7 + 1.6y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The *x*-component of the acceleration vector a_x is

(*a*) 0.8*y* (*b*) –1.6*x* (*c*) 2.5*x* – 1.6 (*d*) 2.56*x* – 4 (*e*) 2.56*x* + 0.8*y*

Answer (*d*) 2.56*x* – 4

```
"u=2.5-1.6x
v=0.7+1.6y
a_x=u(du/dx)+v(du/dy) = (2.5-1.6x)(-1.6)
a_x=-4+2.56x"
```

4-125

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u, v) = (2.5 - 1.6x)\vec{i} + (0.7 + 1.6y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The *x*- and *y*-component of material acceleration a_x and a_y at the point (x = 1 m, y = 1 m), respectively, in m/s^2, are

(*a*) –1.44, 3.68 (*b*) –1.6, 1.5 (*c*) 3.1, –1.32 (*d*) 2.56, –4 (*e*) –0.8, 1.6

Answer (*a*) –1.44, 3.68

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
u=2.5-1.6*x
v=0.7+1.6*y
x=1
y=1
a_x=(2.5-1.6*x)*(-1.6) "a_x=u(du/dx)+v(du/dy)"
a_y=(0.7+1.6*y)*(1.6) "a_y=u(dv/dx)+v(dv/dy)"
```

4-126

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u, v) = (0.65 + 1.7x)\vec{i} + (1.3 - 1.7y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The *y*-component of the acceleration vector a_y is

(*a*) 1.7*y* (*b*) –1.7*y* (*c*) 2.89*y* – 2.21 (*d*) 3.0*x* – 2.73 (*e*) 0.84*y* + 1.42

Answer (*c*) 2.89*y* – 2.21

```
"u=0.65+1.7x
v=1.3-1.7y
a_y=u(dv/dx)+v(dv/dy) = (1.3-1.7y)(-1.7)
a_y=-2.21+2.89y"
```

4-127

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u, v) = (0.65 + 1.7x)\vec{i} + (1.3 - 1.7y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The *x*- and *y*-component of material acceleration a_x and a_y at the point (x = 0 m, y = 0 m), respectively, in m/s^2, are

(*a*) 0.37, –1.85 (*b*) –1.7, 1.7 (*c*) 1.105, –2.21 (*d*) 1.7, –1.7 (*e*) 0.65, 1.3

Answer (*c*) 1.105, –2.21

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
u=0.65+1.7*x
v=1.3-1.7*y
x=0
y=0
a_x=(0.65+1.7*x)*(1.7) "a_x=u(du/dx)+v(du/dy)"
a_y=(1.3-1.7*y)*(-1.7) "a_y=u(dv/dx)+v(dv/dy)"
```

4-128

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u,v) = (0.65 + 1.7x)\vec{i} + (1.3 - 1.7y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The *x*- and *y*-component of velocity *u* and *v* at the point (*x* = 1 m, *y* = 2 m), respectively, in m/s, are

(*a*) 0.54, −2.31 (*b*) −1.9, 0.75 (*c*) 0.598, −2.21 (*d*) 2.35, −2.1 (*e*) 0.65, 1.3

Answer (*d*) 2.35, −2.1

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
u=0.65+1.7*x
v=1.3-1.7*y
x=1
y=2
```

4-129

The actual path traveled by an individual fluid particle over some period is called a

(*a*) Pathline (*b*) Streamtube (*c*) Streamline (*d*) Streakline (*e*) Timeline

Answer (*a*) Pathline

4-130

The locus of fluid particles that have passed sequentially through a prescribed point in the flow is called a

(*a*) Pathline (*b*) Streamtube (*c*) Streamline (*d*) Streakline (*e*) Timeline

Answer (*d*) Streakline

4-131

A curve that is everywhere tangent to the instantaneous local velocity vector is called a

(*a*) Pathline (*b*) Streamtube (*c*) Streamline (*d*) Streakline (*e*) Timeline

Answer (*c*) Streamline

4-132

An array of arrows indicating the magnitude and direction of a vector property at an instant in time is called a

(*a*) Profiler plot (*b*) Vector plot (*c*) Contour plot (*d*) Velocity plot (*e*) Time plot

Answer (*b*) Vector plot

4-133

The CFD stands for

(*a*) Compressible fluid dynamics (*b*) Compressed flow domain (*c*) Circular flow dynamics
(*d*) Convective fluid dynamics (*e*) Computational fluid dynamics

Answer (*e*) Computational fluid dynamics

4-134

Which one is not a fundamental type of motion or deformation an element may undergo in fluid mechanics?

(*a*) Rotation (*b*) Converging (*c*) Translation (*d*) Linear strain (*e*) Shear strain

Answer (*b*) Converging

4-135

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u, v) = (2.5 - 1.6x)\vec{i} + (0.7 + 1.6y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The linear strain rate in the *x*-direction in s^{-1} is

(*a*) –1.6 (*b*) 0.8 (*c*) 1.6 (*d*) 2.5 (*e*) –0.875

Answer (*a*) –1.6

"u=2.5-1.6x
v=0.7+1.6y
epsilon_xx=du/dx=-1.6"

4-136

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u, v) = (2.5 - 1.6x)\vec{i} + (0.7 + 1.6y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The shear strain rate in s^{-1} is

(*a*) –1.6 (*b*) 1.6 (*c*) 2.5 (*d*) 0.7 (*e*) 0

Answer (*e*) 0

"u=2.5-1.6x
v=0.7+1.6y
epsilon_xy=1/2(du/dy+dv/dx)=1/2(0+0)=0"

4-137

A steady, two-dimensional velocity field is given by

$$\vec{V} = (u, v) = (2.5 - 1.6x)\vec{i} + (0.7 + 0.8y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The volumetric strain rate in s^{-1} is

(*a*) 0 (*b*) 3.2 (*c*) –0.8 (*d*) 0.8 (*e*) –1.6

Answer (*c*) –0.8

"u=2.5-1.6x
v=0.7+0.8y
Volumetric strain rate = epsilon_xx+epsilon_yy
epsilon_xx=du/dx=-1.6
epsilon_yy=dv/dy=0.8
Volumetric strain rate = -1.6 + 0.8 = -0.8"

4-138

If the vorticity in a region of the flow is zero, the flow is

(*a*) Motionless (*b*) Incompressible (*c*) Compressible (*d*) Irrotational (*e*) Rotational

Answer (*d*) Irrotational

4-139

The angular velocity of a fluid particle is 20 rad/s. The vorticity of this fluid particle is

(*a*) 20 rad/s (*b*) 40 rad/s (*c*) 80 rad/s (*d*) 10 rad/s (*e*) 5 rad/s

Answer (*b*) 40 rad/s

4-140

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u,v) = (0.75 + 1.2x)\vec{i} + (2.25 - 1.2y)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The vorticity of this flow is

(*a*) 0 (*b*) $1.2y\vec{k}$ (*c*) $-1.2y\vec{k}$ (*d*) $y\vec{k}$ (*e*) $-1.2xy\vec{k}$

Answer (*a*) 0

"u=0.75+1.2x
v=2.25-1.2y
zeta=(dv/dx-du/dy)k=(0-0)k=0"

4-141

A steady, incompressible, two-dimensional velocity field is given by

$$\vec{V} = (u,v) = (2xy + 1)\vec{i} + (-y^2 - 0.6)\vec{j}$$

where the *x*- and *y*-coordinates are in meters and the magnitude of velocity is in m/s. The angular velocity of this flow is

(*a*) 0 (*b*) $-2y\vec{k}$ (*c*) $2y\vec{k}$ (*d*) $-2x\vec{k}$ (*e*) $-x\vec{k}$

Answer (*e*) $-x\vec{k}$

"u=2xy+1
v=-y^2-0.6
zeta=(dv/dx-du/dy)k=(0-2x)k=-2xk
omega=zeta/2=-xk"

4-142

A cart is moving at a constant absolute velocity $\vec{V}_{\text{cart}} = 5\,\text{km/h}$ to the right. A high-speed jet of water at an absolute velocity of $\vec{V}_{\text{jet}} = 15\,\text{km/h}$ to the right strikes the back of the car. The relative velocity of the water is

(*a*) 0 km/h (*b*) 5 km/h (*c*) 10 km/h (*d*) 15 km/h (*e*) 20 km/h

Answer (*c*) 10 km/h

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

V_cart=5 [km/h]
V_jet=15 [km/h]
V_r=V_jet-V_cart

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

Third Edition

Yunus A. Çengel & John M. Cimbala

McGraw-Hill, 2013

CHAPTER 5
BERNOULLI AND ENERGY EQUATIONS

Conservation of Mass

5-1C

Solution We are to name some conserved and non-conserved quantities.

Analysis **Mass, energy, momentum, and electric charge are conserved**, and **volume and entropy are not conserved** during a process.

Discussion Students may think of other answers that may be equally valid.

5-2C

Solution We are to discuss mass and volume flow rates and their relationship.

Analysis *Mass flow rate* is the **amount of mass flowing through a cross-section per unit time** whereas *volume flow rate* is the **amount of volume flowing through a cross-section per unit time**.

Discussion Mass flow rate has dimensions of mass/time while volume flow rate has dimensions of volume/time.

5-3C

Solution We are to discuss the mass flow rate entering and leaving a control volume.

Analysis The amount of mass or energy entering a control volume **does not have to be equal** to the amount of mass or energy leaving during an unsteady-flow process.

Discussion If the process is steady, however, the two mass flow rates must be equal; otherwise the amount of mass would have to increase or decrease inside the control volume, which would make it unsteady.

5-4C

Solution We are to discuss steady flow through a control volume.

Analysis Flow through a control volume is *steady* when it involves **no changes with time at any specified position**.

Discussion This applies to any variable we might consider – pressure, velocity, density, temperature, etc.

5-5C

Solution We are to discuss whether the flow is steady through a given control volume.

Analysis **No**, a flow with the same volume flow rate at the inlet and the exit is **not necessarily steady** (even if the density is constant – see Discussion). ***To be steady, the mass flow rate through the device must remain constant in time, and no variables can change with time at any specified spatial position***.

Discussion If the question had stated that the two *mass* flow rates were equal, then the answer would still be *not necessarily*. As a counter-example, consider the steadily increasing flow of an incompressible liquid through the device. At any instant in time, the mass flow rate in must equal the mass flow rate out since there is nowhere else for the liquid to go. However, the mass flow rate itself is changing with time, and hence the problem is unsteady. Can you think of another counter-example?

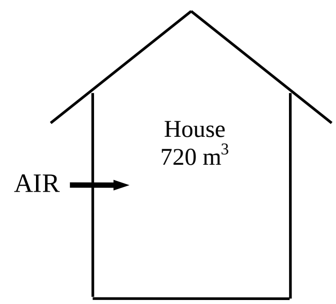
5-6

Solution A house is to be cooled by drawing in cool night time air continuously. For a specified air exchange rate, the required flow rate of the fan and the average discharge speed of air are to be determined.

Assumptions Flow through the fan is steady.

Analysis The volume of the house is given to be $V_{\text{house}} = 720\ \text{m}^3$. Noting that this volume of air is to be replaced every $\Delta t = 20\ \text{min}$, the required volume flow rate of air is

$$\dot{V} = \frac{V_{\text{room}}}{\Delta t} = \frac{720\ \text{m}^3}{20\ \text{min}}\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = \mathbf{0.60\ m^3/s}$$



For the given fan diameter, the average discharge speed is determined to be

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.60\ \text{m}^3/\text{s}}{\pi (0.5\ \text{m})^2/4} = \mathbf{3.06\ m/s}$$

Discussion Note that the air velocity and thus the noise level is low because of the large fan diameter.

5-7E

Solution A garden hose is used to fill a water bucket. The volume and mass flow rates of water, the filling time, and the discharge velocity are to be determined.

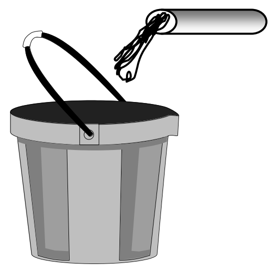
Assumptions **1** Water is an incompressible substance. **2** Flow through the hose is steady. **3** There is no waste of water by splashing.

Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis (*a*) The volume and mass flow rates of water are

$$\dot{V} = AV = (\pi D^2/4)V = [\pi(1/12\text{ ft})^2/4](8\text{ ft/s}) = 0.04363\text{ ft}^3\text{/s} \cong \mathbf{0.0436\ ft^3/s}$$

$$\dot{m} = \rho\dot{V} = (62.4\text{ lbm/ft}^3)(0.04363\text{ ft}^3\text{/s}) = \mathbf{2.72\ lbm/s}$$



(*b*) The time it takes to fill a 20-gallon bucket is

$$\Delta t = \frac{V}{\dot{V}} = \frac{20\text{ gal}}{0.04363\text{ ft}^3\text{/s}}\left(\frac{1\text{ ft}^3}{7.4804\text{ gal}}\right) = \mathbf{61.3\ s}$$

(*c*) The average discharge velocity of water at the nozzle exit is

$$V_e = \frac{\dot{V}}{A_e} = \frac{\dot{V}}{\pi D_e^2/4} = \frac{0.04363\text{ ft}^3\text{/s}}{[\pi(0.5/12\text{ ft})^2/4]} = \mathbf{32\ ft/s}$$

Discussion Note that for a given flow rate, the average velocity is inversely proportional to the square of the velocity. Therefore, when the diameter is reduced by half, the velocity quadruples.

5-8E

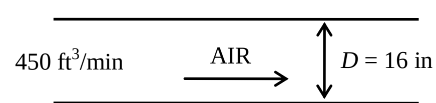
Solution The ducts of an air-conditioning system pass through an open area. The inlet velocity and the mass flow rate of air are to be determined.

Assumptions Flow through the air conditioning duct is steady.

Properties The density of air is given to be 0.082 lbm/ft^3 at the inlet.

Analysis The inlet velocity of air and the mass flow rate through the duct are

$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}_1}{\pi D^2/4} = \frac{450\text{ ft}^3\text{/min}}{\pi(16/12\text{ ft})^2/4} = \mathbf{322\ ft/min} = \mathbf{26.9\ ft/s}$$



$$\dot{m} = \rho_1\dot{V}_1 = (0.082\text{ lbm/ft}^3)(450\text{ ft}^3\text{/min}) = 36.9\text{ lbm/min} = \mathbf{0.615\ lbm/s}$$

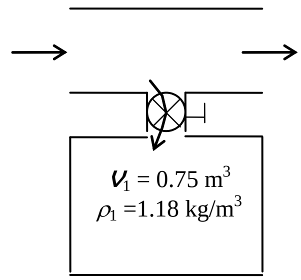
Discussion The mass flow rate though a duct must remain constant in steady flow; however, the volume flow rate varies since the density varies with the temperature and pressure in the duct.

5-9

Solution A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until the density rises to a specified level. The mass of air that entered the tank is to be determined.

Properties The density of air is given to be 1.18 kg/m^3 at the beginning, and 4.95 kg/m^3 at the end.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. The mass balance for this system can be expressed as



Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \;\rightarrow\; m_i = m_2 - m_1 = \rho_2 \boldsymbol{V} - \rho_1 \boldsymbol{V}$

Substituting, $m_i = (\rho_2 - \rho_1)\boldsymbol{V} = [(4.95 - 1.18)\text{ kg/m}^3](0.75\text{ m}^3) = \mathbf{2.83\ kg}$

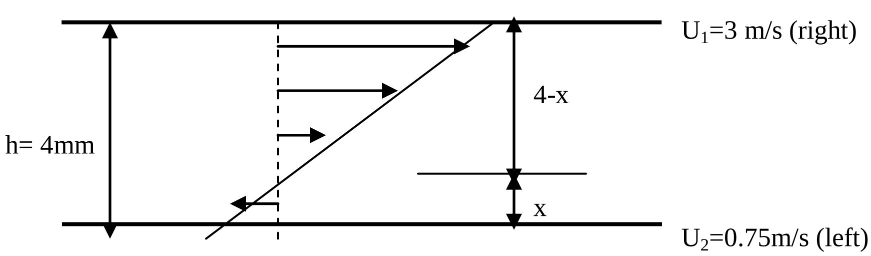
Therefore, **2.83 kg of mass entered the tank**.

Discussion Tank temperature and pressure do not enter into the calculations.

5-10

Solution A Newtonian fluid flows between two parallel plates. The upper plate moves to right and bottom one moves to the left. The net flow rate is to be determined.

Analysis From the similarity of the triangles we write



$$\frac{4-x}{x} = \frac{3}{0.75}$$

$3x = (4-x)(0.75)$

$3x = 3 - 0.75x$

$x = 0.8$ mm

$y = 4 - x = 3.2$ mm

$$\dot{V}_{net} = (3.2 \times 10^{-3})(5\times10^{-2})\frac{3}{2} - (0.8\times10^{-3})(5\times10^{-2})\frac{0.75}{2}$$

$$\dot{V}_{net} = 24\times10^{-6} - 15\times10^{-6} = \mathbf{9\times10^{-6}\ cm^3/s}$$

5-11

Solution Water is pumped out pf a fully-filled semi-circular cross section tank. The time needed to drop the water level to a specified value is to be determined in terms of given parameters.

Analysis From the conservation of mass, we write

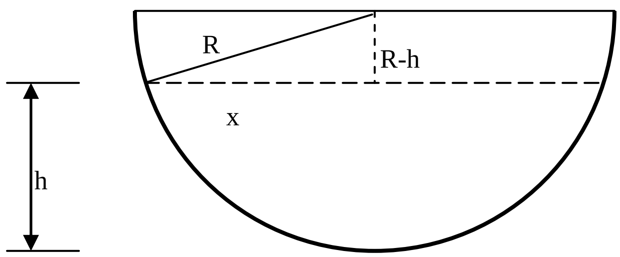
$$Qdt = -A_t dh$$

or

$$dt = -\frac{A_T}{Q}dh = -\frac{\pi x^2}{Kh^2}dh = -\frac{\pi\left[R^2 - (R-h)^2\right]}{Kh^2}dh = -\frac{\pi}{K}\frac{2R-h}{h}dh$$

Integrating from h_1=R to h_2=0, we get

$$t = \frac{\pi}{K}\left(h - 2R\ln(h)\right)_R^H = -\frac{\pi}{K}\left[(R-H) + \ln\left(\frac{H}{R}\right)^{2R}\right]$$



5-12

Solution A desktop computer is to be cooled by a fan at a high elevation where the air density is low. The mass flow rate of air through the fan and the diameter of the casing for a given velocity are to be determined.

Assumptions Flow through the fan is steady.

Properties The density of air at a high elevation is given to be 0.7 kg/m^3.

Analysis The mass flow rate of air is

$$\dot{m}_{\text{air}} = \rho \dot{V}_{\text{air}} = (0.700\ \text{kg/m}^3)(0.400\ \text{m}^3/\text{min}) = 0.280\ \text{kg/min} = \mathbf{0.00467\ kg/s}$$

If the mean velocity is 110 m/min, the diameter of the casing is

$$\dot{V} = AV = \frac{\pi D^2}{4}V \quad \rightarrow \quad D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.400\ \text{m}^3/\text{min})}{\pi(110\ \text{m/min})}} = \mathbf{0.068 m}$$

Therefore, the diameter of the casing must be **at least 5.69 cm** to ensure that the mean velocity does not exceed 110 m/min.

Discussion This problem shows that engineering systems are sized to satisfy given imposed constraints.

5-13

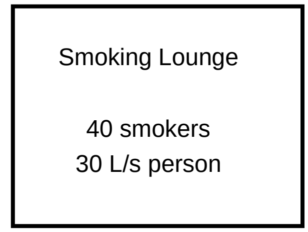
Solution A smoking lounge that can accommodate 40 smokers is considered. The required minimum flow rate of air that needs to be supplied to the lounge and the diameter of the duct are to be determined.

Assumptions Infiltration of air into the smoking lounge is negligible.

Properties The minimum fresh air requirements for a smoking lounge is given to be 30 L/s per person.

Analysis The required minimum flow rate of air that needs to be supplied to the lounge is determined directly from

$$\dot{\boldsymbol{V}}_{\text{air}} = \dot{\boldsymbol{V}}_{\text{air per person}}(\text{No. of persons})$$
$$= (30\ \text{L/s}\cdot\text{person})(40\ \text{persons}) = 1200\ \text{L/s} = \mathbf{1.2\ m^3/s}$$



The volume flow rate of fresh air can be expressed as

$$\dot{\boldsymbol{V}} = VA = V(\pi D^2/4)$$

Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V}} = \sqrt{\frac{4(1.2\ \text{m}^3/\text{s})}{\pi(8\ \text{m/s})}} = \mathbf{0.437\ m}$$

Therefore, the diameter of the fresh air duct should be **at least 43.7 cm** if the velocity of air is not to exceed 8 m/s.

Discussion Fresh air requirements in buildings must be taken seriously to avoid health problems.

5-14

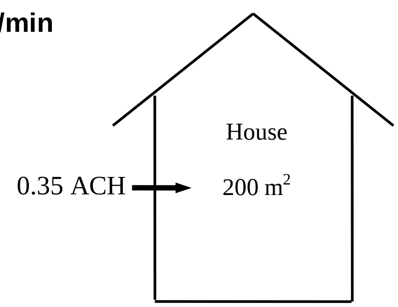
Solution The minimum fresh air requirements of a residential building is specified to be 0.35 air changes per hour. The size of the fan that needs to be installed and the diameter of the duct are to be determined.

Analysis The volume of the building and the required minimum volume flow rate of fresh air are

$$\boldsymbol{V}_{\text{room}} = (2.7\ \text{m})(200\ \text{m}^2) = 540\ \text{m}^3$$
$$\dot{\boldsymbol{V}} = \boldsymbol{V}_{\text{room}} \times \text{ACH} = (540\ \text{m}^3)(0.35/\text{h}) = 189\ \text{m}^3/\text{h} = 189{,}000\ \text{L/h} = \mathbf{3150\ L/min}$$

The volume flow rate of fresh air can be expressed as

$$\dot{\boldsymbol{V}} = VA = V(\pi D^2/4)$$



Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V}} = \sqrt{\frac{4(189/3600\ \text{m}^3/\text{s})}{\pi(5\ \text{m/s})}} = \mathbf{0.116\ m}$$

Therefore, the diameter of the fresh air duct should be **at least 11.6 cm** if the velocity of air is not to exceed 5 m/s.

Discussion Fresh air requirements in buildings must be taken seriously to avoid health problems.

5-15

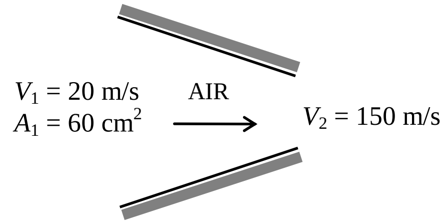
Solution Air is accelerated in a nozzle. The mass flow rate and the exit area of the nozzle are to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 2.21 kg/m^3 at the inlet, and 0.762 kg/m^3 at the exit.

Analysis (*a*) The mass flow rate of air is determined from the inlet conditions to be

$$\dot{m} = \rho_1 A_1 V_1 = (2.21\ \text{kg/m}^3)(0.006\ \text{m}^2)(20\ \text{m/s}) = 0.2652\ \text{kg/s} \cong \mathbf{0.265\ kg/s}$$



(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

Then the exit area of the nozzle is determined to be

$$\dot{m} = \rho_2 A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.2652\ \text{kg/s}}{(0.762\ \text{kg/m}^3)(150\ \text{m/s})} = 0.00232\ \text{m}^2 = \mathbf{23.2\ cm^2}$$

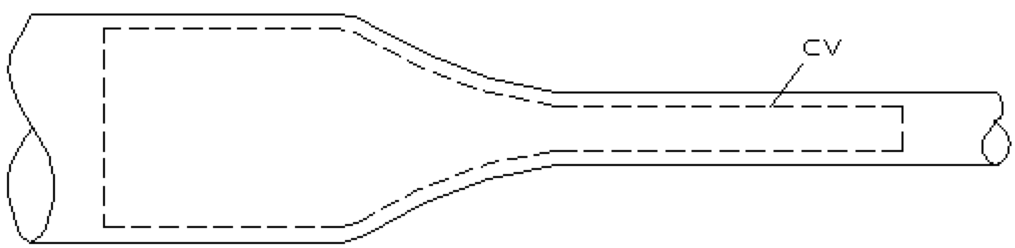
Discussion Since this is a compressible flow, we must equate mass flow rates, not volume flow rates.

5-16

Solution Air flows in a varying crosss section pipe. The speed at a specified section is to be determined.

Assumptions Flow through the pipe is steady.

Analysis



Applying conservation of mass for the cv shown,

$$\frac{\partial}{\partial t}\int_{cv} \rho . d\forall + \int_{cs} \rho . \vec{V} . \vec{n} . dA = 0 \quad , \quad -\rho . V_1 . A_1 + \rho_2 . A_2 . V_2 = 0 \qquad \overline{V_1} = \frac{\rho_2 . A_2 . V_2}{\rho_1 . A_1} \ ,$$

$$\rho = \frac{P_{abs}}{RT} \quad , \quad \rho_1 = \frac{P_{1(abs)}}{RT_1} \quad , \quad \rho_2 = \frac{P_{2(abs)}}{RT_2}$$

$$\frac{\rho_2}{\rho_1} = \frac{P_{2(abs)}}{P_{1(abs)}} \qquad , \qquad \frac{A_2}{A_1} = \frac{\frac{\pi d^2}{4}}{\frac{\pi D^2}{4}} = \left(\frac{d}{D}\right)^2$$

$$\overline{V_1} = \frac{110}{150} . \left(\frac{1}{3}\right)^2 .30 = \mathbf{2.44\ m/s}$$

5-17

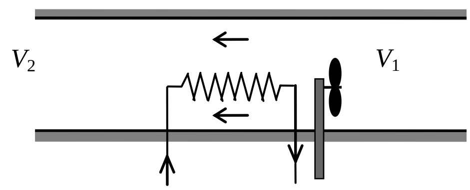
Solution Air is expanded and is accelerated as it is heated by a hair dryer of constant diameter. The percent increase in the velocity of air as it flows through the drier is to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 1.20 kg/m^3 at the inlet, and 1.05 kg/m^3 at the exit.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\dot{m}_1 = \dot{m}_2$$
$$\rho_1 A V_1 = \rho_2 A V_2$$
$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{1.20 \text{ kg/m}^3}{1.05 \text{ kg/m}^3} = 1.14 \qquad \text{(or, an increase of } \mathbf{14\%})$$



Therefore, the air velocity increases **14%** as it flows through the hair drier.

Discussion It makes sense that the velocity *increases* since the density *decreases*, but the mass flow rate is constant.

Mechanical Energy and Efficiency

5-18C

Solution We are to define and discuss turbine, generator, and turbine-generator efficiency.

Analysis Turbine efficiency, generator efficiency, and *combined turbine-generator efficiency* are defined as follows:

$$\boxed{\eta_{\text{turbine}} = \frac{\text{Mechanical energy output}}{\text{Mechanical energy extracted from the fluid}} = \frac{\dot{W}_{\text{shaft,out}}}{|\Delta \dot{E}_{\text{mech,fluid}}|}}$$

$$\boxed{\eta_{\text{generator}} = \frac{\text{Electrical power output}}{\text{Mechanical power input}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft,in}}}}$$

$$\boxed{\eta_{\text{turbine-gen}} = \eta_{\text{turbine}}\eta_{\text{generaor}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{E}_{\text{mech,in}} - \dot{E}_{\text{mech,out}}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta \dot{E}_{\text{mech,fluid}}|}}$$

Discussion Most turbines are connected directly to a generator, so the combined efficiency is a useful parameter.

5-19C

Solution We are to define and discuss mechanical efficiency.

Analysis *Mechanical efficiency* is defined as **the ratio of the mechanical energy output to the mechanical energy input**. A mechanical efficiency of 100% for a hydraulic turbine means that the entire mechanical energy of the fluid is converted to mechanical (shaft) work.

Discussion No real fluid machine is 100% efficient, due to frictional losses, etc. – the second law of thermodynamics.

5-20C

Solution We are to define and discuss pump-motor efficiency.

Analysis The *combined pump-motor efficiency* of a pump/motor system is defined as the ratio of the increase in the mechanical energy of the fluid to the electrical power consumption of the motor,

$$\boxed{\eta_{\text{pump-motor}} = \eta_{\text{pump}}\eta_{\text{motor}} = \frac{\dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}}}{\dot{W}_{\text{elect,in}}} = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{\dot{W}_{\text{pump}}}{\dot{W}_{\text{elect,in}}}}$$

The combined pump-motor efficiency cannot be greater than either of the pump or motor efficiency since both pump and motor efficiencies are less than 1, and the product of two numbers that are less than one is less than either of the numbers.

Discussion Since many pumps are supplied with an integrated motor, pump-motor efficiency is a useful parameter.

5-21C

Solution We are to discuss mechanical energy and how it differs from thermal energy.

Analysis *Mechanical energy* is the **form of energy that can be converted to mechanical work completely and directly by a mechanical device** such as a propeller. It differs from thermal energy in that **thermal energy cannot be converted to work directly and completely**. The forms of mechanical energy of a fluid stream are kinetic, potential, and flow energies.

Discussion It would be nice if we could convert thermal energy completely into work. However, this would violate the second law of thermodynamics.

5-22

Solution Wind is blowing steadily at a certain velocity. The mechanical energy of air per unit mass, the power generation potential, and the actual electric power generation are to be determined.

Assumptions **1** The wind is blowing steadily at a constant uniform velocity. **2** The efficiency of the wind turbine is independent of the wind speed.

Properties The density of air is given to be $\rho = 1.25$ kg/m^3.

Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(8\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.032\text{ kJ/kg}$$

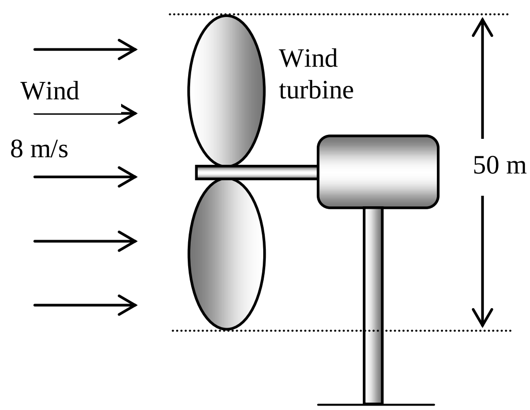
$$\dot{m} = \rho VA = \rho V\frac{\pi D^2}{4} = (1.25\text{ kg/m}^3)(8\text{ m/s})\frac{\pi(50\text{ m})^2}{4} = 19{,}635\text{ kg/s}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (19{,}635\text{ kg/s})(0.032\text{ kJ/kg}) = \mathbf{628\ kW}$$

The actual electric power generation is determined by multiplying the power generation potential by the efficiency,

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}}\dot{W}_{\max} = (0.30)(628\text{ kW}) = \mathbf{188\ kW}$$

Therefore, 283 kW of actual power can be generated by this wind turbine at the stated conditions.



Discussion The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.

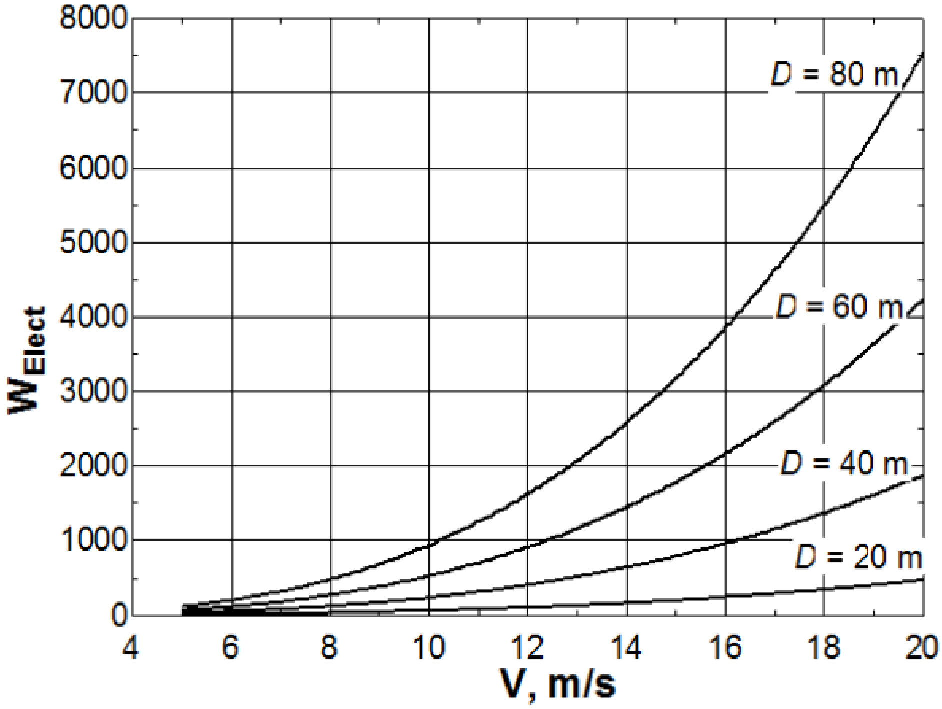
5-23

Solution The previous problem is reconsidered. The effect of wind velocity and the blade span diameter on wind power generation as the velocity varies from 5 m/s to 20 m/s in increments of 5 m/s, and the diameter varies from 20 m to 80 m in increments of 20 m is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
D1=20 "m"
D2=40 "m"
D3=60 "m"
D4=80 "m"
Eta=0.30
rho=1.25 "kg/m3"
m1_dot=rho*V*(pi*D1^2/4); W1_Elect=Eta*m1_dot*(V^2/2)/1000 "kW"
m2_dot=rho*V*(pi*D2^2/4); W2_Elect=Eta*m2_dot*(V^2/2)/1000 "kW"
m3_dot=rho*V*(pi*D3^2/4); W3_Elect=Eta*m3_dot*(V^2/2)/1000 "kW"
m4_dot=rho*V*(pi*D4^2/4); W4_Elect=Eta*m4_dot*(V^2/2)/1000 "kW"
```

D, m	V, m/s	m, kg/s	W_{elect}, kW
20	5	1,963	7
	10	3,927	59
	15	5,890	199
	20	7,854	471
40	5	7,854	29
	10	15,708	236
	15	23,562	795
	20	31,416	1885
60	5	17,671	66
	10	35,343	530
	15	53,014	1789
	20	70,686	4241
80	5	31,416	118
	10	62,832	942
	15	94,248	3181
	20	125,664	7540



Discussion Wind turbine power output is obviously nonlinear with respect to both velocity and diameter.

5-24E

Solution A differential thermocouple indicates that the temperature of water rises a certain amount as it flows through a pump at a specified rate. The mechanical efficiency of the pump is to be determined.

Assumptions **1** The pump is adiabatic so that there is no heat transfer with the surroundings, and the temperature rise of water is completely due to frictional heating. **2** Water is an incompressible substance.

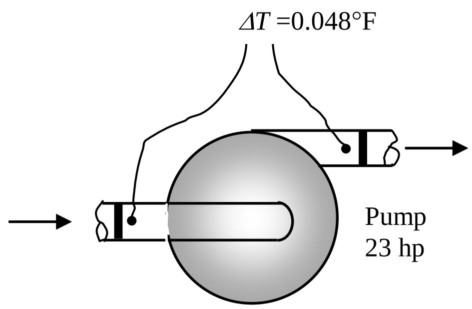
Properties We take the density of water to be ρ = 62.4 lbm/ft^3 and its specific heat to be c = 1.0 Btu/lbm·°F.

Analysis The increase in the temperature of water is due to the conversion of mechanical energy to thermal energy, and the amount of mechanical energy converted to thermal energy is equal to the increase in the internal energy of water,

$$\dot{m} = \rho\dot{V} = (62.4\ \text{lbm/ft}^3)(1.5\ \text{ft}^3/\text{s}) = 93.6\ \text{lbm/s}$$

$$\dot{E}_{\text{mech, loss}} = \Delta\dot{U} = \dot{m}c\Delta T$$

$$= (93.6\ \text{lbm/s})(1.0\ \text{Btu/lbm}\cdot{}^\circ\text{F})(0.048^\circ\text{F})\left(\frac{1\ \text{hp}}{0.7068\ \text{Btu/s}}\right) = 6.36\ \text{hp}$$



The mechanical efficiency of the pump is determined from the general definition of mechanical efficiency,

$$\eta_{\text{pump}} = 1 - \frac{\dot{E}_{\text{mech,loss}}}{\dot{W}_{\text{mech, in}}} = 1 - \frac{6.36\ \text{hp}}{23\ \text{hp}} = 0.724 \quad \text{or} \quad \mathbf{72.4\%}$$

Discussion Note that despite the conversion of more than one-third of the mechanical power input into thermal energy, the temperature of water rises by only a small fraction of a degree. Therefore, the temperature rise of a fluid due to frictional heating is usually negligible in heat transfer analysis.

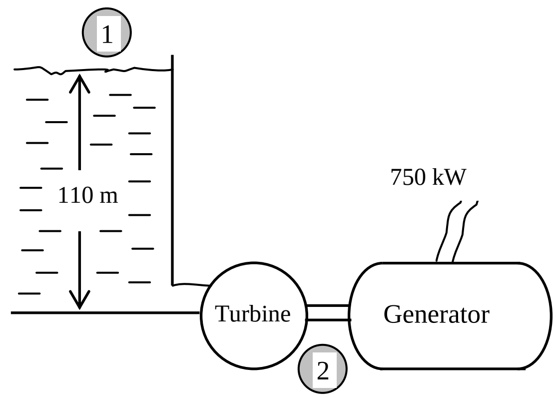
5-25

Solution A hydraulic turbine-generator is generating electricity from the water of a large reservoir. The combined turbine-generator efficiency and the turbine efficiency are to be determined.

Assumptions **1** The elevation of the reservoir remains constant. **2** The mechanical energy of water at the turbine exit is negligible.

Analysis We take the free surface of the reservoir to be point 1 and the turbine exit to be point 2. We also take the turbine exit as the reference level ($z_2 = 0$), and thus the potential energy at points 1 and 2 are $\mathrm{pe}_1 = gz_1$ and $\mathrm{pe}_2 = 0$. The flow energy P/ρ at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$). Further, the kinetic energy at both points is zero ($\mathrm{ke}_1 = \mathrm{ke}_2 = 0$) since the water at point 1 is essentially motionless, and the kinetic energy of water at turbine exit is assumed to be negligible. The potential energy of water at point 1 is

$$pe_1 = gz_1 = (9.81\ \mathrm{m/s^2})(110\ \mathrm{m})\left(\frac{1\ \mathrm{kJ/kg}}{1000\ \mathrm{m^2/s^2}}\right) = 1.079\ \mathrm{kJ/kg}$$



Then the rate at which the mechanical energy of the fluid is supplied to the turbine become

$$\begin{aligned}\left|\Delta\dot{E}_{\mathrm{mech,fluid}}\right| &= \dot{m}(e_{\mathrm{mech,in}} - e_{\mathrm{mech,out}}) = \dot{m}(pe_1 - 0) = \dot{m}pe_1\\ &= (900\ \mathrm{kg/s})(1.079\ \mathrm{kJ/kg})\\ &= 971.2\ \mathrm{kW}\end{aligned}$$

The combined turbine-generator and the turbine efficiency are determined from their definitions,

$$\eta_{\mathrm{turbine\text{-}gen}} = \frac{\dot{W}_{\mathrm{elect,out}}}{|\Delta\dot{E}_{\mathrm{mech,fluid}}|} = \frac{750\ \mathrm{kW}}{971.2\ \mathrm{kW}} = 0.772 \quad \text{or} \quad \mathbf{77.2\%}$$

$$\eta_{\mathrm{turbine}} = \frac{\dot{W}_{\mathrm{shaft,out}}}{|\Delta\dot{E}_{\mathrm{mech,fluid}}|} = \frac{800\ \mathrm{kW}}{971.2\ \mathrm{kW}} = 0.824 \quad \text{or} \quad \mathbf{82.4\%}$$

Therefore, the reservoir supplies 971.2 kW of mechanical energy to the turbine, which converts 800 kW of it to shaft work that drives the generator, which generates 750 kW of electric power.

Discussion This problem can also be solved by taking point 1 to be at the turbine inlet, and using flow energy instead of potential energy. It would give the same result since the flow energy at the turbine inlet is equal to the potential energy at the free surface of the reservoir.

5-26

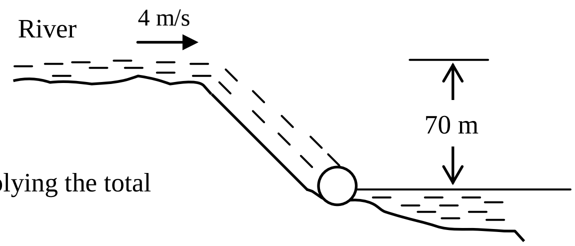
Solution A river is flowing at a specified velocity, flow rate, and elevation. The total mechanical energy of the river water per unit mass, and the power generation potential of the entire river are to be determined.

Assumptions **1** The elevation given is the elevation of the free surface of the river. **2** The velocity given is the average velocity. **3** The mechanical energy of water at the turbine exit is negligible.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis Noting that the sum of the flow energy and the potential energy is constant for a given fluid body, we can take the elevation of the entire river water to be the elevation of the free surface, and ignore the flow energy. Then the total mechanical energy of the river water per unit mass becomes

$$\begin{aligned} e_{\text{mech}} &= pe + ke = gh + \frac{V^2}{2} \\ &= \left((9.81\,\text{m/s}^2)(70\,\text{m}) + \frac{(4\,\text{m/s})^2}{2} \right) \left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2} \right) \\ &= 0.695\,\text{kJ/kg} \end{aligned}$$



The power generation potential of the river water is obtained by multiplying the total mechanical energy by the mass flow rate,

$$\dot{m} = \rho \dot{V} = (1000\,\text{kg/m}^3)(500\,\text{m}^3/\text{s}) = 500{,}000\,\text{kg/s}$$

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m} e_{\text{mech}} = (500{,}000\,\text{kg/s})(0.695\,\text{kg/s}) = 347{,}350\,\text{kW} \cong \mathbf{347\,MW}$$

Therefore, 347 MW of power can be generated from this river as it discharges into the lake if its power potential can be recovered completely.

Discussion Note that the kinetic energy of water is negligible compared to the potential energy, and it can be ignored in the analysis. Also, the power output of an actual turbine will be less than 347 MW because of losses and inefficiencies.

5-27

Solution Water is pumped from a lake to a storage tank at a specified rate. The overall efficiency of the pump-motor unit and the pressure difference between the inlet and the exit of the pump are to be determined.

Assumptions **1** The elevations of the tank and the lake remain constant. **2** Frictional losses in the pipes are negligible. **3** The changes in kinetic energy are negligible. **4** The elevation difference across the pump is negligible.

Properties We take the density of water to be ρ = 1000 kg/m^3.

Analysis (*a*) We take the free surface of the lake to be point 1 and the free surfaces of the storage tank to be point 2. We also take the lake surface as the reference level (z_1 = 0), and thus the potential energy at points 1 and 2 are pe_1 = 0 and $pe_2 = gz_2$. The flow energy at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$). Further, the kinetic energy at both points is zero ($ke_1 = ke_2 = 0$) since the water at both locations is essentially stationary. The mass flow rate of water and its potential energy at point 2 are

$$\dot{m} = \rho\dot{V} = (1000\ \text{kg/m}^3)(0.070\ \text{m}^3/\text{s}) = 70\ \text{kg/s}$$

$$pe_1 = gz_1 = (9.81\ \text{m/s}^2)(18\ \text{m})\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 0.177\ \text{kJ/kg}$$

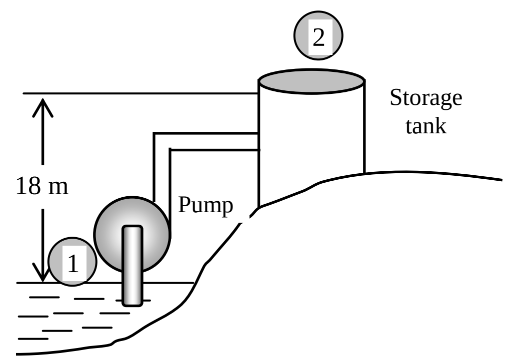
Then the rate of increase of the mechanical energy of water becomes

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}(pe_2 - 0) = \dot{m}pe_2 = (70\ \text{kg/s})(0.177\ \text{kJ/kg}) = 12.4\ \text{kW}$$

The overall efficiency of the combined pump-motor unit is determined from its definition,

$$\eta_{\text{pump-motor}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{12.4\ \text{kW}}{20.4\ \text{kW}} = 0.606 \quad \text{or} \quad \mathbf{60.6\%}$$

(*b*) Now we consider the pump. The change in the mechanical energy of water as it flows through the pump consists of the change in the flow energy only since the elevation difference across the pump and the change in the kinetic energy are negligible. Also, this change must be equal to the useful mechanical energy supplied by the pump, which is 12.4 kW:



$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}\frac{P_2 - P_1}{\rho} = \dot{V}\Delta P$$

Solving for ΔP and substituting,

$$\Delta P = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{V}} = \frac{12.4\ \text{kJ/s}}{0.070\ \text{m}^3/\text{s}}\left(\frac{1\ \text{kPa}\cdot\text{m}^3}{1\ \text{kJ}}\right) = \mathbf{177\ kPa}$$

Therefore, the pump must boost the pressure of water by 177 kPa in order to raise its elevation by 18 m.

Discussion Note that only two-thirds of the electric energy consumed by the pump-motor is converted to the mechanical energy of water; the remaining one-third is wasted because of the inefficiencies of the pump and the motor.

Bernoulli Equation

5-28C

Solution We are to define stagnation pressure and discuss how it can be measured.

Analysis The **sum of the static and dynamic pressures** is called the *stagnation pressure*, and it is expressed as $\boxed{P_{\text{stag}} = P + \rho V^2 / 2}$. The stagnation pressure can be measured by **a Pitot tube whose inlet is normal to the flow**.

Discussion Stagnation pressure, as its name implies, is the pressure obtained when a flowing fluid is brought to rest isentropically, at a so-called *stagnation point*.

5-29C

Solution We are to express the Bernoulli equation in three different ways.

Analysis The Bernoulli equation is expressed in three different ways as follows:

(a) In terms of energies: $\boxed{\frac{P}{\rho} + \frac{V^2}{2} + gz = \text{constant}}$

(b) In terms of pressures: $\boxed{P + \rho \frac{V^2}{2} + \rho gz = \text{constant}}$

(c) in terms of heads: $\boxed{\frac{P}{\rho g} + \frac{V^2}{2g} + z = H = \text{constant}}$

Discussion You could, of course, express it in other ways, but these three are the most useful.

5-30C

Solution We are to discuss the three major assumptions used in the derivation of the Bernoulli equation.

Analysis The three major assumptions used in the derivation of the Bernoulli equation are that the flow is **steady**, **there is negligible frictional effects**, and the flow is **incompressible**.

Discussion If any one of these assumptions is not valid, the Bernoulli equation should not be used. Unfortunately, many people use it anyway, leading to errors.

5-31C

Solution We are to define and discuss static, dynamic, and hydrostatic pressure.

Analysis *Static pressure P* is the **actual pressure of the fluid**. *Dynamic pressure* $\rho V^2/2$ is the **pressure rise when the fluid in motion is brought to a stop isentropically**. *Hydrostatic pressure* ρgz is not pressure in a real sense since its value depends on the reference level selected, and it **accounts for the effects of fluid weight on pressure**. The sum of static, dynamic, and hydrostatic pressures is constant when flow is steady and incompressible, and when frictional effects are negligible.

Discussion The incompressible Bernoulli equation states that the sum of these three pressures is constant along a streamline; this approximation is valid only for steady and incompressible flow with negligible frictional effects.

5-32C

Solution We are to define streamwise acceleration and discuss how it differs from normal acceleration.

Analysis The **acceleration of a fluid particle along a streamline** is called *streamwise acceleration*, and it is due to a change in speed along a streamline. *Normal acceleration* (or centrifugal acceleration), on the other hand, is the **acceleration of a fluid particle in the direction normal to the streamline**, and it is due to a change in direction.

Discussion In a general fluid flow problem, both streamwise and normal acceleration are present.

5-33C

Solution We are to define and discuss pressure head, velocity head, and elevation head.

Analysis The *pressure head* $P/\rho g$ is the **height of a fluid column that produces the static pressure *P***. The *velocity head* $V^2/2$ is the **elevation needed for a fluid to reach the velocity *V* during frictionless free fall**. The *elevation head z* is the **height of a fluid relative to a reference level**.

Discussion It is often convenient in fluid mechanics to work with *head* – pressure expressed as an equivalent column height of fluid.

5-34C

Solution We are to explain how and why a siphon works, and its limitations.

Analysis Siphoning works because of the elevation and thus pressure difference between the inlet and exit of a tube. The pressure at the tube exit and at the free surface of a liquid is the atmospheric pressure. When the tube exit is below the free surface of the liquid, the elevation head difference drives the flow through the tube. At sea level, 1 atm pressure can support about 10.3 m of cold water (cold water has a low vapor pressure). Therefore, siphoning cold water over a 7 m wall is **theoretically feasible**.

Discussion In actual practice, siphoning is also limited by frictional effects in the tube, and by cavitation.

5-35C

Solution We are to discuss the hydraulic grade line in open-channel flow and at the outlet of a pipe.

Analysis For *open-channel flow*, the hydraulic grade line (HGL) **coincides with the free surface of the liquid**. At the exit of a pipe discharging to the atmosphere, HGL **coincides with the elevation of the pipe outlet**.

Discussion We are assuming incompressible flow, and the pressure at the pipe outlet is atmospheric.

5-36C

Solution We are to discuss the effect of liquid density on the operation of a siphon.

Analysis The lower density liquid can go over a higher wall, provided that cavitation pressure is not reached. Therefore, oil may be able to go over a higher wall than water.

Discussion However, frictional losses in the flow of oil in a pipe or tube are much greater than those of water since the viscosity of oil is much greater than that of water. When frictional losses are considered, the water may actually be able to be siphoned over a higher wall than the oil, depending on the tube diameter and length, etc.

5-37C

Solution We are to define hydraulic grade line and compare it to energy grade line.

Analysis **The curve that represents the sum of the static pressure and the elevation heads**, $P/\rho g + z$, is called the *hydraulic grade line* or HGL. The curve that represents the total head of the fluid, $P/\rho g + V^2/2g + z$, is called the *energy line* or EGL. Thus, in comparison, the energy grade line contains an extra kinetic-energy-type term. **For stationary bodies such as reservoirs or lakes, the EL and HGL coincide with the free surface of the liquid**.

Discussion The hydraulic grade line can rise or fall along flow in a pipe or duct as the cross-sectional area increases or decreases, whereas the energy grade line *always* decreases unless energy is added to the fluid (like with a pump).

5-38C

Solution We are to discuss and compare the operation of a manometer.

Analysis As the duct converges to a smaller cross-sectional area, the velocity increases. By Bernoulli's equation, the pressure therefore decreases. Thus **Manometer A** is correct since the pressure on the right side of the manometer is obviously smaller. According to the Bernoulli approximation, the fluid levels in the manometer are independent of the flow direction, and reversing the flow direction would have no effect on the manometer levels. **Manometer A is still correct if the flow is reversed**.

Discussion In reality, it is hard for a fluid to expand without the flow separating from the walls. Thus, reverse flow with such a sharp expansion would not produce as much of a pressure rise as that predicted by the Bernoulli approximation.

5-39C

Solution We are to discuss and compare two different types of manometer arrangements in a flow.

Analysis Arrangement 1 consists of a Pitot probe that measures the stagnation pressure at the pipe centerline, along with a static pressure tap that measures static pressure at the bottom of the pipe. Arrangement 2 is a Pitot-static probe that measures both stagnation pressure and static pressure at *nearly the same location* at the pipe centerline. Because of this, **arrangement 2 is more accurate**. However, it turns out that static pressure in a pipe varies with elevation across the pipe cross section in much the same way as in hydrostatics. Therefore, arrangement 1 is also very accurate, and the elevation difference between the Pitot probe and the static pressure tap is nearly compensated by the change in hydrostatic pressure. Since elevation changes are not important in either arrangement, there is **no change in our analysis when the water is replaced by air**.

Discussion Ignoring the effects of gravity, the pressure at the centerline of a turbulent pipe flow is actually somewhat smaller than that at the wall due to the turbulent eddies in the flow, but this effect is small.

5-40C

Solution We are to discuss the maximum rise of a jet of water from a tank.

Analysis With no losses and a 100% efficient nozzle, the water stream could reach to the water level in the tank, or 20 meters. In reality, friction losses in the hose, nozzle inefficiencies, orifice losses, and air drag would prevent attainment of the maximum theoretical height.

Discussion In fact, the actual maximum obtainable height is much smaller than this ideal theoretical limit.

5-41C

Solution We are to compare siphoning at sea level and on a mountain.

Analysis At sea level, a person can theoretically siphon water over a wall as high as 10.3 m. At the top of a high mountain where the pressure is about half of the atmospheric pressure at sea level, a person can theoretically siphon water over a wall that is only half as high. **An atmospheric pressure of 58.5 kPa is insufficient to support a 8.5 meter high siphon.**

Discussion In actual practice, siphoning is also limited by frictional effects in the tube, and by cavitation.

5-42

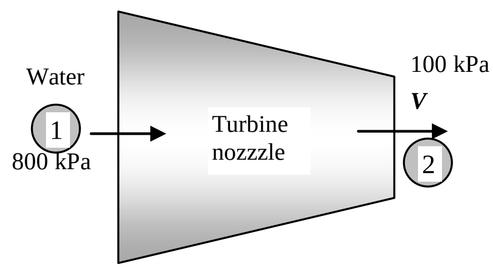
Solution In a power plant, water enters the nozzles of a hydraulic turbine at a specified pressure. The maximum velocity water can be accelerated to by the nozzles is to be determined.

Assumptions **1**The flow of water is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Water enters the nozzle with a low velocity.

Properties We take the density of water to be ρ = 1000 kg/m^3.

Analysis We take points 1 and 2 at the inlet and exit of the nozzle, respectively. Noting that $V_1 \cong 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}=\frac{P_{atm}}{\rho g}+\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{\frac{2(P_1-P_{atm})}{\rho}}$$



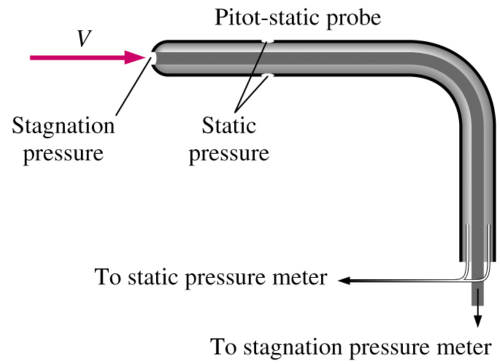
Substituting the given values, the nozzle exit velocity is determined to be

$$V_1=\sqrt{\frac{2(800-100)\text{ kPa}}{1000\text{ kg/m}^3}\left(\frac{1000\text{ N/m}^2}{1\text{ kPa}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}=\mathbf{37.4\,m/s}$$

Discussion This is the maximum nozzle exit velocity, and the actual velocity will be less because of friction between water and the walls of the nozzle.

5-43

Solution The velocity of an aircraft is to be measured by a Pitot-static probe. For a given differential pressure reading, the velocity of the aircraft is to be determined.



Assumptions **1** The air flow over the aircraft is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Standard atmospheric conditions exist. **3** The wind effects are negligible.

Properties The density of the atmosphere at an elevation of 3000 m is ρ = 0.909 kg/m^3.

Analysis We take point 1 at the entrance of the tube whose opening is parallel to flow, and point 2 at the entrance of the tube whose entrance is normal to flow. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{V_1^2}{2g}=\frac{P_2-P_1}{\rho g} \quad\rightarrow\quad \frac{V_1^2}{2}=\frac{P_{stag}-P_1}{\rho}$$

Solving for V_1 and substituting,

$$V_1=\sqrt{\frac{2(P_{stag}-P_1)}{\rho}}=\sqrt{\frac{2(3000\text{ N/m}^2)}{0.909\text{ kg/m}^3}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}=\mathbf{81.2\,m/s}=\mathbf{292\,km/h}$$

since 1 Pa = 1 N/m^2 and 1 m/s = 3.6 km/h.

Discussion Note that the velocity of an aircraft can be determined by simply measuring the differential pressure on a Pitot-static probe.

5-44

Solution A Pitot-static probe is inserted into the duct of an air heating system parallel to flow, and the differential height of the water column is measured. The flow velocity and the pressure rise at the tip of the Pitot-static probe are to be determined.

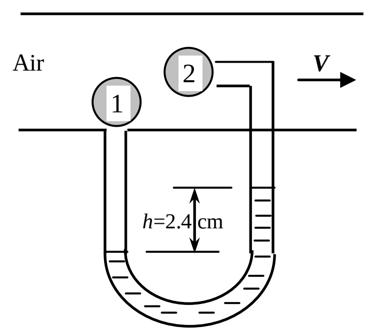
Assumptions **1** The flow through the duct is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Air is an ideal gas.

Properties We take the density of water to be ρ = 1000 kg/m^3. The gas constant of air is R = 0.287 kPa·m^3/kg·K.

Analysis We take point 1 on the side of the probe where the entrance is parallel to flow and is connected to the static arm of the Pitot-static probe, and point 2 at the tip of the probe where the entrance is normal to flow and is connected to the dynamic arm of the Pitot-static probe. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}+\frac{V_1^2}{2g}=\frac{P_2}{\rho g} \quad\rightarrow\quad V=\sqrt{\frac{2(P_2-P_1)}{\rho_{air}}}$$

where the pressure rise at the tip of the Pitot-static probe is



$$P_2-P_1=\rho_w gh=(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.024\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$=235\text{ N/m}^2=\mathbf{235\ Pa}$$

Also, $$\rho_{air}=\frac{P}{RT}=\frac{98\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(45+273\text{ K})}=1.074\text{ kg/m}^3$$

Substituting,

$$V_1=\sqrt{\frac{2(235\text{ N/m}^2)}{1.074\text{ kg/m}^3}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}=\mathbf{20.9\ m/s}$$

Discussion Note that the flow velocity in a pipe or duct can be measured easily by a Pitot-static probe by inserting the probe into the pipe or duct parallel to flow, and reading the differential pressure height. Also note that this is the velocity at the location of the tube. Several readings at several locations in a cross-section may be required to determine the mean flow velocity.

5-45E

Solution The drinking water needs of an office are met by large water bottles with a plastic hose inserted in it. The minimum filling time of an 8-oz glass is to be determined when the bottle is full and when it is near empty.

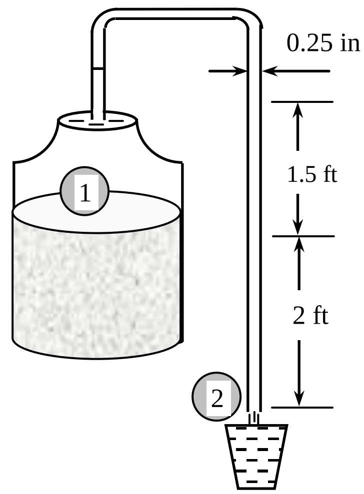
Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** All losses are neglected to obtain the minimum filling time.

Analysis We take point 1 to be at the free surface of water in the bottle and point 2 at the exit of the tube so that $P_1 = P_2 = P_{atm}$ (the bottle is open to the atmosphere and water discharges into the atmosphere), $V_1 \cong 0$ (the bottle is large relative to the tube diameter), and $z_2 = 0$ (we take point 2 as the reference level). Then the Bernoulli equation simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad z_1=\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1}$$

Substituting, the discharge velocity of water and the filling time are determined as follows:

(*a*) *Full bottle* (z_1 = 3.5 ft):



$$V_2=\sqrt{2(32.2\text{ ft/s}^2)(3.5\text{ ft})}=15.0\text{ ft/s}$$

$$A=\pi D^2/4=\pi(0.25/12\text{ ft})^2/4=3.41\times10^{-4}\text{ ft}^2$$

$$\Delta t=\frac{\mathbf{V}}{\dot{\mathbf{V}}}=\frac{\mathbf{V}}{AV_2}=\frac{0.00835\text{ ft}^3}{(3.41\times10^{-4}\text{ ft}^2)(15\text{ ft/s})}=\mathbf{1.6\ s}$$

(*b*) *Empty bottle* (z_1 = 2 ft):

$$V_2=\sqrt{2(32.2\text{ ft/s}^2)(2\text{ ft})}=11.3\text{ ft/s}$$

$$\Delta t=\frac{\mathbf{V}}{\dot{\mathbf{V}}}=\frac{\mathbf{V}}{AV_2}=\frac{0.00835\text{ ft}^3}{(3.41\times10^{-4}\text{ ft}^2)(11.3\text{ ft/s})}=\mathbf{2.2\ s}$$

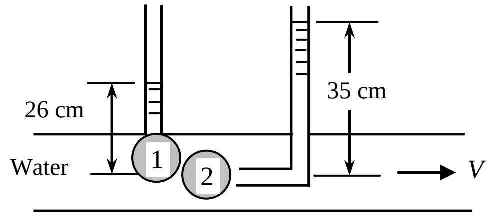
Discussion The siphoning time is determined assuming frictionless flow, and thus this is the *minimum time* required. In reality, the time will be longer because of friction between water and the tube surface.

5-46

Solution The static and stagnation pressures in a horizontal pipe are measured. The velocity at the center of the pipe is to be determined.

Assumptions The flow is steady, incompressible, and irrotational with negligible frictional effects in the short distance between the two pressure measurement locations (so that the Bernoulli equation is applicable).

Analysis We take points 1 and 2 along the centerline of the pipe, with point 1 directly under the piezometer and point 2 at the entrance of the Pitot-static probe (the stagnation point).



This is a steady flow with straight and parallel streamlines, and thus the static pressure at any point is equal to the hydrostatic pressure at that point. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \rightarrow \quad \frac{V_1^2}{2g} = \frac{P_2 - P_1}{\rho g}$$

Substituting the P_1 and P_2 expressions give

$$\frac{V_1^2}{2g} = \frac{P_2 - P_1}{\rho g} = \frac{\rho g(h_{\text{pitot}} + R) - \rho g(h_{\text{piezo}} + R)}{\rho g} = \frac{\rho g(h_{\text{pitot}} - h_{\text{piezo}})}{\rho g} = h_{\text{pitot}} - h_{\text{piezo}}$$

Solving for V_1 and substituting,

$$V_1 = \sqrt{2g(h_{\text{pitot}} - h_{\text{piezo}})} = \sqrt{2(9.81\ \text{m/s}^2)[(0.35 - 0.26)\ \text{m}]} = \mathbf{1.33\ m/s}$$

Discussion Note that to determine the flow velocity, all we need is to measure the height of the excess fluid column in the Pitot-static probe.

5-47

Solution A water tank of diameter D_o and height H open to the atmosphere is initially filled with water. An orifice of diameter D with a smooth entrance (no losses) at the bottom drains to the atmosphere. Relations are to be developed for the time required for the tank to empty completely and half-way.

Assumptions **1** The orifice has a smooth entrance, and thus the frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

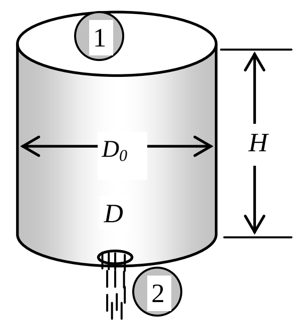
Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of orifice. We take the reference level at the orifice ($z_2 = 0$), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{\text{atm}}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the Bernoulli equation between these two points simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad z_1=\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1}$$

For generality, we express the water height in the tank at any time t by z, and the discharge velocity by $V_2=\sqrt{2gz}$. Note that water surface in the tank moves down as the tank drains, and thus z is a variable whose value changes from H at the beginning to 0 when the tank is emptied completely.

We denote the diameter of the orifice by D, and the diameter of the tank by D_o. The flow rate of water from the tank is obtained by multiplying the discharge velocity by the orifice cross-sectional area,

$$\dot{V}=A_{\text{orifice}}V_2=\frac{\pi D^2}{4}\sqrt{2gz}$$



Then the amount of water that flows through the orifice during a differential time interval dt is

$$dV=\dot{V}dt=\frac{\pi D^2}{4}\sqrt{2gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV=A_{\text{tank}}(-dz)=-\frac{\pi D_0^2}{4}\,dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{2gz}\,dt=-\frac{\pi D_0^2}{4}\,dz \quad\rightarrow\quad dt=-\frac{D_0^2}{D^2}\sqrt{\frac{1}{2gz}}\,dz=-\frac{D_0^2}{D^2\sqrt{2g}}\,z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_i = H$ to $t = t_f$ when $z = z_f$ gives

$$\int_{t=0}^{t_f}dt=-\frac{D_0^2}{D^2\sqrt{2g}}\int_{z=z_1}^{z_f}z^{-1/2}dz \quad\rightarrow\quad t_f=-\frac{D_0^2}{D^2\sqrt{2g}}\left.\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{z_f}=\frac{2D_0^2}{D^2\sqrt{2g}}\left(\sqrt{z_i}-\sqrt{z_f}\right)=\frac{D_0^2}{D^2}\left(\sqrt{\frac{2z_i}{g}}-\sqrt{\frac{2z_f}{g}}\right)$$

Then the discharging time for the two cases becomes as follows:

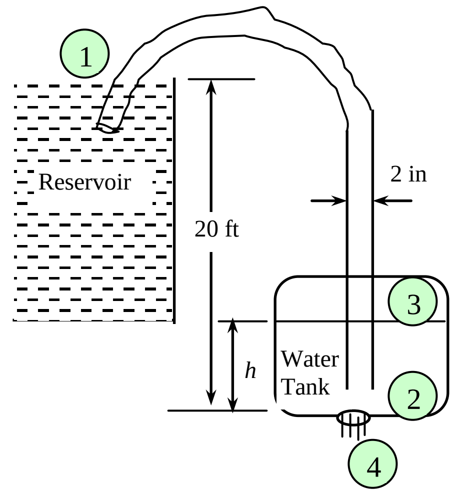
(*a*) *The tank empties halfway*: $z_i = H$ and $z_f = H/2$: $\boxed{t_f=\frac{D_0^2}{D^2}\left(\sqrt{\frac{2H}{g}}-\sqrt{\frac{H}{g}}\right)}$

(*b*) *The tank empties completely*: $z_i = H$ and $z_f = 0$: $\boxed{t_f=\frac{D_0^2}{D^2}\sqrt{\frac{2H}{g}}}$

Discussion Note that the discharging time is inversely proportional to the square of the orifice diameter. Therefore, the discharging time can be reduced to one-fourth by doubling the diameter of the orifice.

5-48E

Solution A siphon pumps water from a large reservoir to a lower tank which is initially empty. Water leaves the tank through an orifice. The height the water will rise in the tank at equilibrium is to be determined.



Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Both the tank and the reservoir are open to the atmosphere. **3** The water level of the reservoir remains constant.

Analysis We take the reference level to be at the bottom of the tank, and the water height in the tank at any time to be h. We take point 1 to be at the free surface of reservoir, point 2 at the exit of the siphon, which is placed at the bottom of the tank, and point 3 at the free surface of the tank, and point 4 at the exit of the orifice at the bottom of the tank. Then z_1 = 20 ft, $z_2 = z_4 = 0$, $z_3 = h$, $P_1 = P_3 = P_4 = P_{atm}$ (the reservoir is open to the atmosphere and water discharges into the atmosphere) $P_2 = P_{atm}+\rho gh$ (the hydrostatic pressure at the bottom of the tank where the siphon discharges), and $V_1 \cong V_3 \cong 0$ (the free surfaces of reservoir and the tank are large relative to the tube diameter). Then the Bernoulli equation between 1-2 and 3-4 simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_{atm}}{\rho g}+z_1=\frac{P_{atm}+\rho gh}{\rho g}+\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1-2gh}=\sqrt{2g(z_1-h)}$$

$$\frac{P_3}{\rho g}+\frac{V_3^2}{2g}+z_3=\frac{P_4}{\rho g}+\frac{V_4^2}{2g}+z_4 \quad\rightarrow\quad h=\frac{V_4^2}{2g} \quad\rightarrow\quad V_4=\sqrt{2gh}$$

Noting that the diameters of the tube and the orifice are the same, the flow rates of water into and out of the tank will be the same when the water velocities in the tube and the orifice are equal since

$$\dot{V}_2=\dot{V}_4 \quad\rightarrow\quad AV_2=AV_4 \quad\rightarrow\quad V_2=V_4$$

Setting the two velocities equal to each other gives

$$V_2=V_4 \quad\rightarrow\quad \sqrt{2g(z_1-h)}=\sqrt{2gh} \quad\rightarrow\quad z_1-h=h \quad\rightarrow\quad h=\frac{z_1}{2}=\frac{20\text{ ft}}{2}=\mathbf{10\ ft}$$

Therefore, the water level in the tank will stabilize when the water level rises to 10 ft.

Discussion This result is obtained assuming negligible friction. The result would be somewhat different if the friction in the pipe and orifice were considered.

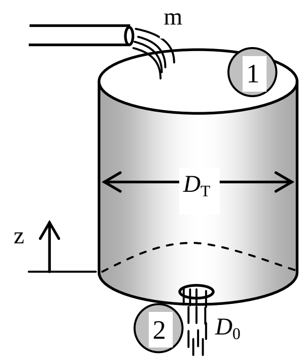
5-49

Solution Water enters an empty tank steadily at a specified rate. An orifice at the bottom allows water to escape. The maximum water level in the tank is to be determined, and a relation for water height z as a function of time is to be obtained.

Assumptions **1** The orifice has a smooth entrance, and thus the frictional losses are negligible. **2** The flow through the orifice is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Analysis (*a*) We take point 1 at the free surface of the tank, and point 2 at the exit of orifice. We take the reference level at the orifice ($z_2 = 0$), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$) (it becomes zero when the water in the tank reaches its maximum level), the Bernoulli equation between these two points simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad z_1=\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1}$$



Then the mass flow rate through the orifice for a water height of z becomes

$$\dot m_{out}=\rho\dot{\mathcal V}_{out}=\rho A_{orifice}V_2=\rho\frac{\pi D_0^2}{4}\sqrt{2gz} \quad\rightarrow\quad z=\frac{1}{2g}\left(\frac{4\dot m_{out}}{\rho\pi D_0^2}\right)^2$$

Setting $z = h_{max}$ and $\dot m_{out} = \dot m_{in}$ (the incoming flow rate) gives the desired relation for the maximum height the water will reach in the tank,

$$h_{max}=\frac{1}{2g}\left(\frac{4\dot m_{in}}{\rho\pi D_0^2}\right)^2$$

(*b*) The amount of water that flows through the orifice and the increase in the amount of water in the tank during a differential time interval dt are

$$dm_{out}=\dot m_{out}dt=\rho\frac{\pi D_0^2}{4}\sqrt{2gz}\,dt$$

$$dm_{tank}=\rho A_{tank}dz=\rho\frac{\pi D_T^2}{4}dz$$

The amount of water that enters the tank during dt is $dm_{in}=\dot m_{in}dt$ (Recall that $\dot m_{in}$ = constant). Substituting them into the conservation of mass relation $dm_{tank}=dm_{in}-dm_{out}$ gives

$$dm_{tank}=\dot m_{in}dt-\dot m_{out}dt \quad\rightarrow\quad \rho\frac{\pi D_T^2}{4}dz=\left(\dot m_{in}-\rho\frac{\pi D_0^2}{4}\sqrt{2gz}\right)dt$$

Separating the variables, and integrating it from $z = 0$ at $t = 0$ to $z = z$ at time $t = t$ gives

$$\frac{\frac14\rho\pi D_T^2\,dz}{\dot m_{in}-\frac14\rho\pi D_0^2\sqrt{2gz}}=dt \quad\rightarrow\quad \int_{z=0}^{z}\frac{\frac14\rho\pi D_T^2\,dz}{\dot m_{in}-\frac14\rho\pi D_0^2\sqrt{2gz}}=\int_{t=0}^{t}dt=t$$

Performing the integration, the desired relation between the water height z and time t is obtained to be

$$\boxed{\frac{\frac12\rho\pi D_T^2}{(\frac14\rho\pi D_0^2\sqrt{2g})^2}\left(\frac14\rho\pi D_0^2\sqrt{2gz}-\dot m_{in}\ln\frac{\dot m_{in}-\frac14\rho\pi D_0^2\sqrt{2gz}}{\dot m_{in}}\right)=t}$$

Discussion Note that this relation is implicit in z, and thus we can't obtain a relation in the form $z = f(t)$. Substituting a z value in the left side gives the time it takes for the fluid level in the tank to reach that level. Equation solvers such as EES can easily solve implicit equations like this.

5-50E

Solution Water flows through a horizontal pipe that consists of two sections at a specified rate. The differential height of a mercury manometer placed between the two pipe sections is to be determined.

Assumptions **1**The flow through the pipe is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The losses in the reducing section are negligible.

Properties The densities of mercury and water are ρ_{Hg} = 847 lbm/ft^3 and ρ_w = 62.4 lbm/ft^3.

Analysis We take points 1 and 2 along the centerline of the pipe over the two tubes of the manometer. Noting that $z_1 = z_2$, the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad P_1-P_2=\frac{\rho_w(V_2^2-V_1^2)}{2} \qquad (1)$$

We let the differential height of the mercury manometer be *h* and the distance between the centerline and the mercury level in the tube where mercury is raised be *s*. Then the pressure difference $P_2 - P_1$ can also be expressed as

$$P_1+\rho_w g(s+h)=P_2+\rho_w gs+\rho_{Hg}gh \qquad\rightarrow\qquad P_1-P_2=(\rho_{Hg}-\rho_w)gh \qquad (2)$$

Combining Eqs. (1) and (2) and solving for *h*,

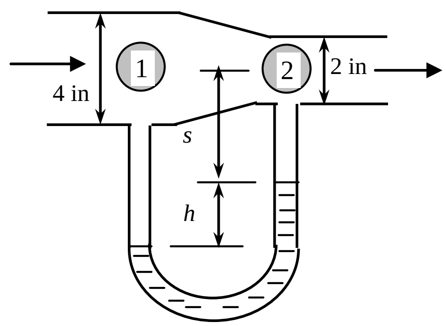
$$\frac{\rho_w(V_2^2-V_1^2)}{2}=(\rho_{Hg}-\rho_w)gh \qquad\rightarrow\qquad h=\frac{\rho_w(V_2^2-V_1^2)}{2g(\rho_{Hg}-\rho_w)}=\frac{V_2^2-V_1^2}{2g(\rho_{Hg}/\rho_w-1)}$$

Calculating the velocities and substituting,

$$V_1=\frac{\dot{V}}{A_1}=\frac{\dot{V}}{\pi D_1^2/4}=\frac{2.4\text{ gal/s}}{\pi(4/12\text{ ft})^2/4}\left(\frac{0.13368\text{ ft}^3}{1\text{ gal}}\right)=3.676\text{ ft/s}$$

$$V_2=\frac{\dot{V}}{A_2}=\frac{\dot{V}}{\pi D_2^2/4}=\frac{2.4\text{ gal/s}}{\pi(2/12\text{ ft})^2/4}\left(\frac{0.13368\text{ ft}^3}{1\text{ gal}}\right)=14.71\text{ ft/s}$$

$$h=\frac{(14.71\text{ ft/s})^2-(3.676\text{ ft/s})^2}{2(32.2\text{ ft/s}^2)(847/62.4-1)}=0.2504\text{ ft }=\mathbf{3.0\,in}$$



Therefore, the differential height of the mercury column will be 3.0 in.

Discussion In reality, there are frictional losses in the pipe, and the pressure at location 2 will actually be smaller than that estimated here, and therefore *h* will be larger than that calculated here.

5-51

Solution An airplane is flying at a certain altitude at a given speed. The pressure on the stagnation point on the nose of the plane is to be determined, and the approach to be used at high velocities is to be discussed.

Assumptions **1** The air flow over the aircraft is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Standard atmospheric conditions exist. **3** The wind effects are negligible.

Properties The density of the atmospheric air at an elevation of 12,000 m is $\rho = 0.312$ kg/m^3.

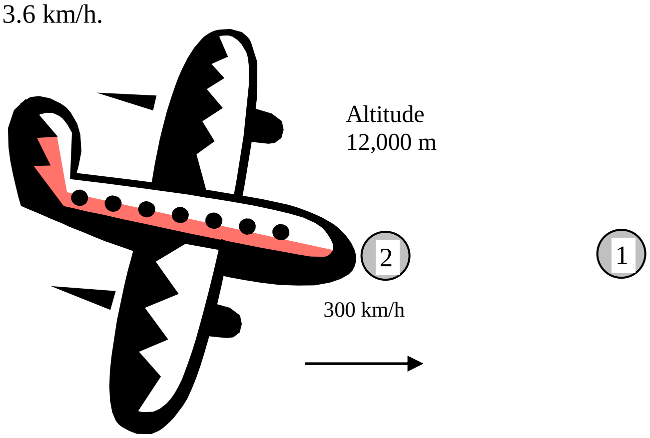
Analysis We take point 1 well ahead of the plane at the level of the nose, and point 2 at the nose where the flow comes to a stop. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\to\quad \frac{V_1^2}{2g}=\frac{P_2-P_1}{\rho g} \quad\to\quad \frac{V_1^2}{2}=\frac{P_{stag}-P_{\text{atm}}}{\rho}=\frac{P_{\text{stag,gage}}}{\rho}$$

Solving for $P_{\text{stag, gage}}$ and substituting,

$$P_{\text{stag,gage}}=\frac{\rho V_1^2}{2}=\frac{(0.312\text{ kg/m}^3)(300/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)=1083\text{ N/m}^2=\mathbf{1083\,Pa}$$

since 1 Pa = 1 N/m^2 and 1 m/s = 3.6 km/h.



Discussion A flight velocity of 1050 km/h = 292 m/s corresponds to a Mach number much greater than 0.3 (the speed of sound is about 340 m/s at room conditions, and lower at higher altitudes, and thus a Mach number of 292/340 = 0.86). Therefore, the flow can no longer be assumed to be incompressible, and the Bernoulli equation given above cannot be used. This problem can be solved using the modified Bernoulli equation that accounts for the effects of compressibility, assuming isentropic flow.

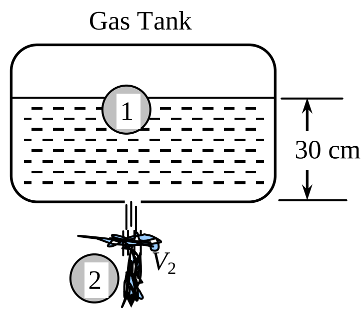
5-52

Solution The bottom of a car hits a sharp rock and a small hole develops at the bottom of its gas tank. For a given height of gasoline, the initial velocity of the gasoline out of the hole is to be determined. Also, the variation of velocity with time and the effect of the tightness of the lid on flow rate are to be discussed.

Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The air space in the tank is at atmospheric pressure. **3** The splashing of the gasoline in the tank during travel is not considered.

Analysis This problem involves the conversion of flow, kinetic, and potential energies to each other without involving any pumps, turbines, and wasteful components with large frictional losses, and thus it is suitable for the use of the Bernoulli equation. We take point 1 to be at the free surface of gasoline in the tank so that $P_1 = P_{atm}$ (open to the atmosphere) $V_1 \cong 0$ (the tank is large relative to the outlet), and $z_1 = 0.3$ m and $z_2 = 0$ (we take the reference level at the hole. Also, $P_2 = P_{atm}$ (gasoline discharges into the atmosphere). Then the Bernoulli equation simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad \rightarrow \quad z_1=\frac{V_2^2}{2g}$$



Solving for V_2 and substituting,

$$V_2=\sqrt{2gz_1}=\sqrt{2(9.81\,\text{m/s}^2)(0.3\,\text{m})}=\mathbf{2.43\,m/s}$$

Therefore, the gasoline will initially leave the tank with a velocity of 2.43 m/s.

Discussion The Bernoulli equation applies along a streamline, and streamlines generally do not make sharp turns. The velocity will be less than 2.43 m/s since the hole is probably sharp-edged and it will cause some head loss. As the gasoline level is reduced, the velocity will decrease since velocity is proportional to the square root of liquid height. If the lid is tightly closed and no air can replace the lost gasoline volume, the pressure above the gasoline level will be reduced, and the velocity will be decreased.

5-53

Solution The water in an above the ground swimming pool is to be emptied by unplugging the orifice of a horizontal pipe attached to the bottom of the pool. The maximum discharge rate of water is to be determined.

Assumptions **1** The orifice has a smooth entrance, and all frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

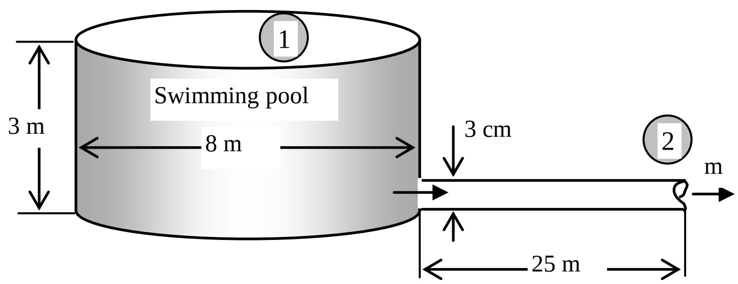
Analysis We take point 1 at the free surface of the pool, and point 2 at the exit of pipe. We take the reference level at the pipe exit ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the Bernoulli equation between these two points simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad z_1=\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1}$$

The maximum discharge rate occurs when the water height in the pool is a maximum, which is the case at the beginning and thus $z_1 = h$. Substituting, the maximum flow velocity and discharge rate become

$$V_{2,\max}=\sqrt{2gh}=\sqrt{2(9.81\,\text{m/s}^2)(3\,\text{m})}=7.67\text{ m/s}$$

$$\dot{V}_{\max}=A_{\text{pipe}}V_{2,\max}=\frac{\pi D^2}{4}V_{2,\max}=\frac{\pi(0.03\,\text{m})^2}{4}(7.67\text{ m/s})=0.00542\,\text{m}^3/\text{s}=\mathbf{5.42\,L/s}$$



Discussion The result above is obtained by disregarding all frictional effects. The actual flow rate will be less because of frictional effects during flow and the resulting pressure drop. Also, the flow rate will gradually decrease as the water level in the pipe decreases.

5-54

Solution The water in an above the ground swimming pool is to be emptied by unplugging the orifice of a horizontal pipe attached to the bottom of the pool. The time it will take to empty the tank is to be determined.

Assumptions **1** The orifice has a smooth entrance, and all frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Analysis We take point 1 at the free surface of water in the pool, and point 2 at the exit of pipe. We take the reference level at the pipe exit ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{\rm atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the Bernoulli equation between these two points simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad z_1=\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1}$$

For generality, we express the water height in the pool at any time t by z, and the discharge velocity by $V_2=\sqrt{2gz}$. Note that water surface in the pool moves down as the pool drains, and thus z is a variable whose value changes from h at the beginning to *0* when the pool is emptied completely.

We denote the diameter of the orifice by D, and the diameter of the pool by D_o. The flow rate of water from the pool is obtained by multiplying the discharge velocity by the orifice cross-sectional area,

$$\dot{V}=A_{\rm orifice}V_2=\frac{\pi D^2}{4}\sqrt{2gz}$$

Then the amount of water that flows through the orifice during a differential time interval dt is

$$dV=\dot{V}dt=\frac{\pi D^2}{4}\sqrt{2gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the pool,

$$dV=A_{\rm tank}(-dz)=-\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the pool during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

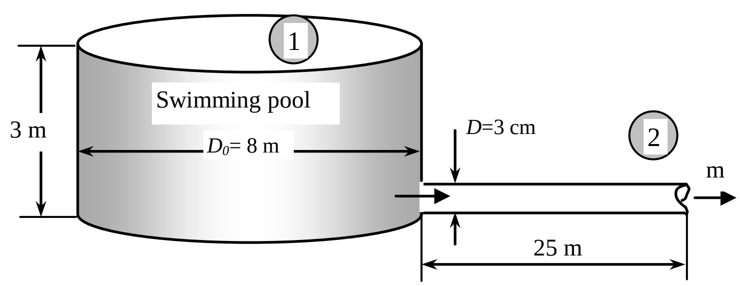
$$\frac{\pi D^2}{4}\sqrt{2gz}\,dt=-\frac{\pi D_0^2}{4}dz \quad\rightarrow\quad dt=-\frac{D_0^2}{D^2}\sqrt{\frac{1}{2gz}}dz=-\frac{D_0^2}{D^2\sqrt{2g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = h$ to $t = t_f$ when $z = 0$ (completely drained pool) gives

$$\int_{t=0}^{t_f}dt=-\frac{D_0^2}{D^2\sqrt{2g}}\int_{z=z_1}^{0}z^{-1/2}dz \quad\rightarrow\quad t_f=-\frac{D_0^2}{D^2\sqrt{2g}}\left.\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0}=\frac{2D_0^2}{D^2\sqrt{2g}}\sqrt{h}=\frac{D_0^2}{D^2}\sqrt{\frac{2h}{g}}$$

Substituting, the draining time of the pool will be

$$t_f=\frac{(8\,{\rm m})^2}{(0.03\,{\rm m})^2}\sqrt{\frac{2(3\,{\rm m})}{9.81\,{\rm m/s^2}}}=55{,}600\,{\rm s}=\mathbf{15.4\,h}$$



Discussion This is the minimum discharging time since it is obtained by neglecting all friction; the actual discharging time will be longer. Note that the discharging time is inversely proportional to the square of the orifice diameter. Therefore, the discharging time can be reduced to one-fourth by doubling the diameter of the orifice.

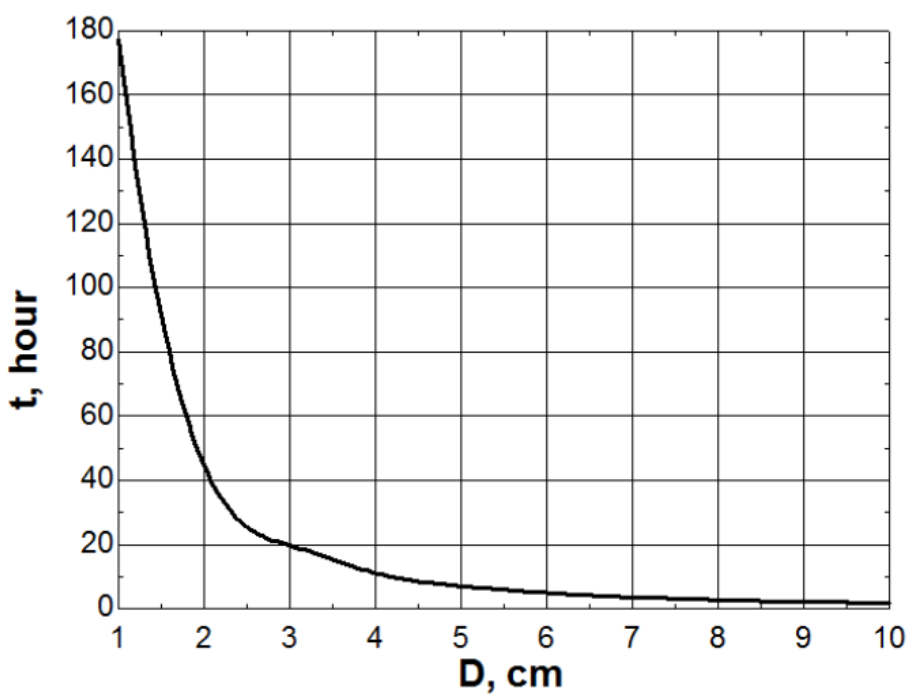
5-55

Solution The previous problem is reconsidered. The effect of the discharge pipe diameter on the time required to empty the pool completely as the diameter varies from 1 to 10 cm in increments of 1 cm is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
h=2 "m"
D=d_pipe/100 "m"
D_pool=10 "m"
V_initial=SQRT(2*g*h) "m/s"
Ac=pi*D^2/4
V_dot=Ac*V_initial*1000 "m3/s"
t=(D_pool/D)^2*SQRT(2*h/g)/3600 "hour"
```

Pipe diameter *D*, m	Discharge time *t*, h
1	177.4
2	44.3
3	19.7
4	11.1
5	7.1
6	4.9
7	3.6
8	2.8
9	2.2
10	1.8



Discussion As you can see from the plot, the discharge time is drastically reduced by increasing the pipe diameter.

5-56

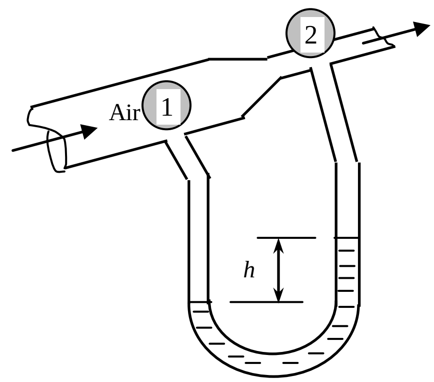
Solution Air flows upward at a specified rate through an inclined pipe whose diameter is reduced through a reducer. The differential height between fluid levels of the two arms of a water manometer attached across the reducer is to be determined.

Assumptions **1** The flow through the duct is steady, incompressible and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Air is an ideal gas. **3** The effect of air column on the pressure change is negligible because of its low density. **4** The air flow is parallel to the entrance of each arm of the manometer, and thus no dynamic effects are involved.

Properties We take the density of water to be ρ = 1000 kg/m^3. The gas constant of air is R = 0.287 kPa·m^3/kg·K.

Analysis We take points 1 and 2 at the lower and upper connection points, respectively, of the two arms of the manometer, and take the lower connection point as the reference level. Noting that the effect of elevation on the pressure change of a gas is negligible, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad P_1-P_2=\rho_{\text{air}}\frac{V_2^2-V_1^2}{2}$$



where $$\rho_{air}=\frac{P}{RT}=\frac{105\,\text{kPa}}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(37+273\,\text{K})}=1.180\,\text{kg/m}^3$$

$$V_1=\frac{\dot{V}}{A_1}=\frac{\dot{V}}{\pi D_1^2/4}=\frac{0.065\,\text{m}^3/\text{s}}{\pi(0.06\,\text{m})^2/4}=22.99\,\text{m/s}$$

$$V_2=\frac{\dot{V}}{A_2}=\frac{\dot{V}}{\pi D_2^2/4}=\frac{0.065\,\text{m}^3/\text{s}}{\pi(0.04\,\text{m})^2/4}=51.73\,\text{m/s}$$

Substituting,

$$P_1-P_2=(1.180\,\text{kg/m}^3)\frac{(51.73\,\text{m/s})^2-(22.99\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)=1267\ \text{N/m}^2=1267\ \text{Pa}$$

The differential height of water in the manometer corresponding to this pressure change is determined from $\Delta P=\rho_{water}gh$ to be

$$h=\frac{P_1-P_2}{\rho_{water}g}=\frac{1267\ \text{N/m}^2}{(1000\ \text{kg/m}^3)(9.81\,\text{m/s}^2)}\left(\frac{1\,\text{kg}\cdot\text{m/s}^2}{1\,\text{N}}\right)=0.1291\,\text{m}=\mathbf{12.9\,cm}$$

Discussion When the effect of air column on pressure change is considered, the pressure change becomes

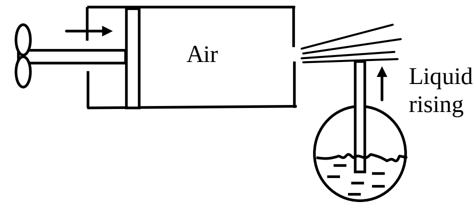
$$\begin{aligned}P_1-P_2&=\frac{\rho_{air}(V_2^2-V_1^2)}{2}+\rho_{air}g(z_2-z_1)\\&=(1.180\,\text{kg/m}^3)\left[\frac{(51.73\,\text{m/s})^2-(22.99\,\text{m/s})^2}{2}+(9.81\,\text{m/s}^2)(0.2\,\text{m})\right]\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\\&=(1267+2)\ \text{N/m}^2=1269\ \text{N/m}^2=1269\ \text{Pa}\end{aligned}$$

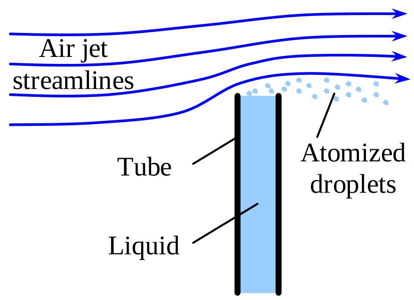
This difference between the two results (1267 and 1269 Pa) is less than 1%. Therefore, the effect of air column on pressure change is, indeed, negligible as assumed. In other words, the pressure change of air in the duct is almost entirely due to velocity change, and the effect of elevation change is negligible. Also, if we were to account for the Δz of air flow, then it would be more proper to account for the Δz of air in the manometer by using ρ_{water} - ρ_{air} instead of ρ_{water} when calculating h. The additional air column in the manometer tends to cancel out the change in pressure due to the elevation difference in the flow in this case.

5-57

Solution A hand-held bicycle pump with a liquid reservoir is used as an atomizer by forcing air at a high velocity through a small hole. We are to explain how the liquid gets sucked up the tube.

Assumptions **1** The flows of air and water are steady and nearly incompressible. **2** Air is an ideal gas. **3** The liquid reservoir is open to the atmosphere. **4** The device is held horizontally. **5** The water velocity through the tube is low (the water in the tube is nearly hydrostatic).





Analysis At first glance, we are tempted to use the Bernoulli equation, thinking that the pressure in the air jet would be lower than atmospheric due to its high speed. However, as stated in the problem statement, the pressure through an incompressible jet exposed to the atmosphere is nearly atmospheric pressure everywhere. Thus, in the absence of the tube, the pressure in the air jet just above the tube would be nearly atmospheric. Meanwhile, the pressure at the liquid surface is also atmospheric. Applying hydrostatics from the liquid surface to the top of the tube reveals that the pressure at the top of the tube must be lower than atmospheric pressure by more than ρgh in order to suck the liquid up the tube. So, what causes the liquid to rise? It turns out that the answer has to do with **streamline curvature**. As the close-up sketch illustrates, the **air streamlines must curve around the top of the tube**. Since pressure decreases towards the center of curvature in a flow with curved streamlines, the pressure at the top of the tube must be less than atmospheric. At high enough air jet speed, the pressure is low enough not only to suck the liquid to the top of the tube, but also to break up the liquid at the top of the tube into small droplets, thereby "atomizing" the liquid into a spray of liquid droplets.

Discussion If the geometry of the top of the tube were known, we could approximate the flow as irrotational and apply the techniques of potential flow analysis (Chap. 10) to estimate the pressure at the top of the tube.

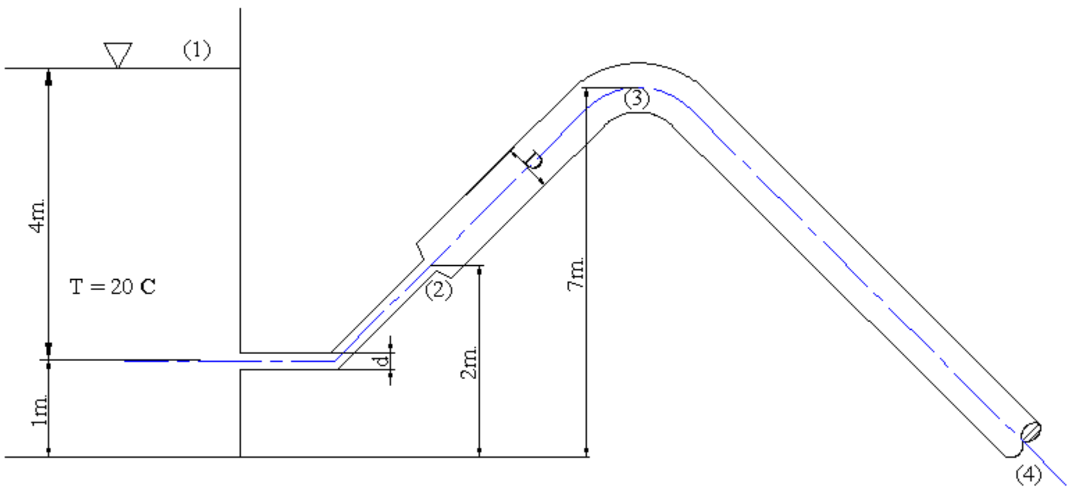
5-58

Solution Water is siphoned from a reservoir. The minimum flow rate that can be achieved without cavitation occurring in the piping system and the maximum elevation of the highest point of the piping system to avoid cavitation are to be determined.

Assumptions **1** The flow through the pipe is steady, incompressible and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Properties We take the density of water to be ρ = 1000 kg/m^3.

Analysis



For T = 20 ^{0}C , P_V = 2.338 x 10^3 Pa (abs)

d = 10 cm. , D = 16 cm.

Applying Bernoulli Eq. between (1) and (4)

$$\frac{P_1}{\gamma}+Z_1+\frac{V_1^2}{2g}=\frac{P_4}{\gamma}+Z_4+\frac{V_4^2}{2g} \quad , \quad V_4=\sqrt{2gh_1}=\sqrt{2g(1+4)}$$

V_4 = 9.904 m/s

On the other hand, from the continuity,

$$A_dV_d=A_DV_D \quad , \quad V_d=(\frac{A_D}{A_d})V_D=\frac{D^2}{d^2}V_D$$

$$V_d=(\frac{16}{10})^2V_D$$

$$V_d=2.56V_D=2.56x9.904=25.35\ m/s$$

We should check if these velocities would be possible,

Bernoulli Eq. from (1) to (2) yields

$$\frac{P_{1m}}{\gamma}+Z_1+\frac{V_1^2}{2g}=\frac{P_{2m}}{\gamma}+Z_2+\frac{V_2^2}{2g}$$

$$\frac{101325}{9810}+5+0=\frac{P_{2m}}{\gamma}+2+\frac{23.35^2}{19.62}$$

$$15.328=\frac{P_{2m}}{\gamma}+29.789 \quad , \quad \frac{P_{2m}}{\gamma}=-14.461m.$$

Since $\frac{P_{2m}}{\gamma} < 0$, the velocity V_d cannot be 23.35 m/s. Applying Bernoulli Eq. from (1) to(2)

$$\frac{101325}{9810} + 5 = 2 + \frac{2.338.10^3}{9810} + \frac{V_{d\max}^2}{2g}$$

$$\frac{V_{d\max}^2}{2g} = 13.09$$

$V_{max} \approx 16$ m/s. Therefore the velocity will never exceed 16 m/s. Accordingly;

$$\dot{V} = A_d V_d = \pi \frac{0.1^2}{4} 16 = \mathbf{0.125\ m^3/s}$$

(b) For a maximum Z_3 , the absolute pressure $P_{3\min} = 2338\ Pa(abs)$

$$\frac{P_{1m}}{\gamma} + Z_1 + \frac{V_1^2}{2g} = cons\tan t \quad \text{, therefore}$$

$$\frac{\mathrm{P}_3}{\gamma} \text{ and } \frac{V_3^2}{2g} \quad \text{must be minimum.}$$

$$\frac{P_{3\min}}{\gamma} = \frac{2338}{9810} = 0.238m.$$

Applying Bernoulli Eq. from (1) to (3)

$$\frac{101325}{9810} + 5 + 0 = Z_{3.\max} + 0.238 + \frac{V_D^2}{2g}$$

From the first part,

$$V_D = \frac{d^2}{D^2} V_d = (\frac{10}{16})^2 16$$

$V_D = 6.25$ m/s

Therefore $\quad Z_{3.\max} = \frac{101325}{9810} + 5 - 0.238 - \frac{6.25^2}{19.62}$

$Z_{3.\max} \approx$ **13 m**

5-59

Solution The gage pressure in the water mains of a city at a particular location is given. It is to be determined if this main can serve water to neighborhoods that are at a given elevation relative to this location.

Assumptions Water is incompressible and thus its density is constant.

Properties We take the density of water to be ρ = 1000 kg/m^3.

Analysis Noting that the gage pressure at a dept of h in a fluid is given by $P_{gage} = \rho_{water} gh$, the height of a fluid column corresponding to a gage pressure of 270 kPa is determined to be

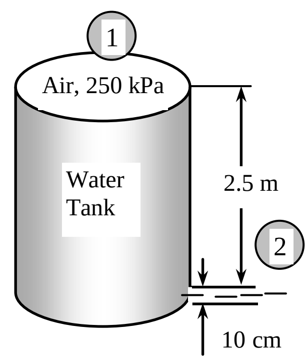
$$h = \frac{P_{gage}}{\rho_{water} g} = \frac{270{,}000\ \text{N/m}^2}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 27.5\ \text{m}$$

which is higher than 25 m. Therefore, this main **can** serve water to neighborhoods that are 25 m above this location.

Discussion Note that h must be much greater than 25 m for water to have enough pressure to serve the water needs of the neighborhood.

5-60

Solution Water discharges to the atmosphere from the orifice at the bottom of a pressurized tank. Assuming frictionless flow, the discharge rate of water from the tank is to be determined.



Assumptions **1** The orifice has a smooth entrance, and thus the frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of orifice, which is also taken to be the reference level (z_2 = 0). Noting that the fluid velocity at the free surface is very low ($V_1 \cong 0$) and water discharges into the atmosphere (and thus $P_2 = P_{atm}$), the Bernoulli equation simplifies to

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad\rightarrow\quad \frac{V_2^2}{2g} = \frac{P_1 - P_2}{\rho g} + z_1$$

Solving for V_2 and substituting, the discharge velocity is determined to

$$V_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho} + 2gz_1} = \sqrt{\frac{2(250-100)\ \text{kPa}}{1000\ \text{kg/m}^3}\left(\frac{1000\ \text{N/m}^2}{1\ \text{kPa}}\right)\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) + 2(9.81\ \text{m/s}^2)(2.5\ \text{m})} = 18.7\ \text{m/s}$$

Then the initial rate of discharge of water becomes

$$\dot{V} = A_{\text{orifice}} V_2 = \frac{\pi D^2}{4} V_2 = \frac{\pi (0.10\ \text{m})^2}{4}(18.7\ \text{m/s}) = \mathbf{0.147\ m^3/s}$$

Discussion Note that this is the maximum flow rate since the frictional effects are ignored. Also, the velocity and the flow rate will decrease as the water level in the tank decreases.

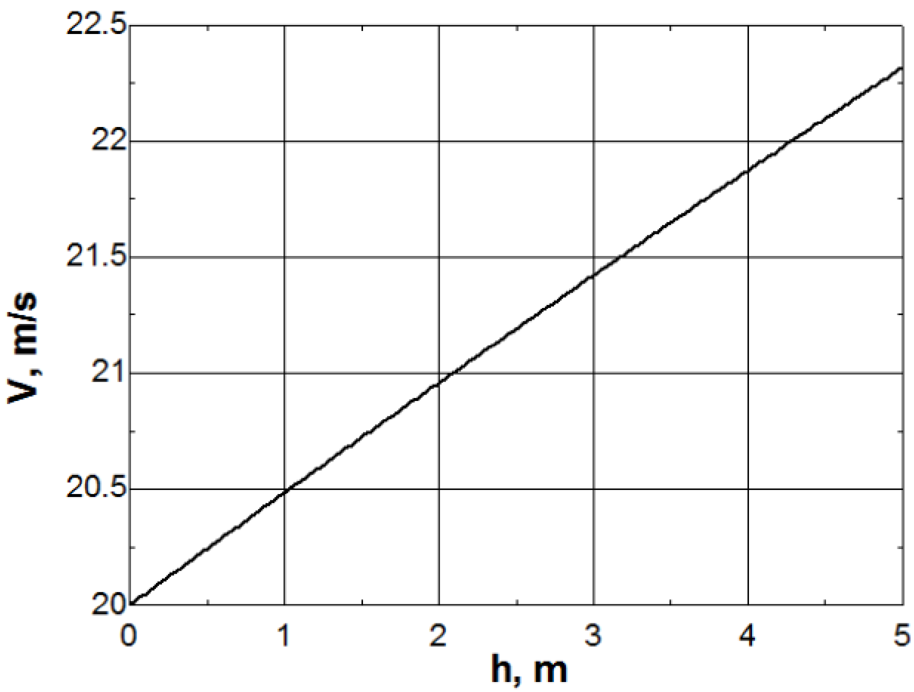
5-61

Solution The previous problem is reconsidered. The effect of water height in the tank on the discharge velocity as the water height varies from 0 to 5 m in increments of 0.5 m is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
d=0.10 "m"
P1=300 "kPa"
P_atm=100 "kPa"
V=SQRT(2*(P1-P_atm)*1000/rho+2*g*h)
Ac=pi*D^2/4
V_dot=Ac*V
```

h, m	*V*, m/s	$\dot{V}$, m^3/s
0.00	20.0	0.157
0.50	20.2	0.159
1.00	20.5	0.161
1.50	20.7	0.163
2.00	21.0	0.165
2.50	21.2	0.166
3.00	21.4	0.168
3.50	21.6	0.170
4.00	21.9	0.172
4.50	22.1	0.174
5.00	22.3	0.175



Discussion Velocity appears to change nearly linearly with *h* in this range of data, but the relationship is *not* linear.

5-62E

Solution Air is flowing through a venturi meter with known diameters and measured pressures. A relation for the flow rate is to be obtained, and its numerical value is to be determined.

Assumptions **1**The flow through the venturi is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The effect of air column on the pressure change is negligible because of its low density, and thus the pressure can be assumed to be uniform at a given cross-section of the venturi meter (independent of elevation change). **3** The flow is horizontal (this assumption is usually unnecessary for gas flow.).

Properties The density of air is given to be $\rho = 0.075$ lbm/ft^3.

Analysis We take point 1 at the main flow section and point 2 at the throat along the centerline of the venturi meter. Noting that $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

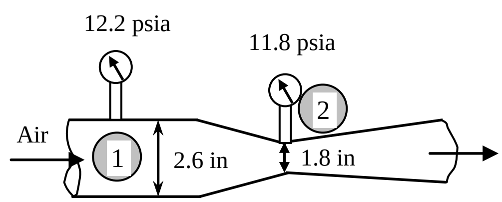
$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad P_1-P_2=\rho\frac{V_2^2-V_1^2}{2} \qquad (1)$$

The flow is assumed to be incompressible and thus the density is constant. Then the conservation of mass relation for this single stream steady flow device can be expressed as

$$\dot{V}_1=\dot{V}_2=\dot{V} \quad\rightarrow\quad A_1V_1=A_2V_2=\dot{V} \quad\rightarrow\quad V_1=\frac{\dot{V}}{A_1} \quad \text{and} \quad V_2=\frac{\dot{V}}{A_2} \qquad (2)$$

Substituting into Eq. (1),

$$P_1-P_2=\rho\frac{(\dot{V}/A_2)^2-(\dot{V}/A_1)^2}{2}=\frac{\rho\dot{V}^2}{2A_2^2}\left(1-\frac{A_2^2}{A_1^2}\right)$$



Solving for $\dot{V}$ gives the desired relation for the flow rate,

$$\dot{V}=A_2\sqrt{\frac{2(P_1-P_2)}{\rho[1-(A_2/A_1)^2]}} \qquad (3)$$

The flow rate for the given case can be determined by substituting the given values into this relation to be

$$\dot{V}=\frac{\pi D_2^2}{4}\sqrt{\frac{2(P_1-P_2)}{\rho[1-(D_2/D_1)^4]}}=\frac{\pi(1.8/12\text{ ft})^2}{4}\sqrt{\frac{2(12.2-11.8)\text{ psi}}{(0.075\text{ lbm/ft}^3)[1-(1.8/2.6)^4]}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)}$$
$$=\mathbf{4.48\ ft^3/s}$$

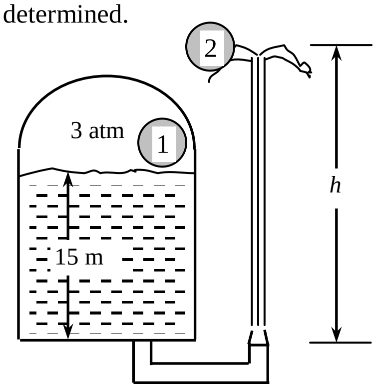
Discussion Venturi meters are commonly used as flow meters to measure the flow rate of gases and liquids by simply measuring the pressure difference P_1 - P_2 by a manometer or pressure transducers. The actual flow rate will be less than the value obtained from Eq. (3) because of the friction losses along the wall surfaces in actual flow. But this difference can be as little as 1% in a well-designed venturi meter. The effects of deviation from the idealized Bernoulli flow can be accounted for by expressing Eq. (3) as

$$\dot{V}=C_cA_2\sqrt{\frac{2(P_1-P_2)}{\rho[1-(A_2/A_1)^2]}}$$

where C_c is the *venturi discharge coefficient* whose value is less than 1 (it is as large as 0.99 for well-designed venturi meters in certain ranges of flow). For Re > 10^5, the value of venturi discharge coefficient is usually greater than 0.96.

5-63

Solution The water height in an airtight pressurized tank is given. A hose pointing straight up is connected to the bottom of the tank. The maximum height to which the water stream could rise is to be determined.



Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The friction between the water and air is negligible.

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take point 1 at the free surface of water in the tank, and point 2 at the top of the water trajectory. Also, we take the reference level at the bottom of the tank. At the top of the water trajectory $\mathbf{V}_2 = 0$, and atmospheric pressure pertains. Noting that z_1 = 20 m, $P_{1,\text{gage}}$ = 2 atm, $P_2 = P_{\text{atm}}$, and that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$), the Bernoulli equation between these two points simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}+z_1=\frac{P_{atm}}{\rho g}+z_2 \quad\rightarrow\quad z_2=\frac{P_1-P_{atm}}{\rho g}+z_1=\frac{P_{1,\text{gage}}}{\rho g}+z_1$$

Substituting,

$$z_2=\frac{3\,\text{atm}}{(1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)}\left(\frac{101{,}325\,\text{N/m}^2}{1\,\text{atm}}\right)\left(\frac{1\,\text{kg}\cdot\text{m/s}^2}{1\,\text{N}}\right)+15=\mathbf{46.0\,m}$$

Therefore, the water jet can rise as high as 46.0 m into the sky from the ground.

Discussion The result obtained by the Bernoulli equation represents the upper limit, and should be interpreted accordingly. It tells us that the water cannot possibly rise more than 46.0 m (giving us an upper limit), and in all likelihood, the rise will be much less because of frictional losses.

5-64

Solution A Pitot-static probe equipped with a water manometer is held parallel to air flow, and the differential height of the water column is measured. The flow velocity of air is to be determined.

Assumptions **1**The flow of air is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The effect of air column on the pressure change is negligible because of its low density, and thus the air column in the manometer can be ignored.

Properties We take the density of water to be ρ = 1000 kg/m^3. The density of air is given to be 1.16 kg/m^3.

Analysis We take point 1 on the side of the probe where the entrance is parallel to flow and is connected to the static arm of the Pitot-static probe, and point 2 at the tip of the probe where the entrance is normal to flow and is connected to the dynamic arm of the Pitot-static probe. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

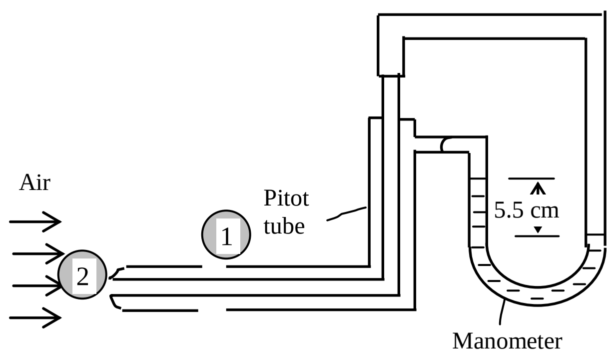
$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}+\frac{V_1^2}{2g}=\frac{P_2}{\rho g} \quad\rightarrow\quad V_1=\sqrt{\frac{2(P_2-P_1)}{\rho_{air}}} \qquad (1)$$

The pressure rise at the tip of the Pitot-static probe is simply the pressure change indicated by the differential water column of the manometer,

$$P_2-P_1=\rho_{\text{water}}gh \qquad (2)$$

Combining Eqs. (1) and (2) and substituting, the flow velocity is determined to be

$$V_1=\sqrt{\frac{2\rho_{\text{water}}gh}{\rho_{\text{air}}}}=\sqrt{\frac{2(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.055\text{ m})}{1.16\text{ kg/m}^3}}=\mathbf{30.5\ m/s}$$



Discussion Note that flow velocity in a pipe or duct can be measured easily by a Pitot-static probe by inserting the probe into the pipe or duct parallel to flow, and reading the differential height. Also note that this is the velocity at the location of the tube. Several readings at several locations in a cross-section may be required to determine the mean flow velocity.

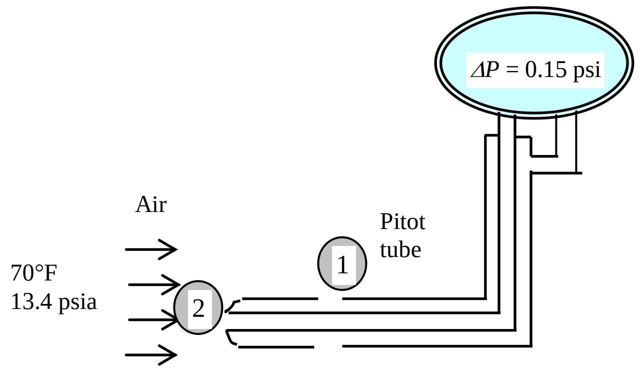
5-65E

Solution A Pitot-static probe equipped with a differential pressure gage is used to measure the air velocity in a duct. For a given differential pressure reading, the flow velocity of air is to be determined.

Assumptions The flow of air is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Properties The gas constant of air is R = 0.3704 psia·ft^3/lbm·R.

Analysis We take point 1 on the side of the probe where the entrance is parallel to flow and is connected to the static arm of the Pitot-static probe, and point 2 at the tip of the probe where the entrance is normal to flow and is connected to the dynamic arm of the Pitot-static probe. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives



$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}+\frac{V_1^2}{2g}=\frac{P_2}{\rho g} \quad\rightarrow\quad V_1=\sqrt{\frac{2(P_2-P_1)}{\rho}}$$

where

$$\rho=\frac{P}{RT}=\frac{13.4\text{ psia}}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(70+460\text{ R})}=0.0683\text{ lbm/ft}^3$$

Substituting the given values, the flow velocity is determined to be

$$V_1=\sqrt{\frac{2(0.15\text{ psi})}{0.0683\text{ lbm/ft}^3}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)}=\mathbf{143\text{ ft/s}}$$

Discussion Note that flow velocity in a pipe or duct can be measured easily by a Pitot-static probe by inserting the probe into the pipe or duct parallel to flow, and reading the pressure differential. Also note that this is the velocity at the location of the tube. Several readings at several locations in a cross-section may be required to determine the mean flow velocity.

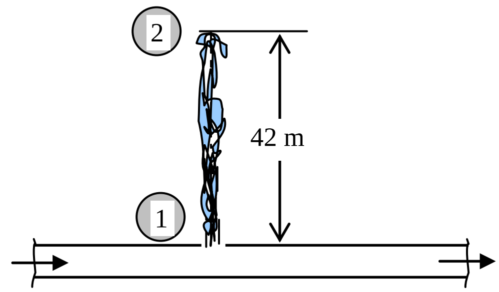
5-66

Solution A water pipe bursts as a result of freezing, and water shoots up into the air a certain height. The gage pressure of water in the pipe is to be determined.

Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The water pressure in the pipe at the burst section is equal to the water main pressure. **3** Friction between the water and air is negligible. **4** The irreversibilities that may occur at the burst section of the pipe due to abrupt expansion are negligible.

Properties We take the density of water to be 1000 kg/m^3.

Analysis This problem involves the conversion of flow, kinetic, and potential energies to each other without involving any pumps, turbines, and wasteful components with large frictional losses, and thus it is suitable for the use of the Bernoulli equation. The water height will be maximum under the stated assumptions. The velocity inside the hose is relatively low ($V_1 \cong 0$) and we take the burst section of the pipe as the reference level ($z_1 = 0$). At the top of the water trajectory $V_2 = 0$, and atmospheric pressure pertains. Then the Bernoulli equation simplifies to



Water Pipe

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}=\frac{P_{atm}}{\rho g}+z_2 \quad\rightarrow\quad \frac{P_1-P_{atm}}{\rho g}=z_2 \quad\rightarrow\quad \frac{P_{1,\text{gage}}}{\rho g}=z_2$$

Solving for $P_{1,\text{gage}}$ and substituting,

$$P_{1,\text{gage}}=\rho g z_2=(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(42\ \text{m})\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right)\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)=\mathbf{412\ kPa}$$

Therefore, the pressure in the main must be at least 412 kPa above the atmospheric pressure.

Discussion The result obtained by the Bernoulli equation represents a limit, since frictional losses are neglected, and should be interpreted accordingly. It tells us that the water pressure (gage) cannot possibly be less than 334 kPa (giving us a lower limit), and in all likelihood, the pressure will be much higher.

5-67

Solution A well-fitting piston with 4 small holes in a sealed water-filled cylinder is pushed to the right at a constant speed while the pressure in the right compartment remains constant. The force that needs to be applied to the piston to maintain this motion is to be determined.

Analysis When frictional effects are negligible, the pressures on both sides of the piston become identical:

$$F=P_{cyl,gage}A_p=P_{cyl,gage}\frac{\pi d_p^2}{4}=(50\ \text{kPa})\frac{\pi(0.12\ \text{m})^2}{4}=\mathbf{0.565\ kN}$$

5-68

Solution We are to generate expressions for inlet pressure and throat pressure and velocity in a converging-diverging duct for the case where P_{outlet} and V_{inlet} are known and irreversibilities are ignored.

Assumptions **1** The flow is steady, incompressible, and two-dimensional. **2** We neglect irreversibilities such as friction. **3** **4** The duct is horizontal – elevation differences do not play a role in this analysis.

Analysis (*a*) We apply conservation of mass from the inlet to the outlet:

$$\rho V_{\text{inlet}} A_{\text{inlet}} = \rho V_{\text{outlet}} A_{\text{outlet}} \quad \rightarrow \quad V_{\text{outlet}} = V_{\text{inlet}} \frac{A_{\text{inlet}}}{A_{\text{outlet}}}$$

where density has dropped out because of the incompressible flow approximation. We do a similar analysis at the throat. Thus, the average inlet velocity and average throat velocity are

$$\boxed{V_{\text{outlet}} = V_{\text{inlet}} \frac{A_{\text{inlet}}}{A_{\text{outlet}}} \qquad V_{\text{throat}} = V_{\text{inlet}} \frac{A_{\text{inlet}}}{A_{\text{throat}}}}$$

To estimate the pressure at inlet, we neglect irreversibilities and apply the Bernoulli equation along a streamline from the inlet to the outlet,

$$P_{\text{inlet}} + \frac{1}{2}\rho V_{\text{inlet}}^{\ 2} + \cancel{\rho g z_{\text{inlet}}} = P_{\text{outlet}} + \frac{1}{2}\rho V_{\text{outlet}}^{\ 2} + \cancel{\rho g z_{\text{outlet}}} \quad \rightarrow \quad P_{\text{inlet}} = P_{\text{outlet}} + \frac{1}{2}\rho\left(V_{\text{outlet}}^{\ 2} - V_{\text{inlet}}^{\ 2}\right)$$

and when we substitute the outlet velocity from conservation of mass we get

$$\boxed{P_{\text{inlet}} = P_{\text{outlet}} + \frac{1}{2}\rho\left(\left(V_{\text{inlet}} \frac{A_{\text{inlet}}}{A_{\text{outlet}}}\right)^2 - V_{\text{inlet}}^{\ 2}\right) = \frac{1}{2}\rho V_{\text{inlet}}^{\ 2}\left(\left(\frac{A_{\text{inlet}}}{A_{\text{throat}}}\right)^2 - 1\right)}$$

In like manner, we calculate the average pressure at any other axial location where the cross-sectional area is known. At the throat, for example,

$$P_{\text{throat}} = P_{\text{outlet}} + \frac{1}{2}\rho\left(V_{\text{outlet}}^{\ 2} - V_{\text{throat}}^{\ 2}\right) = P_{\text{outlet}} + \frac{1}{2}\rho\left(\left(V_{\text{inlet}} \frac{A_{\text{inlet}}}{A_{\text{outlet}}}\right)^2 - \left(V_{\text{inlet}} \frac{A_{\text{inlet}}}{A_{\text{throat}}}\right)^2\right)$$

or, combining some terms,

$$\boxed{P_{\text{throat}} = P_{\text{outlet}} + \frac{1}{2}\rho\left(V_{\text{inlet}} A_{\text{inlet}}\right)^2\left(\left(\frac{1}{A_{\text{outlet}}}\right)^2 - \left(\frac{1}{A_{\text{throat}}}\right)^2\right)}$$

(*b*) In this analysis, we have not accounted for any irreversibilities, such as friction, but in any real flow, friction would lead to higher pressure drop in the duct and thus **the inlet pressure would have to be higher than predicted in order to overcome the additional losses due to friction**.

Discussion We must keep in mind that the Bernoulli equation is only an *approximation*. In Chap. 8 we learn how to approximate the additional pressure drop due to friction along the walls of a duct or pipe.

Energy Equation

5-69C

Solution We are to define and discuss useful pump head.

Analysis *Useful pump head* is the **useful power input to the pump expressed as an equivalent column height of fluid**. It is related to the useful pumping power input by $\boxed{h_{\text{pump}} = \frac{w_{\text{pump,u}}}{g} = \frac{\dot{W}_{\text{pump,u}}}{\dot{m}g}}$.

Discussion Part of the power supplied to the pump is *not* useful, but rather is wasted because of irreversible losses in the pump. This is the reason that pumps have a pump efficiency that is always less than one.

5-70C

Solution We are to analyze whether temperature can decrease during steady adiabatic flow of an incompressible fluid.

Analysis **It is *impossible* for the fluid temperature to decrease** during steady, incompressible, adiabatic flow of an incompressible fluid, since this would require the entropy of an adiabatic system to decrease, which would be a violation of the 2$^{\text{nd}}$ law of thermodynamics.

Discussion The entropy of a fluid *can* decrease, but only if we remove heat.

5-71C

Solution We are to define and discuss irreversible head loss.

Analysis *Irreversible head loss* is the **loss of mechanical energy due to irreversible processes** (such as friction) **in piping expressed as an equivalent column height of fluid**, i.e., head. Irreversible head loss is related to the mechanical energy loss in piping by $\boxed{h_L = \frac{e_{\text{mech loss,piping}}}{g} = \frac{\dot{E}_{\text{mech loss,piping}}}{\dot{m}g}}$.

Discussion h_L is always positive. It can never be negative, since this would violate the second law of thermodynamics.

5-72C

Solution We are to determine if frictional effects are negligible in the steady adiabatic flow of an incompressible fluid if the temperature remains constant.

Analysis **Yes, the *frictional effects* are negligible** if the fluid temperature remains constant during steady, incompressible flow since any irreversibility such as friction would cause the entropy and thus temperature of the fluid to increase during adiabatic flow.

Discussion Thus, this scenario would never occur in real life since all fluid flows have frictional effects.

5-73C

Solution We are to define and discuss the kinetic energy correction factor.

Analysis The *kinetic energy correction factor* is a **correction factor to account for the fact that kinetic energy using average velocity is not the same as the actual kinetic energy using the actual velocity profile** (the square of a sum is not equal to the sum of the squares of its components). The effect of kinetic energy factor is usually negligible, especially for turbulent pipe flows. However, for laminar pipe flows, the effect of α is sometimes significant.

Discussion Even though the effect of ignoring α is usually insignificant, it is wise to keep α in our analyses to increase accuracy and so that we do not forget about it in situations where it *is* significant, such as in some laminar pipe flows.

5-74C

Solution We are to analyze the cause of some strange behavior of a water jet.

Analysis The problem does not state whether the water in the tank is open to the atmosphere or not. Let's assume that the water surface *is* exposed to atmospheric pressure. By the Bernoulli equation, the maximum theoretical height to which the water stream could rise is the tank water level, which is 20 meters above the ground. Since the water rises *above* the tank level, **the tank cover must be airtight, containing pressurized air above the water surface**. In other words, the water in the tank is *not* exposed to atmospheric pressure.

Discussion Alternatively, a pump would have to pressurize the water somewhere in the hose, but this is not allowed, based on the problem statement (only a hose is added).

5-75C

Solution We are to analyze a suggestion regarding a garden hose.

Analysis **Yes**. When water discharges from the hose at waist level, the head corresponding to the waist-knee vertical distance is wasted. When recovered, this elevation head is converted to velocity head, increasing the discharge velocity (and thus the flow rate) of water and thus reducing the filling time.

Discussion If you are still not convinced, imagine holding the hose outlet really high up. If the outlet elevation is greater than the upstream supply head, no water will flow at all. If you are concerned about head losses in the hose, yes, they will increase as the volume flow rate increases, but not enough to change our answer.

5-76C

Solution We are to analyze discharge of water from a tank under different conditions.

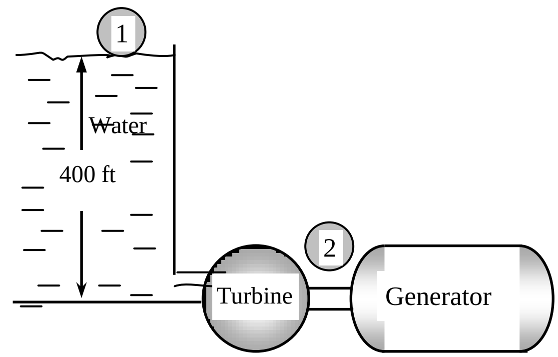
Analysis (*a*) **Yes**, the discharge velocity from the bottom valve will be higher since velocity is proportional to the square root of the vertical distance between the hole and the free surface. (*b*) **No**, the discharge rates of water will be the same since the total available head to drive the flow (elevation difference between the ground and the free surface of water in the tank) is the same for both cases.

Discussion Our answer to Part (b) does not change even if we consider head losses in the hose, because the hose is the same length in either case. Same hose, same length, same flow rate…yields the same head loss through the hose. *Note*: We are ignoring any effects of bends or curves in the hose – assume both cases have the same curves.

5-77E

Solution In a hydroelectric power plant, the elevation difference, the power generation, and the overall turbine-generator efficiency are given. The minimum flow rate required is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water levels at the reservoir and the discharge site remain constant. **3** We assume the flow to be *frictionless* since the *minimum* flow rate is to be determined, $\dot{E}_{\text{mech,loss}} = 0$.



Properties We take the density of water to be ρ = 62.4 lbm/ft^3.

Analysis We take point 1 at the free surface of the reservoir and point 2 at the free surface of the discharge water stream, which is also taken as the reference level ($z_2 = 0$). Also, both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$), the velocities are negligible at both points ($V_1 = V_2 = 0$), and frictional losses are disregarded. Then the energy equation in terms of heads for steady incompressible flow through a control volume between these two points that includes the turbine and the pipes reduces to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad h_{\text{turbine,e}} = z_1$$

Substituting and noting that $\dot{W}_{\text{turbine,elect}} = \eta_{\text{turbine-gen}} \dot{m} g h_{\text{turbine,e}}$, the extracted turbine head and the mass and volume flow rates of water are determined to be

$$h_{\text{turbine,e}} = z_1 = 400 \text{ ft}$$

$$\dot{m} = \frac{\dot{W}_{\text{turbine,elect}}}{\eta_{\text{turbine-gen}} g h_{\text{turbine}}} = \frac{100 \text{ kW}}{0.85(32.2 \text{ ft/s}^2)(400 \text{ ft})}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)\left(\frac{0.9478 \text{ Btu/s}}{1 \text{ kW}}\right) = 216.8 \text{ lbm/s} \cong \mathbf{217 \text{ lbm/s}}$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{216.8 \text{ lbm/s}}{62.4 \text{ lbm/ft}^3} = 3.47 \text{ ft}^3/\text{s}$$

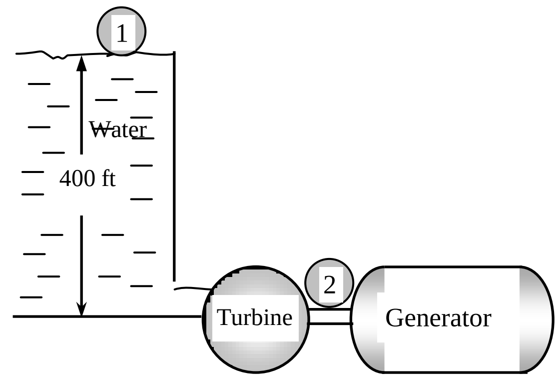
Therefore, the flow rate of water must be at least 3.47 ft^3/s to generate the desired electric power while overcoming friction losses in pipes.

Discussion In an actual system, the flow rate of water will be more because of frictional losses in pipes.

5-78E

Solution In a hydroelectric power plant, the elevation difference, the head loss, the power generation, and the overall turbine-generator efficiency are given. The flow rate required is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water levels at the reservoir and the discharge site remain constant.



Properties We take the density of water to be ρ = 62.4 lbm/ft^3.

Analysis We take point 1 at the free surface of the reservoir and point 2 at the free surface of the discharge water stream, which is also taken as the reference level (z_2 = 0). Also, both 1 and 2 are open to the atmosphere (P_1 = P_2 = P_{atm}), the velocities are negligible at both points (V_1 = V_2 = 0). Then the energy equation in terms of heads for steady incompressible flow through a control volume between these two points that includes the turbine and the pipes reduces to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{turbine,e}}=z_1-h_L$$

Substituting and noting that $\dot{W}_{\text{turbine,elect}}=\eta_{\text{turbine-gen}}\dot{m}gh_{\text{turbine,e}}$, the extracted turbine head and the mass and volume flow rates of water are determined to be

$$h_{\text{turbine,e}}=z_1-h_L=400-36=364\text{ ft}$$

$$\dot{m}=\frac{\dot{W}_{\text{turbine,elect}}}{\eta_{\text{turbine-gen}}gh_{\text{turbine}}}=\frac{100\text{ kW}}{0.85(32.2\text{ ft/s}^2)(364\text{ ft})}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)\left(\frac{0.9478\text{ Btu/s}}{1\text{ kW}}\right)=\mathbf{238\ lbm/s}$$

$$\dot{V}=\frac{\dot{m}}{\rho}=\frac{238\text{ lbm/s}}{62.4\text{ lbm/ft}^3}=3.82\text{ ft}^3/\text{s}$$

Therefore, the flow rate of water must be at least 3.82 ft^3/s to generate the desired electric power while overcoming friction losses in pipes.

Discussion Note that the effect of frictional losses in the pipes is to increase the required flow rate of water to generate a specified amount of electric power.

5-79

Solution A pump is pumping oil at a specified rate. The pressure rise of oil in the pump is measured, and the motor efficiency is specified. The mechanical efficiency of the pump is to be determined.

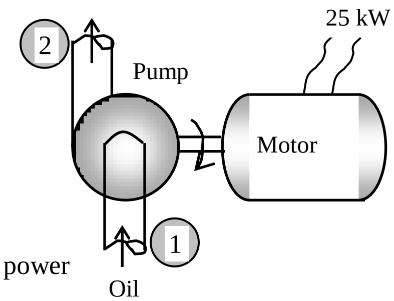
Assumptions **1** The flow is steady and incompressible. **2** The elevation difference across the pump is negligible. **3** All the losses in the pump are accounted for by the pump efficiency and thus $h_L = 0$. **4** The kinetic energy correction factors are given to be $\alpha_1 = \alpha_2 = \alpha = 1.05$.

Properties The density of oil is given to be $\rho = 860$ kg/m^3.

Analysis We take points 1 and 2 at the inlet and the exit of the pump, respectively. Noting that $z_1 = z_2$, the energy equation for the pump reduces to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=\frac{P_2-P_1}{\rho g}+\frac{\alpha(V_2^2-V_1^2)}{2g}$$

where



$$V_1=\frac{\dot{V}}{A_1}=\frac{\dot{V}}{\pi D_1^2/4}=\frac{0.1\,\text{m}^3/\text{s}}{\pi(0.08\,\text{m})^2/4}=19.9\text{ m/s}$$

$$V_2=\frac{\dot{V}}{A_2}=\frac{\dot{V}}{\pi D_2^2/4}=\frac{0.1\,\text{m}^3/\text{s}}{\pi(0.12\,\text{m})^2/4}=8.84\text{ m/s}$$

Substituting, the useful pump head and the corresponding useful pumping power are determined to be

$$h_{\text{pump,u}}=\frac{250{,}000\text{ N/m}^2}{(860\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)+\frac{1.05[(8.84\text{ m/s})^2-(19.9\text{ m/s})^2]}{2(9.81\text{ m/s}^2)}=29.6-17.0=12.6\text{ m}$$

$$\dot{W}_{\text{pump,u}}=\rho\dot{V}gh_{\text{pump,u}}=(860\text{ kg/m}^3)(0.1\text{ m}^3/\text{s})(9.81\text{ m/s}^2)(12.6\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right)=10.6\text{ kW}$$

Then the shaft pumping power and the mechanical efficiency of the pump become

$$\dot{W}_{\text{pump,shaft}}=\eta_{\text{motor}}\dot{W}_{\text{electric}}=(0.90)(25\text{ kW})=22.5\text{ kW}$$

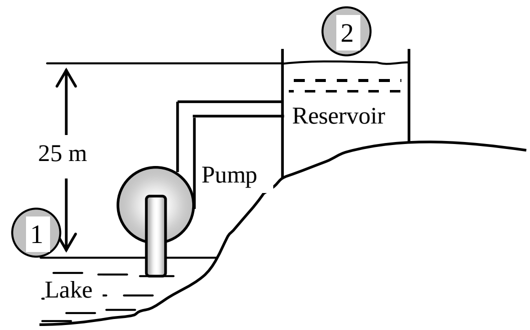
$$\eta_{\text{pump}}=\frac{\dot{W}_{\text{pump,u}}}{\dot{W}_{\text{pump,shaft}}}=\frac{10.6\text{ kW}}{22.5\text{ kW}}=0.471=\mathbf{47.1\%}$$

Discussion The overall efficiency of this pump/motor unit is the product of the mechanical and motor efficiencies, which is $0.9 \times 0.471 = 0.42$.

5-80

Solution Water is pumped from a large lake to a higher reservoir. The head loss of the piping system is given. The mechanical efficiency of the pump is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the lake and the reservoir is constant.



Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis We choose points 1 and 2 at the free surfaces of the lake and the reservoir, respectively, and take the surface of the lake as the reference level ($z_1 = 0$). Both points are open to the atmosphere ($P_1 = P_2 = P_{atm}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\dot{m}\left(\frac{P_1}{\rho}+\alpha_1\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{\text{pump}}=\dot{m}\left(\frac{P_2}{\rho}+\alpha_2\frac{V_2^2}{2}+gz_2\right)+\dot{W}_{\text{turbine}}+\dot{E}_{\text{mech,loss}} \rightarrow \dot{W}_{\text{pump,u}}=\dot{m}gz_2+\dot{E}_{\text{mech loss,piping}}$$

since, in the absence of a turbine, $\dot{E}_{\text{mech, loss}}=\dot{E}_{\text{mech loss,pump}}+\dot{E}_{\text{mech loss,piping}}$ and $\dot{W}_{\text{pump,u}}=\dot{W}_{\text{pump}}-\dot{E}_{\text{mech loss,pump}}$. Noting that $\dot{E}_{\text{mech loss,piping}}=\dot{m}gh_L$, the useful pump power is

$$\begin{aligned}\dot{W}_{\text{pump,u}} &= \dot{m}gz_2+\dot{m}gh_L = \rho\dot{V}g(z_2+h_L)\\ &= (1000\ \text{kg/m}^3)(0.025\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)[(25+5)\ \text{m}]\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\\ &= 7.36\ \text{kN}\cdot\text{m/s} = 7.36\ \text{kW}\end{aligned}$$

Then the mechanical efficiency of the pump becomes

$$\eta_{\text{pump}}=\frac{\dot{W}_{\text{pump,u}}}{\dot{W}_{\text{shaft}}}=\frac{7.36\ \text{kW}}{10\ \text{kW}}=0.736=\mathbf{73.6\%}$$

Discussion A more practical measure of performance of the pump is the overall efficiency, which can be obtained by multiplying the pump efficiency by the motor efficiency.

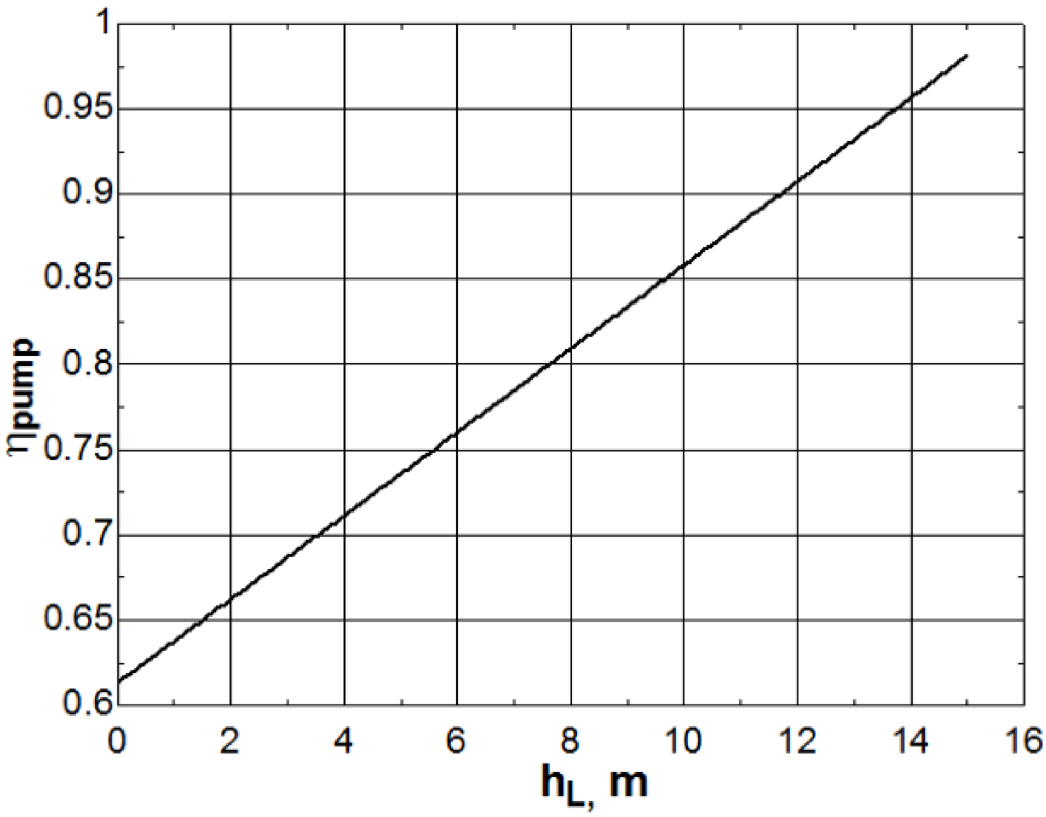
5-81

Solution The previous problem is reconsidered. The effect of head loss on mechanical efficiency of the pump. as the head loss varies 0 to 20 m in increments of 2 m is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
z2=25 "m"
W_shaft=10 "kW"
V_dot=0.025 "m3/s"
W_pump_u=rho*V_dot*g*(z2+h_L)/1000 "kW"
Eta_pump=W_pump_u/W_shaft
```

Head Loss, h_L, m	Pumping power $W_{\text{pump, u}}$	Efficiency η_{pump}
0	6.13	0.613
1	6.38	0.638
2	6.62	0.662
3	6.87	0.687
4	7.11	0.711
5	7.36	0.736
6	7.60	0.760
7	7.85	0.785
8	8.09	0.809
9	8.34	0.834
10	8.58	0.858
11	8.83	0.883
12	9.07	0.907
13	9.32	0.932
14	9.56	0.956
15	9.81	0.981



Discussion Note that the useful pumping power is used to raise the fluid and to overcome head losses. For a given power input, the pump that overcomes more head loss is more efficient.

5-82

Solution A pump with a specified shaft power and efficiency is used to raise water to a higher elevation. The maximum flow rate of water is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the reservoirs is constant. **3** We assume the flow in the pipes to be *frictionless* since the *maximum* flow rate is to be determined, $\dot{E}_{\text{mech loss,piping}} = 0$.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis We choose points 1 and 2 at the free surfaces of the lower and upper reservoirs, respectively, and take the surface of the lower reservoir as the reference level ($z_1 = 0$). Both points are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\dot{m}\left(\frac{P_1}{\rho}+\alpha_1\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{\text{pump}}=\dot{m}\left(\frac{P_2}{\rho}+\alpha_2\frac{V_2^2}{2}+gz_2\right)+\dot{W}_{\text{turbine}}+\dot{E}_{\text{mech,loss}} \;\rightarrow\; \dot{W}_{\text{pump,u}}=\dot{m}gz_2=\rho\dot{V}gz_2$$

since $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss,pump}}$ in this case and $\dot{W}_{\text{pump,u}} = \dot{W}_{\text{pump}} - \dot{E}_{\text{mech loss,pump}}$.

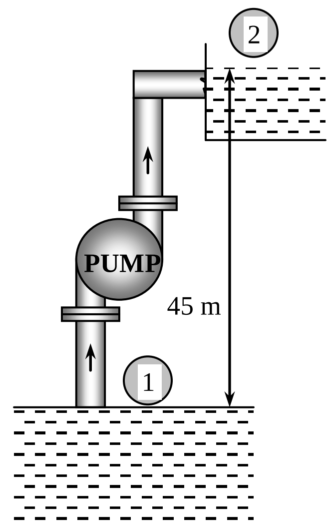
The useful pumping power is

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump}}\dot{W}_{\text{pump,shaft}} = (0.82)(15\text{ hp}) = 12.3\text{ hp}$$

Substituting, the volume flow rate of water is determined to be

$$\dot{V} = \frac{\dot{W}_{\text{pump,u}}}{\rho g z_2} = \frac{12.3\text{ hp}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(45\text{ m})}\left(\frac{745.7\text{ W}}{1\text{ hp}}\right)\left(\frac{1\text{ N}\cdot\text{m/s}}{1\text{ W}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)$$

$$= \mathbf{0.0208\ m^3/s}$$



Discussion This is the maximum flow rate since the frictional effects are ignored. In an actual system, the flow rate of water will be less because of friction in pipes.

5-83

Solution Water flows at a specified rate in a horizontal pipe whose diameter is decreased by a reducer. The pressures are measured before and after the reducer. The head loss in the reducer is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The pipe is horizontal. **3** The kinetic energy correction factors are given to be $\alpha_1 = \alpha_2 = \alpha = 1.05$.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

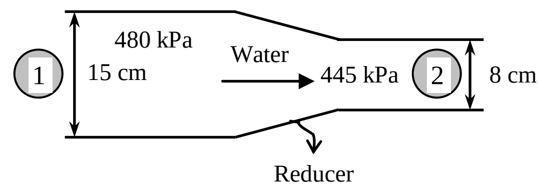
Analysis We take points 1 and 2 along the centerline of the pipe before and after the reducer, respectively. Noting that $z_1 = z_2$, the energy equation for steady incompressible flow through a control volume between these two points reduces to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_L=\frac{P_1-P_2}{\rho g}+\frac{\alpha(V_1^2-V_2^2)}{2g}$$

where

$$V_1=\frac{\dot{V}}{A_1}=\frac{\dot{V}}{\pi D_1^2/4}=\frac{0.035\ \text{m}^3/\text{s}}{\pi(0.15\ \text{m})^2/4}=1.981\ \text{m/s}$$

$$V_2=\frac{\dot{V}}{A_2}=\frac{\dot{V}}{\pi D_2^2/4}=\frac{0.035\ \text{m}^3/\text{s}}{\pi(0.08\ \text{m})^2/4}=6.963\ \text{m/s}$$



Substituting, the head loss in the reducer is determined to be

$$h_L=\frac{(480-445)\ \text{kPa}}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right)\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)+\frac{1.05[(1.981\ \text{m/s})^2-(6.963\ \text{m/s})^2]}{2(9.81\ \text{m/s}^2)}$$
$$=1.183\ \text{m}\cong\mathbf{1.18\ m}$$

Discussion Note that the 1.19 m of the head loss is due to frictional effects and 2.27 m is due to the increase in velocity. This head loss corresponds to a power potential loss of

$$\dot{E}_{\text{mech loss, piping}}=\rho\dot{V}gh_L=(1000\ \text{kg/m}^3)(0.035\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(1.19\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{W}}{1\ \text{N}\cdot\text{m/s}}\right)=406\ \text{W}$$

5-84

Solution A hose connected to the bottom of a tank is equipped with a nozzle at the end pointing straight up. The water is pressurized by a pump, and the height of the water jet is measured. The minimum pressure rise supplied by the pump is to be determined.

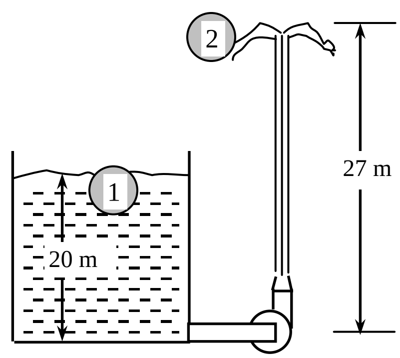
Assumptions **1** The flow is steady and incompressible. **2** Friction between the water and air as well as friction in the hose is negligible. **3** The water surface is open to the atmosphere.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis We take point 1 at the free surface of water in the tank, and point 2 at the top of the water trajectory where $V_2 = 0$ and $P_1 = P_2 = P_{atm}$. Also, we take the reference level at the bottom of the tank. Noting that $z_1 = 20$ m and $z_2 = 27$ m, $h_L = 0$ (to get the minimum value for required pressure rise), and that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$), the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the water stream reduces to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L$$

$$\rightarrow \quad h_{\text{pump,u}} = z_2 - z_1$$



Substituting,

$$h_{\text{pump,u}} = 27 - 20 = 7 \text{ m}$$

A water column height of 7 m corresponds to a pressure rise of

$$\Delta P_{\text{pump,min}} = \rho g h_{\text{pump,u}} = (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(7 \text{ m})\left(\frac{1 \text{ N}}{1000 \text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 68.7 \text{ kN/m}^2 = \mathbf{68.7 kPa}$$

Therefore, the pump must supply a minimum pressure rise of 68.7 kPa.

Discussion The result obtained above represents the minimum value, and should be interpreted accordingly. In reality, a larger pressure rise will need to be supplied to overcome friction.

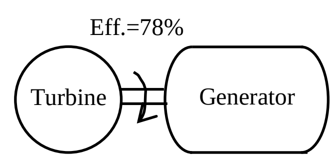
5-85

Solution The available head of a hydraulic turbine and its overall efficiency are given. The electric power output of this turbine is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The available head remains constant.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis When the turbine head is available, the corresponding power output is determined from



$$\dot{W}_{\text{turbine}} = \eta_{\text{turbine}} \dot{m} g h_{\text{turbine}} = \eta_{\text{turbine}} \rho \dot{V} g h_{\text{turbine}}$$

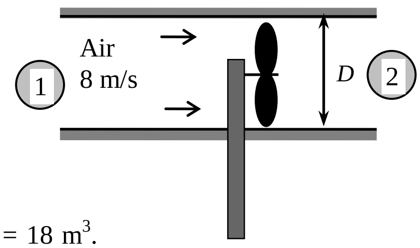
Substituting,

$$\dot{W}_{\text{turbine}} = 0.78(1000 \text{ kg/m}^3)(1.30 \text{ m}^3/\text{s})(9.81 \text{ m/s}^2)(50 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kW}}{1000 \text{ N}\cdot\text{m/s}}\right) = \mathbf{497 kW}$$

Discussion The power output of a hydraulic turbine is proportional to the available turbine head and the flow rate.

5-86

Solution A fan is to ventilate a bathroom by replacing the entire volume of air once every 10 minutes while air velocity remains below a specified value. The wattage of the fan-motor unit, the diameter of the fan casing, and the pressure difference across the fan are to be determined.



Assumptions **1** The flow is steady and incompressible. **2** Frictional losses along the flow (other than those due to the fan-motor inefficiency) are negligible. **3** The fan unit is horizontal so that z = constant along the flow (or, the elevation effects are negligible because of the low density of air). **4** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

Properties The density of air is given to be 1.25 kg/m^3.

Analysis (*a*) The volume of air in the bathroom is $\boldsymbol{V} = 2\text{ m} \times 3\text{ m} \times 3\text{ m} = 18\text{ m}^3$. Then the volume and mass flow rates of air through the casing must be

$$\dot{\boldsymbol{V}} = \frac{\boldsymbol{V}}{\Delta t} = \frac{18\,\text{m}^3}{10 \times 60\,\text{s}} = 0.03\,\text{m}^3/\text{s}$$

$$\dot{m} = \rho \dot{V} = (1.25\,\text{kg/m}^3)(0.03\,\text{m}^3/\text{s}) = 0.0375\ \text{kg/s}$$

We take points 1 and 2 on the inlet and exit sides of the fan, respectively. Point 1 is sufficiently far from the fan so that $P_1 = P_{atm}$ and the flow velocity is negligible ($V_1 = 0$). Also, $P_2 = P_{atm}$. Then the energy equation for this control volume between the points 1 and 2 reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1 \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2 \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech,loss}} \quad \rightarrow \quad \dot{W}_{\text{fan, u}} = \dot{m}\alpha_2 \frac{V_2^2}{2}$$

since $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss,pump}}$ in this case and $\dot{W}_{\text{pump,u}} = \dot{W}_{\text{pump}} - \dot{E}_{\text{mech loss,pump}}$. Substituting,

$$\dot{W}_{\text{fan, u}} = \dot{m}\alpha_2 \frac{V_2^2}{2} = (0.0375\ \text{kg/s})(1.0)\frac{(8\ \text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{W}}{1\,\text{N}\cdot\text{m/s}}\right) = 1.2\,\text{W}$$

and $$\dot{W}_{\text{fan, elect}} = \frac{\dot{W}_{\text{fan, u}}}{\eta_{\text{fan-motor}}} = \frac{1.2\ \text{W}}{0.5} = \mathbf{2.4\,W}$$

Therefore, the electric power rating of the fan/motor unit must be 2.4 W.

(*b*) For air mean velocity to remain below the specified value, the diameter of the fan casing should be

$$\dot{\boldsymbol{V}} = A_2 V_2 = (\pi D_2^2/4)V_2 \quad \rightarrow \quad D_2 = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V_2}} = \sqrt{\frac{4(0.03\,\text{m}^3/\text{s})}{\pi(8\ \text{m/s})}} = 0.069\,\text{m} = \mathbf{6.9\,cm}$$

(*c*) To determine the pressure difference across the fan unit, we take points 3 and 4 to be on the two sides of the fan on a horizontal line. Noting that $z_3 = z_4$ and $V_3 = V_4$ since the fan is a narrow cross-section and neglecting flow loses (other than the loses of the fan unit, which is accounted for by the efficiency), the energy equation for the fan section reduces to

$$\dot{m}\frac{P_3}{\rho} + \dot{W}_{\text{fan, u}} = \dot{m}\frac{P_4}{\rho} \quad \rightarrow \quad P_4 - P_3 = \frac{\dot{W}_{\text{fan, u}}}{\dot{m}/\rho} = \frac{\dot{W}_{\text{fan, u}}}{\dot{V}}$$

Substituting , $$P_4 - P_3 = \frac{1.2\ \text{W}}{0.03\,\text{m}^3/\text{s}}\left(\frac{1\,\text{N}\cdot\text{m/s}}{1\,\text{W}}\right) = 40\ \text{N/m}^2 = \mathbf{40\,Pa}$$

Therefore, the fan will raise the pressure of air by 40 Pa before discharging it.

Discussion Note that only half of the electric energy consumed by the fan-motor unit is converted to the mechanical energy of air while the remaining half is converted to heat because of imperfections.

5-87

Solution Water flows through a horizontal pipe at a specified rate. The pressure drop across a valve in the pipe is measured. The corresponding head loss and the power needed to overcome it are to be determined.

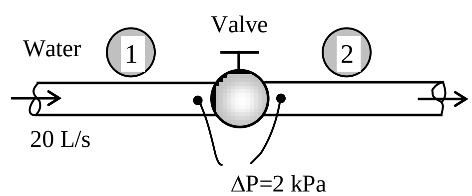
Assumptions **1** The flow is steady and incompressible. **2** The pipe is given to be horizontal (otherwise the elevation difference across the valve is negligible). **3** The mean flow velocities at the inlet and the exit of the valve are equal since the pipe diameter is constant.

Properties We take the density of water to be ρ = 1000 kg/m^3.

Analysis We take the valve as the control volume, and points 1 and 2 at the inlet and exit of the valve, respectively. Noting that $z_1 = z_2$ and $V_1 = V_2$, the energy equation for steady incompressible flow through this control volume reduces to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_L=\frac{P_1-P_2}{\rho g}$$

Substituting,



$$h_L=\frac{2\ \text{kN/m}^2}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)=\mathbf{0.204m}$$

The useful pumping power needed to overcome this head loss is

$$\dot{W}_{\text{pump,u}}=\dot{m}gh_L=\rho\dot{V}gh_L$$

$$=(1000\ \text{kg/m}^3)(0.020\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(0.204\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{W}}{1\ \text{N}\cdot\text{m/s}}\right)=\mathbf{40\ W}$$

Therefore, this valve would cause a head loss of 0.204 m, and it would take 40 W of useful pumping power to overcome it.

Discussion The required useful pumping power could also be determined from

$$\dot{W}_{\text{pump}}=\dot{V}\Delta P=(0.020\ \text{m}^3/\text{s})(2000\ \text{Pa})\left(\frac{1\ \text{W}}{1\ \text{Pa}\cdot\text{m}^3/\text{s}}\right)=\mathbf{40\ W}$$

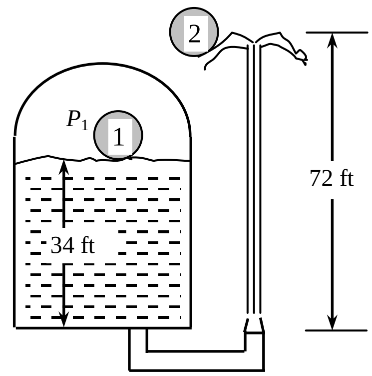
5-88E

Solution A hose connected to the bottom of a pressurized tank is equipped with a nozzle at the end pointing straight up. The minimum tank air pressure (gage) corresponding to a given height of water jet is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Friction between water and air as well as friction in the hose is negligible. **3** The water surface is open to the atmosphere.

Properties We take the density of water to be $\rho = 62.4$ lbm/ft^3.

Analysis We take point 1 at the free surface of water in the tank, and point 2 at the top of the water trajectory where $V_2 = 0$ and $P_1 = P_2 = P_{atm}$. Also, we take the reference level at the bottom of the tank. Noting that $z_1 = 34$ ft and $z_2 = 72$ ft, $h_L = 0$ (to get the minimum value for the required air pressure), and that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$), the energy equation for steady incompressible flow through a control volume between these two points reduces to



$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L$$

or $$\frac{P_1 - P_{atm}}{\rho g} = z_2 - z_1 \rightarrow \frac{P_{1,gage}}{\rho g} = z_2 - z_1$$

Rearranging and substituting, the gage pressure of pressurized air in the tank is determined to be

$$P_{1,\text{gage}} = \rho g(z_2 - z_1) = (62.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(72 - 34\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{psi}}{144\ \text{lbf/ft}^2}\right) = \mathbf{16.5\ psi}$$

Therefore, the gage air pressure on top of the water tank must be at least 10.4 psi.

Discussion The result obtained above represents the minimum value, and should be interpreted accordingly. In reality, a larger pressure will be needed to overcome friction.

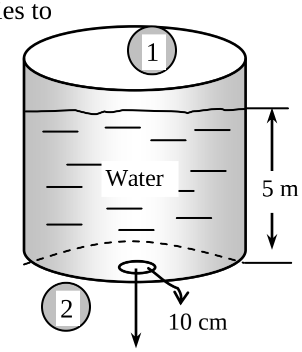
5-89

Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere. The initial discharge velocity from the tank is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The tank is open to the atmosphere. **3** The kinetic energy correction factor at the orifice is given to be $\alpha_2 = \alpha = 1.2$.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of the orifice. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$), the energy equation between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L$$



which yields

$$z_1 + \alpha_2 \frac{V_2^2}{2g} = z_2 + h_L$$

Solving for V_2 and substituting,

$$V_2 = \sqrt{2g(z_1 - z_2 - h_L)/\alpha} = \sqrt{2(9.81\ \text{m/s}^2)(5 - 0.3\ \text{m})/1.2} = \mathbf{8.77\ m/s}$$

Discussion This is the velocity that will prevail at the beginning. The mean flow velocity will decrease as the water level in the tank decreases.

5-90

Solution Water enters a hydraulic turbine-generator system with a known flow rate, pressure drop, and efficiency. The net electric power output is to be determined.

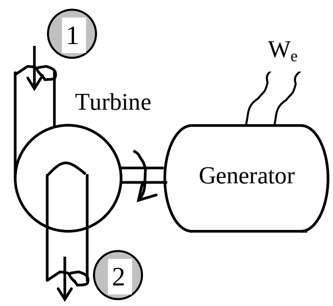
Assumptions **1** The flow is steady and incompressible. **2** All losses in the turbine are accounted for by turbine efficiency and thus h_L = 0. **3** The elevation difference across the turbine is negligible. **4** The effect of the kinetic energy correction factors is negligible, $\alpha_1 = \alpha_2 = \alpha = 1$.

Properties We take the density of water to be 1000 kg/m^3 and the density of mercury to be 13,560 kg/m^3.

Analysis We choose points 1 and 2 at the inlet and the exit of the turbine, respectively. Noting that the elevation effects are negligible, the energy equation in terms of heads for the turbine reduces to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{turbine,e}}=\frac{P_1-P_2}{\rho_{\text{water}}g}+\frac{\alpha(V_1^2-V_2^2)}{2g} \qquad (1)$$

where



$$V_1=\frac{\dot{V}}{A_1}=\frac{\dot{V}}{\pi D_1^2/4}\frac{0.6\ \text{m}^3/\text{s}}{\pi(0.30\ \text{m})^2/4}=8.49\ \text{m/s}$$

$$V_2=\frac{\dot{V}}{A_2}=\frac{\dot{V}}{\pi D_2^2/4}=\frac{0.6\ \text{m}^3/\text{s}}{\pi(0.25\ \text{m})^2/4}=12.2\ \text{m/s}$$

The pressure drop corresponding to a differential height of 1.2 m in the mercury manometer is

$$\begin{aligned}P_1-P_2&=(\rho_{\text{Hg}}-\rho_{\text{water}})gh\\&=[(13{,}560-1000)\ \text{kg/m}^3](9.81\ \text{m/s}^2)(1.2\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\\&=148\ \text{kN/m}^2=148\ \text{kPa}\end{aligned}$$

Substituting into Eq. (1), the turbine head is determined to be

$$h_{\text{turbine,e}}=\frac{148\ \text{kN/m}^2}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)+(1.0)\frac{(8.49\ \text{m/s})^2-(12.2\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)}=15.1-3.9=11.2\ \text{m}$$

Then the net electric power output of this hydroelectric turbine becomes

$$\begin{aligned}\dot{W}_{\text{turbine}}&=\eta_{\text{turbine-gen}}\dot{m}gh_{\text{turbine,e}}=\eta_{\text{turbine-gen}}\rho\dot{V}gh_{\text{turbine,e}}\\&=0.83(1000\ \text{kg/m}^3)(0.6\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(11.2\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right)=\mathbf{55\ kW}\end{aligned}$$

Discussion It appears that this hydroelectric turbine will generate 55 kW of electric power under given conditions. Note that almost half of the available pressure head is discarded as kinetic energy. This demonstrates the need for a larger turbine exit area and better recovery. For example, the power output can be increased to 74 kW by redesigning the turbine and making the exit diameter of the pipe equal to the inlet diameter, $D_2 = D_1$. Further, if a much larger exit diameter is used and the exit velocity is reduced to a very low level, the power generation can increase to as much as 92 kW.

5-91

Solution The velocity profile for turbulent flow in a circular pipe is given. The kinetic energy correction factor for this flow is to be determined.

Analysis The velocity profile is given by $u(r) = u_{\max}(1 - r/R)^{1/n}$ with n = 9 The kinetic energy correction factor is then expressed as

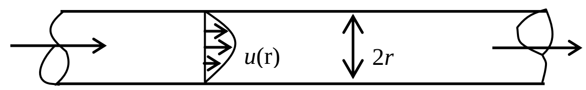
$$\alpha = \frac{1}{A}\int_A \left(\frac{u(r)}{V_{avg}}\right)^3 dA = \frac{1}{AV_{avg}^3}\int_A u(r)^3 dA = \frac{1}{\pi R^2 V_{avg}^3}\int_{r=0}^{R} u_{\max}^3 \left(1-\frac{r}{R}\right)^{\frac{3}{n}}(2\pi r)dr = \frac{2u_{\max}^3}{R^2 V_{avg}^3}\int_{r=0}^{R}\left(1-\frac{r}{R}\right)^{\frac{3}{n}} r dr$$

where the average velocity is

$$V_{avg} = \frac{1}{A}\int_A u(r)dA = \frac{1}{\pi R^2}\int_{r=0}^{R} u_{\max}\left(1-\frac{r}{R}\right)^{1/n}(2\pi r)dr = \frac{2u_{\max}}{R^2}\int_{r=0}^{R}\left(1-\frac{r}{R}\right)^{1/n} r dr$$

From integral tables,

$$\int (a+bx)^n x dx = \frac{(a+bx)^{n+2}}{b^2(n+2)} - \frac{a(a+bx)^{n+1}}{b^2(n+1)}$$



Then,

$$\int_{r=0}^{R} u(r) r dr = \int_{r=0}^{R}\left(1-\frac{r}{R}\right)^{1/n} r dr = \left.\frac{(1-r/R)^{\frac{1}{n}+2}}{\frac{1}{R^2}(\frac{1}{n}+2)} - \frac{(1-r/R)^{\frac{1}{n}+1}}{\frac{1}{R^2}(\frac{1}{n}+1)}\right|_{r=0}^{R} = \frac{n^2R^2}{(n+1)(2n+1)}$$

$$\int_{r=0}^{R} u(r)^3 r dr = \int_{r=0}^{R}\left(1-\frac{r}{R}\right)^{3/n} r dr = \left.\frac{(1-r/R)^{\frac{3}{n}+2}}{\frac{1}{R^2}(\frac{3}{n}+2)} - \frac{(1-r/R)^{\frac{3}{n}+1}}{\frac{1}{R^2}(\frac{3}{n}+1)}\right|_{r=0}^{R} = \frac{n^2R^2}{(n+3)(2n+3)}$$

Substituting,

$$V_{avg} = \frac{2u_{\max}}{R^2}\frac{n^2R^2}{(n+1)(2n+1)} = \frac{2n^2 u_{\max}}{(n+1)(2n+1)} = 0.8167 u_{\max}$$

and

$$\alpha = \frac{2u_{\max}^3}{R^2}\left(\frac{2n^2 u_{\max}}{(n+1)(2n+1)}\right)^{-3}\frac{n^2R^2}{(n+3)(2n+3)} = \frac{(n+1)^3(2n+1)^3}{4n^4(n+3)(2n+3)} = \frac{(9+1)^3(2\times 9+1)^3}{4\times 9^4(9+3)(2\times 9+3)} = 1.037 \cong \mathbf{1.04}$$

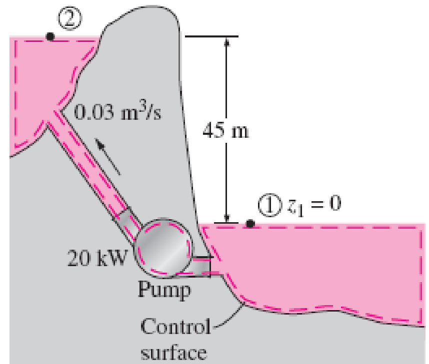
Discussion Note that ignoring the kinetic energy correction factor results in an error of just 4% in this case in the kinetic energy term (which may be small itself). Considering that the uncertainties in some terms are usually more that 4%, we can usually ignore this correction factor in turbulent pipe flow analyses. However, for laminar pipe flow analyses, α is equal to 2.0 for fully developed laminar pipe flow, and ignoring α may lead to significant errors.

5-92

Solution Water is pumped from a lower reservoir to a higher one. The head loss and power loss associated with this process are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the reservoirs is constant.

Properties We take the density of water to be 1000 kg/m^3.



Analysis The mass flow rate of water through the system is

$$\dot{m} = \rho\dot{V} = (1000\ \text{kg/m}^3)(0.030\ \text{m}^3/\text{s}) = 30\ \text{kg/s}$$

We choose points 1 and 2 at the free surfaces of the lower and upper reservoirs, respectively, and take the surface of the lower reservoir as the reference level ($z_1 = 0$). Both points are open to the atmosphere ($P_1 = P_2 = P_{atm}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation for steady incompressible flow for a control volume between 1 and 2 reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1\frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump,u}} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2\frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine,e}} + \dot{E}_{\text{mech,loss}}$$

$$\dot{W}_{\text{pump,u}} = \dot{m}gz_2 + \dot{E}_{\text{mech,loss}} \quad \rightarrow \quad \dot{E}_{\text{mech,loss}} = \dot{W}_{\text{pump,u}} - \dot{m}gz_2$$

Substituting, the lost mechanical power and head loss are calculated as

$$\dot{E}_{\text{mech,loss}} = 20\ \text{kW} - (30\ \text{kg/s})(9.81\ \text{m/s}^2)(45\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = \mathbf{6.76\ kW}$$

Noting that the entire mechanical losses are due to frictional losses in piping and thus $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss,piping}}$, the irreversible head loss is determined to be

$$h_L = \frac{\dot{E}_{\text{mech loss,piping}}}{\dot{m}g} = \frac{6.76\ \text{kW}}{(30\ \text{kg/s})(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)\left(\frac{1000\ \text{N}\cdot\text{m/s}}{1\ \text{kW}}\right) = \mathbf{23.0\ m}$$

Discussion The 6.76 kW of power is used to overcome the friction in the piping system. Note that the pump could raise the water an additional 23 m if there were no irreversible head losses in the system. In this ideal case, the pump would function as a turbine when the water is allowed to flow from the upper reservoir to the lower reservoir and extract 20 kW of power from the water.

5-93

Solution Water from a pressurized tank is supplied to a roof top. The discharge rate of water from the tank is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The effect of the kinetic energy correction factor is negligible and thus $\alpha_2 = 1$ (we examine the effect of this approximation in the discussion).

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of the discharge pipe. Noting that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$) and water discharges into the atmosphere (and thus $P_2 = P_{atm}$), the energy equation written in the head form simplifies to

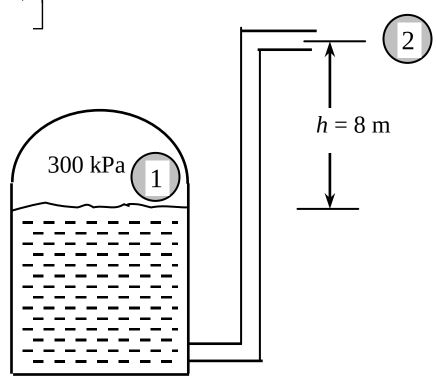
$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump, u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine, e}}+h_L \quad\rightarrow\quad \frac{P_1-P_{atm}}{\rho g}=\alpha_2\frac{V_2^2}{2g}+z_2-z_1+h_L$$

Solving for V_2 and substituting, the discharge velocity is determined to

$$V_2=\sqrt{\frac{1}{\alpha_2}\left[\frac{2P_{1,\text{ gage}}}{\rho}-2g(z_2-z_1+h_L)\right]}$$
$$=\sqrt{\frac{1}{1}\left[\frac{2\times(300\text{ kPa})}{1000\text{ kg/m}^3}\left(\frac{1000\text{ N/m}^2}{1\text{ kPa}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)-2(9.81\text{ m/s}^2)(8+2\text{ m})\right]}$$
$$=20.095\text{ m/s}\cong 20.1\text{ m/s}$$

Then the initial rate of discharge of water becomes

$$\dot{V}=A_{\text{orifice}}V_2=\frac{\pi D^2}{4}V_2=\frac{\pi(0.025\text{ m})^2}{4}(20.095\text{ m/s})$$
$$=0.009864\text{ m}^3\text{/s}\cong \mathbf{0.00986\ m^3/s=9.86\ L/s}$$



Discussion This is the discharge rate that will prevail at the beginning. The mean flow velocity will decrease as the water level in the tank decreases. If we assume that the flow in the hose at the discharge is fully developed and turbulent, $\alpha_2 \approx 1.05$, and the results change to V_2 = 19.610 m/s ≈ 19.6 m/s, and $\dot{V}=0.0096263\text{ m}^3\text{/s}\cong\mathbf{0.00963\ m^3/s=9.63\ L/s}$, a *decrease* (as expected since we are accounting for more losses) of about 2.4%.

5-94

Solution Underground water is pumped to a pool at a given elevation. The maximum flow rate and the pressures at the inlet and outlet of the pump are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the inlet and the outlet of the pump is negligible. **3** We assume the frictional effects in piping to be negligible since the *maximum* flow rate is to be determined, $\dot{E}_{\text{mech loss,pipping}} = 0$. **4** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

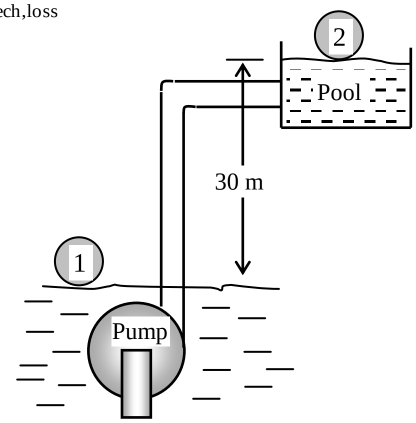
Properties We take the density of water to be 1 kg/L = 1000 kg/m^3.

Analysis (*a*) The pump-motor draws 5-kW of power, and is 78% efficient. Then the useful mechanical (shaft) power it delivers to the fluid is

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump-motor}}\dot{W}_{\text{electric}} = (0.78)(5\text{ kW}) = 3.9\text{ kW}$$

We take point 1 at the free surface of underground water, which is also taken as the reference level ($z_1 = 0$), and point 2 at the free surface of the pool. Also, both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$), the velocities are negligible at both points ($V_1 \cong V_2 \cong 0$), and frictional losses in piping are disregarded. Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1\frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2\frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech,loss}}$$



In the absence of a turbine, $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss,pump}} + \dot{E}_{\text{mech loss,piping}}$ and

$\dot{W}_{\text{pump,u}} = \dot{W}_{\text{pump}} - \dot{E}_{\text{mech loss,pump}}$.

Thus, $\dot{W}_{\text{pump,u}} = \dot{m}gz_2$.

Then the mass and volume flow rates of water become

$$\dot{m} = \frac{\dot{W}_{\text{pump,u}}}{gz_2} = \frac{3.9\text{ kJ/s}}{(9.81\text{ m/s}^2)(30\text{ m})}\left(\frac{1000\text{ m}^2\text{/s}^2}{1\text{ kJ}}\right) = 13.25\text{ kg/s}$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{13.25\text{ kg/s}}{1000\text{ kg/m}^3} = 0.01325\text{ m}^3\text{/s} \cong \mathbf{0.0133\,m^3/s}$$

(*b*) We take points 3 and 4 at the inlet and the exit of the pump, respectively, where the flow velocities are

$$V_3 = \frac{\dot{V}}{A_3} = \frac{\dot{V}}{\pi D_3^2/4} = \frac{0.01325\text{ m}^3\text{/s}}{\pi(0.07\text{ m})^2/4} = 3.443\text{ m/s}, \quad V_4 = \frac{\dot{V}}{A_4} = \frac{\dot{V}}{\pi D_4^2/4} = \frac{0.01325\text{ m}^3\text{/s}}{\pi(0.05\text{ m})^2/4} = 6.748\text{ m/s}$$

We take the pump as the control volume. Noting that $z_3 = z_4$, the energy equation for this control volume reduces to

$$\dot{m}\left(\frac{P_3}{\rho} + \alpha_3\frac{V_3^2}{2} + gz_3\right) + \dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_4}{\rho} + \alpha_4\frac{V_4^2}{2} + gz_4\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech loss,pump}} \rightarrow P_4 - P_3 = \frac{\rho\alpha(V_3^2 - V_4^2)}{2} + \frac{\dot{W}_{\text{pump,u}}}{\dot{V}}$$

Substituting,

$$P_4 - P_3 = \frac{(1000\text{ kg/m}^3)[(3.443\text{ m/s})^2 - (6.748\text{ m/s})^2]}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) + \frac{3.9\text{ kJ/s}}{0.01325\text{ m}^3\text{/s}}\left(\frac{1\text{ kN}\cdot\text{m}}{1\text{ kJ}}\right)$$

$$= (-16.8 + 294.3)\text{ kN/m}^2 = 277.5\text{ kPa} \cong \mathbf{278\,kPa}$$

Discussion In an actual system, the flow rate of water will be less because of friction in the pipes. Also, the effect of flow velocities on the pressure change across the pump is negligible in this case (under 2%) and can be ignored.

5-95

Solution Underground water is pumped to a pool at a given elevation. For a given head loss, the flow rate and the pressures at the inlet and outlet of the pump are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the inlet and the outlet of the pump is negligible. **3** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

Properties We take the density of water to be 1 kg/L = 1000 kg/m^3.

Analysis (*a*) The pump-motor draws 5-kW of power, and is 78% efficient. Then the useful mechanical (shaft) power it delivers to the fluid is

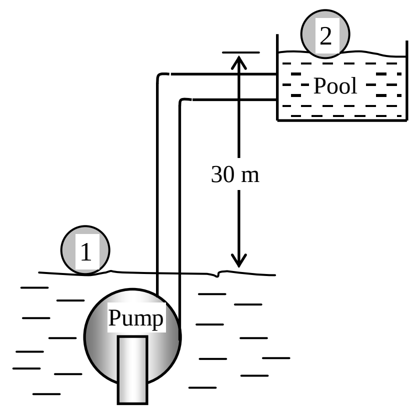
$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump-motor}}\dot{W}_{\text{electric}} = (0.78)(5\text{ kW}) = 3.9\text{ kW}$$

We take point 1 at the free surface of underground water, which is also taken as the reference level ($z_1 = 0$), and point 2 at the free surface of the pool. Also, both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$), and the velocities are negligible at both points ($V_1 \cong V_2 \cong 0$). Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\dot{m}\left(\frac{P_1}{\rho}+\alpha_1\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_2}{\rho}+\alpha_2\frac{V_2^2}{2}+gz_2\right)+\dot{W}_{\text{turbine}}+\dot{E}_{\text{mech,loss}}$$

In the absence of a turbine, $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss,pump}} + \dot{E}_{\text{mech loss,piping}}$ and $\dot{W}_{\text{pump,u}} = \dot{W}_{\text{pump}} - \dot{E}_{\text{mech loss,pump}}$ and thus

$$\dot{W}_{\text{pump,u}} = \dot{m}gz_2 + \dot{E}_{\text{mech loss,piping}}$$



Noting that $\dot{E}_{\text{mech, loss}} = \dot{m}gh_L$, the mass and volume flow rates of water become

$$\dot{m} = \frac{\dot{W}_{\text{pump,u}}}{gz_2+gh_L} = \frac{\dot{W}_{\text{pump,u}}}{g(z_2+h_L)}$$

$$= \frac{3.9\text{ kJ/s}}{(9.81\text{ m/s}^2)(30+4\text{ m})}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ}}\right) = 11.69\text{ kg/s}$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{11.69\text{ kg/s}}{1000\text{ kg/m}^3} = 0.01169\text{ m}^3/\text{s} \cong \mathbf{0.0117\ m^3/s}$$

(*b*) We take points 3 and 4 at the inlet and the exit of the pump, respectively, where the flow velocities are

$$V_3 = \frac{\dot{V}}{A_3} = \frac{\dot{V}}{\pi D_3^2/4} = \frac{0.01169\text{ m}^3/\text{s}}{\pi(0.07\text{ m})^2/4} = 3.038\text{ m/s},\quad V_4 = \frac{\dot{V}}{A_4} = \frac{\dot{V}}{\pi D_4^2/4} = \frac{0.01169\text{ m}^3/\text{s}}{\pi(0.05\text{ m})^2/4} = 5.954\text{ m/s}$$

We take the pump as the control volume. Noting that $z_3 = z_4$, the energy equation for this control volume reduces to

$$\dot{m}\left(\frac{P_3}{\rho}+\alpha_3\frac{V_3^2}{2}+gz_3\right)+\dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_4}{\rho}+\alpha_4\frac{V_4^2}{2}+gz_4\right)+\dot{W}_{\text{turbine}}+\dot{E}_{\text{mech loss,pump}} \rightarrow P_4-P_3 = \frac{\rho\alpha(V_3^2-V_4^2)}{2}+\frac{\dot{W}_{\text{pump,u}}}{\dot{V}}$$

Substituting,

$$P_4 - P_3 = \frac{(1000\text{ kg/m}^3)[(3.038\text{ m/s})^2-(5.954\text{ m/s})^2]}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)+\frac{3.9\text{ kJ/s}}{0.01169\text{ m}^3/\text{s}}\left(\frac{1\text{ kN}\cdot\text{m}}{1\text{ kJ}}\right)$$

$$= (-13.1+333.6)\text{ kN/m}^2 = 320.5\text{ kPa} \cong \mathbf{321\ kPa}$$

Discussion Note that frictional losses in the pipes causes the flow rate of water to decrease. Also, the effect of flow velocities on the pressure change across the pump is negligible in this case (about 1%) and can be ignored.

5-96E

Solution Water is pumped from a lake to a nearby pool by a pump with specified power and efficiency. The head loss of the piping system and the mechanical power used to overcome it are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the lake and the free surface of the pool is constant. **3** All the losses in the pump are accounted for by the pump efficiency and thus h_L represents the losses in piping.

Properties We take the density of water to be $\rho = 62.4 \text{ lbm/ft}^3$.

Analysis The useful pumping power and the corresponding useful pumping head are

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump}}\dot{W}_{\text{pump}} = (0.73)(12 \text{ hp}) = 8.76 \text{ hp}$$

$$h_{\text{pump,u}} = \frac{\dot{W}_{\text{pump,u}}}{\dot{m}g} = \frac{\dot{W}_{\text{pump,u}}}{\rho\dot{V}g}$$

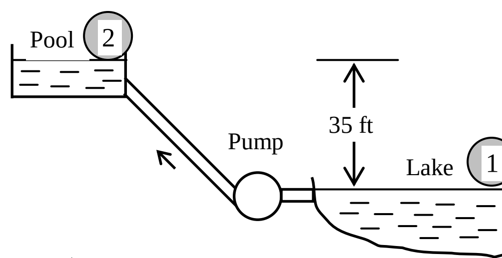
$$= \frac{8.76 \text{ hp}}{(62.4 \text{ lbm/ft}^3)(1.2 \text{ ft}^3\text{/s})(32.2 \text{ ft/s}^2)}\left(\frac{32.2 \text{ lbm}\cdot\text{ft/s}^2}{1 \text{ lbf}}\right)\left(\frac{550 \text{ lbf}\cdot\text{ft/s}}{1 \text{ hp}}\right) = 64.3 \text{ ft}$$

We choose points 1 and 2 at the free surfaces of the lake and the pool, respectively. Both points are open to the atmosphere ($P_1 = P_2 = P_{atm}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad h_L = h_{\text{pump,u}} + z_1 - z_2$$

Substituting, the head loss is determined to be

$$h_L = h_{\text{pump,u}} - (z_2 - z_1) = 64.3 - 35 = \mathbf{29.3 \text{ ft}}$$



Then the power used to overcome it becomes

$$\dot{E}_{\text{mech loss,piping}} = \rho\dot{V}gh_L$$

$$= (62.4 \text{ lbm/ft}^3)(1.2 \text{ ft}^3\text{/s})(32.2 \text{ ft/s}^2)(29.3 \text{ ft})\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1 \text{ hp}}{550 \text{ lbf}\cdot\text{ft/s}}\right)$$

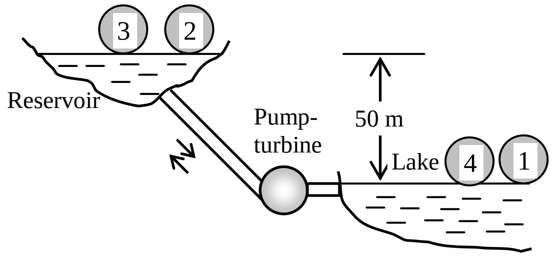
$$= \mathbf{4.0 \text{ hp}}$$

Discussion Note that the pump must raise the water an additional height of 29.3 ft to overcome the frictional losses in pipes, which requires an additional useful pumping power of about 4 hp.

5-97

Solution An entrepreneur is to build a large reservoir above the lake level, and pump water from the lake to the reservoir at night using cheap power, and let the water flow from the reservoir back to the lake during the day, producing power. The potential revenue this system can generate per year is to be determined.

Assumptions **1** The flow in each direction is steady and incompressible. **2** The elevation difference between the lake and the reservoir can be taken to be constant, and the elevation change of reservoir during charging and discharging is disregarded. **3** The given unit prices remain constant. **4** The system operates every day of the year for 10 hours in each mode.



Properties We take the density of water to be ρ = 1000 kg/m^3.

Analysis We choose points 1 and 2 at the free surfaces of the lake and the reservoir, respectively, and take the surface of the lake as the reference level. Both points are open to the atmosphere ($P_1 = P_2 = P_{atm}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation in terms of heads for steady incompressible flow through a control volume between these two points that includes the pump (or the turbine) and the pipes reduces to

Pump mode: $$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \rightarrow$$

$$h_{\text{pump,u}}=z_2+h_L=50+4=54\text{ m}$$

Turbine mode: (switch points 1 and 2 so that 1 is on inlet side) $\rightarrow$ $h_{\text{turbine,e}}=z_1-h_L=50-4=46\text{ m}$

The pump and turbine power corresponding to these heads are

$$\dot W_{\text{pump,elect}}=\frac{\dot W_{\text{pump,u}}}{\eta_{\text{pump-motor}}}=\frac{\rho\dot V g h_{\text{pump,u}}}{\eta_{\text{pump-motor}}}$$
$$=\frac{(1000\text{ kg/m}^3)(2\text{ m}^3/\text{s})(9.81\text{ m/s}^2)(54\text{ m})}{0.75}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right)=1413\text{ kW}$$

$$\dot W_{\text{turbine}}=\eta_{\text{turbine-gen}}\dot m g h_{\text{turbine,e}}=\eta_{\text{turbine-gen}}\rho\dot V g h_{\text{turbine,e}}$$
$$=0.75(1000\text{ kg/m}^3)(2\text{ m}^3/\text{s})(9.81\text{ m/s}^2)(46\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right)=677\text{ kW}$$

Then the power cost of the pump, the revenue generated by the turbine, and the net income (revenue minus cost) per year become

$$\text{Cost}=\dot W_{\text{pump,elect}}\Delta t\times\text{Unit price}=(1413\text{ kW})(365\times 10\text{ h/year})(\$0.06/\text{kWh})=\$309{,}447/\text{year}$$

$$\text{Revenue}=\dot W_{\text{turbine}}\Delta t\times\text{Unit price}=(677\text{ kW})(365\times 10\text{ h/year})(\$0.13/\text{kWh})=\$321{,}237/\text{year}$$

Net income = Revenue – Cost = 321,237 – 309,447 = $11,790/year ≅ **$11,800/year**

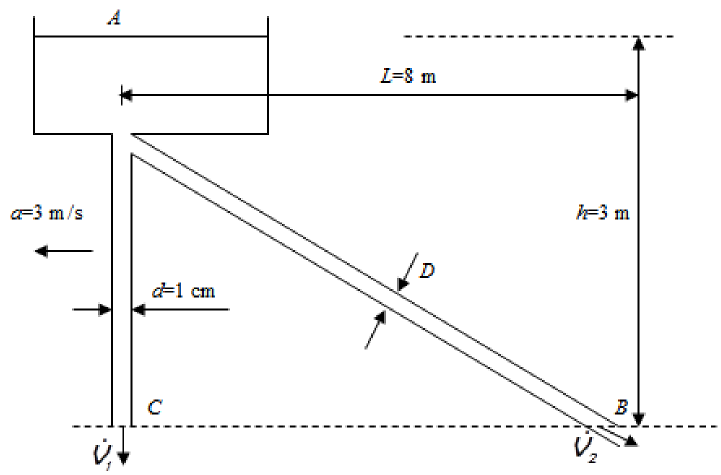
Discussion It appears that this pump-turbine system has a potential annual income of about $11,800. A decision on such a system will depend on the initial cost of the system, its life, the operating and maintenance costs, the interest rate, and the length of the contract period, among other things.

5-98

Solution A tank with two discharge pipes accelerates to the left. The diameter of the inclined pipe is to be determined.

Assumptions 1 The flow is steady and incompressible. **2** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

Analysis



Applying Bernoulli Equation between the points A and B

$$\left(\frac{P_A}{\gamma}+Z_A+\frac{W_A^2}{2g}\right)-\left(\frac{P_B}{\gamma}+Z_B+\frac{W_B^2}{2g}\right)=\frac{a}{g}\Delta L+Negligible\ losses$$

$$h=\frac{W_B^2}{2g}+\frac{a}{g}\Delta L \rightarrow \frac{W_B^2}{2g}=h-\frac{a}{g}\Delta L \qquad (1)$$

Applying Bernoulli Equation between the points A and C

$$h=\frac{W_C^2}{2g} \qquad (2)$$

Let's divide (1) by (2)

$$\frac{\dfrac{W_B^2}{2g}}{\dfrac{W_C^2}{2g}}=\frac{h-\dfrac{a}{g}\Delta L}{h} \text{ or } \left(\frac{W_B}{W_C}\right)^2=1-\frac{a\Delta L}{gh}, \quad \frac{W_B}{W_C}=\sqrt{1-\frac{a\Delta L}{gh}}$$

$$Q_B=Q_C \rightarrow A_BW_B=A_CW_C \rightarrow \frac{A_C}{A_B}=\frac{W_B}{W_C}=\frac{d^2}{D^2}$$

Therefore,

$$\frac{d^2}{D^2}=\sqrt{1-\frac{a\Delta L}{gh}}=\sqrt{1-\frac{3.18}{9,81.3}}=0.429$$

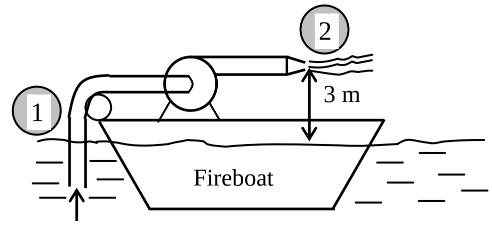
$$D^2=\frac{d^2}{0.429} \rightarrow D=\sqrt{\frac{0.01^2}{0.429^2}} \cong 1.53x10^{-2}\ m \rightarrow D \cong \mathbf{1.53\ cm}$$

5-99

Solution A fireboat is fighting fires by drawing sea water and discharging it through a nozzle. The head loss of the system and the elevation of the nozzle are given. The shaft power input to the pump and the water discharge velocity are to be determined.

Assumptions 1 The flow is steady and incompressible. **2** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

Properties The density of sea water is given to be ρ =1030 kg/m^3.



Analysis We take point 1 at the free surface of the sea and point 2 at the nozzle exit. Noting that $P_1 = P_2 = P_{atm}$ and $V_1 \cong 0$ (point 1 is at the free surface; not at the pipe inlet), the energy equation for the control volume between 1 and 2 that includes the pump and the piping system reduces to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad h_{\text{pump,u}} = z_2 - z_1 + \alpha_2 \frac{V_2^2}{2g} + h_L$$

where the water discharge velocity is

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{0.04\ \text{m}^3/\text{s}}{\pi (0.05\ \text{m})^2/4} = 20.37\ \text{m/s} \cong \mathbf{20.4\ m/s}$$

Substituting, the useful pump head and the corresponding useful pump power are determined to be

$$h_{pump,u} = (3\ \text{m}) + (1)\frac{(20.37\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} + (3\ \text{m}) = 27.15\ \text{m}$$

$$\dot{W}_{\text{pump,u}} = \rho \dot{V} g h_{pump,u}$$
$$= (1030\ \text{kg/m}^3)(0.1\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(27.15\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1\ \text{kN}\cdot\text{m/s}}\right)$$
$$= 27.43\ \text{kW}$$

Then the required shaft power input to the pump becomes

$$\dot{W}_{\text{pump,shaft}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump}}} = \frac{27.43\ \text{kW}}{0.70} = \mathbf{39.2\ kW}$$

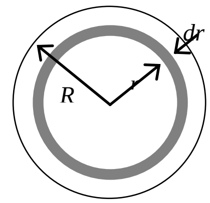
Discussion Note that the pump power is used primarily to increase the kinetic energy of water.

Review Problems

5-100

Solution A fluid is flowing in a circular pipe. A relation is to be obtained for the average fluid velocity in terms of $V(r)$, R, and r.

Analysis Choosing a circular ring of area $dA = 2\pi r dr$ as our differential area, the mass flow rate through a cross-sectional area can be expressed as



$$\dot{m} = \int_A \rho V(r) dA = \int_0^R \rho V(r) 2\pi r\, dr$$

Setting this equal to and solving for V_{avg},

$$\boxed{V_{avg} = \frac{2}{R^2}\int_0^R V(r) r\, dr}$$

Discussion If V were a function of both r and θ, we would also need to integrate with respect to θ.

5-101

Solution Air is accelerated in a nozzle. The density of air at the nozzle exit is to be determined.

Assumptions Flow through the nozzle is steady.

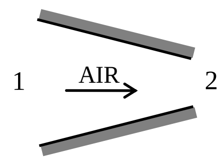
Properties The density of air is given to be 2.50 kg/m^3 at the inlet.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

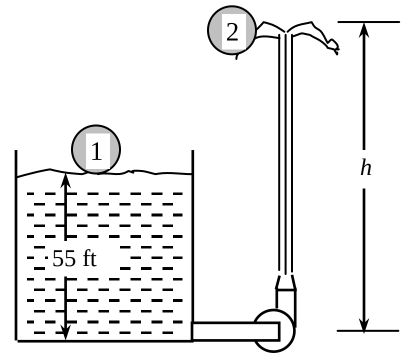
$$\rho_2 = \frac{A_1}{A_2}\frac{V_1}{V_2}\rho_1 = 2\frac{120\text{ m/s}}{330\text{ m/s}}(2.50\text{ kg/m}^3) = \mathbf{1.82\ kg/m^3}$$



Discussion Note that the density of air decreases considerably despite a decrease in the cross-sectional area of the nozzle.

5-102E

Solution A hose is connected to the bottom of a water tank open to the atmosphere. The hose is equipped with a pump and a nozzle at the end. The maximum height to which the water stream could rise is to be determined.



Assumptions **1** The flow is incompressible with negligible friction. **2** The friction between the water and air is negligible. **3** We take the head loss to be zero ($h_L = 0$) to determine the maximum rise of water jet.

Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis We take point 1 at the free surface of the tank, and point 2 at the top of the water trajectory where $V_2 = 0$. We take the reference level at the bottom of the tank. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad z_1 + h_{\text{pump,u}} = z_2$$

where the useful pump head is

$$h_{\text{pump,u}} = \frac{\Delta P_{\text{pump}}}{\rho g} = \frac{10 \text{ psi}}{(62.4 \text{ lbm/ft}^3)(32.2 \text{ ft/s}^2)} \left(\frac{144 \text{ lbf/ft}^2}{1 \text{ psi}} \right) \left(\frac{32.2 \text{ lbm} \cdot \text{ft/s}^2}{1 \text{ lbf}} \right) = 23.1 \text{ ft}$$

Substituting, the maximum height rise of water jet from the ground level is determined to be

$$z_2 = z_1 + h_{\text{pump,u}} = 55 + 23.2 = \mathbf{78.2 \text{ ft}}$$

Discussion The actual rise of water will be less because of the frictional effects between the water and the hose walls and between the water jet and air.

5-103

Solution Water discharges from the orifice at the bottom of a pressurized tank. The time it will take for half of the water in the tank to be discharged and the water level after 10 s are to be determined.

Assumptions **1** The flow is incompressible, and the frictional effects are negligible. **2** The tank air pressure above the water level is maintained constant.

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of orifice. We take the positive direction of z to be upwards with reference level at the orifice (z_2 = 0). Fluid at point 2 is open to the atmosphere (and thus $P_2 = P_{\text{atm}}$) and the velocity at the free surface is very low ($V_1 \cong 0$). Then,

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}+z_1=\frac{P_{atm}}{\rho g}+\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1+2P_{1,\text{gage}}/\rho}$$

or, $V_2=\sqrt{2gz+2P_{1,\text{gage}}/\rho}$ where z is the water height in the tank at any time t. Water surface moves down as the tank drains, and the value of z changes from H initially to 0 when the tank is emptied completely.

We denote the diameter of the orifice by D, and the diameter of the tank by D_o. The flow rate of water from the tank is obtained by multiplying the discharge velocity by the orifice cross-sectional area,

$$\dot{\mathcal{V}}=A_{\text{orifice}}V_2=\frac{\pi D^2}{4}\sqrt{2gz+2P_{1,\text{gage}}/\rho}$$

Then the amount of water that flows through the orifice during a differential time interval dt is

$$d\mathcal{V}=\dot{\mathcal{V}}dt=\frac{\pi D^2}{4}\sqrt{2gz+2P_{1,\text{gage}}/\rho}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$d\mathcal{V}=A_{\text{tank}}(-dz)=-\frac{\pi D_0^2}{4}dz \qquad (2)$$

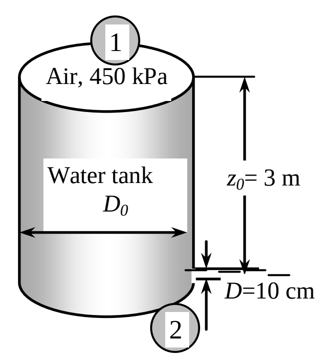
where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{2gz+2P_{1,\text{gage}}/\rho}\,dt=-\frac{\pi D_0^2}{4}dz \quad\rightarrow\quad dt=-\frac{D_0^2}{D^2}\sqrt{\frac{1}{2gz+2P_{1,\text{gage}}/\rho}}\,dz$$

The last relation can be integrated since the variables are separated. Letting t_f be the discharge time and integrating it from t = 0 when $z = z_0$ to $t = t$ when $z = z$ gives

$$\sqrt{\frac{2z_0}{g}+\frac{2P_{1,gage}}{\rho g^2}}-\sqrt{\frac{2z}{g}+\frac{2P_{1,gage}}{\rho g^2}}=\frac{D_0^2}{D^2}t$$

where $$\frac{2P_{1,\text{gage}}}{\rho g^2}=\frac{2(450-100)\text{ kN/m}^2}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)^2}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kN}}\right)=7.274\text{ s}^2$$



The time for half of the water in the tank to be discharged ($z = z_0/2$) is

$$\sqrt{\frac{2(3\text{ m})}{9.81\text{ m/s}^2}+7.274\text{ s}^2}-\sqrt{\frac{2(1.5\text{ m})}{9.81\text{ m/s}^2}+7.274\text{ s}^2}=\frac{(0.1\text{ m})^2}{(2\text{ m})^2}t \quad\rightarrow\quad t = \mathbf{22.0\text{ s}}$$

(b) Water level after 10 s is $\sqrt{\frac{2(3\text{ m})}{9.81\text{ m/s}^2}+7.274\text{ s}^2}-\sqrt{\frac{2z}{9.81\text{ m/s}^2}+7.274\text{ s}^2}=\frac{(0.1\text{ m})^2}{(2\text{ m})^2}(10\text{ s})\rightarrow z = \mathbf{2.31\text{ m}}$

Discussion Note that the discharging time is inversely proportional to the square of the orifice diameter. Therefore, the discharging time can be reduced to one-fourth by doubling the diameter of the orifice.

5-104

Solution Air flows through a pipe that consists of two sections at a specified rate. The differential height of a water manometer placed between the two pipe sections is to be determined.

Assumptions **1**The flow through the pipe is steady, incompressible, and irrotational with negligible friction (so that the Bernoulli equation is applicable). **2** The losses in the reducing section are negligible. **3** The pressure difference across an air column is negligible because of the low density of air, and thus the air column in the manometer can be ignored.

Properties The density of air is given to be ρ_{air} = 1.20 kg/m^3. We take the density of water to be ρ_w = 1000 kg/m^3.

Analysis We take points 1 and 2 along the centerline of the pipe over the two tubes of the manometer. Noting that $z_1 = z_2$ (or, the elevation effects are negligible for gases), the Bernoulli equation between points 1 and 2 gives

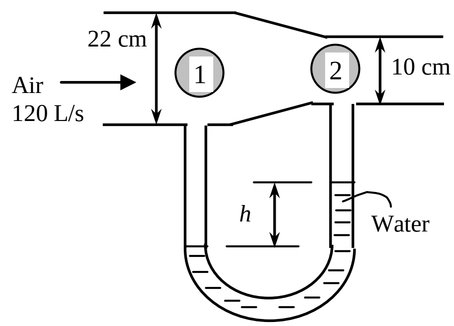
$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad P_1-P_2=\frac{\rho_{air}(V_2^2-V_1^2)}{2} \qquad (1)$$

We let the differential height of the water manometer be h. Then the pressure difference $P_2 - P_1$ can also be expressed as

$$P_1-P_2=\rho_w gh \qquad (2)$$

Combining Eqs. (1) and (2) and solving for h,

$$\frac{\rho_{air}(V_2^2-V_1^2)}{2}=\rho_w gh \quad\rightarrow\quad h=\frac{\rho_{air}(V_2^2-V_1^2)}{2g\rho_w}=\frac{V_2^2-V_1^2}{2g\rho_w/\rho_{air}}$$



Calculating the velocities and substituting,

$$V_1=\frac{\dot{V}}{A_1}=\frac{\dot{V}}{\pi D_1^2/4}=\frac{0.120\ \text{m}^3/\text{s}}{\pi(0.22\ \text{m})^2/4}=3.157\ \text{m/s}$$

$$V_2=\frac{\dot{V}}{A_2}=\frac{\dot{V}}{\pi D_2^2/4}=\frac{0.120\ \text{m}^3/\text{s}}{\pi(0.1\ \text{m})^2/4}=15.28\ \text{m/s}$$

$$h=\frac{(15.28\ \text{m/s})^2 - (3.157\ \text{m/s})^2}{2(9.81\ \text{m/s})^2(1000/1.20)}=0.01367\ \text{m}\cong \mathbf{1.37\ cm}$$

Therefore, the differential height of the water column will be 1.37 cm.

Discussion Note that the differential height of the manometer is inversely proportional to the density of the manometer fluid. Therefore, heavy fluids such as mercury are used when measuring large pressure differences.

5-105

Solution Air flows through a horizontal duct of variable cross-section. For a given differential height of a water manometer placed between the two pipe sections, the downstream velocity of air is to be determined, and an error analysis is to be conducted.

Assumptions **1** The flow through the duct is steady, incompressible, and irrotational with negligible friction (so that the Bernoulli equation is applicable). **2** The losses in this section of the duct are negligible. **3** The pressure difference across an air column is negligible because of the low density of air, and thus the air column in the manometer can be ignored.

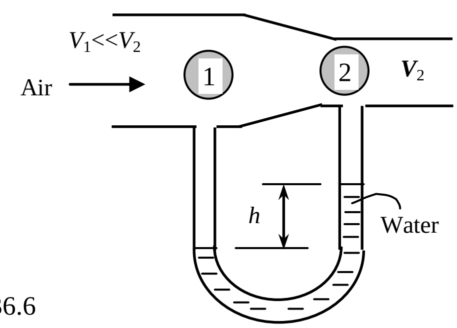
Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K. We take the density of water to be ρ_w = 1000 kg/m^3.

Analysis We take points 1 and 2 along the centerline of the duct over the two tubes of the manometer. Noting that $z_1 = z_2$ (or, the elevation effects are negligible for gases) and $V_1 \cong 0$, the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{\frac{2(P_1-P_2)}{\rho_{air}}} \qquad (1)$$

where $\quad P_1-P_2=\rho_w gh$

and $\quad \rho_{air}=\frac{P}{RT}=\frac{100\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})}=1.17\text{ kg/m}^3$



Substituting into (1), the downstream velocity of air V_2 is determined to be

$$V_2=\sqrt{\frac{2\rho_w gh}{\rho_{air}}}=\sqrt{\frac{2(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.08\text{ m})}{1.17\text{ kg/m}^3}}=\mathbf{36.6\text{ m/s}} \qquad (2)$$

Therefore, the velocity of air increases from a low level in the first section to 36.6 m/s in the second section.

Error Analysis We observe from Eq. (2) that the velocity is proportional to the square root of the differential height of the manometer fluid. That is, $V_2 = k\sqrt{h}$.

Taking the differential: $\quad dV_2=\frac{1}{2}k\frac{dh}{\sqrt{h}}$

Dividing by V_2: $\quad \frac{dV_2}{V_2}=\frac{1}{2}k\frac{dh}{\sqrt{h}}\frac{1}{k\sqrt{h}} \quad\rightarrow\quad \frac{dV_2}{V_2}=\frac{dh}{2h}=\frac{\pm 2\text{ mm}}{2\times 80\text{ mm}}=\mathbf{\pm\ 0.013}$

Therefore, the uncertainty in the velocity corresponding to an uncertainty of 2 mm in the differential height of water is 1.3%, which corresponds to 0.013×(36.6 m/s) = 0.5m/s. Then the discharge velocity can be expressed as

$$V_2=\mathbf{36.6\pm 0.5\text{ m/s}}$$

Discussion The error analysis does not include the effects of friction in the duct; the error due to frictional losses is most likely more severe than the error calculated here.

5-106

Solution A tap is opened on the wall of a very large tank that contains air. The maximum flow rate of air through the tap is to be determined, and the effect of a larger diameter lead section is to be assessed.

Assumptions Flow through the tap is steady, incompressible, and irrotational with negligible friction (so that the flow rate is maximum, and the Bernoulli equation is applicable).

Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K.

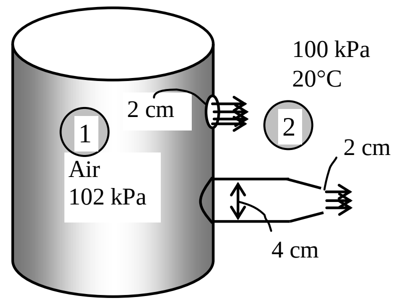
Analysis The density of air in the tank is

$$\rho_{air} = \frac{P}{RT} = \frac{102\ \text{kPa}}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = 1.21\ \text{kg/m}^3$$

We take point 1 in the tank, and point 2 at the exit of the tap along the same horizontal line. Noting that $z_1 = z_2$ (or, the elevation effects are negligible for gases) and $V_1 \cong 0$, the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \rightarrow \quad \frac{P_1}{\rho g} = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} \quad \rightarrow \quad V_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho_{air}}}$$

Substituting, the discharge velocity and the flow rate becomes



$$V_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho_{air}}} = \sqrt{\frac{2(102-100)\ \text{kN/m}^2}{1.21\ \text{kg/m}^3}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)} = 57.5\ \text{m/s}$$

$$\dot{V} = AV_2 = \frac{\pi D_2^2}{4} V_2 = \frac{\pi (0.02\ \text{m})^2}{4}(57.5\ \text{m/s}) = \mathbf{0.0181\ m^3/s}$$

This is the *maximum* flow rate since it is determined by assuming frictionless flow. The actual flow rate will be less.

Adding a 2-m long larger diameter lead section will have **no effect** on the flow rate since the flow is frictionless (by using the Bernoulli equation, it can be shown that the velocity in this section increases, but the pressure decreases, and there is a smaller pressure difference to drive the flow through the tab, with zero net effect on the discharge rate).

Discussion If the pressure in the tank were 300 kPa, the flow is no longer incompressible, and thus the problem in that case should be analyzed using compressible flow theory.

5-107

Solution Water is flowing through a venturi meter with known diameters and measured pressures. The flow rate of water is to be determined for the case of frictionless flow.

Assumptions **1** The flow through the venturi is steady, incompressible, and irrotational with negligible friction (so that the Bernoulli equation is applicable). **2** The flow is horizontal so that elevation along the centerline is constant. **3** The pressure is uniform at a given cross-section of the venturi meter (or the elevation effects on pressure measurement are negligible).

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis We take point 1 at the main flow section and point 2 at the throat along the centerline of the venturi meter. Noting that $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

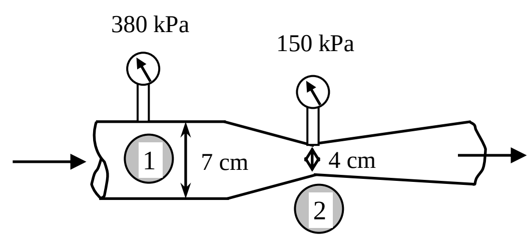
$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\to\quad P_1-P_2=\rho\frac{V_2^2-V_1^2}{2} \qquad (1)$$

The flow is assumed to be incompressible and thus the density is constant. Then the conservation of mass relation for this single stream steady flow device can be expressed as

$$\dot{V}_1=\dot{V}_2=\dot{V} \quad\to\quad A_1V_1=A_2V_2=\dot{V} \quad\to\quad V_1=\frac{\dot{V}}{A_1} \quad\text{and}\quad V_2=\frac{\dot{V}}{A_2} \qquad (2)$$

Substituting into Eq. (1),

$$P_1-P_2=\rho\frac{(\dot{V}/A_2)^2-(\dot{V}/A_1)^2}{2}=\frac{\rho\dot{V}^2}{2A_2^2}\left(1-\frac{A_2^2}{A_1^2}\right)$$



Solving for $\dot{V}$ gives the desired relation for the flow rate,

$$\dot{V}=A_2\sqrt{\frac{2(P_1-P_2)}{\rho[1-(A_2/A_1)^2]}} \qquad (3)$$

The flow rate for the given case can be determined by substituting the given values into this relation to be

$$\dot{V}=\frac{\pi D_2^2}{4}\sqrt{\frac{2(P_1-P_2)}{\rho[1-(D_2/D_1)^4]}}=\frac{\pi(0.04\text{ m})^2}{4}\sqrt{\frac{2(380-150)\text{ kN/m}^2}{(1000\text{ kg/m}^3)[1-(4/7)^4]}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kN}}\right)}=\mathbf{0.0285\ m^3/s}$$

Discussion Venturi meters are commonly used as flow meters to measure the flow rate of gases and liquids by simply measuring the pressure difference P_1 - P_2 by a manometer or pressure transducers. The actual flow rate will be less than the value obtained from Eq. (3) because of the friction losses along the wall surfaces in actual flow. But this difference can be as little as 1% in a well-designed venturi meter. The effects of deviation from the idealized Bernoulli flow can be accounted for by expressing Eq. (3) as

$$\dot{V}=C_dA_2\sqrt{\frac{2(P_1-P_2)}{\rho[1-(A_2/A_1)^2]}}$$

where C_d is the *venturi discharge coefficient* whose value is less than 1 (it is as large as 0.99 for well-designed venturi meters in certain ranges of flow). For Re > 10^5, the value of venturi discharge coefficient is usually greater than 0.96.

5-108

Solution Water flows through the enlargement section of a horizontal pipe at a specified rate. For a given head loss, the pressure change across the enlargement section is to be determined.

Assumptions **1** The flow through the pipe is steady and incompressible. **2** The pipe is horizontal. **3** The kinetic energy correction factors are given to be $\alpha_1 = \alpha_2 = \alpha = 1.05$.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

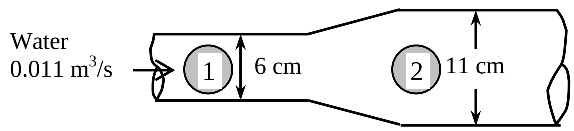
Analysis We take points 1 and 2 at the inlet and exit of the enlargement section along the centerline of the pipe. Noting that $z_1 = z_2$, the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad P_2 - P_1 = \rho \frac{\alpha(V_1^2 - V_2^2)}{2} - \rho g h_L$$

where the inlet and exit velocities are

$$V_1 = \frac{\dot{V}}{A_1} = \frac{\dot{V}}{\pi D_1^2/4} = \frac{0.011\ \text{m}^3/\text{s}}{\pi(0.06\ \text{m})^2/4} = 3.890\ \text{m/s}$$

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{0.011\ \text{m}^3/\text{s}}{\pi(0.11\ \text{m})^2/4} = 1.157\ \text{m/s}$$



Substituting, the change in static pressure across the enlargement section is determined to be

$$P_2 - P_1 = (1000\ \text{kg/m}^3)\left(\frac{1.05[(3.890\ \text{m/s})^2 - (1.157\ \text{m/s})^2]}{2} - (9.81\ \text{m/s}^2)(0.65\ \text{m})\right)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right)$$

$$= \mathbf{0.865\ kPa}$$

Therefore, the water pressure increases by 0.865 kPa across the enlargement section.

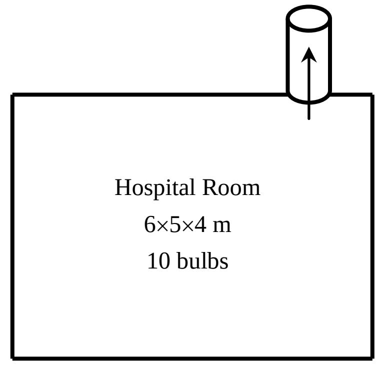
Discussion Note that the pressure increases despite the head loss in the enlargement section. This is due to dynamic pressure being converted to static pressure. But the total pressure (static + dynamic) decreases by 0.65 m (or 6.38 kPa) as a result of frictional effects.

5-109

Solution The air in a hospital room is to be replaced every 20 minutes. The minimum diameter of the duct is to be determined if the air velocity is not to exceed a certain value.

Assumptions **1** The volume occupied by the furniture etc in the room is negligible. **2** The incoming conditioned air does not mix with the air in the room.

Analysis The volume of the room is



$$\mathbf{V} = (6\text{ m})(5\text{ m})(4\text{ m}) = 120\text{ m}^3$$

To empty this air in 20 min, the volume flow rate must be

$$\dot{\mathbf{V}} = \frac{\mathbf{V}}{\Delta t} = \frac{120\text{ m}^3}{20\times 60\text{ s}} = 0.10\text{ m}^3/\text{s}$$

If the mean velocity is 5 m/s, the diameter of the duct is

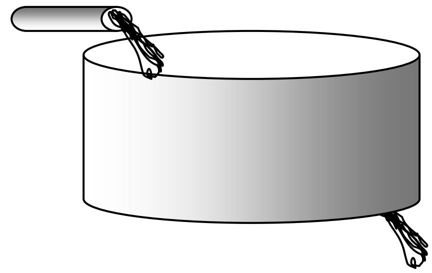
$$\dot{\mathbf{V}} = AV = \frac{\pi D^2}{4}V \quad \rightarrow \quad D = \sqrt{\frac{4\dot{\mathbf{V}}}{\pi V}} = \sqrt{\frac{4(0.10\text{ m}^3/\text{s})}{\pi(5\text{ m/s})}} = \mathbf{0.16\,m}$$

Therefore, the diameter of the duct must be at least 0.16 m to ensure that the air in the room is exchanged completely within 20 min while the mean velocity does not exceed 5 m/s.

Discussion This problem shows that engineering systems are sized to satisfy certain constraints imposed by certain considerations.

5-110

Solution The rate of accumulation of water in a pool and the rate of discharge are given. The rate supply of water to the pool is to be determined.



Assumptions **1** Water is supplied and discharged steadily. **2** The rate of evaporation of water is negligible. **3** No water is supplied or removed through other means.

Analysis The conservation of mass principle applied to the pool requires that the rate of increase in the amount of water in the pool be equal to the difference between the rate of supply of water and the rate of discharge. That is,

$$\frac{dm_{\text{pool}}}{dt} = \dot{m}_i - \dot{m}_e \quad \rightarrow \quad \dot{m}_i = \frac{dm_{\text{pool}}}{dt} + \dot{m}_e \quad \rightarrow \quad \dot{V}_i = \frac{d\dot{V}_{\text{pool}}}{dt} + \dot{V}_e$$

since the density of water is constant and thus the conservation of mass is equivalent to conservation of volume. The rate of discharge of water is

$$\dot{V}_e = A_e V_e = (\pi D^2/4)V_e = [\pi(0.05\,\text{m})^2/4](5\,\text{m/s}) = 0.00982\,\text{m}^3/\text{s}$$

The rate of accumulation of water in the pool is equal to the cross-section of the pool times the rate at which the water level rises,

$$\frac{dV_{\text{pool}}}{dt} = A_{\text{cross-section}}V_{\text{level}} = (3\,\text{m}\times 4\,\text{m})(0.015\,\text{m/min}) = 0.18\,\text{m}^3/\text{min} = 0.00300\,\text{m}^3/\text{s}$$

Substituting, the rate at which water is supplied to the pool is determined to be

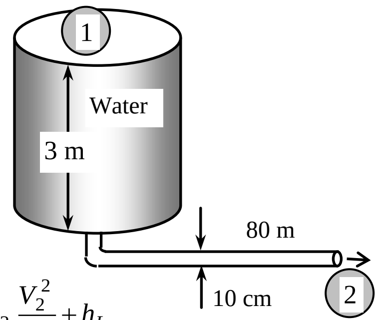
$$\dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e = 0.003 + 0.00982 = 0.01282\ \text{m}^3/\text{s} \cong \mathbf{0.0128\,m^3/s}$$

Therefore, water is supplied at a rate of 0.01282 m^3/s = 12.82 L/s.

Discussion This is a very simple application of the conservation of mass equations.

5-111

Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere through a long pipe with a specified head loss. The initial discharge velocity is to be determined.



Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** There are no pumps or turbines in the system. **4** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of the pipe. We take the reference level at the centerline of the orifice ($z_2 = 0$. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad z_1 = \alpha_2 \frac{V_2^2}{2g} + h_L$$

where $\alpha_2 = 1$ and the head loss is given to be $h_L = 1.5$ m. Solving for V_2 and substituting, the discharge velocity of water is determined to be

$$V_2 = \sqrt{2g(z_1 - h_L)} = \sqrt{2(9.81\,\text{m/s}^2)(3-1.5)\,\text{m}} = \mathbf{5.42\ m/s}$$

Discussion Note that this is the discharge velocity at the beginning, and the velocity will decrease as the water level in the tank drops. The head loss in that case will change since it depends on velocity.

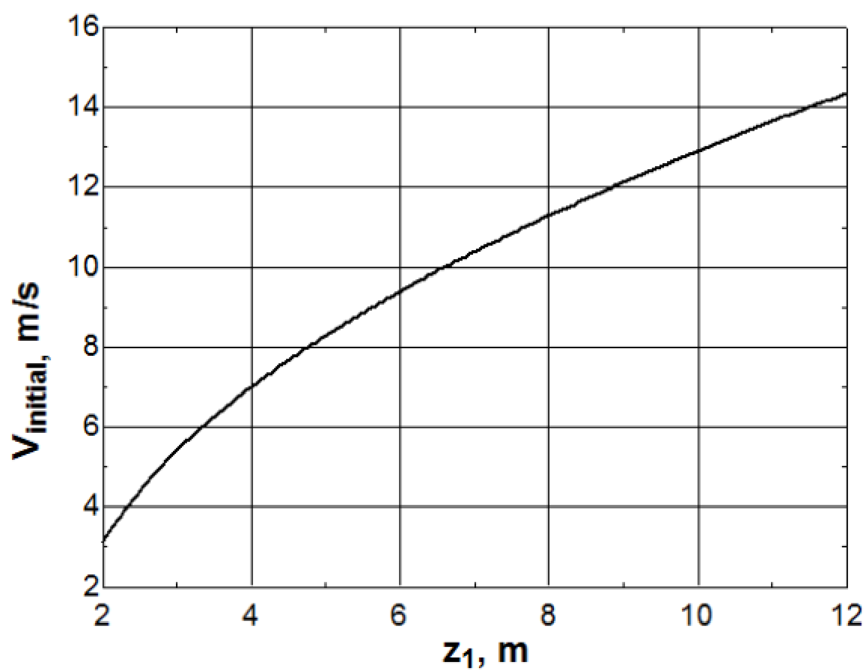
5-112

Solution The previous problem is reconsidered. The effect of the tank height on the initial discharge velocity of water from the completely filled tank as the tank height varies from 2 to 20 m in increments of 2 m at constant heat loss is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
h_L=1.5 "m"
D=0.10 "m"
V_initial=SQRT(2*g*(z1-h_L)) "m/s"
```

Tank height, $z1$, m	Head Loss, h_L, m	Initial velocity V_{initial}, m/s
2	1.5	3.13
3	1.5	5.42
4	1.5	7.00
5	1.5	8.29
6	1.5	9.40
7	1.5	10.39
8	1.5	11.29
9	1.5	12.13
10	1.5	12.91
11	1.5	13.65
12	1.5	14.35



Discussion The dependence of V on height is not linear, but rather V changes as the square root of z_1.

5-113

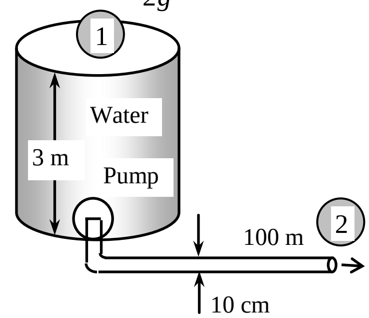
Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere through a long pipe equipped with a pump with a specified head loss. The required pump head to assure a certain velocity is to be determined.

Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of the pipe. We take the reference level at the centerline of the orifice ($z_2 = 0$), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad z_1 + h_{\text{pump,u}} = \alpha_2 \frac{V_2^2}{2g} + h_L$$

where $\alpha_2 = 1$ and the head loss is given to be $h_L = 1.5$ m. Solving for $h_{\text{pump, u}}$ and substituting, the required useful pump head is determined to be



$$h_{pum,u} = \sqrt{\frac{V_2^2}{2g} - z_1 + h_L} = \sqrt{\frac{(6.5\text{ m/s})^2}{2(9.81\text{ m/s}^2)} - (3\text{ m}) + (1.5\text{ m})} = \mathbf{0.808m}$$

Discussion Note that this is the required useful pump head at the beginning, and it will need to be increased as the water level in the tank drops to make up for the lost elevation head to maintain the constant discharge velocity.

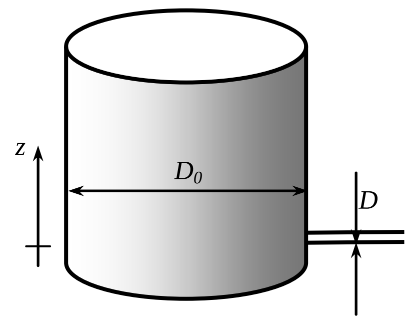
5-114

Solution A water tank open to the atmosphere is initially filled with water. The tank discharges to the atmosphere through a long pipe connected to a valve. The initial discharge velocity from the tank and the time required to empty the tank are to be determined.

Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The tank is considered to be empty when the water level drops to the center of the valve.

Analysis (*a*) Substituting the known quantities, the discharge velocity can be expressed as

$$V = \sqrt{\frac{2gz}{1.5 + fL/D}} = \sqrt{\frac{2gz}{1.5 + 0.015(80\text{ m})/(0.10\text{ m})}} = \sqrt{0.1481gz}$$



Then the initial discharge velocity becomes

$$V_1 = \sqrt{0.1481gz_1} = \sqrt{0.1481(9.81\text{ m/s}^2)(2\text{ m})} = \mathbf{1.705\text{ m/s}}$$

where z is the water height relative to the center of the orifice at that time.

(*b*) The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{V} = A_{\text{pipe}}V_2 = \frac{\pi D^2}{4}\sqrt{0.1481gz}$$

Then the amount of water that flows through the pipe during a differential time interval dt is

$$dV = \dot{V}dt = \frac{\pi D^2}{4}\sqrt{0.1481gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV = A_{\tan k}(-dz) = -\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{0.1481gz}\,dt = -\frac{\pi D_0^2}{4}dz \;\rightarrow\; dt = -\frac{D_0^2}{D^2}\frac{dz}{\sqrt{0.1481gz}} = -\frac{D_0^2}{D^2\sqrt{0.1481g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained tank) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2\sqrt{0.1481g}}\int_{z=z_1}^{0} z^{-1/2}dz \;\rightarrow\; t_f = -\frac{D_0^2}{D^2\sqrt{0.1481g}}\left|\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0} = \frac{2D_0^2}{D^2\sqrt{0.1481g}}z_1^{\frac{1}{2}}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{2D_0^2}{D^2}\sqrt{\frac{z_1}{0.1212g}} = \frac{2(8\text{ m})^2}{(0.1\text{ m})^2}\sqrt{\frac{2\text{ m}}{0.1481(9.81\text{ m/s}^2)}} = 15{,}018\text{ s} = \mathbf{4.17\text{ h}}$$

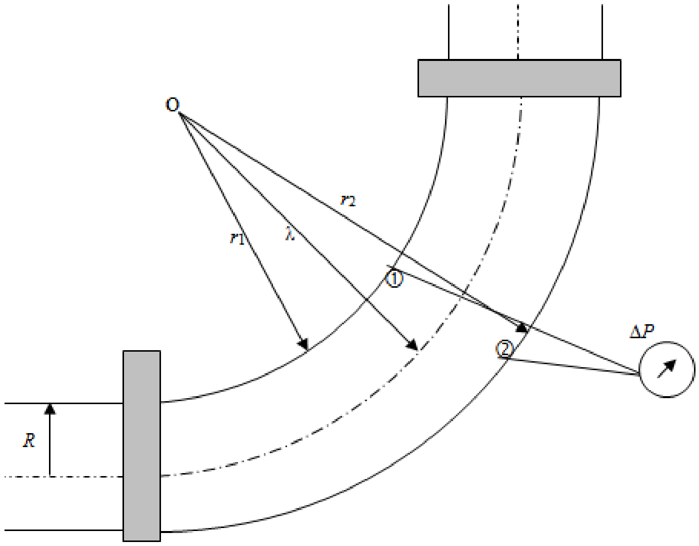
Discussion The draining time can be shortened considerably by installing a pump in the pipe.

5-115

Solution Elbow-type flow meters are used to measure flow rates. A relation for the flow rate as a function of given parameters is to be obtained.

Assumptions **1** The flow is steady and incompressible.

Analysis



Bernoulli equation in the direction r normal to streamlines for steady, incompressible flow is given by (Eq. 5-43)

$$\frac{P}{\rho} + \int \frac{V^2}{r} dr + gz = K \text{ (constant)}$$

where r is the local radius of curvature. Substituting $V = C/r$ and performing integration gives

$$\frac{P}{\rho} + \frac{V^2}{2} + gz = K_1 \text{ (another constant)} \qquad (1)$$

Applying to points 1 and 2 and taking $z_1 = z_2$ gives

$$\frac{P_1}{\rho} + \frac{V_1^2}{2} = \frac{P_2}{\rho} + \frac{V_2^2}{2} \text{ or } \frac{\Delta P}{\rho} = \frac{V_1^2 - V_2^2}{2} \qquad (1)$$

Substituting $V_1 = \dfrac{C}{r_1}$ and $V_2 = \dfrac{C}{r_2}$ into Eq. 1 and solving for C,

$$C = \sqrt{2\frac{\Delta p}{\rho}} \frac{r_1 r_2}{\sqrt{r_2^2 - r_1^2}} \qquad (2)$$

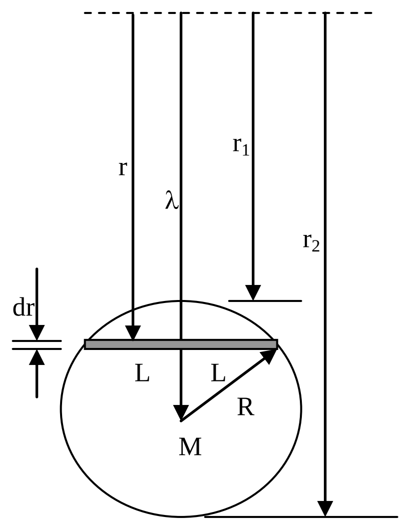
But from the figure, $r_1 = \lambda - R$ and $r_2 = \lambda + R$. Substituting, Eq. 2 becomes

$$C = (\lambda^2 - R^2)\sqrt{\frac{\Delta p}{2\rho\lambda R}} \qquad (3)$$

The flow rate is given by $Q = \int_A V dA$. Let us determine what dA would be. Considering the cross-sectional area of the elbow below we can write

$$dA = (2L)dr$$

From the sketch, $L = \sqrt{R^2 - (\lambda - r)^2}$



$$dA = 2\sqrt{R^2 - (\lambda - r)^2}\,dr$$

$$Q = \int V(2L)dr = 2\int \frac{C}{r}\sqrt{R^2 - (\lambda - r)^2}\,dr$$

$$Q = 2C\int_{r_1=\lambda-R}^{r_2=\lambda+R} \frac{\sqrt{R^2 - (\lambda - r)^2}}{r}dr$$

$$= 2\pi C\left(\lambda - \sqrt{\lambda^2 - R^2}\right)$$

Combining with Eq. 3, we obtain the flow rate to be

$$Q = \pi\sqrt{\frac{2\Delta P}{\rho g \lambda R}}\left(\lambda^2 - R^2\right)\left(\lambda - \sqrt{\lambda^2 - R^2}\right)$$

5-116

Solution A cylindrical water tank with a valve at the bottom contains air at the top part and water. The water height in the tank when water stops flowing out of the fully open valve is to be determined.

Assumptions **1** The flow is incompressible.

Analysis Applying Bernoulli Eq. from free surface to the point A would give

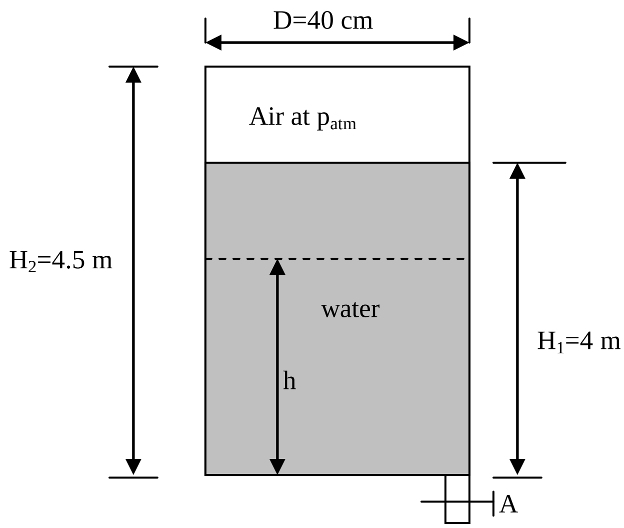
$$\frac{P_{air}}{\gamma}+\frac{V_{air}^{\ 2}}{2g}+Z_{air}=\frac{P_{atm}}{\gamma}+\frac{V_A^{\ 2}}{2g}+Z_A$$

$$P_{air,initial}.\boldsymbol{V}_{air,initial}=P_{air}\boldsymbol{V}_{air}=mRT=\text{constant}$$

$$P_{atm}\frac{\pi D^2}{4}(H_1-H_2)=P_{air}(H_1-h)$$

$$100000\times(4.5-4)=P_{air}(4.5-h)$$

$$P_{air}=\frac{50000}{4.5-h}$$



Plugging P_{air} into the Bernoulli Equation,

$$\frac{50000}{9810(4.5-h)}+0+h=\frac{10000}{9810}+\frac{V_A^{\ 2}}{2g}+Z_A$$

$$\frac{5.097}{4.5-h}+h=10.193+\frac{V_A^{\ 2}}{2g}$$

In case of no flow, V_A=0, that is

$$5.097+4.5h-h^2=45.87-10.193h$$

$$h^2-14.693h+40.773=0$$

$$h_1=10.98m \qquad \text{(not possible)}$$

$$h_2=\mathbf{3.71m}$$

Therefore, when the water level becomes h=3.71m the flow would stop. The discharged water volume is then

$$V=\pi\frac{D^2}{4}(H_2-h)=\pi\frac{0.4^2}{4}(4-3.71)=0.0364\ m^3$$

5-117

Solution A compressor supplies air to a rigid tank. A relation for the variation of pressure in the tank with time is to be obtained and and the time it will take for the absolute pressure in the tank to triple is to be determined.

Analysis (a) Applying conservation of mass to the CV enclosing tank, we get

$$\frac{\partial}{\partial t}\int_{CV}\rho d\forall + \int_{CS}\rho\vec{V}\vec{n}dA = 0\,, \text{ and } \forall\frac{d\rho}{dt} - \rho Q = 0$$

Therefore, we write

$$\int_{\rho_0}^{\rho}\frac{d\rho}{\rho} = \frac{Q}{\forall}\int_0^t dt \qquad \ln\frac{\rho}{\rho_0} = \frac{Q}{\forall}t$$

$$\frac{\rho}{\rho_0} = e^{\frac{Q}{\forall}t} \qquad (1)$$

On the other hand, considering an adiabatic flow we have,

$$\frac{P}{\rho^k} = \frac{P_0}{\rho_0{}^k} \Rightarrow \left(\frac{\rho}{\rho_0}\right)^k = \frac{P}{P_0} \qquad (2)$$

From the Equation 1;

$$\left(\frac{\rho}{\rho_0}\right)^k = e^{\frac{Q}{\forall}kt} = \frac{P}{P_0}$$

Therefore

$$P = P_0 e^{\frac{Q}{\forall}kt}$$

(b) $P = 3P_0$

$$3P_0 = P_0 e^{\frac{Q}{\forall}kt}\,, \text{ or } \ln 3 = \frac{Q}{\forall}kt$$

$$t = \frac{\forall}{Q}.\frac{\ln 3}{k} = \frac{1.5}{0.05}.\frac{\ln 3}{1.4} = \mathbf{23.5\,s}$$

5-118

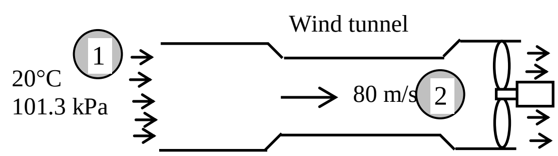
Solution A wind tunnel draws atmospheric air by a large fan. For a given air velocity, the pressure in the tunnel is to be determined.

Assumptions **1**The flow through the pipe is steady, incompressible, and irrotational with negligible friction (so that the Bernoulli equation is applicable). **2** Air is and ideal gas.

Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K.

Analysis We take point 1 in atmospheric air before it enters the wind tunnel (and thus $P_1 = P_{atm}$ and $V_1 \cong 0$), and point 2 in the wind tunnel. Noting that $z_1 = z_2$ (or, the elevation effects are negligible for gases), the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad P_2=P_1-\frac{\rho V_2^2}{2} \qquad (1)$$



where

$$\rho=\frac{P}{RT}=\frac{101.3\,\text{kPa}}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}=1.205\,\text{kg/m}^3$$

Substituting, the pressure in the wind tunnel is determined to be

$$P_2=(101.3\,\text{kPa})-(1.205\,\text{kg/m}^3)\frac{(80\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right)=\mathbf{97.4\,kPa}$$

Discussion Note that the velocity in a wind tunnel increases at the expense of pressure. In reality, the pressure will be even lower because of losses.

Fundamentals of Engineering (FE) Exam Problems

5-119

Water flows in a 5-cm-diameter pipe at a velocity of 0.75 m/s. The mass flow rate of water in the pipe is

(*a*) 353 kg/min (*b*) 75 kg/min (*c*) 37.5 kg/min (*d*) 1.47 kg/min (*e*) 88.4 kg/min

Answer (*e*) 88.4 kg/min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.05 [m]
V=0.75 [m/s]
rho=1000 [kg/m^3]
A_c=pi*D^2/4
m_dot=rho*A_c*V*Convert(kg/s, kg/min)
```

5-120

Air at 100 kPa and 20°C flows in a 12-cm-diameter pipe at a rate of 9.5 kg/min. The velocity of air in the pipe is

(*a*) 1.4 m/s (*b*) 6.0 m/s (*c*) 9.5 m/s (*d*) 11.8 m/s (*e*) 14.0 m/s

Answer (*d*) 11.8 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=100 [kPa]
T=(20+273.15) [K]
D=0.12 [m]
m_dot=9.5 [kg/min]*Convert(kg/min, kg/s)
R=0.287 [kJ/kg-K]
rho=P/(R*T)
A_c=pi*D^2/4
V=m_dot/(rho*A_c)
```

5-121

A water tank initially contains 140 L of water. Now, equal rates of cold and hot water enter the tank for a period of 30 minutes while warm water is discharged from the tank at a rate of 25 L/min. The amount of water in the tank at the end of this 30-min period is 50 L. The rate of hot water entering the tank is

(*a*) 33 L/min (*b*) 25 L/min (*c*) 11 L/min (*d*) 7 L/min (*e*) 5 L/min

Answer (*c*) 11 L/min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V1=140 [L]
time=30 [min]
V_dot_out=25 [L/min]
V2=50 [L]
V_out=V_dot_out*time
V_in-V_out=V2-V1
V_in=V_cold+V_hot
V_cold=V_hot
V_dot_hot=V_hot/time
```

5-122

Water enters a 4-cm-diameter pipe at a velocity of 1 m/s. The diameter of the pipe is reduced to 3 cm at the exit. The velocity of the water at the exit is

(*a*) 1.78 m/s (*b*) 1.25 m/s (*c*) 1 m/s(*d*) 0.75 m/s (*e*) 0.50 m/s

Answer (*a*) 1.78 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D1=0.04 [m]
V1=1 [m/s]
D2=0.03 [m]
A_c1=pi*D1^2/4
A_c2=pi*D2^2/4
A_c1*V1=A_c2*V2
```

5-123

The pressure of water is increased from 100 kPa to 900 kPa by a pump. The mechanical energy increase of water is

(*a*) 0.9 kJ/kg (*b*) 0.5 kJ/kg (*c*) 500 kJ/kg (*d*) 0.8 kJ/kg (*e*) 800 kJ/kg

Answer (*d*) 0.8 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 [kPa]
P2=900 [kPa]
rho=1000 [kg/m^3]
DELTAe=(P2-P1)/rho
```

5-124

A 75-m-high water body that is open to the atmosphere is available. Water is run through a turbine at a rate of 200 L/s at the bottom of the water body. The pressure difference across the turbine is

(*a*) 736 kPa (*b*) 0.736 kPa (*c*) 1.47 kPa (*d*) 1470 kPa (*e*) 368 kPa

Answer (*a*) 736 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h=75 [m]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
e=g*h*Convert(J/kg, kJ/kg)
DELTAP=e*rho
```

5-125

A pump is used to increase the pressure of water from 100 kPa to 900 kPa at a rate of 160 L/min. If the shaft power input to the pump is 3 kW, the efficiency of the pump is

(*a*) 0.532 (*b*) 0.660 (*c*) 0.711 (*d*) 0.747 (*e*) 0.855

Answer (*c*) 0.711

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 [kPa]
P2=900 [kPa]
V_dot=160 [L/min]*Convert(L/min, m^3/s)
W_dot_shaft=3 [kW]
rho=1000 [kg/m^3]
m_dot=rho*V_dot
DELTAE_dot=m_dot*(P2-P1)/rho
eta_pump=DELTAE_dot/W_dot_shaft
```

5-126

A hydraulic turbine is used to generate power by using the water in a dam. The elevation difference between the free surfaces upstream and downstream of the dam is 120 m. The water is supplied to the turbine at a rate of 150 kg/s. If the shaft power output from the turbine is 155 kW, the efficiency of the turbine is

(*a*) 0.77 (*b*) 0.80 (*c*) 0.82 (*d*) 0.85 (*e*) 0.88

Answer (*e*) 0.88

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h=120 [m]
m_dot=150 [kg/s]
W_dot_shaft=155 [kW]
g=9.81 [m/s^2]
DELTAE_dot=m_dot*g*h*Convert(W, kW)
eta_turbine=W_dot_shaft/DELTAE_dot
```

5-127

The motor of a pump consumes 1.05 hp of electricity. The pump increases the pressure of water from 120 kPa to 1100 kPa at a rate of 35 L/min. If the motor efficiency is 94 percent, the pump efficiency is

(*a*) 0.75 (*b*) 0.78 (*c*) 0.82 (*d*) 0.85 (*e*) 0.88

Answer (*b*) 0.78

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W_dot_elec=1.05 [hp]*Convert(hp, kW)
P1=120 [kPa]
P2=1100 [kPa]
V_dot=35 [L/min]*Convert(L/min, m^3/s)
eta_motor=0.94
rho=1000 [kg/m^3]
m_dot=rho*V_dot
DELTAE_dot=m_dot*(P2-P1)/rho
eta_motor_pump=DELTAE_dot/W_dot_elec
eta_pump=eta_motor_pump/eta_motor
```

5-128

The efficiency of a hydraulic turbine-generator unit is specified to be 85 percent. If the generator efficiency is 96 percent, the turbine efficiency is

(*a*) 0.816 (*b*) 0.850 (*c*) 0.862 (*d*) 0.885 (*e*) 0.960

Answer (*d*) 0.885

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
eta_turbine_gen=0.85
eta_gen=0.96
eta_turbine=eta_turbine_gen/eta_gen
```

5-129

Which parameter is not related in the Bernoulli equation?

(*a*) Density (*b*) Velocity (*c*) Time (*d*) Pressure (*e*) Elevation

Answer (*c*) Time

5-130

Consider incompressible, frictionless flow of a fluid in a horizontal piping. The pressure and velocity of a fluid is measured to be 150 kPa and 1.25 m/s at a specified point. The density of the fluid is 700 kg/m^3. If the pressure is 140 kPa at another point, the velocity of the fluid at that point is

(*a*) 1.26 m/s (*b*) 1.34 m/s (*c*) 3.75 m/s (*d*) 5.49 m/s (*e*) 7.30 m/s

Answer (*d*) 5.49 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=150 [kPa]
V1=1.25 [m/s]
rho=700 [kg/m^3]
P2=140 [kPa]
P1/rho+V1^2/2*Convert(m^2/s^2, kJ/kg)=P2/rho+V2^2/2*Convert(m^2/s^2, kJ/kg)
```

5-131

Consider incompressible, frictionless flow of water in a vertical piping. The pressure is 240 kPa at 2 m from the ground level. The velocity of water does not change during this flow. The pressure at 15 m from the ground level is

(*a*) 227 kPa (*b*) 174 kPa (*c*) 127 kPa (*d*) 120 kPa (*e*) 113 kPa

Answer (*e*) 113 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=240 [kPa]
z1=2 [m]
z2=15 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P1/rho+g*z1*Convert(m^2/s^2, kJ/kg)=P2/rho+g*z2*Convert(m^2/s^2, kJ/kg)
```

5-132

Consider water flow in a piping network. The pressure, velocity, and elevation at a specified point (point 1) of the flow are 150 kPa, 1.8 m/s, and 14 m. The pressure and velocity at point 2 are 165 kPa and 2.4 m/s. Neglecting frictional effects, the elevation at point 2 is

(*a*) 12.4 m (*b*) 9.3 m (*c*) 14.2 m (*d*) 10.3 m (*e*) 7.6 m

Answer (*a*) 12.4 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=150 [kPa]
V1=1.8 [m/s]
z1=14 [m]
P2=165 [kPa]
V2=2.4 [m/s]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P1/rho+(V1^2/2+g*z1)*Convert(m^2/s^2, kJ/kg)=P2/rho+(V2^2/2+g*z2)*Convert(m^2/s^2, kJ/kg)
```

5-133

The static and stagnation pressures of a fluid in a pipe are measured by a piezometer and a pitot tube to be 200 kPa and 210 kPa, respectively. If the density of the fluid is 550 kg/m^3, the velocity of the fluid is

(*a*) 10 m/s (*b*) 6.03 m/s (*c*) 5.55 m/s (*d*) 3.67 m/s (*e*) 0.19 m/s

Answer (*b*) 6.03 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P_stag=210 [kPa]
P=200 [kPa]
rho=550 [kg/m^3]
P_stag=P+rho*V^2/2*Convert(Pa, kPa)
```

5-134

The static and stagnation pressures of a fluid in a pipe are measured by a piezometer and a pitot tube. The heights of the fluid in the piozemeter and pitot tube are measured to be 2.2 m and 2.0 m, respectively. If the density of the fluid is 5000 kg/m^3, the velocity of the fluid in the pipe is

(*a*) 0.92 m/s (*b*) 1.43 m/s (*c*) 1.65 m/s (*d*) 1.98 m/s (*e*) 2.39 m/s

Answer (*d*) 1.98 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_piezo=2 [m]
h_pitot=2.2 [m]
rho=5000 [kg/m^3]
g=9.81 [m/s^2]
P=rho*g*h_piezo
P_stag=rho*g*h_pitot
P_stag=P+rho*V^2/2
```

5-135

The difference between the heights of energy grade line (EGL) and hydraulic grade line (HGL) is equal to

(*a*) z (*b*) $P/\rho g$ (*c*) $V^2/2g$(*d*) $z + P/\rho g$ (*e*) $z + V^2/2g$

Answer (*c*) $V^2/2g$

5-136

Water at 120 kPa (gage) is flowing in a horizontal pipe at a velocity of 1.15 m/s. The pipe makes a 90° angle at the exit and the water exits the pipe vertically into the air. The maximum height the water jet can rise is

(*a*) 6.9 m (*b*) 7.8 m (*c*) 9.4 m (*d*) 11.5 m (*e*) 12.3 m

Answer (*e*) 12.3 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=120000 [Pa]
V1=1.15 [m/s]
z1=0 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P2=0 [Pa]
V2=0 [m/s]
P1/(rho*g)+V1^2/(2*g)+z1=P2/(rho*g)+V2^2/(2*g)+z2
```

5-137

Water is withdrawn at the bottom of a large tank open to the atmosphere. The water velocity is 6.6 m/s. The minimum height of the water in the tank is

(*a*) 2.22 m (*b*) 3.04 m (*c*) 4.33 m (*d*) 5.75 m (*e*) 6.60 m

Answer (*a*) 2.22 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V2=6.6 [m/s]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P1=0 [Pa]
P2=0 [Pa]
V1=0 [m/s]
z2=0 [m]
P1/(rho*g)+V1^2/(2*g)+z1=P2/(rho*g)+V2^2/(2*g)+z2
```

5-138

Water at 80 kPa (gage) enters a horizontal pipe at a velocity of 1.7 m/s. The pipe makes a 90° angle at the exit and the water exits the pipe vertically into the air. Take the correction factor to be 1. If the irreversible head loss between the inlet and exit of the pipe is 3 m, the height the water jet can rise is

(*a*) 3.4 m (*b*) 5.3 m (*c*) 8.2 m (*d*) 10.5 m (*e*) 12.3 m

Answer (*b*) 5.3 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=80000 [Pa]
V1=1.7 [m/s]
h_L=3 [m]
z1=0 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P2=0 [Pa]
V2=0 [m/s]
P1/(rho*g)+V1^2/(2*g)+z1=P2/(rho*g)+V2^2/(2*g)+z2+h_L
```

5-139

Seawater is to be pumped into a large tank at a rate of 165 kg/min. The tank is open to the atmosphere and the water enters the tank from a 80-m-height. The overall efficiency of the motor-pump unit is 75 percent and the motor consumes electricity at a rate of 3.2 kW. Take the correction factor to be 1. If the irreversible head loss in the piping is 7 m, the velocity of the water at the tank inlet is

(*a*) 2.34 m/s (*b*) 4.05 m/s (*c*) 6.21 m/s (*d*) 8.33 m/s (*e*) 10.7 m/s

Answer (*c*) 6.21 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_dot=(165/60) [kg/s]
z2=80 [m]
eta_motor_pump=0.75
W_dot_elec=3200 [W]
h_L=7 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
z1=0 [m]
V1=0 [m/s]
P1=0 [Pa]
P2=0 [Pa]
h_pump_u=eta_motor_pump*W_dot_elec/(m_dot*g)
P1/(rho*g)+V1^2/(2*g)+z1+h_pump_u=P2/(rho*g)+V2^2/(2*g)+z2+h_L
```

5-140

Water enters a pump at 350 kPa at a rate of 1 kg/s. The water leaving the pump enters a turbine in which the pressure is reduced and electricity is produced. The shaft power input to the pump is 1 kW and the shaft power output from the turbine is 1 kW. Bothe the pump and turbine are 90 percent efficient. If the elevation and velocity of the water remain constant throughout the flow and the irreversible head loss is 1 m, the pressure of water at the turbine exit is

(*a*) 350 kPa (*b*) 100 kPa (*c*) 173 kPa (*d*) 218 kPa (*e*) 129 kPa

Answer (*e*) 129 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_dot=1 [kg/s]
P1=350000 [Pa]
W_dot_pump=1000 [W]
W_dot_turbine=1000 [W]
eta_pump=0.90
eta_turbine=0.90
h_L=1 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
z1=0 [m]
z2=0 [m]
V1=1 [m/s]
V2=1 [m/s]
h_pump_u=eta_pump*W_dot_pump/(m_dot*g)
h_turbine_e=W_dot_turbine/(eta_turbine*m_dot*g)
P1/(rho*g)+V1^2/(2*g)+z1+h_pump_u=P2/(rho*g)+V2^2/(2*g)+z2+h_turbine_e+h_L
```

5-141

An adibatic pump is used to increase the pressure of water from 100 kPa to 500 kPa at a rate of 400 L/min. If the efficiency of the pump is 75 percent, the maximum temperature rise of the water across the pump is

(*a*) 0.096°C (*b*) 0.058°C (*c*) 0.035°C (*d*) 1.52°C (*e*) 1.27°C

Answer (*a*) 0.096°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 [kPa]
P2=500 [kPa]
V_dot=400 [L/min]*Convert(L/min, m^3/s)
eta_pump=0.75
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
c=4.18 [kJ/kg-C]
m_dot=rho*V_dot
DELTAE_dot_mech=m_dot*(P2-P1)/rho
W_dot_pump=DELTAE_dot_mech/eta_pump
E_dot_mech_loss=W_dot_pump-DELTAE_dot_mech
DELTAT=DELTAE_dot_mech/(m_dot*c)
```

5-142

The shaft power from a 90 percent-efficient turbine is 500 kW. If the mass flow rate through the turbine is 575 kg/s, the extracted head removed from the fluid by the turbine is

(*a*) 48.7 m (*b*) 57.5 m (*c*) 147 m (*d*) 139 m (*e*) 98.5 m

Answer (*e*) 98.5 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
eta_turbine=0.90
W_dot_turbine=500000 [W]
m_dot=575 [kg/s]
g=9.81 [m/s^2]
h_turbine_e=W_dot_turbine/(eta_turbine*m_dot*g)
```

Design and Essay Problems

5-143 to 5-147

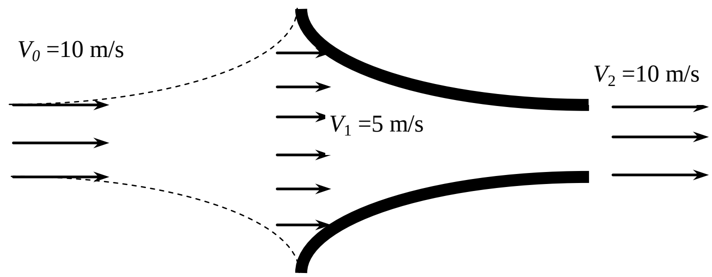
Solution Students' essays and designs should be unique and will differ from each other.

5-148

Solution We are to evaluate a proposed modification to a wind turbine.

Analysis Using the mass and the Bernoulli equations, it can be shown that **this is a bad idea** – the velocity at the exit of nozzle is equal to the wind velocity. (The velocity at nozzle inlet is much lower). Sample calculation using EES using a wind velocity of 10 m/s:

```
V0=10 "m/s"
rho=1.2 "kg/m3"
g=9.81 "m/s2"
A1=2 "m2"
A2=1 "m2"
A1*V1=A2*V2
P1/rho+V1^2/2=V2^2/2
m=rho*A1*V1
m*V0^2/2=m*V2^2/2
```



Results: V_1 = 5 m/s, V_2 = 10 m/s, m = 12 kg/s (mass flow rate).

Discussion Students' approaches may be different, but they should come to the same conclusion – this does not help.

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

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CHAPTER 6
MOMENTUM ANALYSIS OF FLOW SYSTEMS

Newton's Laws and Conservation of Momentum

6-1C

Solution We are to express Newton's three laws.

Analysis *Newton's first law* states that "**a body at rest remains at rest, and a body in motion remains in motion at the same velocity in a straight path when the net force acting on it is zero.**" Therefore, a body tends to preserve its state or inertia. *Newton's second law* states that "**the acceleration of a body is proportional to the net force acting on it and is inversely proportional to its mass.**" *Newton's third law states* "***when a body exerts a force on a second body, the second body exerts an equal and opposite force on the first.***"

Discussion As we shall see in later chapters, the differential equation of fluid motion is based on Newton's second law.

6-2C

Solution We are to discuss Newton's second law for rotating bodies.

Analysis Newton's second law of motion, also called the *angular momentum equation*, is expressed as "**the rate of change of the angular momentum of a body is equal to the net torque acting it**." For a non-rigid body with zero net torque, the angular momentum remains constant, but the **angular velocity changes in accordance with $I\omega$ = *constant*** where I is the moment of inertia of the body.

Discussion Angular momentum is analogous to linear momentum in this way: Linear momentum does not change unless a force acts on it. Angular momentum does not change unless a torque acts on it.

6-3C

Solution We are to discuss if momentum is a vector, and its direction.

Analysis Since momentum ($m\vec{V}$) is the product of a vector (velocity) and a scalar (mass), **momentum must be a vector that points in the same direction as the velocity vector**.

Discussion In the general case, we must solve three components of the linear momentum equation, since it is a vector equation.

6-4C

Solution We are to discuss the conservation of momentum principle.

Analysis The *conservation of momentum principle* is expressed as "**the momentum of a system remains constant when the net force acting on it is zero, and thus the momentum of such systems is conserved**". The momentum of a body remains constant if the net force acting on it is zero.

Discussion Momentum is not conserved in general, because when we apply a force, the momentum changes.

Linear Momentum Equation

6-5C

Solution We are to compare the reaction force on two fire hoses.

Analysis **The fireman who holds the hose backwards so that the water makes a U-turn before being discharged will experience a greater reaction force**. This is because of the vector nature of the momentum equation. Specifically, the inflow and outflow terms end up with the same sign (they add together) for the case with the U-turn, whereas they have opposite signs (one partially cancels out the other) for the case without the U-turn.

Discussion Direction is not an issue with the conservation of mass or energy equations, since they are scalar equations.

6-6C

Solution We are to discuss surface forces in a control volume analysis.

Analysis All surface forces arise as **the control volume is isolated from its surroundings for analysis, and the effect of any detached object is accounted for by a force at that location**. We can minimize the number of surface forces exposed by **choosing the control volume (wisely) such that the forces that we are not interested in remain internal, and thus they do not complicate the analysis**. A well-chosen control volume exposes only the forces that are to be determined (such as reaction forces) and a minimum number of other forces.

Discussion There are many choices of control volume for a given problem. Although there are not really "wrong" and "right" choices of control volume, there certainly are "wise" and "unwise" choices of control volume.

6-7C

Solution We are to discuss the importance of the RTT, and its relationship to the linear momentum equation.

Analysis **The relationship between the time rates of change of an extensive property for a system and for a control volume** is expressed by the *Reynolds transport theorem* (RTT), which provides the link between the system and control volume concepts. The linear momentum equation is obtained by **setting** $b = \vec{V}$ **and thus** $B = m\vec{V}$ **in the Reynolds transport theorem**.

Discussion Newton's second law applies directly to a system of fixed mass, but we use the RTT to transform from the system formulation to the control volume formulation.

6-8C

Solution We are to discuss the momentum flux correction factor, and its significance.

Analysis The *momentum-flux correction factor* β enables us to express the momentum flux in terms of the mass flow rate and mean flow velocity as $\boxed{\int_{A_c} \rho\vec{V}(\vec{V}\cdot\vec{n})dA_c = \beta\dot{m}\vec{V}_{avg}}$. The value of β is unity for uniform flow, such as a jet flow, nearly unity for fully developed turbulent pipe flow (between 1.01 and 1.04), but about 1.3 for fully developed laminar pipe flow. **So it is significant and should be considered in laminar flow**; it is often ignored in turbulent flow.

Discussion Even though β is nearly unity for many turbulent flows, it is wise not to ignore it.

6-9C

Solution We are to discuss the momentum equation for steady one-D flow with no external forces.

Analysis The momentum equation for steady flow for the case of no external forces is

$$\boxed{\sum \vec{F} = 0 = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}}$$

where the left hand side is the **net force acting on the control volume** (which is zero here), the first term on the right hand side is the **incoming momentum flux**, and the second term on the right is the **outgoing momentum flux** by mass.

Discussion This is a special simplified case of the more general momentum equation, since there are no forces. In this case we can say that momentum is conserved.

6-10C

Solution We are to explain why we can usually work with gage pressure rather than absolute pressure.

Analysis In the application of the momentum equation, we can disregard the atmospheric pressure and work with gage pressures only since the **atmospheric pressure acts in all directions**, and its effect cancels out in every direction.

Discussion In some applications, it is better to use absolute pressure everywhere, but for most practical engineering problems, it is simpler to use gage pressure everywhere, with no loss of accuracy.

6-11C

Solution We are to discuss if the upper limit of a rocket's velocity is limited to V, its discharge velocity.

Analysis **No, *V* is not the upper limit to the rocket's ultimate velocity**. Without friction the rocket velocity will continue to increase (i.e., it will continue to accelerate) as more gas is expelled out the nozzle.

Discussion This is a simple application of Newton's second law. As long as there is a force acting on the rocket, it will continue to accelerate, regardless of how that force is generated.

6-12C

Solution We are to describe how a helicopter can hover.

Analysis A helicopter hovers because **the strong downdraft of air, caused by the overhead propeller blades, manifests a momentum in the air stream**. This momentum must be countered by the helicopter lift force.

Discussion In essence, the helicopter stays aloft by pushing down on the air with a net force equal to its weight.

6-13C

Solution We are to discuss the power required for a helicopter to hover at various altitudes.

Analysis Since the air density decreases, **it requires more energy for a helicopter to hover at higher altitudes**, because more air must be forced into the downdraft by the helicopter blades to provide the same lift force. Therefore, it takes more power for a helicopter to hover on the top of a high mountain than it does at sea level.

Discussion This is consistent with the limiting case – if there were no air, the helicopter would not be able to hover at all. There would be no air to push down.

6-14C

Solution We are to discuss helicopter performance in summer versus winter.

Analysis In winter the air is generally colder, and thus denser. Therefore, less air must be driven by the blades to provide the same helicopter lift, requiring less power. **Less energy is required in the winter**.

Discussion However, it is also harder for the blades to move through the denser cold air, so there is more torque required of the engine in cold weather.

6-15C

Solution We are to discuss if the force required to hold a plate stationary doubles when the jet velocity doubles.

Analysis **No, the force will not double**. In fact, the force required to hold the plate against the horizontal water stream **will increase by a factor of 4** when the velocity is doubled since

$$F = \dot{m}V = (\rho AV)V = \rho AV^2$$

and thus the *force is proportional to the square of the velocity*.

Discussion You can think of it this way: Since momentum flux is mass flow rate times velocity, a doubling of the velocity doubles both the mass flow rate *and* the velocity, increasing the momentum flux by a factor of four.

6-16C

Solution We are to describe and discuss body forces and surface forces.

Analysis The forces acting on the control volume consist of *body forces* that **act throughout the entire body of the control volume** (such as gravity, electric, and magnetic forces) and *surface forces* that **act on the control surface** (such as the pressure forces and reaction forces at points of contact). The *net force* acting on a control volume is the **sum of all body *and* surface forces**. Fluid **weight is a body force**, and **pressure is a surface force** (acting per unit area).

Discussion In a general fluid flow, the flow is influenced by both body and surface forces.

6-17C

Solution We are to discuss the acceleration of a cart hit by a water jet.

Analysis **The acceleration is not be constant since the force is not constant**. The impulse force exerted by the water on the plate is $F = \dot{m}V = (\rho AV)V = \rho AV^2$, where V is the relative velocity between the water and the plate, which is moving. The magnitude of the plate acceleration is thus $a = F/m$**.** But as the plate begins to move, V decreases, so the acceleration must also decrease.

Discussion It is the *relative* velocity of the water jet on the cart that contributes to the cart's acceleration.

6-18C

Solution We are to discuss the maximum possible velocity of a cart hit by a water jet.

Analysis The **maximum possible velocity for the plate is the velocity of the water jet**. As long as the plate is moving slower than the jet, the water exerts a force on the plate, which causes it to accelerate, until terminal jet velocity is reached.

Discussion Once the relative velocity is zero, the jet supplies no force to the cart, and thus it cannot accelerate further.

6-19

Solution The velocity distribution for turbulent flow of water in a pipe is considered. The darg force exerted on the pipe by water flow is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

Properties We take the density of water to be 1000 kg/m^3.

Analysis Applying x-component of momentum equation,

$$\frac{\partial}{\partial t}\int_{cv}\vec{V}\rho dV+\int_{cs}\vec{V}\rho\vec{V}\vec{n}dA=\sum F_{cv}$$

$$\rho U_{ave}(-U_{ave})A_1+\rho\int_0^R u^2 dA=p_1A_1-p_2A_2-F_D$$

Recalling that $U_{ave}=0.816U_{\max}$ for turbulent flow, we write

$$u=\frac{U_{ave}}{0.816}(1-r/R)^{1/7}=1.2255U_{ave}(1-r/R)^{1/7}$$

Therefore,

$$-\rho U_{ave}^2A_1+\rho\int_0^R 1.5U_{ave}^2(1-r/R)^{2/7}2\pi rdr=(p_1-p_2)A-F_D$$

$$-\rho U_{ave}^2A_1+3\pi U_{ave}^2\rho\int_0^R(1-r/R)^{2/7}rdr=(p_1-p_2)A-F_D$$

$$\int_0^R(1-r/R)^{2/7}rdr=R^2\int_0^1\left(1-\frac{r}{R}\right)^{2/7}\frac{r}{R}d\left(\frac{r}{R}\right)$$

Setting $1-r/R=v$, we obtain

$$\int_0^R(1-r/R)^{2/7}rdr=-R^2\int_0^1(v^{2/7}-v^{9/7})dv=-R^2\left[\frac{7}{9}\left(1-\frac{r}{R}\right)^{9/7}-\frac{7}{16}\left(1-\frac{r}{R}\right)^{16/7}\right]_0^1$$

Finally we get

$$\int_0^R(1-r/R)^{2/7}rdr=R^2\frac{49}{144}$$

Therefore, the momentum equation takes the form

$$-\rho U_{ave}^2A_1+3\pi U_{ave}^2\rho R^2\frac{49}{144}=\Delta pA-F_D$$

$$F_D=\Delta pA+\rho U_{ave}^2A_1-\rho\pi U_{ave}^2R^2\frac{49}{48}$$

$$F_D=10{,}000\times\frac{\pi 0.1^2}{4}+1000\times 3^2\frac{\pi 0.1^2}{4}-1000\pi 3^2\left(\frac{0.1}{2}\right)^2\frac{49}{48}\approx\mathbf{77\ N}$$

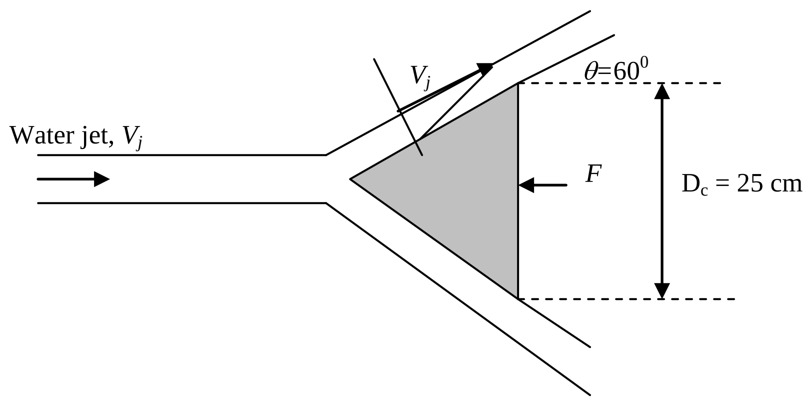
6-20

Solution A horizontal water jet is deflected by a stationary cone. The horizontal force needed to hold the cone stationary is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

Properties We take the density of water to be 1000 kg/m^3.

Analysis



Considering front view of the cone , we apply the conservation of mass,

$$0 = -Q_j + \int_A V_{exit} dA \Rightarrow Q_j \approx \frac{V_j}{2} \pi D_c h$$

$$\frac{\pi}{4} 0.025^2 \times 40 = \frac{40}{2} \pi 0.25 h \Rightarrow h = 1.25 \times 10^{-3} \ m = 1.25 \ mm$$

Linear momentum equation in the x direction gives

$$\int \vec{V} \rho \vec{V} \vec{n} dA = \sum F_x$$

$$-V_j \rho V_j A_1 + \int_0^h (V Cos\theta) \rho V (\pi D_c) dY = -F$$

Since velocity is linear, it is in the form of $V = a + bY$. It is given that V=0 when Y=0, and V=V_j when Y=h=1.25x10^{-3} m. Therefore we obtain

$$V = 32000Y$$

$$-1000 \times 40^2 \frac{\pi}{4} 0.025^2 + 1000 \times \pi 0.25 \times (Cos60) \times 32000^2 \int_0^{1.25\times 10^{-3}} Y^2 dY = -F$$

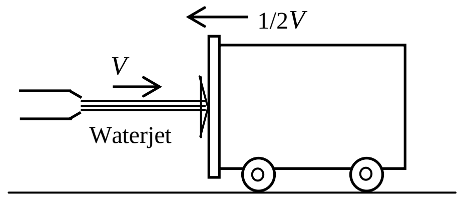
$$F = -785 + 262 = \mathbf{523\ N}$$

6-21

Solution A water jet of velocity V impinges on a plate moving toward the water jet with velocity ½V. The force required to move the plate towards the jet is to be determined in terms of F acting on the stationary plate.

Assumptions **1** The flow is steady and incompressible. **2** The plate is vertical and the jet is normal to plate. **3** The pressure on both sides of the plate is atmospheric pressure (and thus its effect cancels out). **4** Fiction during motion is negligible. **5** There is no acceleration of the plate. **6** The water splashes off the sides of the plate in a plane normal to the jet. **6** Jet flow is nearly uniform and thus the effect of the momentum-flux correction factor is negligible, $\beta \cong 1$.

Analysis We take the plate as the control volume. The relative velocity between the plate and the jet is V when the plate is stationary, and 1.5V when the plate is moving with a velocity ½V towards the plate. Then the momentum equation for steady flow in the horizontal direction reduces to



$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V} \quad \rightarrow \quad -F_R = -\dot{m}_i V_i \quad \rightarrow \quad F_R = \dot{m}_i V_i$$

Stationary plate: $(V_i = V \quad \text{and} \quad \dot{m}_i = \rho A V_i = \rho A V) \rightarrow \quad F_R = \rho A V^2 = F$

Moving plate: $(V_i = 1.5V \quad \text{and} \quad \dot{m}_i = \rho A V_i = \rho A(1.5V))$

$$\rightarrow \quad F_R = \rho A(1.5V)^2 = 2.25\rho A V^2 = 2.25F$$

Therefore, the force required to hold the plate stationary against the oncoming water jet becomes **2.25 times greater** when the jet velocity becomes 1.5 times greater.

Discussion Note that when the plate is stationary, V is also the jet velocity. But if the plate moves toward the stream with velocity ½V, then the *relative* velocity is 1.5V, and the amount of mass striking the plate (and falling off its sides) per unit time also increases by 50%.

6-22

Solution A 90° elbow deflects water upwards and discharges it to the atmosphere at a specified rate. The gage pressure at the inlet of the elbow and the anchoring force needed to hold the elbow in place are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Frictional effects are negligible in the calculation of the pressure drop (so that the Bernoulli equation can be used). **3** The weight of the elbow and the water in it is negligible. **4** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero. **5** The momentum-flux correction factor for each inlet and outlet is given to be $\beta = 1.03$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis (*a*) We take the elbow as the control volume, and designate the entrance by 1 and the outlet by 2. We also designate the horizontal coordinate by *x* (with the direction of flow as being the positive direction) and the vertical coordinate by *z*. The continuity equation for this one-inlet one-outlet steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m} = 40$ kg/s. Noting that $\dot{m} = \rho AV$, the mean inlet and outlet velocities of water are

$$V_1 = V_2 = V = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{\rho(\pi D^2/4)} = \frac{40\text{ kg/s}}{(1000\text{ kg/m}^3)[\pi(0.1\text{ m})^2/4]} = 5.093\text{ m/s}$$

Noting that $V_1 = V_2$ and $P_2 = P_{atm}$, the Bernoulli equation for a streamline going through the center of the reducing elbow is expressed as

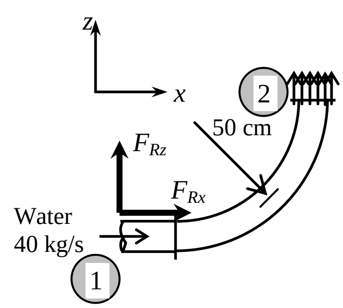
$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \;\rightarrow\; P_1 - P_2 = \rho g(z_2 - z_1) \;\rightarrow\; P_{1,\text{gage}} = \rho g(z_2 - z_1)$$

Substituting,

$$P_{1,\text{gage}} = (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.50\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 4.905\text{ kN/m}^2 \cong \mathbf{4.91 kPa}$$

(*b*) The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. We let the *x*- and *z*- components of the anchoring force of the elbow be F_{Rx} and F_{Rz}, and assume them to be in the positive directions. We also use gage pressures to avoid dealing with the atmospheric pressure which acts on all surfaces. Then the momentum equations along the *x* and *y* axes become

$$F_{Rx} + P_{1,\text{gage}}A_1 = 0 - \beta\dot{m}(+V_1) = -\beta\dot{m}V$$
$$F_{Rz} = \beta\dot{m}(+V_2) = \beta\dot{m}V$$



Solving for F_{Rx} and F_{Rz}, and substituting the given values,

$$F_{Rx} = -\beta\dot{m}V - P_{1,\text{gage}}A_1$$
$$= -1.03(40\text{ kg/s})(5.093\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) - (4905\text{ N/m}^2)[\pi(0.1\text{ m})^2/4]$$
$$= -248.4\text{ N}$$

$$F_{Ry} = \beta\dot{m}V = 1.03(40\text{ kg/s})(5.093\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 209.8\text{ N}$$

and $F_R = \sqrt{F_{Rx}^2 + F_{Ry}^2} = \sqrt{(-248.4)^2 + (209.8)^2} = \mathbf{325 N}$, $\quad \theta = \tan^{-1}\dfrac{F_{Ry}}{F_{Rx}} = \tan^{-1}\dfrac{209.8}{-248.4} = -40.2° = \mathbf{140°}$

Discussion Note that the magnitude of the anchoring force is 325 N, and its line of action makes 140° from the positive *x* direction. Also, a negative value for F_{Rx} indicates the assumed direction is wrong, and should be reversed.

6-23

Solution A 180° elbow forces the flow to make a U-turn and discharges it to the atmosphere at a specified rate. The gage pressure at the inlet of the elbow and the anchoring force needed to hold the elbow in place are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Frictional effects are negligible in the calculation of the pressure drop (so that the Bernoulli equation can be used). **3** The weight of the elbow and the water in it is negligible. **4** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero. **5** The momentum-flux correction factor for each inlet and outlet is given to be $\beta = 1.03$.

Properties We take the density of water to be 1000 kg/m³.

Analysis (*a*) We take the elbow as the control volume, and designate the entrance by 1 and the outlet by 2. We also designate the horizontal coordinate by *x* (with the direction of flow as being the positive direction) and the vertical coordinate by *z*. The continuity equation for this one-inlet one-outlet steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m} = 40$ kg/s. Noting that $\dot{m} = \rho AV$, the mean inlet and outlet velocities of water are

$$V_1 = V_2 = V = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{\rho(\pi D^2/4)} = \frac{40\text{ kg/s}}{(1000\text{ kg/m}^3)[\pi(0.1\text{ m})^2/4]} = 5.093\text{ m/s}$$

Noting that $V_1 = V_2$ and $P_2 = P_{atm}$, the Bernoulli equation for a streamline going through the center of the reducing elbow is expressed as

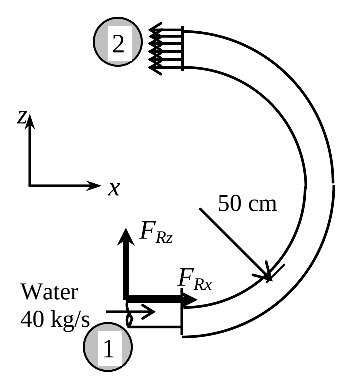
$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \;\rightarrow\; P_1 - P_2 = \rho g(z_2 - z_1) \;\rightarrow\; P_{1,\text{gage}} = \rho g(z_2 - z_1)$$

Substituting,

$$P_{1,\text{gage}} = (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(1.0\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 9.810\text{ kN/m}^2 \cong \mathbf{9.81 kPa}$$

(*b*) The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. We let the *x*- and *z*- components of the anchoring force of the elbow be F_{Rx} and F_{Rz}, and assume them to be in the positive directions. We also use gage pressures to avoid dealing with the atmospheric pressure which acts on all surfaces. Then the momentum equations along the *x* and *z* axes become

$$F_{Rx} + P_{1,\text{gage}}A_1 = \beta\dot{m}(-V_2) - \beta\dot{m}(+V_1) = -2\beta\dot{m}V$$
$$F_{Rz} = 0$$



Solving for F_{Rx} and substituting the given values,

$$\begin{aligned} F_{Rx} &= -2\beta\dot{m}V - P_{1,\text{gage}}A_1 \\ &= -2\times 1.03(40\text{ kg/s})(5.093\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) - (9810\text{ N/m}^2)[\pi(0.1\text{ m})^2/4] \\ &= -497\text{ N} \end{aligned}$$

and $F_R = F_{Rx}$ **= – 497 N** since the *y*-component of the anchoring force is zero. Therefore, the anchoring force has a magnitude of 497 N and it acts in the negative *x* direction.

Discussion Note that a negative value for F_{Rx} indicates the assumed direction is wrong, and should be reversed.

6-24E

Solution A horizontal water jet strikes a vertical stationary plate normally at a specified velocity. For a given anchoring force needed to hold the plate in place, the flow rate of water is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water splatters off the sides of the plate in a plane normal to the jet. **3** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water is the atmospheric pressure which is disregarded since it acts on the entire control surface. **4** The vertical forces and momentum fluxes are not considered since they have no effect on the horizontal reaction force. **5** Jet flow is nearly uniform and thus the effect of the momentum-flux correction factor is negligible, $\beta \cong 1$.

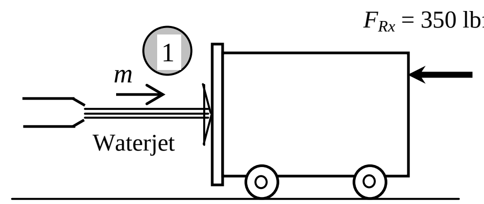
Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis We take the plate as the control volume such that it contains the entire plate and cuts through the water jet and the support bar normally, and the direction of flow as the positive direction of *x* axis. The momentum equation for steady flow in the *x* (flow) direction reduces in this case to

$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V} \quad \to \quad -F_{Rx} = -\dot{m}V_1 \quad \to \quad F_R = \dot{m}V_1$$

We note that the reaction force acts in the opposite direction to flow, and we should not forget the negative sign for forces and velocities in the negative *x*-direction. Solving for $\dot{m}$ and substituting the given values,

$$\dot{m} = \frac{F_{Rx}}{V_1} = \frac{350\ \text{lbf}}{25\ \text{ft/s}}\left(\frac{32.2\ \text{lbm}\cdot\text{ft/s}^2}{1\ \text{lbf}}\right) = 450.8\ \text{lbm/s}$$



Then the volume flow rate becomes

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{450.8\ \text{lbm/s}}{62.4\ \text{lbm/ft}^3} = \mathbf{7.22\ ft^3/s}$$

Therefore, the volume flow rate of water under stated assumptions must be 7.22 ft^3/s.

Discussion In reality, some water will be scattered back, and this will add to the reaction force of water. The flow rate in that case will be less.

6-25

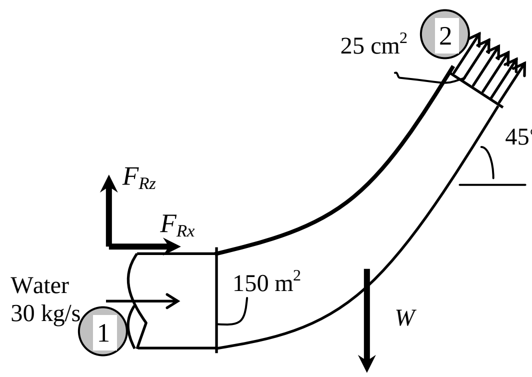
Solution A reducing elbow deflects water upwards and discharges it to the atmosphere at a specified rate. The anchoring force needed to hold the elbow in place is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Frictional effects are negligible in the calculation of the pressure drop (so that the Bernoulli equation can be used). **3** The weight of the elbow and the water in it is considered. **4** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero. **5** The momentum-flux correction factor for each inlet and outlet is given to be $\beta = 1.03$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis The weight of the elbow and the water in it is

$$W = mg = (50\text{ kg})(9.81\text{ m/s}^2) = 490.5\text{ N} = 0.4905\text{ kN}$$



We take the elbow as the control volume, and designate the entrance by 1 and the outlet by 2. We also designate the horizontal coordinate by x (with the direction of flow as being the positive direction) and the vertical coordinate by z. The continuity equation for this one-inlet one-outlet steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m} = 30$ kg/s. Noting that $\dot{m} = \rho AV$, the inlet and outlet velocities of water are

$$V_1 = \frac{\dot{m}}{\rho A_1} = \frac{30\text{ kg/s}}{(1000\ \text{ kg/m}^3)(0.0150\text{ m}^2)} = 2.0\text{ m/s}$$

$$V_2 = \frac{\dot{m}}{\rho A_2} = \frac{30\text{ kg/s}}{(1000\ \text{ kg/m}^3)(0.0025\text{ m}^2)} = 12\text{ m/s}$$

Taking the center of the inlet cross section as the reference level ($z_1 = 0$) and noting that $P_2 = P_{atm}$, the Bernoulli equation for a streamline going through the center of the reducing elbow is expressed as

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \ \rightarrow\ P_1 - P_2 = \rho g\left(\frac{V_2^2 - V_1^2}{2g} + z_2 - z_1\right) \ \rightarrow\ P_{1,\text{ gage}} = \rho g\left(\frac{V_2^2 - V_1^2}{2g} + z_2\right)$$

Substituting,

$$P_{1,\text{ gage}} = (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)\left(\frac{(12\text{ m/s})^2 - (2\text{ m/s})^2}{2(9.81\text{ m/s}^2)} + 0.4\right)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 73.9\text{ kN/m}^2 = 73.9\text{ kPa}$$

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. We let the x- and z- components of the anchoring force of the elbow be F_{Rx} and F_{Rz}, and assume them to be in the positive directions. We also use gage pressures to avoid dealing with the atmospheric pressure which acts on all surfaces. Then the momentum equations along the x and z axes become

$$F_{Rx} + P_{1,\text{gage}}A_1 = \beta\dot{m}V_2\cos\theta - \beta\dot{m}V_1 \quad \text{and} \quad F_{Rz} - W = \beta\dot{m}V_2\sin\theta$$

Solving for F_{Rx} and F_{Rz}, and substituting the given values,

$$F_{Rx} = \beta\dot{m}(V_2\cos\theta - V)_1 - P_{1,\text{ gage}}A_1 = 1.03(30\text{ kg/s})[(12\cos 45°\text{-}2)\text{ m/s}]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) - \left(73.9\text{ kN/m}^2\right)\left(0.0150\text{ m}^2\right)$$

$$= -0.908\text{ kN}$$

$$F_{Rz} = \beta\dot{m}V_2\sin\theta + W = 1.03(30\text{ kg/s})(12\sin 45°\text{ m/s})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) + 0.4905\text{ kN} = 0.753\text{ kN}$$

$$F_R = \sqrt{F_{Rx}^2 + F_{Rz}^2} = \sqrt{(-0.908)^2 + (0.753)^2} = \mathbf{1.18\,kN}, \quad \theta = \tan^{-1}\frac{F_{Rz}}{F_{Rx}} = \tan^{-1}\frac{0.753}{-0.908} = \mathbf{-39.7°}$$

Discussion Note that the magnitude of the anchoring force is 1.18 kN, and its line of action makes –39.7° from +x direction. Negative value for F_{Rx} indicates the assumed direction is wrong.

6-26

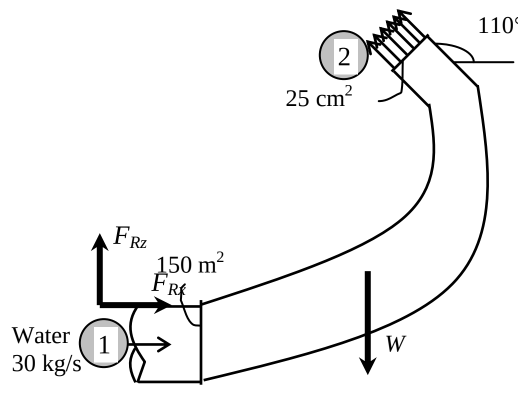
Solution A reducing elbow deflects water upwards and discharges it to the atmosphere at a specified rate. The anchoring force needed to hold the elbow in place is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Frictional effects are negligible in the calculation of the pressure drop (so that the Bernoulli equation can be used). **3** The weight of the elbow and the water in it is considered. **4** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero. **5** The momentum-flux correction factor for each inlet and outlet is given to be $\beta = 1.03$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis The weight of the elbow and the water in it is

$$W = mg = (50\text{ kg})(9.81\text{ m/s}^2) = 490.5\text{ N} = 0.4905\text{ kN}$$



We take the elbow as the control volume, and designate the entrance by 1 and the outlet by 2. We also designate the horizontal coordinate by x (with the direction of flow as being the positive direction) and the vertical coordinate by z. The continuity equation for this one-inlet one-outlet steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m} = 30\text{ kg/s}$. Noting that $\dot{m} = \rho AV$, the inlet and outlet velocities of water are

$$V_1 = \frac{\dot{m}}{\rho A_1} = \frac{30\text{ kg/s}}{(1000\text{ kg/m}^3)(0.0150\text{ m}^2)} = 2.0\text{ m/s}$$

$$V_2 = \frac{\dot{m}}{\rho A_2} = \frac{30\text{ kg/s}}{(1000\text{ kg/m}^3)(0.0025\text{ m}^2)} = 12\text{ m/s}$$

Taking the center of the inlet cross section as the reference level ($z_1 = 0$) and noting that $P_2 = P_{atm}$, the Bernoulli equation for a streamline going through the center of the reducing elbow is expressed as

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \;\rightarrow\; P_1 - P_2 = \rho g\left(\frac{V_2^2 - V_1^2}{2g} + z_2 - z_1\right) \;\rightarrow\; P_{1,\text{ gage}} = \rho g\left(\frac{V_2^2 - V_1^2}{2g} + z_2\right)$$

or, $$P_{1,\text{ gage}} = (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)\left(\frac{(12\text{ m/s})^2 - (2\text{ m/s})^2}{2(9.81\text{ m/s}^2)} + 0.4\right)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 73.9\text{ kN/m}^2 = 73.9\text{ kPa}$$

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. We let the x- and y- components of the anchoring force of the elbow be F_{Rx} and F_{Rz}, and assume them to be in the positive directions. We also use gage pressures to avoid dealing with the atmospheric pressure which acts on all surfaces. Then the momentum equations along the x and z axes become

$$F_{Rx} + P_{1,\text{gage}}A_1 = \beta\dot{m}V_2\cos\theta - \beta\dot{m}V_1 \quad\text{and}\quad F_{Ry} - W = \beta\dot{m}V_2\sin\theta$$

Solving for F_{Rx} and F_{Rz}, and substituting the given values,

$$F_{Rx} = \beta\dot{m}(V_2\cos\theta - V_1) - P_{1,\text{ gage}}A_1$$

$$= 1.03(30\text{ kg/s})[(12\cos 110° - 2)\text{ m/s}]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) - (73.9\text{ kN/m}^2)(0.0150\text{ m}^2) = -1.297\text{ kN}$$

$$F_{Rz} = \beta\dot{m}V_2\sin\theta + W = 1.03(30\text{ kg/s})(12\sin 110°\text{ m/s})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) + 0.4905\text{ kN} = 0.8389\text{ kN}$$

$$F_R = \sqrt{F_{Rx}^2 + F_{Rz}^2} = \sqrt{(-1.297)^2 + 0.8389^2} = \mathbf{1.54\ kN}$$

and

$$\theta = \tan^{-1}\frac{F_{Rz}}{F_{Rx}} = \tan^{-1}\frac{0.8389}{-1.297} = \mathbf{-32.9°}$$

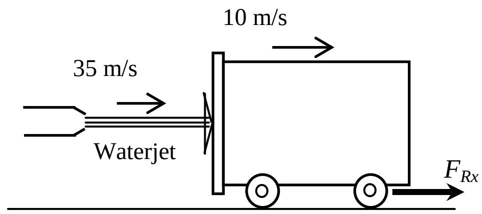
Discussion Note that the magnitude of the anchoring force is 1.54 kN, and its line of action makes –32.9° from +x direction. Negative value for F_{Rx} indicates assumed direction is wrong, and should be reversed.

6-27

Solution Water accelerated by a nozzle strikes the back surface of a cart moving horizontally at a constant velocity. The braking force and the power wasted by the brakes are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water splatters off the sides of the plate in all directions in the plane of the back surface. **3** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water is the atmospheric pressure which is disregarded since it acts on all surfaces. **4** Fiction during motion is negligible. **5** There is no acceleration of the cart. **7** The motions of the water jet and the cart are horizontal. **6** Jet flow is nearly uniform and thus the effect of the momentum-flux correction factor is negligible, $\beta \cong 1$.

Analysis We take the cart as the control volume, and the direction of flow as the positive direction of x axis. The relative velocity between the cart and the jet is



$$V_r = V_{\text{jet}} - V_{\text{cart}} = 35 - 10 = 25\ \text{m/s}$$

Therefore, we can view the cart as being stationary and the jet moving with a velocity of 25 m/s. Noting that water leaves the nozzle at 20 m/s and the corresponding mass flow rate relative to nozzle exit is 30 kg/s, the mass flow rate of water striking the cart corresponding to a water jet velocity of 25 m/s relative to the cart is

$$\dot{m}_r = \frac{V_r}{V_{\text{jet}}}\dot{m}_{\text{jet}} = \frac{25\ \text{m/s}}{35\ \text{m/s}}(30\ \text{kg/s}) = 21.43\ \text{kg/s}$$

The momentum equation for steady flow in the x (flow) direction reduces in this case to

$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V} \quad \rightarrow \quad F_{Rx} = -\dot{m}_i V_i \quad \rightarrow \quad F_{\text{brake}} = -\dot{m}_r V_r$$

We note that the brake force acts in the opposite direction to flow, and we should not forget the negative sign for forces and velocities in the negative x-direction. Substituting the given values,

$$F_{\text{brake}} = -\dot{m}_r V_r = -(21.43\ \text{kg/s})(+25\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = -535.8\ \text{N} \cong \mathbf{-536\ N}$$

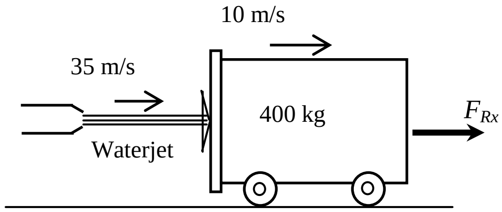
The negative sign indicates that the braking force acts in the opposite direction to motion, as expected. Noting that work is force times distance and the distance traveled by the cart per unit time is the cart velocity, the power wasted by the brakes is

$$\dot{W} = F_{\text{brake}} V_{\text{cart}} = (535.8\ \text{N})(10\ \text{m/s})\left(\frac{1\ \text{W}}{1\ \text{N}\cdot\text{m/s}}\right) = 5358\ \text{W} \cong \mathbf{5.36\ kW}$$

Discussion Note that the power wasted is equivalent to the maximum power that can be generated as the cart velocity is maintained constant.

6-28

Solution Water accelerated by a nozzle strikes the back surface of a cart moving horizontally. The acceleration of the cart if the brakes fail is to be determined.



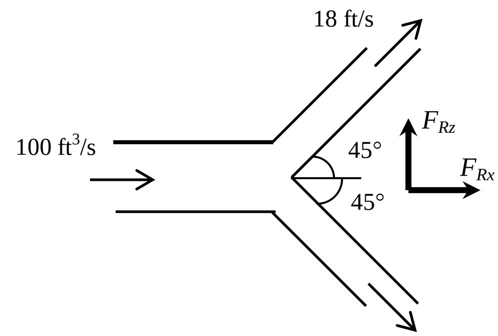
Analysis The braking force was determined in previous problem to be 535.8 N. When the brakes fail, this force will propel the cart forward, and the acceleration will be

$$a = \frac{F}{m_{\text{cart}}} = \frac{535.8\text{ N}}{400\text{ kg}}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{1.34\ m/s^2}$$

Discussion This is the acceleration at the moment the brakes fail. The acceleration will decrease as the relative velocity between the water jet and the cart (and thus the force) decreases.

6-29E

Solution A water jet hits a stationary splitter, such that half of the flow is diverted upward at 45°, and the other half is directed down. The force required to hold the splitter in place is to be determined.



Assumptions **1** The flow is steady and incompressible. **2** The water jet is exposed to the atmosphere, and thus the pressure of the water jet before and after the split is the atmospheric pressure which is disregarded since it acts on all surfaces. **3** The gravitational effects are disregarded. **4** The flow is nearly uniform at all cross sections, and thus the effect of the momentum-flux correction factor is negligible, $\beta \cong 1$.

Properties We take the density of water to be 62.4 lbm/ft³.

Analysis The mass flow rate of water jet is

$$\dot{m} = \rho \dot{V} = (62.4\ \text{lbm/ft}^3)(100\ \text{ft}^3/\text{s}) = 6240\ \text{lbm/s}$$

We take the splitting section of water jet, including the splitter as the control volume, and designate the entrance by 1 and the outlet of either arm by 2 (both arms have the same velocity and mass flow rate). We also designate the horizontal coordinate by x with the direction of flow as being the positive direction and the vertical coordinate by z.

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m} \vec{V} - \sum_{\text{in}} \beta \dot{m} \vec{V}$. We let the x- and y- components of the anchoring force of the splitter be F_{Rx} and F_{Rz}, and assume them to be in the positive directions. Noting that $V_2 = V_1 = V$ and $\dot{m}_2 = \frac{1}{2}\dot{m}$, the momentum equations along the x and z axes become

$$F_{Rx} = 2(\tfrac{1}{2}\dot{m})V_2 \cos\theta - \dot{m}V_1 = \dot{m}V(\cos\theta - 1)$$
$$F_{Rz} = \tfrac{1}{2}\dot{m}(+V_2 \sin\theta) + \tfrac{1}{2}\dot{m}(-V_2 \sin\theta) - 0 = 0$$

Substituting the given values,

$$F_{Rx} = (6240\ \text{lbm/s})(18\ \text{ft/s})(\cos 45° - 1)\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = -1021.67\ \text{lbf} \cong \mathbf{-1020\ lbf}$$

$$F_{Rz} = \mathbf{0}$$

The negative value for F_{Rx} indicates the assumed direction is wrong, and should be reversed. Therefore, a force of 1020 lbf must be applied to the splitter in the opposite direction to flow to hold it in place. No holding force is necessary in the vertical direction. This can also be concluded from the symmetry.

Discussion In reality, the gravitational effects will cause the upper stream to slow down and the lower stream to speed up after the split. But for short distances, these effects are indeed negligible.

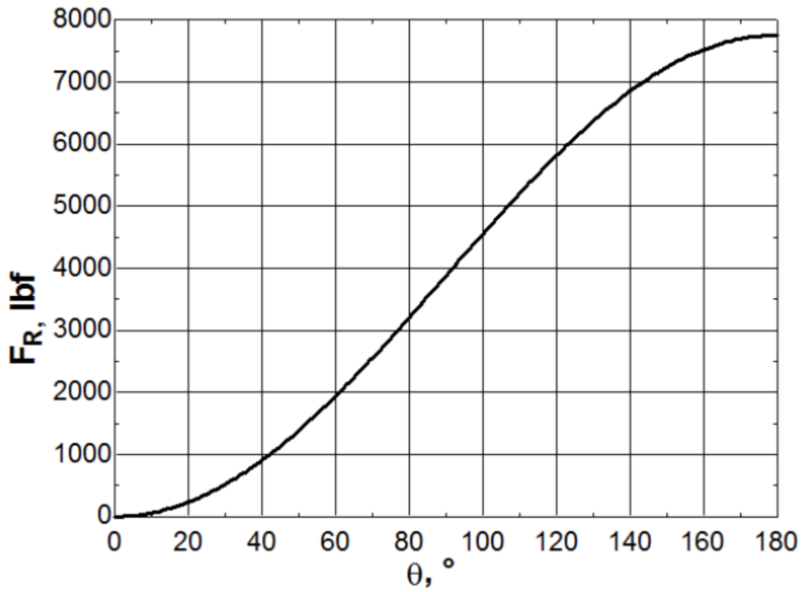
6-30E

Solution The previous problem is reconsidered. The effect of splitter angle on the force exerted on the splitter as the half splitter angle varies from 0 to 180° in increments of 10° is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=32.2 "ft/s2"
rho=62.4 "lbm/ft3"
V_dot=100 "ft3/s"
V=20 "ft/s"
m_dot=rho*V_dot
F_R=-m_dot*V*(cos(theta)-1)/g "lbf"
```

θ, °	$\dot{m}$, lbm/s	F_R, lbf
0	6240	0
10	6240	59
20	6240	234
30	6240	519
40	6240	907
50	6240	1384
60	6240	1938
70	6240	2550
80	6240	3203
90	6240	3876
100	6240	4549
110	6240	5201
120	6240	5814
130	6240	6367
140	6240	6845
150	6240	7232
160	6240	7518
170	6240	7693
180	6240	7752



Discussion The force rises from zero at $\theta = 0^{\circ}$ to a maximum at $\theta = 180^{\circ}$, as expected, but the relationship is not linear.

6-31

Solution A horizontal water jet impinges normally upon a vertical plate which is held on a frictionless track and is initially stationary. The initial acceleration of the plate, the time it takes to reach a certain velocity, and the velocity at a given time are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water always splatters in the plane of the retreating plate. **3** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water is the atmospheric pressure which is disregarded since it acts on all surfaces. **4** The track is nearly frictionless, and thus fiction during motion is negligible. **5** The motions of the water jet and the cart are horizontal. **6** The velocity of the jet relative to the plate remains constant, $V_r = V_{jet} = V$. **7** Jet flow is nearly uniform and thus the effect of the momentum-flux correction factor is negligible, $\beta \cong 1$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis (*a*) We take the vertical plate on the frictionless track as the control volume, and the direction of flow as the positive direction of *x* axis. The mass flow rate of water in the jet is

$$\dot{m} = \rho VA = (1000\ \text{kg/m}^3)(18\ \text{m/s})[\pi(0.05\ \text{m})^2/4] = 35.34\ \text{kg/s}$$

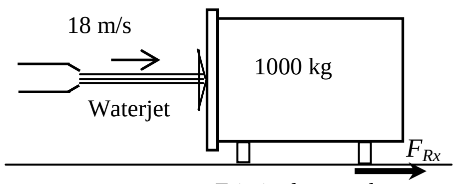
The momentum equation for steady flow in the *x* (flow) direction reduces in this case to

$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V} \quad \rightarrow \quad F_{Rx} = -\dot{m}_i V_i \quad \rightarrow \quad F_{Rx} = -\dot{m}V$$

where F_{Rx} is the reaction force required to hold the plate in place. When the plate is released, an equal and opposite impulse force acts on the plate, which is determined to

$$F_{\text{plate}} = -F_{Rx} = \dot{m}V = (35.34\ \text{kg/s})(18\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = 636\ \text{N}$$

Then the initial acceleration of the plate becomes



$$a = \frac{F_{\text{plate}}}{m_{\text{plate}}} = \frac{636\ \text{N}}{1000\ \text{kg}}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = \mathbf{0.636\ m/s^2}$$

This acceleration will remain constant during motion since the force acting on the plate remains constant.

(*b*) Noting that $a = dV/dt = \Delta V/\Delta t$ since the acceleration *a* is constant, the time it takes for the plate to reach a velocity of 9 m/s is

$$\Delta t = \frac{\Delta V_{\text{plate}}}{a} = \frac{(9-0)\ \text{m/s}}{0.636\ \text{m/s}^2} = \mathbf{14.2\ s}$$

(*c*) Noting that $a = dV/dt$ and thus $dV = adt$ and that the acceleration *a* is constant, the plate velocity in 20 s becomes

$$V_{\text{plate}} = V_{0,\text{plate}} + a\Delta t = 0 + (0.636\ \text{m/s}^2)(20\ \text{s}) = \mathbf{12.7\ m/s}$$

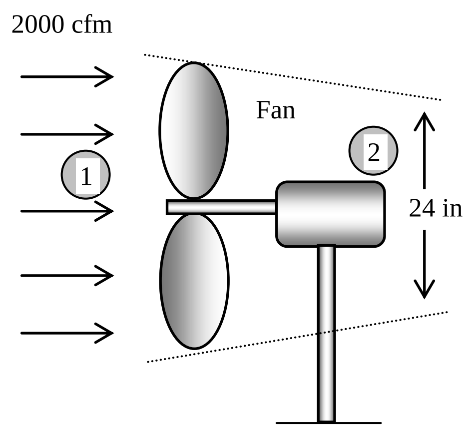
Discussion The assumption that the relative velocity between the water jet and the plate remains constant is valid only for the initial moments of motion when the plate velocity is low unless the water jet is moving with the plate at the same velocity as the plate.

6-32E

Solution A fan moves air at sea level at a specified rate. The force required to hold the fan and the minimum power input required for the fan are to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** Standard atmospheric conditions exist so that the pressure at sea level is 1 atm. **3** Air leaves the fan at a uniform velocity at atmospheric pressure. **4** Air approaches the fan through a large area at atmospheric pressure with negligible velocity. **5** The frictional effects are negligible, and thus the entire mechanical power input is converted to kinetic energy of air (no conversion to thermal energy through frictional effects). **6** Wind flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties The gas constant of air is R = 0.3704 psi·ft^3/lbm·R. The standard atmospheric pressure at sea level is 1 atm = 14.7 psi.



Analysis (*a*) We take the control volume to be a horizontal hyperbolic cylinder bounded by streamlines on the sides with air entering through the large cross-section (section 1) and the fan located at the narrow cross-section at the end (section 2), and let its centerline be the x axis. The density, mass flow rate, and discharge velocity of air are

$$\rho = \frac{P}{RT} = \frac{14.7\text{ psi}}{(0.3704\text{ psi}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530\text{ R})} = 0.0749\text{ lbm/ft}^3$$

$$\dot{m} = \rho\dot{V} = (0.0749\text{ lbm/ft}^3)(2000\text{ ft}^3/\text{min}) = 149.8\text{ lbm/min} = 2.50\text{ lbm/s}$$

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{2000\text{ ft}^3/\text{min}}{\pi(2\text{ ft})^2/4} = 636.6\text{ ft/min} = 10.6\text{ ft/s}$$

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. Letting the reaction force to hold the fan be F_{Rx} and assuming it to be in the positive x (i.e., the flow) direction, the momentum equation along the x axis becomes

$$F_{Rx} = \dot{m}(V_2) - 0 = \dot{m}V = (2.50\text{ lbm/s})(10.6\text{ ft/s})\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{0.820\ lbf}$$

Therefore, a force of 0.82 lbf must be applied (through friction at the base, for example) to prevent the fan from moving in the horizontal direction under the influence of this force.

(*b*) Noting that $P_1 = P_2 = P_{\text{atm}}$ and $V_1 \cong 0$, the energy equation for the selected control volume reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump,u}} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech,loss}} \quad \rightarrow \quad \dot{W}_{\text{fan, u}} = \dot{m}\frac{V_2^2}{2}$$

Substituting,

$$\dot{W}_{\text{fan,u}} = \dot{m}\frac{V_2^2}{2} = (2.50\text{ lbm/s})\frac{(10.6\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\text{ W}}{0.73756\text{ lbf}\cdot\text{ft/s}}\right) = \mathbf{5.91 W}$$

Therefore, a useful mechanical power of 5.91 W must be supplied to air. This is the *minimum* required power input required for the fan.

Discussion The actual power input to the fan will be larger than 5.91 W because of the fan inefficiency in converting mechanical power to kinetic energy.

6-33E

Solution A horizontal water jet strikes a bent plate, which deflects the water by 135° from its original direction. The force required to hold the plate against the water stream is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water is the atmospheric pressure, which is disregarded since it acts on all surfaces. **3** Frictional and gravitational effects are negligible. **4** There is no splattering of water or the deformation of the jet, and the reversed jet leaves at the same velocity and flow rate. **5** Jet flow is nearly uniform and thus the momentum-flux correction factor is nearly unity, $\beta \cong 1$.

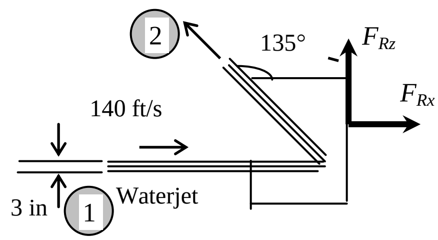
Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis We take the plate together with the curved water jet as the control volume, and designate the jet inlet by 1 and the outlet by 2. We also designate the horizontal coordinate by x (with the direction of incoming flow as being the positive direction), and the vertical coordinate by z. The equation of conservation of mass for this one-inlet one-outlet steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m}$ where

$$\dot{m} = \rho VA = \rho V[\pi D^2/4] = (62.4\ \text{lbm/ft}^3)(140\ \text{ft/s})[\pi(3/12\ \text{ft})^2/4] = 428.8\ \text{lbm/s}$$

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V}$. We let the x- and z- components of the anchoring force of the plate be F_{Rx} and F_{Rz}, and assume them to be in the positive directions. Then the momentum equations along the x and y axes become

$$F_{Rx} = \dot{m}(-V_2)\cos 45° - \dot{m}(+V_1) = -\dot{m}V(1+\cos 45°)$$
$$F_{Rz} = \dot{m}(+V_2)\sin 45° = \dot{m}V\sin 45°$$



Substituting the given values,

$$F_{Rx} = -(428.8\ \text{lbm/s})(140\ \text{ft/s})(1+\cos 45°)\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)$$
$$= -3182.64\ \text{lbf} \cong -3180\ \text{lbf}$$

$$F_{Rz} = (428.8\ \text{lbm/s})(140\ \text{ft/s})\sin 45°\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = 1318.29\ \text{lbf} \cong 1320\ \text{lbf}$$

and

$$F_R = \sqrt{F_{Rx}^2 + F_{Rz}^2} = \sqrt{(-3182.64)^2 + 1318.29^2} = 3444.86\ \text{lbf} \cong \mathbf{3440\ lbf}, \quad \theta = \tan^{-1}\frac{F_{Ry}}{F_{Rx}} = \tan^{-1}\frac{1318.29}{-3182.64} = 157.50° \cong \mathbf{158}°$$

Discussion Note that the magnitude of the anchoring force is 3440 lbf, and its line of action is 158° from the positive x direction. Also, a negative value for F_{Rx} indicates the assumed direction is wrong; the actual anchoring force is to the left. This makes sense when we think about it; with the water jet striking the plate from left to right, one would have to push to the left in order to hold the plat in place.

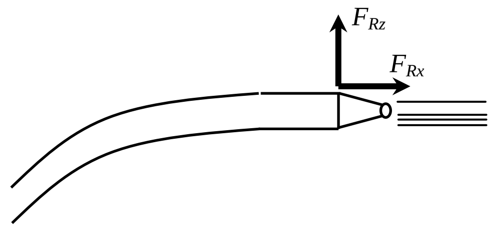
6-34

Solution Firemen are holding a nozzle at the end of a hose while trying to extinguish a fire. The average water outlet velocity and the resistance force required of the firemen to hold the nozzle are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water jet is exposed to the atmosphere, and thus the pressure of the water jet is the atmospheric pressure, which is disregarded since it acts on all surfaces. **3** Gravitational effects and vertical forces are disregarded since the horizontal resistance force is to be determined. **5** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis (*a*) We take the nozzle and the horizontal portion of the hose as the system such that water enters the control volume vertically and outlets horizontally (this way the pressure force and the momentum flux at the inlet are in the vertical direction, with no contribution to the force balance in the horizontal direction), and designate the entrance by 1 and the outlet by 2. We also designate the horizontal coordinate by x (with the direction of flow as being the positive direction). The average outlet velocity and the mass flow rate of water are determined from



$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\pi D^2/4} = \frac{12\ \text{m}^3/\text{min}}{\pi(0.08\ \text{m})^2/4} = 2387\ \text{m/min} = 39.79\ \text{m/s} \cong \mathbf{39.8\ m/s}$$

$$\dot{m} = \rho\dot{V} = (1000\ \text{kg/m}^3)(12\ \text{m}^3/\text{min}) = 12{,}000\ \text{kg/min} = 200\ \text{kg/s}$$

(*b*) The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. We let horizontal force applied by the firemen to the nozzle to hold it be F_{Rx}, and assume it to be in the positive x direction. Then the momentum equation along the x direction gives

$$F_{Rx} = \dot{m}V_e - 0 = \dot{m}V = (200\ \text{kg/s})(39.79\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{-7958\ N}$$

Therefore, the firemen must be able to resist a force of 7958 N to hold the nozzle in place.

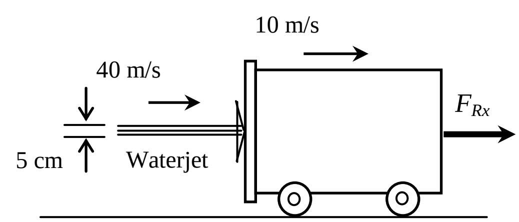
Discussion The force of 7958 N is equivalent to the weight of about 810 kg. That is, holding the nozzle requires the strength of holding a weight of 810 kg, which cannot be done by a single person. This demonstrates why several firemen are used to hold a hose with a high flow rate.

6-35

Solution A horizontal jet of water with a given velocity strikes a flat plate that is moving in the same direction at a specified velocity. The force that the water stream exerts against the plate is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water splatters in all directions in the plane of the plate. **3** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water is the atmospheric pressure, which is disregarded since it acts on all surfaces. **4** The vertical forces and momentum fluxes are not considered since they have no effect on the horizontal force exerted on the plate. **5** The velocity of the plate, and the velocity of the water jet relative to the plate, are constant. **6** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties We take the density of water to be 1000 kg/m^3.



Analysis We take the plate as the control volume, and the flow direction as the positive direction of x axis. The relative velocity between the plate and the jet is

$$V_r = V_{\text{jet}} - V_{\text{plate}} = 40 - 10 = 30 \text{ m/s}$$

Therefore, we can view the plate as being stationary and the jet to be moving with a velocity of 30 m/s. The mass flow rate of water relative to the plate [i.e., the flow rate at which water strikes the plate] is

$$\dot{m}_r = \rho V_r A = \rho V_r \frac{\pi D^2}{4} = (1000 \text{ kg/m}^3)(30 \text{ m/s})\frac{\pi (0.05 \text{ m})^2}{4} = 58.90 \text{ kg/s}$$

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m} \vec{V} - \sum_{\text{in}} \beta \dot{m} \vec{V}$. We let the horizontal reaction force applied to the plate in the negative x direction to counteract the impulse of the water jet be F_{Rx}. Then the momentum equation along the x direction gives

$$-F_{Rx} = 0 - \dot{m} V_i \quad \rightarrow \quad F_{Rx} = \dot{m}_r V_r = (58.90 \text{ kg/s})(30 \text{ m/s})\left(\frac{1 \text{ N}}{1 \text{kg} \cdot \text{m/s}^2}\right) = \mathbf{1767 N}$$

Therefore, the water jet applies a force of 1767 N on the plate in the direction of motion, and an equal and opposite force must be applied on the plate if its velocity is to remain constant.

Discussion Note that we used the relative velocity in the determination of the mass flow rate of water in the momentum analysis since water will enter the control volume at this rate. (In the limiting case of the plate and the water jet moving at the same velocity, the mass flow rate of water relative to the plate will be zero since no water will be able to strike the plate).

6-36

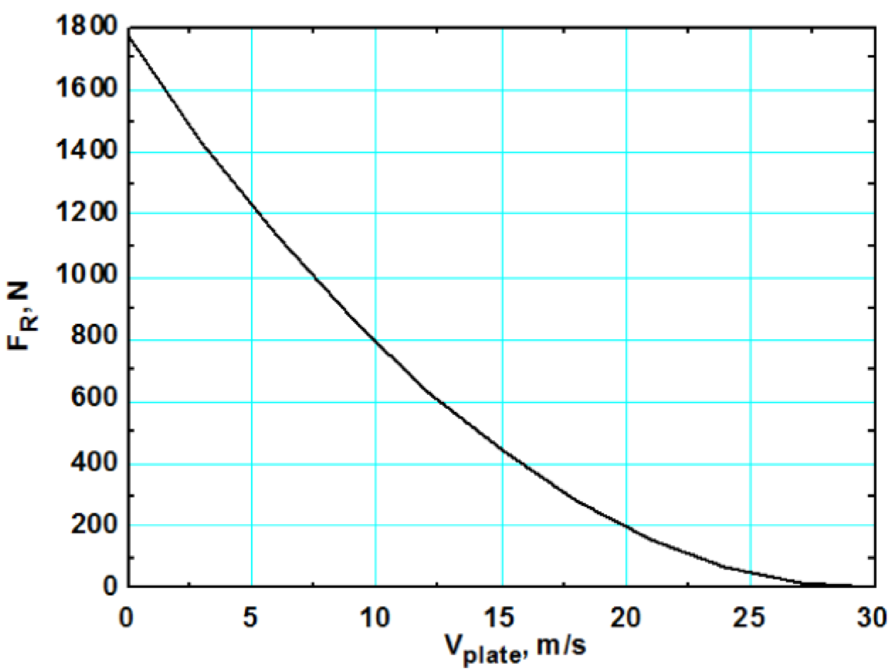
Solution The previous problem is reconsidered. The effect of the plate velocity on the force exerted on the plate as the plate velocity varies from 0 to 30 m/s in increments of 3 m/s is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
rho=1000 "kg/m3"
D=0.05 "m"
V_jet=30 "m/s"

Ac=pi*D^2/4
V_r=V_jet-V_plate
m_dot=rho*Ac*V_r
F_R=m_dot*V_r "N"
```

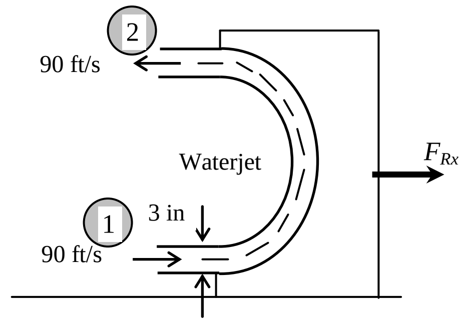
V_{plate}, m/s	V_r, m/s	F_R, N
0	30	1767
3	27	1431
6	24	1131
9	21	866
12	18	636
15	15	442
18	12	283
21	9	159
24	6	70.7
27	3	17.7
30	0	0



Discussion When the plate velocity reaches 30 m/s, there is no relative motion between the jet and the plate; hence, there can be no force acting.

6-37E

Solution A horizontal water jet strikes a curved plate, which deflects the water back to its original direction. The force required to hold the plate against the water stream is to be determined.



Assumptions **1** The flow is steady and incompressible. **2** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water is the atmospheric pressure, which is disregarded since it acts on all surfaces. **3** Friction between the plate and the surface it is on is negligible (or the friction force can be included in the required force to hold the plate). **4** There is no splashing of water or the deformation of the jet, and the reversed jet leaves horizontally at the same velocity and flow rate. **5** Jet flow is nearly uniform and thus the momentum-flux correction factor is nearly unity, $\beta \cong 1$.

Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis We take the plate together with the curved water jet as the control volume, and designate the jet inlet by 1 and the outlet by 2. We also designate the horizontal coordinate by x (with the direction of incoming flow as being the positive direction). The continuity equation for this one-inlet one-outlet steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m}$ where

$$\dot{m} = \rho VA = \rho V[\pi D^2/4] = (62.4\ \text{lbm/ft}^3)(90\ \text{ft/s})[\pi(3/12\ \text{ft})^2/4] = 275.7\ \text{lbm/s}$$

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. Letting the reaction force to hold the plate be F_{Rx} and assuming it to be in the positive direction, the momentum equation along the x axis becomes

$$F_{Rx} = \dot{m}(-V_2) - \dot{m}(+V_1) = -2\dot{m}V$$

Substituting,

$$F_{Rx} = -2(275.7\ \text{lbm/s})(90\ \text{ft/s})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = -1541\ \text{lbf} \cong \mathbf{-1540\ lbf}$$

Therefore, a force of 1540 lbf must be applied on the plate in the negative x direction to hold it in place.

Discussion Note that a negative value for F_{Rx} indicates the assumed direction is wrong (as expected), and should be reversed. Also, there is no need for an analysis in the vertical direction since the fluid streams are horizontal.

6-38

Solution A helicopter hovers at sea level while being loaded. The volumetric air flow rate and the required power input during unloaded hover, and the rpm and the required power input during loaded hover are to be determined.

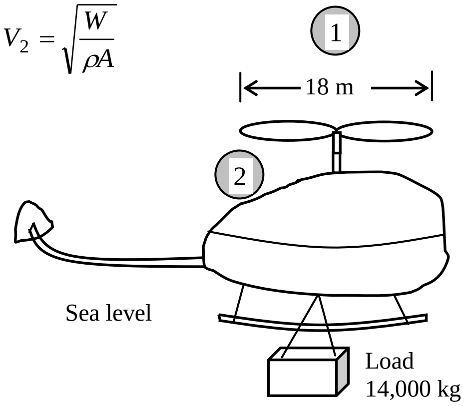
Assumptions **1** The flow of air is steady and incompressible. **2** Air leaves the blades at a uniform velocity at atmospheric pressure. **3** Air approaches the blades from the top through a large area at atmospheric pressure with negligible velocity. **4** The frictional effects are negligible, and thus the entire mechanical power input is converted to kinetic energy of air (no conversion to thermal energy through frictional effects). **5** The change in air pressure with elevation is negligible because of the low density of air. **6** There is no acceleration of the helicopter, and thus the lift generated is equal to the total weight. **7** Air flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties The density of air is given to be 1.18 kg/m^3.

Analysis (*a*) We take the control volume to be a vertical hyperbolic cylinder bounded by streamlines on the sides with air entering through the large cross-section (section 1) at the top and the fan located at the narrow cross-section at the bottom (section 2), and let its centerline be the *z* axis with upwards being the positive direction.

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V}$. Noting that the only force acting on the control volume is the total weight *W* and it acts in the negative *z* direction, the momentum equation along the *z* axis gives

$$-W = \dot{m}(-V_2) - 0 \quad \rightarrow \quad W = \dot{m}V_2 = (\rho A V_2)V_2 = \rho A V_2^2 \quad \rightarrow \quad V_2 = \sqrt{\frac{W}{\rho A}}$$



where *A* is the blade span area,

$$A = \pi D^2/4 = \pi(18\text{ m})^2/4 = 254.5\text{ m}^2$$

Then the discharge velocity, volume flow rate, and the mass flow rate of air in the unloaded mode become

$$V_{2,\text{unloaded}} = \sqrt{\frac{m_{\text{unloaded}}g}{\rho A}} = \sqrt{\frac{(12{,}000\text{ kg})(9.81\text{ m/s}^2)}{(1.18\text{ kg/m}^3)(254.5\text{ m}^2)}} = 19.80\text{ m/s}$$

$$\dot{V}_{\text{unloaded}} = AV_{2,\text{unloaded}} = (254.5\text{ m}^2)(19.80\text{ m/s}) = \mathbf{5039\ m^3/s}$$

$$\dot{m}_{\text{unloaded}} = \rho\dot{V}_{\text{unloaded}} = (1.18\text{ kg/m}^3)(5039\text{ m}^3/\text{s}) = 5946\text{ kg/s}$$

Noting that $P_1 = P_2 = P_{\text{atm}}$, $V_1 \cong 0$, the elevation effects are negligible, and the frictional effects are disregarded, the energy equation for the selected control volume reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump,u}} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech,loss}} \quad \rightarrow \quad \dot{W}_{\text{fan, u}} = \dot{m}\frac{V_2^2}{2}$$

Substituting,

$$\dot{W}_{\text{unloadedfan,u}} = \left(\dot{m}\frac{V_2^2}{2}\right)_{\text{unloaded}} = (5946\text{ kg/s})\frac{(19.80\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right) = 1165\text{ kW} \cong \mathbf{1170\ kW}$$

(*b*) We now repeat the calculations for the *loaded* helicopter, whose mass is 12,000+14,000 = 26,000 kg:

$$V_{2,\text{loaded}} = \sqrt{\frac{m_{\text{loaded}}g}{\rho A}} = \sqrt{\frac{(26{,}000\text{ kg})(9.81\text{ m/s}^2)}{(1.18\text{ kg/m}^3)(254.5\text{ m}^2)}} = 29.14\text{ m/s}$$

$$\dot{m}_{\text{loaded}} = \rho\dot{V}_{\text{loaded}} = \rho A V_{2,\text{loaded}} = (1.18\text{ kg/m}^3)(254.5\text{ m}^2)(29.14\text{ m/s}) = 8752\text{ kg/s}$$

$$\dot{W}_{\text{loadedfan,u}} = \left(\dot{m}\frac{V_2^2}{2}\right)_{\text{loaded}} = (8752\text{ kg/s})\frac{(29.14\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right) = 3716\text{ kW} \cong \mathbf{3720\ kW}$$

Noting that the average flow velocity is proportional to the overhead blade rotational velocity, the rpm of the loaded helicopter blades becomes

$$V_2 = k\dot{n} \quad \rightarrow \quad \frac{V_{2,\text{loaded}}}{V_{2,\text{unloaded}}} = \frac{\dot{n}_{\text{loaded}}}{\dot{n}_{\text{unloaded}}} \quad \rightarrow \quad \dot{n}_{\text{loaded}} = \frac{V_{2,\text{loaded}}}{V_{2,\text{unloaded}}}\dot{n}_{\text{unloaded}} = \frac{29.14}{19.80}(550\text{ rpm}) = \mathbf{809\ rpm}$$

Discussion The actual power input to the helicopter blades will be considerably larger than the calculated power input because of the fan inefficiency in converting mechanical power to kinetic energy.

6-39

Solution A helicopter hovers on top of a high mountain where the air density considerably lower than that at sea level. The blade rotational velocity to hover at the higher altitude and the percent increase in the required power input to hover at high altitude relative to that at sea level are to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** The air leaves the blades at a uniform velocity at atmospheric pressure. **3** Air approaches the blades from the top through a large area at atmospheric pressure with negligible velocity. **4** The frictional effects are negligible, and thus the entire mechanical power input is converted to kinetic energy of air. **5** The change in air pressure with elevation while hovering at a given location is negligible because of the low density of air. **6** There is no acceleration of the helicopter, and thus the lift generated is equal to the total weight. **7** Air flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties The density of air is given to be 1.18 kg/m^3 at sea level, and 0.928 kg/m^3 on top of the mountain.

Analysis (*a*) We take the control volume to be a vertical hyperbolic cylinder bounded by streamlines on the sides with air entering through the large cross-section (section 1) at the top and the fan located at the narrow cross-section at the bottom (section 2), and let its centerline be the *z* axis with upwards being the positive direction.

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V}$. Noting that the only force acting on the control volume is the total weight *W* and it acts in the negative *z* direction, the momentum equation along the *z* axis gives

$$-W = \dot{m}(-V_2) - 0 \quad \rightarrow \quad W = \dot{m}V_2 = (\rho A V_2)V_2 = \rho A V_2^2 \quad \rightarrow \quad V_2 = \sqrt{\frac{W}{\rho A}}$$

where *A* is the blade span area. Then for a given weight *W*, the ratio of discharge velocities becomes

$$\frac{V_{2,\text{mountain}}}{V_{2,\text{sea}}} = \frac{\sqrt{W/\rho_{\text{mountain}}A}}{\sqrt{W/\rho_{\text{sea}}A}} = \sqrt{\frac{\rho_{\text{sea}}}{\rho_{\text{mountain}}}} = \sqrt{\frac{1.18 \text{ kg/m}^3}{0.928 \text{ kg/m}^3}} = 1.1276$$

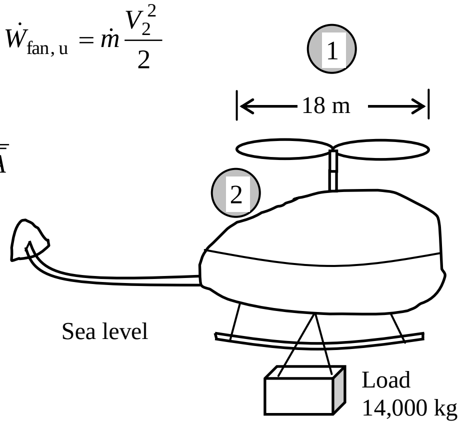
Noting that the average flow velocity is proportional to the overhead blade rotational velocity, the rpm of the helicopter blades on top of the mountain becomes

$$\dot{n} = kV_2 \rightarrow \frac{\dot{n}_{\text{mountain}}}{\dot{n}_{\text{sea}}} = \frac{V_{2,\text{mountain}}}{V_{2,\text{sea}}} \rightarrow \dot{n}_{\text{mountain}} = \frac{V_{2,\text{mountain}}}{V_{2,\text{sea}}}\dot{n}_{\text{sea}} = (1.1276)(550 \text{ rpm}) = 620.2 \text{ rpm} \cong \mathbf{620\,rpm}$$

Noting that $P_1 = P_2 = P_{\text{atm}}$, $V_1 \cong 0$, the elevation effect are negligible, and the frictional effects are disregarded, the energy equation for the selected control volume reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump,u}} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech,loss}} \rightarrow \dot{W}_{\text{fan, u}} = \dot{m}\frac{V_2^2}{2}$$

or $$\dot{W}_{\text{fan,u}} = \dot{m}\frac{V_2^2}{2} = \rho A V_2 \frac{V_2^2}{2} = \rho A \frac{V_2^3}{2} = \tfrac{1}{2}\rho A\left(\sqrt{\frac{W}{\rho A}}\right)^3 = \tfrac{1}{2}\rho A\left(\frac{W}{\rho A}\right)^{1.5} = \frac{W^{1.5}}{2\sqrt{\rho A}}$$



Then the ratio of the required power input on top of the mountain to that at sea level becomes

$$\frac{\dot{W}_{\text{mountain fan,u}}}{\dot{W}_{\text{sea fan,u}}} = \frac{0.5W^{1.5}/\sqrt{\rho_{\text{mountain}}A}}{0.5W^{1.5}/\sqrt{\rho_{\text{sea}}A}} = \sqrt{\frac{\rho_{\text{sea}}}{\rho_{\text{mountain}}}} = \sqrt{\frac{1.18 \text{ kg/m}^3}{0.928 \text{ kg/m}^3}} = 1.128$$

Therefore, the required power input will increase by approximately **12.8%** on top of the mountain relative to the sea level.

Discussion Note that both the rpm and the required power input to the helicopter are inversely proportional to the square root of air density. Therefore, more power is required at higher elevations for the helicopter to operate because air is less dense, and more air must be forced by the blades into the downdraft.

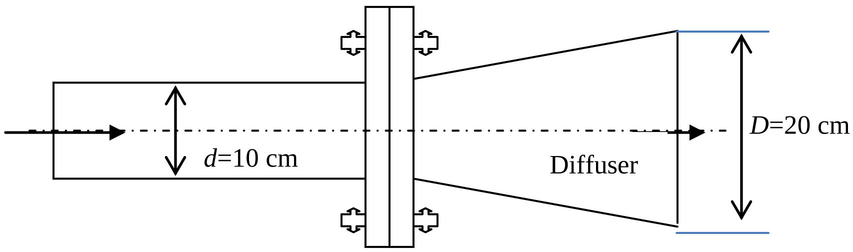
6-40

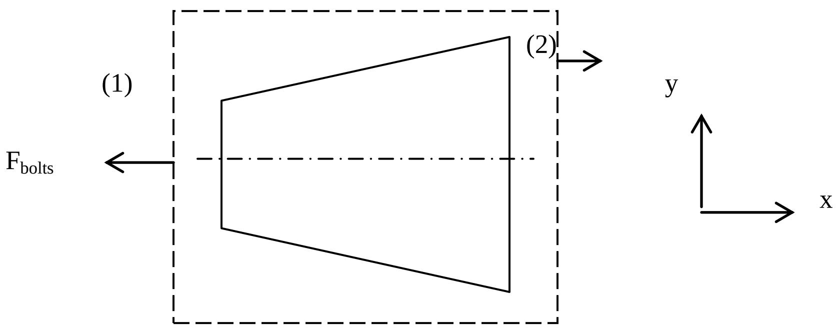
Solution Water flowing in a pipe is slowed down by a diffuser. The force exerted on the bolts due to water flow is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Frictional effects are negligible (so that the Bernoulli equation can be used). **3** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

Properties We take the density of water to be 1000 kg/m^3.

Analysis





Writing Bernoulli equation between 1-2 ;

$$P_1 + \frac{1}{2}\rho V_1^{\ 2} \ + \gamma z_1 = P_2 + \frac{1}{2}\rho V_2^{\ 2} \ + \gamma z_2 + negligible\ losses$$

$$V_1 = \frac{0.1}{\pi 0.1^2 / 4} = 12.73\ \text{m/s}\ , \quad V_2 = \frac{0.1}{\pi 0.2^2 / 4} = 3.18\ \text{m/s}$$

$$P_1 = \frac{1}{2}\rho\left(V_2^{\ 2} - V_1^{\ 2}\right) = \frac{1}{2}1000\left(12.73^2 - 3.18^2\right) = 75970\ Pa$$

Applying linear momentum to the CV gives;

$$\frac{\partial}{\partial t}\int_{cv} \vec{V}\rho dV + \int_{cs} \vec{V}\rho \vec{V}\vec{n} dA = \sum F_x \quad and \quad V_1\rho(-V_1)A_1 + V_2\rho(+V_2)A_2 = P_1 A_1 - F_{bolts}$$

$$\rho Q\left(V_2 - V_1\right) = P_1 A_1 - F_{bolts}$$

$$F_{bolts} = 75970\pi \frac{0.1^2}{4} - 1000 \times 0.1 \times \left(3.18 - 12.73\right)$$

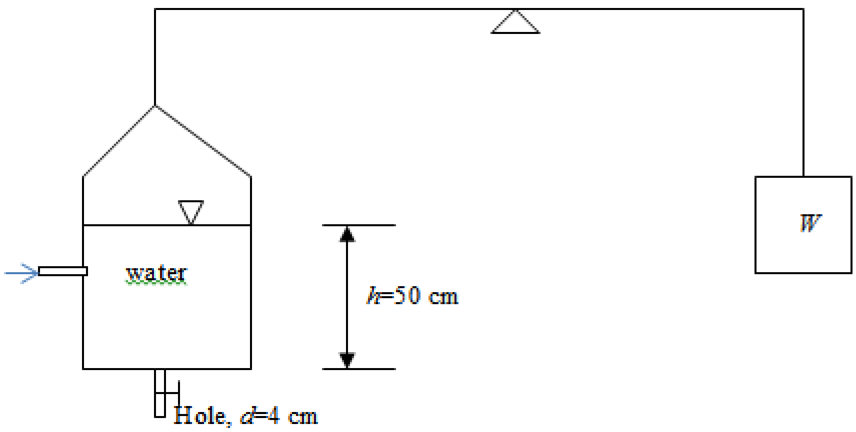
$F_{bolts} = 1552 \leftarrow$,fluid force on bolts **1552 N** to right.

6-41

Solution The weight of a water tank is balanced by a counterweight. Water enters the tank horizontally and there is a hole at the bottom of the tank. The amount of mass that must be added to or removed from the counterweight to maintain balance when the hole at the bottom is opened is to be determined.

Properties We take the density of water to be 1000 kg/m^3.

Analysis



The flowrate from the hole, just after the valve is opened,

$$Q = CA_h\sqrt{2gh} = 0.90\pi\frac{0.04^2}{4}\sqrt{2\times 9.81\times 0.5} = 3.54\times 10^{-3}\ \ m^3/s$$

$$V_{hole} = 0.90\times\sqrt{2\times 9.81\times 0.5} = 2.82\ m/s$$

Reaction force due to momentum change

$$F = \rho Q V_h = 1000\times 3.54\times 10^{-3}\times 2.82 \approx 10\ N$$

Therefore the tank will sink just after the valve is opened, thus m = $\dfrac{10}{9.81}$ ≅ **1 kg** must be removed from the counterweight.

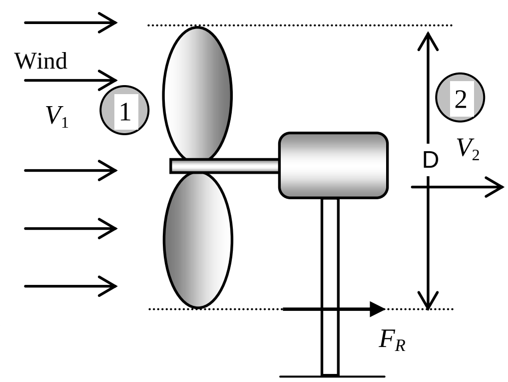
6-42

Solution A wind turbine with a given span diameter and efficiency is subjected to steady winds. The power generated and the horizontal force on the supporting mast of the turbine are to be determined.

Assumptions **1** The wind flow is steady and incompressible. **2** The efficiency of the turbine-generator is independent of wind speed. **3** The frictional effects are negligible, and thus none of the incoming kinetic energy is converted to thermal energy. **4** Wind flow is uniform and thus the momentum-flux correction factor is nearly unity, $\beta \cong 1$.

Properties The density of air is given to be 1.25 kg/m^3.

Analysis (*a*) The power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:



$$V_1 = (30\text{ km/h})\left(\frac{1\text{ m/s}}{3.6\text{ km/h}}\right) = 8.333\text{ m/s}$$

$$\dot{m} = \rho_1 V_1 A_1 = \rho_1 V_1 \frac{\pi D^2}{4} = (1.25\text{ kg/m}^3)(8.333\text{ m/s})\frac{\pi(60\text{ m})^2}{4} = 29{,}452\text{ kg/s}$$

$$\dot{W}_{\max} = \dot{m}ke_1 = \dot{m}\frac{V_1^2}{2} = (29{,}452\text{ kg/s})\frac{(8.333\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right) = 1023\text{ kW}$$

Then the actual power produced becomes

$$\dot{W}_{\text{act}} = \eta_{\text{wind turbine}}\dot{W}_{\max} = (0.32)(1023\text{ kW}) = \mathbf{327\text{ kW}}$$

(*b*) The frictional effects are assumed to be negligible, and thus the portion of incoming kinetic energy not converted to electric power leaves the wind turbine as outgoing kinetic energy. Therefore,

$$\dot{m}ke_2 = \dot{m}ke_1(1-\eta_{\text{wind turbine}}) \quad \rightarrow \quad \dot{m}\frac{V_2^2}{2} = \dot{m}\frac{V_1^2}{2}(1-\eta_{\text{wind turbine}})$$

or

$$V_2 = V_1\sqrt{1-\eta_{\text{wind turbine}}} = (8.333\text{ m/s})\sqrt{1-0.32} = 6.872\text{ m/s}$$

We choose the control volume around the wind turbine such that the wind is normal to the control surface at the inlet and the outlet, and the entire control surface is at the atmospheric pressure. The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. Writing it along the *x*-direction (without forgetting the negative sign for forces and velocities in the negative *x*-direction) and assuming the flow velocity through the turbine to be equal to the wind velocity give

$$F_R = \dot{m}V_2 - \dot{m}V_1 = \dot{m}(V_2 - V_1) = (29{,}452\text{ kg/s})(6.872 - 8.333\text{ m/s})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{-43.0\text{ kN}}$$

The negative sign indicates that the reaction force acts in the negative *x* direction, as expected.

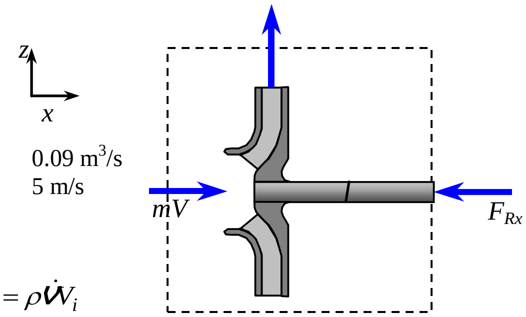
Discussion This force acts on top of the tower where the wind turbine is installed, and the bending moment it generates at the bottom of the tower is obtained by multiplying this force by the tower height.

6-43

Solution Water enters a centrifugal pump axially at a specified rate and velocity, and leaves in the normal direction along the pump casing. The force acting on the shaft in the axial direction is to be determined.

Properties We take the density of water to be 1000 kg/m^3.

Assumptions **1** The flow is steady and incompressible. **2** The forces acting on the piping system in the horizontal direction are negligible. **3** The atmospheric pressure is disregarded since it acts on all surfaces.



Analysis We take the pump as the control volume, and the inlet direction of flow as the positive direction of *x* axis. The momentum equation for steady flow in the *x* (flow) direction reduces in this case to

$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V} \quad \rightarrow \quad -F_{Rx} = -\dot{m}V_i \quad \rightarrow \quad F_{Rx} = \dot{m}V_i = \rho \dot{\mathcal{V}} V_i$$

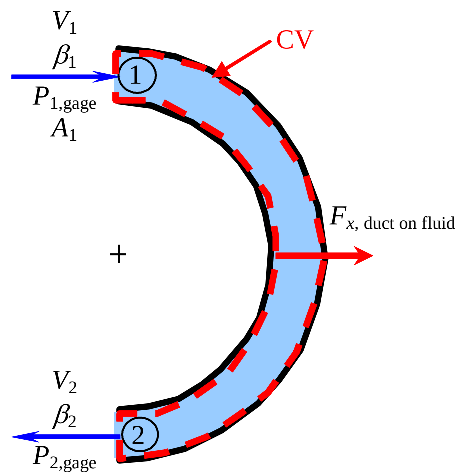
Note that the reaction force acts in the opposite direction to flow, and we should not forget the negative sign for forces and velocities in the negative *x*-direction. Substituting the given values,

$$F_{\text{brake}} = (1000\ \text{kg/m}^3)(0.09\ \text{m}^3/\text{s})(5\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{450\ N}$$

Discussion To find the total force acting on the shaft, we also need to do a force balance for the vertical direction, and find the vertical component of the reaction force.

6-44

Solution A curved duct deflects a fluid. The horizontal force exerted on the duct by the fluid is to be determined.



Assumptions **1** The flow is steady and incompressible. **2** The momentum-flux correction factor for each inlet and outlet is given to account for frictional effects and the non-uniformity of the inlet and outlet velocity profiles.

Properties The density of the liquid is ρ = 998.2 kg/m^3. Viscosity does not enter our analysis.

Analysis (*a*) We take the fluid within the duct as the control volume (see sketch), and designate the inlet by 1 and the outlet by 2. We also designate the horizontal coordinate by *x* (with the direction of flow as being the positive direction). Conservation of mass for this one-inlet one-outlet steady flow system is $\dot{m} = \dot{m}_1 = \dot{m}_2$. The mass flow rate is $\dot{m} = \rho V_1 A_1 = \rho V_2 A_2$, and the average speed at the outlet is thus $V_2 = \dfrac{\dot{m}}{\rho A_2} = \dfrac{\rho V_1 A_1}{\rho A_2} = \dfrac{V_1 A_1}{A_2}$. Since $A_1 = A_2$ in this problem, $V_2 = V_1$. The momentum equation for steady flow in the *x*-direction is $\sum F_{x\text{ on CV}} = \sum_{\text{out}} \beta \dot{m} u - \sum_{\text{in}} \beta \dot{m} u$, where *u* is the horizontal velocity component: $u_1 = V_1$ and $u_2 = -V_1$. The total force on the control volume consists of pressure forces at the inlet and outlet plus the total of all forces (including pressure and viscous forces) acting on the control volume by the duct walls. Calling this force $F_{x,\text{ duct on fluid}}$, and assuming it to be in the positive *x*-direction, we write

$$P_{1,\text{gage}} A_1 + P_{2,\text{gage}} A_2 + F_{x,\text{ duct on fluid}} = \dot{m}\beta_2\left(-V_1\right) - \dot{m}\beta_1 V_1$$

Note that we must be careful with the signs for forces (including pressure forces) and velocities. Solving for $F_{x,\text{ duct on fluid}}$, and plugging in $\dot{m} = \rho V_1 A_1$, we get

$$F_{x,\text{ duct on fluid}} = -\left(P_{1,\text{gage}} + P_{2,\text{gage}}\right) A_1 - \rho V_1^2 A_1 \left(\beta_1 + \beta_2\right)$$

Finally, the force exerted by the fluid on the duct is the negative of this, i.e.,

$$\boxed{F_x = F_{x,\text{ fluid on duct}} = -F_{x,\text{ duct on fluid}} = \left(P_{1,\text{gage}} + P_{2,\text{gage}}\right) A_1 + \rho V_1^2 A_1 \left(\beta_1 + \beta_2\right)}$$

(*b*) We verify our expression by plugging in the given values:

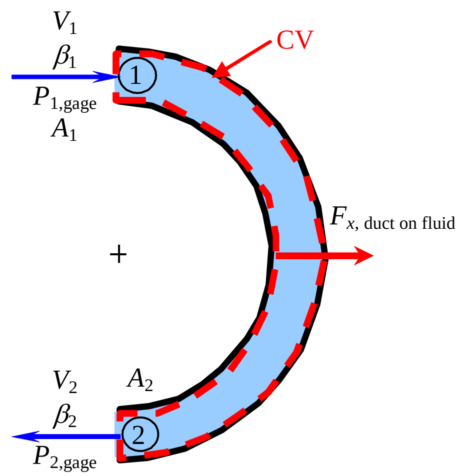
$$F_x = \left(78470 \frac{\text{N}}{\text{m}^2} + 65230 \frac{\text{N}}{\text{m}^2}\right)\left(0.025\text{ m}^2\right) + \left(998.2 \frac{\text{kg}}{\text{m}^3}\right)\left(10 \frac{\text{m}}{\text{s}}\right)^2 \left(0.025\text{ m}^2\right)\left(1.01 + 1.03\right)\left(\frac{\text{N}}{\text{kg}\cdot\text{m/s}^2}\right)$$

$$= 8683.3\text{ N} \cong \mathbf{8680\text{ N}}$$

where we give the final answer to three significant digits in keeping with our convention.

Discussion The direction agrees with our intuition. The fluid is trying to push the duct to the right. A negative value for $F_{x,\text{ fluid on duct}}$ indicates that the initially assumed direction was incorrect, but did not hinder our solution.

6-45



Solution A curved duct deflects a fluid. The horizontal force exerted on the duct by the fluid is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The momentum-flux correction factor for each inlet and outlet is given to account for frictional effects and the non-uniformity of the inlet and outlet velocity profiles.

Properties The density of the liquid is ρ = 998.2 kg/m^3. Viscosity does not enter our analysis.

Analysis (*a*) We take the fluid within the duct as the control volume (see sketch), and designate the inlet by 1 and the outlet by 2. We also designate the horizontal coordinate by *x* (with the direction of flow as being the positive direction). Conservation of mass for this one-inlet one-outlet steady flow system is $\dot{m} = \dot{m}_1 = \dot{m}_2$. The mass flow rate is $\dot{m} = \rho V_1 A_1 = \rho V_2 A_2$, and the average speed at the outlet is thus $V_2 = \dfrac{\dot{m}}{\rho A_2} = \dfrac{\rho V_1 A_1}{\rho A_2} = \dfrac{V_1 A_1}{A_2}$. The momentum equation for steady flow in the *x*-direction is $\sum F_{x \text{ on CV}} = \sum_{\text{out}} \beta \dot{m} u - \sum_{\text{in}} \beta \dot{m} u$, where *u* is the horizontal velocity component: $u_1 = V_1$ and $u_2 = -V_2$. The total force on the control volume consists of pressure forces at the inlet and outlet plus the total of all forces (including pressure and viscous forces) acting on the control volume by the duct walls. Calling this force $F_{x,\text{ duct on fluid}}$, and assuming it to be in the positive *x*-direction, we write

$$P_{1,\text{gage}} A_1 + P_{2,\text{gage}} A_2 + F_{x,\text{ duct on fluid}} = \dot{m}\beta_2\left(-V_2\right) - \dot{m}\beta_1 V_1$$

Note that we must be careful with the signs for forces (including pressure forces) and velocities. Solving for $F_{x,\text{ duct on fluid}}$, and plugging in $\dot{m} = \rho V_1 A_1$ and $V_2 = \dfrac{V_1 A_1}{A_2}$, we get

$$F_{x,\text{ duct on fluid}} = -P_{1,\text{gage}} A_1 - P_{2,\text{gage}} A_2 - \rho V_1^2 A_1 \left(\beta_1 + \beta_2 \frac{A_1}{A_2} \right)$$

Finally, the force exerted by the fluid on the duct is the negative of this, i.e.,

$$\boxed{F_x = F_{x,\text{ fluid on duct}} = -F_{x,\text{ duct on fluid}} = P_{1,\text{gage}} A_1 + P_{2,\text{gage}} A_2 + \rho V_1^2 A_1 \left(\beta_1 + \beta_2 \frac{A_1}{A_2} \right)}$$

(*b*) We verify our expression by plugging in the given values:

$$F_x = \left(88340 \frac{\text{N}}{\text{m}^2}\left(0.025 \text{ m}^2\right) + 67480 \frac{\text{N}}{\text{m}^2}\left(0.015 \text{ m}^2\right) \right)$$
$$+ \left(998.2 \frac{\text{kg}}{\text{m}^3} \right)\left(20 \frac{\text{m}}{\text{s}} \right)^2 \left(0.025 \text{ m}^2\right)\left(1.02 + 1.04 \frac{0.025 \text{ m}^2}{0.015 \text{ m}^2} \right)\left(\frac{\text{N}}{\text{kg}\cdot\text{m/s}^2} \right)$$
$$= 30704.5 \text{ N} \cong \mathbf{30{,}700 \text{ N to the right}}$$

where we give the final answer to three significant digits in keeping with our convention.

Discussion The direction agrees with our intuition. The fluid is trying to push the duct to the right. A negative value for $F_{x,\text{ fluid on duct}}$ indicates that the initially assumed direction was incorrect, but did not hinder our solution.

6-46

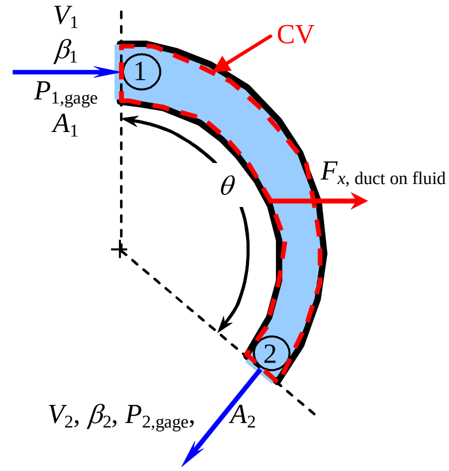
Solution The curved duct analysis of the previous problem is to be examined more closely. Namely, we are to explain how the pressure can actually *rise* from inlet to outlet.

Analysis The simple answer is that **when the area ratio is greater than 1, the duct acts as a diffuser**. From the Bernoulli approximation, we know that as the velocity decreases along the duct (since area increases), the pressure increases. Irreversibilities act counter to this, and cause the pressure to drop along the duct. For large enough area ratio, the diffuser effect "wins". In this problem, the pressure changes from decreasing to increasing at an area ratio of around 1.3.

Discussion The curvature does not affect the fact that the duct acts like a diffuser – the area still increases. In fact, some diffusers are curved, e.g., centrifugal pumps and the draft tubes of hydroelectric turbines, as discussed in the turbomachinery chapter.

6-47

Solution A curved duct deflects a fluid. The horizontal force exerted on the duct by the fluid is to be determined.



Assumptions **1** The flow is steady and incompressible. **2** The momentum-flux correction factor for each inlet and outlet is given to account for frictional effects and the non-uniformity of the inlet and outlet velocity profiles.

Properties The density is given as ρ = 998.2 kg/m^3. Viscosity does not enter our analysis.

Analysis (a) We take the fluid within the duct as the control volume (see sketch), and designate the inlet by 1 and the outlet by 2. We also designate the horizontal coordinate by x (with the direction of flow as being the positive direction). Conservation of mass for this one-inlet one-outlet steady flow system is $\dot{m} = \dot{m}_1 = \dot{m}_2$. The mass flow rate is $\dot{m} = \rho V_1 A_1 = \rho V_2 A_2$, and the average speed at the outlet is thus $V_2 = \dfrac{\dot{m}}{\rho A_2} = \dfrac{\rho V_1 A_1}{\rho A_2} = \dfrac{V_1 A_1}{A_2}$. The momentum equation for steady flow in the x-direction is $\sum F_{x\text{ on CV}} = \sum_{\text{out}} \beta \dot{m} u - \sum_{\text{in}} \beta \dot{m} u$, where u is the horizontal velocity component. The total force on the control volume consists of pressure forces at the inlet and outlet plus the total of all forces (including pressure and viscous forces) acting on the control volume by the duct walls. Calling this force $F_{x,\text{ duct on fluid}}$, and assuming it to be in the positive x-direction, we write

$$P_{1,\text{gage}} A_1 - P_{2,\text{gage}} A_2 \cos\theta + F_{x,\text{ duct on fluid}} = \dot{m}\beta_2 V_2 \cos\theta - \dot{m}\beta_1 V_1$$

Note that we must be careful with the signs for forces (including pressure forces) and velocities. Solving for $F_{x,\text{ duct on fluid}}$, and plugging in $\dot{m} = \rho V_1 A_1$, we get

$$F_{x,\text{ duct on fluid}} = -P_{1,\text{gage}} A_1 + P_{2,\text{gage}} A_2 \cos\theta + \rho V_1 A_1 \left(\beta_2 V_2 \cos\theta - \beta_1 V_1\right)$$

$$\text{where } V_2 = \frac{V_1 A_1}{A_2}$$

Finally, the force exerted by the fluid on the duct is the negative of this, i.e.,

$$\boxed{\begin{aligned} &F_x = F_{x,\text{ fluid on duct}} = -F_{x,\text{ duct on fluid}} = P_{1,\text{gage}} A_1 - P_{2,\text{gage}} A_2 \cos\theta + \rho V_1 A_1 \left(\beta_1 V_1 - \beta_2 V_2 \cos\theta\right) \\ &\text{where } V_2 = \frac{V_1 A_1}{A_2} \end{aligned}}$$

(b) We verify our expression by plugging in the given values:

$$V_2 = \frac{V_1 A_1}{A_2} = \frac{(6\text{ m/s})(0.025\text{ m}^2)}{0.050\text{ m}^2} = 3\text{ m/s}$$

and

$$\begin{aligned} F_x &= (78{,}470\text{ N/m}^2)(0.025\text{ m}^2) - (65{,}230\text{ N/m}^2)(0.050\text{ m}^2)\cos(135°) \\ &\quad + (998.2\text{ kg/m}^3)(6\text{ m/s})(0.025\text{ m}^2)\left[(1.01)(6\text{ m/s}) - (1.03)(3\text{ m/s})\cos(135°)\right]\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) \\ &= 5502\text{ N} \cong \mathbf{5500 N\ to\ the\ right} \end{aligned}$$

where we give the final answer to three significant digits in keeping with our convention.

(c) We predict that **the force would maximize at θ = 180$^\text{o}$ because the water is turned completely around**.

Discussion The direction agrees with our intuition. The fluid is trying to push the duct to the right. A negative value for $F_{x,\text{ fluid on duct}}$ indicates that the initially assumed direction was incorrect, but did not hinder our solution.

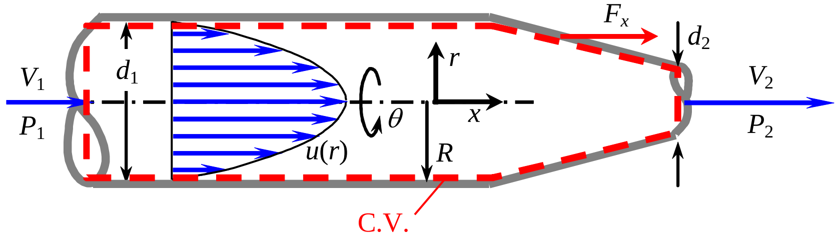
6-48

Solution A fireman's hose accelerates water from high pressure, low velocity to atmospheric pressure, high velocity at the nozzle exit plane. The horizontal force exerted on the duct by the water is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The momentum-flux correction factor for each inlet and outlet is given to account for frictional effects and the non-uniformity of the inlet and outlet velocity profiles.

Properties The density of the water is $\rho = 998.2$ kg/m^3. Viscosity does not enter our analysis.

Analysis (*a*) We take the fluid within the duct as the control volume (see sketch), and designate the inlet by 1 and the outlet by 2. We also designate the horizontal coordinate by *x* (with the direction of flow as being the positive direction). Conservation of mass for this one-inlet one-outlet steady flow system is $\dot{m} = \dot{m}_1 = \dot{m}_2$. The mass flow rate is $\dot{m} = \rho V_1 A_1 = \rho V_2 A_2$, and the average speed at the outlet is thus $V_2 = \dfrac{\dot{m}}{\rho A_2} = \dfrac{\rho V_1 A_1}{\rho A_2} = \dfrac{V_1 A_1}{A_2}$. The momentum equation for steady flow in the *x*-direction is $\sum F_{x \text{ on CV}} = \sum_{\text{out}} \beta \dot{m} u - \sum_{\text{in}} \beta \dot{m} u$, where *u* is the horizontal velocity component: $u_1 = V_1$ and $u_2 = V_2$. The total force on the control volume consists of pressure forces at the inlet and outlet plus the total of all forces (including pressure and viscous forces) acting on the control volume by the duct walls. Calling this force $F_{x,\text{ duct on fluid}}$, and assuming it to be in the positive *x*-direction, we write



$$P_{1,\text{gage}} A_1 - P_{2,\text{gage}} A_2 + F_{x,\text{ duct on fluid}} = \dot{m}\beta_2 \left(V_2\right) - \dot{m}\beta_1 V_1$$

Note that we must be careful with the signs for forces (including pressure forces) and velocities. Solving for $F_{x,\text{ duct on fluid}}$, and plugging in $\dot{m} = \rho V_1 A_1$ and $V_2 = \dfrac{V_1 A_1}{A_2}$, we get

$$F_{x,\text{ duct on fluid}} = P_{2,\text{gage}} \frac{A_2}{A_1} A_1 - P_{1,\text{gage}} A_1 + \rho V_1^2 A_1 \left(\beta_2 \frac{A_1}{A_2} - \beta_1 \right)$$

Finally, the force exerted by the fluid on the duct is the negative of this, i.e.,

$$\boxed{\begin{aligned} F_x = F_{x,\text{ fluid on duct}} = -F_{x,\text{ duct on fluid}} &= \left(P_{1,\text{gage}} - P_{2,\text{gage}} \frac{A_2}{A_1} \right) A_1 + \rho V_1^2 A_1 \left(\beta_1 - \beta_2 \frac{A_1}{A_2} \right) \\ &= A_1 \left[\left(P_{1,\text{gage}} - P_{2,\text{gage}} \frac{A_2}{A_1} \right) + \rho V_1^2 \left(\beta_1 - \beta_2 \frac{A_1}{A_2} \right) \right] \end{aligned}}$$

(*b*) We verify our expression by plugging in the given values:

$$F_x = \frac{\pi}{4}(0.10\text{ m})^2 \left[\left(123{,}000 \frac{\text{N}}{\text{m}^2} - 0 \frac{\text{N}}{\text{m}^2} \left(\frac{0.05}{0.10} \right)^2 \right) + \left(998.2 \frac{\text{kg}}{\text{m}^3} \right) \left(4 \frac{\text{m}}{\text{s}} \right)^2 \left(1.03 - 1.02 \left(\frac{0.10}{0.050} \right)^2 \right) \left(\frac{\text{N}}{\text{kg} \cdot \text{m/s}^2} \right) \right]$$

$$= 583.455\text{ N} \cong \mathbf{583\ N\ to\ the\ right}$$

where we give the final answer to three significant digits in keeping with our convention.

Discussion The direction agrees with our intuition. The fluid is trying to push the duct to the right. A negative value for $F_{x,\text{ fluid on duct}}$ indicates that the initially assumed direction was incorrect, but did not hinder our solution.

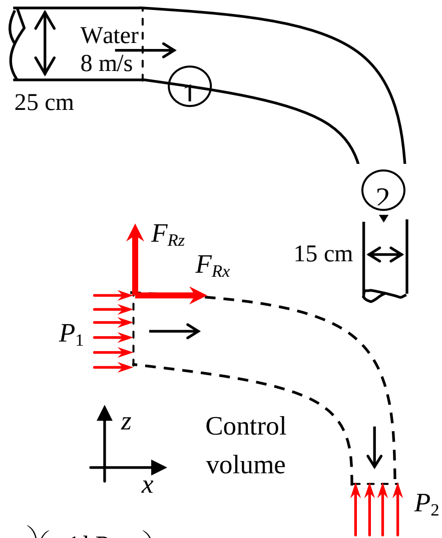
6-49

Solution A 90° reducer elbow deflects water downwards into a smaller diameter pipe. The resultant force exerted on the reducer by water is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Frictional effects are negligible in the calculation of the pressure drop (so that the Bernoulli equation can be used). **3** The weight of the elbow and the water in it is disregarded since the gravitational effects are negligible. **4** The momentum-flux correction factor for each inlet and outlet is given to be $\beta = 1.04$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take the water within the elbow as the control volume (see sketch), and designate the inlet by 1 and the outlet by 2. We also designate the horizontal coordinate by x (with the direction of flow as being the positive direction) and the vertical coordinate by z. The continuity equation for this one-inlet one-outlet steady flow system is $\dot{m} = \dot{m}_1 = \dot{m}_2$. Noting that $\dot{m} = \rho AV$, the mass flow rate of water and its outlet velocity are



$$\dot{m} = \rho V_1 A_1 = \rho V_1 (\pi D_1^2/4) = (1000\ \text{kg/m}^3)(8\ \text{m/s})[\pi(0.25\ \text{m})^2/4] = 392.7\ \text{kg/s}$$

$$V_2 = \frac{\dot{m}}{\rho A_2} = \frac{\dot{m}}{\rho \pi D_2^2/4} = \frac{392.7\ \text{kg/s}}{(1000\ \text{kg/m}^3)[\pi(0.15\ \text{m})^2/4]} = 22.22\ \text{m/s}$$

The Bernoulli equation for a streamline going through the center of the reducing elbow is expressed as

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \rightarrow \quad P_2 = P_1 + \rho g\left(\frac{V_1^2 - V_2^2}{2g} + z_1 - z_2\right)$$

Using gage pressures and substituting, the gage pressure at the outlet becomes

$$P_2 = (300\ \text{kPa}) + (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)\left(\frac{(8\ \text{m/s})^2 - (22.22\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} + 0.5\right)\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) = 90.04\ \text{kPa}$$

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V}$. We let the x- and z- components of the resultant of the reaction forces exerted by the reducer elbow on water be F_{Rx} and F_{Rz}, and assume them to be in the positive directions. Noting that the atmospheric pressure acts from all directions and its effect cancels out, the momentum equations along the x and z axes become

$$F_{Rx} + P_{1,\text{gage}} A_1 = 0 - \beta_1 \dot{m} V_1$$
$$F_{Rz} + P_{2,\text{gage}} A_2 = \beta_2 \dot{m}(-V_2) - 0$$

Note that we should not forget the negative sign for forces (including pressure forces) and velocities in the negative x or z direction. Solving for F_{Rx} and F_{Rz}, and substituting the given values,

$$F_{Rx} = -\beta \dot{m} V_1 - P_{1,\text{ gage}} A_1 = -1.04(392.7\ \text{kg/s})(8\ \text{m/s})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) - (300\ \text{kN/m}^2)\frac{\pi(0.25\ \text{m})^2}{4} = -17.99\ \text{kN}$$

$$F_{Rz} = -\beta \dot{m} V_2 + P_{2,\text{ gage}} A_1 = -1.04(392.7\ \text{kg/s})(22.22\ \text{m/s})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) + (90.04\ \text{kN/m}^2)\frac{\pi(0.15\ \text{m})^2}{4} = -7.484\ \text{kN}$$

The force exerted by the water on the reducer elbow is the negative of this, i.e.,

$$\boldsymbol{F_{x,\ \text{water on reducer}} = 18.0\ \text{kN}} \text{ and } \boldsymbol{F_{z,\ \text{water on reducer}} = 7.48\ \text{kN}}$$

The magnitude of the resultant force exerted by the water on the reducer, and its line of action from the $+x$ direction are

$$F_R = \sqrt{F_{Rx}^2 + F_{Rz}^2} = \sqrt{(-17.99)^2 + (-7.484)^2} = \mathbf{19.5\ kN}$$

$$\theta = \tan^{-1}\frac{F_{Rz}}{F_{Rx}} = \tan^{-1}\frac{-7.484}{-17.99} = \mathbf{22.6^\circ}$$

Discussion The direction agrees with our intuition. The water is trying to push the reducer to the right and up. Negative values for F_{Rx} and F_{Rz} indicate that the initially assumed directions were incorrect, but did not hinder our solution.

6-50

Solution The flow rate in a channel is controlled by a sluice gate by raising or lowering a vertical plate. A relation for the force acting on a sluice gate of width w for steady and uniform flow is to be developed.

Assumptions **1** The flow is steady, incompressible, frictionless, and uniform (and thus the Bernoulli equation is applicable.) **2** Wall shear forces at channel walls are negligible. **3** The channel is exposed to the atmosphere, and thus the pressure at free surfaces is the atmospheric pressure. **4** The flow is horizontal. **5** Water flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Analysis We take point 1 at the free surface of the upstream flow before the gate and point 2 at the free surface of the downstream flow after the gate. We also take the bottom surface of the channel as the reference level so that the elevations of points 1 and 2 are y_1 and y_2, respectively. The application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+y_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+y_2 \quad\rightarrow\quad V_2^2-V_1^2=2g(y_1-y_2) \qquad (1)$$

The flow is assumed to be incompressible and thus the density is constant. Then the conservation of mass relation for this single stream steady flow device can be expressed as

$$\dot{V}_1=\dot{V}_2=\dot{V} \quad\rightarrow\quad A_1V_1=A_2V_2=\dot{V} \quad\rightarrow\quad V_1=\frac{\dot{V}}{A_1}=\frac{\dot{V}}{wy_1} \quad \text{and} \quad V_2=\frac{\dot{V}}{A_2}=\frac{\dot{V}}{wy_2} \qquad (2)$$

Substituting into Eq. (1),

$$\left(\frac{\dot{V}}{wy_2}\right)^2-\left(\frac{\dot{V}}{wy_1}\right)^2=2g(y_1-y_2) \rightarrow \dot{V}=w\sqrt{\frac{2g(y_1-y_2)}{1/y_2^2-1/y_1^2}} \quad\rightarrow\quad \dot{V}=wy_2\sqrt{\frac{2g(y_1-y_2)}{1-y_2^2/y_1^2}} \qquad (3)$$

Substituting Eq. (3) into Eqs. (2) gives the following relations for velocities,

$$V_1=\frac{y_2}{y_1}\sqrt{\frac{2g(y_1-y_2)}{1-y_2^2/y_1^2}} \quad \text{and} \quad V_2=\sqrt{\frac{2g(y_1-y_2)}{1-y_2^2/y_1^2}} \qquad (4)$$

We choose the control volume as the water body surrounded by the vertical cross-sections of the upstream and downstream flows, free surfaces of water, the inner surface of the sluice gate, and the bottom surface of the channel. The momentum equation for steady flow is $\sum \vec{F}=\sum_{\text{out}}\beta\dot{m}\vec{V}-\sum_{\text{in}}\beta\dot{m}\vec{V}$. The force acting on the sluice gate F_{Rx} is horizontal since the wall shear at the surfaces is negligible, and it is equal and opposite to the force applied on water by the sluice gate. Noting that the pressure force acting on a vertical surface is equal to the product of the pressure at the centroid of the surface and the surface area, the momentum equation along the x direction gives

$$-F_{Rx}+P_1A_1-P_2A_2=\dot{m}V_2-\dot{m}V_1 \quad\rightarrow\quad -F_{Rx}+\left(\rho g\frac{y_1}{2}\right)(wy_1)-\left(\rho g\frac{y_2}{2}\right)(wy_2)=\dot{m}(V_2-V_1)$$

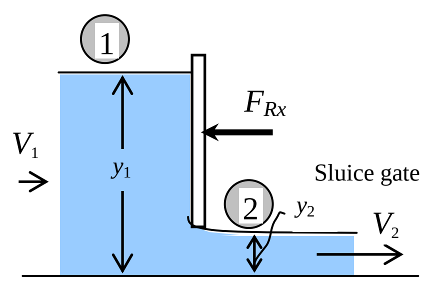
Rearranging, the force acting on the sluice gate is determined to be

$$\boxed{F_{Rx}=\dot{m}(V_1-V_2)+\frac{w}{2}\rho g(y_1^2-y_2^2)} \qquad (5)$$

where V_1 and V_2 are given in Eq. (4). Thus,

$$\boxed{F_{Rx}=\dot{m}\left[\frac{y_2}{y_1}\sqrt{\frac{2g(y_1-y_2)}{1-y_2^2/y_1^2}}-\sqrt{\frac{2g(y_1-y_2)}{1-y_2^2/y_1^2}}\right]+\frac{w}{2}\rho g(y_1^2-y_2^2)}$$

or, simplifying, $\boxed{F_{Rx}=\dot{m}\left(\frac{y_2}{y_1}-1\right)\sqrt{\frac{2g(y_1-y_2)}{1-y_2^2/y_1^2}}+\frac{w}{2}\rho g(y_1^2-y_2^2)}$.



Discussion Note that for $y_1 >> y_2$, Eq. (3) simplifies to $\dot{V}=y_2w\sqrt{2gy_1}$ or $V_2=\sqrt{2gy_1}$ which is the Toricelli equation for frictionless flow from a tank through a hole a distance y_1 below the free surface.

Angular Momentum Equation

6-51C

Solution We are to discuss how the angular momentum equation is obtained from the RTT.

Analysis The *angular momentum equation* is obtained by **replacing *B* in the Reynolds transport theorem by the total angular momentum $\vec{H}_{sys}$, and *b* by the angular momentum per unit mass $\vec{r} \times \vec{V}$.**

Discussion The RTT is a general equation that holds for any property *B*, either scalar or (as in this case) vector.

6-52C

Solution We are to express the angular momentum equation in scalar form about a specified axis.

Analysis The angular momentum equation about a given fixed axis in this case can be expressed in scalar form as $\boxed{\sum M = \sum_{\text{out}} r\dot{m}V - \sum_{\text{in}} r\dot{m}V}$ where *r* is the moment arm, *V* is the magnitude of the radial velocity, and $\dot{m}$ is the mass flow rate.

Discussion This is a simplification of the more general angular momentum equation (many terms have dropped out).

6-53C

Solution We are to express the angular momentum equation for a specific (restricted) control volume.

Analysis The angular momentum equation in this case is expressed as $\boxed{I\vec{\alpha} = -\vec{r} \times \dot{m}\vec{V}}$ where $\vec{\alpha}$ is the angular acceleration of the control volume, and $\vec{r}$ is the vector from the axis of rotation to any point on the line of action of $\vec{F}$.

Discussion This is a simplification of the more general angular momentum equation (many terms have dropped out).

6-54C

Solution We are to compare the angular momentum of two rotating bodies

Analysis **No. The two bodies do not necessarily have the same angular momentum**. Two rigid bodies having the same mass and angular speed may have different angular momentums unless they also have the same moment of inertia *I*.

Discussion The reason why flywheels have most of their mass at the outermost radius, is to maximize the angular momentum.

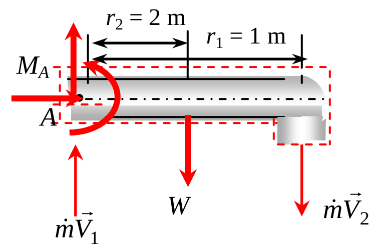
6-55

Solution Water is pumped through a piping section. The moment acting on the elbow for the cases of downward and upward discharge is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero. **3** Effects of water falling down during upward discharge is disregarded. **4** Pipe outlet diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take the entire pipe as the control volume, and designate the inlet by 1 and the outlet by 2. We also take the x and y coordinates as shown. The control volume and the reference frame are fixed. The conservation of mass equation for this one-inlet one-outlet steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m}$, and $V_1 = V_2 = V$ since A_c = constant. The mass flow rate and the weight of the horizontal section of the pipe are



$$\dot{m} = \rho A_c V = (1000\ \text{kg/m}^3)[\pi(0.15\,\text{m})^2/4](7\,\text{m/s}) = 123.7\,\text{kg/s}$$

$$W = mg = (15\ \text{kg/m})(2\,\text{m})(9.81\,\text{m/s}^2)\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = 294.3\,\text{N/m}$$

(*a*) **Downward discharge**: To determine the moment acting on the pipe at point *A*, we need to take the moment of all forces and momentum flows about that point. This is a steady and uniform flow problem, and all forces and momentum flows are in the same plane. Therefore, the angular momentum equation in this case can be expressed as $\sum M = \sum_{\text{out}} r\dot{m}V - \sum_{\text{in}} r\dot{m}V$ where r is the moment arm, all moments in the counterclockwise direction are positive, and all in the clockwise direction are negative.

The free body diagram of the pipe section is given in the figure. Noting that the moments of all forces and momentum flows passing through point *A* are zero, the only force that will yield a moment about point *A* is the weight *W* of the horizontal pipe section, and the only momentum flow that will yield a moment is the outlet stream (both are negative since both moments are in the clockwise direction). Then the angular momentum equation about point *A* becomes

$$M_A - r_1 W = -r_2 \dot{m} V_2$$

Solving for M_A and substituting,

$$M_A = r_1 W - r_2 \dot{m} V_2 = (1\,\text{m})(294.3\,\text{N}) - (2\,\text{m})(123.7\,\text{kg/s})(7\,\text{m/s})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{-1438\,N\cdot m}$$

The negative sign indicates that the assumed direction for M_A is wrong, and should be reversed. Therefore, a moment of 70 N·m acts at the stem of the pipe in the clockwise direction.

(*b*) **Upward discharge**: The moment due to discharge stream is positive in this case, and the moment acting on the pipe at point *A* is

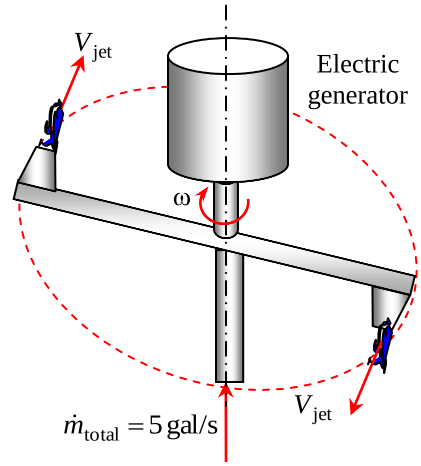
$$M_A = r_1 W + r_2 \dot{m} V_2 = (1\,\text{m})(294.3\,\text{N}) + (2\,\text{m})(123.7\,\text{kg/s})(7\,\text{m/s})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{2026\,N\cdot m}$$

Discussion Note direction of discharge can make a big difference in the moments applied on a piping system. This problem also shows the importance of accounting for the moments of momentums of flow streams when performing evaluating the stresses in pipe materials at critical cross-sections.

6-56E

Solution A two-armed sprinkler is used to generate electric power. For a specified flow rate and rotational speed, the power produced is to be determined.

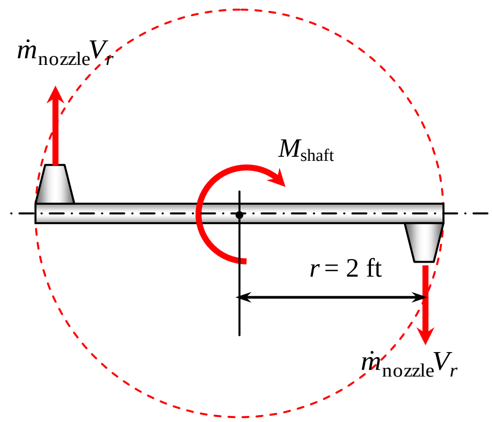
Assumptions **1** The flow is cyclically steady (i.e., steady from a frame of reference rotating with the sprinkler head). **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** Generator losses and air drag of rotating components are neglected. **4** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.



Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis We take the disk that encloses the sprinkler arms as the control volume, which is a stationary control volume. The conservation of mass equation for this steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Noting that the two nozzles are identical, we have $\dot{m}_{\text{nozzle}} = \dot{m}/2$ or $\dot{V}_{\text{nozzle}} = \dot{V}_{total}/2$ since the density of water is constant. The average jet outlet velocity relative to the nozzle is

$$V_{\text{jet}} = \frac{\dot{V}_{\text{nozzle}}}{A_{\text{jet}}} = \frac{2.5 \text{ gal/s}}{[\pi(0.5/12 \text{ ft})^2/4]}\left(\frac{1 \text{ ft}^3}{7.480 \text{ gal}}\right) = 245.1 \text{ ft/s}$$



The angular and tangential velocities of the nozzles are

$$\omega = 2\pi\dot{n} = 2\pi(180 \text{ rev/min})\left(\frac{1 \text{ min}}{60 \text{ s}}\right) = 18.85 \text{ rad/s}$$

$$V_{\text{nozzle}} = r\omega = (2 \text{ ft})(18.85 \text{ rad/s}) = 37.70 \text{ ft/s}$$

The velocity of water jet relative to the control volume (or relative to a fixed location on earth) is

$$V_r = V_{\text{jet}} - V_{\text{nozzle}} = 245.1 - 37.70 = 207.4 \text{ ft/s}$$

The angular momentum equation can be expressed as $\sum M = \sum_{\text{out}} r\dot{m}V - \sum_{\text{in}} r\dot{m}V$ where all moments in the counterclockwise direction are positive, and all in the clockwise direction are negative. Then the angular momentum equation about the axis of rotation becomes

$$-M_{\text{shaft}} = -2r\dot{m}_{\text{nozzle}}V_r \qquad \text{or} \qquad M_{\text{shaft}} = r\dot{m}_{\text{total}}V_r$$

Substituting, the torque transmitted through the shaft is determined to be

$$M_{\text{shaft}} = r\dot{m}_{\text{total}}V_r = (2 \text{ ft})(41.71 \text{ lbm/s})(207.4 \text{ ft/s})\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right) = 537.3 \text{ lbf}\cdot\text{ft}$$

since $\dot{m}_{\text{total}} = \rho\dot{V}_{\text{total}} = (62.4 \text{ lbm/ft}^3)(5/7.480 \text{ ft}^3/\text{s}) = 41.71 \text{ lbm/s}$. Then the power generated becomes

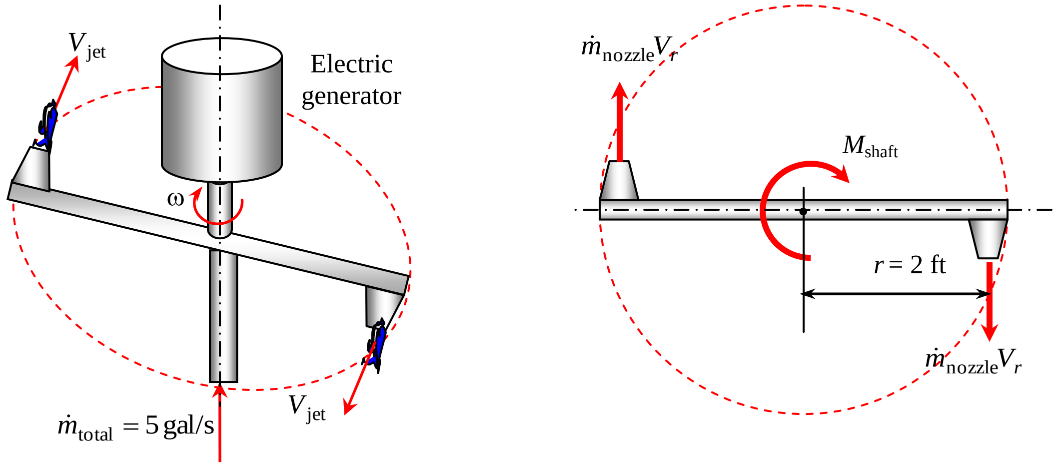
$$\dot{W} = 2\pi\dot{n}M_{\text{shaft}} = \omega M_{\text{shaft}} = (18.85 \text{ rad/s})(537.3 \text{ lbf}\cdot\text{ft})\left(\frac{1 \text{ kW}}{737.56 \text{ lbf}\cdot\text{ft/s}}\right) = \mathbf{13.7 \text{ kW}}$$

Therefore, this sprinkler-type turbine has the potential to produce 13.7 kW of power.

Discussion This is, of course, the maximum possible power. The actual power generated would be much smaller than this due to all the irreversible losses that we have ignored in this analysis.

6-57E

Solution A two-armed sprinkler is used to generate electric power. For a specified flow rate and rotational speed, the moment acting on the rotating head when the head is stuck is to be determined.



Assumptions **1** The flow is uniform and steady. **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis We take the disk that encloses the sprinkler arms as the control volume, which is a stationary control volume. The conservation of mass equation for this steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Noting that the two nozzles are identical, we have $\dot{m}_{nozzle} = \dot{m}/2$ or $\dot{V}_{nozzle} = \dot{V}_{total}/2$ since the density of water is constant. The average jet outlet velocity relative to the nozzle is

$$V_{jet} = \frac{\dot{V}_{nozzle}}{A_{jet}} = \frac{2.5 \text{ gal/s}}{[\pi(0.5/12 \text{ ft})^2/4]}\left(\frac{1 \text{ ft}^3}{7.480 \text{ gal}}\right) = 245.1 \text{ ft/s}$$

The angular momentum equation can be expressed as $\sum M = \sum_{out} r\dot{m}V - \sum_{in} r\dot{m}V$ where all moments in the counterclockwise direction are positive, and all in the clockwise direction are negative. Then the angular momentum equation about the axis of rotation becomes

$$-M_{shaft} = -2r\dot{m}_{nozzle}V_{jet} \qquad \text{or} \qquad M_{shaft} = r\dot{m}_{total}V_{jet}$$

Substituting, the torque transmitted through the shaft is determined to be

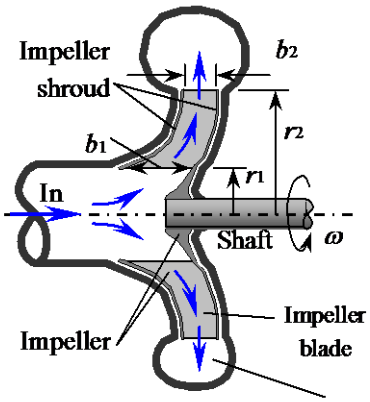
$$M_{shaft} = r\dot{m}_{total}V_{jet} = (2 \text{ ft})(41.71 \text{ lbm/s})(245.1 \text{ ft/s})\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{635 \text{ lbf}\cdot\text{ft}}$$

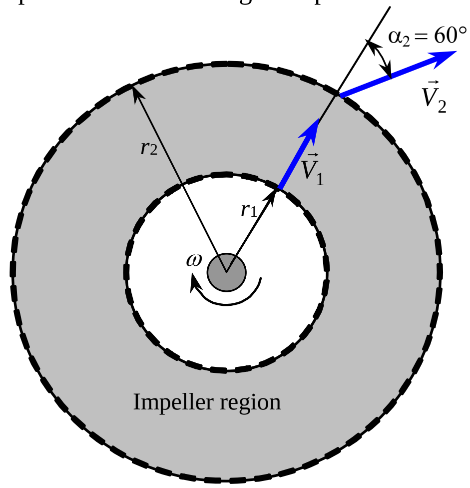
since $\dot{m}_{total} = \rho\dot{V}_{total} = (62.4 \text{ lbm/ft}^3)(5/7.480 \text{ ft}^3\text{/s}) = 41.71 \text{ lbm/s}$.

Discussion When the sprinkler is stuck and thus the angular velocity is zero, the torque developed is maximum since $V_{nozzle} = 0$ and thus $V_r = V_{jet} = 245.1$ ft/s, giving $M_{shaft,max}$ = 635 lbf·ft. But the power generated is zero in this case since the shaft does not rotate.

6-58

Solution A centrifugal pump is used to supply water at a specified rate and angular speed. The minimum power consumption of the pump is to be determined.





Assumptions **1** The flow is steady in the mean. **2** Irreversible losses are negligible.

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take the impeller region as the control volume. The normal velocity components at the inlet and the outlet are

$$V_{1,n} = \frac{\dot{V}}{2\pi r_1 b_1} = \frac{0.15\ \text{m}^3/\text{s}}{2\pi\left(\left(0.13/2\right)\ \text{m}\right)(0.080\ \text{m})} = 4.5910\ \text{m/s}$$

$$V_{2,n} = \frac{\dot{V}}{2\pi r_2 b_2} = \frac{0.15\ \text{m}^3/\text{s}}{2\pi\left(\left(0.30/2\right)\ \text{m}\right)(0.035\ \text{m})} = 4.54728\ \text{m/s}$$

The tangential components of absolute velocity are:

$\alpha_1 = 0°$: $\quad V_{1,t} = V_{1,n}\tan\alpha_1 = 0$

$\alpha_2 = 60°$: $\quad V_{2,t} = V_{2,n}\tan\alpha_1 = (4.54728\ \text{m/s})\tan 60° = 7.87613\ \text{m/s}$

The angular velocity of the propeller is

$$\omega = 2\pi\dot{n} = 2\pi(1200\ \text{rev/min})\left(\frac{1\,\text{min}}{60\,\text{s}}\right) = 125.7\ \text{rad/s}$$

$$\dot{m} = \rho\dot{V} = (1000\ \text{kg/m}^3)(0.15\,\text{m}^3/\text{s}) = 150\ \text{kg/s}$$

Normal velocity components $V_{1,n}$ and $V_{2,n}$ as well pressure acting on the inner and outer circumferential areas pass through the shaft center, and thus they do not contribute to torque. Only the tangential velocity components contribute to torque, and the application of the angular momentum equation gives

$$\mathrm{T}_{\text{shaft}} = \dot{m}(r_2V_{2,t} - r_1V_{1,t}) = (150\ \text{kg/s})[(0.30\ \text{m})(7.87613\ \text{m/s})\text{-}0]\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 0.354426\ \text{kN}\cdot\text{m}$$

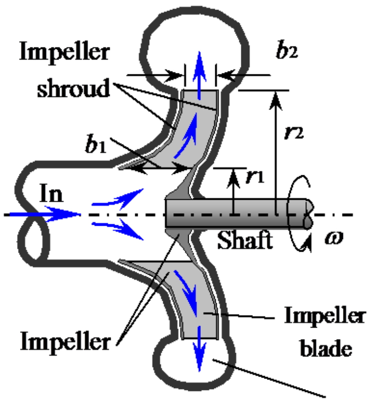
Then the shaft power becomes

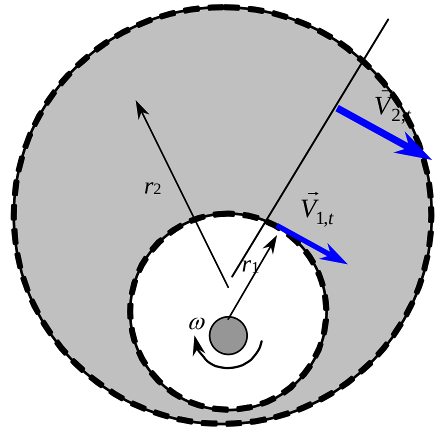
$$\dot{W} = \omega\mathrm{T}_{\text{shaft}} = (125.7\ \text{rad/s})(0.354426\ \text{kN}\cdot\text{m})\left(\frac{1\ \text{kW}}{1\ \text{kN}\cdot\text{m/s}}\right) = 44.5513\ \text{kW} \cong \mathbf{44.6\ kW}$$

Discussion Note that the irreversible losses are not considered in analysis. In reality, the required power input will be larger.

6-59

Solution A centrifugal blower is used to deliver atmospheric air. For a given angular speed and power input, the volume flow rate of air is to be determined.





Assumptions **1** The flow is steady in the mean. **2** Irreversible losses are negligible. **3** The tangential components of air velocity at the inlet and the outlet are said to be equal to the impeller velocity at respective locations.

Properties The gas constant of air is 0.287 kPa·m³/kg·K. The density of air at 20°C and 95 kPa is

$$\rho = \frac{P}{RT} = \frac{95\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 1.130\text{ kg/m}^3$$

Analysis In the idealized case of the tangential fluid velocity being equal to the blade angular velocity both at the inlet and the outlet, we have $V_{1,t} = \omega r_1$ and $V_{2,t} = \omega r_2$, and the torque is expressed as

$$\mathrm{T}_{\text{shaft}} = \dot{m}(r_2 V_{2,t} - r_1 V_{1,t}) = \dot{m}\omega(r_2^2 - r_1^2) = \rho\dot{V}\omega(r_2^2 - r_1^2)$$

where the angular velocity is

$$\omega = 2\pi\dot{n} = 2\pi(900\text{ rev/min})\left(\frac{1\text{ min}}{60\text{ s}}\right) = 94.25\text{ rad/s}$$

Then the shaft power becomes

$$\dot{W}_{\text{shaft}} = \omega\mathrm{T}_{\text{shaft}} = \rho\dot{V}\omega^2(r_2^2 - r_1^2)$$

Solving for $\dot{V}$ and substituting, the volume flow rate of air is determined to

$$\dot{V} = \frac{\dot{W}_{\text{shaft}}}{\rho\omega^2(r_2^2 - r_1^2)} = \frac{120\text{ N}\cdot\text{m/s}}{(1.130\text{ kg/m}^3)(94.25\text{ rad/s})^2[(0.30\text{ m})^2 - (0.18\text{ m})^2]}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{0.2075\,m^3/s}$$

The normal velocity components at the inlet and the outlet are

$$V_{1,n} = \frac{\dot{V}}{2\pi r_1 b_1} = \frac{0.2075\text{ m}^3/\text{s}}{2\pi(0.18\text{ m})(0.061\text{ m})} = \mathbf{3.01\,m/s}$$

$$V_{2,n} = \frac{\dot{V}}{2\pi r_2 b_2} = \frac{0.2075\text{ m}^3/\text{s}}{2\pi(0.30\text{ m})(0.034\text{ m})} = \mathbf{3.24\,m/s}$$

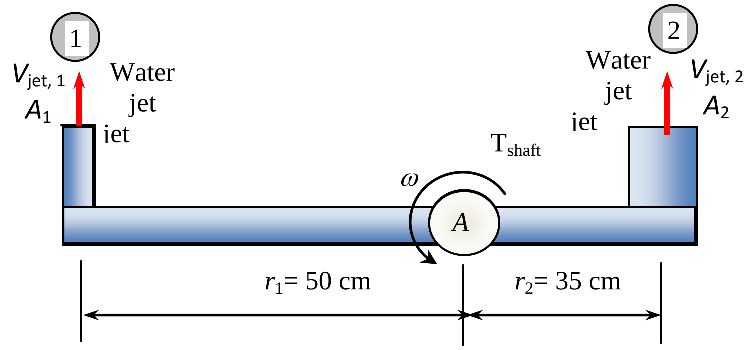
Discussion Note that the irreversible losses are not considered in this analysis. In reality, the flow rate and the normal components of velocities will be smaller.

6-60

Solution Water enters a two-armed sprinkler vertically, and leaves the nozzles horizontally. For a specified flow rate, the rate of rotation of the sprinkler and the torque required to prevent the sprinkler from rotating are to be determined.

Assumptions **1** The flow is uniform and cyclically steady (i.e., steady from a frame of reference rotating with the sprinkler head). **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** Frictional effects and air drag of rotating components are neglected. **4** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 1000 kg/m^3 = 1 kg/L.



Analysis We take the disk that encloses the sprinkler arms as the control volume, which is a stationary control volume. The conservation of mass equation for this symmetrical steady flow system is $\dot m_1 + \dot m_2 = \dot m$ or $\dot V_{\text{jet},1} + \dot V_{\text{jet},2} = \dot V_{\text{total}}$ since the density of water is constant. Both jets are at the same elevation and pressure, and the frictional effects are said to be negligible. Also, the head losses in each arm are equal, so that mass flow rates in each nozzle are identical. Then,

$$V_{\text{jet},r,1} = \frac{\dot V_{\text{total}}/2}{A_{\text{jet},1}} = \frac{(35/2)\ \text{L/s}}{3\times10^{-4}\ \text{m}^2}\left(\frac{1\ \text{m}^3}{1000\ \text{L}}\right) = 58.33\ \text{m/s}$$

$$V_{\text{jet},r,2} = \frac{\dot V_{\text{total}}/2}{A_{\text{jet},2}} = \frac{(35/2)\ \text{L/s}}{5\times10^{-4}\ \text{m}^2}\left(\frac{1\ \text{m}^3}{1000\ \text{L}}\right) = 35.00\ \text{m/s}$$

Noting that $V_{\text{nozzle}} = \omega r = 2\pi \dot n r$ and assuming the sprinkler to rotate in the clock-wise direction, the absolute water jet speeds in the tangential direction can be expressed as

$$V_{\text{jet},1} = V_{\text{jet},r,1} + V_{\text{nozzle},1} = V_{\text{jet},r,1} + \omega r_1 \quad \text{(Nozzle and water jet move in the same direction).}$$

$$V_{\text{jet},2} = V_{\text{jet},r,2} - V_{\text{nozzle},2} = V_{\text{jet},r,2} - \omega r_2 \quad \text{(Nozzle and water jet move in opposite directions).}$$

The angular momentum equation about the axis of rotation can be expressed as $\sum M = \sum_{\text{out}} r\dot m V - \sum_{\text{in}} r\dot m V$ where r is the average moment arm, V is the average absolute speed (relative to an inertial reference frame), all moments in the counterclockwise direction are positive, and all moments in the clockwise direction are negative. Momentum flows in the clockwise direction are also negative. Then the angular momentum equation becomes

$$T_{\text{shaft}} = -r_1\dot m_{\text{jet},1}V_{\text{jet},1} + r_2\dot m_{\text{jet},2}V_{\text{jet},2} \rightarrow T_{\text{shaft}} = -r_1\dot m_{\text{jet},1}(V_{\text{jet},r,1} + \omega r_1) + r_2\dot m_{\text{jet},2}(V_{\text{jet},r,2} - \omega r_2)$$

Noting that $\dot m_{\text{jet}} = \rho A V_{\text{jet},r}$ and rearranging,

$$\dot m_{jet,1} = \rho \dot V = 1000\times17.5/1000 = 17.5\ \text{kg/s} = \dot m_{jet,2}$$

(*a*) In the case of free spin with no frictional effects, we have $T_{\text{shaft}} = 0$ and thus $0 = (r_2A_2 - r_1A_1)V_{\text{jet},r} - (r_1^2A_1 + r_2^2A_2)\omega$. Then angular speed and the rate of rotation of sprinkler head becomes

$$0 = -0.5\times17.5\times(58.33 + 0.5\omega) + 0.35\times17.5\times(35 - 0.35\omega)$$

$$\omega = -45.41\ \text{rad/s} = \mathbf{434\ rpm}\ \text{(counterclockwise)}$$

(*b*) When the sprinkler is prevented from rotating, we have ω= 0. Then the required torque becomes

$$T_{\text{shaft}} = -r_1\dot m_{\text{jet},1}V_{\text{jet},r,1} + r_2\dot m_{\text{jet},2}V_{\text{jet},r,2} = -0.5\times17.5\times58.33 + 0.35\times17.5\times35 = \mathbf{-296\ N\cdot m}$$

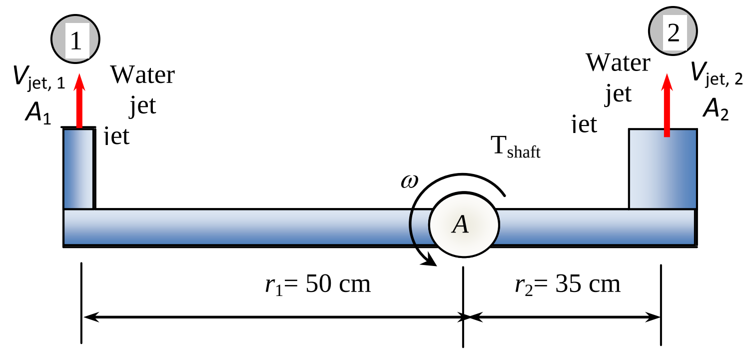
Discussion The rate of rotation determined in (*a*) will be lower in reality because of frictional effects and air drag.

6-61

Solution Water enters a two-armed sprinkler vertically, and leaves the nozzles horizontally. For a specified flow rate, the rate of rotation of the sprinkler and the torque required to prevent the sprinkler from rotating are to be determined.

Assumptions **1**The flow is uniform and cyclically steady (i.e., steady from a frame of reference rotating with the sprinkler head). **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** Frictional effects and air drag of rotating components are neglected. **4** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 1000 kg/m^3 = 1 kg/L.



Analysis We take the disk that encloses the sprinkler arms as the control volume, which is a stationary control volume. The conservation of mass equation for this symmetrical steady flow system is $\dot{m}_1 + \dot{m}_2 = \dot{m}$ or $\dot{V}_{\text{jet},1} + \dot{V}_{\text{jet},2} = \dot{V}_{\text{total}}$ since the density of water is constant. Both jets are at the same elevation and pressure, and the frictional effects are said to be negligible. Also, the head losses in each arm are equal, so that mass flow rates in each nozzle are identical. Then,

$$V_{\text{jet},r,1} = \frac{\dot{V}_{\text{total}}/2}{A_{\text{jet},1}} = \frac{(50/2)\text{ L/s}}{3\times10^{-4}\text{ m}^2}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right) = 83.33\text{ m/s}$$

$$V_{\text{jet},r,2} = \frac{\dot{V}_{\text{total}}/2}{A_{\text{jet},2}} = \frac{(50/2)\text{ L/s}}{5\times10^{-4}\text{ m}^2}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right) = 50.0\text{ m/s}$$

Noting that $V_{\text{nozzle}} = \omega r = 2\pi\dot{n}r$ and assuming the sprinkler to rotate in the clock-wise direction, the absolute water jet speeds in the tangential direction can be expressed as

$$V_{\text{jet},1} = V_{\text{jet},r,1} + V_{\text{nozzle},1} = V_{\text{jet},r,1} + \omega r_1 \quad \text{(Nozzle and water jet move in the same direction).}$$

$$V_{\text{jet},2} = V_{\text{jet},r,2} - V_{\text{nozzle},2} = V_{\text{jet},r,2} - \omega r_2 \quad \text{(Nozzle and water jet move in opposite directions).}$$

The angular momentum equation about the axis of rotation can be expressed as $\sum M = \sum_{\text{out}} r\dot{m}V - \sum_{\text{in}} r\dot{m}V$ where r is the average moment arm, V is the average absolute speed (relative to an inertial reference frame), all moments in the counterclockwise direction are positive, and all moments in the clockwise direction are negative. Momentum flows in the clockwise direction are also negative. Then the angular momentum equation becomes

$$T_{\text{shaft}} = -r_1\dot{m}_{\text{jet},1}V_{\text{jet},1} + r_2\dot{m}_{\text{jet},2}V_{\text{jet},2} \rightarrow T_{\text{shaft}} = -r_1\dot{m}_{\text{jet},1}(V_{\text{jet},r,1} + \omega r_1) + r_2\dot{m}_{\text{jet},2}(V_{\text{jet},r,2} - \omega r_2)$$

Noting that $\dot{m}_{\text{jet}} = \rho A V_{\text{jet},r}$ and rearranging,

$$\dot{m}_{jet,1} = \rho\dot{V} = 1000\times 25/1000 = 25\text{ kg/s} = \dot{m}_{jet,2}$$

(*a*) In the case of free spin with no frictional effects, we have $T_{\text{shaft}} = 0$ and thus $0 = (r_2A_2 - r_1A_1)V_{\text{jet},r} - (r_1^2A_1 + r_2^2A_2)\omega$.
Then angular speed and the rate of rotation of sprinkler head becomes

$$0 = -0.5\times 25\times(83.33 + 0.5\omega) + 0.35\times 25\times(50 - 0.35\omega)$$

$$\omega = -64.89\text{ rad/s} = \mathbf{620\ rpm}\text{ (counterclockwise)}$$

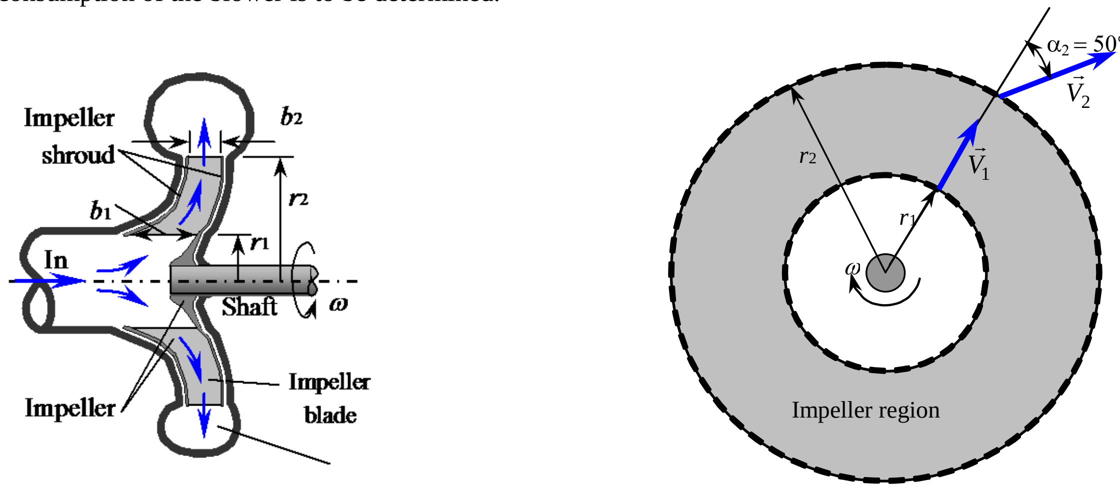
(*b*) When the sprinkler is prevented from rotating, we have ω= 0. Then the required torque becomes

$$T_{\text{shaft}} = -r_1\dot{m}_{\text{jet},1}V_{\text{jet},r,1} + r_2\dot{m}_{\text{jet},2}V_{\text{jet},r,2} = -0.5\times 25\times 83.33 + 0.35\times 25\times 50 = \mathbf{-604\ N\cdot m}$$

Discussion The rate of rotation determined in (*a*) will be lower in reality because of frictional effects and air drag.

6-62

Solution A centrifugal blower is used to deliver atmospheric air at a specified rate and angular speed. The minimum power consumption of the blower is to be determined.



Assumptions **1** The flow is steady in the mean. **2** Irreversible losses are negligible.

Properties The density of air is given to be 1.25 kg/m^3.

Analysis We take the impeller region as the control volume. The normal velocity components at the inlet and the outlet are

$$V_{1,n} = \frac{\dot{V}}{2\pi r_1 b_1} = \frac{0.70\ \text{m}^3/\text{s}}{2\pi(0.20\ \text{m})(0.082\ \text{m})} = 6.793\ \text{m/s}$$

$$V_{2,n} = \frac{\dot{V}}{2\pi r_2 b_2} = \frac{0.70\ \text{m}^3/\text{s}}{2\pi(0.45\ \text{m})(0.056\ \text{m})} = 4.421\ \text{m/s}$$

The tangential components of absolute velocity are:

$\alpha_1 = 0°$: $\quad V_{1,t} = V_{1,n}\tan\alpha_1 = 0$

$\alpha_2 = 60°$: $\quad V_{2,t} = V_{2,n}\tan\alpha_1 = (4.421\ \text{m/s})\tan 50° = 5.269\ \text{m/s}$

The angular velocity of the propeller is

$$\omega = 2\pi\dot{n} = 2\pi(700\ \text{rev/min})\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = 73.30\ \text{rad/s}$$

$$\dot{m} = \rho\dot{V} = (1.25\ \text{kg/m}^3)(0.7\ \text{m}^3/\text{s}) = 0.875\ \text{kg/s}$$

Normal velocity components $V_{1,n}$ and $V_{2,n}$ as well pressure acting on the inner and outer circumferential areas pass through the shaft center, and thus they do not contribute to torque. Only the tangential velocity components contribute to torque, and the application of the angular momentum equation gives

$$\text{T}_{\text{shaft}} = \dot{m}(r_2V_{2,t} - r_1V_{1,t}) = (0.875\ \text{kg/s})[(0.45\ \text{m})(5.269\ \text{m/s}) - 0]\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = 2.075\ \text{N}\cdot\text{m}$$

Then the shaft power becomes

$$\dot{W} = \omega\text{T}_{\text{shaft}} = (73.30\ \text{rad/s})(2.075\ \text{N}\cdot\text{m})\left(\frac{1\ \text{W}}{1\ \text{N}\cdot\text{m/s}}\right) = \mathbf{152\ W}$$

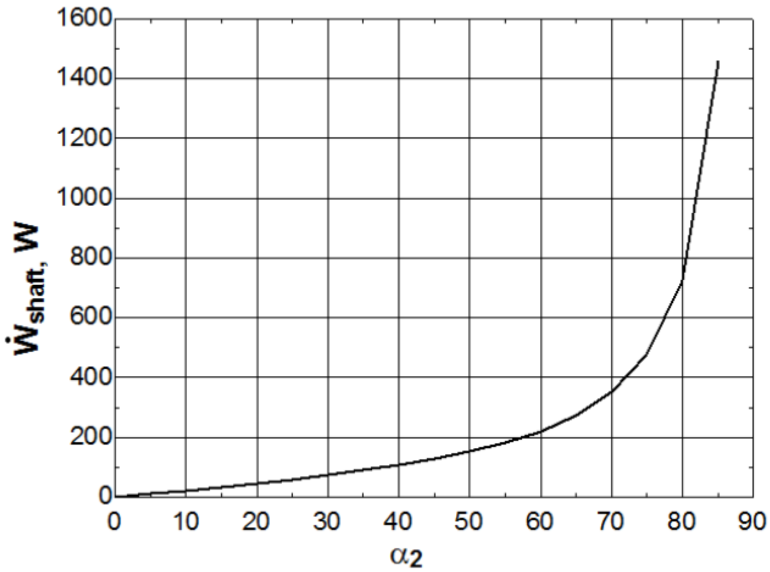
Discussion The actual required shaft power is greater than this, due to the friction and other irreversibilities that we have neglected in our analysis. Nevertheless, this is a good first approximation.

6-63

Solution The previous problem is reconsidered. The effect of discharge angle α_2 on the minimum power input requirements as α_2 varies from 0° to 85° in increments of 5° is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
rho=1.25 "kg/m3"
r1=0.20 "m"
b1=0.082 "m"
r2=0.45 "m"
b2=0.056 "m"
V_dot=0.70 "m3/s"
V1n=V_dot/(2*pi*r1*b1) "m/s"
V2n=V_dot/(2*pi*r2*b2) "m/s"
Alpha1=0
V1t=V1n*tan(Alpha1) "m/s"
V2t=V2n*tan(Alpha2) "m/s"
n_dot=700 "rpm"
omega=2*pi*n_dot/60 "rad/s"
m_dot=rho*V_dot "kg/s"
T_shaft=m_dot*(r2*V2t-r1*V1t) "Nm"
W_dot_shaft=omega*T_shaft "W"
```

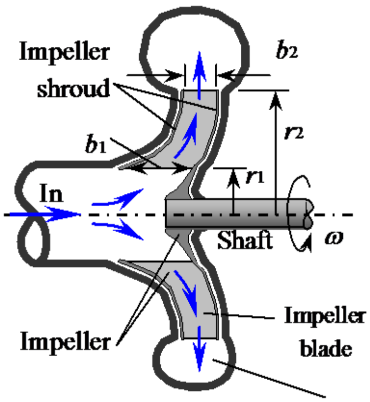


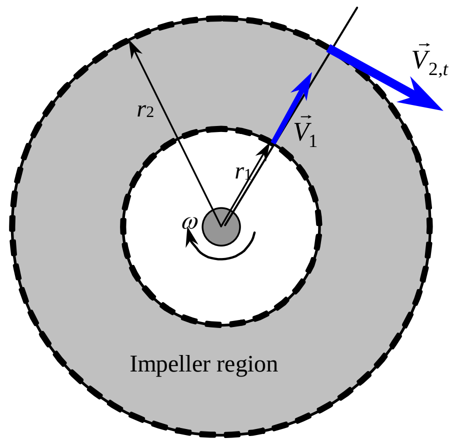
Angle, α_2°	$V_{2,t}$, m/s	Torque, T_{shaft}, Nm	Shaft power, $\dot{W}_{shaft}$, W
0	0.00	0.00	0
5	0.39	0.15	11
10	0.78	0.31	23
15	1.18	0.47	34
20	1.61	0.63	46
25	2.06	0.81	60
30	2.55	1.01	74
35	3.10	1.22	89
40	3.71	1.46	107
45	4.42	1.74	128
50	5.27	2.07	152
55	6.31	2.49	182
60	7.66	3.02	221
65	9.48	3.73	274
70	12.15	4.78	351
75	16.50	6.50	476
80	25.07	9.87	724
85	50.53	19.90	1459

Discussion When $\alpha_2 = 0$, the shaft power is also zero as expected, since there is no turning at all. As α_2 approaches 90°, the required shaft power rises rapidly towards infinity. We can never reach $\alpha_2 = 90°$ because this would mean zero flow normal to the outlet, which is impossible.

6-64E

Solution Water enters the impeller of a centrifugal pump radially at a specified flow rate and angular speed. The torque applied to the impeller is to be determined.





Assumptions **1** The flow is steady in the mean. **2** Irreversible losses are negligible.

Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis Water enters the impeller normally, and thus $V_{1,t} = 0$. The tangential component of fluid velocity at the outlet is given to be $V_{2,t} = 110$ ft/s. The inlet radius r_1 is unknown, but the outlet radius is given to be r_2 = 1 ft. The angular velocity of the propeller is

$$\omega = 2\pi\dot{n} = 2\pi(500 \text{ rev/min})\left(\frac{1\text{ min}}{60\text{ s}}\right) = 52.36 \text{ rad/s}$$

The mass flow rate is

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (62.4 \text{ lbm/ft}^3)(45/60 \text{ ft}^3/\text{s}) = 46.8 \text{ lbm/s}$$

Only the tangential velocity components contribute to torque, and the application of the angular momentum equation gives

$$\text{T}_{\text{shaft}} = \dot{m}(r_2V_{2,t} - r_1V_{1,t}) = (46.8 \text{ lbm/s})[(1 \text{ ft})(110 \text{ ft/s}) - 0]\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 159.9 \text{ lbf}\cdot\text{ft} \cong \mathbf{160\ lbf\cdot ft}$$

Discussion This shaft power input corresponding to this torque is

$$\dot{W} = 2\pi\dot{n}T_{\text{shaft}} = \omega T_{\text{shaft}} = (52.36 \text{ rad/s})(159.9 \text{ lbf}\cdot\text{ft})\left(\frac{1\text{ kW}}{737.56\text{ lbf}\cdot\text{ft/s}}\right) = \mathbf{11.3\ kW}$$

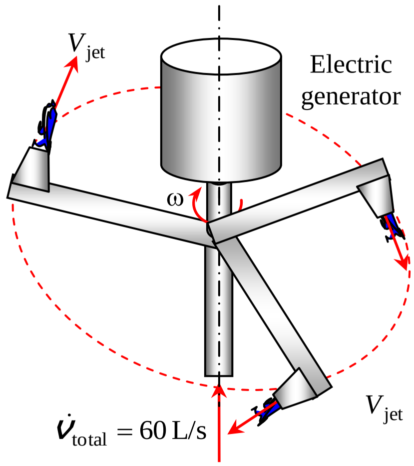
Therefore, the minimum power input to this pump should be 11.3 kW.

6-65

Solution A three-armed sprinkler is used to water a garden. For a specified flow rate and resistance torque, the angular velocity of the sprinkler head is to be determined.

Assumptions **1** The flow is uniform and cyclically steady (i.e., steady from a frame of reference rotating with the sprinkler head). **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** Air drag of rotating components are neglected. **4** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 1000 kg/m^3 = 1 kg/L.



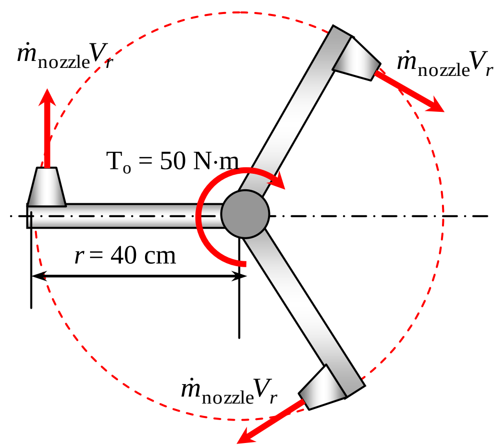
Analysis We take the disk that encloses the sprinkler arms as the control volume, which is a stationary control volume. The conservation of mass equation for this steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Noting that the three nozzles are identical, we have $\dot{m}_{\text{nozzle}} = \dot{m}/3$ or $\dot{V}_{\text{nozzle}} = \dot{V}_{total}/3$ since the density of water is constant. The average jet outlet velocity relative to the nozzle and the mass flow rate are

$$V_{\text{jet}} = \frac{\dot{V}_{\text{nozzle}}}{A_{\text{jet}}} = \frac{60\text{ L/s}}{3[\pi(0.015\text{ m})^2/4]}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right) = 113.2\text{ m/s}$$

$$\dot{m}_{\text{total}} = \rho\dot{V}_{\text{total}} = (1\text{ kg/L})(60\text{ L/s}) = 60\text{ kg/s}$$

The angular momentum equation can be expressed as

$$\sum M = \sum_{\text{out}} r\dot{m}V - \sum_{\text{in}} r\dot{m}V$$



where all moments in the counterclockwise direction are positive, and all in the clockwise direction are negative. Then the angular momentum equation about the axis of rotation becomes

$$-T_0 = -3r\dot{m}_{\text{nozzle}}V_r \qquad \text{or} \qquad T_0 = r\dot{m}_{\text{total}}V_r$$

Solving for the relative velocity V_r and substituting,

$$V_r = \frac{T_0}{r\dot{m}_{\text{total}}} = \frac{50\text{ N}\cdot\text{m}}{(0.40\text{ m})(60\text{ kg/s})}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = 2.08\text{ m/s}$$

Then the tangential and angular velocity of the nozzles become

$$V_{\text{nozzle}} = V_{\text{jet}} - V_r = 113.2 - 2.08 = 111.1\text{ m/s}$$

$$\omega = \frac{V_{\text{nozzle}}}{r} = \frac{111.1\text{ m/s}}{0.4\text{ m}} = \mathbf{278\ rad/s}$$

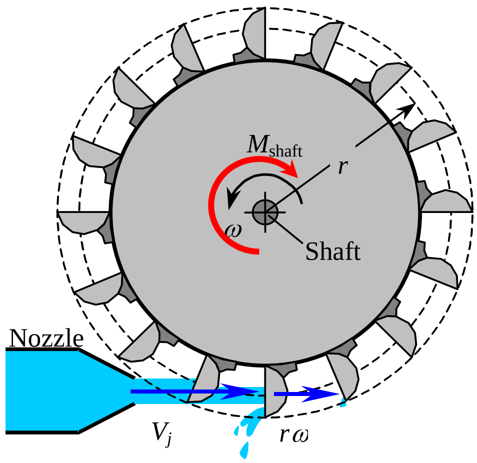
$$\dot{n} = \frac{\omega}{2\pi} = \frac{278\text{ rad/s}}{2\pi}\left(\frac{60\text{ s}}{1\text{ min}}\right) = 2652\text{ rpm} \cong \mathbf{2650\ rpm}$$

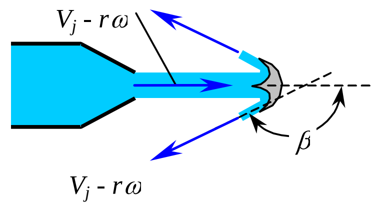
Therefore, this sprinkler will rotate at 2650 revolutions per minute (to three significant digits).

Discussion The actual rotation rate will be somewhat lower than this due to air friction as the arms rotate.

6-66

Solution A Pelton wheel is considered for power generation in a hydroelectric power plant. A relation is to be obtained for power generation, and its numerical value is to be obtained.





Assumptions **1** The flow is uniform and cyclically steady. **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** Friction and losses due to air drag of rotating components are neglected. **4** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 1000 kg/m^3 = 1 kg/L.

Analysis The tangential velocity of buckets corresponding to an angular velocity of $\omega = 2\pi\dot{n}$ is $V_{\text{bucket}} = r\omega$. Then the relative velocity of the jet (relative to the bucket) becomes

$$V_r = V_j - V_{\text{bucket}} = V_j - r\omega$$

We take the imaginary disk that contains the Pelton wheel as the control volume. The inlet velocity of the fluid into this control volume is V_r, and the component of outlet velocity normal to the moment arm is $V_r\cos\beta$. The angular momentum equation can be expressed as $\sum M = \sum_{\text{out}} r\dot{m}V - \sum_{\text{in}} r\dot{m}V$ where all moments in the counterclockwise direction are positive, and all in the clockwise direction are negative. Then the angular momentum equation about the axis of rotation becomes

$$-M_{\text{shaft}} = r\dot{m}V_r\cos\beta - r\dot{m}V_r \quad \text{or} \quad M_{\text{shaft}} = r\dot{m}V_r(1-\cos\beta) = r\dot{m}(V_j - r\omega)(1-\cos\beta)$$

Noting that $\dot{W}_{\text{shaft}} = 2\pi\dot{n}M_{\text{shaft}} = \omega M_{\text{shaft}}$ and $\dot{m} = \rho\dot{V}$, the shaft power output of a Pelton turbine becomes

$$\boxed{\dot{W}_{\text{shaft}} = \rho\dot{V}r\omega(V_j - r\omega)(1-\cos\beta)}$$

which is the desired relation. For given values, the shaft power output is determined to be

$$\dot{W}_{\text{shaft}} = (1000\text{ kg/m}^3)(10\text{ m}^3\text{/s})(2\text{ m})(15.71\text{ rad/s})(50 - 2\times 15.71\text{m/s})(1-\cos 160°)\left(\frac{1\text{MW}}{10^6\text{ N}\cdot\text{m/s}}\right) = \mathbf{11.3MW}$$

where $$\omega = 2\pi\dot{n} = 2\pi(150\text{ rev/min})\left(\frac{1\text{min}}{60\text{ s}}\right) = 15.71\text{ rad/s}$$

Discussion The actual power will be somewhat lower than this due to air drag and friction. Note that this is the *shaft* power; the electrical power generated by the generator connected to the shaft is be lower due to generator inefficiencies.

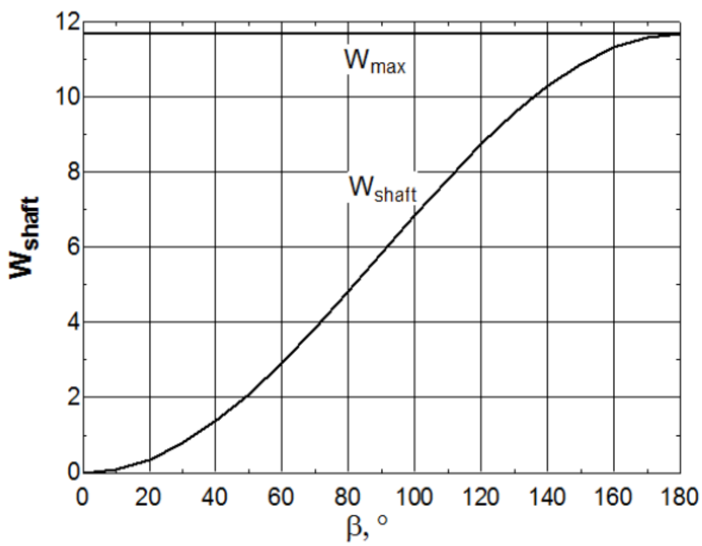
6-67

Solution The previous problem is reconsidered. The effect of β on the power generation as β varies from 0° to 180° is to be determined, and the fraction of power loss at 160° is to be assessed.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
rho=1000 "kg/m3"
r=2 "m"
V_dot=10 "m3/s"
V_jet=50 "m/s"
n_dot=150 "rpm"
omega=2*pi*n_dot/60
V_r=V_jet-r*omega
m_dot=rho*V_dot
W_dot_shaft=m_dot*omega*r*V_r*(1-cos(Beta))/1E6 "MW"
W_dot_max=m_dot*omega*r*V_r*2/1E6 "MW"
Efficiency=W_dot_shaft/W_dot_max
```

Angle, β°	Max power, $\dot{W}_{max}$, MW	Actual power, $\dot{W}_{shaft}$, MW	Efficiency, η
0	11.7	0.00	0.000
10	11.7	0.09	0.008
20	11.7	0.35	0.030
30	11.7	0.78	0.067
40	11.7	1.37	0.117
50	11.7	2.09	0.179
60	11.7	2.92	0.250
70	11.7	3.84	0.329
80	11.7	4.82	0.413
90	11.7	5.84	0.500
100	11.7	6.85	0.587
110	11.7	7.84	0.671
120	11.7	8.76	0.750
130	11.7	9.59	0.821
140	11.7	10.31	0.883
150	11.7	10.89	0.933
160	11.7	11.32	0.970
170	11.7	11.59	0.992
180	11.7	11.68	1.000



Discussion The efficiency of a Pelton wheel for β =160° is 0.97. Therefore, at this angle, only 3% of the power is lost.

Review Problems

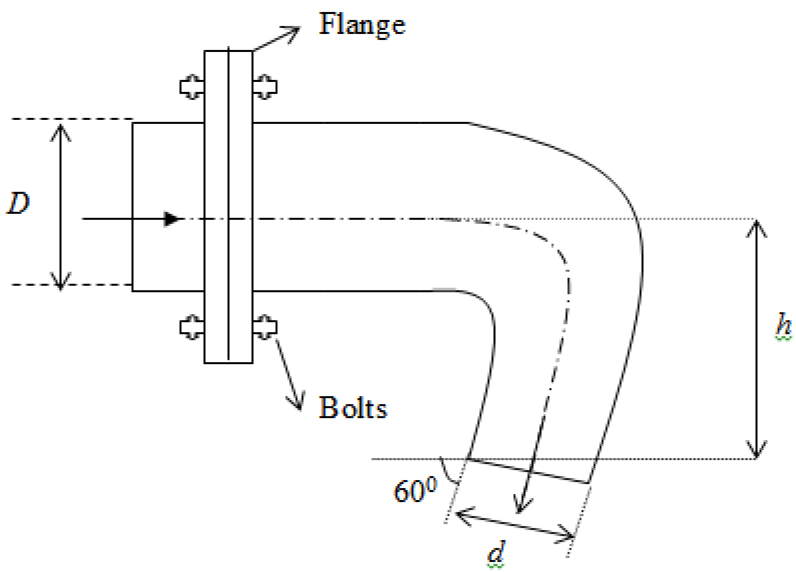
6-68

Solution Water is deflected by an elbow. The force acting on the flanges of the elbow and the angle its line of action makes with the horizontal are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Frictional effects are negligible (so that the Bernoulli equation can be used). **3** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

Properties We take the density of water to be 1000 kg/m^3.

Analysis



Writing Bernoulli equation between 1 (elbow entrance)-2 (exit) ;

$$P_1 + \frac{1}{2}\rho V_1^{\,2} + \gamma z_1 = P_2 + \frac{1}{2}\rho V_2^{\,2} + \gamma z_2 + negligible\ losses$$

$$V_1 = \frac{0.16}{\pi 0.3^2/4} = 2.26\ m/s\ ,\ V_2 = 20.37\ \ m/s\ ,\ z_1\text{=0.5m , } z_2\text{=0m ,}P_2\text{=0}$$

$$P_1 = \frac{1}{2}\rho\left(V_2^{\,2} - V_1^{\,2}\right) - \gamma z_1 = \frac{1}{2}1000\left(20.37^2 - 2.26^2\right) - 9810\times 0.5$$

$$P_1 = 20010\ Pa \cong 200\ kPa$$

Linear momentum equation for the CV gives;

$$\frac{\partial}{\partial t}\int_{cv}\vec{V}\rho dV + \int_{cs}\vec{V}\rho\vec{V}\vec{n}dA = \sum F_x$$

<u>x component</u>

$$V_1\rho(-V_1)A_1 + (-V_2\cos\theta)\ \rho(V_2)A_2 = P_1A_1 + R_x$$

$$-\rho Q V_1^2 - \rho Q V_2^2\cos\theta - P_1A_1 = R_x$$

$$-\rho Q(V_1^2 - V_2^2\cos\theta) - P_1A_1 = R_x$$

$$-1000\times 0.16\left(2.26^2 + 20.37^2\cos 60\right) - 200010\pi\frac{0.3^2}{4} = R_x$$

$$R_x = -48150 \;\; N \; (left)$$

<u>y component</u>

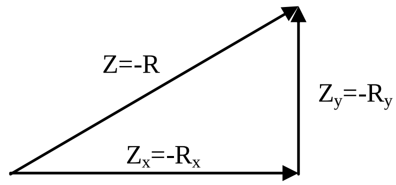
$$0 + (-V_2 \sin\theta)\rho V_2 A_2 = -W_{water,cv} + R_y$$

$$-\rho Q V_2^2 \sin\theta + W_{cv} = R_y$$

$$-1000 \times 20.37^2 \sin 60 + 9810 \times 0.03 = R_y$$

$$R_y = -359052 \;\; N \; (down)$$

These forces are exerted by elbow on water confined by CV. The force exerted by water on elbow is therefore;



$$Z = -R = \sqrt{48150^2 + 359052^2} = 362266 \text{ N} \cong \mathbf{362 kN}$$

and

$$\tan\beta = \frac{R_y}{R_x} = \frac{359{,}052 \text{ N}}{48{,}150 \text{ N}} = 7.457 \rightarrow \beta = \mathbf{82.4}°$$

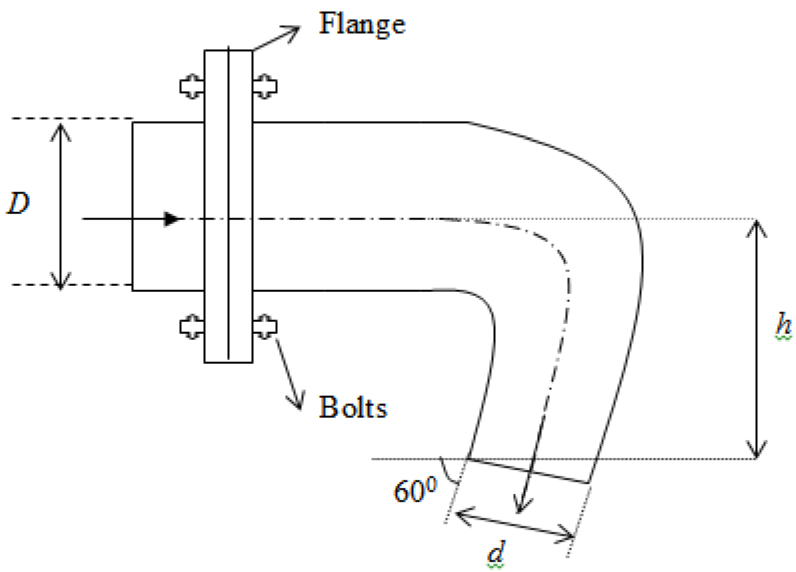
6-69

Solution Water is deflected by an elbow. The force acting on the flanges of the elbow and the angle its line of action makes with the horizontal are to be determined by taking into consideration of the weight of the elbow.

Assumptions **1** The flow is steady and incompressible. **2** Frictional effects are negligible (so that the Bernoulli equation can be used). **3** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

Properties We take the density of water to be 1000 kg/m^3.

Analysis



Writing Bernoulli equation between 1 (elbow entrance)-2 (exit) ;

$$P_1 + \frac{1}{2}\rho V_1^{\,2} + \gamma z_1 = P_2 + \frac{1}{2}\rho V_2^{\,2} + \gamma z_2 + \textit{negligible losses}$$

$$V_1 = \frac{0.16}{\pi 0.3^2/4} = 2.26\ m/s\ ,\ V_2 = 20.37\ \ m/s\ ,\ z_1\text{=0.5m},\ z_2\text{=0m},\ P_2\text{=0}$$

$$P_1 = \frac{1}{2}\rho\left(V_2^{\,2} - V_1^{\,2}\right) - \gamma z_1 = \frac{1}{2}1000\left(20.37^2 - 2.26^2\right) - 9810 \times 0.5$$

$$P_1 = 20010\ Pa \cong 200\ kPa$$

Linear momentum equation for the CV gives;

$$\frac{\partial}{\partial t}\int_{cv} \vec{V}\rho dV + \int_{cs} \vec{V}\rho\vec{V}\vec{n}dA = \sum F_x$$

<u>x component</u>

$$V_1\rho(-V_1)A_1 + (-V_2\cos\theta)\ \rho(V_2)A_2 = P_1A_1 + R_x$$

$$-\rho QV_1^2 - \rho QV_2^2\cos\theta - P_1A_1 = R_x$$

$$-\rho Q(V_1^2 - V_2^2\cos\theta) - P_1A_1 = R_x$$

$$-1000 \times 0.16\left(2.26^2 + 20.37^2\cos 60\right) - 200010\pi\frac{0.3^2}{4} = R_x$$

$$R_x = -48150\ \ N\ (left)$$

To include elbow weight we must modify y-momentum equation as follows:

<u>y component:</u>

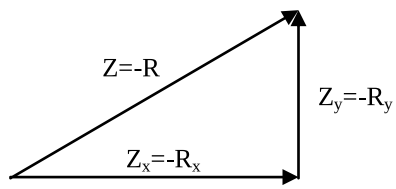
$$0+(-V_2\sin\theta)\rho V_2 A_2 = -W_{water,cv} - W_{elbow} + R_y$$

$$-\rho Q V_2^2 \sin\theta + W_{water,cv} + W_{elbow} = R_y$$

$$-1000\times 20.37^2 \sin 60 + 9810\times 0.03 + 5\times 9.81 = R_y$$

$$R_y \approx -359003 \ \ N \ (down)$$

These forces are exerted by elbow on water confined by CV. The force exerted by water on elbow is therefore;



$$Z=-R=\sqrt{48150^2+359052^2}=362266 \ \text{N} \cong \mathbf{362 kN}$$

and

$$\tan\beta = \frac{R_y}{R_x} = \frac{359{,}003 \text{ N}}{48{,}150 \text{ N}} = 7.457 \rightarrow \beta = \mathbf{82.4}°$$

Therefore we could neglect the weight of the elbow.

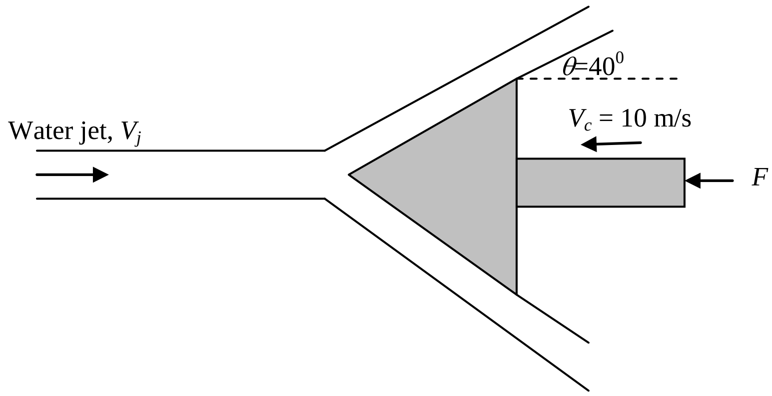
6-70

Solution A horizontal water jet is deflected by a cone. The external force needed to maintain the motion of the cone is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The flow is uniform ine each section.

Properties We take the density of water to be 1000 kg/m^3.

Analysis



We chose a CV moving with V_c to left. W_1 and W_2 are the relative velocities with respect to the CV. Conservation of mass gives

$$0=\frac{\partial}{\partial t}\int_{cv}\rho dV+\int_{cs}\rho.\left(\overbrace{\vec{V}-\vec{V}_{cv}}^{\vec{W}}\right)\vec{n}dA \text{ ....1}$$

Conservation of momentum gives

$$F=\frac{\partial}{\partial t}\int_{cv}W\rho dV+\int_{cs}\vec{W}\rho\vec{W}\vec{n}dA \text{ ....2}$$

From Eq. 1

$$-W_1A_1+W_2A_2=0$$

$W_2A_2=W_1A_1=(V_j+V_c)A_j$, therefore $A_1=A_2=A_j$

From eq.2

$$F_x=-\int_{A_1}W_1\rho.(-W_1).dA_1+\int_{A_2}(W_2Cos\theta)\rho(+W_2)dA_2$$

$$W_1=W_2=V_j+V_c=25+10=35\ m/s$$

Then, $F_x=-\rho.W_1.A_j+\rho.W_2{}^2.A_jCos\theta$

$$= (Cos\theta-1).\rho.W_2{}^2A_j$$

$$=(Cos40-1)\times1000\times35^2\frac{\pi}{4}0.12^2$$

F_x = -3241 N

Therefore,

F_x = **3241 N** to left

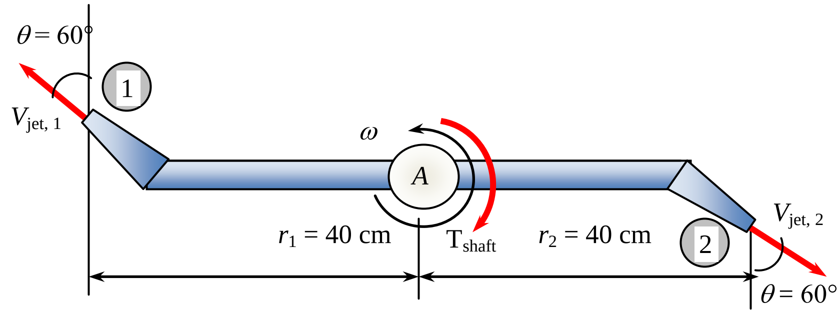
6-71

Solution Water enters a two-armed sprinkler vertically, and leaves the nozzles horizontally at an angle to tangential direction. For a specified flow rate and discharge angle, the rate of rotation of the sprinkler and the torque required to prevent the sprinkler from rotating are to be determined. √EES

Assumptions **1** The flow is uniform and cyclically steady (i.e., steady from a frame of reference rotating with the sprinkler head). **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** Frictional effects and air drag of rotating components are neglected. **4** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 1000 kg/m^3 = 1 kg/L.

Analysis We take the disk that encloses the sprinkler arms as the control volume, which is a stationary control volume. The conservation of mass equation for this symmetrical steady flow system is $\dot m_1 = \dot m_2 = \dot m$. Noting that the two nozzles are identical, we have $\dot m_{\text{jet}} = \dot m_{\text{total}}/2$ or $\dot V_{\text{jet}} = \dot V_{\text{total}}/2$ since the density of water is constant. The average jet outlet velocity relative to the nozzle is



$$V_{\text{jet},r} = \frac{\dot V_{\text{jet}}}{A_{\text{jet}}} = \frac{10/2\ \text{L/s}}{\pi(0.012\ \text{m})^2/4}\left(\frac{1\ \text{m}^3}{1000\ \text{L}}\right) = 44.21\ \text{m/s}$$

The angular momentum equation about the axis of rotation can be expressed as $\sum M = \sum_{\text{out}} r\dot m V - \sum_{\text{in}} r\dot m V$ where r is the average moment arm, V is the average absolute speed (relative to an inertial reference frame), all moments in the counterclockwise direction are positive, and all moments in the clockwise direction are negative. Momentum flows in the clockwise direction (as in this case) are also negative. The absolute water jet speed in the tangential direction is the difference between the tangential component of the water jet speed and the nozzle speed $V_{\text{nozzle}} = \omega r = 2\pi\dot n r$. Thus, $V_{\text{jet},t} = V_{\text{jet},r,\text{t}} - V_{\text{nozzle}} = V_{\text{jet},r}\cos\theta - \omega r$. Then the angular momentum equation becomes

$$-\text{T}_{\text{shaft}} = -r_1\dot m_{\text{jet},1}V_{\text{jet},t,1} - r_2\dot m_{\text{jet},2}V_{\text{jet},t,2} \quad \to \quad \text{T}_{\text{shaft}} = r_1\dot m_{\text{jet},1}(V_{\text{jet},r,1}\cos\theta_1 - \omega r_1) + r_2\dot m_{\text{jet},2}(V_{\text{jet},r,2}\cos\theta_2 - \omega r_2)$$

Noting that $r_1 = r_2 = r$, $\theta_1 = \theta_2 = \theta$, $V_{\text{jet},r,1} = V_{\text{jet},r,2} = V_{\text{jet},r}$, and, the angular momentum equation becomes

$$\text{T}_{\text{shaft}} = r\rho\dot V_{\text{total}}(V_{\text{jet},r}\cos\theta - \omega r)$$

(*a*) In the case of free spin with no frictional effects, we have $\text{T}_{\text{shaft}} = 0$ and thus $V_{\text{jet},r}\cos\theta - \omega r = 0$. Then angular speed and the rate of rotation of sprinkler head becomes

$$\omega = \frac{V_{\text{jet, r}}\cos\theta}{r} = \frac{(44.21\ \text{m/s})\cos 60°}{0.40\ \text{m}} = 55.26\ \text{rad/s} \quad \text{and} \quad \dot n = \frac{\omega}{2\pi} = \frac{55.26\ \text{rad/s}}{2\pi}\left(\frac{60\ \text{s}}{1\ \text{min}}\right) = \mathbf{528\ rpm}$$

(*b*) When the sprinkler is prevented from rotating, we have $\omega = 0$. Then the required torque becomes

$$\text{T}_{\text{shaft}} = r\rho\dot V_{\text{total}}V_{\text{jet},r}\cos\theta = (0.4\ \text{m})(1\ \text{kg/L})(10\ \text{L/s})(44.21\ \text{m/s})\cos 60°\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{88.4\ N\cdot m}$$

Discussion The rate of rotation determined in (*a*) will be lower in reality because of frictional effects and air drag.

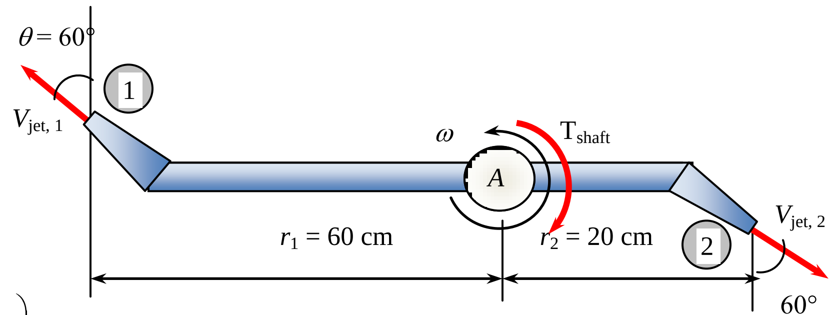
6-72

Solution Water enters a two-armed sprinkler vertically, and leaves the nozzles horizontally at an angle to tangential direction. For a specified flow rate and discharge angle, the rate of rotation of the sprinkler and the torque required to prevent the sprinkler from rotating are to be determined.

Assumptions **1** The flow is uniform and cyclically steady (i.e., steady from a frame of reference rotating with the sprinkler head). **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** Frictional effects and air drag of rotating components are neglected. **4** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 1000 kg/m^3 = 1 kg/L.

Analysis We take the disk that encloses the sprinkler arms as the control volume, which is a stationary control volume. The conservation of mass equation for this symmetrical steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Noting that the two nozzles are identical, we have $\dot{m}_{\text{jet}} = \dot{m}_{\text{total}}/2$ or $\dot{V}_{\text{jet}} = \dot{V}_{\text{total}}/2$ since the density of water is constant. The average jet outlet velocity relative to the nozzle is



$$V_{\text{jet},r} = \frac{\dot{V}_{\text{jet}}}{A_{\text{jet}}} = \frac{10/2\text{ L/s}}{\pi(0.012\text{ m})^2/4}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right) = 44.21\text{ m/s}$$

The angular momentum equation about the axis of rotation can be expressed as $\sum M = \sum_{\text{out}} r\dot{m}V - \sum_{\text{in}} r\dot{m}V$ where r is the average moment arm, V is the average absolute speed (relative to an inertial reference frame), all moments in the counterclockwise direction are positive, and all moments in the clockwise direction are negative. Momentum flows in the clockwise direction (as in this case) are also negative. The absolute water jet speed in the tangential direction is the difference between the tangential component of the water jet speed and the nozzle speed $V_{\text{nozzle}} = \omega r = 2\pi \dot{n} r$. Thus, $V_{\text{jet},t} = V_{\text{jet},r,\text{t}} - V_{\text{nozzle}} = V_{\text{jet},r}\cos\theta - \omega r$. Then the angular momentum equation becomes

$$-\text{T}_{\text{shaft}} = -r_1\dot{m}_{\text{jet},1}V_{\text{jet},t,1} - r_2\dot{m}_{\text{jet},2}V_{\text{jet},t,2} \quad \rightarrow \quad \text{T}_{\text{shaft}} = r_1\dot{m}_{\text{jet},1}(V_{\text{jet},r,1}\cos\theta_1 - \omega r_1) + r_2\dot{m}_{\text{jet},2}(V_{\text{jet},r,2}\cos\theta_2 - \omega r_2)$$

Noting that $\theta_1 = \theta_2 = \theta$, $V_{\text{jet},r,1} = V_{\text{jet},r,2} = V_{\text{jet},r}$, and, the angular momentum equation becomes

$$\text{T}_{\text{shaft}} = \rho\dot{V}_{\text{jet}}[r_1(V_{\text{jet},r}\cos\theta - \omega r_1) + r_2(V_{\text{jet},r}\cos\theta - \omega r_2)] \quad \text{or} \quad \text{T}_{\text{shaft}} = \rho\dot{V}_{\text{jet}}[(r_1 + r_2)V_{\text{jet},r}\cos\theta - (r_1^2 + r_2^2)\omega]$$

(*a*) In the case of free spin with no frictional effects, we have T$_{\text{shaft}}$ = 0 and thus $0 = (r_1 + r_2)V_{\text{jet},r}\cos\theta - (r_1^2 + r_2^2)\omega$. Then angular speed and the rate of rotation of sprinkler head becomes

$$\omega = \frac{(r_1 + r_2))V_{\text{jet, r}}\cos\theta}{r_1^2 + r_2^2} = \frac{(0.6 + 0.2\text{m})(44.21\text{ m/s})\cos 60°}{(0.60\text{ m})^2 + (0.20\text{ m})^2} = 44.21\text{ rad/s}$$

and

$$\dot{n} = \frac{\omega}{2\pi} = \frac{44.21\text{ rad/s}}{2\pi}\left(\frac{60\text{ s}}{1\text{ min}}\right) = 422.2\text{ rpm} \cong \mathbf{422\ rpm}$$

(*b*) When the sprinkler is prevented from rotating, we have ω = 0. Then the required torque becomes

$$\text{T}_{\text{shaft}} = (r_1 + r_2)\rho\dot{V}_{\text{jet}}V_{\text{jet},r}\cos\theta = (0.6 + 0.2\text{ m})(1\text{ kg/L})(10/2\text{ L/s})(44.21\text{ m/s})\cos 60°\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{88.4\ N \cdot m}$$

Discussion The rate of rotation determined in (*a*) will be lower in reality because of frictional effects and air drag.

6-73

Solution A horizontal water jet strikes a vertical stationary flat plate normally at a specified velocity. For a given flow velocity, the anchoring force needed to hold the plate in place is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water splatters off the sides of the plate in a plane normal to the jet. **3** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water is the atmospheric pressure which is disregarded since it acts on the entire control surface. **4** The vertical forces and momentum fluxes are not considered since they have no effect on the horizontal reaction force. **5** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

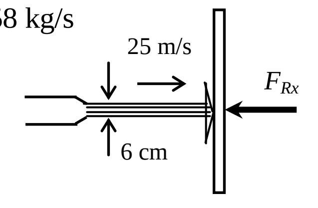
Properties We take the density of water to be 1000 kg/m^3.

Analysis We take the plate as the control volume such that it contains the entire plate and cuts through the water jet and the support bar normally, and the direction of flow as the positive direction of *x* axis. We take the reaction force to be in the negative *x* direction. The momentum equation for steady flow in the *x* (flow) direction reduces in this case to

$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V} \quad \rightarrow \quad -F_{Rx} = -\beta_i \dot{m}_i V_i \quad \rightarrow \quad F_{Rx} = \beta_i \dot{m} V$$

We note that the reaction force acts in the opposite direction to flow, and we should not forget the negative sign for forces and velocities in the negative *x*-direction. The mass flow rate of water is

$$\dot{m} = \rho \dot{\mathbf{V}} = \rho A V = \rho \frac{\pi D^2}{4} V = (1000 \text{ kg/m}^3) \frac{\pi (0.06 \text{ m})^2}{4} (25 \text{ m/s}) = 70.6858 \text{ kg/s}$$



Substituting, the reaction force is determined to be

$$F_{Rx} = (1)(70.6858 \text{ kg/s})(25 \text{ m/s}) = 1767 \text{ N} \cong \mathbf{1770\ N}$$

Therefore, a force of approximately 1770 N must be applied to the plate in the opposite direction to the flow to hold it in place.

Discussion In reality, some water may be scattered back, and this would add to the reaction force of water. If we do not approximate the water jet as uniform, the momentum flux correction factor β would factor in. For example, if β = 1.03 (approximate value for fully developed pipe flow), the force would *increase* by 3%. This is because the actual nonuniform jet has *more* momentum than the uniform jet.

6-74

Solution Steady developing laminar flow is considered in a constant horizontal diameter discharge pipe. A relation is to be obtained for the horizontal force acting on the bolts that hold the pipe.

Assumptions **1** The flow is steady, laminar, and incompressible. **2** The flow is fully developed at the end of the pipe section considered. **3** The velocity profile at the pipe inlet is uniform and thus the momentum-flux correction factor is β_1 = 1. **4** The momentum-flux correction factor is β = 2 at the outlet.

Analysis We take the developing flow section of the pipe (including the water inside) as the control volume. We assume the reaction force to act in the positive direction. Noting that the flow is incompressible and thus the average velocity is constant $V_1 = V_2 = V$, the momentum equation for steady flow in the z (flow) direction in this case reduces to

$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m} \vec{V} - \sum_{\text{in}} \beta \dot{m} \vec{V}$$

$$-F_R + P_1 A_c - P_2 A_c = \dot{m}\beta_2 V - \dot{m}\beta_1 V$$

$$\begin{aligned} F_R &= (P_1 - P_2)A_c + \dot{m}V(\beta_1 - \beta_2) \\ &= (P_1 - P_2)A_c + \dot{m}V(1-2) \\ &= (P_1 - P_2)\pi D^2/4 - \dot{m}V \end{aligned}$$

Or, using the definition of the mass flow rate,

$$F_R = (P_1 - P_2)\pi D^2/4 - [\rho\pi D^2/4]V^2$$

Or,

$$\boxed{F_R = \frac{\pi D^2}{4}\left[(P_1 - P_2) - \rho V^2\right]}$$

Discussion Note that the cause of this reaction force is non-uniform velocity profile at the end.

6-75

Solution A fireman was hit by a nozzle held by a tripod with a rated holding force. The accident is to be investigated by calculating the water velocity, the flow rate, and the nozzle velocity.

Assumptions **1** The flow is steady and incompressible. **2** The water jet is exposed to the atmosphere, and thus the pressure of the water jet is the atmospheric pressure, which is disregarded since it acts on all surfaces. **3** Gravitational effects and vertical forces are disregarded since the horizontal resistance force is to be determined. **4** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$. **5** Upstream pressure and momentum effects are ignored.

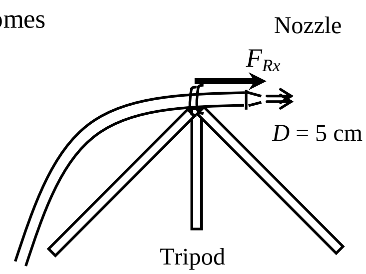
Properties We take the density of water to be 1000 kg/m^3.

Analysis We take the nozzle and the horizontal portion of the hose as the system such that water enters the control volume vertically and outlets horizontally (this way the pressure force and the momentum flux at the inlet are in the vertical direction, with no contribution to the force balance in the horizontal direction, and designate the entrance by 1 and the outlet by 2. We also designate the horizontal coordinate by x (with the direction of flow as being the positive direction).

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V}$. We let the horizontal force applied by the tripod to the nozzle to hold it be F_{Rx}, and assume it to be in the positive x direction. Then the momentum equation along the x direction becomes

$$F_{Rx} = \dot{m}V_e - 0 = \dot{m}V = \rho AVV = \rho\frac{\pi D^2}{4}V^2 \quad \rightarrow \quad (1800\text{ N})\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = (1000\text{ kg/m}^3)\frac{\pi(0.05\text{ m})^2}{4}V^2$$

Solving for the water outlet velocity gives V = **30.3 m/s.** Then the water flow rate becomes



$$\dot{V} = AV = \frac{\pi D^2}{4}V = \frac{\pi(0.05\text{ m})^2}{4}(30.3\text{ m/s}) = \mathbf{0.0595\ m^3/s}$$

When the nozzle was released, its acceleration must have been

$$a_{\text{nozzle}} = \frac{F}{m_{\text{nozzle}}} = \frac{1800\text{ N}}{10\text{ kg}}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = 180\text{ m/s}^2$$

Assuming the reaction force acting on the nozzle and thus its acceleration to remain constant, the time it takes for the nozzle to travel 60 cm and the nozzle velocity at that moment were (note that both the distance x and the velocity V are zero at time t = 0)

$$x = \tfrac{1}{2}at^2 \quad \rightarrow \quad t = \sqrt{\frac{2x}{a}} = \sqrt{\frac{2(0.6\text{ m})}{180\text{ m/s}^2}} = 0.0816\ s$$

$$V = at = (180\text{ m/s}^2)(0.0816\text{ s}) = \mathbf{14.7\ m/s}$$

Thus we conclude that the nozzle hit the fireman with a velocity of 14.7 m/s.

Discussion Engineering analyses such as this one are frequently used in accident reconstruction cases, and they often form the basis for judgment in courts.

6-76

Solution During landing of an airplane, the thrust reverser is lowered in the path of the exhaust jet, which deflects the exhaust and provides braking. The thrust of the engine and the braking force produced after the thrust reverser is deployed are to be determined.

Assumptions **1** The flow of exhaust gases is steady and one-dimensional. **2** The exhaust gas stream is exposed to the atmosphere, and thus its pressure is the atmospheric pressure. **3** The velocity of exhaust gases remains constant during reversing. **4** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Analysis (*a*) The thrust exerted on an airplane is simply the momentum flux of the combustion gases in the reverse direction,

$$\text{Thrust} = \dot{m}_{ex}V_{ex} = (18\text{ kg/s})(300\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{5400N}$$

(*b*) We take the thrust reverser as the control volume such that it cuts through both exhaust streams normally and the connecting bars to the airplane, and the direction of airplane as the positive direction of *x* axis. The momentum equation for steady flow in the *x* direction reduces to

$$\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V} \quad \rightarrow \quad F_{Rx} = \dot{m}(V)\cos 20° - \dot{m}(-V) \quad \rightarrow \quad F_{Rx} = (1+\cos 20°)\dot{m}V_i$$

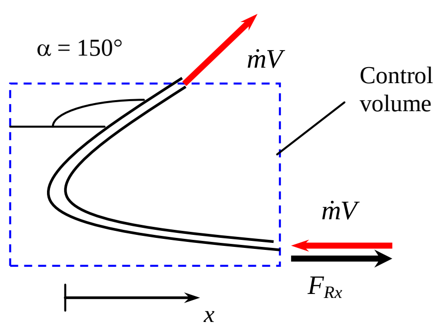
Substituting, the reaction force is determined to be

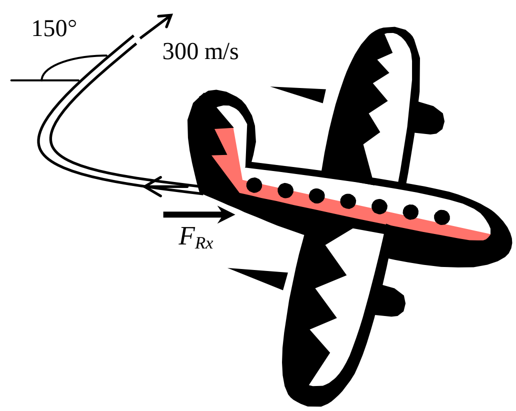
$$F_{Rx} = (1+\cos 30°)(18\text{ kg/s})(300\text{ m/s}) = 10{,}077\text{ N}$$

The breaking force acting on the plane is equal and opposite to this force,

$$F_{\text{breaking}} = 10{,}077\text{ N} \cong \mathbf{10{,}100\ N}$$

Therefore, a braking force of 10,100 N develops in the opposite direction to flight.





Discussion This problem can be solved more generally by measuring the reversing angle from the direction of exhaust gases (α = 0 when there is no reversing). When α < 90°, the reversed gases are discharged in the negative *x* direction, and the momentum equation reduces to

$$F_{Rx} = \dot{m}(-V)\cos\alpha - \dot{m}(-V) \quad \rightarrow \quad F_{Rx} = (1-\cos\alpha)\dot{m}V_i$$

This equation is also valid for α >90° since cos(180°-α) = - cosα. Using α = 150°, for example, gives
$F_{Rx} = (1-\cos 150)\dot{m}V_i = (1+\cos 30)\dot{m}V_i$, which is identical to the solution above.

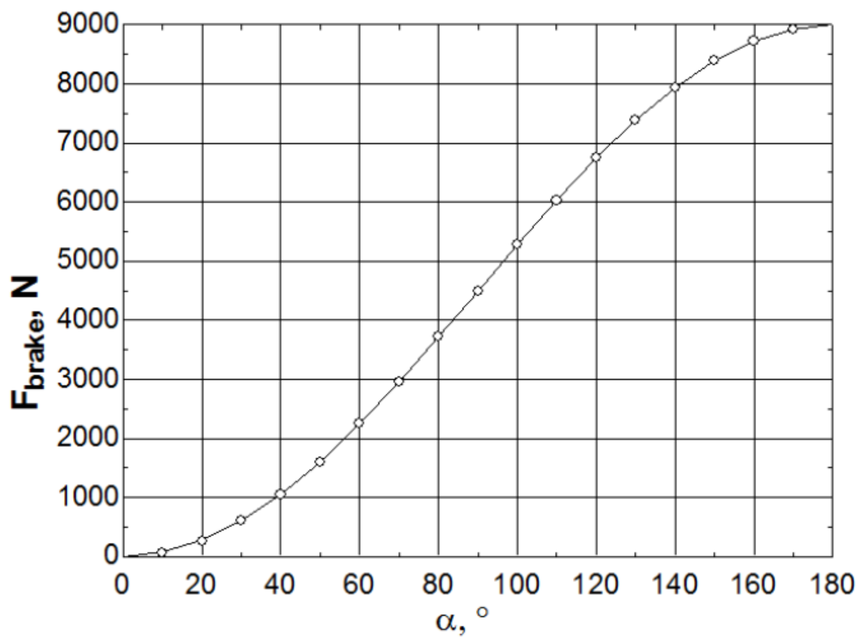
6-77

Solution The previous problem is reconsidered. The effect of thrust reverser angle on the braking force exerted on the airplane as the reverser angle varies from 0 (no reversing) to 180° (full reversing) in increments of 10° is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
V_jet=250 "m/s"
m_dot=18 "kg/s"
F_Rx=(1-cos(alpha))*m_dot*V_jet "N"
```

Reversing angle, α°	Braking force F_{brake}, N
0	0
10	68
20	271
30	603
40	1053
50	1607
60	2250
70	2961
80	3719
90	4500
100	5281
110	6039
120	6750
130	7393
140	7947
150	8397
160	8729
170	8932
180	9000



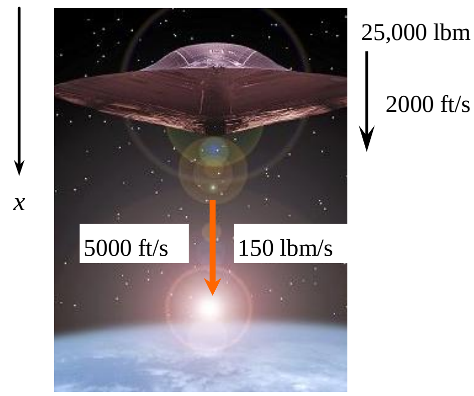
Discussion As expected, the braking force is zero when the angle is zero (no deflection), and maximum when the angle is 180° (completely reversed). Of course, it is impossible to completely reverse the flow, since the jet exhaust cannot be directed back into the engine.

6-78E

Solution The rocket of a spacecraft is fired in the opposite direction to motion. The deceleration, the velocity change, and the thrust are to be determined.

Assumptions **1** The flow of combustion gases is steady and one-dimensional during the firing period, but the flight of the spacecraft is unsteady. **2** There are no external forces acting on the spacecraft, and the effect of pressure force at the nozzle outlet is negligible. **3** The mass of discharged fuel is negligible relative to the mass of the spacecraft, and thus the spacecraft may be treated as a solid body with a constant mass. **4** The nozzle is well-designed such that the effect of the momentum-flux correction factor is negligible, and thus $\beta \cong 1$.

Analysis (*a*) We choose a reference frame in which the control volume moves with the spacecraft. Then the velocities of fluid steams become simply their relative velocities (relative to the moving body). We take the direction of motion of the spacecraft as the positive direction along the *x* axis. There are no external forces acting on the spacecraft, and its mass is nearly constant. Therefore, the spacecraft can be treated as a solid body with constant mass, and the momentum equation in this case is



$$0 = \frac{d(m\vec{V})_{CV}}{dt} + \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V} \quad \rightarrow \quad m_{\text{space}}\frac{d\vec{V}_{\text{space}}}{dt} = -\dot{m}_f\vec{V}_f$$

Noting that the motion is on a straight line and the discharged gases move in the positive *x* direction (to slow down the spacecraft), we write the momentum equation using magnitudes as

$$m_{\text{space}}\frac{dV_{\text{space}}}{dt} = -\dot{m}_f V_f \quad \rightarrow \quad \frac{dV_{\text{space}}}{dt} = -\frac{\dot{m}_f}{m_{\text{space}}}V_f$$

Substituting, the deceleration of the spacecraft during the first 5 seconds is determined to be

$$a_{\text{space}} = \frac{dV_{\text{space}}}{dt} = -\frac{\dot{m}_f}{m_{\text{space}}}V_f = -\frac{150\ \text{lbm/s}}{25{,}000\ \text{lbm}}(5000\ \text{ft/s}) = \mathbf{-30.0\ ft/s^2}$$

(*b*) Knowing the deceleration, which is constant, the velocity change of the spacecraft during the first 5 seconds is determined from the definition of acceleration $a_{\text{space}} = dV_{\text{space}}/dt$ to be

$$dV_{\text{space}} = a_{\text{space}}dt \quad \rightarrow \quad \Delta V_{\text{space}} = a_{\text{space}}\Delta t = (-30.0\ \text{ft/s}^2)(5\ \text{s}) = \mathbf{-150\ ft/s}$$

(*c*) The thrust exerted on the system is simply the momentum flux of the combustion gases in the reverse direction,

$$\text{Thrust} = F_R = -\dot{m}_f V_f = -(150\ \text{lbm/s})(5000\ \text{ft/s})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = -23{,}290\ \text{lbf} \cong \mathbf{-23{,}300\ lbf}$$

Therefore, if this spacecraft were attached somewhere, it would exert a force of 23,300 lbf (equivalent to the weight of 23,300 lbm of mass on earth) to its support in the negative *x* direction.

Discussion In Part (*b*) we approximate the deceleration as constant. However, since mass is lost from the spacecraft during the time in which the jet is on, a more accurate solution would involve solving a differential equation. Here, the time span is short, and the lost mass is likely negligible compared to the total mass of the spacecraft, so the more complicated analysis is not necessary.

6-79

Solution An ice skater is holding a flexible hose (essentially weightless) which directs a stream of water horizontally at a specified velocity. The velocity and the distance traveled in 5 seconds, and the time it takes to move 5 m and the velocity at that moment are to be determined.

Assumptions **1** Friction between the skates and ice is negligible. **2** The flow of water is steady and one-dimensional (but the motion of skater is unsteady). **3** The ice skating arena is level, and the water jet is discharged horizontally. **4** The mass of the hose and the water in it is negligible. **5** The skater is standing still initially at $t = 0$. **6** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis (*a*) The mass flow rate of water through the hose is

$$\dot{m} = \rho A V = \rho \frac{\pi D^2}{4} V = (1000\ \text{kg/m}^3)\frac{\pi (0.02\ \text{m})^2}{4}(10\ \text{m/s}) = 3.14\ \text{kg/s}$$

The thrust exerted on the skater by the water stream is simply the momentum flux of the water stream, and it acts in the reverse direction,

$$F = \text{Thrust} = \dot{m}V = (3.14\ \text{kg/s})(10\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = 31.4\ \text{ N (constant)}$$

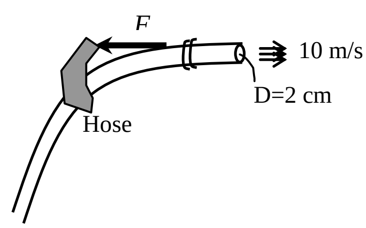
The acceleration of the skater is determined from Newton's 2$^{\text{nd}}$ law of motion $F = ma$ where m is the mass of the skater,

$$a = \frac{F}{m} = \frac{31.4\ \text{N}}{60\ \text{kg}}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 0.523\ \text{m/s}^2$$

Note that thrust and thus the acceleration of the skater is constant. The velocity of the skater and the distance traveled in 5 s are

$$V_{\text{skater}} = at = (0.523\ \text{m/s}^2)(5\ \text{s}) = \mathbf{2.62\ m/s}$$

$$x = \tfrac{1}{2}at^2 = \tfrac{1}{2}(0.523\ \text{m/s}^2)(5\ \text{s})^2 = \mathbf{6.54\ m}$$



(*b*) The time it will take to move 5 m and the velocity at that moment are

$$x = \tfrac{1}{2}at^2 \quad \rightarrow \quad t = \sqrt{\frac{2x}{a}} = \sqrt{\frac{2(5\ \text{m})}{0.523\ \text{m/s}^2}} = \mathbf{4.4\ s}$$

$$V_{\text{skater}} = at = (0.523\ \text{m/s}^2)(4.4\ \text{s}) = \mathbf{2.3\ m/s}$$

Discussion In reality, the velocity of the skater will be lower because of friction on ice and the resistance of the hose to follow the skater. Also, in the $\beta\dot{m}V$ expressions, V is the fluid stream speed relative to a fixed point. Therefore, the correct expression for thrust is $F = \dot{m}(V_{jet} - V_{skater})$, and the analysis above is valid only when the skater speed is low relative to the jet speed. An exact analysis would result in a differential equation.

6-80

Solution A water jet hits a stationary cone, such that the flow is diverted equally in all directions at 45°. The force required to hold the cone in place against the water stream is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water jet is exposed to the atmosphere, and thus the pressure of the water jet before and after the split is the atmospheric pressure which is disregarded since it acts on all surfaces. **3** The gravitational effects are disregarded. **4** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis The mass flow rate of water jet is

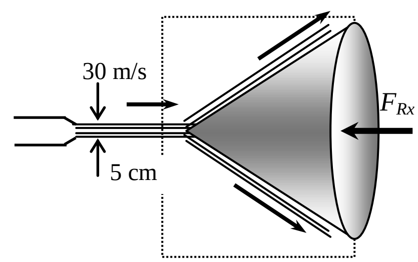
$$\dot{m} = \rho\dot{V} = \rho AV = \rho\frac{\pi D^2}{4}V = (1000\ \text{kg/m}^3)\frac{\pi(0.05\ \text{m})^2}{4}(30\ \text{m/s}) = 58.90\ \text{kg/s}$$

We take the diverting section of water jet, including the cone as the control volume, and designate the entrance by 1 and the outlet after divergence by 2. We also designate the horizontal coordinate by x with the direction of flow as being the positive direction and the vertical coordinate by y.

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. We let the x- and y- components of the anchoring force of the cone be F_{Rx} and F_{Ry}, and assume them to be in the positive directions. Noting that $V_2 = V_1 = V$ and $\dot{m}_2 = \dot{m}_1 = \dot{m}$, the momentum equations along the x and y axes become

$$F_{Rx} = \dot{m}V_2\cos\theta - \dot{m}V_1 = \dot{m}V(\cos\theta - 1)$$

$$F_{Ry} = 0 \quad \text{(because of symmetry about x axis)}$$



Substituting the given values,

$$F_{Rx} = (58.90\ \text{kg/s})(30\ \text{m/s})(\cos 45° - 1)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= \mathbf{-518\ N}$$

$$F_{Ry} = \mathbf{0}$$

The negative value for F_{Rx} indicates that the assumed direction is wrong, and should be reversed. Therefore, a force of 518 N must be applied to the cone in the opposite direction to flow to hold it in place. No holding force is necessary in the vertical direction due to symmetry and neglecting gravitational effects.

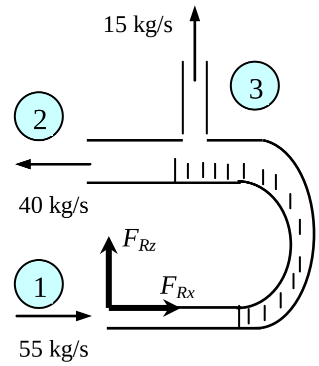
Discussion In reality, the gravitational effects will cause the upper part of flow to slow down and the lower part to speed up after the split. But for short distances, these effects are negligible.

6-81

Solution Water is flowing into and discharging from a pipe U-section with a secondary discharge section normal to return flow. Net *x*- and *z*- forces at the two flanges that connect the pipes are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The weight of the U-turn and the water in it is negligible. **4** The momentum-flux correction factor for each inlet and outlet is given to be $\beta = 1.03$.

Properties We take the density of water to be 1000 kg/m^3.



Analysis The flow velocities of the 3 streams are

$$V_1 = \frac{\dot{m}_1}{\rho A_1} = \frac{\dot{m}_1}{\rho(\pi D_1^2/4)} = \frac{55\text{ kg/s}}{(1000\text{ kg/m}^3)[\pi(0.05\text{ m})^2/4]} = 28.01\text{ m/s}$$

$$V_2 = \frac{\dot{m}_2}{\rho A_2} = \frac{\dot{m}_2}{\rho(\pi D_2^2/4)} = \frac{40\text{ kg/s}}{(1000\text{ kg/m}^3)[\pi(0.10\text{ m})^2/4]} = 5.093\text{ m/s}$$

$$V_3 = \frac{\dot{m}_3}{\rho A_3} = \frac{\dot{m}_3}{\rho(\pi D_3^2/4)} = \frac{15\text{ kg/s}}{(1000\text{ kg/m}^3)[\pi(0.03\text{ m})^2/4]} = 21.22\text{ m/s}$$

We take the entire U-section as the control volume. We designate the horizontal coordinate by *x* with the direction of incoming flow as being the positive direction and the vertical coordinate by *z*. The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V}$. We let the *x*- and *z*- components of the anchoring force of the cone be F_{Rx} and F_{Rz}, and assume them to be in the positive directions. Then the momentum equations along the *x* and *z* axes become

$$F_{Rx} + P_1A_1 + P_2A_2 = \beta\dot{m}_2(-V_2) - \beta\dot{m}_1V_1 \quad\rightarrow\quad F_{Rx} = -P_1A_1 - P_2A_2 - \beta(\dot{m}_2V_2 + \dot{m}_1V_1)$$

$$F_{Rz} + 0 = \dot{m}_3V_3 - 0 \quad\rightarrow\quad F_{Rz} = \beta\dot{m}_3V_3$$

Substituting the given values,

$$\begin{aligned} F_{Rx} &= -[(200-100)\text{ kN/m}^2]\frac{\pi(0.05\text{ m})^2}{4} - [(150-100)\text{ kN/m}^2]\frac{\pi(0.10\text{ m})^2}{4} \\ &\quad -1.03\left[(40\text{ kg/s})(5.093\text{ m/s})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) + (55\text{ kg/s})(28.01\text{ m/s})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\right] \\ &= -2.386\text{ kN} \cong \mathbf{-2390 N} \end{aligned}$$

$$F_{Rz} = 1.03(15\text{ kg/s})(21.22\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 327.8\text{ N} \cong \mathbf{328 N}$$

The negative value for F_{Rx} indicates the assumed direction is wrong, and should be reversed. Therefore, a force of 2390 N acts on the flanges in the opposite direction. A vertical force of 328 N acts on the flange in the vertical direction.

Discussion To assess the significance of gravity forces, we estimate the weight of the weight of water in the U-turn and compare it to the vertical force. Assuming the length of the U-turn to be 0.5 m and the average diameter to be 7.5 cm, the mass of the water becomes

$$m = \rho \boldsymbol{V} = \rho AL = \rho\frac{\pi D^2}{4}L = (1000\text{ kg/m}^3)\frac{\pi(0.075\text{ m})^2}{4}(0.5\text{ m}) = 2.2\text{ kg}$$

whose weight is 2.2×9.81 = 22 N, which is much less than 328, but still significant. Therefore, disregarding the gravitational effects is a reasonable assumption if great accuracy is not required.

6-82

Solution Indiana Jones is to ascend a building by building a platform, and mounting four water nozzles pointing down at each corner. The minimum water jet velocity needed to raise the system, the time it will take to rise to the top of the building and the velocity of the system at that moment, the additional rise when the water is shut off, and the time he has to jump from the platform to the roof are to be determined.

Assumptions **1** The air resistance is negligible. **2** The flow of water is steady and one-dimensional (but the motion of platform is unsteady). **3** The platform is still initially at t = 0. **4** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties We take the density of water to be 1000 kg/m^3.

Analysis (a) The total mass flow rate of water through the 4 hoses and the total weight of the platform are

$$\dot{m} = \rho AV = 4\rho\frac{\pi D^2}{4}V = 4(1000\ \text{kg/m}^3)\frac{\pi(0.04\ \text{m})^2}{4}(18\ \text{m/s}) = 90.4779\ \text{kg/s}$$

$$W = mg = (150\ \text{kg})(9.81\ \text{m/s}^2)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = 1471.5\ \text{N}$$

We take the platform as the system. The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V}$. The minimum water jet velocity needed to raise the platform is determined by setting the net force acting on the platform equal to zero,

$$-W = \dot{m}(-V_{\min}) - 0 \quad \rightarrow \quad W = \dot{m}V_{\min} = \rho AV_{\min}V_{\min} = 4\rho\frac{\pi D^2}{4}V_{\min}^2$$

Solving for $V_{\min}$ and substituting,

$$V_{\min} = \sqrt{\frac{W}{\rho\pi D^2}} = \sqrt{\frac{1471.5\ \text{N}}{(1000\ \text{kg/m}^3)\pi(0.04\ \text{m})^2}\left(\frac{1\text{kg}\cdot\text{m/s}^2}{1\text{N}}\right)} = 17.1098\ \text{m/s} \cong \mathbf{17.1\ m/s}$$

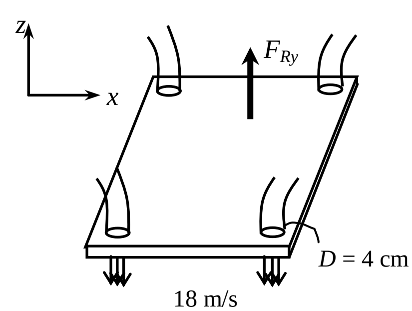
(b) We let the vertical reaction force (assumed upwards) acting on the platform be F_{Rz}. Then the momentum equation in the vertical direction becomes

$$F_{Rz} - W = \dot{m}(-V) - 0 = \dot{m}V \quad \rightarrow \quad F_{Rz} = W - \dot{m}V = (1471.5\ \text{N}) - (90.4779\frac{\text{kg}}{\text{s}})(18\frac{\text{m}}{\text{s}})\left(\frac{1\text{kg}\cdot\text{m/s}^2}{1\text{N}}\right) = -157.101\ \text{N}$$

The upward thrust acting on the platform is equal and opposite to this reaction force, and thus F = 156.6 N. Then the acceleration and the ascending time to rise 10 m and the velocity at that moment become

$$a = \frac{F}{m} = \frac{157.101\ \text{N}}{150\ \text{kg}}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 1.0473\ \text{m/s}^2$$

$$x = \tfrac{1}{2}at^2 \quad \rightarrow \quad t = \sqrt{\frac{2x}{a}} = \sqrt{\frac{2(10\ \text{m})}{1.0473\ \text{m/s}^2}} = 4.36989\ \text{s} \cong \mathbf{4.37\ \ s}$$



and

$$V = at = (1.0473\ \text{m/s}^2)(4.36989\ \text{s}) = 4.5766\ \text{m/s}$$

(c) When the water is shut off at 10 m height (where the velocity is 4.57 m/s), the platform will decelerate under the influence of gravity, and the time it takes to come to a stop and the additional rise above 10 m become

$$V = V_0 - gt = 0 \quad \rightarrow \quad t = \frac{V_0}{g} = \frac{4.5766\ \text{m/s}}{9.81\ \text{m/s}^2} = 0.46652\ \text{s}$$

$$z = V_0t - \tfrac{1}{2}gt^2 = (4.5766\ \text{m/s})(0.46652\ \text{s}) - \tfrac{1}{2}(9.81\ \text{m/s}^2)(0.46652\ \text{s})^2 = 1.0675\ \text{m} \cong \mathbf{1.07\ m}$$

Therefore, Jones has 2×0.46652 = 0.93304 ≈ **0.933 s** to jump off from the platform to the roof since it takes another 0.466 s for the platform to descend to the 10 m level.

Discussion Like most stunts in the Indiana Jones movies, this would not be practical in reality.

6-83E

Solution A box-enclosed fan is faced down so the air blast is directed downwards, and it is to be hovered by increasing the blade rpm. The required blade rpm, air outlet velocity, the volumetric flow rate, and the minimum mechanical power are to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** The air leaves the blades at a uniform velocity at atmospheric pressure, and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$. **3** Air approaches the blades from the top through a large area at atmospheric pressure with negligible velocity. **4** The frictional effects are negligible, and thus the entire mechanical power input is converted to kinetic energy of air (no conversion to thermal energy through frictional effects). **5** The change in air pressure with elevation is negligible because of the low density of air. **6** There is no acceleration of the fan, and thus the lift generated is equal to the total weight.

Properties The density of air is given to be 0.078 lbm/ft^3.

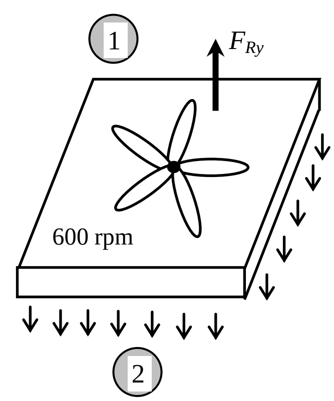
Analysis (*a*) We take the control volume to be a vertical hyperbolic cylinder bounded by streamlines on the sides with air entering through the large cross-section (section 1) at the top and the fan located at the narrow cross-section at the bottom (section 2), and let its centerline be the *z* axis with upwards being the positive direction.

The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V}$. Noting that the only force acting on the control volume is the total weight *W* and it acts in the negative *z* direction, the momentum equation along the *z* axis gives

$$-W = \dot{m}(-V_2) - 0 \quad \rightarrow \quad W = \dot{m}V_2 = (\rho A V_2)V_2 = \rho A V_2^2 \quad \rightarrow \quad V_2 = \sqrt{\frac{W}{\rho A}}$$

where *A* is the blade span area,

$$A = \pi D^2/4 = \pi(3\text{ ft})^2/4 = 7.069\text{ ft}^2$$



Then the discharge velocity to produce 5 lbf of upward force becomes

$$V_2 = \sqrt{\frac{5\text{ lbf}}{(0.078\text{ lbm/ft}^3)(7.069\text{ ft}^2)}\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)} = \mathbf{17.1\ ft/s}$$

(*b*) The volume flow rate and the mass flow rate of air are determined from their definitions,

$$\dot{V} = AV_2 = (7.069\text{ ft}^2)(17.1\text{ ft/s}) = \mathbf{121\ ft^3/s}$$

$$\dot{m} = \rho\dot{V} = (0.078\text{ lbm/ft}^3)(121\text{ ft}^3\text{/s}) = 9.43\text{ lbm/s}$$

(*c*) Noting that $P_1 = P_2 = P_{atm}$, $V_1 \cong 0$, the elevation effects are negligible, and the frictional effects are disregarded, the energy equation for the selected control volume reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump,u}} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech,loss}} \quad \rightarrow \quad \dot{W}_{\text{fan, u}} = \dot{m}\frac{V_2^2}{2}$$

Substituting,

$$\dot{W}_{\text{fan,u}} = \dot{m}\frac{V_2^2}{2} = (9.43\text{ lbm/s})\frac{(18.0\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\text{ W}}{0.73756\text{ lbf}\cdot\text{ft/s}}\right) = \mathbf{64.3\ W}$$

Therefore, the minimum mechanical power that must be supplied to the air stream is 64.3 W.

Discussion The actual power input to the fan will be considerably larger than the calculated power input because of the fan inefficiency in converting mechanical work to kinetic energy.

6-84

Solution A plate is maintained in a horizontal position by frictionless vertical guide rails. The underside of the plate is subjected to a water jet. The minimum mass flow rate $\dot{m}_{\text{min}}$ to just levitate the plate is to be determined, and a relation is to be obtained for the steady state upward velocity. Also, the integral that relates velocity to time when the water is first turned on is to be obtained.

Assumptions **1** The flow of water is steady and one-dimensional. **2** The water jet splatters in the plane of he plate. **3** The vertical guide rails are frictionless. **4** Times are short, so the velocity of the rising jet can be considered to remain constant with height. **5** At time t = 0, the plate is at rest. **6** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Analysis (*a*) We take the plate as the system. The momentum equation for steady flow is $\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V}$. Noting that $\dot{m} = \rho A V_J$ where *A* is the cross-sectional area of the water jet and $W = m_p g$, the minimum mass flow rate of water needed to raise the plate is determined by setting the net force acting on the plate equal to zero,

$$-W = 0 - \dot{m}_{\text{min}} V_{\text{J}} \quad \rightarrow \quad W = \dot{m}_{\text{min}} V_{\text{J}} \;\rightarrow\; m_p g = \dot{m}_{\text{min}} (\dot{m}_{\text{min}} / AV_{\text{J}}) \rightarrow \quad \dot{m}_{\text{min}} = \sqrt{\rho A m_p g}$$

For $\dot{m} > \dot{m}_{\text{min}}$, a relation for the steady state upward velocity *V* is obtained setting the upward impulse applied by water jet to the weight of the plate (during steady motion, the plate velocity *V* is constant, and the velocity of water jet relative to plate is $V_{\text{J}} - V$),

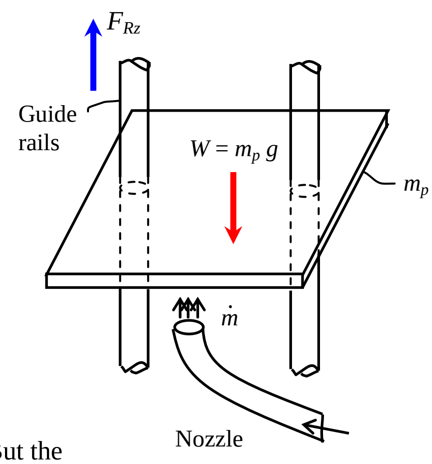
$$W = \dot{m}(V_J - V) \quad \rightarrow \quad m_p g = \rho A (V_J - V)^2 \quad \rightarrow \; V_J - V = \sqrt{\frac{m_p g}{\rho A}} \quad \rightarrow \; V = \frac{\dot{m}}{\rho A} - \sqrt{\frac{m_p g}{\rho A}}$$

(*b*) At time $t = 0$ the plate is at rest ($V = 0$), and it is subjected to water jet with $\dot{m} > \dot{m}_{\text{min}}$ and thus the net force acting on it is greater than the weight of the plate, and the difference between the jet impulse and the weight will accelerate the plate upwards. Therefore, Newton's 2nd law $F = ma = mdV/dt$ in this case can be expressed as

$$\dot{m}(V_J - V) - W = m_p a \quad \rightarrow \quad \rho A (V_J - V)^2 - m_p g = m_p \frac{dV}{dt}$$

Separating the variables and integrating from $t = 0$ when $V = 0$ to $t = t$ when $V = V$ gives the desired integral,

$$\int_0^V \frac{m_p dV}{\rho A (V_J - V)^2 - m_p g} = \int_{t=0}^{t} dt \quad \rightarrow \quad \boxed{t = \int_0^V \frac{m_p dV}{\rho A (V_J - V)^2 - m_p g}}$$



Discussion This integral can be performed with the help of integral tables. But the relation obtained will be implicit in *V*.

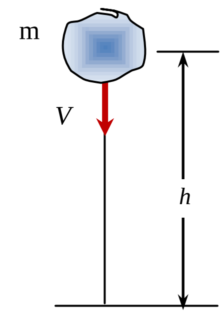
6-85

Solution A walnut is to be cracked by dropping it from a certain height to a hard surface. The minimum height required is to be determined.

Assumptions **1** The force remains constant during the cracking period of the walnut. **2** The air resistance is negligible.

Analysis We take the *x* axis as the upward vertical direction. Newton's 2nd law $F = ma = mdV/dt$ can be expressed as

$$F\Delta t = m\Delta V = m(V_{\text{strike}} - V_{\text{final}}) \quad \rightarrow \quad V_{\text{strike}} = \frac{F\Delta t}{m}$$



since the force remains constant and the final velocity is zero. Substituting,

$$V_{\text{strike}} = \frac{F\Delta t}{m} = \frac{(200\text{ N})(0.002\text{ s})}{0.050\text{ kg}}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = 8\text{ m/s}$$

The elevation that will result at this value of velocity can be determined from the conservation of energy principle (in this case potential energy being converted to kinetic energy) to be

$$pe_{\text{initial}} = ke_{\text{final}} \quad \rightarrow \quad mgh = \tfrac{1}{2}mV_{\text{strike}}^2 \quad \rightarrow \quad h = \frac{V_{\text{strike}}^2}{2g}$$

Substituting, the required height at which the walnut needs to be dropped becomes

$$h = \frac{V_{\text{strike}}^2}{2g} = \frac{(8\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = \mathbf{3.26\ m}$$

Discussion Note that a greater height will be required in reality because of air friction.

6-86

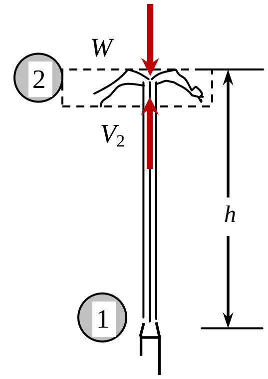
Solution A vertical water jet strikes a horizontal stationary plate normally. The maximum weight of the plate that can be supported by the water jet at a specified height is to be determined.

Assumptions **1** The flow of water at the nozzle outlet is steady and incompressible. **2** The water splatters in directions normal to the approach direction of the water jet. **3** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water leaving the control volume is atmospheric pressure . **4** Friction between the water and air is negligible. **5** The effect of the momentum-flux correction factor is negligible, and thus $\beta \cong 1$ for the jet.

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take the x axis as the upward vertical direction. We also take point 1 at the point where the water jet leaves the nozzle, and point 2 at the point where the jet strikes the flat plate. Noting that water jet is exposed to the atmosphere, we have $P_1 = P_2 = P_{atm}$, Also, $z_1 = 0$ and $z_2 = h$. Then the Bernoulli Equation simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad \rightarrow \quad 0+\frac{V_1^2}{2g}+0=0+\frac{V_2^2}{2g}+h \quad \rightarrow \quad V_2=\sqrt{V_1^2-2gh}$$



Substituting, the jet velocity when the jet strikes the flat plate is determined to be

$$V_2=\sqrt{(15\text{ m/s})^2-2(9.81\text{ m/s}^2)(2\text{ m})}=13.63\text{ m/s}$$

The mass flow rate of water is

$$\dot{m}=\rho\dot{V}=\rho A_c V=\rho\frac{\pi D^2}{4}V=(1000\text{ kg/m}^3)\frac{\pi(0.07\text{ m})^2}{4}(15\text{ m/s})=57.73\text{ kg/s}$$

We take the thin region below the flat plate as the control volume such that it cuts through the incoming water jet. The weight W of the flat plate acts downward as a vertical force on the CV. Noting that water jet splashes out horizontally after it strikes the plate, the momentum equation for steady flow in the x (flow) direction reduces to

$$\sum\vec{F}=\sum_{\text{out}}\beta\dot{m}\vec{V}-\sum_{\text{in}}\beta\dot{m}\vec{V} \quad \rightarrow \quad -W=-\dot{m}V_2 \quad \rightarrow \quad W=\dot{m}V_2$$

Substituting, the weight of the flat plate is determined to be

$$W=(57.73\text{ kg/s})(13.36\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)=\mathbf{771N}$$

Discussion Note that this weight corresponds to a plate mass of 771/9.81 = 78.5 kg of mass. Also, a smaller mass will be held in balance at a greater height and a larger mass at a smaller height.

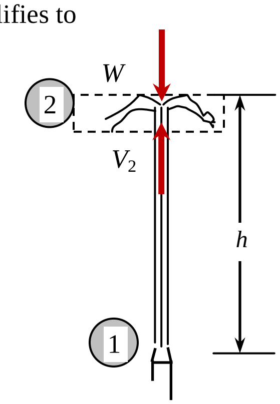
6-87

Solution A vertical water jet strikes a horizontal stationary plate normally. The maximum weight of the plate that can be supported by the water jet at a specified height is to be determined.

Assumptions **1** The flow of water at the nozzle outlet is steady and incompressible. **2** The water splatters in directions normal to the approach direction of the water jet. **3** The water jet is exposed to the atmosphere, and thus the pressure of the water jet and the splattered water leaving the control volume is atmospheric pressure . **4** Friction between the water and air is negligible. **5** The effect of the momentum-flux correction factor is negligible, and thus $\beta \cong 1$ for the jet.

Properties We take the density of water to be 1000 kg/m^3.

Analysis We take the x axis as the upward vertical direction. We also take point 1 at the point where the water jet leaves the nozzle, and point 2 at the point where the jet strikes the flat plate. Noting that water jet is exposed to the atmosphere, we have $P_1 = P_2 = P_{atm}$, Also, $z_1 = 0$ and $z_2 = h$. Then the Bernoulli Equation simplifies to



$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad 0+\frac{V_1^2}{2g}+0=0+\frac{V_2^2}{2g}+h \quad\rightarrow\quad V_2=\sqrt{V_1^2-2gh}$$

Substituting, the jet velocity when the jet strikes the flat plate is determined to be

$$V_2=\sqrt{(15\text{ m/s})^2-2(9.81\text{ m/s}^2)(8\text{ m})}=8.249\text{ m/s}$$

The mass flow rate of water is

$$\dot m=\rho\dot V=\rho A_c V=\rho\frac{\pi D^2}{4}V=(1000\text{ kg/m}^3)\frac{\pi(0.07\text{ m})^2}{4}(15\text{ m/s})=57.73\text{ kg/s}$$

We take the thin region below the flat plate as the control volume such that it cuts through the incoming water jet. The weight W of the flat plate acts downward as a vertical force on the CV. Noting that water jet splashes out horizontally after it strikes the plate, the momentum equation for steady flow in the x (flow) direction reduces to

$$\sum\vec F=\sum_{\text{out}}\beta\dot m\vec V-\sum_{\text{in}}\beta\dot m\vec V \quad\rightarrow\quad -W=-\dot m V_2 \quad\rightarrow\quad W=\dot m V_2$$

Substituting, the weight of the flat plate is determined to be

$$W=(57.73\text{ kg/s})(8.249\text{ m/s})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)=\mathbf{476\ N}$$

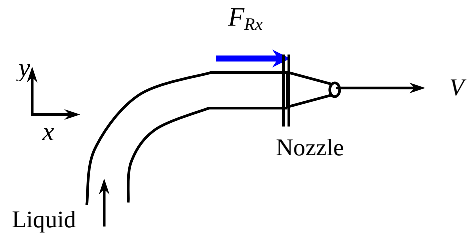
Discussion Note that this weight corresponds to a plate mass of 476/9.81 = 48.5 kg of mass. Also, a smaller mass will be held in balance at a greater height and a larger mass at a smaller height.

6-88

Solution It is to be shown that the force exerted by a liquid jet of velocity V on a stationary nozzle is proportional to V^2, or alternatively, to $\dot{m}^2$.

Assumptions **1** The flow is steady and incompressible. **2** The nozzle is given to be stationary. **3** The nozzle involves a 90° turn and thus the incoming and outgoing flow streams are normal to each other. **4** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

Analysis We take the nozzle as the control volume, and the flow direction at the outlet as the x axis. Note that the nozzle makes a 90° turn, and thus it does not contribute to any pressure force or momentum flux term at the inlet in the x direction. Noting that $\dot{m} = \rho AV$ where A is the nozzle outlet area and V is the average nozzle outlet velocity, the momentum equation for steady flow in the x direction reduces to



$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m}\vec{V} - \sum_{\text{in}} \beta \dot{m}\vec{V} \quad \rightarrow \quad F_{Rx} = \beta \dot{m}_{out} V_{out} = \beta \dot{m} V$$

where F_{Rx} is the reaction force on the nozzle due to liquid jet at the nozzle outlet. Then,

$$\dot{m} = \rho AV \quad \rightarrow \quad F_{Rx} = \beta \dot{m} V = \beta \rho AVV = \beta \rho AV^2 \quad \text{or} \quad \boxed{F_{Rx} = \beta \dot{m} V = \beta \dot{m} \frac{\dot{m}}{\rho A} = \beta \frac{\dot{m}^2}{\rho A}}$$

Therefore, **the force exerted by a liquid jet of velocity *V* on this stationary nozzle is proportional to V^2, or alternatively, to $\dot{m}^2$.**

Discussion If there were not a 90° turn, we would need to take into account the momentum flux and pressure contributions at the inlet.

6-89

Solution A parachute slows a soldier from his terminal velocity V_T to his landing velocity of V_F. A relation is to be developed for the soldier's velocity after he opens the parachute at time $t = 0$.

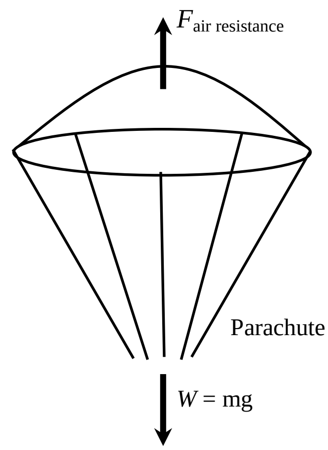
Assumptions **1** The air resistance is proportional to the velocity squared (i.e. $F = -kV^2$). **2** The variation of the air properties with altitude is negligible. **3** The buoyancy force applied by air to the person (and the parachute) is negligible because of the small volume occupied and the low density of air. **4** The final velocity of the soldier is equal to its terminal velocity with his parachute open.

Analysis The terminal velocity of a free falling object is reached when the air resistance (or air drag) equals the weight of the object, less the buoyancy force applied by the fluid, which is negligible in this case,

$$F_{\text{air resistance}} = W \quad \rightarrow \quad kV_F^2 = mg \quad \rightarrow \quad k = \frac{mg}{V_F^2}$$

This is the desired relation for the constant of proportionality k. When the parachute is deployed and the soldier starts to decelerate, the net downward force acting on him is his weight less the air resistance,

$$F_{\text{net}} = W - F_{\text{air resistance}} = mg - kV^2 = mg - \frac{mg}{V_F^2}V^2 = mg\left(1 - \frac{V^2}{V_F^2}\right)$$



Substituting it into Newton's 2nd law relation $F_{\text{net}} = ma = m\dfrac{dV}{dt}$ gives

$$mg\left(1 - \frac{V^2}{V_F^2}\right) = m\frac{dV}{dt}$$

Canceling m and separating variables, and integrating from $t = 0$ when $V = V_T$ to $t = t$ when $V = V$ gives

$$\frac{dV}{1 - V^2/V_F^2} = g dt \quad \rightarrow \quad \int_{V_T}^{V} \frac{dV}{V_F^2 - V^2} = \frac{g}{V_F^2}\int_0^t dt$$

Using $\displaystyle\int \frac{dx}{a^2 - x^2} = \frac{1}{2a}\ln\frac{a+x}{a-x}$ from integral tables and applying the integration limits,

$$\frac{1}{2V_F}\left(\ln\frac{V_F + V}{V_F - V} - \ln\frac{V_F + V_T}{V_F - V_T}\right) = \frac{gt}{V_F^2}$$

Rearranging, the velocity can be expressed explicitly as a function of time as

$$\boxed{V = V_F \frac{V_T + V_F + (V_T - V_F)e^{-2gt/V_F}}{V_T + V_F - (V_T - V_F)e^{-2gt/V_F}}}$$

Discussion Note that as $t \rightarrow \infty$, the velocity approaches the landing velocity of V_F, as expected.

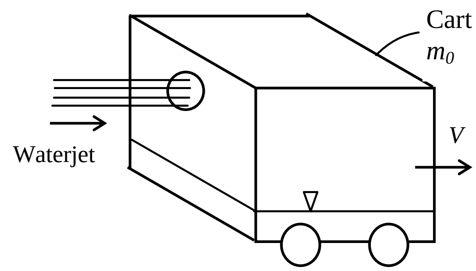
6-90

Solution An empty cart is to be driven by a horizontal water jet that enters from a hole at the rear of the cart. A relation is to be developed for cart velocity versus time.

Assumptions **1** The flow of water is steady, one-dimensional, incompressible, and horizontal. **2** All the water which enters the cart is retained. **3** The path of the cart is level and frictionless. **4** The cart is initially empty and stationary, and thus $V = 0$ at time $t = 0$. **5** Friction between water jet and air is negligible, and the entire momentum of water jet is used to drive the cart with no losses. **6** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Analysis We note that the water jet velocity V_J is constant, but the car velocity V is variable. Noting that $\dot{m} = \rho A(V_J - V)$ where A is the cross-sectional area of the water jet and V_J - V is the velocity of the water jet relative to the cart, the mass of water in the cart at any time t is

$$m_w = \int_0^t \dot{m}dt = \int_0^t \rho A(V_J - V)dt = \rho AV_J t - \rho A\int_0^t Vdt \quad (1)$$



Also,

$$\frac{dm_w}{dt} = \dot{m} = \rho A(V_J - V)$$

We take the cart as the moving control volume. The net force acting on the cart in this case is equal to the momentum flux of the water jet. Newton's 2nd law $F = ma = d(mV)/dt$ in this case can be expressed as

$$F = \frac{d(m_{\text{total}}V)}{dt} \quad \text{where} \quad F = \sum_{\text{in}} \beta\dot{m}V - \sum_{\text{out}} \beta\dot{m}V = (\dot{m}V)_{in} = \dot{m}V_J = \rho A(V_J - V)V_J$$

and

$$\frac{d(m_{\text{total}}V)}{dt} = \frac{d[(m_c + m_w)V]}{dt} = m_c\frac{dV}{dt} + \frac{d(m_w V)}{dt} = m_c\frac{dV}{dt} + m_w\frac{dV}{dt} + V\frac{dm_w}{dt}$$

$$= (m_c + m_w)\frac{dV}{dt} + \rho A(V_J - V)V$$

Note that in $\beta\dot{m}V$ expressions, we used the fluid stream velocity relative to a fixed point. Substituting,

$$\rho A(V_J - V)V_J = (m_c + m_w)\frac{dV}{dt} + \rho A(V_J - V)V \quad \rightarrow \quad \rho A(V_J - V)(V_J - V) = (m_c + m_w)\frac{dV}{dt}$$

Noting that m_w is a function of t (as given by Eq. 1) and separating variables,

$$\frac{dV}{\rho A(V_J - V)^2} = \frac{dt}{m_c + m_w} \quad \rightarrow \quad \frac{dV}{\rho A(V_J - V)^2} = \frac{dt}{m_c + \rho AV_J t - \rho A\int_0^t Vdt}$$

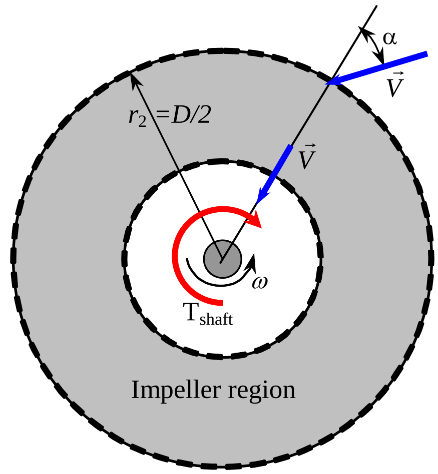
Integrating from $t = 0$ when $V = 0$ to $t = t$ when $V = V$ gives the desired integral,

$$\boxed{\int_0^V \frac{dV}{\rho A(V_J - V)^2} = \int_o^t \frac{dt}{m_c + \rho AV_J t - \rho A\int_0^t Vdt}}$$

Discussion Note that the time integral involves the integral of velocity, which complicates the solution.

6-91

Solution Water enters the impeller of a turbine through its outer edge of diameter D with velocity V making an angle α with the radial direction at a mass flow rate of $\dot{m}$, and leaves the impeller in the radial direction. The maximum power that can be generated is to be shown to be $\dot{W}_{\text{shaft}} = \pi \dot{n} \dot{m} DV \sin\alpha$.



Assumptions **1** The flow is steady in the mean. **2** Irreversible losses are negligible.

Analysis We take the impeller region as the control volume. The tangential velocity components at the inlet and the outlet are $V_{1,t} = 0$ and $V_{2,t} = V\sin\alpha$.

Normal velocity components as well pressure acting on the inner and outer circumferential areas pass through the shaft center, and thus they do not contribute to torque. Only the tangential velocity components contribute to torque, and the application of the angular momentum equation gives

$$\mathrm{T}_{\text{shaft}} = \dot{m}(r_2 V_{2,t} - r_1 V_{1,t}) = \dot{m} r_2 V_{2,t} - 0 = \dot{m} D(V\sin\alpha)/2$$

The angular velocity of the propeller is $\omega = 2\pi\dot{n}$. Then the shaft power becomes

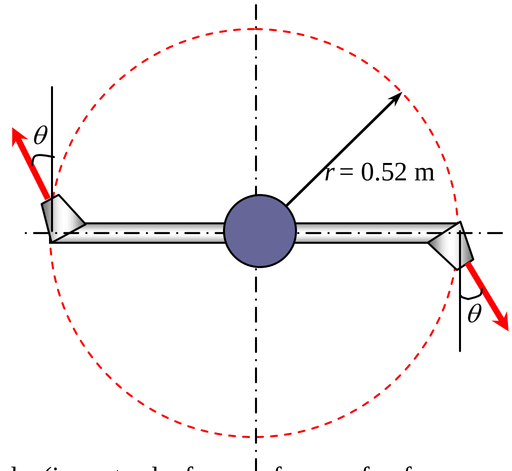
$$\dot{W}_{\text{shaft}} = \omega \mathrm{T}_{\text{shaft}} = 2\pi\dot{n}\dot{m} D(V\sin\alpha)/2$$

Simplifying, the maximum power generated becomes $\boxed{\dot{W}_{\text{shaft}} = \pi\dot{n}\dot{m} DV\sin\alpha}$ which is the desired relation.

Discussion The actual power is less than this due to irreversible losses that are not taken into account in our analysis.

6-92

Solution A two-armed sprinkler is used to water a garden. For specified flow rate and discharge angles, the rates of rotation of the sprinkler head are to be determined.



Assumptions **1** The flow is uniform and cyclically steady (i.e., steady from a frame of reference rotating with the sprinkler head). **2** The water is discharged to the atmosphere, and thus the gage pressure at the nozzle outlet is zero. **3** Frictional effects and air drag of rotating components are neglected. **4** The nozzle diameter is small compared to the moment arm, and thus we use average values of radius and velocity at the outlet.

Properties We take the density of water to be 1000 kg/m^3 = 1 kg/L.

Analysis We take the disk that encloses the sprinkler arms as the control volume, which is a stationary control volume. The conservation of mass equation for this steady flow system is $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Noting that the two nozzles are identical, we have $\dot{m}_{\text{nozzle}} = \dot{m}/2$ or $\dot{V}_{\text{nozzle}} = \dot{V}_{total}/2$ since the density of water is constant. The average jet outlet velocity relative to the nozzle is

$$V_{\text{jet}} = \frac{\dot{V}_{\text{nozzle}}}{A_{\text{jet}}} = \frac{75\text{ L/s}}{2[\pi(0.02\text{ m})^2/4]}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right) = 119.4\text{ m/s}$$

The angular momentum equation can be expressed as $\sum M = \sum_{\text{out}} r\dot{m}V - \sum_{\text{in}} r\dot{m}V$. Noting that there are no external moments acting, the angular momentum equation about the axis of rotation becomes

$$0 = -2r\dot{m}_{\text{nozzle}}V_r \cos\theta \quad \rightarrow \quad V_r = 0 \quad \rightarrow \quad V_{\text{jet,t}} - V_{\text{nozzle}} = 0$$

Noting that the tangential component of jet velocity is $V_{\text{jet,t}} = V_{\text{jet}}\cos\theta$, we have

$$V_{\text{nozzle}} = V_{\text{jet}}\cos\theta = (119.4\text{ m/s})\cos\theta$$

Also noting that $V_{\text{nozzle}} = \omega r = 2\pi\dot{n}r$, and angular speed and the rate of rotation of sprinkler head become

1) θ = 0°: $\omega = \frac{V_{\text{nozzle}}}{r} = \frac{(119.4\text{ m/s})\cos 0}{0.52\text{ m}} = \mathbf{230\ rad/s}$ and $\dot{n} = \frac{\omega}{2\pi} = \frac{230\text{ rad/s}}{2\pi}\left(\frac{60\text{ s}}{1\text{ min}}\right) = \mathbf{2193\ rpm}$

2) θ = 30°: $\omega = \frac{V_{\text{nozzle}}}{r} = \frac{(119.4\text{ m/s})\cos 30°}{0.52\text{ m}} = \mathbf{199\ rad/s}$ and $\dot{n} = \frac{\omega}{2\pi} = \frac{199\text{ rad/s}}{2\pi}\left(\frac{60\text{ s}}{1\text{ min}}\right) = \mathbf{1899\ rpm}$

3) θ = 60°: $\omega = \frac{V_{\text{nozzle}}}{r} = \frac{(119.4\text{ m/s})\cos 60°}{0.52\text{ m}} = \mathbf{115\ rad/s}$ and $\dot{n} = \frac{\omega}{2\pi} = \frac{115\text{ rad/s}}{2\pi}\left(\frac{60\text{ s}}{1\text{ min}}\right) = \mathbf{1096\ rpm}$

Discussion Final results are given to three significant digits, as usual. The rate of rotation in reality will be lower because of frictional effects and air drag.

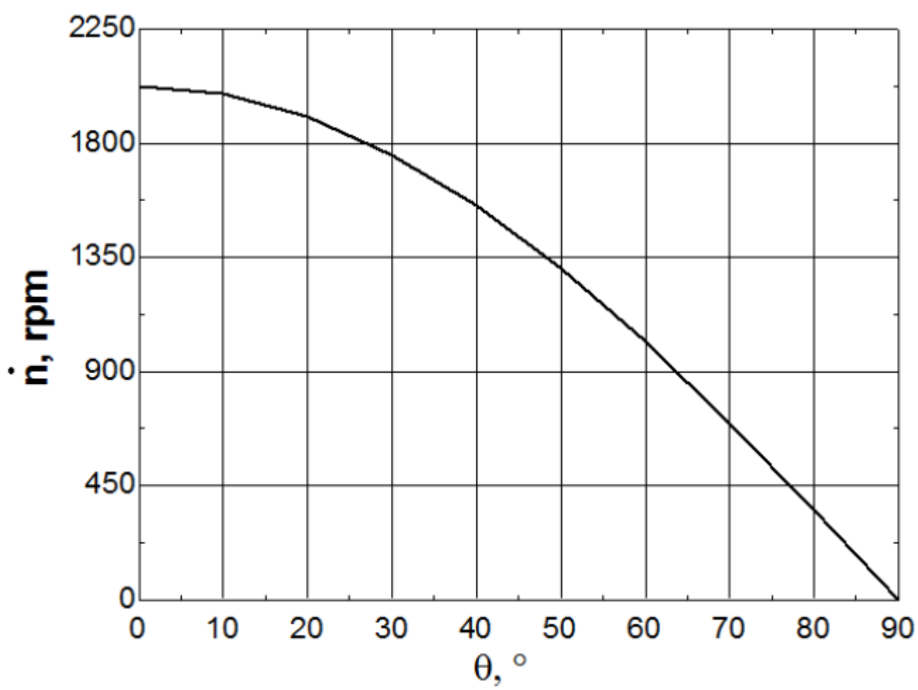
6-93

Solution The previous problem is reconsidered. The effect of discharge angle θ on the rate of rotation $\dot{n}$ as θ varies from 0 to 90° in increments of 10° is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
D=0.02 "m"
r=0.45 "m"
n_nozzle=2 "number of nozzles"
Ac=pi*D^2/4
V_jet=V_dot/Ac/n_nozzle
V_nozzle=V_jet*cos(theta)
V_dot=0.060 "m3/s"
omega=V_nozzle/r
n_dot=omega*60/(2*pi)
```

Angle, $\theta°$	V_{nozzle}, m/s	ω rad/s	$\dot{n}$ rpm
0	95.5	212	2026
10	94.0	209	1996
20	89.7	199	1904
30	82.7	184	1755
40	73.2	163	1552
50	61.4	136	1303
60	47.7	106	1013
70	32.7	73	693
80	16.6	37	352
90	0.0	0	0



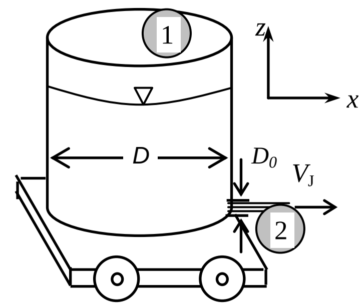
Discussion The maximum rpm occurs when $\theta = 0^{o}$, as expected, since this represents purely tangential outflow. When $\theta = 90^{o}$, the rpm drops to zero, as also expected, since the outflow is purely radial and therefore there is no torque to spin the sprinkler.

6-94

Solution A stationary water tank placed on wheels on a frictionless surface is propelled by a water jet that leaves the tank through a smooth hole. Relations are to be developed for the acceleration, the velocity, and the distance traveled by the tank as a function of time as water discharges.

Assumptions **1** The orifice has a smooth entrance, and thus the frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational (so that the Bernoulli equation is applicable). **3** The surface under the wheeled tank is level and frictionless. **4** The water jet is discharged horizontally and rearward. **5** The mass of the tank and wheel assembly is negligible compared to the mass of water in the tank. **4** Jet flow is nearly uniform and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Analysis (*a*) We take point 1 at the free surface of the tank, and point 2 at the outlet of the hole, which is also taken to be the reference level ($z_2 = 0$) so that the water height above the hole at any time is z. Noting that the fluid velocity at the free surface is very low ($V_1 \cong 0$), it is open to the atmosphere ($P_1 = P_{atm}$), and water discharges into the atmosphere (and thus $P_2 = P_{atm}$), the Bernoulli equation simplifies to



$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \rightarrow \quad z = \frac{V_J^2}{2g} + 0 \quad \rightarrow \quad V_J = \sqrt{2gz}$$

The discharge rate of water from the tank through the hole is

$$\dot{m} = \rho A V_J = \rho \frac{\pi D_0^2}{4} V_J = \rho \frac{\pi D_0^2}{4} \sqrt{2gz}$$

The momentum equation for steady flow is $\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}$. Applying it to the water tank, the horizontal force that acts on the tank is determined to be

$$F = \dot{m} V_e - 0 = \dot{m} V_J = \rho \frac{\pi D_0^2}{4} 2gz = \rho g z \frac{\pi D_0^2}{2}$$

The acceleration of the water tank is determined from Newton's 2nd law of motion $F = ma$ where m is the mass of water in the tank, $m = \rho \boldsymbol{V}_{tank} = \rho(\pi D^2/4)z$,

$$a = \frac{F}{m} = \frac{\rho g z \left(\pi D_0^2/2\right)}{\rho z \left(\pi D^2/4\right)} \quad \rightarrow \quad \boxed{a = 2g \frac{D_0^2}{D^2}}$$

Note that the acceleration of the tank is constant.

(*b*) Noting that $a = dV/dt$ and thus $dV = adt$ and acceleration a is constant, the velocity is expressed as

$$V = at \quad \rightarrow \quad \boxed{V = 2g \frac{D_0^2}{D^2} t}$$

(*c*) Noting that $V = dx/dt$ and thus $dx = Vdt$, the distance traveled by the water tank is determined by integration to be

$$dx = Vdt \quad \rightarrow \quad dx = 2g \frac{D_0^2}{D^2} t dt \quad \rightarrow \quad \boxed{x = g \frac{D_0^2}{D^2} t^2}$$

since $x = 0$ at $t = 0$.

Discussion In reality, the flow rate discharge velocity and thus the force acting on the water tank will be less because of the frictional losses at the hole. But these losses can be accounted for by incorporating a discharge coefficient.

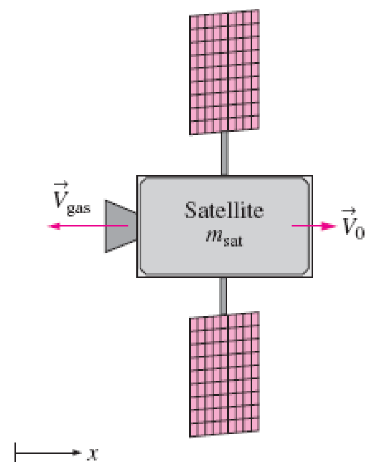
6-95

Solution The rocket of a satellite is fired in the opposite direction to motion. The thrust exerted on the satellite, the acceleration, and the velocity change are to be determined.

Assumptions **1** The flow of combustion gases is steady and one-dimensional during the firing period, but the motion of the satellite is unsteady. **2** There are no external forces acting on the spacecraft, and the effect of pressure force at the nozzle outlet is negligible. **3** The mass of discharged fuel is negligible relative to the mass of the spacecraft, and thus, the spacecraft may be treated as a solid body with a constant mass. **4** The nozzle is well designed such that the effect of the momentum-flux correction factor is negligible, and thus $\beta \cong 1$.

Analysis (*a*) For convenience, we choose an inertial reference frame that moves with the satellite at the same initial velocity. Then the velocities of fluid stream relative to an inertial reference frame become simply the velocities relative to the satellite. We take the direction of motion of the satellite as the positive direction along the *x*-axis. There are no external forces acting on the satellite, and its mass is essentially constant. Therefore, the satellite can be treated as a solid body with constant mass, and the momentum equation in this case is

$$\vec{F}_{\text{thrust}} = m_{\text{satellite}}\vec{a}_{\text{satellite}} = \sum_{\text{in}} \beta \dot{m}\vec{V} - \sum_{\text{out}} \beta \dot{m}\vec{V}$$



The fuel discharge rate is

$$\dot{m}_f = \frac{m_f}{\Delta t} = \frac{100\text{ kg}}{3\text{ s}} = 33.33\text{ kg/s}$$

Then the thrust exerted on the satellite in the positive x direction becomes

$$F_{\text{thrust}} = 0 - \dot{m}_f V_f = -(33.33\text{ kg/s})(-3000\text{ m/s})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{100\,kN}$$

(*b*) Noting that the net force acting on the satellite is thrust, the acceleration of the satellite in the direction of thrust during the first 2 s is determined to be

$$a_{\text{satellite}} = \frac{F_{\text{thrust}}}{m_{\text{satellite}}} = \frac{100\text{ kN}}{3400\text{ kg}}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kN}}\right) = 29.41\text{ m/s}^2 \cong \mathbf{29.4\,m/s^2}$$

(*c*) Knowing acceleration, which is constant, the velocity change of the satellite during the first 2 s is determined from the definition of acceleration $a_{\text{satellite}} = dV_{\text{satellite}}/dt$,

$$\Delta V_{\text{satellite}} = a_{\text{satellite}}\Delta t = (29.41\text{ m/s}^2)(3\text{ s}) = \mathbf{88.2\,m/s}$$

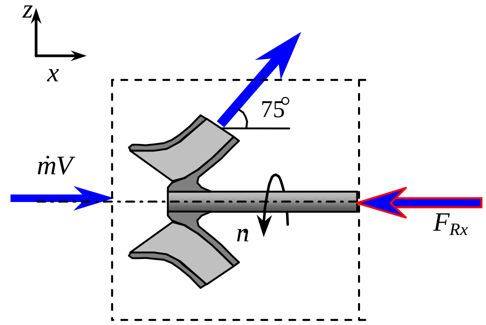
Discussion Note that if this satellite were attached somewhere, it would exert a force of 100 kN (equivalent to the weight of 10 tons of mass) to its support. This can be verified by taking the satellite as the system and applying the momentum equation.

6-96

Solution Water enters a centrifugal pump axially at a specified rate and velocity, and leaves at an angle from the axial direction. The force acting on the shaft in the axial direction is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The forces acting on the piping system in the horizontal direction are negligible. **3** The atmospheric pressure is disregarded since it acts on all surfaces. **4** Water flow is nearly uniform at the outlet and thus the momentum-flux correction factor can be taken to be unity, $\beta \cong 1$.

Properties We take the density of water to be 1000 kg/m^3.



Analysis From conservation of mass we have $\dot{m}_1 = \dot{m}_2 = \dot{m}$, and thus $\dot{V}_1 = \dot{V}_2$ and $A_{c1}V_1 = A_{c2}V_2$. Noting that the discharge area is half the inlet area, the discharge velocity is twice the inlet velocity. That is,

$$A_{c1}V_2 = \frac{A_{c1}}{A_{c2}}V_1 = 2V_1 = 2(7\text{ m/s}) = 14\text{ m/s}$$

We take the pump as the control volume, and the inlet direction of flow as the positive direction of x axis. The linear momentum equation in this case in the x direction reduces to

$$\sum \vec{F} = \sum_{\text{out}} \beta\dot{m}\vec{V} - \sum_{\text{in}} \beta\dot{m}\vec{V} \quad \rightarrow \quad -F_{Rx} = \dot{m}V_2\cos\theta - \dot{m}V_1 \quad \rightarrow \quad F_{Rx} = \dot{m}(V_1 - V_2\cos\theta)$$

where the mass flow rate it

$$\dot{m} = \rho\dot{V} = (1000\text{ kg/m}^3)(0.30\text{ m}^3/\text{s}) = 300\text{ kg/s}$$

Substituting the known quantities, the reaction force is determined to be

$$F_{\text{Rx}} = (300\text{ kg/s})[(7\text{ m/s}) - (14\text{ m/s})\cos 75]\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{1013N}$$

Discussion Note that at this angle of discharge, the bearing is not subjected to any horizontal loading. Therefore, the loading in the system can be controlled by adjusting the discharge angle.

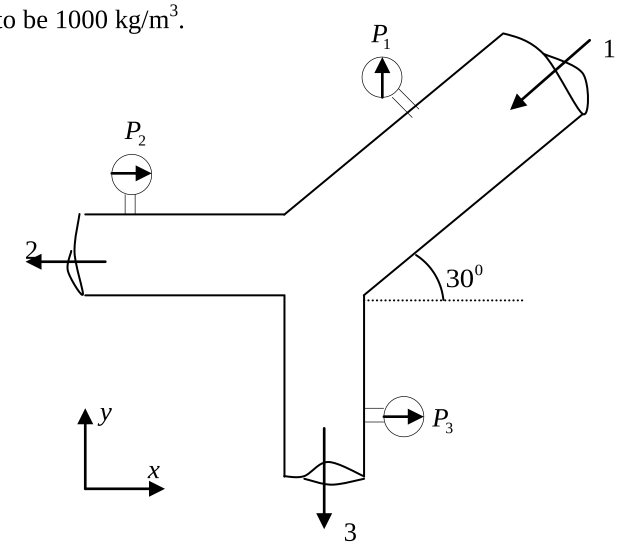
6-97

Solution Water flows through a splitter. The external force needed to hold the device fixed is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

Properties We take the density of water to be 1000 kg/m^3.

Analysis



Velocities at each section are

$$V_1 = \frac{0{,}08}{\pi .0.12^2/4} = 7.07\ m/s$$

$$V_2 = \frac{0{,}05}{\pi .0.12^2/4} = 4.42\ m/s$$

$$V_3 = \frac{0{,}03}{\pi .0.1^2/4} = 3.82\ m/s$$

Applying linear momentum equation to the CV,

$$\frac{\partial}{\partial t}\int_{CV} V\rho dV + \int_{CS} \vec{V}.\rho\vec{V}.\vec{n}dA = \sum F$$

x-component:

$$-(V_1.\cos\theta)\rho(-V_1)A_1 + (-V_2)\rho(V_2)A_2 = -P_1A_1\cos\theta + P_2A_2 + Rx$$

$$-\rho\dot{V}_1 Cos\theta V_1 - \rho\dot{V}_2 V_2 + P_1A_1Cos\theta - P_2A_2 = R_x$$

$$R_X = (1000)(0.08.\cos 30)(7.07) - 1000\times(0.05)\times(4.42) + 100000\pi\frac{0.12^2}{4}\cos 30 - 90000\pi\frac{0.12^2}{4}$$

$$R_X = 230\ \text{N}$$

y-component:

$$(-V_1.\sin\theta)\rho(-V_1)A_1 - V_3\rho(V_3)A_3 = -P_1A_1\sin\theta + P_3A_3 + R_y$$

$$R_y = \rho\dot{V}_1V_1Sin\theta - \rho\dot{V}_3V_3 + P_1A_1\sin\theta - P_3A_3$$

$$R_y = 1000\times 0.08\times 7.07\times Sin30 - 1000\times 0.03\times 3.82 + 100000\pi\frac{0.12^2}{4} - 80000\frac{\pi 0.1^2}{4}$$

$$R_y = 671\quad N\uparrow$$

The resultant force is then

$$R = \sqrt{230^2 + 671^2} = \mathbf{709N}$$

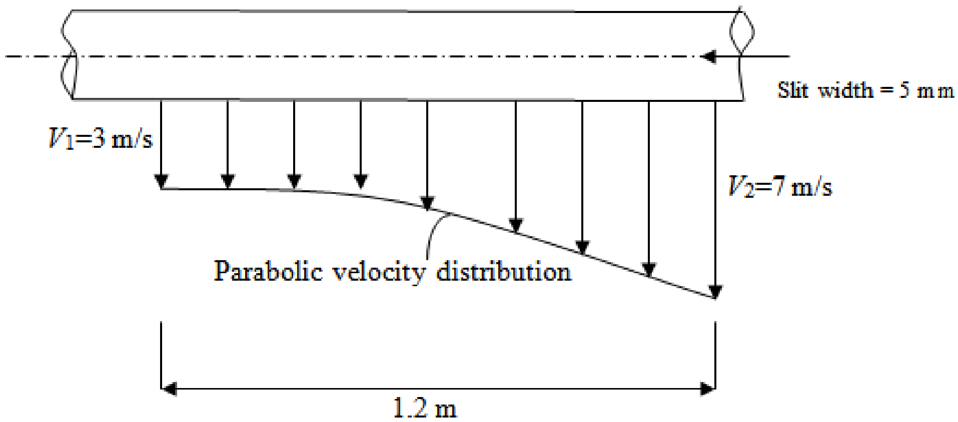
6-98

Solution Water is discharged from a pipe through a rectangular slit underneath of the pipe. The rate of discharge through the slit and the vertical force acting on the pipe due to this discharge process are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

Properties We take the density of water to be 1000 kg/m^3.

Analysis



(a) From the conservation of mass,

$$\frac{\partial}{\partial t}\int \rho dV + \int_{CS} \rho \vec{V}.\vec{n} dA = 0$$

$$\rho(-V_3)A_3 + \int_0^L \rho V dA = 0$$

$$V = a + bx + cx^2 \ , \ x = 0 \ V = V_1 = 3 \ m/s \quad , \quad a = 3 \ m/s$$

$$x = 0 \qquad \frac{dv}{dx} = 0 \quad , \quad b + 2cx = 0 \quad , \quad b = 0$$

$$x = L \quad , \qquad V = V_2 = 7 \ m/s$$

$$7 = 3 + c.1.2^2 \rightarrow c = 2.77$$

$$V(y) = 3 + 2.77x^2$$

$$-\rho\dot{V} + \int_0^L \rho.(3 + 2.77x^2)tdx = 0$$

$$\dot{V} = \int_0^{1,2} (3 + 2.77x^2) \times 0.01 dx = 0.005(3x + 0.923x^3)\Big|_0^{1,2}$$

$$\dot{V} = \mathbf{0.026 m^3/s}$$

(b) Momentum equations: y-component:

$$0 + \int_0^L (-V)\rho(V)dA = R_y$$

$$R_y = -\int_0^L \rho V^2 t dx = -\rho.t\int_0^L (3 + 2.77x^2)^2 dx$$

$$R_y = -\rho t \int_0^L \left(9 + 16.62x^2 + 7.673x^4\right) dx$$

$$R_y - \rho t \left[9x + 5.54x^3 + 1.5346x^5\right]\Big|_0^{1.2}$$

$$R_y = -1000 \times 0.005 \times (24.19) \Rightarrow$$

$$R_y = \mathbf{-121\ N} \ \downarrow$$

Fundamentals of Engineering (FE) Exam Problems

6-99

When determining the thrust developed by a jet engine, a wise choice of control volume is

(*a*) Fixed control volume (*b*) Moving control volume (*c*) Deforming control volume

(*d*) Moving or deforming control volume (*e*) None of these

Answer (*c*) Deforming control volume

6-100

Consider an airplane cruising at 850 km/h to the right. If the velocity of exhaust gases is 700 km/h to the left relative to the ground, the velocity of the exhaust gases relative to the nozzle exit is

(*a*) 1550 km/h (*b*) 850 km/h (*c*) 700 km/h (*d*) 350 km/h (*e*) 150 km/h

Answer (*a*) 1550 km/h

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

V_airplane=850 [km/h]

V_exhaust=-700 [km/h]

V_r=V_airplane-V_exhaust

6-101

Consider water flow through a horizontal, short garden hose at a rate of 30 kg/min. The velocity at the inlet is 1.5 m/s and that at the outlet is 14.5 m/s. Disregard the weight of the hose and water. Taking the momentum-flux correction factor to be 1.04 at both the inlet and the outlet, the anchoring force required to hold the hose in place is

(*a*) 2.8 N (*b*) 8.6 N (*c*) 17.5 N (*d*) 27.9 N (*e*) 43.3 N

Answer (*d*) 27.9 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_dot=(30/60) [kg/s]
V1=1.5 [m/s]
V2=14.5 [m/s]
beta=1.04
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
A1=m_dot/(rho*V1)
P_1_gage=rho*(V2^2-V1^2)/2
A1=pi*D^2/4
F+P_1_gage*A1=m_dot*beta*(V2-V1)
```

6-102

Consider water flow through a horizontal, short garden hose at a rate of 30 kg/min. The velocity at the inlet is 1.5 m/s and that at the outlet is 11.5 m/s. The hose makes a 180° turn before the water is discharged. Disregard the weight of the hose and water. Taking the momentum-flux correction factor to be 1.04 at both the inlet and the outlet, the anchoring force required to hold the hose in place is

(*a*) 7.6 N (*b*) 28.4 N (*c*) 16.6 N (*d*) 34.1 N (*e*) 11.9 N

Answer (*b*) 28.4 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_dot=(30/60) [kg/s]
V1=1.5 [m/s]
V2=-11.5 [m/s]
beta=1.04
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
A1=m_dot/(rho*V1)
P_1_gage=rho*(V2^2-V1^2)/2
A1=pi*D^2/4
F+P_1_gage*A1=m_dot*beta*(V2-V1)
```

6-103

A water jet strikes a stationary vertical plate horizontally at a rate of 5 kg/s with a velocity of 35 km/h. Assume the water stream moves in the vertical direction after the strike. The force needed to prevent the plate from moving horizontally is

(*a*) 15.5 N (*b*) 26.3 N (*c*) 19.7 N (*d*) 34.2 N (*e*) 48.6 N

Answer (*e*) 48.6 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

m_dot=5 [kg/s]

V1=35 [km/h]*Convert(km/h, m/s)

F=m_dot*V1

6-104

Consider water flow through a horizontal, short garden hose at a rate of 40 kg/min. The velocity at the inlet is 1.5 m/s and that at the outlet is 16 m/s. The hose makes a 90° turn to a vertical direction before the water is discharged. Disregard the weight of the hose and water. Taking the momentum-flux correction factor to be 1.04 at both the inlet and the outlet, the reaction force in the vertical direction required to hold the hose in place is

(*a*) 11.1 N (*b*) 10.1 N (*c*) 9.3 N(*d*) 27.2 N (*e*) 28.9 N

Answer (*a*) 11.1 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

m_dot=(40/60) [kg/s]

V1=1.5 [m/s]

V2=16 [m/s]

beta=1.04

F_vertical=m_dot*beta*V2

6-105

Consider water flow through a horizontal, short pipe at a rate of 80 kg/min. The velocity at the inlet is 1.5 m/s and that at the outlet is 16.5 m/s. The pipe makes a 90° turn to a vertical direction before the water is discharged. Disregard the weight of the pipe and water. Taking the momentum-flux correction factor to be 1.04 at both the inlet and the outlet, the reaction force in the horizontal direction required to hold the pipe in place is

(*a*) 73.7 N (*b*) 97.1 N (*c*) 99.2 N (*d*) 122 N (*e*) 153 N

Answer (*d*) 122 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

m_dot=(80/60) [kg/s]

V1=1.5 [m/s]

V2=16.5 [m/s]

theta_2=90 [degree]

beta=1.04

rho=1000 [kg/m^3]

g=9.81 [m/s^2]

A1=m_dot/(rho*V1)

P_1_gage=rho*(V2^2-V1^2)/2

A1=pi*D^2/4

F_horizontal+P_1_gage*A1=m_dot*beta*(V2*Cos(theta_2)-V1)

6-106

A water jet strikes a stationary vertical plate vertically at a rate of 18 kg/s with a velocity of 24 m/s. The mass of the plate is 10 kg. Assume the water stream moves in the horizontal direction after the strike. The force needed to prevent the plate from moving vertically is

(*a*) 192 N (*b*) 240 N (*c*) 334 N (*d*) 432 N (*e*) 530 N

Answer (*c*) 334 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

m_dot=18 [kg/s]

V1=24 [m/s]

m_plate=10 [kg]

g=9.81 [m/s^2]

F_vertical-m_plate*g=m_dot*(0-V1)

6-107

The velocity of wind at a wind turbine is measured to be 6 m/s. The blade span diameter is 24 m and the efficiency of the wind turbine is 29 percent. The density of air is 1.22 kg/m^3. The horizontal force exerted by the wind on the supporting mast of the wind turbine is

(*a*) 2524 N (*b*) 3127 N (*c*) 3475 N (*d*) 4138 N (*e*) 4313 N

Answer (*b*) 3127 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V1=6 [m/s]
D=24 [m]
eta_turbine=0.29
rho=1.22 [kg/m^3]
A=pi*D^2/4
m_dot=rho*A*V1
eta_turbine=1-KE_2/KE_1
KE_1=m_dot*V1^2/2
KE_2=m_dot*V2^2/2
F=m_dot*(V2-V1)
```

6-108

The velocity of wind at a wind turbine is measured to be 8 m/s. The blade span diameter is 12 m. The density of air is 1.2 kg/m^3. If the horizontal force exerted by the wind on the supporting mast of the wind turbine is 1620 N, the efficiency of the wind turbine is

(*a*) 27.5% (*b*) 31.7% (*c*) 29.5% (*d*) 35.1% (*e*) 33.8%

Answer (*e*) 33.8%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V1=8 [m/s]
D=12 [m]
F=-1620 [N]
rho=1.2 [kg/m^3]
A=pi*D^2/4
m_dot=rho*A*V1
KE_1=m_dot*V1^2/2
F=m_dot*(V2-V1)
KE_2=m_dot*V2^2/2
eta_turbine=1-KE_2/KE_1
```

6-109

The shaft of a turbine rotates at a speed of 800 rpm. If the torque of the shaft is 350 N·m, the shaft power is

(*a*) 112 kW (*b*) 176 kW (*c*) 293 kW (*d*) 350 kW (*e*) 405 kW

Answer (*c*) 293 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_dot=(800/60) [1/s]
M=3500 [N-m]
W_dot_shaft=2*pi*n_dot*M
```

6-110

A 3-cm-diameter horizontal pipe attached to a surface makes a 90° turn to a vertical upward direction before the water is discharged at a velocity of 9 m/s. The horizontal section is 5-m-long and the vertical section is 4-m long. Neglecting the mass of the water contained in the pipe, the bending moment acting on the base of the pipe on the wall is

(*a*) 286 N·m (*b*) 229 N·m (*c*) 207 N·m (*d*) 175 N·m (*e*) 124 N·m

Answer (*a*) 286 N·m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.03 [m]
V=9 [m/s]
L_horizontal=5 [m]
L_vertical=4 [m]
rho=1000 [kg/m^3]
A=pi*D^2/4
m_dot=rho*A*V
r=L_horizontal
M=r*m_dot*V
```

6-111

A 3-cm-diameter horizontal pipe attached to a surface makes a 90° turn to a vertical upward direction before the water is discharged at a velocity of 6 m/s. The horizontal section is 5-m-long and the vertical section is 4-m long. Neglecting the mass of the pipe and considering the weight of the water contained in the pipe, the bending moment acting on the base of the pipe on the wall is

(*a*) 11.9 N·m (*b*) 46.7 N·m (*c*) 127 N·m (*d*) 104 N·m (*e*) 74.8 N·m

Answer (*e*) 74.8 N·m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.03 [m]
V=6 [m/s]
L_horizontal=5 [m]
L_vertical=4 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
A=pi*D^2/4
m_dot=rho*A*V
Vol_horizontal=pi*D^2/4*L_horizontal
m_water_horizontal=rho*Vol_horizontal
W_horizontal=m_water_horizontal*g
Vol_vertical=pi*D^2/4*L_vertical
m_water_vertical=rho*Vol_vertical
W_vertical=m_water_vertical*g
r_horizontal=L_horizontal/2
r_vertical=L_horizontal+D/2
r=L_horizontal
M-r_horizontal*W_horizontal+r_vertical*W_vertical=r*m_dot*V
```

6-112

A large lawn sprinkler with four identical arms is to be converted into a turbine to generate electric power by attaching a generator to its rotating head. Water enters the sprinkler from the base along the axis of rotation at a rate of 15 kg/s and leaves the nozzles in the tangential direction at a velocity of 50 m/s relative to the rotating nozzle. The sprinkler rotates at a rate of 400 rpm in a horizontal plane. The normal distance between the axis of rotation and the center of each nozzle is 30 cm. Estimate the electric power produced.

(*a*) 5430 W (*b*) 6288 W (*c*) 6634 W (*d*) 7056 W (*e*) 7875 W

Answer (*d*) 7056 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_dot=15 [kg/s]
V_jet_r=50 [m/s]
n_dot=(400/60) [1/s]
r=0.30 [m]
omega=2*pi*n_dot
V_nozzle=r*omega
V_jet=V_jet_r-V_nozzle
T_shaft=r*m_dot*V_jet
W_dot=2*pi*n_dot*T_shaft
```

6-113

Consider the impeller of a centrifugal pump with a rotational speed of 900 rpm and a flow rate of 95 kg/min. The impeller radii at the inlet and outlet are 7 cm and 16 cm, respectively. Assuming that the tangential fluid velocity is equal to the blade angular velocity both at the inlet and the exit, the power requirement of the pump is

(*a*) 83 W (*b*) 291 W (*c*) 409 W (*d*) 756 W (*e*) 1125 W

Answer (*b*) 291 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_dot=(900/60) [1/s]
m_dot=(95/60) [kg/s]
r1=0.07 [m]
r2=0.16 [m]
omega=2*pi*n_dot
T_shaft=m_dot*omega*(r2^2-r1^2)
W_dot=2*pi*n_dot*T_shaft
```

6-114

Water enters the impeller of a centrifugal pump radially at a rate of 450 L/min when the shaft is rotating at 400 rpm. The tangential component of absolute velocity of water at the exit of the 70-cm outer diameter impeller is 55 m/s. The torque applied to the impeller is

(*a*) 144 N·m (*b*) 93.6 N·m (*c*) 187 N·m (*d*) 112 N·m (*e*) 235 N·m

Answer (*a*) 144 N·m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_dot=(0.450/60) [m^3/s]
n_dot=(40/60) [1/s]
D2=0.70 [m]
V_2_t=55 [m/s]
r2=D2/2
omega=2*pi*n_dot
rho=1000 [kg/m^3]
m_dot=rho*V_dot
T_shaft=m_dot*r2*V_2_t
```

Design and Essay Problem

6-115

Solution Students' essays and designs should be unique and will differ from each other.

Solutions Manual for

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CHAPTER 7
DIMENSIONAL ANALYSIS AND MODELING

Dimensions and Units, Primary Dimensions

7-1C

Solution We are to list the seven primary dimensions and explain their significance.

Analysis The seven primary dimensions are **mass, length, time, temperature, electrical current, amount of light, and amount of matter**. Their significance is that **all other dimensions can be formed by combinations of these seven primary dimensions**.

Discussion One of the first steps in a dimensional analysis is to write down the primary dimensions of every variable or parameter that is important in the problem.

7-2C

Solution We are to explain the difference between a dimension and a unit, and give examples.

Analysis **A dimension is a measure of a physical quantity (without numerical values), while a unit is a way to assign a number to that dimension**. Examples are numerous – length and meter, temperature and ^{o}C, weight and lbf, mass and kg, time and second, power and watt,…

Discussion When performing dimensional analysis, it is important to recognize the difference between dimensions and units.

7-3

Solution We are to write the primary dimensions of the universal ideal gas constant.

Analysis From the given equation,

Primary dimensions of the universal ideal gas constant:

$$\{R_u\} = \left\{\frac{\text{pressure} \times \text{volume}}{\text{mol} \times \text{temperature}}\right\} = \left\{\frac{\frac{\text{m}}{\text{t}^2\text{L}} \times \text{L}^3}{\text{N} \times \text{T}}\right\} = \left\{\frac{\mathbf{mL^2}}{\mathbf{t^2TN}}\right\}$$

Or, in exponent form, $\{R_u\}$ = $\{\mathbf{m^1\,L^2\,t^{-2}\,T^{-1}\,N^{-1}}\}$.

Discussion The standard value of R_u is 8314.3 J/kmol·K. You can verify that these units agree with the dimensions of the result.

7-4

Solution We are to determine the primary dimensions of each variable.

Analysis

(*a*) Energy is force times length (the same dimensions as work),

Primary dimensions of energy:

$$\{E\} = \{\text{force} \times \text{length}\} = \left\{\frac{\text{mass} \times \text{length}}{\text{time}^2} \times \text{length}\right\} = \left\{\frac{\mathbf{mL^2}}{\mathbf{t^2}}\right\} \tag{1}$$

Or, in exponent form, $\{E\}$ = $\{\mathbf{m^1\,L^2\,t^{-2}}\}$.

(*b*) Specific energy is energy per unit mass,

Primary dimensions of specific energy:

$$\{e\} = \left\{\frac{\text{energy}}{\text{mass}}\right\} = \left\{\frac{\text{mass} \times \text{length}^2}{\text{time}^2} \times \frac{1}{\text{mass}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2}}\right\} \tag{2}$$

Or, in exponent form, $\{e\}$ = $\{\mathbf{L^2\,t^{-2}}\}$.

(*c*) Power is the rate of change of energy, i.e. energy per unit time,

Primary dimensions of power:

$$\{\dot{W}\} = \left\{\frac{\text{energy}}{\text{time}}\right\} = \left\{\frac{\text{mass} \times \text{length}^2}{\text{time}^2} \times \frac{1}{\text{time}}\right\} = \left\{\frac{\mathbf{mL^2}}{\mathbf{t^3}}\right\} \tag{3}$$

Or, in exponent form, $\{\dot{W}\}$ = $\{\mathbf{m^1\,L^2\,t^{-3}}\}$.

Discussion In dimensional analysis it is important to distinguish between energy, specific energy, and power.

7-5

Solution We are to append the given table with other parameters and their primary dimensions.

Analysis Students' tables will differ, but they should add entries such as angular velocity, kinematic viscosity, work, energy, power, specific heat, thermal conductivity, torque or moment, stress, etc.

Discussion This problem should be assigned as an ongoing homework problem throughout the semester, and then collected towards the end. Individual instructors can determine how many entries to be required in the table.

7-6

Solution We are to write the primary dimensions of several variables in the force-length-time system.

Analysis We start with the dimensions of force in the mass-length-time (m-L-t) system: $\{F\} = \{m^1L^1t^{-2}\}$, from which we solve for the dimensions of mass in the force-length-time (F-L-t) system,

Primary dimensions of mass in the F-L-t system: $$\{F\} = \{m^1L^1t^{-2}\} \quad \rightarrow \quad \{m\} = \{F^1L^{-1}t^2\}$$

We plug in the above conversion to go from the m-L-t system to the F-L-t system for each of the variables:

Density: $$\{\rho\} = \{m^1L^{-3}\} \quad \rightarrow \quad \{\rho\} = \{F^1L^{-1}t^2L^{-3}\} = \{F^1L^{-4}t^2\}$$

Surface tension: $$\{\sigma_s\} = \{m^1t^{-2}\} \quad \rightarrow \quad \{\sigma_s\} = \{F^1L^{-1}t^2t^{-2}\} = \{F^1L^{-1}\}$$

Viscosity: $$\{\mu\} = \{m^1L^{-1}t^{-1}\} \quad \rightarrow \quad \{\mu\} = \{F^1L^{-1}t^2L^{-1}t^{-1}\} = \{F^1L^{-2}t^1\}$$

Discussion Sometimes the F-L-t system is easier to use than the m-L-t system. Neither one is “right” or “wrong” – it is a matter of personal preference, although the m-L-t system is the more popular one, especially in fluid mechanics.

7-7

Solution We are to write the primary dimensions of atomic weight.

Analysis By definition, atomic weight is mass per mol,

Primary dimensions of atomic weight: $$\{M\} = \left\{\frac{\text{mass}}{\text{mol}}\right\} = \left\{\frac{\mathbf{m}}{\mathbf{N}}\right\} \qquad (1)$$

Or, in exponent form, $\{M\} = \mathbf{\{m^1\,N^{-1}\}}$.

Discussion In terms of primary dimensions, atomic mass is *not* dimensionless, although many authors treat it as such. Note that mass and amount of matter are defined as two separate primary dimensions.

7-8

Solution We are to write the primary dimensions of the universal ideal gas constant in the alternate system where force replaces mass as a primary dimension.

Analysis From Newton's second law, force equals mass times acceleration. Thus, mass is written in terms of force as

Primary dimensions of mass in the alternate system:

$$\{\text{mass}\} = \left\{\frac{\text{force}}{\text{acceleration}}\right\} = \left\{\frac{\text{F}}{\text{L/t}^2}\right\} = \left\{\frac{\text{Ft}^2}{\text{L}}\right\} \qquad (1)$$

We substitute Eq. 1 into the results of Problem 7-4,

Primary dimensions of the universal ideal gas constant:

$$\{R_u\} = \left\{\frac{\text{mL}^2}{\text{Nt}^2\text{T}}\right\} = \left\{\frac{\dfrac{\text{Ft}^2}{\text{L}}\text{L}^2}{\text{Nt}^2\text{T}}\right\} = \left\{\frac{\mathbf{FL}}{\mathbf{TN}}\right\} \qquad (2)$$

Or, in exponent form, $\{R_u\}$ = $\mathbf{\{F^1\,L^1\,T^{-1}\,N^{-1}\}}$.

Discussion The standard value of R_u is 8314.3 J/kmol·K. You can verify that these units agree with the dimensions of Eq. 2.

7-9

Solution We are to write the primary dimensions of the specific ideal gas constant, and verify are result by comparing to the standard SI units of R_{air}.

Analysis We can approach this problem two ways. If we have already worked through Problem 7-4, we can use our results. Namely,

Primary dimensions of specific ideal gas constant:

$$\{R_{\text{gas}}\} = \left\{\frac{R_u}{M}\right\} = \left\{\frac{\dfrac{\text{mL}^2}{\text{Nt}^2\text{T}}}{\dfrac{\text{m}}{\text{N}}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2T}}\right\} \qquad (1)$$

Or, in exponent form, $\{R_{\text{gas}}\}$ = $\mathbf{\{L^2\,t^{-2}\,T^{-1}\}}$. Alternatively, we can use either form of the ideal gas law,

Primary dimensions of specific ideal gas constant:

$$\{R_{\text{gas}}\} = \left\{\frac{\text{pressure}\times\text{volume}}{\text{mass}\times\text{temperature}}\right\} = \left\{\frac{\dfrac{\text{m}}{\text{t}^2\text{L}}\times\text{L}^3}{\text{m}\times\text{T}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2T}}\right\} \qquad (2)$$

For air, R_{air} = 287.0 J/kg·K. We transform these units into primary dimensions,

Primary dimensions of the specific ideal gas constant for air:

$$\{R_{\text{air}}\} = \left\{287.0\frac{\text{J}}{\text{kg}\times\text{K}}\right\} = \left\{\frac{\dfrac{\text{mL}^2}{\text{t}^2}}{\text{m}\times\text{T}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2T}}\right\} \qquad (3)$$

Equation 3 agrees with Eq. 1 and Eq. 2, increasing our confidence that we have performed the algebra correctly.

Discussion Notice that numbers, like the value 287.0 in Eq. 3 have no influence on the dimensions.

7-10

Solution We are to write the primary dimensions of torque and list its units.

Analysis Torque is the product of a length and a force,

Primary dimensions of torque:
$$\left\{\vec{M}\right\} = \left\{\text{length}\times\text{mass}\frac{\text{length}}{\text{time}^2}\right\} = \left\{\mathbf{m}\frac{\mathbf{L^2}}{\mathbf{t^2}}\right\} \qquad (1)$$

Or, in exponent form, $\{\vec{M}\}$ = $\mathbf{\{m^1\ L^2\ t^{-2}\}}$. The common units of torque are newton-meter (SI) and inch-pound (English). In *primary units*, however, we write the primary SI units according to Eq. 1,

Primary SI units:
$$\text{Units of torque} = \mathbf{kg\cdot m^2/s^2}$$

and in primary English units,

Primary English units:
$$\text{Units of torque} = \mathbf{lbm\cdot ft^2/s^2}$$

Discussion Since torque is the product of a force and a length, it has the same dimensions as energy. Primary *units* are not required for dimensional analysis, but are often useful for unit conversions and for verification of proper units when solving a problem.

7-11

Solution We are to determine the primary dimensions of electrical voltage.

Analysis From the hint,

Primary dimensions of voltage:
$$\{\text{voltage}\} = \left\{\frac{\text{power}}{\text{current}}\right\} = \left\{\frac{\dfrac{\text{mass}\times\text{length}^2}{\text{time}^3}}{\text{current}}\right\} = \left\{\frac{\mathbf{mL^2}}{\mathbf{t^3I}}\right\} \qquad (1)$$

Or, in exponent form, $\{E\}$ = $\mathbf{\{m^1\,L^2\,t^{-3}\,I^{-1}\}}$.

Discussion We see that all dimensions, even those of electrical properties, can be expressed in terms of primary dimensions.

7-12

Solution We are to write the primary dimensions of electrical resistance.

Analysis From Ohm's law, we see that resistance has the dimensions of voltage difference divided by electrical current,

Primary dimensions of resistance:
$$\{R\} = \left\{\frac{\Delta E}{I}\right\} = \left\{\frac{\dfrac{\text{mass}\times\text{length}^2}{\text{time}^3\times\text{current}}}{\text{current}}\right\} = \left\{\frac{\mathbf{mL^2}}{\mathbf{t^3I^2}}\right\}$$

Or, in exponent form, $\{R\}$ = $\mathbf{\{m^1\,L^2\,t^{-3}\,I^{-2}\}}$, where we have also used the result of the previous problem.

Discussion *All* dimensions can be written in terms of primary dimensions.

7-13

Solution We are to determine the primary dimensions of each variable.

Analysis

(*a*) Acceleration is the rate of change of velocity,

Primary dimensions of acceleration:

$$\{a\} = \left\{\frac{\text{velocity}}{\text{time}}\right\} = \left\{\frac{\text{length}}{\text{time}} \times \frac{1}{\text{time}}\right\} = \left\{\frac{\mathbf{L}}{\mathbf{t}^2}\right\} \qquad (1)$$

Or, in exponent form, $\{a\}$ = $\{\mathbf{L}^1\,\mathbf{t}^{-2}\}$.

(*b*) Angular velocity is the rate of change of angle,

Primary dimensions of angular velocity: $$\{\omega\} = \left\{\frac{\text{angle}}{\text{time}}\right\} = \left\{\frac{1}{\text{time}}\right\} = \left\{\frac{\mathbf{1}}{\mathbf{t}}\right\} \qquad (2)$$

Or, in exponent form, $\{\omega\}$ = $\{\mathbf{t}^{-1}\}$.

(*c*) Angular acceleration is the rate of change of angular velocity,

Primary dimensions of angular acceleration:

$$\{\alpha = \dot{\omega}\} = \left\{\frac{\text{angular velocity}}{\text{time}}\right\} = \left\{\frac{1}{\text{time}} \times \frac{1}{\text{time}}\right\} = \left\{\frac{\mathbf{1}}{\mathbf{t}^2}\right\} \qquad (3)$$

Or, in exponent form, $\{\alpha\}$ = $\{\mathbf{t}^{-2}\}$.

Discussion In Part (*b*) we note that the unit of angle is radian, which is a dimensionless unit. Therefore the dimensions of angle are unity.

7-14

Solution We are to write the primary dimensions of angular momentum and list its units.

Analysis Angular momentum is the product of length, mass, and velocity,

Primary dimensions of angular momentum:

$$\{\vec{H}\} = \left\{\text{length} \times \text{mass} \times \frac{\text{length}}{\text{time}}\right\} = \left\{\frac{\mathbf{mL}^2}{\mathbf{t}}\right\} \qquad (1)$$

Or, in exponent form, $\{\vec{H}\}$ = $\{\mathbf{m}^1\,\mathbf{L}^2\,\mathbf{t}^{-1}\}$. We write the primary SI units according to Eq. 1,

Primary SI units: $$\text{Units of angular momentum} = \frac{\mathbf{kg \cdot m^2}}{\mathbf{s}}$$

and in primary English units,

Primary English units: $$\text{Units of angular momentum} = \frac{\mathbf{lbm \cdot ft^2}}{\mathbf{s}}$$

Discussion Primary units are not required for dimensional analysis, but are often useful for unit conversions and for verification of proper units when solving a problem.

7-15

Solution We are to determine the primary dimensions of each variable.

Analysis
(a) Specific heat is energy per unit mass per unit temperature,

Primary dimensions of specific heat at constant pressure:

$$\left\{c_p\right\} = \left\{\frac{\text{energy}}{\text{mass}\times\text{temperature}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m}\times\text{T}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2T}}\right\} \qquad (1)$$

Or, in exponent form, $\{c_p\}$ = $\{\mathbf{L^2\, t^{-2}\, T^{-1}}\}$.

(b) Specific weight is density times gravitational acceleration,

Primary dimensions of specific weight: $\left\{\rho g\right\} = \left\{\frac{\text{mass}}{\text{volume}}\frac{\text{length}}{\text{time}^2}\right\} = \left\{\frac{\mathbf{m}}{\mathbf{L^2t^2}}\right\} \qquad (2)$

Or, in exponent form, $\{\rho g\}$ = $\{\mathbf{m^1\, L^{-2}\, t^{-2}}\}$.

(c) Specific enthalpy has dimensions of energy per unit mass,

Primary dimensions of specific enthalpy: $\left\{h\right\} = \left\{\frac{\text{energy}}{\text{mass}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2}}\right\} \qquad (3)$

Or, in exponent form, $\{h\}$ = $\{\mathbf{L^2\, t^{-2}}\}$.

Discussion As a check, from our study of thermodynamics we know that $dh = c_p dT$ for an ideal gas. Thus, the dimensions of dh must equal the dimensions of c_p times the dimensions of dT. Comparing Eqs. 1 and 3 above, we see that this is indeed the case.

7-16

Solution We are to determine the primary dimensions of thermal conductivity.

Analysis The primary dimensions of $\dot{Q}_{\text{conduction}}$ are energy/time, and the primary dimensions of dT/dx are temperature/length. From the given equation,

Primary dimensions of thermal conductivity:

$$\left\{k\right\} = \left\{\frac{\frac{\text{energy}}{\text{time}}}{\text{length}^2\times\frac{\text{temperature}}{\text{length}}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^3}}{\text{L}\times\text{T}}\right\} = \left\{\frac{\mathbf{mL}}{\mathbf{t^3T}}\right\} \qquad (1)$$

Or, in exponent form, $\{k\}$ = $\{\mathbf{m^1\, L^1\, t^{-3}\, T^{-1}}\}$. We obtain a value of k from a reference book. E.g. k_{copper} = 401 W/m·K. These units have dimensions of power/length·temperature. Since power is energy/time, we see immediately that Eq. 1 is correct. Alternatively, we can transform the units of k into primary units,

Primary SI units of thermal conductivity:

$$k_{\text{copper}} = 401\frac{\text{W}}{\text{m K}}\left(\frac{\text{N m}}{\text{s W}}\right)\left(\frac{\text{kg m}}{\text{N s}^2}\right) = 401\frac{\mathbf{kg\cdot m}}{\mathbf{s^3\cdot K}} \qquad (2)$$

Discussion We have used the principle of dimensional homogeneity to determine the primary dimensions of k. Namely, we utilized the fact that the dimensions of both terms of the given equation must be identical.

7-17

Solution We are to determine the primary dimensions of each variable.

Analysis

(*a*) Heat generation rate is energy per unit volume per unit time,

Primary dimensions of heat generation rate: $$\{\dot{g}\} = \left\{\frac{\text{energy}}{\text{volume}\times\text{time}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{L}^3\text{t}}\right\} = \left\{\frac{\mathbf{m}}{\mathbf{Lt^3}}\right\} \tag{1}$$

Or, in exponent form, { $\dot{g}$ } = $\{\mathbf{m^1\,L^{-1}\,t^{-3}}\}$.

(*b*) Heat flux is energy per unit area per unit time,

Primary dimensions of heat flux: $$\{\dot{q}\} = \left\{\frac{\text{energy}}{\text{area}\times\text{time}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{L}^2\text{t}}\right\} = \left\{\frac{\mathbf{m}}{\mathbf{t^3}}\right\} \tag{2}$$

Or, in exponent form, { $\dot{q}$ } = $\{\mathbf{m^1\,t^{-3}}\}$.

(*c*) Heat flux is energy per unit area per unit time per unit temperature,

Primary dimensions of heat transfer coefficient: $$\{h\} = \left\{\frac{\text{energy}}{\text{area}\times\text{time}\times\text{temperature}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{L}^2\times\text{t}\times\text{T}}\right\} = \left\{\frac{\mathbf{m}}{\mathbf{t^3T}}\right\} \tag{3}$$

Or, in exponent form, $\{h\}$ = $\{\mathbf{m^1\,t^{-3}\,T^{-1}}\}$.

Discussion In the field of heat transfer it is critical that one be careful with the dimensions (and units) of heat transfer variables.

7-18

Solution We are to choose three properties or constants and write out their names, their SI units, and their primary dimensions.

Analysis There are many options. For example,

Students may choose c_v (specific heat at constant volume). The units are kJ/kg·K, which is energy per mass per temperature. Thus,

Primary dimensions of specific heat at constant volume:

$$\{c_v\} = \left\{\frac{\text{energy}}{\text{mass}\times\text{temperature}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m}\times\text{T}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2T}}\right\} \tag{1}$$

Or, in exponent form, $\{c_v\}$ = $\{\mathbf{L^2\,t^{-2}\,T^{-1}}\}$.

Students may choose v (specific volume). The units are m^3/kg, which is volume per mass. Thus,

Primary dimensions of specific volume:
$$\{v\} = \left\{\frac{\text{volume}}{\text{mass}}\right\} = \left\{\frac{\mathbf{L^3}}{\mathbf{m}}\right\} \tag{2}$$

Or, in exponent form, $\{v\}$ = $\{\mathbf{m^{-1}\,L^3}\}$.

Students may choose h_{fg} (latent heat of vaporization). The units are kJ/kg, which is energy per mass. Thus,

Primary dimensions of latent heat of vaporization:

$$\{h_{fg}\} = \left\{\frac{\text{energy}}{\text{mass}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2}}\right\} \tag{3}$$

Or, in exponent form, $\{h_{fg}\}$ = $\{\mathbf{L^2\,t^{-2}}\}$. (The same dimensions hold for h_f and h_g.)

Students may choose s_f (specific entropy of a saturated liquid). The units are kJ/kg·K, which is energy per mass per temperature. Thus,

Primary dimensions of specific entropy of a saturated liquid:

$$\{s_f\} = \left\{\frac{\text{energy}}{\text{mass}\times\text{temperature}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m}\times\text{T}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2T}}\right\} \tag{4}$$

Or, in exponent form, $\{s_f\}$ = $\{\mathbf{L^2\,t^{-2}\,T^{-1}}\}$. (The same dimensions hold for s_{fg} and s_g.)

Discussion Students' answers will vary. There are some other choices.

7-19E

Solution We are to choose three properties or constants and write out their names, their SI units, and their primary dimensions.

Analysis There are many options. For example,

Students may choose c_v (specific heat at constant volume). The units are Btu/lbm·R, which is energy per mass per temperature. Thus,

Primary dimensions of specific heat at constant volume:

$$\{c_v\} = \left\{\frac{\text{energy}}{\text{mass}\times\text{temperature}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m}\times\text{T}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2T}}\right\} \quad (1)$$

Or, in exponent form, $\{c_v\}$ = $\{\mathbf{L^2\,t^{-2}\,T^{-1}}\}$.

Students may choose v (specific volume). The units are ft^3/lbm, which is volume per mass. Thus,

Primary dimensions of specific volume:

$$\{v\} = \left\{\frac{\text{volume}}{\text{mass}}\right\} = \left\{\frac{\mathbf{L^3}}{\mathbf{m}}\right\} \quad (2)$$

Or, in exponent form, $\{v\}$ = $\{\mathbf{m^{-1}\,L^3}\}$.

Students may choose h_{fg} (latent heat of vaporization). The units are Btu/lbm, which is energy per mass. Thus,

Primary dimensions of latent heat of vaporization:

$$\{h_{fg}\} = \left\{\frac{\text{energy}}{\text{mass}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2}}\right\} \quad (3)$$

Or, in exponent form, $\{h_{fg}\}$ = $\{\mathbf{L^2\,t^{-2}}\}$. (The same dimensions hold for h_f and h_g.)

Students may choose s_f (specific entropy of a saturated liquid). The units are Btu/lbm·R, which is energy per mass per temperature. Thus,

Primary dimensions of specific entropy of a saturated liquid:

$$\{s_f\} = \left\{\frac{\text{energy}}{\text{mass}\times\text{temperature}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m}\times\text{T}}\right\} = \left\{\frac{\mathbf{L^2}}{\mathbf{t^2T}}\right\} \quad (4)$$

Or, in exponent form, $\{s_f\}$ = $\{\mathbf{L^2\,t^{-2}\,T^{-1}}\}$. (The same dimensions hold for s_{fg} and s_g.)

Discussion Students' answers will vary. There are some other choices.

Dimensional Homogeneity

7-20C

Solution We are to explain the law of dimensional homogeneity.

Analysis The law of dimensional homogeneity states that **every additive term in an equation must have the same dimensions**. As a simple counter example, an equation with one term of dimensions length and another term of dimensions temperature would clearly violate the law of dimensional homogeneity – you cannot add length and temperature. All terms in the equation must have the *same* dimensions.

Discussion If in the solution of an equation you realize that the dimensions of two terms are not equivalent, this is a sure sign that you have made a mistake somewhere!

7-21

Solution We are to determine the primary dimensions of the gradient operator, and then verify that primary dimensions of each additive term in the equation are the same.

Analysis

(*a*) By definition, the gradient operator is a three-dimensional derivative operator. For example, in Cartesian coordinates,

Gradient operator in Cartesian coordinates:

$$\vec{\nabla} = \left(\frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z} \right) = \vec{i}\,\frac{\partial}{\partial x} + \vec{j}\,\frac{\partial}{\partial y} + \vec{k}\,\frac{\partial}{\partial z}$$

Therefore its dimensions must be 1/length. Thus,

Primary dimensions of the gradient operator: $$\left\{ \vec{\nabla} \right\} = \left\{ \frac{\mathbf{1}}{\mathbf{L}} \right\}$$

Or, in exponent form, $\{ \vec{\nabla} \} = \mathbf{\{ L^{-1} \}}$.

(*b*) Similarly, the primary dimensions of a time derivative ($\partial/\partial t$) are 1/time. Also, the primary dimensions of velocity are length/time, and the primary dimensions of acceleration are length/time^2. Thus each term in the given equation can be written in terms of primary dimensions,

$$\{\vec{a}\} = \left\{ \frac{\text{length}}{\text{time}^2} \right\} \qquad \{\vec{a}\} = \left\{ \frac{\text{L}}{\text{t}^2} \right\}$$

$$\left\{ \frac{\partial \vec{V}}{\partial t} \right\} = \left\{ \frac{\frac{\text{length}}{\text{time}}}{\text{time}} \right\} = \left\{ \frac{\text{length}}{\text{time}^2} \right\} \qquad \left\{ \frac{\partial \vec{V}}{\partial t} \right\} = \left\{ \frac{\text{L}}{\text{t}^2} \right\}$$

$$\left\{ \left(\vec{V} \cdot \vec{\nabla} \right) \vec{V} \right\} = \left\{ \frac{\text{length}}{\text{time}} \times \frac{1}{\text{length}} \times \frac{\text{length}}{\text{time}} \right\} = \left\{ \frac{\text{length}}{\text{time}^2} \right\} \qquad \left\{ \left(\vec{V} \cdot \vec{\nabla} \right) \vec{V} \right\} = \left\{ \frac{\text{L}}{\text{t}^2} \right\}$$

Indeed, **all three additive terms have the same dimensions, namely $\mathbf{\{L^1\,t^{-2}\}}$**.

Discussion If the dimensions of any of the terms were different from the others, it would be a sure sign that an error was made somewhere in deriving or copying the equation.

7-22

Solution We are to determine the primary dimensions of each additive term in the equation, and we are to verify that the equation is dimensionally homogeneous.

Analysis The primary dimensions of the time derivative ($\partial/\partial t$) are 1/time. The primary dimensions of the gradient vector are 1/length, and the primary dimensions of velocity are length/time. Thus each term in the equation can be written in terms of primary dimensions,

$$\left\{\frac{\vec{F}}{m}\right\} = \left\{\frac{\text{force}}{\text{mass}}\right\} = \left\{\frac{\dfrac{\text{mass}\times\text{length}}{\text{time}^2}}{\text{mass}}\right\} \qquad \left\{\frac{\vec{F}}{m}\right\} = \left\{\frac{\text{L}}{\text{t}^2}\right\}$$

$$\left\{\frac{\partial\vec{V}}{\partial t}\right\} = \left\{\frac{\dfrac{\text{length}}{\text{time}}}{\text{time}}\right\} \qquad \left\{\frac{\partial\vec{V}}{\partial t}\right\} = \left\{\frac{\text{L}}{\text{t}^2}\right\}$$

$$\left\{\left(\vec{V}\cdot\vec{\nabla}\right)\vec{V}\right\} = \left\{\frac{\text{length}}{\text{time}}\times\frac{1}{\text{length}}\times\frac{\text{length}}{\text{time}}\right\} \qquad \left\{\left(\vec{V}\cdot\vec{\nabla}\right)\vec{V}\right\} = \left\{\frac{\text{L}}{\text{t}^2}\right\}$$

Indeed, **all three additive terms have the same dimensions, namely {$\mathbf{L^1\,t^{-2}}$}**.

Discussion The dimensions are, in fact, those of acceleration.

7-23

Solution We are to determine the primary dimensions of each additive term, and we are to verify that the equation is dimensionally homogeneous.

Analysis The primary dimensions of the velocity components are length/time. The primary dimensions of coordinates r and z are length, and the primary dimensions of coordinate θ are unity (it is a dimensionless angle). Thus each term in the equation can be written in terms of primary dimensions,

$$\left\{\frac{1}{r}\frac{\partial(ru_r)}{\partial r}\right\} = \left\{\frac{1}{\text{length}}\times\frac{\text{length}\dfrac{\text{length}}{\text{time}}}{\text{length}}\right\} = \left\{\frac{1}{\text{time}}\right\} \qquad \left\{\frac{1}{r}\frac{\partial(ru_r)}{\partial r}\right\} = \left\{\frac{1}{\text{t}}\right\}$$

$$\left\{\frac{1}{r}\frac{\partial u_\theta}{\partial\theta}\right\} == \left\{\frac{1}{\text{length}}\times\frac{\dfrac{\text{length}}{\text{time}}}{1}\right\} = \left\{\frac{1}{\text{time}}\right\} \qquad \left\{\frac{1}{r}\frac{\partial u_\theta}{\partial\theta}\right\} = \left\{\frac{1}{\text{t}}\right\}$$

$$\left\{\frac{\partial u_z}{\partial z}\right\} = \left\{\frac{\dfrac{\text{length}}{\text{time}}}{\text{length}}\right\} = \left\{\frac{1}{\text{time}}\right\} \qquad \left\{\frac{\partial u_z}{\partial z}\right\} = \left\{\frac{1}{\text{t}}\right\}$$

Indeed, **all three additive terms have the same dimensions, namely {$\mathbf{t^{-1}}$}**.

Discussion If the dimensions of any of the terms were different from the others, it would be a sure sign that an error was made somewhere in deriving or copying the equation.

7-24

Solution We are to determine the primary dimensions of each additive term, and we are to verify that the equation is dimensionally homogeneous.

Analysis The primary dimensions of the time derivative (d/dt) are 1/time. The primary dimensions of density are mass/length^3, those of volume are length^3, those of area are length^2, and those of velocity are length/time. The primary dimensions of unit vector $\vec{n}$ are unity, i.e. {1} (in other words $\vec{n}$ has no dimensions). Finally, the primary dimensions of b, which is defined as B per unit mass, are {B/m}. Thus each term in the equation can be written in terms of primary dimensions,

$$\left\{\frac{dB_{\text{sys}}}{dt}\right\} = \left\{\frac{B}{\text{time}}\right\} \qquad \left\{\frac{dB_{\text{sys}}}{dt}\right\} = \left\{\frac{B}{\text{t}}\right\}$$

$$\left\{\frac{d}{dt}\int_{\text{CV}} \rho b dV\right\} = \left\{\frac{1}{\text{time}} \times \frac{\text{mass}}{\text{length}^3} \times \frac{B}{\text{mass}} \times \text{length}^3\right\} \qquad \left\{\frac{d}{dt}\int_{\text{CV}} \rho b dV\right\} = \left\{\frac{B}{\text{t}}\right\}$$

$$\left\{\int_{\text{CS}} \rho b \vec{V}_{\text{r}} \cdot \vec{n} dA\right\} = \left\{\frac{\text{mass}}{\text{length}^3} \times \frac{B}{\text{mass}} \times \frac{\text{length}}{\text{time}} \times 1 \times \text{length}^2\right\} \qquad \left\{\int_{\text{CS}} \rho b \vec{V}_{\text{r}} \cdot \vec{n} dA\right\} = \left\{\frac{B}{\text{t}}\right\}$$

Indeed, **all three additive terms have the same dimensions, namely {$\boldsymbol{B}$ $\mathbf{t}^{-1}$}**.

Discussion The RTT for property B has dimensions of rate of change of B.

7-25

Solution We are to determine the primary dimensions of the first three additive term, and we are to verify that those terms are dimensionally homogeneous. Then we are to evaluate the dimensions of the adsorption coefficient.

Analysis The primary dimensions of the time derivative (d/dt) are 1/time. Those of A_s are length^2, those of V are length^3, those of c are mass/length^3, and those of $\dot{V}$ are length^3/time. Thus the primary dimensions of the first three terms are

$$\left\{V\frac{dc}{dt}\right\} = \left\{\text{length}^3 \frac{\frac{\text{mass}}{\text{length}^3}}{\text{time}}\right\} = \left\{\frac{\text{mass}}{\text{time}}\right\} \qquad \left\{V\frac{dc}{dt}\right\} = \left\{\frac{\text{m}}{\text{t}}\right\}$$

$$\{S\} = \left\{\frac{\text{mass}}{\text{time}}\right\} \qquad \{S\} = \left\{\frac{\text{m}}{\text{t}}\right\}$$

$$\{\dot{V}c\} = \left\{\frac{\text{length}^3}{\text{time}} \times \frac{\text{mass}}{\text{length}^3}\right\} = \left\{\frac{\text{mass}}{\text{time}}\right\} \qquad \{\dot{V}c\} = \left\{\frac{\text{m}}{\text{t}}\right\}$$

Indeed, the first three additive terms have the same dimensions, namely **{$\mathbf{m}^1$ $\mathbf{t}^{-1}$}**. Since the equation must be dimensionally homogeneous, the last term must have the same dimensions as well. We use this fact to find the dimensions of k_w,

$$\{cA_s k_w\} = \left\{\frac{\text{mass}}{\text{time}}\right\} \qquad \{k_w\} = \left\{\frac{\frac{\text{mass}}{\text{time}}}{cA_s}\right\} = \left\{\frac{\frac{\text{mass}}{\text{time}}}{\frac{\text{mass}}{\text{length}^3} \times \text{length}^2}\right\} \qquad \{k_w\} = \left\{\frac{\mathbf{L}}{\mathbf{t}}\right\}$$

Or, in exponent form, $\{k_w\}$ = **{$\mathbf{L}^1$ $\mathbf{t}^{-1}$}**. The dimensions of wall adsorption coefficient are those of velocity.

Discussion In fact, some authors call k_w a "deposition velocity".

7-26

Solution We are to determine the primary dimensions of each additive term in Eq. 1, and we are to verify that the equation is dimensionally homogeneous.

Analysis The primary dimensions of the material derivative (*D*/*Dt*) are 1/time. The primary dimensions of volume are length^3, and the primary dimensions of velocity are length/time. Thus each term in the equation can be written in terms of primary dimensions,

$$\left\{\frac{1}{V}\frac{DV}{Dt}\right\} = \left\{\frac{1}{\text{length}^3}\times\frac{\text{length}^3}{\text{time}}\right\} = \left\{\frac{1}{\text{time}}\right\} \qquad \left\{\frac{1}{V}\frac{DV}{Dt}\right\} = \left\{\frac{1}{\text{t}}\right\}$$

$$\left\{\frac{\partial u}{\partial x}\right\} = \left\{\frac{\frac{\text{length}}{\text{time}}}{\text{length}}\right\} = \left\{\frac{1}{\text{time}}\right\} \qquad \left\{\frac{\partial u}{\partial x}\right\} = \left\{\frac{1}{\text{t}}\right\}$$

$$\left\{\frac{\partial v}{\partial y}\right\} = \left\{\frac{\frac{\text{length}}{\text{time}}}{\text{length}}\right\} = \left\{\frac{1}{\text{time}}\right\} \qquad \left\{\frac{\partial v}{\partial y}\right\} = \left\{\frac{1}{\text{t}}\right\}$$

$$\left\{\frac{\partial w}{\partial z}\right\} = \left\{\frac{\frac{\text{length}}{\text{time}}}{\text{length}}\right\} = \left\{\frac{1}{\text{time}}\right\} \qquad \left\{\frac{\partial w}{\partial z}\right\} = \left\{\frac{1}{\text{t}}\right\}$$

Indeed, **all four additive terms have the same dimensions, namely {t^{-1}}**.

Discussion If the dimensions of any of the terms were different from the others, it would be a sure sign that an error was made somewhere in deriving or copying the equation.

7-27

Solution We are to determine the primary dimensions of each additive term in the equation, and we are to verify that the equation is dimensionally homogeneous.

Analysis The primary dimensions of heat transfer rate are energy/time. The primary dimensions of mass flow rate are mass/time, and those of specific heat are energy/mass·temperature, as found in Problem 7-14. Thus each term in the equation can be written in terms of primary dimensions,

$$\left\{\dot{Q}\right\} = \left\{\frac{\text{energy}}{\text{time}}\right\} = \left\{\frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{t}}\right\} = \left\{\frac{\text{mL}^2}{\text{t}^3}\right\} \qquad \left\{\dot{Q}\right\} = \left\{\frac{\text{mL}^2}{\text{t}^3}\right\}$$

$$\left\{\dot{m}C_pT_{\text{out}}\right\} = \left\{\frac{\text{mass}}{\text{time}} \times \frac{\text{energy}}{\text{mass} \times \text{temperature}} \times \text{temperature}\right\} = \left\{\frac{\text{m}}{\text{t}} \times \frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m} \times \text{T}} \times \text{T}\right\} = \left\{\frac{\text{mL}^2}{\text{t}^3}\right\} \qquad \left\{\dot{m}C_pT_{\text{out}}\right\} = \left\{\frac{\text{mL}^2}{\text{t}^3}\right\}$$

$$\left\{\dot{m}C_pT_{\text{in}}\right\} = \left\{\frac{\text{mass}}{\text{time}} \times \frac{\text{energy}}{\text{mass} \times \text{temperature}} \times \text{temperature}\right\} = \left\{\frac{\text{m}}{\text{t}} \times \frac{\frac{\text{mL}^2}{\text{t}^2}}{\text{m} \times \text{T}} \times \text{T}\right\} = \left\{\frac{\text{mL}^2}{\text{t}^3}\right\} \qquad \left\{\dot{m}C_pT_{\text{in}}\right\} = \left\{\frac{\text{mL}^2}{\text{t}^3}\right\}$$

Indeed, **all three additive terms have the same dimensions, namely $\{m^1 L^2 t^{-3}\}$**.

Discussion We could also have left the temperature difference in parentheses as a temperature difference (same dimensions as the individual temperatures), and treated the equation as having only two terms.

Nondimensionalization of Equations

7-28C

Solution We are to give the primary reason for nondimensionalizing an equation.

Analysis The primary reason for nondimensionalizing an equation is **to reduce the number of parameters in the problem**.

Discussion As shown in the examples in the text, nondimensionalization of an equation reduces the number of independent parameters in the problem, simplifying the analysis.

7-29

Solution We are to nondimensionalize the equation, and identify the dimensionless parameters that appear in the nondimensionalized equation.

Assumptions **1** The flow is steady. **2** The flow is incompressible.

Analysis We plug the nondimensionalized variables into the equation. For example, $u = u^*U$ and $x = x^*L$ in the first term. The result is

$$\frac{U}{L}\frac{\partial u^*}{\partial x^*}+\frac{U}{L}\frac{\partial v^*}{\partial y^*}+\frac{U}{L}\frac{\partial w^*}{\partial z^*}=0$$

or, after simplifying,

Nondimensionalized incompressible flow relationship:

$$\boxed{\frac{\partial u^*}{\partial x^*}+\frac{\partial v^*}{\partial y^*}+\frac{\partial w^*}{\partial z^*}=0} \qquad (1)$$

There are no nondimensional parameters in the nondimensionalized equation. The original equation comes from pure kinematics – there are no fluid properties involved in the equation, and therefore it is not surprising that no nondimensional parameters appear in the nondimensionalized form of the equation, Eq. 1.

Discussion We show in Chap. 9 that the equation given in this problem is the differential equation for conservation of mass for an incompressible flow field – the *incompressible continuity equation*.

7-30

Solution We are to nondimensionalize the equation of motion and identify the dimensionless parameters that appear in the nondimensionalized equation.

Analysis First, we must expand the first material derivative term since the nondimensionalization is not identical for the individual terms. Then we plug in the nondimensionalized variables. For example, $u = u^*V$ and $x = x^*L$ in the first term on the right. The result is

$$\frac{1}{V}\frac{DV}{Dt} = \frac{1}{V}\left(\frac{\partial V}{\partial t} + \left(\vec{V}\cdot\vec{\nabla}\right)V\right) = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}$$

$$\frac{1}{L^3 V^*}\left(L^3 f\frac{\partial V^*}{\partial t^*} + \frac{L^3 V}{L}\left(\overrightarrow{V^*}\cdot\overrightarrow{\nabla^*}\right)V^*\right) = \frac{V}{L}\frac{\partial u^*}{\partial x^*} + \frac{V}{L}\frac{\partial v^*}{\partial y^*} + \frac{V}{L}\frac{\partial w^*}{\partial z^*}$$

or, after simplifying (multiply each term by L/V),

$$\frac{1}{V^*}\left(\left(\frac{fL}{V}\right)\frac{\partial V^*}{\partial t^*} + \left(\overrightarrow{V^*}\cdot\overrightarrow{\nabla^*}\right)V^*\right) = \frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} + \frac{\partial w^*}{\partial z^*} \qquad (1)$$

We recognize the nondimensional parameter in parentheses in Eq. 1 as St, the **Strouhal number**, and we re-write Eq. 1 as

Nondimensionalized oscillating compressible flow relationship: $\boxed{\frac{1}{V^*}\left(\mathrm{St}\frac{\partial V^*}{\partial t^*} + \left(\overrightarrow{V^*}\cdot\overrightarrow{\nabla^*}\right)V^*\right) = \frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} + \frac{\partial w^*}{\partial z^*}}$

Discussion We show in Chap. 9 that the given equation of motion is the differential equation for conservation of mass for an unsteady, compressible flow field – the *general continuity equation*. We may also use angular frequency ω (radians per second) in place of physical frequency f (cycles per second), with the same result.

7-31

Solution We are to determine the primary dimensions of the stream function, nondimensionalize the variables, and then re-write the definition of ψ in nondimensionalized form.

Assumptions **1** The flow is incompressible. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis (*a*) We use that fact that all equations must be dimensionally homogeneous. We solve for the dimensions of ψ,

Primary dimensions of stream function: $$\{\psi\} = \{u\}\times\{y\} = \left\{\frac{\mathrm{L}}{\mathrm{t}}\times\mathrm{L}\right\} = \left\{\frac{\mathbf{L}^2}{\mathbf{t}}\right\}$$

Or, in exponent form, $\{\psi\}$ = $\{\mathbf{L}^2\,\mathbf{t}^{-1}\}$.

(*b*) We nondimensionalize the variables by inspection according to their dimensions,

Nondimensionalized variables:

$$\boxed{x^* = \frac{x}{L} \qquad y^* = \frac{y}{L} \qquad u^* = u\frac{t}{L} \qquad v^* = v\frac{t}{L} \qquad \psi^* = \psi\frac{t}{L^2}}$$

(*c*) We generate the nondimensionalized equations,

$$u^*\left(\frac{L}{t}\right) = \frac{\partial\psi^*\left(\frac{L^2}{t}\right)}{\partial y^*(L)} \qquad v^*\left(\frac{L}{t}\right) = -\frac{\partial\psi^*\left(\frac{L^2}{t}\right)}{\partial x^*(L)}$$

We notice that every term in both parts of the above equation contains the ratio *L*/*t*. We divide every term by *L*/*t* to get the final nondimensionalized form of the equations,

Nondimensionalized stream function equations: $$\boxed{u^* = \frac{\partial\psi^*}{\partial y^*} \qquad v^* = -\frac{\partial\psi^*}{\partial x^*}}$$

No dimensionless groups have arisen in this nondimensionalization.

Discussion Since all the nondimensionalized variables scale with *L* and *t*, no dimensionless groups have arisen.

7-32

Solution We are to nondimensionalize the equation of motion and identify the dimensionless parameters that appear in the nondimensionalized equation.

Analysis We plug the nondimensionalized variables into the equation. For example, $t = t^*/\omega$ and $\vec{V} = V_\infty \vec{V}^*$ in the first term on the right hand side. The result is

$$\omega^2 L\left(\vec{F}/m\right)^* = \omega U \frac{\partial \vec{V}^*}{\partial t^*} + \frac{U^2}{L}\left(\vec{V}^* \cdot \vec{\nabla}^*\right)\vec{V}^*$$

or, after simplifying by multiplying each term by L/V_∞^2,

$$\left(\frac{\omega L}{V_\infty}\right)^2 \left(\vec{F}/m\right)^* = \left(\frac{\omega L}{V_\infty}\right)\frac{\partial \vec{V}^*}{\partial t^*} + \left(\vec{V}^* \cdot \vec{\nabla}^*\right)\vec{V}^* \qquad (1)$$

We recognize the nondimensional parameter in parentheses in Eq. 1 as St, the **Strouhal number**. We re-write Eq. 1 as

Nondimensionalized Newton's second law for incompressible oscillatory flow:

$$\boxed{(\mathrm{St})^2 \left(\vec{F}/m\right)^* = (\mathrm{St})\frac{\partial \vec{V}^*}{\partial t^*} + \left(\vec{V}^* \cdot \vec{\nabla}^*\right)\vec{V}^*}$$

Discussion We used angular frequency ω in this problem. The same result would be obtained if we used physical frequency. Equation 1 is the basis for forming the differential equation for conservation of linear momentum for an unsteady, incompressible flow field.

7-33

Solution We are to nondimensionalize the Bernoulli equation and generate an expression for the pressure coefficient.

Assumptions **1** The flow is incompressible. **2** Gravitational terms in the Bernoulli equation are negligible compared to the other terms.

Analysis We nondimensionalize the equation by dividing each term by the **dynamic pressure**, $\frac{1}{2}\rho V_\infty^2$,

Nondimensionalization:

$$\frac{P}{\frac{1}{2}\rho V_\infty^2} + \frac{V^2}{V_\infty^2} = \frac{P_\infty}{\frac{1}{2}\rho V_\infty^2} + 1$$

Rearranging,

Pressure coefficient:

$$\boxed{C_p = \frac{P - P_\infty}{\frac{1}{2}\rho V_\infty^2} = 1 - \frac{V^2}{V_\infty^2}}$$

Discussion Pressure coefficient is a useful dimensionless parameter that is inversely related to local air speed – as local air speed V increases, C_p decreases.

7-34

Solution We are to nondimensionalize all the variables, and then re-write the equation in nondimensionalized form.

Assumptions **1** The air in the room is well mixed so that c is only a function of time.

Analysis
(a) We nondimensionalize the variables by inspection according to their dimensions,

Nondimensionalized variables:

$$V^* = \frac{V}{L^3},\ c^* = \frac{c}{c_{\text{limit}}},\ t^* = t\frac{\dot{V}}{L^3},\ A_s{}^* = \frac{A_s}{L^2},\ k_w{}^* = k_w\frac{L^2}{\dot{V}},\ \text{and}\ S^* = \frac{S}{c_{\text{limit}}\dot{V}}$$

(b) We substitute these into the equation to generate the nondimensionalized equation,

$$V^*L^3\frac{d\left(c^*c_{\text{limit}}\right)}{d\left(t^*\dfrac{L^3}{\dot{V}}\right)} = S^*c_{\text{limit}}\dot{V} - \dot{V}c^*c_{\text{limit}} - \left(c^*c_{\text{limit}}\right)\left(A_s{}^*L^2\right)\left(k_w{}^*\frac{\dot{V}}{L^2}\right) \qquad (1)$$

We notice that every term in Eq. 1 contains the quantity $\dot{V}c_{\text{limit}}$. We divide every term by this quantity to get a nondimensionalized form of the equation,

Nondimensionalized equation:

$$\boxed{V^*\frac{dc^*}{dt^*} = S^* - c^* - c^*A_s{}^*k_w{}^*}$$

No dimensionless groups have arisen in this nondimensionalization.

Discussion Since all the characteristic scales disappear, no dimensionless groups have arisen. Since there are no dimensionless parameters, one solution in nondimensionalized variables is valid for all combinations of L, $\dot{V}$, and c_{limit}.

Dimensional Analysis and Similarity

7-35C

Solution We are to list the three primary purposes of dimensional analysis.

Analysis The three primary purposes of *dimensional analysis* are:

1. **To generate nondimensional parameters that help in the design of experiments and in the reporting of experimental results**.
2. **To obtain scaling laws so that prototype performance can be predicted from model performance**.
3. **To (sometimes) predict trends in the relationship between parameters**.

Discussion Dimensional analysis is most useful for difficult problems that cannot be solved analytically.

7-36C

Solution We are to list and describe the three necessary conditions for complete similarity between a model and a prototype.

Analysis The three necessary conditions for complete similarity between a model and a prototype are:

1. **Geometric similarity** – the model must be the **same shape** as the prototype, but scaled by some constant scale factor.
2. **Kinematic similarity** – **the velocity at any point in the model flow must be proportional** (by a constant scale factor) to the velocity at the corresponding point in the prototype flow.
3. **Dynamic similarity** – **all *forces* in the model flow scale by a constant factor to corresponding forces in the prototype flow**.

Discussion Complete similarity is achievable only when all three of the above similarity conditions are met.

7-37

Solution For a scale model of a submarine being tested in air, we are to calculate the wind tunnel speed required to achieve similarity with the prototype submarine that moves through water at a given speed.

Assumptions **1** Compressibility of the air is assumed to be negligible. **2** The wind tunnel walls are far enough away so as to not interfere with the aerodynamic drag on the model sub. **3** The model is geometrically similar to the prototype.

Properties For water at $T = 15^\circ$C and atmospheric pressure, $\rho = 999.1$ kg/m^3 and $\mu = 1.138 \times 10^{-3}$ kg/m·s. For air at $T = 25^\circ$C and atmospheric pressure, $\rho = 1.184$ kg/m^3 and $\mu = 1.849 \times 10^{-5}$ kg/m·s.

Analysis Similarity is achieved when the Reynolds number of the model is equal to that of the prototype,

Similarity:

$$\mathrm{Re_m} = \frac{\rho_\mathrm{m} V_\mathrm{m} L_\mathrm{m}}{\mu_\mathrm{m}} = \mathrm{Re_p} = \frac{\rho_\mathrm{p} V_\mathrm{p} L_\mathrm{p}}{\mu_\mathrm{p}} \qquad (1)$$

We solve Eq. 1 for the unknown wind tunnel speed,

$$V_\mathrm{m} = V_\mathrm{p}\left(\frac{\mu_\mathrm{m}}{\mu_\mathrm{p}}\right)\left(\frac{\rho_\mathrm{p}}{\rho_\mathrm{m}}\right)\left(\frac{L_\mathrm{p}}{L_\mathrm{m}}\right) = (0.440\ \mathrm{m/s})\left(\frac{1.849\times10^{-5}\ \mathrm{kg/m\cdot s}}{1.138\times10^{-3}\ \mathrm{kg/m\cdot s}}\right)\left(\frac{999.1\,\mathrm{kg/m^3}}{1.184\,\mathrm{kg/m^3}}\right)(5) = \mathbf{30.2\,m/s}$$

Discussion At this air temperature, the speed of sound is around 346 m/s. Thus the Mach number in the wind tunnel is equal to 30.2/346 = 0.0873. This is sufficiently low that the incompressible flow approximation is reasonable.

7-38

Solution For a scale model of a submarine being tested in air, we are to calculate the wind tunnel speed required to achieve similarity with the prototype submarine that moves through water at a given speed.

Assumptions **1** Compressibility of the air is assumed to be negligible. **2** The wind tunnel walls are far enough away so as to not interfere with the aerodynamic drag on the model sub. **3** The model is geometrically similar to the prototype.

Properties For water at T = 15°C and atmospheric pressure, ρ = 999.1 kg/m^3 and μ = 1.138 × 10^{-3} kg/m·s. For air at T = 25°C and atmospheric pressure, ρ = 1.184 kg/m^3 and μ = 1.849 × 10^{-5} kg/m·s.

Analysis Similarity is achieved when the Reynolds number of the model is equal to that of the prototype,

Similarity:
$$\mathrm{Re_m} = \frac{\rho_\mathrm{m} V_\mathrm{m} L_\mathrm{m}}{\mu_\mathrm{m}} = \mathrm{Re_p} = \frac{\rho_\mathrm{p} V_\mathrm{p} L_\mathrm{p}}{\mu_\mathrm{p}} \tag{1}$$

We solve Eq. 1 for the unknown wind tunnel speed,

$$V_\mathrm{m} = V_\mathrm{p}\left(\frac{\mu_\mathrm{m}}{\mu_\mathrm{p}}\right)\left(\frac{\rho_\mathrm{p}}{\rho_\mathrm{m}}\right)\left(\frac{L_\mathrm{p}}{L_\mathrm{m}}\right) = (0.440\ \mathrm{m/s})\left(\frac{1.849\times10^{-5}\ \mathrm{kg/m\cdot s}}{1.138\times10^{-3}\ \mathrm{kg/m\cdot s}}\right)\left(\frac{999.1\,\mathrm{kg/m^3}}{1.184\,\mathrm{kg/m^3}}\right)(24) = \mathbf{145\,m/s}$$

At this air temperature, the speed of sound is around 346 m/s. Thus the Mach number in the wind tunnel is equal to 145/346 = 0.419. **The Mach number is sufficiently high that the incompressible flow approximation is not reasonable**. The wind tunnel should be run at a flow speed at which the Mach number is less than one-third of the speed of sound. At this lower speed, the Reynolds number of the model will be too small, but the results may still be usable, either by extrapolation to higher Re, or if we are fortunate enough to have Reynolds number independence, as discussed in Section 7-5.

Discussion It is also unlikely that a small instructional wind tunnel can achieve such a high speed.

7-39

Solution We are to estimate the drag on a prototype submarine in water, based on aerodynamic drag measurements performed in a wind tunnel.

Assumptions **1** The model is geometrically similar. **2** The wind tunnel is run at conditions which ensure similarity between model and prototype.

Properties For water at T = 15°C and atmospheric pressure, ρ = 999.1 kg/m^3 and μ = 1.138 × 10^{-3} kg/m·s. For air at T = 25°C and atmospheric pressure, ρ = 1.184 kg/m^3 and μ = 1.849 × 10^{-5} kg/m·s.

Analysis Since the Reynolds numbers have been matched, the nondimensionalized drag coefficient of the model equals that of the prototype,

$$\frac{F_{D,\mathrm{m}}}{\rho_\mathrm{m} V_\mathrm{m}^2 L_\mathrm{m}^2} = \frac{F_{D,\mathrm{p}}}{\rho_\mathrm{p} V_\mathrm{p}^2 L_\mathrm{p}^2} \tag{1}$$

We solve Eq. 1 for the unknown aerodynamic drag force on the prototype, $F_{D,\mathrm{p}}$,

$$F_{D,\mathrm{p}} = F_{D,\mathrm{m}}\left(\frac{\rho_\mathrm{p}}{\rho_\mathrm{m}}\right)\left(\frac{V_\mathrm{p}}{V_\mathrm{m}}\right)^2\left(\frac{L_\mathrm{p}}{L_\mathrm{m}}\right)^2 = (5.70\ \mathrm{N})\left(\frac{999.1\,\mathrm{kg/m^3}}{1.184\,\mathrm{kg/m^3}}\right)\left(\frac{0.440\,\mathrm{m/s}}{30.2\,\mathrm{m/s}}\right)^2(5)^2 = \mathbf{25.5\,N}$$

where we have used the wind tunnel speed calculated in Problem 7-39.

Discussion Although the prototype moves at a much slower speed than the model, the density of water is much higher than that of air, and the prototype is eight times larger than the model. When all of these factors are combined, the drag force on the prototype is much larger than that on the model.

7-40E

Solution For a prototype parachute and its model we are to calculate drag coefficient, and determine the wind tunnel speed that ensures dynamic similarity. Then we are to estimate the aerodynamic drag on the model.

Assumptions **1** The model is geometrically similar to the prototype.

Properties For air at 60°F and standard atmospheric pressure, ρ = 0.07633 lbm/ft^3 and μ = 1.213 × 10^{-5} lbm/ft·s.

Analysis (*a*) The aerodynamic drag on the prototype parachute is equal to the total weight. We can then easily calculate the drag coefficient C_D,

Drag coefficient:
$$C_D = \frac{F_{D,\mathrm{p}}}{\frac{1}{2}\rho_\mathrm{p} V_\mathrm{p}^2 A_\mathrm{p}} = \frac{230\ \mathrm{lbf}}{\frac{1}{2}\left(0.07633\ \mathrm{lbm/ft^3}\right)\left(18\ \mathrm{ft/s}\right)^2 \pi \frac{\left(24\ \mathrm{ft}\right)^2}{4}}\left(\frac{32.2\ \mathrm{lbm\ ft}}{\mathrm{lbf\ s^2}}\right) = \mathbf{1.32}$$

(*b*) We must match model and prototype Reynolds numbers in order to achieve dynamic similarity,

Similarity:
$$\mathrm{Re_m} = \frac{\rho_\mathrm{m} V_\mathrm{m} L_\mathrm{m}}{\mu_\mathrm{m}} = \mathrm{Re_p} = \frac{\rho_\mathrm{p} V_\mathrm{p} L_\mathrm{p}}{\mu_\mathrm{p}} \qquad (1)$$

We solve Eq. 1 for the unknown wind tunnel speed,

Wind tunnel speed:
$$V_\mathrm{m} = V_\mathrm{p}\left(\frac{\mu_\mathrm{m}}{\mu_\mathrm{p}}\right)\left(\frac{\rho_\mathrm{p}}{\rho_\mathrm{m}}\right)\left(\frac{L_\mathrm{p}}{L_\mathrm{m}}\right) = \left(18\ \mathrm{ft/s}\right)(1)(1)(12) = \mathbf{216\ ft/s} \qquad (2)$$

(*c*) As discussed in the text, if the fluid is the same and dynamic similarity between the model and the prototype is achieved, the aerodynamic drag force on the model is the same as that on the prototype. Thus,

Aerodynamic drag on model:
$$F_{D,\mathrm{m}} = F_{D,\mathrm{p}} = \mathbf{230\ lbf} \qquad (3)$$

Discussion We should check that the wind tunnel speed of Eq. 2 is not too high that the incompressibility approximation becomes invalid. The Mach number at this speed is about 216/1120 = 0.193. Since this is less than 0.3, compressibility is not an issue in this model test. The drag force on the model is quite large, and a fairly hefty drag balance must be available to measure such a large force.

7-41

Solution We are to discuss why one would pressurize a wind tunnel.

Analysis As we see in some of the example problems and homework problems in this chapter, it is often difficult to achieve a high-enough wind tunnel speed to match the Reynolds number between a small model and a large prototype. Even if we were able to match the speed, the Mach number would often be too high. A pressurized wind tunnel has higher density air. At the same Reynolds number, the larger density leads to a lower air speed requirement. In other words, **a pressurized wind tunnel can achieve higher Reynolds numbers for the same scale model**.

If the pressure were to be increased by a factor of 1.8, the air density would also go up by a factor of 1.8 (ideal gas law), assuming that the air temperature remains constant. Then the Reynolds number, Re = $\rho VL/\mu$, would go up by approximately **1.8**. Note that we are also assuming that the viscosity does not change significantly with pressure, which is a reasonable assumption.

Discussion The speed of sound is not a strong function of pressure, so Mach number is not affected significantly by pressurizing the wind tunnel. However, the *power* requirement for the wind tunnel blower increases significantly as air density is increased, so this must be taken into account when designing the wind tunnel.

7-42E

Solution The concept of similarity will be utilized to determine the speed of the wind tunnel.

Assumptions **1** Compressibility of the air is ignored (the validity of this assumption will be discussed later). **2** The wind tunnel walls are far enough away so as to not interfere with the aerodynamic drag on the model car. **3** The model is geometrically similar to the prototype. **4** Both the air in the wind tunnel and the air flowing over the prototype car are at standard atmospheric pressure.

Properties For air at T = 25°C and atmospheric pressure, ρ = 1.184 kg/m^3 and μ = 1.849 × 10^{-5} kg/m·s.

Analysis Since there is only one independent Π in this problem, similarity is achieved if $\Pi_{2,m} = \Pi_{2,p}$, where Π_2 is the Reynolds number. Thus, we can write

$$\Pi_{2,\mathrm{m}} = \mathrm{Re}_\mathrm{m} = \frac{\rho_\mathrm{m} V_\mathrm{m} L_\mathrm{m}}{\mu_\mathrm{m}} = \Pi_{2,\mathrm{p}} = \mathrm{Re}_\mathrm{p} = \frac{\rho_\mathrm{p} V_\mathrm{p} L_\mathrm{p}}{\mu_\mathrm{p}}$$

which can be solved for the unknown wind tunnel speed for the model tests, V_m,

$$V_\mathrm{m} = V_\mathrm{p}\left(\frac{\mu_\mathrm{m}}{\mu_\mathrm{p}}\right)\left(\frac{\rho_\mathrm{p}}{\rho_\mathrm{m}}\right)\left(\frac{L_\mathrm{p}}{L_\mathrm{m}}\right) = (60.0\text{ mph})(1)(1)(3) = \mathbf{180\ mph}$$

Thus, to ensure similarity, **the wind tunnel should be run at 180 miles per hour** (to three significant digits).

Discussion This speed is quite high, and the wind tunnel may not be able to run at that speed. The Mach number is Ma = (180 mph)/(774 mph) = 0.23, so compressible effects can be reasonably ignored. We were never given the actual length of either car, but the ratio of L_p to L_m is known because the prototype is three times larger than the scale model. The problem statement contains a mixture of SI and English units, but it does not matter since we use ratios in the algebra.

7-43E

Solution We are to estimate the drag on a prototype car, based on aerodynamic drag measurements performed in a wind tunnel.

Assumptions **1** The model is geometrically similar. **2** The wind tunnel is run at conditions which ensure similarity between model and prototype.

Properties For air at T = 25°C and atmospheric pressure, ρ = 1.184 kg/m^3 and μ = 1.849 × 10^{-5} kg/m·s.

Analysis Following the example in the text, since the Reynolds numbers have been matched, the nondimensionalized drag coefficient of the model equals that of the prototype,

$$\frac{F_{D,\mathrm{m}}}{\rho_\mathrm{m} V_\mathrm{m}^2 L_\mathrm{m}^2} = \frac{F_{D,\mathrm{p}}}{\rho_\mathrm{p} V_\mathrm{p}^2 L_\mathrm{p}^2} \qquad (1)$$

We solve Eq. 1 for the unknown aerodynamic drag force on the prototype, $F_{D,\mathrm{p}}$,

$$F_{D,\mathrm{p}} = F_{D,\mathrm{m}}\left(\frac{\rho_\mathrm{p}}{\rho_\mathrm{m}}\right)\left(\frac{V_\mathrm{p}}{V_\mathrm{m}}\right)^2\left(\frac{L_\mathrm{p}}{L_\mathrm{m}}\right)^2 = (33.5\text{ lbf})(1)\left(\frac{60.0\text{ mph}}{180\text{ mph}}\right)^2(3)^2 = \mathbf{33.5\ lbf}$$

where we have used the wind tunnel speed calculated in the previous problem. **The predicted drag on the prototype car is 33.5 lbf** (to three significant digits).

Discussion Since the air properties of the wind tunnel are identical to those of the air flowing around the prototype car, it turns out that the aerodynamic drag force on the prototype is the same as that on the model. This would not be the case if the wind tunnel air were at a different temperature or pressure compared to that of the prototype.

7-44

Solution We are to discuss whether cold or hot air in a wind tunnel is better, and we are to support our answer by comparing air at two given temperatures.

Properties For air at atmospheric pressure and at T = 10°C, ρ = 1.246 kg/m^3 and μ = 1.778 × 10^{-5} kg/m·s. At T = 40°C, ρ = 1.127 kg/m^3 and μ = 1.918 × 10^{-5} kg/m·s.

Analysis As we see in some of the example problems and homework problems in this chapter, it is often difficult to achieve a high-enough wind tunnel speed to match the Reynolds number between a small model and a large prototype. Even if we were able to match the speed, the Mach number would often be too high. Cold air has higher density than warm air. In addition, the viscosity of cold air is lower than that of hot air. Thus, at the same Reynolds number, the colder air leads to a lower air speed requirement. In other words, **a cold wind tunnel can achieve higher Reynolds numbers than can a hot wind tunnel for the same scale model, all else being equal**. We support our conclusion by comparing air at two temperatures,

Comparison of Reynolds numbers:

$$\frac{\mathrm{Re}_{\text{cold}}}{\mathrm{Re}_{\text{hot}}} = \frac{\dfrac{\rho_{\text{cold}} VL}{\mu_{\text{cold}}}}{\dfrac{\rho_{\text{hot}} VL}{\mu_{\text{hot}}}} = \frac{\rho_{\text{cold}}}{\rho_{\text{hot}}}\frac{\mu_{\text{hot}}}{\mu_{\text{cold}}} = \left(\frac{1.246\ \text{kg/m}^3}{1.127\ \text{kg/m}^3}\right)\left(\frac{1.918\times10^{-5}\ \text{kg/m}\cdot\text{s}}{1.778\times10^{-5}\ \text{kg/m}\cdot\text{s}}\right) = \mathbf{1.19}$$

Thus we see that **the colder wind tunnel can achieve approximately 19% higher Reynolds number, all else being equal**.

Discussion There are other issues however. First of all, the denser air of the cold wind tunnel is harder to pump – the cold wind tunnel may not be able to achieve the same wind speed as the hot wind tunnel. Furthermore, the speed of sound is proportional to the square root of temperature. Thus, at colder temperatures, the Mach number is higher than at warmer temperatures for the same value of V, and compressibility effects are therefore more significant at lower temperatures.

7-45E

Solution We are to calculate the speed and angular velocity (rpm) of a spinning baseball in a water channel such that flow conditions are dynamically similar to that of the actual baseball moving and spinning in air.

Properties For air at T = 20°C and atmospheric pressure, ρ = 1.204 kg/m^3 and μ = 1.825 × 10^{-5} kg/m·s. For water at T = 20°C and atmospheric pressure, ρ = 998.0 kg/m^3 and μ = 1.002 × 10^{-3} kg/m·s.

Analysis The model (in the water) and the prototype (in the air) are actually the same baseball, so their characteristic lengths are equal, $L_m = L_p$. We match Reynolds number,

$$\mathrm{Re_m} = \frac{\rho_m V_m L_m}{\mu_m} = \mathrm{Re_p} = \frac{\rho_p V_p L_p}{\mu_p} \tag{1}$$

and solve for the required water tunnel speed for the model tests, V_m,

$$V_m = V_p\left(\frac{\mu_m}{\mu_p}\right)\left(\frac{\rho_p}{\rho_m}\right)\left(\frac{L_p}{L_m}\right) = (85.0\text{ mph})\left(\frac{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}}{1.825\times10^{-5}\text{ kg/m}\cdot\text{s}}\right)\left(\frac{1.204\text{ kg/m}^3}{998.0\text{ kg/m}^3}\right)(1) = \mathbf{5.63\,mph} \tag{2}$$

We also match Strouhal numbers, recognizing that $\dot{n}$ is proportional to f,

$$\mathrm{St_m} = \frac{f_m L_m}{V_m} = \mathrm{St_p} = \frac{f_p L_p}{V_p} \quad \rightarrow \quad \frac{\dot{n}_m L_m}{V_m} = \frac{\dot{n}_p L_p}{V_p} \tag{3}$$

from which we solve for the required spin rate in the water tunnel,

$$\dot{n}_m = \dot{n}_p\left(\frac{L_p}{L_m}\right)\left(\frac{V_m}{V_p}\right) = (320\text{ rpm})(1)\left(\frac{5.63\text{ mph}}{85.0\text{ mph}}\right) = \mathbf{21.2\,rpm} \tag{4}$$

The water tunnel needs to be run at 5.63 mph, and the baseball needs to be spun at 21.2 rpm for dynamic similarity.

Discussion Because of the difference in fluid properties between air and water, the required water tunnel speed is much lower than that in air. In addition, the spin rate is much lower, making flow visualization easier.

Dimensionless Parameters and the Method of Repeating Variables

7-46

Solution We are to verify that the Archimedes number is dimensionless.

Analysis Archimedes number is defined as

Archimedes number:
$$\mathrm{Ar} = \frac{\rho_s g L^3}{\mu^2}\left(\rho_s - \rho\right) \qquad (1)$$

We know the primary dimensions of density, gravitational acceleration, length, and viscosity. Thus,

Primary dimensions of Archimedes number:
$$\left\{\mathrm{Ar}\right\} = \left\{ \frac{\frac{\mathrm{m}}{\mathrm{L}^3}\frac{\mathrm{L}}{\mathrm{t}^2}\mathrm{L}^3}{\frac{\mathrm{m}^2}{\mathrm{L}^2\mathrm{t}^2}} \frac{\mathrm{m}}{\mathrm{L}^3} \right\} = \left\{\mathbf{1}\right\} \qquad (2)$$

Discussion If the primary dimensions were *not* unity, we would assume that we made an error in the dimensions of one or more of the parameters.

7-47

Solution We are to verify that the Grashof number is dimensionless.

Analysis Grashof number is defined as

Grashof number:
$$\mathrm{Gr} = \frac{g\beta\left|\Delta T\right| L^3 \rho^2}{\mu^2} \qquad (1)$$

We know the primary dimensions of density, gravitational acceleration, length, temperature, and viscosity. The dimensions of coefficient of thermal expansion β are 1/temperature. Thus,

Primary dimensions of Grashof number:
$$\left\{\mathrm{Gr}\right\} = \left\{ \frac{\frac{\mathrm{L}}{\mathrm{t}^2}\frac{1}{\mathrm{T}}\mathrm{T}\mathrm{L}^3\frac{\mathrm{m}^2}{\mathrm{L}^6}}{\frac{\mathrm{m}^2}{\mathrm{L}^2\mathrm{t}^2}} \right\} = \left\{\mathbf{1}\right\} \qquad (2)$$

Discussion If the primary dimensions were *not* unity, we would assume that we made an error in the dimensions of one or more of the parameters.

7-48

Solution We are to verify that the Rayleigh number is dimensionless, and determine what other established nondimensional parameter is formed by the ratio of Ra and Gr.

Analysis Rayleigh number is defined as

Rayleigh number:
$$\mathrm{Ra} = \frac{g\beta\left|\Delta T\right|L^3\rho^2 c_p}{k\mu} \qquad (1)$$

We know the primary dimensions of density, gravitational acceleration, length, temperature, and viscosity. The dimensions of coefficient of thermal expansion β are 1/temperature, those of specific heat c_p are length^2/time^2·temperature (Problem 7-14), and those of thermal conductivity k are mass·length/time^3·temperature. Thus,

Primary dimensions of Rayleigh number:
$$\{\mathrm{Ra}\} = \left\{ \frac{\frac{\mathrm{L}}{\mathrm{t}^2}\frac{1}{\mathrm{T}}\mathrm{TL}^3\frac{\mathrm{m}^2}{\mathrm{L}^6}\frac{\mathrm{L}^2}{\mathrm{t}^2\mathrm{T}}}{\frac{\mathrm{mL}}{\mathrm{t}^3\mathrm{T}}\frac{\mathrm{m}}{\mathrm{Lt}}} \right\} = \{\mathbf{1}\} \qquad (2)$$

We take the ratio of Ra and Gr:

Ratio of Rayleigh number and Grashof number:
$$\frac{\mathrm{Ra}}{\mathrm{Gr}} = \frac{\dfrac{g\beta\left|\Delta T\right|L^3\rho^2 c_p}{k\mu}}{\dfrac{g\beta\left|\Delta T\right|L^3\rho^2}{\mu^2}} = \frac{c_p\mu}{k} \qquad (3)$$

We recognize Eq. 3 as the **Prandtl number**,

Prandtl number:
$$\boxed{\mathrm{Pr} = \frac{\mathrm{Ra}}{\mathrm{Gr}} = \frac{c_p\mu}{k} = \frac{\rho c_p\mu}{\rho k} = \frac{\nu}{\alpha}} \qquad (4)$$

Discussion Many of the established nondimensional parameters are formed by the ratio or product of two (or more) other established nondimensional parameters.

7-49

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; n = 5,

List of relevant parameters: $$f_k = f\left(V, \rho, \mu, D\right) \qquad n = 5$$

Step 2 The primary dimensions of each parameter are listed,

f_k	V	ρ	μ	D
$\{\mathrm{t}^{-1}\}$	$\{\mathrm{L}^1\mathrm{t}^{-1}\}$	$\{\mathrm{m}^1\mathrm{L}^{-3}\}$	$\{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\}$	$\{\mathrm{L}^1\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since j = 3. Following the guidelines outlined in this chapter, we elect not to pick the viscosity. We choose

Repeating parameters: V, ρ, and D

Step 5 The dependent Π is generated:

$$\Pi_1 = f_k V^{a_1} \rho^{b_1} D^{c_1} \qquad \{\Pi_1\} = \left\{\left(\mathrm{t}^{-1}\right)\left(\mathrm{L}^1\mathrm{t}^{-1}\right)^{a_1}\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{b_1}\left(\mathrm{L}^1\right)^{c_1}\right\}$$

mass: $\{\mathrm{m}^0\} = \{\mathrm{m}^{b_1}\}$ $\qquad 0 = b_1 \qquad b_1 = 0$

time: $\{\mathrm{t}^0\} = \{\mathrm{t}^{-1}\mathrm{t}^{-a_1}\}$ $\qquad 0 = -1 - a_1 \qquad a_1 = -1$

length: $\{\mathrm{L}^0\} = \{\mathrm{L}^{a_1}\mathrm{L}^{-3b_1}\mathrm{L}^{c_1}\}$ $\qquad 0 = a_1 - 3b_1 + c_1 \qquad c_1 = 1$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{f_k D}{V} = \mathrm{St}$$

where we have identified this Pi as the **Strouhal number**.

The second Pi (the only independent Π in this problem) is generated:

$$\Pi_2 = \mu V^{a_2} \rho^{b_2} D^{c_2} \qquad \{\Pi_2\} = \left\{\left(\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\right)\left(\mathrm{L}^1\mathrm{t}^{-1}\right)^{a_2}\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{b_2}\left(\mathrm{L}^1\right)^{c_2}\right\}$$

mass: $\{\mathrm{m}^0\} = \{\mathrm{m}^1\mathrm{m}^{b_2}\}$ $\qquad 0 = 1 + b_2 \qquad b_2 = -1$

time: $\{\mathrm{t}^0\} = \{\mathrm{t}^{-1}\mathrm{t}^{-a_2}\}$ $\qquad 0 = -1 - a_2 \qquad a_2 = -1$

length: $\{L^0\} = \{L^{-1} L^{a_2} L^{-3b_2} L^{c_2}\}$ $\quad 0 = -1 + a_2 - 3b_2 + c_2$ $\quad c_2 = -1$

$0 = -1 - 1 + 3 + c_2$

which yields

Π_2: $$\Pi_2 = \frac{\mu}{\rho V D}$$

We recognize this Π as the inverse of the **Reynolds number**. So, after inverting,

Modified Π_2: $$\Pi_2 = \frac{\rho V D}{\mu} = \text{Reynolds number} = \text{Re}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\text{St} = f(\text{Re})}$$

Discussion We cannot tell from dimensional analysis the exact form of the functional relationship. However, experiments verify that the Strouhal number is indeed a function of Reynolds number.

7-50

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are six parameters in this problem; $n = 6$,

List of relevant parameters: $$f_k = f\left(V, \rho, \mu, D, c\right) \qquad n = 6 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

f_k	V	ρ	μ	D	c
$\{t^{-1}\}$	$\{L^1t^{-1}\}$	$\{m^1L^{-3}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^1\}$	$\{L^1t^{-1}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 6 - 3 = 3$$

Step 4 We need to choose three repeating parameters since $j = 3$. Following the guidelines outlined in this chapter, we elect not to pick the viscosity. We choose

Repeating parameters: V, ρ, and D

Step 5 The dependent Π is generated:

$$\Pi_1 = f_k V^{a_1} \rho^{b_1} D^{c_1} \qquad \{\Pi_1\} = \left\{\left(t^{-1}\right)\left(L^1t^{-1}\right)^{a_1}\left(m^1L^{-3}\right)^{b_1}\left(L^1\right)^{c_1}\right\}$$

mass: $\{m^0\} = \{m^{b_1}\}$ $\qquad 0 = b_1 \qquad b_1 = 0$

time: $\{t^0\} = \{t^{-1}t^{-a_1}\}$ $\qquad 0 = -1 - a_1 \qquad a_1 = -1$

length: $\{L^0\} = \{L^{a_1}L^{-3b_1}L^{c_1}\}$ $\qquad 0 = a_1 - 3b_1 + c_1 \qquad c_1 = 1$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{f_k D}{V} = \mathrm{St}$$

where we have identified this Pi as the **Strouhal number**.

The second Pi (the first independent Π in this problem) is generated:

$$\Pi_2 = \mu V^{a_2} \rho^{b_2} D^{c_2} \qquad \{\Pi_2\} = \left\{\left(m^1L^{-1}t^{-1}\right)\left(L^1t^{-1}\right)^{a_2}\left(m^1L^{-3}\right)^{b_2}\left(L^1\right)^{c_2}\right\}$$

mass: $\{m^0\} = \{m^1m^{b_2}\}$ $\qquad 0 = 1 + b_2 \qquad b_2 = -1$

time: $\{t^0\} = \{t^{-1}t^{-a_2}\}$ $\qquad 0 = -1 - a_2 \qquad a_2 = -1$

length: $\{L^0\} = \{L^{-1}L^{a_2}L^{-3b_2}L^{c_2}\}$ $\quad 0 = -1 + a_2 - 3b_2 + c_2$ $\quad 0 = -1 - 1 + 3 + c_2$ $\quad c_2 = -1$

which yields

Π_2: $$\Pi_2 = \frac{\mu}{\rho VD}$$

We recognize this Π as the inverse of the **Reynolds number**. So, after inverting,

Modified Π_2: $$\Pi_2 = \frac{\rho VD}{\mu} = \text{Reynolds number} = \text{Re}$$

The third Pi (the second independent Π in this problem) is generated:

$$\Pi_3 = cV^{a_3}\rho^{b_3}D^{c_3} \qquad \{\Pi_3\} = \left\{(L^1t^{-1})(L^1t^{-1})^{a_3}(m^1L^{-3})^{b_3}(L^1)^{c_3}\right\}$$

mass: $\{m^0\} = \{m^{b_3}\}$ $\quad 0 = b_3$ $\quad b_3 = 0$

time: $\{t^0\} = \{t^{-1}t^{-a_3}\}$ $\quad 0 = -1 - a_3$ $\quad a_3 = -1$

length: $\{L^0\} = \{L^1L^{a_3}L^{-3b_3}L^{c_3}\}$ $\quad 0 = 1 + a_3 - 3b_3 + c_3$ $\quad 0 = 1 - 1 + c_3$ $\quad c_3 = 0$

which yields

Π_3: $$\Pi_3 = \frac{c}{V}$$

We recognize this Π as the inverse of the **Mach number**. So, after inverting,

Modified Π_3: $$\Pi_3 = \frac{V}{c} = \text{Mach number} = \text{Ma}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\text{St} = f(\text{Re}, \text{Ma})}$$

Discussion We have shown all the details. After you become comfortable with the method of repeating variables, you can do some of the algebra in your head.

7-51

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; n = 5,

List of relevant parameters: $$\dot{W} = f\left(\omega, \rho, \mu, D\right) \qquad n = 5 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

$\dot{W}$	ω	ρ	μ	D
$\{m^1L^2t^{-3}\}$	$\{t^{-1}\}$	$\{m^1L^{-3}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^1\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since j = 3. Following the guidelines outlined in this chapter, we elect not to pick the viscosity. We choose

Repeating parameters: ω, ρ, and D

Step 5 The dependent Π is generated:

$$\Pi_1 = \dot{W}\omega^{a_1}\rho^{b_1}D^{c_1} \qquad \{\Pi_1\} = \left\{\left(m^1L^2t^{-3}\right)\left(t^{-1}\right)^{a_1}\left(m^1L^{-3}\right)^{b_1}\left(L^1\right)^{c_1}\right\}$$

mass: $\{m^0\} = \{m^1m^{b_1}\}$ $\qquad 0 = 1 + b_1 \qquad b_1 = -1$

time: $\{t^0\} = \{t^{-3}t^{-a_1}\}$ $\qquad 0 = -3 - a_1 \qquad a_1 = -3$

length: $\{L^0\} = \{L^2L^{-3b_1}L^{c_1}\}$ $\qquad 0 = 2 - 3b_1 + c_1 \qquad c_1 = -5$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{\dot{W}}{\rho D^5\omega^3} = N_P$$

where we have defined this Pi as the **power number** (Table 7-5).

The second Pi (the only independent Π in this problem) is generated:

$$\Pi_2 = \mu\omega^{a_2}\rho^{b_2}D^{c_2} \qquad \{\Pi_2\} = \left\{\left(m^1L^{-1}t^{-1}\right)\left(t^{-1}\right)^{a_2}\left(m^1L^{-3}\right)^{b_2}\left(L^1\right)^{c_2}\right\}$$

mass: $\{m^0\} = \{m^1m^{b_2}\}$ $\qquad 0 = 1 + b_2 \qquad b_2 = -1$

time: $\{t^0\} = \{t^{-1}t^{-a_2}\}$ $\quad 0 = -1 - a_2 \quad a_2 = -1$

length: $\{L^0\} = \{L^{-1}L^{-3b_2}L^{c_2}\}$ $\quad 0 = -1 - 3b_2 + c_2$, $\quad 0 = -1 + 3 + c_2 \quad c_2 = -2$

which yields

Π_2:
$$\Pi_2 = \frac{\mu}{\rho D^2 \omega}$$

Since Dω is the speed of the tip of the rotating stirrer blade, we recognize this Π as the inverse of a **Reynolds number**. So, after inverting,

Modified Π_2:
$$\Pi_2 = \frac{\rho D^2 \omega}{\mu} = \frac{\rho(D\omega)D}{\mu} = \text{Reynolds number} = \text{Re}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:
$$\boxed{N_P = f(\text{Re})} \qquad (2)$$

Discussion After some practice you should be able to do some of the algebra with the exponents in your head. Also, we usually expect a type of Reynolds number when we combine viscosity with a density, a length, and some kind of speed, be it angular speed or linear speed.

7-52

Solution We are to determine the dimensionless relationship between the given parameters

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The dimensional analysis is identical to the previous problem except that we add two additional independent parameters, both of which have dimensions of length. The two Πs of the previous problem remain. We get two additional Πs since n is now equal to 7 instead of 5. There is no need to go through all the algebra for the two additional Πs – since their dimensions match those of one of the repeating variables (D), we know that all the exponents in the Π will be zero except the exponent for D, which will be –1. The two additional Πs are

Π_3 and Π_4:

$$\Pi_3 = \frac{D_{\text{tank}}}{D} \qquad \Pi_4 = \frac{h_{\text{surface}}}{D}$$

The final functional relationship is

Relationship between Π*s*:

$$\boxed{N_P = f\left(\text{Re}, \frac{D_{\text{tank}}}{D}, \frac{h_{\text{surface}}}{D}\right)} \qquad (1)$$

Discussion We could also manipulate our Πs so that we have other length ratios like $h_{\text{surface}}/D_{\text{tank}}$, etc. Any such combination is acceptable.

7-53

Solution We are to use dimensional analysis to find the functional relationship between the given parameters, namely, energy E as a function of mass m and speed of light c.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters.

Step 1 There are three parameters in this problem; $n = 3$,

List of relevant parameters: $$E = \text{function}(m, c) \qquad n = 3$$

Step 2 The primary dimensions of each parameter are listed,

E	m	c
$\{\mathrm{m}^1\mathrm{L}^2\mathrm{t}^{-2}\}$	$\{\mathrm{m}^1\}$	$\{\mathrm{L}^1\mathrm{t}^{-1}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $j = 3$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 3 - 3 = 0$$

This is clearly incorrect (we cannot have zero Πs). So we reduce j by one,

Reduction: $j = 2$

and the expected number of Πs is

Number of expected Πs: $$k = n - j = 3 - 2 = 1$$

Step 4 We need to choose two repeating parameters since $j = 2$. Following the guidelines, we have only one choice,

Repeating parameters: m and c

Step 5 The dependent Π is generated:

$$\Pi_1 = Em^{a_1}c^{b_1} \qquad \{\Pi_1\} = \left\{\left(\mathrm{m}^1\mathrm{L}^2\mathrm{t}^{-2}\right)\left(\mathrm{m}^1\right)^{a_1}\left(\mathrm{L}^1\mathrm{t}^{-1}\right)^{b_1}\right\}$$

time: $$\{\mathrm{t}^0\} = \{\mathrm{t}^{-2}\mathrm{t}^{-b_1}\} \qquad 0 = -2 - b_1 \qquad b_1 = -2$$

mass: $$\{\mathrm{m}^0\} = \{\mathrm{m}^1\mathrm{m}^{a_1}\} \qquad 0 = 1 + a_1 \qquad a_1 = -1$$

length: $$\{\mathrm{L}^0\} = \{\mathrm{L}^2\mathrm{L}^{b_1}\} \qquad 0 = 2 + b_1 \qquad b_1 = -2$$

Thus,

Π_1: $$\Pi_1 = \frac{E}{mc^2}$$

Step 6 Since there is only one Π, it must be a constant and we write the final functional relationship as

Relationship between Π*s*: $$\boxed{\Pi_1 = \frac{E}{mc^2} = \text{constant},\ \ \text{or}\ \ E = \text{constant}\cdot mc^2}$$

Comparing to Einstein's famous equation, $\boxed{E = mc^2}$, we see that the constant is unity.

Discussion In this case, since the number of variables is small, we were able to generate the exact form of Einstein's equation to within an unknown constant. The constant cannot be obtained through dimensional analysis alone.

7-54

Solution We are to create a scale for volume flow rate and then define an appropriate Richardson number.

Analysis By "back of the envelope" reasoning (or by inspection), we define a volume flow rate scale as L^2V. Then the Richardson number can be defined as

Richardson number:
$$\boxed{\mathrm{Ri} = \frac{L^5 g\Delta\rho}{\rho\dot{V}^2} = \frac{L^5 g\Delta\rho}{\rho\left(L^2V\right)^2} = \frac{Lg\Delta\rho}{\rho V^2}} \qquad (1)$$

Discussion It is perhaps more clear from the form of Eq. 1 that Richardson number is a ratio of buoyancy forces to inertial forces.

7-55

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are six parameters in this problem; $n = 6$,

List of relevant parameters: $$u = f\left(\mu, V, h, \rho, y\right) \qquad n = 6 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

u	μ	V	h	ρ	y
$\{L^1t^{-1}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^1t^{-1}\}$	$\{L^1\}$	$\{m^1L^{-3}\}$	$\{L^1\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 6 - 3 = 3$$

Step 4 We need to choose three repeating parameters since $j = 3$. Following the guidelines outlined in this chapter, we elect not to pick the viscosity. It is better to pick a fixed length (h) rather than a variable length (y); otherwise y would appear in each Pi, which would not be desirable. We choose

Repeating parameters: V, ρ, and h

Step 5 The dependent Π is generated:

$$\Pi_1 = uV^{a_1}\rho^{b_1}h^{c_1} \qquad \{\Pi_1\} = \left\{\left(L^1t^{-1}\right)\left(L^1t^{-1}\right)^{a_1}\left(m^1L^{-3}\right)^{b_1}\left(L^1\right)^{c_1}\right\}$$

mass: $$\{m^0\} = \{m^{b_1}\} \qquad 0 = b_1 \qquad b_1 = 0$$

time: $$\{t^0\} = \{t^{-1}t^{-a_1}\} \qquad 0 = -1 - a_1 \qquad a_1 = -1$$

length: $$\{L^0\} = \{L^1L^{a_1}L^{-3b_1}L^{c_1}\} \qquad 0 = 1 + a_1 - 3b_1 + c_1 \qquad c_1 = 0$$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{u}{V}$$

The second Pi (the first independent Π in this problem) is generated:

$$\Pi_2 = \mu V^{a_2}\rho^{b_2}h^{c_2} \qquad \{\Pi_2\} = \left\{\left(m^1L^{-1}t^{-1}\right)\left(L^1t^{-1}\right)^{a_2}\left(m^1L^{-3}\right)^{b_2}\left(L^1\right)^{c_2}\right\}$$

mass: $$\{m^0\} = \{m^1m^{b_2}\} \qquad 0 = 1 + b_2 \qquad b_2 = -1$$

time: $\left\{t^0\right\} = \left\{t^{-1}t^{-a_2}\right\}$ $\quad 0 = -1 - a_2 \quad a_2 = -1$

length: $\left\{L^0\right\} = \left\{L^{-1}L^{a_2}L^{-3b_2}L^{c_2}\right\}$ $\quad 0 = -1 + a_2 - 3b_2 + c_2$ $\quad c_2 = -1$

$0 = -1 - 1 + 3 + c_2$

which yields

Π_2:
$$\Pi_2 = \frac{\mu}{\rho V h}$$

We recognize this Π as the inverse of the **Reynolds number**. So, after inverting,

Modified Π_2:
$$\Pi_2 = \frac{\rho V h}{\mu} = \text{Reynolds number} = \text{Re}$$

The third Pi (the second independent Π in this problem) is generated:

$$\Pi_3 = yV^{a_3}\rho^{b_3}h^{c_3} \qquad \left\{\Pi_3\right\} = \left\{\left(L^1\right)\left(L^1t^{-1}\right)^{a_3}\left(m^1L^{-3}\right)^{b_3}\left(L^1\right)^{c_3}\right\}$$

mass: $\left\{m^0\right\} = \left\{m^{b_3}\right\}$ $\quad 0 = b_3 \quad b_3 = 0$

time: $\left\{t^0\right\} = \left\{t^{-a_3}\right\}$ $\quad 0 = -a_3 \quad a_3 = 0$

length: $\left\{L^0\right\} = \left\{L^1L^{a_3}L^{-3b_3}L^{c_3}\right\}$ $\quad 0 = 1 + a_3 - 3b_3 + c_3$ $\quad c_3 = -1$

$0 = 1 + c_3$

which yields

Π_3:
$$\Pi_3 = \frac{y}{h}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:
$$\boxed{\frac{u}{V} = f\left(\text{Re}, \frac{y}{h}\right)} \qquad (2)$$

Discussion We notice in the first and third Πs that when the parameter on which we are working has the same dimensions as one of the repeating parameters, the Π is simply the ratio of those two parameters (here u/V and y/h).

7-56

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are seven parameters in this problem; n = 7,

List of relevant parameters:
$$u = f\left(\mu, V, h, \rho, y, t\right) \qquad n = 7 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

u	μ	V	h	ρ	y	t
$\{L^1t^{-1}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^1t^{-1}\}$	$\{L^1\}$	$\{m^1L^{-3}\}$	$\{L^1\}$	$\{t^1\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction:
$$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs:
$$k = n - j = 7 - 3 = 4$$

Step 4 We need to choose three repeating parameters since j = 3. Following the guidelines outlined in this chapter, we do not pick the viscosity. It is better to pick a fixed length (h) rather than a variable length (y); otherwise y would appear in each Pi, which would not be desirable. It would also not be wise to have time appear in each parameter. We choose

Repeating parameters: V, ρ, and h

Step 5 The Πs are generated. The first three Πs are identical to those of the previous problem, so we do not include the details here. The fourth Π is formed by joining the new parameter t to the repeating variables,

$$\Pi_4 = tV^{a_4}\rho^{b_4}h^{c_4} \qquad \{\Pi_4\} = \left\{\left(t^1\right)\left(L^1t^{-1}\right)^{a_4}\left(m^1L^{-3}\right)^{b_4}\left(L^1\right)^{c_4}\right\}$$

mass: $\{m^0\} = \{m^{b_4}\}$ $\qquad 0 = b_4 \qquad b_4 = 0$

time: $\{t^0\} = \{t^1t^{-a_4}\}$ $\qquad 0 = 1 - a_4 \qquad a_4 = 1$

length: $\{L^0\} = \{L^{a_1}L^{-3b_4}L^{c_4}\}$ $\qquad 0 = a_4 - 3b_4 + c_4 \qquad c_4 = -1$

This Π is thus

Π_4:
$$\Pi_4 = \frac{tV}{h}$$

Step 6 Combining this result with the first three Πs from the previous problem,

Relationship between Πs:
$$\boxed{\frac{u}{V} = f\left(\mathrm{Re}, \frac{y}{h}, \frac{tV}{h}\right)} \qquad (2)$$

Discussion As $t \to \infty$, Π_4 becomes irrelevant and the result degenerates into that of the previous problem.

7-57

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are four parameters in this problem; $n = 4$,

List of relevant parameters: $$c = f\left(k, T, R_{\text{gas}}\right) \qquad n = 4 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed; the ratio of specific heats k is dimensionless.

$$\begin{array}{cccc} c & k & T & R_{\text{gas}} \\ \left\{\mathrm{L}^1\mathrm{t}^{-1}\right\} & \left\{1\right\} & \left\{\mathrm{T}^1\right\} & \left\{\mathrm{L}^2\mathrm{t}^{-2}\mathrm{T}^{-1}\right\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (T, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 4 - 3 = 1$$

Thus we expect only one Π.

Step 4 We need to choose three repeating parameters since $j = 3$. We only have one choice in this problem, since there are only three independent parameters on the right-hand side of Eq. 1. However, one of these is already dimensionless, so it is a Π all by itself. In this situation we reduce j by one and continue,

Reduction: $$j = 3 - 1 = 2$$

If this revised value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 4 - 2 = 2$$

We now expect two Πs. We choose two repeating parameters since $j = 2$,

Repeating parameters: T and R_{gas}

Step 5 The dependent Π is generated:

$$\Pi_1 = c T^{a_1} R_{\text{gas}}{}^{b_1} \qquad \left\{\Pi_1\right\} = \left\{\left(\mathrm{L}^1\mathrm{t}^{-1}\right)\left(\mathrm{T}^1\right)^{a_1}\left(\mathrm{L}^2\mathrm{t}^{-2}\mathrm{T}^{-1}\right)^{b_1}\right\}$$

time: $$\left\{\mathrm{t}^0\right\} = \left\{\mathrm{t}^{-1-2b_1}\right\} \qquad 0 = -1 - 2b_1 \qquad b_1 = -1/2$$

temperature: $$\left\{\mathrm{T}^0\right\} = \left\{\mathrm{T}^{a_1 - b_1}\right\} \qquad a_1 = b_1 \qquad a_1 = -1/2$$

length: $$\left\{\mathrm{L}^0\right\} = \left\{\mathrm{L}^1\mathrm{L}^{2b_1}\right\} \qquad 0 = 1 + 2b_1 \qquad b_1 = -1/2$$

Fortunately the two results for exponent b_1 agree. The dependent Π is thus

Π_1: $$\Pi_1 = \frac{c}{\sqrt{R_{\text{gas}} T}}$$

The independent Π is already known,

Π_2: $$\Pi_2 = k$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\frac{c}{\sqrt{R_{\text{gas}}T}} = f\left(k\right)} \qquad (2)$$

Discussion We cannot tell from dimensional analysis the exact form of the functional relationship. However, in this case the result agrees with the known equation for speed of sound in an ideal gas, $c = \sqrt{kR_{\text{gas}}T}$.

7-58

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; $n = 5$,

List of relevant parameters: $$c = f\left(k, T, R_u, M\right) \qquad n = 5 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed; the ratio of specific heats k is dimensionless.

c	k	T	R_u	M
$\{\mathrm{L}^1\mathrm{t}^{-1}\}$	$\{1\}$	$\{\mathrm{T}^1\}$	$\{\mathrm{m}^1\mathrm{L}^2\mathrm{t}^{-2}\mathrm{T}^{-1}\mathrm{N}^{-1}\}$	$\{\mathrm{m}^1\mathrm{N}^{-1}\}$

Step 3 As a first guess, j is set equal to 5, the number of primary dimensions represented in the problem (m, T, L, N, and t).

Reduction: $$j = 5$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 5 - 5 = 0$$

Obviously we cannot have zero Πs. We check that we have not missed a relevant parameter. Convinced that we have included all the relevant parameters we reduce j by 1:

Reduction: $$j = 5 - 1 = 4$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 5 - 4 = 1$$

Step 4 We need to choose four repeating parameters since $j = 4$. We only have one choice in this problem, since there are only four independent parameters on the right-hand side of Eq. 1. However, one of these is already dimensionless, so it is a Π all by itself. In this situation we reduce j by one (*again*) and continue,

Reduction: $$j = 4 - 1 = 3$$

If this revised value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 5 - 3 = 2$$

We now expect two Πs. Since $j = 3$ we choose three repeating parameters,

Repeating parameters: T, M, and R_u

Step 5 The dependent Π is generated:

$$\Pi_1 = cT^{a_1}M^{b_1}R_u^{c_1} \qquad \{\Pi_1\} = \left\{\left(\mathrm{L}^1\mathrm{t}^{-1}\right)\left(\mathrm{T}^1\right)^{a_1}\left(\mathrm{m}^1\mathrm{n}^{-1}\right)^{b_1}\left(\mathrm{m}^1\mathrm{L}^2\mathrm{t}^{-2}\mathrm{T}^{-1}\mathrm{N}^{-1}\right)^{c_1}\right\}$$

time: $$\{\mathrm{t}^0\} = \{\mathrm{t}^{-1}\mathrm{t}^{-2c_1}\} \qquad 0 = -1 - 2c_1 \qquad c_1 = -1/2$$

mass: $$\{\mathrm{m}^0\} = \{\mathrm{m}^{b_1}\mathrm{m}^{c_1}\} \qquad 0 = b_1 + c_1 \quad b_1 = -c_1 \qquad b_1 = 1/2$$

amount of matter:	$\left\{N^0\right\} = \left\{N^{-b_1} N^{-c_1}\right\}$	$0 = -b_1 - c_1 \quad b_1 = -c_1$	$b_1 = 1/2$
temperature:	$\left\{T^0\right\} = \left\{T^{a_1} T^{-c_1}\right\}$	$0 = a_1 - c_1 \quad a_1 = c_1$	$a_1 = -1/2$
length:	$\left\{L^0\right\} = \left\{L^1 L^{2c_1}\right\}$	$0 = 1 + 2c_1$	$c_1 = -1/2$

Fortunately the two results for exponent b_1 agree, and the two results for exponent c_1 agree. (If they did not agree, we would search for algebra mistakes. Finding none we would suspect that *j* is not correct or that we are missing a relevant parameter in the problem.) The dependent Π is thus

Π_1:
$$\Pi_1 = \frac{c\sqrt{M}}{\sqrt{R_u T}}$$

The independent Π is already known,

Π_2:
$$\Pi_2 = k$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:
$$\boxed{\Pi_1 = \frac{c\sqrt{M}}{\sqrt{R_u T}} = f\left(k\right)} \qquad (2)$$

Discussion Since we know that $R_{gas} = R_u/M$, we see that the result here is the same as that of the previous problem.

7-59

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are three parameters in this problem; $n = 3$,

List of relevant parameters: $$c = f\left(T, R_{\text{gas}}\right) \qquad n = 3 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

$$\begin{array}{ccc} c & T & R_{\text{gas}} \\ \left\{\mathrm{L}^1\mathrm{t}^{-1}\right\} & \left\{\mathrm{T}^1\right\} & \left\{\mathrm{L}^2\mathrm{t}^{-2}\mathrm{T}^{-1}\right\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (T, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 3 - 3 = 0$$

Obviously this is not correct, so we re-examine our initial assumptions. We can add another variable, k (the ratio of specific heats) to our List of relevant parameters. This problem would then be identical to Problem 7-58. Instead, for instructional purposes we reduce j by one and continue,

Reduction: $$j = 3 - 1 = 2$$

If this revised value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 3 - 2 = 1$$

We now expect only one Π.

Step 4 We need to choose two repeating parameters since $j = 2$. We only have one choice in this problem, since there are only two independent parameters on the right-hand side of Eq. 1,

Repeating parameters: T and R_{gas}

Step 5 The dependent Π is generated:

$$\Pi_1 = cT^{a_1}R_{\text{gas}}{}^{b_1} \qquad \left\{\Pi_1\right\} = \left\{\left(\mathrm{L}^1\mathrm{t}^{-1}\right)\left(\mathrm{T}^1\right)^{a_1}\left(\mathrm{L}^2\mathrm{t}^{-2}\mathrm{T}^{-1}\right)^{b_1}\right\}$$

time: $$\left\{\mathrm{t}^0\right\} = \left\{\mathrm{t}^{-1-2b_1}\right\} \qquad 0 = -1 - 2b_1 \qquad b_1 = -1/2$$

temperature: $$\left\{\mathrm{T}^0\right\} = \left\{\mathrm{T}^{a_1-b_1}\right\} \qquad a_1 = b_1 \qquad a_1 = -1/2$$

length: $$\left\{\mathrm{L}^0\right\} = \left\{\mathrm{L}^1\mathrm{L}^{2b_1}\right\} \qquad 0 = 1 + 2b_1 \qquad b_1 = -1/2$$

Fortunately the two results for exponent b_1 agree. The dependent Π is thus

Π_1:

$$\Pi_1 = \frac{c}{\sqrt{R_{\text{gas}}T}}$$

Step 6 Since there is only one Π, it is a function of nothing. This is only possible if we set the Π equal to a constant. We write the final functional relationship as

Relationship between Π*s*:

$$\boxed{\Pi_1 = \frac{c}{\sqrt{R_{\text{gas}}T}} = \text{constant}} \qquad (2)$$

Discussion Our result represents an interesting case of "luck". Although we failed to include the ratio of specific heats k in our analysis, we nevertheless obtain the correct result. In fact, if we set the constant in Eq. 2 as the square root of k, our result agrees with the known equation for speed of sound in an ideal gas, $c = \sqrt{kR_{\text{gas}}T}$.

7-60

Solution We are to use dimensional analysis to find the functional relationship between the given parameters, and compare to the known equation for an ideal gas.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are three parameters in this problem; $n = 3$,

List of relevant parameters: $$c = f(P, \rho) \qquad n = 3 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

$$\begin{array}{ccc} c & P & \rho \\ \{\mathrm{L}^1\mathrm{t}^{-1}\} & \{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-2}\} & \{\mathrm{m}^1\mathrm{L}^{-3}\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 3 - 3 = 0$$

Obviously this is not correct, so we re-examine our initial assumptions. If we are convinced that c is a function of only P and ρ, we reduce j by one and continue,

Reduction: $$j = 3 - 1 = 2$$

If this revised value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 3 - 2 = 1$$

We now expect only one Π.

Step 4 We need to choose two repeating parameters since $j = 2$. We only have one choice in this problem, since there are only two independent parameters on the right-hand side of Eq. 1,

Repeating parameters: P and ρ

Step 5 The dependent Π is generated:

$$\Pi_1 = cP^{a_1}\rho^{b_1} \qquad \{\Pi_1\} = \left\{\left(\mathrm{L}^1\mathrm{t}^{-1}\right)\left(\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-2}\right)^{a_1}\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{b_1}\right\}$$

time: $$\{\mathrm{t}^0\} = \{\mathrm{t}^{-1}\mathrm{t}^{-2a_1}\} \qquad 0 = -1 - 2a_1 \qquad a_1 = -1/2$$

mass: $$\{\mathrm{m}^0\} = \{\mathrm{m}^{a_1}\mathrm{m}^{b_1}\} \qquad 0 = a_1 + b_1 \qquad b_1 = 1/2$$

length: $$\{\mathrm{L}^0\} = \{\mathrm{L}^1\mathrm{L}^{-2a_1}\mathrm{L}^{-3b_1}\} \qquad \begin{array}{l} 0 = 1 - a_1 - 3b_1 \\ 0 = 1 + \frac{1}{2} - \frac{3}{2} \end{array} \qquad 0 = 0$$

Fortunately the exponents for length agree with those of mass and time. The dependent Π is thus

Π_1: $$\Pi_1 = c\sqrt{\frac{\rho}{P}}$$

Step 6 Since there is only one Π, it is a function of nothing. This is only possible if we set the Π equal to a constant. We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\Pi_1 = c\sqrt{\frac{\rho}{P}} = \text{constant, or } c = \text{constant}\sqrt{\frac{P}{\rho}}} \qquad (2)$$

The ideal gas equation is $P = \rho R_{\text{gas}}T$, or $P/\rho = R_{\text{gas}}T$. Thus, Eq. 2 can be written as

Alternative result using ideal gas law: $$\boxed{c = \text{constant}\sqrt{R_{\text{gas}}T}} \qquad (3)$$

Equation 3 is indeed consistent with the equation $c = \sqrt{kR_{\text{gas}}T}$.

Discussion There is no way to obtain the value of the constant in Eq. 2 or 3 solely by dimensional analysis, but it turns out that the constant is the square root of *k*.

7-61

Solution We are to use dimensional analysis to find a functional relationship between F_D and variables V, L, and μ.

Assumptions **1** We assume Re << 1 so that the creeping flow approximation applies. **2** Gravitational effects are irrelevant. **3** No parameters other than those listed in the problem statement are relevant to the problem.

Analysis We follow the step-by-step method of repeating variables.

Step 1 There are four variables and constants in this problem; n = 4. They are listed in functional form, with the dependent variable listed as a function of the independent variables and constants:

List of relevant parameters: $$F_D = f\left(V, L, \mu\right) \qquad n = 4$$

Step 2 The primary dimensions of each parameter are listed.

F_D	V	L	μ
$\left\{\mathrm{m}^1\mathrm{L}^1\mathrm{t}^{-2}\right\}$	$\left\{\mathrm{L}^1\mathrm{t}^{-1}\right\}$	$\left\{\mathrm{L}^1\right\}$	$\left\{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\right\}$

Step 3 As a first guess, we set j equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the number of Πs expected is

Number of expected Πs: $$k = n - j = 4 - 3 = 1$$

Step 4 Now we need to choose three repeating parameters since j = 3. Since we cannot choose the dependent variable, our only choices are V, L, and μ.

Step 5 Now we combine these repeating parameters into a product with the dependent variable F_D to create the dependent Π,

Dependent Π: $$\Pi_1 = F_D V^{a_1} L^{b_1} \mu^{c_1} \qquad (1)$$

We apply the primary dimensions of Step 2 into Eq. 1 and force the Π to be dimensionless,

Dimensions of Π_1:
$$\left\{\Pi_1\right\} = \left\{\mathrm{m}^0\mathrm{L}^0\mathrm{t}^0\right\} = \left\{F_D V^{a_1} L^{b_1} \mu^{c_1}\right\} = \left\{\left(\mathrm{m}^1\mathrm{L}^1\mathrm{t}^{-2}\right)\left(\mathrm{L}^1\mathrm{t}^{-1}\right)^{a_1}\left(\mathrm{L}^1\right)^{b_1}\left(\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\right)^{c_1}\right\}$$

Now we equate the exponents of each primary dimension to solve for exponents a_1 through c_1.

mass: $\left\{\mathrm{m}^0\right\} = \left\{\mathrm{m}^1\mathrm{m}^{c_1}\right\}$ $\qquad 0 = 1 + c_1 \qquad c_1 = -1$

time: $\left\{\mathrm{t}^0\right\} = \left\{\mathrm{t}^{-2}\mathrm{t}^{-a_1}\mathrm{t}^{-c_1}\right\}$ $\qquad 0 = -2 - a_1 - c_1 \qquad a_1 = -1$

length: $\left\{\mathrm{L}^0\right\} = \left\{\mathrm{L}^1\mathrm{L}^{a_1}\mathrm{L}^{b_1}\mathrm{L}^{-c_1}\right\}$ $\qquad 0 = 1 + a_1 + b_1 - c_1 \qquad b_1 = -1$

Equation 1 thus becomes

Π_1: $$\Pi_1 = \frac{F_D}{\mu V L} \qquad (2)$$

Step 6 We now write the functional relationship between the nondimensional parameters. In the case at hand, there is only one Π, which is a function of *nothing*. This is possible only if the Π is constant. Putting Eq. 2 into standard functional form, *Relationship between* Πs: $$\Pi_1 = \frac{F_D}{\mu VL} = f\left(\text{nothing}\right) = \text{constant} \qquad (3)$$

or

Result of dimensional analysis: $$\boxed{F_D = \text{constant} \cdot \mu VL} \qquad (4)$$

Thus we have shown that for creeping flow around an object, the aerodynamic drag force is simply a constant multiplied by μVL, regardless of the shape of the object.

Discussion This result is very significant because all that is left to do is find the constant, which will be a function of the shape of the object (and its orientation with respect to the flow).

7-62

Solution We are to find the functional relationship between the given parameters and name any established dimensionless parameters.

Assumptions **1** The given parameters are the only ones relevant to the flow at hand.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; n = 5,

List of relevant parameters: $$V = f\left(d_p, \left(\rho_p - \rho\right), \mu, g\right) \qquad n = 5 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

V	d_p	$(\rho_p - \rho)$	μ	g
$\{L^1t^{-1}\}$	$\{L^1\}$	$\{m^1L^{-3}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^1t^{-2}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since j = 3. We pick length scale d_p, density difference (ρ_p - ρ), and gravitational constant g.

Repeating parameters: $$d_p,\ \left(\rho_p\text{-}\rho\right), \text{ and } g$$

Step 5 The Πs are generated. Note that for the first Π we do the algebra in our heads since the relationship is very simple. The dependent Π is

Π_1 = *a Froude number*: $$\Pi_1 = \frac{V}{\sqrt{gd_p}}$$

This Π is a type of **Froude number**. Similarly, the Π formed with viscosity is generated,

$$\Pi_2 = \mu d_p^{\,a}\left(\rho_p - \rho\right)^b g^c \qquad \{\Pi_2\} = \left\{\left(m^1L^{-1}t^{-1}\right)\left(L^1\right)^a\left(m^1L^{-3}\right)^b\left(L^1t^{-2}\right)^c\right\}$$

mass: $$\{m^0\} = \{m^1m^b\} \qquad 0 = 1 + b \qquad b = -1$$

time: $$\{t^0\} = \{t^{-1}t^{-2c}\} \qquad 0 = -1 - 2c \qquad c = -\frac{1}{2}$$

length: $$\{L^0\} = \{L^{-1}L^aL^{-3b}L^c\} \qquad \begin{aligned} 0 &= -1 + a - 3b + c \\ 0 &= -1 + a + 3 - \tfrac{1}{2} \end{aligned} \qquad a = -\frac{3}{2}$$

which yields

Π_2:

$$\Pi_2 = \frac{\mu}{\left(\rho_p - \rho\right) d_p^{\frac{3}{2}} \sqrt{g}}$$

We recognize this Π as the inverse of a kind of **Reynolds number** if we split the d_p terms to separate them into a length scale and (when combined with g) a velocity scale. The final form is

Modified Π_2 = *a Reynolds number*:

$$\Pi_2 = \frac{\left(\rho_p - \rho\right) d_p \sqrt{g d_p}}{\mu}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:

$$\boxed{\frac{V}{\sqrt{g d_p}} = f\left(\frac{\left(\rho_p - \rho\right) d_p \sqrt{g d_p}}{\mu}\right)} \qquad (2)$$

Discussion We cannot determine the *form* of the relationship by purely dimensional reasoning since there are two Πs. However, in Chap. 10 we shall see that Π_1 is a constant times Π_2.

7-63

Solution We are to develop an equation for the settling speed of an aerosol particle falling in air under creeping flow conditions.

Assumptions **1** The particle falls at steady speed *V*. **2** The Reynolds number is small enough that the creeping flow approximation is valid.

Analysis We start by recognizing that as the particle falls at steady settling speed, its net weight *W* must equal the aerodynamic drag F_D on the particle. We also know that *W* is proportional to $(\rho_p - \rho)gd_p^3$. Thus,

Equating forces: $$W = \text{constant}_1\left(\rho_p - \rho\right)gd_p^{\,3} = F_D = \text{constant}_2\,\mu V d_p \qquad (1)$$

where we have converted the notation of the previous problem, and we have defined two different constants. The two constants in Eq. 1 can be combined into one new constant for simplicity. Solving for *V*,

Settling speed: $$\boxed{V = \text{constant}\frac{\left(\rho_p - \rho\right)gd_p^{\,2}}{\mu}} \qquad (2)$$

If we divide both sides of Eq. 2 by $\sqrt{gd_p}$ we see that the functional relationship given by Eq. 2 of the previous problem is consistent.

Discussion This result is valid only if the Reynolds number is much smaller than one, as will be discussed in Chap. 10. If the particle is less dense than the fluid (e.g. bubbles rising in water), our result is still valid, but the particle rises instead of falls.

7-64

Solution We are to determine how the settling speed of an aerosol particle falling in air under creeping flow conditions changes when certain parameters are doubled.

Assumptions **1** The particle falls at steady speed *V*. **2** The Reynolds number is small enough that the creeping flow approximation is valid.

Analysis From the results of the previous problem, we see that **if particle size doubles, the settling speed increases by a factor of 2^2 = 4**. Similarly, **if density difference doubles, the settling speed increases by a factor of 2^1 = 2**.

Discussion This result is valid only if the Reynolds number remains much smaller than unity, as will be discussed in Chap. 10. As the particle's settling speed increases by a factor of 2 or 4, the Reynolds number will also increase by that same factor. If the new Reynolds number is not small enough, the creeping flow approximation will be invalid and our results will not be correct, although the error will probably be small.

7-65

Solution We are to generate a nondimensional relationship between the given parameters.

Assumptions **1** The flow is fully developed. **2** The fluid is incompressible. **3** No other parameters are significant in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters.

Step 1 All the relevant parameters in the problem are listed in functional form:

List of relevant parameters: $\Delta P = f\left(V, \varepsilon, \rho, \mu, D, L\right) \qquad n = 7$

Step 2 The primary dimensions of each parameter are listed:

$$\begin{array}{ccccccc} \Delta P & V & \varepsilon & \rho & \mu & D & L \\ \left\{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-2}\right\} & \left\{\mathrm{L}^1\mathrm{t}^{-1}\right\} & \left\{\mathrm{L}^1\right\} & \left\{\mathrm{m}^1\mathrm{L}^{-3}\right\} & \left\{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\right\} & \left\{\mathrm{L}^1\right\} & \left\{\mathrm{L}^1\right\} \end{array}$$

Step 3 As a first guess, *j* is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $j = 3$

If this value of *j* is correct, the expected number of Πs is

Number of expected Π*s*: $k = n - j = 7 - 3 = 4$

Step 4 We need to choose three repeating parameters since *j* = 3. Following the guidelines listed in Table 7-3, we cannot pick the dependent variable, Δ*P*. We cannot choose any two of parameters ε, *L*, and *D* since their dimensions are identical. It is not desirable to have μ or ε appear in all the Πs. The best choice of repeating parameters is thus *V*, *D*, and ρ.

Repeating parameters: *V*, *D*, and ρ

Step 5 The dependent Π is generated:

$$\Pi_1 = \Delta P V^{a_1} D^{b_1} \rho^{c_1} \qquad \left\{\Pi_1\right\} = \left\{\left(\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-2}\right)\left(\mathrm{L}^1\mathrm{t}^{-1}\right)^{a_1}\left(\mathrm{L}^1\right)^{b_1}\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{c_1}\right\}$$

mass: $\left\{\mathrm{m}^0\right\} = \left\{\mathrm{m}^1\mathrm{m}^{c_1}\right\} \qquad 0 = 1 + c_1 \qquad c_1 = -1$

time: $\left\{\mathrm{t}^0\right\} = \left\{\mathrm{t}^{-2}\mathrm{t}^{-a_1}\right\} \qquad 0 = -2 - a_1 \qquad a_1 = -2$

length: $\left\{\mathrm{L}^0\right\} = \left\{\mathrm{L}^{-1}\mathrm{L}^{a_1}\mathrm{L}^{b_1}\mathrm{L}^{-3c_1}\right\} \qquad \begin{array}{l} 0 = -1 + a_1 + b_1 - 3c_1 \\ 0 = -1 - 2 + b_1 + 3 \end{array} \qquad b_1 = 0$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{\Delta P}{\rho V^2}$$

From Table 7-5, the established nondimensional parameter most similar to our Π_1 is the **Euler number** Eu. No manipulation is required.

We form the second Π with μ. By now we know that we will generate a **Reynolds number**,

$$\Pi_2 = \mu V^{a_2} D^{b_2} \rho^{c_2} \qquad \Pi_2 = \frac{\rho V D}{\mu} = \text{Reynolds number} = \text{Re}$$

The final two Π groups are formed with ε and then with L. The algebra is trivial for these cases since their dimension (length) is identical to that of one of the repeating variables (D). The results are

$$\Pi_3 = \varepsilon V^{a_3} D^{b_3} \rho^{c_3} \qquad \Pi_3 = \frac{\varepsilon}{D} = \text{Roughness ratio}$$

$$\Pi_4 = L V^{a_4} D^{b_4} \rho^{c_4} \qquad \Pi_4 = \frac{L}{D} = \text{Length-to-diameter ratio or aspect ratio}$$

Step 6 We write the final functional relationship as

Relationship between Πs:

$$\boxed{\text{Eu} = \frac{\Delta P}{\rho V^2} = f\left(\text{Re}, \frac{\varepsilon}{D}, \frac{L}{D}\right)} \qquad (1)$$

Discussion The result applies to both laminar and turbulent fully developed pipe flow; it turns out, however, that the second independent Π (roughness ratio) is not nearly as important in laminar pipe flow as in turbulent pipe flow. Since ΔP drops linearly with distance down the pipe, we know that ΔP is linearly proportional to L/D. It is not possible to determine the functional relationships between the other Πs by dimensional reasoning alone.

7-66

Solution We are to determine by what factor volume flow rate increases in the case of fully developed laminar pipe flow when pipe diameter is doubled.

Assumptions **1** The flow is steady. **2** The flow is fully developed, meaning that dP/dx is constant and the velocity profile does not change downstream.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters.

Step 1 All the relevant parameters in the problem are listed in functional form:

List of relevant parameters: $$\dot{V} = f\left(D, \mu, \frac{dP}{dx}\right) \qquad n = 4$$

Step 2 The primary dimensions of each parameter are listed:

$$\begin{array}{cccc} \dot{V} & D & \mu & dP/dx \\ \{L^3t^{-1}\} & \{L^1\} & \{m^1L^{-1}t^{-1}\} & \{m^1L^{-2}t^{-2}\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 4 - 3 = 1$$

Step 4 We need to choose three repeating parameters since $j = 3$. Here we must pick all three independent parameters,

Repeating parameters: D, μ, and dP/dx

Step 5 The Π is generated:

$$\Pi_1 = \dot{V}\left(dP/dx\right)^{a_1} D^{b_1} \mu^{c_1} \qquad \{\Pi_1\} = \left\{\left(L^3t^{-1}\right)\left(m^1L^{-2}t^{-2}\right)^{a_1}\left(L^1\right)^{b_1}\left(m^1L^{-1}t^{-1}\right)^{c_1}\right\}$$

mass: $$\{m^0\} = \{m^{a_1}m^{c_1}\} \qquad 0 = a_1 + c_1 \qquad c_1 = -a_1$$

time: $$\{t^0\} = \{t^{-1}t^{-2a_1}t^{-c_1}\} \qquad \begin{array}{l} 0 = -1 - 2a_1 - c_1 \\ 0 = -1 - 2a_1 + a_1 \end{array} \qquad \begin{array}{l} a_1 = -1 \\ c_1 = 1 \end{array}$$

length: $$\{L^0\} = \{L^3L^{-2a_1}L^{b_1}L^{-c_1}\} \qquad \begin{array}{l} 0 = 3 - 2a_1 + b_1 - c_1 \\ b_1 = -3 + 2a_1 + c_1 \end{array} \qquad b_1 = -4$$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{\dot{V}\mu}{D^4 \dfrac{dP}{dx}}$$

Step 6 Since there is only one Π, we set it equal to a constant. We write the final functional relationship as

Relationship between Π*s:* $\boxed{\Pi_1 = \text{constant}, \qquad \dot{V} = \text{constant}\,\frac{D^4}{\mu}\frac{dP}{dx}}$ (1)

We see immediately that **if the pipe diameter is doubled with all other parameters fixed, the volume flow rate will increase by a factor of 2^4 = 16**.

Discussion We will see in Chap. 9 that the constant is $\pi/8$. There is no way to obtain the value of the constant from dimensional analysis alone.

7-67

Solution We are to determine the units of a parameter, and then use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis (*a*) The units of G are easily obtained by knowing the units of the other variables. Since $F = G\dfrac{m_1 m_2}{r^2}$, $G = \dfrac{Fr^2}{m_1 m_2}$ and the units of G are therefore $\dfrac{\text{N}\cdot\text{m}^2}{\text{kg}^2}\left(\dfrac{\text{kg}\cdot\text{m/s}^2}{\text{N}}\right) = \dfrac{\text{m}^3}{\text{kg}\cdot\text{s}^2}$. $\boxed{G \text{ has units of } \dfrac{\text{m}^3}{\text{kg}\cdot\text{s}^2}}$. *Note*: This can be easily verified by looking up the universal gravitational constant in a textbook or on the Internet.

(*b*) The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; $n = 5$,

List of relevant parameters: $$F = F(G, m_1, m_2, r) \qquad n = 5$$

Step 2 The primary dimensions of each parameter are listed,

F	G	m_1	m_2	r
$\{\text{m}^1\text{L}^1\text{t}^{-2}\}$	$\{\text{L}^3\text{m}^{-1}\text{t}^{-2}\}$	$\{\text{m}^1\}$	$\{\text{m}^1\}$	$\{\text{L}^1\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since $j = 3$. Following the guidelines, we cannot pick both masses since they have the same dimensions. We choose

Repeating parameters: G, m_1, and r

Step 5 The dependent Π is generated:

$$\Pi_1 = FG^{a_1} m_1^{b_1} r^{c_1} \qquad \{\Pi_1\} = \left\{\left(\frac{\text{mL}}{\text{t}^2}\right)\left(\frac{\text{L}^3}{\text{mt}^2}\right)^{a_1}(\text{m})^{b_1}(\text{L})^{c_1}\right\}$$

time: $$\{\text{t}^0\} = \{\text{t}^{-2}\text{t}^{-2a_1}\} \qquad 0 = -2 - 2a_1 \qquad a_1 = -1$$

mass: $$\{\text{m}^0\} = \{\text{m}^1\text{m}^{-a_1}\text{m}^{b_1}\} \qquad 0 = 1 - a_1 + b_1 \qquad b_1 = -2$$

length: $$\{\text{L}^0\} = \{\text{L}^1\text{L}^{3a_1}\text{L}^{c_1}\} \qquad 0 = 1 + 3a_1 + c_1 \qquad c_1 = 2$$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{Fr^2}{Gm_1^2}$$

The second Pi (the only independent Π in this problem) is generated using the remaining variable m_2. This one is trivial and is done "by eye" since one of the repeating variables is also a mass,

Π_2:
$$\Pi_2 = \frac{m_2}{m_1}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:
$$\boxed{\Pi_1 = f\left(\Pi_2\right) \quad \text{or} \quad \frac{Fr^2}{Gm_1^{\;2}} = f\left(\frac{m_2}{m_1}\right)}$$

(c) Finally, comparing to the known law of universal gravitation, we see that **the relationship is in fact linear**, namely,

Relationship between Π*s*:
$$\boxed{\Pi_1 = \Pi_2 \quad \text{or} \quad \frac{Fr^2}{Gm_1^{\;2}} = \frac{m_2}{m_1} \quad \text{which yields } F = G\frac{m_1 m_2}{r^2}}$$

Discussion Not only is the relationship linear, but the constant of proportionality is unity. There is no way to get this result from dimensional analysis alone.

7-68

Solution We are to use dimensional analysis to find the functional relationship between the given parameters, namely, damping coefficient ζ as a function of spring constant k, mass m, and damping coefficient c.

Assumptions **1** The given parameters are the only relevant ones in the problem. **2** ζ is already dimensionless.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs). In this situation, ζ is already dimensionless, and so it is automatically one of the Πs. We can therefore form the other Π from the remaining three variables. Or, we can go through the normal procedure. We elect to show the full procedure below, but it is simpler to form a nondimensional parameter from variables k, m, and c.

Step 1 There are four parameters in this problem; n = 4,

List of relevant parameters: $\zeta = \zeta(k, m, c) \qquad n = 4$

Step 2 The primary dimensions of each parameter are listed,

$$\begin{array}{cccc} \zeta & k & m & c \\ \{m^0L^0t^0\} = \{1\} & \{m^1L^0t^{-2}\} & \{m^1\} & \{m^1L^0t^{-1}\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $j = 3$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $k = n - j = 4 - 3 = 1$

Step 4 We need to choose three repeating parameters since j = 3. Following the guidelines, we have only one choice,

Repeating parameters: k, m, and c

Step 5 The dependent Π is generated:

$$\Pi_1 = \zeta k^{a_1} m^{b_1} c^{c_1} \qquad \{\Pi_1\} = \left\{(1)\left(m^1t^{-2}\right)^{a_1}\left(m^1\right)^{b_1}\left(m^1t^{-1}\right)^{c_1}\right\}$$

time: $\{t^0\} = \{t^{-2a_1}t^{-c_1}\} \qquad 0 = -2a_1 - c_1 \qquad a_1 = -c_1/2$

mass: $\{m^0\} = \{m^{a_1}m^{b_1}m^{c_1}\} \qquad 0 = a_1 + b_1 + c_1 \qquad b_1 = -c_1/2$

length: $\{L^0\} = \{L^0\}$ - -

Here we have the curious situation that the length dimension drops out, and so we have only two equations, but three unknown exponents. Thus there is a whole family of possible answers. In other words, we are free to choose any value of c_1 that we wish, and the other two coefficients can then be determined. For example, if we pick c_1 = 1, then a_1 = b_1 = -1/2, and the Π is thus

Π$_1$ for c_1 = 1: $$\Pi_1 = \frac{\zeta c}{\sqrt{km}}$$

If we choose a different value of c_1 = 1, we get a different ∏. For example, if we pick c_1 = -1, then a_1 = b_1 = 1/2, and the Π is thus

Π_1 for c_1 = -1: $$\Pi_1 = \frac{\zeta\sqrt{km}}{c}$$

It turns out that the latter is more appropriate to match with standard dynamic system analysis. However, there is no way to know this from dimensional analysis alone. Jen must remember at least one other thing from her dynamic systems class – that the damping ratio increases with damping coefficient (which makes sense on physical grounds), and thus the latter equation for Π_1 makes the most physical sense.

Step 6 Since there is only one Π, it must be a constant and we write the final functional relationship as

Relationship between Π*s*: $$\boxed{\Pi_1 = \frac{\zeta\sqrt{km}}{c} = \text{constant, or } \zeta = \text{constant}\frac{c}{\sqrt{km}}}$$

Discussion It turns out that the constant is ½, and the correct definition for ζ is $\zeta = \dfrac{c}{2\sqrt{km}}$. Thus, our analysis yields the correct equation to within a constant. Note, however, that any other choice of exponent c_1 is also valid from a purely dimensional point of view. For example, if we pick c_1 = -2, then $a_1 = b_1 = 1$, and the result is simply the square of the previous result, i.e., $\Pi_1 = \dfrac{\zeta km}{c^2} = \text{constant, or } \zeta = \text{constant}\dfrac{c^2}{km}$. While this result is correct dimensionally (ζ is still dimensionless), it does not agree with the standard definition.

7-69

Solution We are to use dimensional analysis to find the functional relationship between the given parameters, namely, voltage drop ΔE as a function of electrical current I and electrical resistance R.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters.

Step 1 There are three parameters in this problem; n = 3,

List of relevant parameters: $\Delta E = \text{function}(I, R) \qquad n = 3$

Step 2 The primary dimensions of each parameter are listed [we get these dimensions from a previous problem],

$$\begin{array}{ccc} \Delta E & I & R \\ \{\text{m}^1\text{L}^2\text{t}^{-3}\text{I}^{-1}\} & \{\text{I}^1\} & \{\text{m}^1\text{L}^2\text{t}^{-3}\text{I}^{-2}\} \end{array}$$

Step 3 As a first guess, j is set equal to 4, the number of primary dimensions represented in the problem (m, L, t, and I).

Reduction: $j = 4$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $k = n - j = 3 - 4 = -1$

This is clearly incorrect, so we reduce j by one, yielding $k = n - j = 3 - 3 = 0$. This is again clearly incorrect (we cannot have zero Πs). So,

Reduction: $j = 2$

and the expected number of Πs is

Number of expected Π*s*: $k = n - j = 3 - 2 = 1$

Step 4 We need to choose two repeating parameters since j = 2. Following the guidelines, we have only one choice,

Repeating parameters: I and R

Step 5 The dependent Π is generated:

$$\Pi_1 = \Delta E I^{a_1} R^{b_1} \qquad \{\Pi_1\} = \left\{\left(\text{m}^1\text{L}^2\text{t}^{-3}\text{I}^{-1}\right)\left(\text{I}^1\right)^{a_1}\left(\text{m}^1\text{L}^2\text{t}^{-3}\text{I}^{-2}\right)^{b_1}\right\}$$

time: $\{\text{t}^0\} = \{\text{t}^{-3}\text{t}^{-3b_1}\}$ $\quad 0 = -3 - 3b_1 \quad b_1 = -1$

mass: $\{\text{m}^0\} = \{\text{m}^1\text{m}^{b_1}\}$ $\quad 0 = 1 + b_1 \quad b_1 = -1$

length: $\{\text{L}^0\} = \{\text{L}^2\text{L}^{2b_1}\}$ $\quad 0 = 2 + 2b_1 \quad b_1 = -1$

current: $\{\text{I}^0\} = \{\text{I}^{-1}\text{I}^{a_1}\text{I}^{-2b_1}\}$ $\quad 0 = -1 + a_1 - 2b_1 \quad a_1 = -1$

Thus,

Π_1: $$\Pi_1 = \frac{\Delta E}{IR}$$

Step 6 Since there is only one $\prod$, it must be a constant and we write the final functional relationship as

Relationship between Π*s*: $$\boxed{\Pi_1 = \frac{\Delta E}{IR} = \text{constant}, \text{ or } \Delta E = \text{constant} \cdot IR}$$

Comparing to Ohm's law, $\boxed{\Delta E = IR}$, we see that the constant is unity.

Discussion In this case, since the number of variables is small, we were able to generate the exact form of Ohm's law to within an unknown constant. The constant cannot be obtained through dimensional analysis alone.

7-70

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; $n = 5$,

List of relevant parameters: $$\delta = f\left(x, V, \rho, \mu\right) \qquad n = 5 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

δ	x	V	ρ	μ
$\{L^1\}$	$\{L^1\}$	$\{L^1t^{-1}\}$	$\{m^1L^{-3}\}$	$\{m^1L^{-1}t^{-1}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since j = 3. We pick length scale x, density ρ, and freestream velocity V.

Repeating parameters: x, ρ, and V

Step 5 The Πs are generated. Note that for the first Π we can do the algebra in our heads since the relationship is very simple. Namely, the dimensions of δ are identical to those of one of the repeating variables (x). In such a case we know that all the exponents in the Π group are zero except the one for x, which is –1. The dependent Π is thus

Π_1: $$\Pi_1 = \frac{\delta}{x}$$

The second Π is formed with viscosity,

$$\Pi_2 = \mu x^a \rho^b V^c \qquad \{\Pi_2\} = \left\{\left(m^1L^{-1}t^{-1}\right)\left(L^1\right)^a\left(m^1L^{-3}\right)^b\left(L^1t^{-1}\right)^c\right\}$$

mass: $\{m^0\} = \{m^1m^b\}$ $\qquad 0 = 1 + b \qquad b = -1$

time: $\{t^0\} = \{t^{-1}t^{-c}\}$ $\qquad 0 = -1 - c \qquad c = -1$

length: $\{L^0\} = \{L^{-1}L^aL^{-3b}L^c\}$ $\qquad 0 = -1 + a - 3b + c \qquad a = -1$

$$0 = -1 + a + 3 - 1$$

which yields

Π_2: $$\Pi_2 = \frac{\mu}{\rho V x}$$

We recognize this Π as the inverse of the **Reynolds number**, *Modified* Π_2 = *Reynolds number based on x*: $$\Pi_2 = \mathrm{Re}_x = \frac{\rho V x}{\mu}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\frac{\delta}{x} = f\left(\mathrm{Re}_x\right)}$$

Discussion We cannot determine the *form* of the relationship by purely dimensional reasoning since there are two Πs. However, in Chap. 10 we shall see that for a laminar boundary layer, Π_1 is proportional to the square root of Π_2.

7-71

Solution We are to find the functional relationship between the given parameters and name any established dimensionless parameters.

Assumptions **1** The given parameters are the only ones relevant to the flow at hand.

Analysis The method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are six parameters in this problem; n = 6,

List of relevant parameters: $$\Delta P = f\left(\rho, \omega, D, \mu, \dot{V}\right) \qquad n = 6$$

Step 2 The primary dimensions of each parameter are listed,

$$\begin{array}{cccccc} \Delta P & \rho & \omega & D & \mu & \dot{V} \\ \left\{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-2}\right\} & \left\{\mathrm{m}^1\mathrm{L}^{-3}\right\} & \left\{\mathrm{t}^{-1}\right\} & \left\{\mathrm{L}^1\right\} & \left\{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\right\} & \left\{\mathrm{L}^3\mathrm{t}^{-1}\right\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 6 - 3 = 3$$

Step 4 We need to choose three repeating parameters since j = 3. We pick fluid density ρ, length scale D, and angular velocity ω.

Repeating parameters: ρ, ω, and D

Step 5 The Πs are generated. The dependent Π is generated using ΔP and the three repeating variables,

$$\Pi_1 = \Delta P \rho^{a_1} \omega^{b_1} D^{c_1} \qquad \left\{\Pi_1\right\} = \left\{\left(\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-2}\right)\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{a_1}\left(\mathrm{t}^{-1}\right)^{b_1}\left(\mathrm{L}^1\right)^{c_1}\right\}$$

time: $$\left\{\mathrm{t}^0\right\} = \left\{\mathrm{t}^{-2}\mathrm{t}^{-b_1}\right\} \qquad 0 = -2 - b_1 \qquad b_1 = -2$$

mass: $$\left\{\mathrm{m}^0\right\} = \left\{\mathrm{m}^1\mathrm{m}^{a_1}\right\} \qquad 0 = 1 + a_1 \qquad a_1 = -1$$

length: $$\left\{\mathrm{L}^0\right\} = \left\{\mathrm{L}^{-1}\mathrm{L}^{-3a_1}\mathrm{L}^{c_1}\right\} \qquad 0 = -1 - 3a_1 + c_1 \qquad c_1 = -2$$

Thus,

Π_1 = *a kind of* **pressure coefficient**: $$\Pi_1 = \frac{\Delta P}{\rho \omega^2 D^2}$$

The second Π is obtained using viscosity and the three repeating variables,

$$\Pi_2 = \mu \rho^{a_2} \omega^{b_2} D^{c_2} \qquad \left\{\Pi_1\right\} = \left\{\left(\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\right)\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{a_2}\left(\mathrm{t}^{-1}\right)^{b_2}\left(\mathrm{L}^1\right)^{c_2}\right\}$$

time: $$\left\{\mathrm{t}^0\right\} = \left\{\mathrm{t}^{-1}\mathrm{t}^{-b_2}\right\} \qquad 0 = -1 - b_2 \qquad b_2 = -1$$

mass: $$\left\{\mathrm{m}^0\right\} = \left\{\mathrm{m}^1\mathrm{m}^{a_2}\right\} \qquad 0 = 1 + a_2 \qquad a_2 = -1$$

length: $$\left\{\mathrm{L}^0\right\} = \left\{\mathrm{L}^{-1}\mathrm{L}^{-3a_2}\mathrm{L}^{c_2}\right\} \qquad 0 = -1 - 3a_2 + c_2 \qquad c_2 = -2$$

Thus,

Π_2: $$\Pi_2 = \frac{\mu}{\rho \omega D^2}$$

We recognize this Π as the inverse of a kind of **Reynolds number**. The modified form of the Π is thus

Modified Π_2 *= a Reynolds number*: $$\Pi_2 = \frac{\rho \omega D^2}{\mu}$$

In similar fashion we obtain the third nondimensional parameter, combining volume flow rate and the three repeating variables. This one is trivial and the algebra can be performed in our heads, yielding

Π_3: $$\Pi_3 = \frac{\dot{V}}{\omega D^3}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\frac{\Delta P}{\rho \omega^2 D^2} = f\left(\frac{\rho \omega D^2}{\mu}, \frac{\dot{V}}{\omega D^3}\right)}$$

Discussion You may choose different repeating variables, and may generate different nondimensional groups. If you do the algebra correctly, your answer is not "wrong" – you just may not get the same dimensionless groups.

7-72

Solution We are to find the functional relationship between the given parameters and name any established dimensionless parameters.

Assumptions **1** The given parameters are the only ones relevant to the flow at hand.

Analysis The method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; n = 5,

List of relevant parameters: $$\mathrm{T} = f(\rho, \omega, D, \mu) \qquad n = 5$$

Step 2 The primary dimensions of each parameter are listed,

T	ρ	ω	D	μ
$\{\mathrm{m^1L^2t^{-2}}\}$	$\{\mathrm{m^1L^{-3}}\}$	$\{\mathrm{t^{-1}}\}$	$\{\mathrm{L^1}\}$	$\{\mathrm{m^1L^{-1}t^{-1}}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since j = 3. We pick fluid density ρ, length scale D, and angular velocity ω.

Repeating parameters: ρ, ω, and D

Step 5 The Πs are generated. The dependent Π is generated by combining T with the three repeating variables,

$$\Pi_1 = T\rho^{a_1}\omega^{b_1}D^{c_1} \qquad \{\Pi_1\} = \left\{\left(\mathrm{m^1L^2t^{-2}}\right)\left(\mathrm{m^1L^{-3}}\right)^{a_1}\left(\mathrm{t^{-1}}\right)^{b_1}\left(\mathrm{L^1}\right)^{c_1}\right\}$$

time: $\{\mathrm{t^0}\} = \{\mathrm{t^{-2}t^{-b_1}}\}$ $\quad 0 = -2 - b_1$ $\quad b_1 = -2$

mass: $\{\mathrm{m^0}\} = \{\mathrm{m^1m^{a_1}}\}$ $\quad 0 = 1 + a_1$ $\quad a_1 = -1$

length: $\{\mathrm{L^0}\} = \{\mathrm{L^2L^{-3a_1}L^{c_1}}\}$ $\quad 0 = 2 - 3a_1 + c_1$ $\quad c_1 = -5$

Thus,

Π_1 = *some kind of torque coefficient*: $$\Pi_1 = \frac{\mathrm{T}}{\rho\omega^2D^5}$$

The second Π is obtained using viscosity and the three repeating variables,

$$\Pi_2 = \mu\rho^{a_2}\omega^{b_2}D^{c_2} \qquad \{\Pi_1\} = \left\{\left(\mathrm{m^1L^{-1}t^{-1}}\right)\left(\mathrm{m^1L^{-3}}\right)^{a_2}\left(\mathrm{t^{-1}}\right)^{b_2}\left(\mathrm{L^1}\right)^{c_2}\right\}$$

time: $\{\mathrm{t^0}\} = \{\mathrm{t^{-1}t^{-b_2}}\}$ $\quad 0 = -1 - b_2$ $\quad b_2 = -1$

mass: $\{\mathrm{m^0}\} = \{\mathrm{m^1m^{a_2}}\}$ $\quad 0 = 1 + a_2$ $\quad a_2 = -1$

length: $\{\mathrm{L^0}\} = \{\mathrm{L^{-1}L^{-3a_2}L^{c_2}}\}$ $\quad 0 = -1 - 3a_2 + c_2$ $\quad c_2 = -2$

Thus,

Π_2: $$\Pi_2 = \frac{\mu}{\rho \omega D^2}$$

We recognize this Π as the inverse of a kind of **Reynolds number**. The modified form of the Π is thus

Modified Π_2 = *a Reynolds number*: $$\Pi_2 = \frac{\rho \omega D^2}{\mu}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\frac{\mathrm{T}}{\rho \omega^2 D^5} = f\left(\frac{\rho \omega D^2}{\mu}\right)}$$

Discussion You may choose different repeating variables, and may generate different nondimensional groups. If you do the algebra correctly, your answer is not "wrong" – you just may not get the same dimensionless groups.

7-73

Solution We are to repeat the previous problem, but with the fluid being a compressible gas rather than a liquid.

Assumptions **1** The given parameters are the only ones relevant to the flow at hand. **2** The speed of sound in the gas is an additional relevant parameter.

Analysis The method of repeating variables is employed to obtain the nondimensional parameters (the Πs). In this case, the key is to add speed of sound c to the list of independent variables since the fluid is compressible.

Step 1 There are now six parameters in this problem; $n = 6$,

List of relevant parameters: $\mathrm{T} = f\left(\rho, \omega, D, \mu, c\right) \qquad n = 6$

Step 2 The primary dimensions of each parameter are listed,

T	ρ	ω	D	μ	c
$\{\mathrm{m}^1\mathrm{L}^2\mathrm{t}^{-2}\}$	$\{\mathrm{m}^1\mathrm{L}^{-3}\}$	$\{\mathrm{t}^{-1}\}$	$\{\mathrm{L}^1\}$	$\{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\}$	$\{\mathrm{L}^1\mathrm{t}^{-1}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $j = 3$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $k = n - j = 6 - 3 = 3$

Step 4 We need to choose three repeating parameters since $j = 3$. We pick fluid density ρ, length scale D, and angular velocity ω.

Repeating parameters: ρ, ω, and D

Step 5 The Πs are generated. The dependent Π is generated by combining T with the three repeating variables,

$$\Pi_1 = T\rho^{a_1}\omega^{b_1}D^{c_1} \qquad \{\Pi_1\} = \left\{\left(\mathrm{m}^1\mathrm{L}^2\mathrm{t}^{-2}\right)\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{a_1}\left(\mathrm{t}^{-1}\right)^{b_1}\left(\mathrm{L}^1\right)^{c_1}\right\}$$

time: $\{\mathrm{t}^0\} = \{\mathrm{t}^{-2}\mathrm{t}^{-b_1}\}$ $\qquad 0 = -2 - b_1 \qquad b_1 = -2$

mass: $\{\mathrm{m}^0\} = \{\mathrm{m}^1\mathrm{m}^{a_1}\}$ $\qquad 0 = 1 + a_1 \qquad a_1 = -1$

length: $\{\mathrm{L}^0\} = \{\mathrm{L}^2\mathrm{L}^{-3a_1}\mathrm{L}^{c_1}\}$ $\qquad 0 = 2 - 3a_1 + c_1 \qquad c_1 = -5$

Thus,

Π_1 = some kind of torque coefficient: $\Pi_1 = \dfrac{\mathrm{T}}{\rho\omega^2 D^5}$

The second Π is obtained using viscosity and the three repeating variables,

$$\Pi_2 = \mu\rho^{a_2}\omega^{b_2}D^{c_2} \qquad \{\Pi_1\} = \left\{\left(\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-1}\right)\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{a_2}\left(\mathrm{t}^{-1}\right)^{b_2}\left(\mathrm{L}^1\right)^{c_2}\right\}$$

time: $\{\mathrm{t}^0\} = \{\mathrm{t}^{-1}\mathrm{t}^{-b_2}\}$ $\qquad 0 = -1 - b_2 \qquad b_2 = -1$

mass: $\{\mathrm{m}^0\} = \{\mathrm{m}^1\mathrm{m}^{a_2}\}$ $\qquad 0 = 1 + a_2 \qquad a_2 = -1$

length: $\{\mathrm{L}^0\} = \{\mathrm{L}^{-1}\mathrm{L}^{-3a_2}\mathrm{L}^{c_2}\}$ $\qquad 0 = -1 - 3a_2 + c_2 \qquad c_2 = -2$

Thus,

Π_2: $$\Pi_2 = \frac{\mu}{\rho\omega D^2}$$

We recognize this Π as the inverse of a kind of **Reynolds number**. The modified form of the Π is thus

Modified Π_2 = *a Reynolds number*: $$\Pi_2 = \frac{\rho\omega D^2}{\mu}$$

In similar fashion we obtain the third nondimensional parameter by combining speed of sound with the three repeating variables.

$$\Pi_3 = c\rho^{a_3}\omega^{b_3}D^{c_3} \qquad \{\Pi_3\} = \left\{\left(\mathrm{L}^1\mathrm{t}^{-1}\right)\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{a_3}\left(\mathrm{t}^{-1}\right)^{b_3}\left(\mathrm{L}^1\right)^{c_3}\right\}$$

time: $\{\mathrm{t}^0\} = \{\mathrm{t}^{-1}\mathrm{t}^{-b_3}\}$ $\qquad 0 = -1 - b_3 \qquad b_3 = -1$

mass: $\{\mathrm{m}^0\} = \{\mathrm{m}^{a_3}\}$ $\qquad 0 = a_3 \qquad a_3 = 0$

length: $\{\mathrm{L}^0\} = \{\mathrm{L}^1\mathrm{L}^{-3a_3}\mathrm{L}^{c_3}\}$ $\qquad 0 = 1 - 3a_3 + c_3 \qquad c_3 = -1$

Thus,

Π_3: $$\Pi_3 = \frac{c}{\omega D}$$

Since ωD is a speed – the tip speed of the propeller, we recognize this Π as the inverse of a kind of **Mach number**. The modified form of the Π is thus

Modified Π_3 = *a Mach number*: $$\Pi_3 = \frac{\omega D}{c}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\frac{\mathrm{T}}{\rho\omega^2 D^5} = f\left(\frac{\rho\omega D^2}{\mu}, \frac{\omega D}{c}\right)}$$

Discussion We notice that the first two Πs are identical to those of the previous problem, since we have the same variables and we chose the same repeating parameters. The addition of speed of sound leads to a third Π.

7-74

Solution We are to use dimensional analysis to find the functional relationship between the given parameters, namely, turbulent viscous dissipation rate ε (rate of energy loss per unit mass) as a function of length scale ℓ and velocity scale u' of the large-scale turbulent eddies.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are three parameters in this problem; $n = 3$,

List of relevant parameters: $\varepsilon = \text{function of}\left(\ell, u'\right) \qquad n = 3$

Step 2 The primary dimensions of each parameter are listed,

$$\begin{array}{ccc} \varepsilon & \ell & u' \\ \left\{\mathrm{m}^0\mathrm{L}^2\mathrm{t}^{-3}\right\} & \left\{\mathrm{m}^0\mathrm{L}^1\mathrm{t}^0\right\} & \left\{\mathrm{m}^0\mathrm{L}^1\mathrm{t}^{-1}\right\} \end{array}$$

Step 3 As a first guess, j is set equal to 2, the number of primary dimensions represented in the problem (L and t).

Reduction: $j = 2$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $k = n - j = 3 - 2 = 1$

Step 4 We need to choose three repeating parameters since $j = 2$. Following the guidelines, we have only one choice,

Repeating parameters: ℓ and u'

Step 5 The dependent Π is generated:

$$\Pi_1 = \varepsilon \ell^a \left(u'\right)^b \qquad \left\{\Pi_1\right\} = \left\{\left(\mathrm{L}^2\mathrm{t}^{-3}\right)\left(\mathrm{L}^1\right)^a\left(\mathrm{L}^1\mathrm{t}^{-1}\right)^b\right\}$$

time: $\left\{\mathrm{t}^0\right\} = \left\{\mathrm{t}^{-3}\mathrm{t}^{-b}\right\} \qquad 0 = -3 - b \qquad b = -3$

length: $\left\{\mathrm{L}^0\right\} = \left\{\mathrm{L}^2\mathrm{L}^a\mathrm{L}^b\right\} \qquad 0 = 2 + a + b \qquad a = -2 - b = 1$

So, the dependent (and only) Π becomes

Π_1:

$$\Pi_1 = \varepsilon \frac{\ell}{\left(u'\right)^3}$$

Step 6 Since there is only one ∏, it must be a constant and we write the final functional relationship as

Relationship between Πs:

$$\boxed{\Pi_1 = \varepsilon \frac{\ell}{\left(u'\right)^3} = \text{constant, or } \varepsilon = \text{constant}\frac{\left(u'\right)^3}{\ell}}$$

Discussion The constant cannot be obtained from dimensional analysis. Indeed, the constant is not really a constant at all, but depends on the specifics of the turbulent flow being examined. It is common in the study of turbulence to write the final result in order-of-magnitude form, namely, $\boxed{\varepsilon \sim \frac{(u')^3}{\ell}}$, where the tilde means "is of order of magnitude".

7-75

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are four parameters in this problem; $n = 4$,

List of relevant parameters: $$\dot{Q} = f\left(\dot{m}, c_p, \left(T_{\text{out}} - T_{\text{in}}\right)\right) \qquad n = 4 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

$\dot{Q}$	$\dot{m}$	c_p	T_{out}- T_{in}
$\{\text{m}^1\text{L}^2\text{t}^{-3}\}$	$\{\text{m}^1\text{t}^{-1}\}$	$\{\text{L}^2\text{t}^{-2}\text{T}^{-1}\}$	$\{\text{T}^1\}$

Step 3 As a first guess, j is set equal to 4, the number of primary dimensions represented in the problem (m, T, L, and t).

Reduction: $$j = 4$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 4 - 4 = 0$$

Obviously this is not correct, so we re-examine our initial assumptions. We are convinced that our list of parameters is sufficient, so we reduce j by one and continue,

Reduction: $$j = 4 - 1 = 3$$

If this revised value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 4 - 3 = 1$$

We now expect only one Π.

Step 4 We need to choose three repeating parameters since $j = 3$. We only have one choice in this problem, since there are only three independent parameters on the right-hand side of Eq. 1,

Repeating parameters: $$\dot{m},\ c_p, \text{ and } \left(T_{\text{out}} - T_{\text{in}}\right)$$

Step 5 The dependent Π is generated:

$$\Pi_1 = \dot{Q}\dot{m}^{a_1} c_p^{\ b_1} \left(T_{\text{out}} - T_{\text{in}}\right)^{c_1} \qquad \{\Pi_1\} = \left\{\left(\text{m}^1\text{L}^2\text{t}^{-3}\right)\left(\text{m}^1\text{t}^{-1}\right)^{a_1}\left(\text{L}^2\text{t}^{-2}\text{T}^{-1}\right)^{b_1}\left(\text{T}^1\right)^{c_1}\right\}$$

mass: $\{\text{m}^0\} = \{\text{m}^{1+a_1}\}$ $\quad 0 = 1 + a_1 \quad$ $a_1 = -1$

length: $\{\text{L}^0\} = \{\text{L}^2\text{L}^{2b_1}\}$ $\quad 0 = 2 + 2b_1 \quad$ $b_1 = -1$

temperature: $\{\text{T}^0\} = \{\text{T}^{-b_1+c_1}\}$ $\quad c_1 = b_1 \quad$ $c_1 = -1$

time: $\{\text{t}^0\} = \{\text{t}^{-3-a_1-2b_1}\}$ $\quad 0 = -3 - a_1 - 2b_1 \quad$ $3 = 1 + 2$

Fortunately the result for the time exponents is consistent with that of the other dimensions. The dependent Π is thus

Π_1:

$$\Pi_1 = \frac{\dot{Q}}{\dot{m}c_p\left(T_{\text{out}} - T_{\text{in}}\right)}$$

Step 6 Since there is only one Π, it is a function of nothing. This is only possible if we set the Π equal to a constant. We write the final functional relationship as

Relationship between Π*s*:

$$\boxed{\Pi_1 = \frac{\dot{Q}}{\dot{m}c_p\left(T_{\text{out}} - T_{\text{in}}\right)} = \text{constant}} \qquad (2)$$

Discussion When there is only one Π, we know the functional relationship to within some (unknown) constant. In this particular case, comparing to Eq. 1 of Problem 7-24, we see that the constant is unity, $\dot{Q} = \dot{m}c_p\left(T_{\text{out}} - T_{\text{in}}\right)$. There is no way to obtain the constant in Eq. 2 from dimensional analysis; however, *one* experiment would be sufficient to determine the constant.

7-76

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; $n = 5$,

List of relevant parameters: $$h = f\left(\omega, \rho, g, R\right) \qquad n = 5 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

h	ω	ρ	g	R
$\{\mathrm{L}^1\}$	$\{\mathrm{t}^{-1}\}$	$\{\mathrm{m}^1\mathrm{L}^{-3}\}$	$\{\mathrm{L}^1\mathrm{t}^{-2}\}$	$\{\mathrm{L}^1\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since $j = 3$. Following the guidelines outlined in this chapter, we elect not to pick the viscosity. We choose

Repeating parameters: ω, ρ, and R

Step 5 The dependent Π is generated:

$$\Pi_1 = h\omega^{a_1}\rho^{b_1}R^{c_1} \qquad \{\Pi_1\} = \left\{\left(\mathrm{L}^1\right)\left(\mathrm{t}^{-1}\right)^{a_1}\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{b_1}\left(\mathrm{L}^1\right)^{c_1}\right\}$$

mass: $$\{\mathrm{m}^0\} = \{\mathrm{m}^{b_1}\} \qquad 0 = b_1 \qquad b_1 = 0$$

time: $$\{\mathrm{t}^0\} = \{\mathrm{t}^{-a_1}\} \qquad 0 = -a_1 \qquad a_1 = 0$$

length: $$\{\mathrm{L}^0\} = \{\mathrm{L}^1\mathrm{L}^{-3b_1}\mathrm{L}^{c_1}\} \qquad 0 = 1 - 3b_1 + c_1 \qquad c_1 = -1$$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{h}{R}$$

The second Pi (the only independent Π in this problem) is generated:

$$\Pi_2 = g\omega^{a_2}\rho^{b_2}R^{c_2} \qquad \{\Pi_2\} = \left\{\left(\mathrm{L}^1\mathrm{t}^{-2}\right)\left(\mathrm{t}^{-1}\right)^{a_2}\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{b_2}\left(\mathrm{L}^1\right)^{c_2}\right\}$$

mass: $$\{\mathrm{m}^0\} = \{\mathrm{m}^{b_2}\} \qquad 0 = b_2 \qquad b_2 = 0$$

time: $\{t^0\} = \{t^{-2}t^{-a_2}\}$ $\quad 0 = -2 - a_2 \quad a_2 = -2$

length: $\{L^0\} = \{L^1L^{-3b_2}L^{c_2}\}$ $\quad 0 = 1 - 3b_2 + c_2$, $\quad 0 = 1 + c_2 \quad c_2 = -1$

which yields

Π_2:
$$\Pi_2 = \frac{g}{\omega^2 R}$$

If we take Π_2 to the power –1/2 and recognize that ωR is the speed of the rim, we see that Π_2 can be modified into a **Froude number**,

Modified Π_2:
$$\Pi_2 = \mathrm{Fr} = \frac{\omega R}{\sqrt{gR}}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:
$$\boxed{\frac{h}{R} = f(\mathrm{Fr})} \qquad (2)$$

Discussion In the generation of the first Π, h and R have the same dimensions. Thus, we could have immediately written down the result, $\Pi_1 = h/R$. Notice that density ρ does not appear in the result. Thus, density is not a relevant parameter after all.

7-77

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are seven parameters in this problem; $n = 7$,

List of relevant parameters: $$h = f\left(\omega, \rho, g, R, t, \mu\right) \qquad n = 7 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

h	ω	ρ	g	R	t	μ
$\{L^1\}$	$\{t^{-1}\}$	$\{m^1L^{-3}\}$	$\{L^1t^{-2}\}$	$\{L^1\}$	$\{t^1\}$	$\{m^1L^{-1}t^{-1}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 7 - 3 = 4$$

Step 4 We need to choose three repeating parameters since $j = 3$. For convenience we choose the same repeating parameters that we used in the previous problem,

Repeating parameters: ω, ρ, and R

Step 5 The first two Πs are identical to those of the previous problem:

Π_1: $$\Pi_1 = \frac{h}{R}$$

and

Π_2: $$\Pi_2 = \frac{\omega R}{\sqrt{gR}}$$

where Π_2 is identified as a form of the **Froude number**. The third Π is formed with time t. Since repeating parameter ω has dimensions of 1/time, it is the only one that remains in the Π. Thus, without the formal algebra,

Π_3: $$\Pi_3 = \omega t$$

Finally, Π_4 is generated with liquid viscosity,

$$\Pi_4 = \mu\omega^{a_4}\rho^{b_4}R^{c_4} \qquad \{\Pi_4\} = \left\{\left(m^1L^{-1}t^{-1}\right)\left(t^{-1}\right)^{a_4}\left(m^1L^{-3}\right)^{b_4}\left(L^1\right)^{c_4}\right\}$$

mass: $$\{m^0\} = \{m^1m^{b_4}\} \qquad 0 = 1 + b_4 \qquad b_4 = -1$$

time: $$\{t^0\} = \{t^{-1}t^{-a_4}\} \qquad 0 = -1 - a_4 \qquad a_4 = -1$$

length: $\left\{L^0\right\} = \left\{L^{-1}L^{-3b_1}L^{c_1}\right\}$ $\quad 0 = -1 - 3b_4 + c_4$ $\quad c_4 = 1 + 3b_4$ $\quad c_4 = -2$

The final Π is thus

Π_4: $$\Pi_4 = \frac{\mu}{\rho\Omega R^2} \qquad (2)$$

If we invert Π_4 and recognize that ωR is the speed of the rim, it becomes clear that Π_4 of Eq. 2 can be modified into a **Reynolds number**,

Modified Π_4: $$\Pi_4 = \frac{\rho\omega R^2}{\mu} = \mathrm{Re} \qquad (3)$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\frac{h}{R} = f\left(\mathrm{Fr}, \omega t, \mathrm{Re}\right)} \qquad (4)$$

Discussion Notice that this time density ρ *does* appear in the result. There are other acceptable answers, but this one has the most established dimensionless groups.

Experimental Testing and Incomplete Similarity

7-78C

Solution We are to discuss some situations in which a model should be larger than its prototype.

Analysis There are many possible situations, and students' examples should vary. Generally, any flow field that is very small and/or very fast benefits from simulation with a larger model. In most cases these are situations in which we want the model to be larger and slower so that experimental measurements and flow visualization are easier. Here are a few examples:

- **Modeling a hard disk drive.**
- **Modeling insect flight.**
- **Modeling the settling of very small particles in air or water.**
- **Modeling the motion of water droplets in clouds.**
- **Modeling flow through very fine tubing.**
- **Modeling biological systems like blood flow through capillaries, flow in the bronchi of lungs, etc.**

Discussion You can think of several more examples.

7-79C

Solution We are to discuss the purpose of a moving ground belt and suggest an alternative.

Analysis From the frame of reference of a moving car, both the air and the ground approach the car at freestream speed. When we test a model car in a wind tunnel, the air approaches at freestream speed, but the ground (floor of the wind tunnel) is *stationary*. Therefore we are not modeling the same flow. A boundary layer builds up on the wind tunnel floor, and the flow under the car cannot be expected to be the same as that under a real car. A moving ground belt solves this problem. Another way to say the same thing is to say that without the moving ground belt, there would not be kinematic similarity between the underside of the model and the underside of the prototype.

If a moving ground belt is unavailable, we could instead **install a false wall – i.e., a thin flat plate just above the boundary layer on the floor of the wind tunnel**. A sketch is shown in Fig. 1. At least then the boundary layer will be very thin and will not have as much influence on the flow under the model.

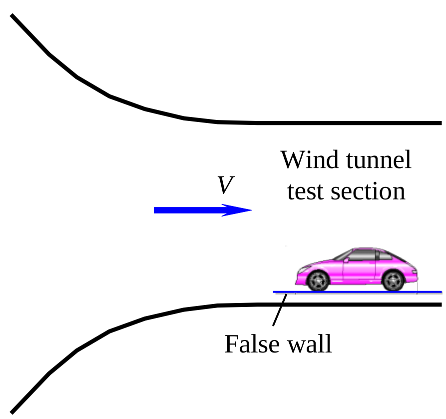


FIGURE 1
A false wall along the floor of a wind tunnel to reduce the size of the ground boundary layer.

Discussion We discuss boundary layer growth in Chap. 10.

7-80C

Solution We are to discuss whether Reynolds number independence has been achieved, and whether the researchers can be confident about it.

Analysis We remove the last four data points from Table 7-7 and from Fig. 7-41. From the remaining data it appears that the drag coefficient is beginning to level off, but is still decreasing with Re. Thus, **the researchers do not know if they have achieved Reynolds number independence or not**.

Discussion The wind tunnel speed is too low to achieve Reynolds number independence.

7-81C

Solution We are to define wind tunnel blockage and discuss its acceptable limit. We are also to discuss the source of measurement errors at high values of blockage.

Analysis Wind tunnel blockage is defined as the **ratio of model frontal area to cross-sectional area of the test-section**. The rule of thumb is that the blockage should be no more than 7.5%. If the blockage were significantly higher than this value, the flow would have to accelerate around the model much more than if the model were in an unbounded situation. Hence, similarity would not be achieved. We might expect the aerodynamic drag on the model to be too high since the *effective* freestream speed is too large due to the blockage.

Discussion There are formulas to correct for wind tunnel blockage, but they become less and less reliable as blockage increases.

7-82C

Solution We are to discuss the rule of thumb concerning Mach number and incompressibility.

Analysis **The rule of thumb is that the Mach number must stay below about 0.3 in order for the flow field to be considered "incompressible"**. What this really means is that compressibility effects, although present at all Mach numbers, are negligibly small compared to other effects driving the flow. If Ma is larger than about 0.3 in a wind tunnel test, the model flow field loses both kinematic and dynamic similarity, and the measured results are questionable. Of course, the error increases as Ma increases.

Discussion Compressible flow is discussed in detail in Chap. 12. There you will see where the value 0.3 comes from.

7-83

Solution We are to calculate and plot C_D as a function of Re for a given set of wind tunnel measurements, and determine if dynamic similarity and/or Reynolds number independence have been achieved. Finally, we are to estimate the aerodynamic drag force acting on the prototype car.

Assumptions **1** The model car is geometrically similar to the prototype car. **2** The aerodynamic drag on the strut holding the model car is negligible.

Properties For air at atmospheric pressure and at $T = 25^{o}C$, $\rho = 1.184$ kg/m^3 and $\mu = 1.849 \times 10^{-5}$ kg/(m s).

Analysis We calculate C_D and Re for the last data point listed in the given table (at the fastest wind tunnel speed),

Model drag coefficient at last data point:

$$C_{D,\mathrm{m}} = \frac{F_{D,\mathrm{m}}}{\frac{1}{2}\rho_\mathrm{m} V_\mathrm{m}^{\;2} A_\mathrm{m}} = \frac{4.91\text{ N}}{\frac{1}{2}\left(1.184\text{ kg/m}^3\right)\left(55\text{ m/s}\right)^2 \frac{\left(1.69\text{ m}\right)\left(1.30\text{ m}\right)}{16^2}}\left(\frac{\text{kg m}}{\text{s}^2\text{N}}\right) = 0.319$$

and

Model Reynolds number at last data point:

$$\mathrm{Re}_\mathrm{m} = \frac{\rho_\mathrm{m} V_\mathrm{m} W_\mathrm{m}}{\mu_\mathrm{m}} = \frac{\left(1.184\text{ kg/m}^3\right)\left(55\text{ m/s}\right)\left(\frac{1.69}{16}\text{ m}\right)}{1.849\times10^{-5}\text{ kg/m}\cdot\text{s}} = 3.72\times10^5$$

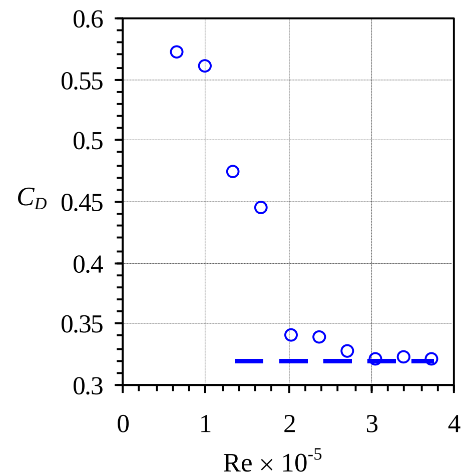


FIGURE 1
Aerodynamic drag coefficient as a function of Reynolds number – results nondimensionalized from wind tunnel test data on a model car.

We repeat the above calculations for all the data points in the given table, and we plot C_D verses Re in Fig. 1.

Have we achieved dynamic similarity? Well, we have *geometric* similarity between model and prototype, but the Reynolds number of the prototype car is

Reynolds number of prototype car:
$$\mathrm{Re}_\mathrm{p} = \frac{\rho_\mathrm{p} V_\mathrm{p} W_\mathrm{p}}{\mu_\mathrm{p}} = \frac{\left(1.184\text{ kg/m}^3\right)\left(31.3\text{ m/s}\right)\left(1.69\text{ m}\right)}{1.849\times10^{-5}\text{ kg/m}\cdot\text{s}} = 3.39\times10^6 \qquad (2)$$

where the width and speed of the prototype are used in the calculation of Re_p. Comparison of Eqs. 1 and 2 reveals that the prototype Reynolds number is more than eight times larger than that of the model. Since we cannot match the independent Πs in the problem, **dynamic similarity has not been achieved**.

Have we achieved Reynolds number independence? From Fig. 1 we see that Yes, **Reynolds number independence has indeed been achieved** – at Re greater than about 3×10^5, C_D has leveled off to a value of about 0.32 (to two significant digits).

Since we have achieved Reynolds number independence, we can extrapolate to the full scale prototype, assuming that C_D remains constant as Re is increased to that of the full scale prototype.

Aerodynamic drag on the prototype:

$$F_{D,\mathrm{p}} = \frac{1}{2}\rho_\mathrm{p} V_\mathrm{p}^{\;2} A_\mathrm{p} C_{D,\mathrm{p}} = \frac{1}{2}\left(1.184\text{ kg/m}^3\right)\left(31.3\text{ m/s}\right)^2\left(1.69\text{ m}\right)\left(1.30\text{ m}\right)0.32\left(\frac{\text{s}^2\text{N}}{\text{kg m}}\right) = \mathbf{408\ N}$$

Discussion We give our final result to two significant digits.

7-84

Solution We are to nondimensionalize experimental pipe data, plot the data, and determine if Reynolds number independence has been achieved. We are then to extrapolate to a higher speed.

Assumptions **1** The flow is fully developed. **2** The flow is steady and incompressible.

Properties For water at T = 20°C and atmospheric pressure, ρ = 998.0 kg/m^3 and μ = 1.002 × 10^{-3} kg/m·s.

Analysis (*a*) We convert each data point in the table from V and ΔP to Reynolds number and Euler number. The calculations at the last (highest speed) data point are shown here:

Reynolds number:

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{\left(998.0 \text{ kg/m}^3\right)\left(50 \text{ m/s}\right)\left(0.104 \text{ m}\right)}{1.002\times10^{-3} \text{ kg/m}\cdot\text{s}} = 5.18\times10^6 \quad (1)$$

and

Euler number:

$$\text{Eu} = \frac{\Delta P}{\rho V^2} = \frac{758,700 \text{ N/m}^2}{\left(998.0 \text{ kg/m}^3\right)\left(50 \text{ m/s}\right)^2}\left(\frac{\text{kg m}}{\text{s}^2\text{N}}\right) = 0.304 \quad (2)$$

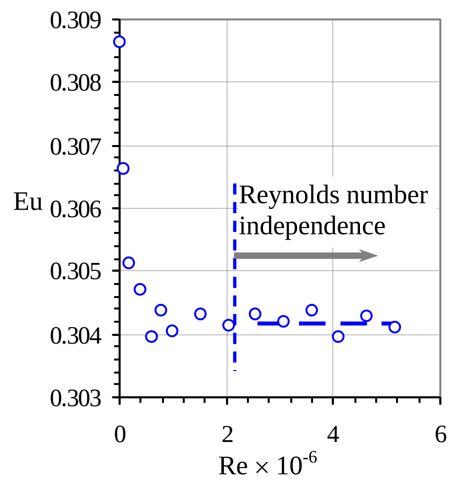


FIGURE 1
Nondimensionalized experimental data from a section of pipe.

We plot Eu versus Re in Fig. 1. Although there is experimental scatter in the data, it appears that Reynolds number independence has been achieved beyond a Reynolds number of about 2 × 10^6. The average value of Eu based on the last 6 data points is 0.3042.

(*b*) We extrapolate to higher speeds. At V = 80 m/s, we calculate ΔP, assuming that Eu remains constant to higher values of Re,

Extrapolated value:

$$\Delta P = \text{Eu}\times\rho V^2 = 0.3042\left(998.0 \text{ kg/m}^3\right)\left(80 \text{ m/s}\right)^2\left(\frac{\text{s}^2\text{N}}{\text{kg m}}\right) = \mathbf{1,940,000 \text{ N/m}^2} \quad (3)$$

Discussion It is shown in Chap. 8 that Reynolds number independence is indeed achieved at high-enough values of Re. The threshold value above which Re independence is achieved is a function of relative roughness height, ε/D.

7-85

Solution We are to calculate the wind tunnel blockage of a model truck in a wind tunnel and determine if it is within acceptable limits.

Assumptions **1** The frontal area is equal to truck width times height. (Note that the actual area of the truck may be somewhat smaller than this due to rounded corners and the air gap under the truck, but a truck looks nearly like a rectangle from the front, so this is not a bad approximation.)

Analysis Wind tunnel blockage is defined as the ratio of model frontal area to cross-sectional area of the test-section,

$$Blockage = \frac{A_{\text{model}}}{A_{\text{wind tunnel}}} = \frac{(0.159\text{ m})(0.257\text{ m})}{(0.90\text{ m})(0.85\text{ m})} = 0.0534 \cong \mathbf{5.3\%}$$

The rule of thumb is that the blockage should be no more than 7.5%. Since we are well below this value, we need not worry about blockage effects.

Discussion The *length* of the model does not enter our analysis since we are only concerned with the frontal area of the model.

7-86E

Solution We are to calculate the size and scale of the model truck to be constructed, and calculate its maximum Reynolds number. Then we are to determine whether this model in this wind tunnel will achieve Reynolds number independence.

Assumptions **1** The model will be constructed carefully so as to achieve approximate geometric similarity. **2** The wind tunnel air is at the same temperature and pressure as that flowing over the prototype truck.

Properties For air at $T = 80^\circ$F and atmospheric pressure, $\rho = 0.07350$ lbm/ft^3 and $\mu = 1.248 \times 10^{-5}$ lbm/ft·s.

Analysis (*a*) The rule of thumb about blockage is that we should keep the blockage below 7.5%. Thus, the frontal area of the model truck must be no more than $0.075 \times A_{\text{wind tunnel}}$. The ratio of height to width of the full-scale truck is $H_p/W_p = 12/8.33 = 1.44$. Thus, for the geometrically similar model truck,

Equation for model truck width:
$$W_m = \frac{A_m}{H_m} = \frac{7.5\% A_{\text{wind tunnel}}}{1.44 W_m} \qquad (1)$$

We solve Eq. 1 for W_m,

Model truck width:
$$W_m = \sqrt{\frac{7.5\% A_{\text{wind tunnel}}}{1.44}} = \sqrt{\frac{0.075\left(400\text{ in}^2\right)}{1.44}} = 4.56\text{ in} \qquad (2)$$

Scaling the height and length geometrically,

Model truck dimensions:
$$W_m = \mathbf{4.56\text{ in}},\ H_m = \mathbf{6.57\text{ in}},\ L_m = \mathbf{28.5\text{ in}} \qquad (3)$$

These dimensions represent a model that is scaled at approximately 1:22.

(*b*) At the maximum speed, with Re based on truck width,

Maximum Re:
$$\text{Re} = \frac{\rho W_m V_{\max}}{\mu} = \frac{\left(0.07350\text{ lbm/ft}^3\right)\left(4.56\text{ in}\right)\left(145\text{ ft/s}\right)}{1.248\times10^{-5}\text{ lbm/ft}\cdot\text{s}}\left(\frac{1\text{ ft}}{12\text{ in}}\right) = \mathbf{3.25\times10^5} \qquad (4)$$

(*c*) Based on the data of Fig. 7-41, **this Reynolds number is shy of the value needed to achieve Reynolds number independence**.

Discussion The students should run at the highest wind tunnel speed. Their measured values of C_D will probably be higher than those of the prototype, but the *relative* difference in C_D due to their modifications should still be valid.

7-87

Solution We are to show that Froude number and Reynolds number are the dimensionless parameters that appear in a problem involving shallow water waves.

Assumptions **1** Wave speed c is a function only of depth h, gravitational acceleration g, fluid density ρ, and fluid viscosity μ.

Analysis We perform a dimensional analysis using the method of repeating variables.

Step 1 There are five parameters in this problem; $n = 5$,

List of relevant parameters: $$c = f(h, \rho, \mu, g) \qquad n = 5$$

Step 2 The primary dimensions of each parameter are listed,

$$\begin{array}{ccccc} c & h & \rho & \mu & g \\ \{L^1t^{-1}\} & \{L^1\} & \{m^1L^{-3}\} & \{m^1L^{-1}t^{-1}\} & \{L^1t^{-2}\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since $j = 3$. We pick length scale h, density difference ρ, and gravitational constant g.

Repeating parameters: h, ρ, and g

Step 5 The Πs are generated. Note that for the first Π we do the algebra in our heads since the relationship is very simple. The dependent Π is

Π_1 = *Froude number*: $$\Pi_1 = \mathrm{Fr} = \frac{c}{\sqrt{gh}} \qquad (1)$$

This Π is the *Froude number*. Similarly, the Π formed with viscosity is generated,

$$\Pi_2 = \mu h^a \rho^b g^c \qquad \{\Pi_2\} = \left\{(m^1L^{-1}t^{-1})(L^1)^a(m^1L^{-3})^b(L^1t^{-2})^c\right\}$$

mass: $$\{m^0\} = \{m^1m^b\} \qquad 0 = 1 + b \qquad b = -1$$

time: $$\{t^0\} = \{t^{-1}t^{-2c}\} \qquad 0 = -1 - 2c \qquad c = -\frac{1}{2}$$

length: $$\{L^0\} = \{L^{-1}L^aL^{-3b}L^c\} \qquad \begin{array}{l} 0 = -1 + a - 3b + c \\ 0 = -1 + a + 3 - \frac{1}{2} \end{array} \qquad a = -\frac{3}{2}$$

which yields

Π_2: $$\Pi_2 = \frac{\mu}{\rho h^{\frac{3}{2}}\sqrt{g}}$$

We can manipulate this Π into the Reynolds number if we invert it and then multiply by Fr (Eq. 1) The final form is

Modified Π_2 = *Reynolds number*: $$\Pi_2 = \mathrm{Re} = \frac{\rho c h}{\mu}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\mathrm{Fr} = \frac{c}{\sqrt{gh}} = f\left(\mathrm{Re}\right) \quad \text{where} \quad \mathrm{Re} = \frac{\rho c h}{\mu}}$$

Discussion As discussed in this chapter, it is often difficult to match both Fr and Re between a model and a prototype.

Review Problems

7-88C

Solution We are to find at least three established nondimensional parameters not listed in Table 7-5, and list these following the format of that table.

Analysis Students' responses will vary. Here are some examples:

Name	Definition	Ratio of significance
Bingham number	$\mathrm{Bm} = \dfrac{\tau L}{\mu V}$	$\dfrac{\text{yield stress}}{\text{viscous stress}}$
Elasticity number	$\mathrm{El} = \dfrac{t_c \mu}{\rho L^2}$	$\dfrac{\text{elastic force}}{\text{inertial force}}$
Galileo number	$\mathrm{Ga} = \dfrac{g D^3 \rho^2}{\mu^2}$	$\dfrac{\text{gravitational force}}{\text{viscous force}}$

In the above, t_c is a characteristic time.

Discussion There are many more established dimensionless parameters in the literature. Some sneaky students may make up their own!

7-89C

Solution We are to think of and describe a prototype and model flow in which there is geometric but not kinematic similarity even though $\mathrm{Re}_m = \mathrm{Re}_p$.

Analysis Students' responses will vary. Here are some examples:

- A model car is being tested in a wind tunnel such that there is geometric similarity and the wind tunnel speed is adjusted so that $\mathrm{Re}_m = \mathrm{Re}_p$. However, there is not a moving ground belt, so there is not kinematic similarity between the model and prototype.
- A model airplane is being tested in a wind tunnel such that there is geometric similarity and the wind tunnel speed is adjusted so that $\mathrm{Re}_m = \mathrm{Re}_p$. However, the Mach numbers are quite different, and therefore kinematic similarity is not achieved.
- A model of a river or waterfall or other open surface flow problem in which there is geometric similarity and the speed is adjusted so that $\mathrm{Re}_m = \mathrm{Re}_p$. However, the Froude numbers do not match and therefore the velocity fields are not similar and kinematic similarity is not achieved.

Discussion There are many more acceptable cases that students may imagine.

7-90C

Solution

(*a*) **False**: Kinematic similarity is a necessary but not sufficient condition for dynamic similarity.
(*b*) **True**: You cannot have dynamic similarity if the model and prototype are not geometrically similar.
(*c*) **True**: You cannot have kinematic similarity if the model and prototype are not geometrically similar.
(d) **False**: It is possible to have kinematic similarity (scaled velocities at corresponding points), yet not have dynamic similarity (forces do not scale at corresponding points).

7-91

Solution We are to determine the primary dimensions of each variable, and then show that Hooke's law is dimensionally homogeneous.

Analysis

(*a*) Moment of inertia has dimensions of length^4,

Primary dimensions of moment of inertia:
$$\{I\} = \{\text{length}^4\} = \{\mathbf{L^4}\} \qquad (1)$$

(*b*) Modulus of elasticity has the same dimensions as pressure,

Primary dimensions of modulus of elasticity:
$$\{E\} = \left\{\frac{\text{force}}{\text{area}}\right\} = \left\{\frac{\text{mass}\times\text{length}}{\text{time}^2}\times\frac{1}{\text{length}^2}\right\} = \left\{\frac{\mathbf{m}}{\mathbf{Lt^2}}\right\} \qquad (2)$$

Or, in exponent form, $\{E\}$ = $\{\mathbf{m^1\,L^{-1}\,t^{-2}}\}$.

(*c*) Strain is defined as change in length per unit length, so it is dimensionless.

Primary dimensions of strain:
$$\{\varepsilon\} = \left\{\frac{\text{length}}{\text{length}}\right\} = \{\mathbf{1}\} \qquad (3)$$

(*d*) Stress is force per unit area, again just like pressure.

Primary dimensions of stress:
$$\{\sigma\} = \left\{\frac{\text{force}}{\text{area}}\right\} = \left\{\frac{\text{mass}\times\text{length}}{\text{time}^2\times\text{length}^2}\right\} = \left\{\frac{\mathbf{m}}{\mathbf{Lt^2}}\right\} \qquad (4)$$

Or, in exponent form, $\{\sigma\}$ = $\{\mathbf{m^1\,L^{-1}\,t^{-2}}\}$.

(*e*) Hooke's law is $\sigma = E\varepsilon$. We write the primary dimensions of both sides:

Primary dimensions of Hooke's law:
$$\{\sigma\} = \left\{\frac{\mathbf{m}}{\mathbf{Lt^2}}\right\} = \{E\varepsilon\} = \left\{\frac{\text{m}}{\text{Lt}^2}\times 1\right\} = \left\{\frac{\mathbf{m}}{\mathbf{Lt^2}}\right\} \qquad (5)$$

Or, in exponent form, the dimensions of both sides of the equation are $\{\mathbf{m^1\,L^{-1}\,t^{-2}}\}$. Thus we see that **Hooke's law is indeed dimensionally homogeneous**.

Discussion If the dimensions of Eq. 5 were not homogeneous, we would surely expect that we made an error somewhere.

7-92

Solution We are to find the functional relationship between the given parameters and name any established dimensionless parameters.

Assumptions **1** The given parameters are the only ones relevant to the flow at hand.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; n = 5,

List of relevant parameters: $$z_d = f\left(F, L, E, I\right) \qquad n = 5 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

z_d	F	L	E	I
$\{L^1\}$	$\{m^1L^1t^{-2}\}$	$\{L^1\}$	$\{m^1L^{-1}t^{-2}\}$	$\{L^4\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since j = 3. We cannot pick both length L and moment of inertia I since their dimensions differ only by a power. We also notice that we cannot choose F, L, and E since these three parameters can form a Π all by themselves. So, we set j = 3 – 1 = 2, and we choose two repeating parameters, expecting 5 – 2 = 3 Πs,

Repeating parameters: L and E

Step 5 The Πs are generated. Note that for the first Π we do the algebra in our heads since z_d has the same dimensions as L. The dependent Π is

Π$_1$: $$\Pi_1 = \frac{z_d}{L}$$

This Π is not an established dimensionless group, although it is a ratio of two lengths, **similar to an aspect ratio**.

We form the second Π with force F:

$$\Pi_2 = FL^aE^b \qquad \{\Pi_2\} = \left\{\left(m^1L^1t^{-2}\right)\left(L^1\right)^a\left(m^1L^{-1}t^{-2}\right)^b\right\}$$

mass: $$\{m^0\} = \{m^1m^b\} \qquad 0 = 1 + b \qquad b = -1$$

time: $$\{t^0\} = \{t^{-2}t^{-2b}\} \qquad 0 = -2 - 2b \qquad b = -1$$

length: $$\{L^0\} = \{L^1L^aL^{-b}\} \qquad \begin{gathered} 0 = 1 + a - b \\ a = -1 + b \end{gathered} \qquad a = -2$$

which yields

Π$_2$: $$\Pi_2 = \frac{F}{L^2E}$$

We do not recognize Π_2 as a named dimensionless parameter.

The final Π is formed with moment of inertia. Since $\{I\} = \{L^4\}$, there is no need to go through the algebra – we write

Π_3:
$$\Pi_3 = \frac{I}{L^4}$$

Again, we do not recognize Π_2 as a named dimensionless parameter.

Step 6 We write the final functional relationship as

Relationship between Π*s*:
$$\boxed{\frac{z_d}{L} = f\left(\frac{F}{L^2 E}, \frac{I}{L^4}\right)} \qquad (2)$$

Discussion We cannot determine the *form* of the relationship by purely dimensional reasoning since there are three Πs.

7-93

Solution We are to generate dimensionless relationships among given parameters, and then we are to discuss how ΔP decreases if the time is doubled.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis (*a*) We perform dimensional analyses using the method of repeating variables. First we analyze ΔP:

Step 1 There are four parameters in this problem; $n = 4$,

List of relevant parameters: $$\Delta P = f\left(t, c, E\right) \qquad n = 4 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

ΔP	t	c	E
$\{m^1L^{-1}t^{-2}\}$	$\{t^1\}$	$\{L^1t^{-1}\}$	$\{m^1L^2t^{-2}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 4 - 3 = 1$$

Step 4 We need to choose three repeating parameters since $j = 3$. We pick all the independent parameters – time t, speed of sound c, and energy E,

Repeating parameters: t, c, and E

Step 5 The dependent Π is generated:

$$\Pi_1 = \Delta P \times t^{a_1} c^{b_1} E^{c_1} \qquad \{\Pi_1\} = \left\{\left(m^1L^{-1}t^{-2}\right)\left(t^1\right)^{a_1}\left(L^1t^{-1}\right)^{b_1}\left(m^1L^2t^{-2}\right)^{c_1}\right\}$$

mass: $$\{m^0\} = \{m^1m^{c_1}\} \qquad 0 = 1 + c_1 \qquad c_1 = -1$$

length: $$\{L^0\} = \{L^{-1}L^{b_1}L^{2c_1}\} \qquad \begin{aligned} 0 &= -1 + b_1 + 2c_1 \\ b_1 &= 1 - 2c_1 \end{aligned} \qquad b_1 = 3$$

time: $$\{t^0\} = \{t^{-2}t^{a_1}t^{-b_1}t^{-2c_1}\} \qquad \begin{aligned} 0 &= -2 + a_1 - b_1 - 2c_1 \\ a_1 &= 2 + b_1 + 2c_1 \end{aligned} \qquad a_1 = 3$$

The dependent Π is thus

Π_1 for ΔP: $$\Pi_1 = \frac{t^3c^3\Delta P}{E}$$

This Π is not an established one, so we leave it as is.

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\Delta P = \text{constant}\,\frac{E}{t^3c^3}} \qquad (2)$$

We perform a similar dimensional analysis using the same repeating variables, but this time for radius r. We do not show the algebra since the Π can be found by inspection. We get

Π_1 for r: $$\Pi_1 = \frac{r}{ct}$$

Since this is the only Π, it must be equal to a constant,

Relationship between Π*s*: $$r = \text{constant} \cdot ct \qquad (3)$$

(b) From Eq. 2 we see that **if t is doubled, ΔP decreases by a factor of $2^3 = 8$**.

Discussion The pressure rise across the blast wave decays rapidly with time (and with distance from the explosion). The speed of sound depends on temperature. If the explosion is of sufficient strength, T will increase significantly and c will not remain constant.

7-94

Solution We are to find an alternate definition of Archimedes number, and list it following the format of Table 7-5. Then we are to find an established Π group that is similar.

Analysis Students' responses will vary. There seems to be a plethora of definitions of Archimedes number. Here is the one most appropriate for buoyant fluids:

Name	**Definition**	**Ratio of significance**
Archimedes number	$\text{Ar} = \dfrac{gL\Delta\rho}{\rho V^2}$	$\dfrac{\text{buoyant force}}{\text{inertial force}}$

In the above, $\Delta\rho$ is a characteristic density difference in the fluid (due to buoyancy) and ρ is a characteristic or average density of the fluid. A glance through Table 7-5 shows that the **Richardson number** is very similar to this alternative definition of Ar. In fact, the alternate form of Ri (Problem 7-55) is *identical* to our new Ar.

Discussion Some students may find other definitions that are also valid. For example, $\Delta\rho/\rho$ may be replaced by $\Delta T/T$.

7-95

Solution We are to generate a dimensionless relationship between the given parameters.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane. **3** The flow is fully developed.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; $n = 5$,

List of relevant parameters: $$u = f\left(h, \frac{dP}{dx}, \mu, y\right) \qquad n = 5 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

u	h	dP/dx	μ	y
$\{L^1 t^{-1}\}$	$\{L^1\}$	$\{m^1 L^{-2} t^{-2}\}$	$\{m^1 L^{-1} t^{-1}\}$	$\{L^1\}$

Step 3 As a first guess, *j* is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of *j* is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since $j = 3$. We cannot pick both *h* and *y* since they have the same dimensions. We choose

Repeating parameters: *h*, *dP/dx*, and μ

Step 5 The dependent Π is generated:

$$\Pi_1 = u h^{a_1} \left(\frac{dP}{dx}\right)^{b_1} \mu^{c_1} \qquad \{\Pi_1\} = \left\{(L^1 t^{-1})(L^1)^{a_1} (m^1 L^{-2} t^{-2})^{b_1} (m^1 L^{-1} t^{-1})^{c_1}\right\}$$

mass: $\{m^0\} = \{m^{b_1} m^{c_1}\}$ $\qquad 0 = b_1 + c_1 \qquad c_1 = -b_1$

time: $\{t^0\} = \{t^{-1} t^{-2b_1} t^{-c_1}\}$ $\qquad 0 = -1 - 2b_1 - c_1 \qquad b_1 = -1$

$0 = -1 - b_1 \qquad c_1 = 1$

length: $\{L^0\} = \{L^1 L^{a_1} L^{-2b_1} L^{-c_1}\}$ $\qquad 0 = 1 + a_1 - 2b_1 - c_1 \qquad a_1 = -2$

$0 = 1 + a_1 + 1$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{\mu u}{h^2 \dfrac{dP}{dx}}$$

The independent Π is generated with variable *y*. Since $\{y\} = \{L\}$, and this is the same as one of the repeating variables (*h*), Π_2 is simply y/h,

Π_2:

$$\Pi_2 = \frac{y}{h}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:

$$\boxed{\Pi_1 = \frac{\mu u}{h^2 \dfrac{dP}{dx}} = f\left(\frac{y}{h}\right)} \qquad (2)$$

Discussion If we were to solve this problem exactly (using the methods of Chap. 9) we would see that the functional relationship of Eq. 2 is correct.

7-96

Solution We are to generate a dimensionless relationship between the given parameters and then analyze the behavior of u_{max} when an independent variable is doubled.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane. **3** The flow is fully developed.

Analysis (*a*) A step-by-step dimensional analysis procedure could be performed. However, we notice that u_{max} has the same dimensions as *u*. Therefore the algebra would be identical to that of the previous problem except that there is only one Π instead of two since *y* is no longer a parameter. The result is

Relationship between Πs:

$$\Pi_1 = \frac{\mu u_{\max}}{h^2 \dfrac{dP}{dx}} = \text{constant} = C \qquad (1)$$

or

Final relationship for u_{max}:

$$\boxed{u_{\max} = C\frac{h^2}{\mu}\frac{dP}{dx}} \qquad (2)$$

Alternatively, we can use the results of the previous problem directly. Namely, since we know that the maximum velocity occurs at the centerline, $y/h = 1/2$ there, and is a constant. Hence, Eq. 2 of the previous problem reduces to Eq. 1 of the present problem.

(*b*) If *h* doubles, we see from Eq. 2 that u_{max} will increase by a factor of $2^2 = 4$.

(*c*) If dP/dx doubles, we see from Eq. 2 that u_{max} will increase by a factor of $2^1 = 2$.

(*d*) Since there is only one Π in this problem, we would need to conduct only *one experiment* to determine the constant *C* in Eq. 2.

Discussion The constant turns out to be -1/8, but there is no way to determine this from dimensional analysis alone. To obtain the constant, we would need to either do an experiment, or solve the problem exactly using the methods discussed in Chap. 9.

7-97

Solution We are to generate a relationship for Darcy friction factor f in terms of Euler number Eu. We are then to plot f as a function of Re and discuss whether Reynolds number independence has been achieved.

Assumptions **1** The flow is fully developed. **2** The flow is steady and incompressible.

Properties For water at T = 20°C and atmospheric pressure, ρ = 998.0 kg/m^3 and $\mu = 1.002 \times 10^{-3}$ kg/m·s.

Analysis (*a*) Since the flow is fully developed, the control volume cuts through two cross sections in which the velocity profiles are identical. The flow is also steady, so the control volume momentum equation in the horizontal (x) direction reduces to

Conservation of momentum:

$$\sum F_x = \sum F_{x,\text{ pressure}} + \sum F_{x,\text{ shear stress}} = 0 \qquad (1)$$

We multiply pressure by cross-sectional area to obtain the pressure force, and wall shear stress times inner pipe wall surface area to obtain the shear stress force,

$$\sum F_{x,\text{ pressure}} = \Delta P \frac{\pi D^2}{4} \qquad \sum F_{x,\text{ shear stress}} = -\tau_w \pi DL \qquad (2)$$

Note the negative sign in the shear stress term since τ_w points to the left. We substitute Eq. 2 into Eq. 1. After some algebra,

Result:

$$\Delta P = \frac{4\tau_w L}{D} \qquad (3)$$

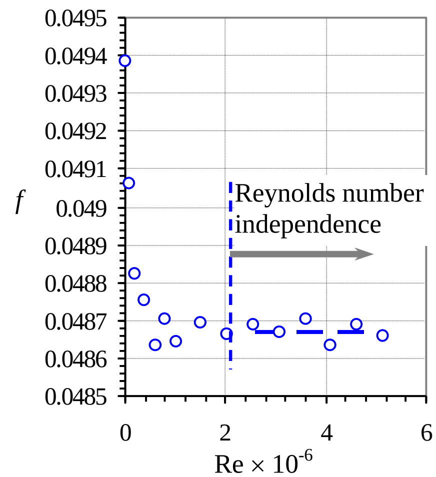


FIGURE 1
Nondimensionalized experimental data from a section of pipe.

Finally, we divide both sides of Eq. 3 by ρV^2 to convert ΔP into an Euler number,

Nondimensional relationship:

$$\text{Eu} = \frac{\Delta P}{\rho V^2} = \frac{4\tau_w L}{\rho V^2 D} = \frac{1}{2}\frac{L}{D}\left(\frac{8\tau_w}{\rho V^2}\right) \qquad (4)$$

We recognize the term in parentheses on the right as the Darcy friction factor. Thus,

Final nondimensional relationship:

$$\boxed{\text{Eu} = \frac{1}{2}\frac{L}{D} f} \quad \text{or} \quad \boxed{f = 2\frac{D}{L}\text{Eu}} \qquad (5)$$

(*b*) We use Eq. 5 to calculate f at each data point of Table P7-74. We plot f as a function of Re in Fig. 1. We see that the behavior of f mimics that of Eu (as it must because of Eq. 5 where we see that f is just a constant times Eu). Since Eu shows Reynolds number independence for Re greater than about 2×10^6, so does f. **We see Reynolds number independence for Re greater than about 2×10^6**. From the plot, the extrapolated value of f at large Re is about 0.04867, which agrees with Eq. 5 when we plug in the Re-independent value of Eu,

Extrapolated value of f:

$$f = 2\frac{D}{L}\text{Eu} = 2\frac{0.104\text{ m}}{1.3\text{ m}}(0.3042) = \mathbf{0.0487} \qquad (6)$$

Discussion We show in Chap. 8 (the Moody chart) that f does indeed flatten out at high enough values of Re, depending on the relative roughness height, ε/D.

7-98

Solution We are to create characteristic scales so that we can define a desired established dimensionless parameter.

Analysis (*a*) For Froude number we need a velocity scale, a length scale, and gravity. We already have a length scale and gravity. We create a velocity scale as $\dot{V}'/L$. We then define a Froude number as

Froude number:
$$\boxed{\mathrm{Fr} = \frac{V}{\sqrt{gL}} = \frac{\dot{V}'}{L\sqrt{gL}} = \frac{\dot{V}'}{\sqrt{gL^3}}}$$

(*b*) For Reynolds number we need a velocity scale, a length scale, and kinematic viscosity. Of these we only have the kinematic viscosity, so we need to create a velocity scale and a length scale. After a "back of the envelope" analysis, we create a velocity scale as $\dot{V}'/L$ where L is some undefined characteristic length scale. Thus,

Reynolds number:
$$\boxed{\mathrm{Re} = \frac{LV}{\nu} = \frac{L\dot{V}'}{L\nu} = \frac{\dot{V}'}{\nu}}$$

Note that in this case, the length scales drop out, so it doesn't matter that we could not define a length scale from the given parameters.

(*c*) For Richardson number we need a length scale, the gravitational constant, a volume flow rate, a density, and a density difference. Of these we have all but the volume flow rate, so we create a volume flow rate scale as $\dot{V}'\,L$. Thus,

Richardson number:
$$\boxed{\mathrm{Ri} = \frac{L^5 g\Delta\rho}{\rho\dot{V}^2} = \frac{L^5 g\Delta\rho}{\rho\left(\dot{V}'L\right)^2} = \frac{L^3 g\Delta\rho}{\rho\left(\dot{V}'\right)^2}}$$

Discussion You can verify that each of the parameters above is dimensionless.

7-99

Solution We are to find the functional relationship between the given parameters and name any established dimensionless parameters.

Assumptions **1** The given parameters are the only ones relevant to the flow at hand.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are seven parameters in this problem; $n = 7$,

List of relevant parameters: $$V = f\left(d, D, \rho, \mu, h, g\right) \qquad n = 7 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

V	d	D	ρ	μ	h	g
$\{L^1t^{-1}\}$	$\{L^1\}$	$\{L^1\}$	$\{m^1L^{-3}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^1\}$	$\{L^1t^{-2}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 7 - 3 = 4$$

Step 4 We need to choose three repeating parameters since $j = 3$. We pick length scale h, fluid density ρ, and gravitational constant g.

Repeating parameters: h, ρ, and g

Step 5 The Πs are generated. Note that in this case we do the algebra in our heads since these relationships are very simple. The dependent Π is

Π_1 = *a Froude number*: $$\Pi_1 = \frac{V}{\sqrt{gh}}$$

This Π is a type of **Froude number**. Similarly, the two length-scale Πs are obtained easily,

Π_2: $$\Pi_2 = \frac{d}{h}$$

and

Π_3: $$\Pi_3 = \frac{D}{h}$$

Finally, the Π formed with viscosity is generated,

$$\Pi_4 = \mu h^{a_4} \rho^{b_4} g^{c_4} \qquad \{\Pi_4\} = \left\{\left(m^1L^{-1}t^{-1}\right)\left(L^1\right)^{a_4}\left(m^1L^{-3}\right)^{b_4}\left(L^1t^{-2}\right)^{c_4}\right\}$$

mass: $$\{m^0\} = \{m^1m^{b_4}\} \qquad 0 = 1 + b_4 \qquad b_4 = -1$$

time: $$\{t^0\} = \{t^{-1}t^{-2c_4}\} \qquad 0 = -1 - 2c_4 \qquad c_4 = -\frac{1}{2}$$

length: $\left\{\mathrm{L}^0\right\} = \left\{\mathrm{L}^{-1}\mathrm{L}^{a_4}\mathrm{L}^{-3b_4}\mathrm{L}^{c_4}\right\}$

$$0 = -1 + a_4 - 3b_4 + c_4$$
$$0 = -1 + a_4 + 3 - \tfrac{1}{2}$$
$$a_4 = -\frac{3}{2}$$

which yields

Π_4:
$$\Pi_4 = \frac{\mu}{\rho h^{\frac{3}{2}}\sqrt{g}}$$

We recognize this Π as the inverse of a kind of **Reynolds number**. We also split the *h* terms to separate them into a length scale and (when combined with *g*) a velocity scale. The final form is

Modified Π_4 *= a Reynolds number*:
$$\Pi_4 = \frac{\rho h\sqrt{gh}}{\mu}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:
$$\boxed{\frac{V}{\sqrt{gh}} = f\left(\frac{d}{h}, \frac{D}{h}, \frac{\rho h\sqrt{gh}}{\mu}\right)} \qquad (2)$$

Discussion You may choose different repeating variables, and may generate different nondimensional groups. If you do the algebra correctly, your answer is not "wrong" – you just may not get the same dimensionless groups.

7-100

Solution We are to find a dimensionless relationship among the given parameters.

Assumptions **1** The given parameters are the only ones relevant to the flow at hand.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are seven parameters in this problem; n = 7,

List of relevant parameters: $$t_{\text{empty}} = f\left(d, D, \rho, \mu, h, g\right) \qquad n = 7 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

t_{empty}	d	D	ρ	μ	h	g
$\{t^1\}$	$\{L^1\}$	$\{L^1\}$	$\{m^1L^{-3}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^1\}$	$\{L^1t^{-2}\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 7 - 3 = 4$$

Step 4 We need to choose three repeating parameters since j = 3. We pick length scale h, fluid density ρ, and gravitational constant g. (Note: these are the same repeating parameters as in the previous problem.)

Repeating parameters: h, ρ, and g

Step 5 The Πs are generated. We leave out the details since the algebra is trivial and can be done by inspection in most cases. The dependent Π is

Π_1: $$\Pi_1 = t_{\text{empty}}\sqrt{\frac{g}{h}}$$

The rest of the Πs are identical to those of the previous problem.

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{t_{\text{empty}}\sqrt{\frac{g}{h}} = f\left(\frac{d}{h}, \frac{D}{h}, \frac{\rho h\sqrt{gh}}{\mu}\right)} \qquad (2)$$

Discussion You may choose different repeating variables, and may generate different nondimensional groups. If you do the algebra correctly, your answer is not "wrong" – you just may not get the same dimensionless groups.

7-101

Solution We are to calculate the temperature of water in a model test to ensure similarity with the prototype, and we are to predict the time required to empty the prototype tank.

Assumptions **1** The parameters specified in the previous problem are the only parameters relevant to the problem. **2** The model and prototype are geometrically similar.

Properties For ethylene glycol at 60°C, $\nu = \mu/\rho = 4.75\times10^{-6}$ m^2/s (given).

Analysis

(*a*) We use the functional relationship obtained in the previous problem,

Dimensionless relationship:
$$t_{\text{empty}}\sqrt{\frac{g}{h}} = f\left(\frac{d}{h}, \frac{D}{h}, \frac{\rho h\sqrt{gh}}{\mu}\right) \qquad (1)$$

Since the model and prototype are geometrically similar, $(d/h)_{\text{model}} = (d/h)_{\text{prototype}}$ and $(D/h)_{\text{model}} = (D/h)_{\text{prototype}}$. Thus, we are left with only one Π to match to ensure similarity. Namely, the Reynolds number parameter in Eq. 1 must be matched between model and prototype. Since g remains the same in either case, and using "m" for model and "p" for prototype,

Similarity:
$$\left(\frac{\rho h\sqrt{gh}}{\mu}\right)_{\text{m}} = \left(\frac{\rho h\sqrt{gh}}{\mu}\right)_{\text{p}} \quad \text{or} \quad \frac{\rho_{\text{m}}}{\mu_{\text{m}}} = \frac{\rho_{\text{p}}}{\mu_{\text{p}}}\left(\frac{h_{\text{p}}}{h_{\text{m}}}\right)^{\frac{3}{2}} \qquad (2)$$

We recognize that $\nu = \mu/\rho$, and we know that $h_{\text{p}}/h_{\text{m}} = 4$. Thus, Eq. 2 reduces to

Similarity:
$$\nu_{\text{m}} = \nu_{\text{p}}\left(\frac{h_{\text{p}}}{h_{\text{m}}}\right)^{\frac{-3}{2}} = 4.75\times10^{-6}\ \text{m}^2/\text{s}\,(4)^{\frac{-3}{2}} = 5.94\times10^{-7}\ \text{m}^2/\text{s} \qquad (3)$$

For similarity we need to find the temperature of water where the kinematic viscosity is 5.94×10^{-7} m^2/s. By interpolation from the property tables, **the designers should run the model tests at a water temperature of 45.8°C**.

(*b*) At dynamically similar conditions, Eq. 1 yields

At dynamically similar conditions:

$$\left(t_{\text{empty}}\sqrt{\frac{g}{h}}\right)_{\text{p}} = \left(t_{\text{empty}}\sqrt{\frac{g}{h}}\right)_{\text{m}} \rightarrow t_{\text{empty,p}} = t_{\text{empty,m}}\sqrt{\frac{h_{\text{p}}}{h_{\text{m}}}} = 3.27\ \text{min}\sqrt{4} = \mathbf{6.54\ min}$$

Discussion We set up Eqs. 3 and 5 in terms of ratios of h_{p} to h_{m} so that the actual dimensions are not needed – just the *ratio* is needed, and it is given.

7-102

Solution For the simplified case in which V depends only on h and g, we are to determine how V increases when h is doubled.

Assumptions **1** The given parameters are the only ones relevant to the problem.

Analysis We employ the dimensional analysis results of Problem 7-90. Dropping d, D, ρ, and μ from the list of parameters, we are left with n = 3,

List of relevant parameters:
$$V = f\left(h, g\right) \qquad n = 3 \qquad (1)$$

We perform the analysis in our heads –only one Π remains, and it is therefore set to a constant. The final result of the dimensional analysis is

Relationship between Π*s*:
$$\frac{V}{\sqrt{gh}} = \text{constant} \qquad (2)$$

Thus, when h is doubled, we can easily calculate the factor by which V increases,

Increase in V:
$$\boxed{\frac{V_2}{\sqrt{gh_2}} = \frac{V_1}{\sqrt{gh_1}} \quad \text{or} \quad V_2 = V_1\sqrt{\frac{h_2}{h_1}} = V_1\sqrt{2}} \qquad (3)$$

Thus, **when *h* increases by a factor of 2, *V* increases by a factor of** $\sqrt{2}$.

Discussion We don't need to know the constant in Eq. 2 to solve the problem. However, it turns out that the constant is $\sqrt{2}$ (see Chap. 5).

7-103

Solution We are to verify the dimensions of particle relaxation time τ_p, and then identify the established dimensionless parameter formed by nondimensionalization of τ_p.

Analysis First we obtain the primary dimensions of τ_p,

Primary dimensions of τ_p:
$$\left\{\tau_p\right\} = \left\{\frac{\frac{\text{m}}{\text{L}^3}\times\text{L}^2}{\frac{\text{m}}{\text{Lt}}}\right\} = \left\{\mathbf{t}\right\}$$

A characteristic time scale for the air flow is L/V. Thus, we nondimensionalize τ_p,

Nondimensionalized particle relaxation time:
$$\tau_p{}^* = \tau_p = \frac{\rho_p d_p{}^2}{18\mu}\frac{V}{L}$$

From Table 7-5 we recognize this as the **Stokes number**, Stk,

Stokes number:
$$\boxed{\text{Stk} = \frac{\rho_p d_p{}^2 V}{18\mu L}}$$

Discussion Stokes number is useful when studying the flow of aerosol particles.

7-104

Solution We are to compare the primary dimensions of each given property in mass-based and force-based primary dimensions, and discuss.

Analysis From previous problems and examples in this chapter, we can write down the primary dimensions of each property in the mass-based system. We use the fundamental definitions of these quantities to generate the primary dimensions in the force-based system:

(*a*) For pressure *P* the primary dimensions are

Mass-based primary dimensions	**Force-based primary dimensions**
$\boxed{\{P\} = \left\{\frac{\text{m}}{\text{t}^2\text{L}}\right\}}$	$\boxed{\{P\} = \left\{\frac{\text{force}}{\text{area}}\right\} = \left\{\frac{\text{F}}{\text{L}^2}\right\}}$

(*b*) For moment $\vec{M}$ the primary dimensions are

Mass-based primary dimensions	**Force-based primary dimensions**
$\boxed{\{\vec{M}\} = \left\{\text{m}\frac{\text{L}^2}{\text{t}^2}\right\}}$	$\boxed{\{\vec{M}\} = \{\text{force} \times \text{moment arm}\} = \{\text{FL}\}}$

(*c*) For energy *E* the primary dimensions are

Mass-based primary dimensions	**Force-based primary dimensions**
$\boxed{\{E\} = \left\{\text{m}\frac{\text{L}^2}{\text{t}^2}\right\}}$	$\boxed{\{E\} = \{\text{force} \times \text{distance}\} = \{\text{FL}\}}$

We see that (in these three examples anyway), the forced-base cases have only two primary dimensions represented (F and L), whereas the mass-based cases have three primary dimensions represented (m, L, and t). Some authors would prefer the force-based system because of its **reduced complexity** when dealing with forces, pressures, energies, etc.

Discussion Not all variables have a simpler form in the force-based system. Mass itself for example has primary dimensions of {m} in the mass-based system, but has primary dimensions of $\{\text{Ft}^2/\text{L}\}$ in the force-based system. In problems involving mass, mass flow rates, and/or density, the force-based system may not have any advantage.

7-105

Solution We are to determine the relationship between four established nondimensional parameters, and then try to form the Stanton number by some combination of only *two* other established dimensionless parameters.

Analysis We manipulate Re, Nu, and Pr, guided by the known result. After some trial and error,

Stanton number:
$$\boxed{\mathrm{St}=\frac{\mathrm{Nu}}{\mathrm{Re}\times\mathrm{Pr}}=\frac{\frac{Lh}{k}}{\frac{\rho VL}{\mu}\times\frac{\mu c_P}{k}}=\frac{h}{\rho c_p V}} \qquad (1)$$

We recognize from Table 7-5 (or from the previous problem) that *Peclet number* is equal to the product of Reynolds number and Prandtl number. Thus,

Stanton number:
$$\boxed{\mathrm{St}=\frac{\mathrm{Nu}}{\mathrm{Pe}}=\frac{\frac{Lh}{k}}{\frac{\rho LVc_P}{k}}=\frac{h}{\rho c_p V}} \qquad (2)$$

Discussion Not all named, established dimensionless parameters are independent of other named, established dimensionless parameters.

7-106

Solution We are to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis First we do some thinking. If we imagine traveling at the same speed as the bottom plate, the flow would be identical to that of Problem 7-56 except that the top plate speed would be ($V_{\text{top}} - V_{\text{bottom}}$) instead of just V. The step-by-step method of repeating variables is otherwise identical to that of Problem 7-56, and the details are not included here. The final functional relationship is

Relationship between Πs:
$$\boxed{\frac{u}{V_{\text{top}}-V_{\text{bottom}}}=f\left(\mathrm{Re},\frac{y}{h}\right)} \qquad (1)$$

where

Reynolds number:
$$\boxed{\mathrm{Re}=\frac{\rho\left(V_{\text{top}}-V_{\text{bottom}}\right)h}{\mu}} \qquad (2)$$

Discussion It is always wise to look for shortcuts like this to save us time.

7-107

Solution We are to determine the primary dimensions of electrical charge.

Analysis The fundamental definition of electrical current is charge per unit time. Thus,

Primary dimensions of charge:
$$\{q\}=\{\text{current}\times\text{time}\}=\{\mathbf{I\ t}\} \qquad (1)$$

Or, in exponent form, $\{q\}$ = $\{\mathbf{t}^1\ \mathbf{I}^1\}$.

Discussion We see that all dimensions, even those of electrical properties, can be expressed in terms of primary dimensions.

7-108

Solution We are to determine the primary dimensions of electrical capacitance.

Analysis Electrical capacitance C is measured in units of farads (F). By definition, a one-farad capacitor with an applied electric potential of one volt across it will store one coulomb of electrical charge. Thus,

Primary dimensions of capacitance:

$$\{C\} = \{\text{charge / voltage}\} = \left\{ \frac{\text{I t}}{\dfrac{\text{mL}^2}{\text{t}^3\text{I}}} \right\} = \left\{ \frac{\mathbf{I^2 t^4}}{\mathbf{mL^2}} \right\} \qquad (1)$$

where the primary dimensions of voltage are obtained from Problem 7-10, and those of electric charge are obtained from the previous problem. Or, in exponent form, $\{C\}$ = $\mathbf{\{m^{-1}\,L^{-2}\,t^4\,I^2\}}$.

Discussion We see that all dimensions, even those of electrical properties, can be expressed in terms of primary dimensions.

7-109

Solution We are to determine the primary dimensions of electrical time constant RC, and discuss the significance of our result.

Analysis The primary dimensions of electrical resistance are obtained from Problem 7-11. Those of electrical capacitance C are obtained from the previous problem. Thus,

Primary dimensions of electrical time constant RC:

$$\{RC\} = \{\text{resistance} \times \text{capacitance}\} = \left\{ \frac{\text{mL}^2}{\text{t}^3\text{I}^2} \times \frac{\text{I}^2\text{t}^4}{\text{mL}^2} \right\} = \{\mathbf{t}\} \qquad (1)$$

Thus we see that the primary dimensions of RC are those of *time*. This explains why a resistor and capacitor in series is often used in timing circuits.

Discussion The cut-off frequency of the low-pass filter is proportional to $1/R{\cdot}C$. If the resistor and the capacitor were to swap places we would have a high-pass rather than a low-pass filter.

7-110

Solution We are to determine the primary dimensions of both sides of the equation, and we are to verify that the equation is dimensionally homogeneous.

Analysis The primary dimensions of the time derivative (d/dt) are 1/time. The primary dimensions of capacitance are current^2×time^4 / (mass×length^2), as obtained from Problem 7-101. Thus both sides of the equation can be written in terms of primary dimensions,

$$\{I\} = \{\text{current}\} \qquad\qquad \{I\} = \{\text{I}\}$$

$$C\frac{dE}{dt} = \left\{ \frac{\text{current}^2 \times \text{time}^4}{\text{mass} \times \text{length}^2} \frac{\dfrac{\text{mass} \times \text{length}^2}{\text{current} \times \text{time}^3}}{\text{time}} \right\} = \{\text{current}\} \qquad\qquad \left\{ C\frac{dE}{dt} \right\} = \{\text{I}\}$$

Indeed, **both sides of the equation have the same dimensions, namely {*I*}**.

Discussion Current is one of our seven primary dimensions. These results verify our algebra in Problem 7-101.

7-111

Solution We are to find the functional relationship between the given parameters, and then answer some questions about scaling.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis

(*a*) The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are five parameters in this problem; $n = 5$,

List of relevant parameters: $$w = f\left(q_p, E_f, \mu, D_p\right) \qquad n = 5 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

w	q_p	E_f	μ	D_p
$\{L^1t^{-1}\}$	$\{I^1t^1\}$	$\{m^1L^1t^{-3}I^{-1}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^1\}$

where the primary dimensions of voltage are obtained from Problem 7-10, and those of electric charge are obtained from Problem 7-100.

Step 3 As a first guess, j is set equal to 4, the number of primary dimensions represented in the problem (m, L, t, and I).

Reduction: $$j = 4$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 4 = 1$$

Step 4 We need to choose four repeating parameters since $j = 4$. We only have one choice in this problem, since there are only four independent parameters on the right-hand side of Eq. 1,

Repeating parameters: q_p, E, D_p, and μ

Step 5 The dependent Π is generated:

$$\Pi_1 = w q_p^{a_1} E_f^{b_1} \mu^{c_1} D_p^{d_1} \qquad \{\Pi_1\} = \left\{\left(L^1t^{-1}\right)\left(I^1t^1\right)^{a_1}\left(m^1L^1t^{-3}I^{-1}\right)^{b_1}\left(m^1L^{-1}t^{-1}\right)^{c_1}\left(L^1\right)^{d_1}\right\}$$

current: $$\{I^0\} = \{I^{a_1}I^{-b_1}\} \qquad 0 = a_1 - b_1 \qquad a_1 = b_1$$

mass: $$\{m^0\} = \{m^{b_1}m^{c_1}\} \qquad 0 = b_1 + c_1 \qquad c_1 = -b_1 = -a_1$$

time: $$\{t^0\} = \{t^{-1}t^{a_1}t^{-3b_1}t^{-c_1}\} \qquad 0 = -1 + a_1 - 3b_1 - c_1 \qquad a_1 = b_1 = -1, \quad c_1 = 1$$

length: $$\{L^0\} = \{L^1L^{b_1}L^{-c_1}L^{d_1}\} \qquad 0 = 1 + b_1 - c_1 + d_1 \qquad d_1 = 1$$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{w \mu D_p}{q_p E_f}$$

Step 6 Since there is only one Π, it is a function of nothing. This is only possible if we set the Π equal to a constant. We write the final functional relationship as

Relationship between Πs:

$$\boxed{\Pi_1 = \frac{w\mu D_p}{q_p E} = \text{constant}} \qquad (2)$$

(*b*) We re-write Eq. 2 as

Equation for w:

$$\boxed{w = \text{constant}\,\frac{q_p E_f}{\mu D_p}} \qquad (3)$$

Thus, **if we double the electric field strength, the drift velocity will increase by a factor of 2**.

(c) Also from Eq. 3 we see that **if we double the particle size, the drift velocity will decrease by a factor of 2**.

Discussion These results agree with our intuition. Certainly we would expect the drift velocity to increase if we increase the field strength. Also, larger particles have more aerodynamic drag, so for the same charge, we would expect a larger dust particle to drift more slowly than a smaller dust particle.

7-112

Solution We are to generate a nondimensional relationship between the given parameters.

Assumptions **1** The fluid is incompressible. **2** No other parameters are significant in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters.

Step 1 All the relevant parameters in the problem are listed in functional form:

List of relevant parameters: $F = f\left(V_1, \Delta P, \rho, \mu, A_1, A_2, L\right) \qquad n = 8$

Step 2 The primary dimensions of each parameter are listed:

F	V_1	ΔP	ρ	μ	A_1	A_1	L
$\{m^1L^1t^{-2}\}$	$\{L^1t^{-1}\}$	$\{m^1L^{-1}t^{-2}\}$	$\{m^1L^{-3}\}$	$\{m^1L^{-1}t^{-1}\}$	$\{L^2\}$	$\{L^2\}$	$\{L^1\}$

Step 3 As a first guess, *j* is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $j = 3$

If this value of *j* is correct, the expected number of Πs is

Number of expected Π*s*: $k = n - j = 8 - 3 = 5$

Step 4 We need to choose three repeating parameters since *j* = 3. Following the guidelines listed in Table 7-3, we cannot pick the dependent variable, *F*. We cannot choose any two of parameters A_1, A_2, and *L* since length^1 and length^2 are related by an exponent. It is not desirable to have μ or ΔP appear in all the Πs. The best choice of repeating parameters is thus V_1, ρ, and one of the length scales. We choose A_1.

Repeating parameters: V_1, A_1, and ρ

Step 5 The dependent Π is generated:

$$\Pi_1 = F V_1^{a_1} A_1^{b_1} \rho^{c_1} \qquad \{\Pi_1\} = \left\{\left(m^1L^1t^{-2}\right)\left(L^1t^{-1}\right)^{a_1}\left(L^2\right)^{b_1}\left(m^1L^{-3}\right)^{c_1}\right\}$$

mass: $\{m^0\} = \{m^1 m^{c_1}\} \qquad 0 = 1 + c_1 \qquad c_1 = -1$

time: $\{t^0\} = \{t^{-2} t^{-a_1}\} \qquad 0 = -2 - a_1 \qquad a_1 = -2$

length: $\{L^0\} = \{L^1 L^{a_1} L^{2b_1} L^{-3c_1}\}$

$$0 = 1 + a_1 + 2b_1 - 3c_1$$
$$0 = 1 - 2 + 2b_1 + 3 \qquad b_1 = -1$$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{F}{\rho V_1^2 A_1}$$

From Table 7-5, the established nondimensional parameter most similar to our Π_1 is a kind of **force coefficient** (similar to a lift or drag coefficient) which we shall call C_F. No manipulation is required, although a constant of ½ is often placed in the denominator in parameters like this.

We form the second Π with ΔP.

$$\Pi_2 = \Delta P V_1^{a_2} A_1^{b_2} \rho^{c_2} \qquad \{\Pi_2\} = \left\{ \left(\mathrm{m^1L^{-1}t^{-2}}\right)\left(\mathrm{L^1t^{-1}}\right)^{a_2} \left(\mathrm{L^2}\right)^{b_2} \left(\mathrm{m^1L^{-3}}\right)^{c_2} \right\}$$

mass: $\{\mathrm{m}^0\} = \{\mathrm{m^1m}^{c_2}\}$ $\qquad 0 = 1 + c_2 \qquad c_2 = -1$

time: $\{\mathrm{t}^0\} = \{\mathrm{t^{-2}t}^{-a_2}\}$ $\qquad 0 = -2 - a_2 \qquad a_2 = -2$

length: $\{\mathrm{L}^0\} = \{\mathrm{L^{-1}L^{a_2}L^{2b_2}L^{-3c_2}}\}$ $\qquad 0 = -1 + a_2 + 2b_2 - 3c_2 \qquad b_2 = 0$

$$0 = -1 - 2 + 2b_2 + 3$$

The second Π (first independent Π) is thus

Π_2:
$$\Pi_2 = \frac{\Delta P}{\rho V_1^2}$$

From Table 7-5, the established nondimensional parameter most similar to our Π_2 is the **Euler number** Eu. No manipulation is required.

We form the third Π with μ. By now we know that we will generate a type of **Reynolds number**. Here, since we chose area A_1 instead of a length, the length scale in our Reynolds number is the square root of A_1.

$$\Pi_3 = \mu V_1^{a_3} A_1^{b_3} \rho^{c_3} \qquad \Pi_3 = \frac{\rho V_1 \sqrt{A_1}}{\mu} = \text{Reynolds number} = \text{Re}$$

Note: In hindsight, it would probably have been better to use diameter d_1 as a parameter instead of A_1.

The final two Π groups are formed with A_2 and then with L. The algebra is trivial for these cases since their dimensions contain nothing but length). The results are

$$\Pi_4 = \frac{A_2}{A_1} = \text{Area ratio} \qquad \Pi_5 = \frac{L}{\sqrt{A_1}} = \text{A type of length-to-diameter ratio}$$

Step 6 We write the final functional relationship as

Relationship between Πs:
$$\boxed{C_F = \frac{F}{\rho V_1^2 A_1} = f\left(\frac{\Delta P}{\rho V_1^2}, \text{Re}, \frac{A_2}{A_1}, \frac{L}{\sqrt{A_1}}\right) \text{ where Re} = \frac{\rho V_1 \sqrt{A_1}}{\mu}}$$

Or, if the constant ½ is applied,

Alternate relationship between Πs:
$$\boxed{C_F = \frac{F}{\frac{1}{2}\rho V_1^2 A_1} = f\left(\frac{\Delta P}{\rho V_1^2}, \text{Re}, \frac{A_2}{A_1}, \frac{L}{\sqrt{A_1}}\right) \text{ where Re} = \frac{\rho V_1 \sqrt{A_1}}{\mu}}$$

Or, if we had chosen the inlet diameter d_1 as one of the variables instead of area A_1, we would have gotten

Alternate relationship between Πs:
$$\boxed{C_F = \frac{F}{\frac{1}{2}\rho V_1^2 d_1^2} = f\left(\frac{\Delta P}{\rho V_1^2}, \text{Re}, \frac{A_2}{d_1^2}, \frac{L}{d_1}\right) \text{ where Re} = \frac{\rho V_1 d_1}{\mu}}$$

Discussion The result applies to both laminar and turbulent flow. Any of the above is acceptable as there is never really only one "right" answer in dimensional analysis.

7-113

Solution We are to generate a dimensionless functional relationship between the given parameters and then compare our results with a known exact analytical solution.

Assumptions **1** There is no flow (hydrostatics). **2** The parameters listed here are the only relevant parameters in the problem.

Analysis (*a*) We perform a dimensional analysis using the method of repeating variables.

Step 1 There are five parameters in this problem; n = 6,

List of relevant parameters: $$h = f\left(\rho, g, \sigma_s, D, \phi\right) \qquad n = 6 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

h	ρ	g	σ_s	D	ϕ
$\{L^1\}$	$\{m^1L^{-3}\}$	$\{L^1t^{-2}\}$	$\{m^1t^{-2}\}$	$\{L^1\}$	$\{1\}$

Note that the dimensions of the contact angle are unity (angles are dimensionless).

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 6 - 3 = 3$$

Step 4 We need to choose three repeating parameters since j = 3. We cannot choose ϕ since it is dimensionless. We choose a length (D) and a density (ρ). We'd rather have gravitational constant g than surface tension σ_s in our Πs. So, we choose

Repeating parameters: $$\rho,\ g,\ D$$

Step 5 The dependent Π is generated. Since h has the same dimensions as D, we immediately write

Π_1: $$\Pi_1 = \frac{h}{D}$$

The first independent Π is generated by combining σ_s with the repeating parameters,

$$\Pi_2 = \sigma_s \rho^{a_2} g^{b_2} D^{c_2} \qquad \{\Pi_2\} = \left\{\left(m^1t^{-2}\right)\left(m^1L^{-3}\right)^{a_2}\left(L^1t^{-2}\right)^{b_2}\left(L^1\right)^{c_2}\right\}$$

mass: $$\{m^0\} = \{m^1m^{a_2}\} \qquad 0 = 1 + a_2 \qquad a_2 = -1$$

time: $$\{t^0\} = \{t^{-2}t^{-2b_2}\} \qquad 0 = -2 - 2b_2 \qquad b_2 = -1$$

length: $$\{L^0\} = \{L^{-3a_2}L^{b_2}L^{c_2}\} \qquad \begin{aligned} 0 &= -3a_2 + b_2 + c_2 \\ c_2 &= 3a_2 - b_2 \end{aligned} \qquad c_2 = -2$$

The first independent Π is thus

Π_2: $$\Pi_2 = \frac{\sigma_s}{\rho g D^2}$$

Finally, the third Π (second independent Π) is simply angle ϕ itself since it is dimensionless,

Π_3: $\Pi_3 = \phi$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\frac{h}{D} = f\left(\frac{\sigma_s}{\rho g D^2}, \phi\right)} \qquad (2)$$

(*b*) From Chap. 2 we see that the exact analytical solution is

Exact relationship: $$h = \frac{4\sigma_s}{\rho g D}\cos\phi \qquad (3)$$

Comparing Eqs. 2 and 3, we see that they are indeed of the same form. In fact,

Functional relationship: $$\boxed{\Pi_1 = \text{constant} \times \Pi_2 \times \cos\Pi_3} \qquad (4)$$

Discussion We cannot determine the constant in Eq. 4 by dimensional analysis. However, one experiment is enough to establish the constant. Or, in this case we can find the constant exactly. Viscosity is not relevant in this problem since there is no fluid motion.

7-114

Solution We are to find a functional relationship for the time scale required for the liquid to climb up the capillary tube.

Assumptions **1** t_{rise} is a function of the same parameters listed in the previous problem, but there is another relevant parameter.

Analysis Since this is an unsteady problem, the rise time will surely depend also on fluid viscosity μ. The list of parameters now involves seven parameters,

List of relevant parameters: $$t_{\text{rise}} = f\left(\rho, g, \sigma_s, D, \phi, \mu\right) \qquad n = 7 \qquad (1)$$

and we expect four Πs. We choose the same repeating parameters and the algebra is similar to that of the previous problem. It turns out that

Π_1: $$\Pi_1 = t_{\text{rise}}\sqrt{\frac{g}{D}}$$

The second and third Π are the same as those of the previous problem. Finally, the fourth Π is formed by combining μ with the repeating parameters. We expect some kind of Reynolds number. We can do the algebra in our head. Specifically, a velocity scale can be formed as $\sqrt{gD}$. Thus,

Π_4: $$\Pi_4 = \text{Re} = \frac{\rho D\sqrt{gD}}{\mu}$$

The final functional relationship is

Relationship between Πs: $$\boxed{t_{\text{rise}}\sqrt{\frac{g}{D}} = f\left(\frac{\sigma_s}{\rho g D^2}, \phi, \text{Re}\right)} \qquad (2)$$

Discussion If we would have defined a time scale as $\sqrt{D/g}$, we could have written Π_1 by inspection as well, saving ourselves some algebra.

7-115

Solution We are to use dimensional analysis to find the functional relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are four parameters in this problem; $n = 4$,

List of relevant parameters: $$I = f\left(P, c, \rho\right) \qquad n = 4 \qquad (1)$$

Step 2 The dimensions of I are those of power per area. The primary dimensions of each parameter are listed,

$$\begin{array}{cccc} I & P & c & \rho \\ \left\{\mathrm{m^1t^{-3}}\right\} & \left\{\mathrm{m^1L^{-1}t^{-2}}\right\} & \left\{\mathrm{L^1t^{-1}}\right\} & \left\{\mathrm{m^1L^{-3}}\right\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 4 - 3 = 1$$

Step 4 We need to choose three repeating parameters since $j = 3$. *The problem is that the three independent parameters form a* Π *all by themselves* ($c^2\rho/P$ is dimensionless). Let's see what happens if we don't notice this, and we pick all three independent parameters as repeating variables,

Repeating parameters: P, ρ, and c

Step 5 The Π is generated:

$$\Pi_1 = I \times P^a \rho^b c^c \qquad \left\{\Pi_1\right\} = \left\{\left(\mathrm{m^1t^{-3}}\right)\left(\mathrm{m^1L^{-1}t^{-2}}\right)^a \left(\mathrm{m^1L^{-3}}\right)^b \left(\mathrm{L^1t^{-1}}\right)^c\right\}$$

mass: $$\left\{\mathrm{m^0}\right\} = \left\{\mathrm{m^1m^a m^b}\right\} \qquad 0 = 1 + a + b \qquad a = -1 - b$$

time: $$\left\{\mathrm{t^0}\right\} = \left\{\mathrm{t^{-3}t^{-2a}t^{-c}}\right\} \qquad \begin{array}{l} 0 = -3 - 2a - c \\ c = -3 - 2a \end{array} \qquad c = -1 + 2b$$

length: $$\left\{\mathrm{L^0}\right\} = \left\{\mathrm{L^{-a}L^{-3b}L^c}\right\} \qquad \begin{array}{l} 0 = -a - 3b + c \\ c = a + 3b \end{array} \qquad c = -1 + 2b$$

This is a situation in which two of the equations agree, but we cannot solve for unique exponents. If we knew b, we could get a and c. The problem is that any value of b we choose will make the Π dimensionless. For example, if we choose $b = 1$, we find that $a = -2$ and $c = 1$, yielding

Π_1 for the case with $b = 1$: $$\Pi_1 = \frac{I\rho c}{P^2}$$

Since there is only one Π, we write

Functional relationship for the case with $b = 1$: $$I = \text{constant} \times \frac{P^2}{\rho c} \qquad (2)$$

However, if we choose a different value of b, say $b = -1$, then $a = 0$ and $c = -3$, yielding

Π_1 for the case with b = –1: $\Pi_1 = \dfrac{I}{\rho c^3}$

Since there is only one Π, we write

Functional relationship for the case with b = –1: $I = \text{constant} \times \rho c^3$ (3)

Similarly, you can come up with a whole family of possible answers, depending on your choice of b. We double check our algebra and realize that any value of b works. Hence *the problem is indeterminate with three repeating variables*.

We go back now and realize that something is wrong. As stated previously, the problem is that the three independent parameters can form a dimensionless group all by themselves. This is another case where we have to reduce j by 1. Setting j = 3 – 1 = 2, we choose two repeating parameters,

Repeating parameters: ρ and c

We jump to Step 5 of the method of repeating variables,

Step 5 The first Π is generated:

$$\Pi_1 = I\rho^a c^b \qquad \{\Pi_1\} = \left\{\left(\text{m}^1\text{t}^{-3}\right)\left(\text{m}^1\text{L}^{-3}\right)^a\left(\text{L}^1\text{t}^{-1}\right)^b\right\}$$

mass: $\{\text{m}^0\} = \{\text{m}^1\text{m}^a\}$ $\quad 0 = 1 + a \quad$ $a = -1$

time: $\{\text{t}^0\} = \{\text{t}^{-3}\text{t}^{-b}\}$ $\quad 0 = -3 - b \quad$ $b = -3$

length: $\{\text{L}^0\} = \{\text{L}^{-3a}\text{L}^b\}$ $\quad 0 = -3a + b$, $b = 3a \quad$ $b = -3$

Fortunately, the results for time and length agree. The dependent Π is thus

Π_1: $\Pi_1 = \dfrac{I}{\rho c^3}$

We form the second Π with sound pressure P,

$$\Pi_2 = P\rho^e c^f \qquad \{\Pi_2\} = \left\{\left(\text{m}^1\text{L}^{-1}\text{t}^{-2}\right)\left(\text{m}^1\text{L}^{-3}\right)^e\left(\text{L}^1\text{t}^{-1}\right)^f\right\}$$

mass: $\{\text{m}^0\} = \{\text{m}^1\text{m}^e\}$ $\quad 0 = 1 + e \quad$ $e = -1$

time: $\{\text{t}^0\} = \{\text{t}^{-2}\text{t}^{-f}\}$ $\quad 0 = -2 - f \quad$ $f = -2$

length: $\{\text{L}^0\} = \{\text{L}^{-1}\text{L}^{-3e}\text{L}^f\}$ $\quad 0 = -1 - 3e + f$, $f = 1 + 3e \quad$ $f = -2$

The second Π is thus

Π_2: $\Pi_2 = \dfrac{P}{\rho c^2}$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $\boxed{\dfrac{I}{\rho c^3} = f\left(\dfrac{P}{\rho c^2}\right)}$ (4)

(*b*) We try the force-based primary dimension system instead.

Step 1 There are four parameters in this problem; $n = 4$,

List of relevant parameters: $$I = f(P, c, \rho) \qquad n = 4 \qquad (5)$$

Step 2 The dimensions of *I* are those of power per area. The primary dimensions of each parameter are listed,

$$\begin{array}{cccc} I & P & c & \rho \\ \{F^1L^{-1}t^{-1}\} & \{F^1L^{-2}\} & \{L^1t^{-1}\} & \{F^1t^2L^{-4}\} \end{array}$$

Step 3 As a first guess, *j* is set equal to 3, the number of primary dimensions represented in the problem (F, L, and t). Again, however, the three independent parameters form a dimensionless group all by themselves. Thus we lower *j* by 1.

Reduction: $$j = 2$$

If this value of *j* is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 4 - 2 = 2$$

Step 4 We need to choose two repeating parameters since $j = 2$. We pick the same two parameters as in Part (*a*),

Repeating parameters: ρ and c

Step 5 The first Π is generated:

$$\Pi_1 = I \times \rho^b c^c \qquad \{\Pi_1\} = \left\{(F^1L^{-1}t^{-1})(F^1t^2L^{-4})^b (L^1t^{-1})^c\right\}$$

force: $\{F^0\} = \{F^1F^b\}$ $\quad 0 = 1 + b \quad$ $b = -1$

time: $\{t^0\} = \{t^{-1}t^{2b}t^{-c}\}$ $\quad 0 = -1 + 2b - c$, $c = -1 + 2b \quad$ $c = -3$

length: $\{L^0\} = \{L^{-1}L^{-4b}L^c\}$ $\quad 0 = -1 - 4b + c$, $c = 1 + 4b \quad$ $c = -3$

Again the two results for length and time agree. The dependent Π is thus

Π_1: $$\Pi_1 = \frac{I}{\rho c^3}$$

We form the second Π with sound pressure *P*,

$$\Pi_2 = P \times \rho^e c^f \qquad \{\Pi_2\} = \left\{(F^1L^{-2})(F^1t^2L^{-4})^e (L^1t^{-1})^f\right\}$$

force: $\{F^0\} = \{F^1F^e\}$ $\quad 0 = 1 + e \quad$ $e = -1$

time: $\{t^0\} = \{t^{-2e}t^{-f}\}$ $\quad 0 = -2e - f$, $f = -2e \quad$ $f = -2$

length: $\{L^0\} = \{L^{-2}L^{-4e}L^f\}$ $\quad 0 = -2 - 4e + f$ $\quad f = -2$

$f = 2 + 4e$

The second Π is thus

Π_2: $$\Pi_2 = \frac{P}{\rho c^2}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*: $$\boxed{\frac{I}{\rho c^3} = f\left(\frac{P}{\rho c^2}\right)} \qquad (6)$$

Discussion Equations 4 and 6 are the same. This exercise shows that you should get the same results using mass-based or force-based primary dimensions.

7-116

Solution We are to find the dimensionless relationship between the given parameters.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are now five parameters in this problem; $n = 5$,

List of relevant parameters: $$I = f\left(P, c, \rho, r\right) \qquad n = 5 \qquad (1)$$

Step 2 The dimensions of I are those of power per area. The primary dimensions of each parameter are listed,

I	P	c	ρ	r
$\{m^1t^{-3}\}$	$\{m^1L^{-1}t^{-2}\}$	$\{L^1t^{-1}\}$	$\{m^1L^{-3}\}$	$\{L^1\}$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 5 - 3 = 2$$

Step 4 We need to choose three repeating parameters since $j = 3$. We pick the three simplest independent parameters (r instead of P),

Repeating parameters: r, ρ, and c

Step 5 The first Π is generated:

$$\Pi_1 = I \times r^a \rho^b c^c \qquad \{\Pi_1\} = \left\{\left(m^1t^{-3}\right)\left(L^1\right)^a\left(m^1L^{-3}\right)^b\left(L^1t^{-1}\right)^c\right\}$$

mass: $\{m^0\} = \{m^1m^b\}$ $\qquad 0 = 1 + b \qquad b = -1$

time: $\{t^0\} = \{t^{-3}t^{-c}\}$ $\qquad 0 = -3 - c \qquad c = -3$

length: $\{L^0\} = \{L^aL^{-3b}L^c\}$ $\qquad 0 = a - 3b + c$, $a = 3b - c \qquad a = 0$

The first Π is thus

Π_1: $$\Pi_1 = \frac{I}{\rho c^3}$$

We form the second Π with sound pressure P,

$$\Pi_2 = P \times r^d \rho^e c^f \qquad \{\Pi_2\} = \left\{\left(m^1L^{-1}t^{-2}\right)\left(L^1\right)^d\left(m^1L^{-3}\right)^e\left(L^1t^{-1}\right)^f\right\}$$

mass: $\{m^0\} = \{m^1m^e\}$ $\qquad 0 = 1 + e \qquad e = -1$

time: $\{t^0\} = \{t^{-2}t^{-f}\}$ $\quad 0 = -2 - f \quad$ $f = -2$

length: $\{L^0\} = \{L^{-1}L^{d}L^{-3e}L^{f}\}$ $\quad$ $0 = -1 + d - 3e + f$, $d = 1 + 3e - f$ $\quad$ $d = 0$

The second Π is thus

Π_2:
$$\Pi_2 = \frac{P}{\rho c^2}$$

Step 6 We write the final functional relationship as

Relationship between Π*s*:
$$\boxed{\frac{I}{\rho c^3} = f\left(\frac{P}{\rho c^2}\right)} \tag{2}$$

Discussion This is an interesting case in which we added another independent parameter (r), yet this new parameter does not even appear in the final functional relationship! The list of independent parameters is thus *over specified*. (It turns out that P is a function of r, so r is not needed in the problem.) The result here is identical to the result of the previous problem. It turns out that the function in Eq. 2 is a constant times $\Pi_2^{\,2}$, which yields the correct analytical equation for I, namely

Analytical result:
$$I = \text{constant} \times \frac{P^2}{\rho c} \tag{3}$$

7-117

Solution We are to calculate the speed of a robotic tuna to match the Reynolds number of a real tuna.

Assumptions **1** The fluid in the model and prototype tests is the same (seawater) so that fluid properties such as density and viscosity do not change from model to prototype.

Analysis We match Reynolds number between the model and prototype,

Reynolds number matching:

$$\mathrm{Re}_\mathrm{m} = \frac{\rho_\mathrm{m} V_\mathrm{m} L_\mathrm{m}}{\mu_\mathrm{m}} = \mathrm{Re}_\mathrm{p} = \frac{\rho_\mathrm{p} V_\mathrm{p} L_\mathrm{p}}{\mu_\mathrm{p}}$$

from which we solve for the required speed of the model,

Required model speed:

$$V_\mathrm{m} = V_\mathrm{p}\left(\frac{\mu_\mathrm{m}}{\mu_\mathrm{p}}\right)\left(\frac{\rho_\mathrm{p}}{\rho_\mathrm{m}}\right)\left(\frac{L_\mathrm{p}}{L_\mathrm{m}}\right) = (10\ \mathrm{m/s})(1)(1)\left(\frac{2.0\ \mathrm{m}}{1.0\ \mathrm{m}}\right) = \mathbf{20.0\ m/s}$$

Discussion Note that both the increased size and increased speed influence the required speed of the model. We round the final answer to two significant digits in keeping with the precision of the given information.

7-118

Solution The pressure difference between the inside of a soap bubble and the outside air is to be analyzed with dimensional analysis and the method of repeating variables using the force-based system of primary dimensions.

Assumptions **1** The soap bubble is neutrally buoyant in the air, and gravity is not relevant. **2** No other variables or constants are important in this problem.

Analysis The step-by-step method of repeating variables is employed.

Step 1 There are three variables and constants in this problem; n = 3,

$$\Delta P = f\left(R, \sigma_s\right) \qquad n = 3 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed. The dimensions of pressure are force per area and those of surface tension are force per length.

ΔP	R	σ_s
$\{\mathrm{F}^1\mathrm{L}^{-2}\}$	$\{\mathrm{L}^1\}$	$\{\mathrm{F}^1\mathrm{L}^{-1}\}$

Step 3 As a first guess, j is set equal to 2, the number of primary dimensions represented in the problem (F and L).

Reduction: $$j = 2$$

If this value of j is correct, the expected number of Πs is

Number of expected Π*s*: $$k = n - j = 3 - 2 = 1$$

Step 4 We choose two repeating parameters since j = 2. Our only choice is R and σ_s since ΔP is the dependent variable.

Step 5 The dependent Π is generated:

$$\Pi_1 = \Delta P R^{a_1} \sigma_s^{b_1} \qquad \{\Pi_1\} = \{\mathrm{F}^0\mathrm{L}^0\} = \left\{\left(\mathrm{F}^1\mathrm{L}^{-2}\right)\mathrm{L}^{a_1}\left(\mathrm{F}^1\mathrm{L}^{-1}\right)^{b_1}\right\}$$

force: $\{\mathrm{F}^0\} = \{\mathrm{F}^1\mathrm{F}^{b_1}\}$ $\qquad 0 = 1 + b_1 \qquad b_1 = -1$

length: $\{\mathrm{L}^0\} = \{\mathrm{L}^{-2}\mathrm{L}^{a_1}\mathrm{L}^{-b_1}\}$ $\qquad 0 = -2 + a_1 - b_1$, $\quad a_1 = 2 + b_1 \qquad a_1 = 1$

Eq. 1 thus becomes

Π_1: $$\Pi_1 = \frac{\Delta P R}{\sigma_s} \qquad (2)$$

From Table 7-5, the established nondimensional parameter most similar to Eq. 2 is the **Weber number**, defined as a pressure times a length divided by surface tension. There is no need to further manipulate this Π.

Step 6 We now write the functional relationship between the nondimensional parameters. Since there is only one Π, it is a function of *nothing*, which means it must be a constant,

Relationship between Π*s*: $$\Pi_1 = \frac{\Delta P R}{\sigma_s} = f\left(\text{nothing}\right) = \text{constant} \quad \rightarrow \quad \boxed{\Delta P = \text{constant}\frac{\sigma_s}{R}} \qquad (3)$$

The result using force-based primary dimensions is indeed identical to the previous result using the mass-based system.

Discussion Because only two primary dimensions are represented in the problem when using the force-based system, the algebra is in fact a lot easier.

7-119

Solution We are to a third established nondimensional parameter that is formed by the product or ratio of two given established nondimensional parameters.

Analysis

(*a*) The product of Reynolds number and Prandtl number yields

Reynolds number times Prandtl number:
$$\mathrm{Re}\times\mathrm{Pr}=\frac{\rho LV}{\mu}\times\frac{c_P\mu}{k}=\frac{\rho LVc_P}{k} \qquad (1)$$

We recognize Eq. 1 as the *Peclet number*,

Peclet number:
$$\boxed{\mathrm{Pe}=\mathrm{Re}\times\mathrm{Pr}=\frac{\rho LVc_P}{k}=\frac{LV}{\alpha}} \qquad (2)$$

(*b*) The ratio of Schmidt number and Prandtl number yields

Schmidt number divided by Prandtl number:
$$\frac{\mathrm{Sc}}{\mathrm{Pr}}=\frac{\dfrac{\mu}{\rho D_{AB}}}{\dfrac{c_P\mu}{k}}=\frac{k}{\rho c_P D_{AB}} \qquad (3)$$

We recognize Eq. 3 as the *Lewis number*,

Lewis number:
$$\boxed{\mathrm{Le}=\frac{\mathrm{Sc}}{\mathrm{Pr}}=\frac{k}{\rho c_P D_{AB}}=\frac{\alpha}{D_{AB}}} \qquad (4)$$

(*c*) The product of Reynolds number and Schmidt number yields

Reynolds number times Schmidt number:
$$\mathrm{Re}\times\mathrm{Sc}=\frac{\rho LV}{\mu}\times\frac{\mu}{\rho D_{AB}}=\frac{LV}{D_{AB}} \qquad (5)$$

We recognize Eq. 5 as the *Sherwood number*,

Sherwood number:
$$\boxed{\mathrm{Sh}=\mathrm{Re}\times\mathrm{Sc}=\frac{LV}{D_{AB}}} \qquad (6)$$

Discussion Can you find any other such combinations from Table 7-5?

7-120

Solution We are to find the functional relationship between the given parameters, and then answer some questions about scaling.

Assumptions **1** The given parameters are the only relevant ones in the problem.

Analysis

(*a*) The step-by-step method of repeating variables is employed to obtain the nondimensional parameters (the Πs).

Step 1 There are four parameters in this problem; n = 4,

List of relevant parameters: $$\delta P = f\left(\rho, \dot{V}, D\right) \qquad n = 4 \qquad (1)$$

Step 2 The primary dimensions of each parameter are listed,

$$\begin{array}{cccc} \delta P & \rho & \dot{V} & D \\ \{\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-2}\} & \{\mathrm{m}^1\mathrm{L}^{-3}\} & \{\mathrm{L}^3\mathrm{t}^{-1}\} & \{\mathrm{L}^1\} \end{array}$$

Step 3 As a first guess, j is set equal to 3, the number of primary dimensions represented in the problem (m, L, and t).

Reduction: $$j = 3$$

If this value of j is correct, the expected number of Πs is

Number of expected Πs: $$k = n - j = 4 - 3 = 1$$

Step 4 We need to choose three repeating parameters since j = 3. We only have one choice in this problem, since there are only three independent parameters on the right-hand side of Eq. 1,

Repeating parameters: ρ, $\dot{V}$, and D

Step 5 The dependent Π is generated:

$$\Pi_1 = \delta P \rho^{a_1} \dot{V}^{b_1} D^{c_1} \qquad \{\Pi_1\} = \left\{\left(\mathrm{m}^1\mathrm{L}^{-1}\mathrm{t}^{-2}\right)\left(\mathrm{m}^1\mathrm{L}^{-3}\right)^{a_1}\left(\mathrm{L}^3\mathrm{t}^{-1}\right)^{b_1}\left(\mathrm{L}^1\right)^{c_1}\right\}$$

mass: $$\{\mathrm{m}^0\} = \{\mathrm{m}^{1+a_1}\} \qquad 0 = 1 + a_1 \qquad a_1 = -1$$

time: $$\{\mathrm{t}^0\} = \{\mathrm{t}^{-2-b_1}\} \qquad 0 = -2 - b_1 \qquad b_1 = -2$$

length: $$\{\mathrm{L}^0\} = \{\mathrm{L}^{-1}\mathrm{L}^{-3a_1}\mathrm{L}^{3b_1}\mathrm{L}^{c_1}\} \qquad 0 = -1 - 3a_1 + 3b_1 + c_1 \qquad c_1 = 4$$

The dependent Π is thus

Π_1: $$\Pi_1 = \frac{D^4 \delta P}{\rho \dot{V}^2}$$

Step 6 Since there is only one Π, it is a function of nothing. This is only possible if we set the Π equal to a constant. We write the final functional relationship as

Relationship between Πs: $$\boxed{\Pi_1 = \frac{D^4 \delta P}{\rho \dot{V}^2} = \text{constant}} \qquad (2)$$

(*b*) We re-write Eq. 2 as

Equation for δP: $$\delta P = \text{constant}\,\frac{\rho \dot{V}^2}{D^4} \qquad (3)$$

Thus, if we double the size of the cyclone, the pressure drop will *decrease* by a factor of $2^4 = 16$ **(Answer: The pressure drop will change by a factor of 1/16.)**.

(c) Also from Eq. 3 we see that if we double the volume flow rate, the pressure drop will increase by a factor of $2^2 = 4$ **(Answer: The pressure drop will change by a factor of 4.)**.

Discussion The pressure drop would be smallest for the *largest* cyclone operating at the *smallest* volume flow rate. (This agrees with our intuition.)

Fundamentals of Engineering (FE) Exam Problems

7-121

Which one is not a primary dimension?

(*a*) Velocity (*b*) Time (*c*) Electric current (*d*) Temperature (*e*) Mass

Answer (*a*) Velocity

7-122

The primary dimensions of kinematic viscosity is

(*a*) $m \cdot L/t^2$ (*b*) $m/L \cdot t$ (*c*) L^2/t (*d*) $L^2/m \cdot t$ (*e*) $L/m \cdot t^2$

Answer (*c*) L^2/t

7-123

Thermal conductivity of a substance may be defined as the rate of heat transfer per unit length per unit temperature difference. The primary dimensions of thermal conductivity is

(*a*) $m^2 \cdot L/t^2 \cdot T$ (*b*) $m^2 \cdot L^2/t \cdot T$ (*c*) $L^2/m \cdot t^2 \cdot T$ (*d*) $m \cdot L/t^3 \cdot T$ (*e*) $m \cdot L^2/t^3 \cdot T$

Answer (*d*) $m \cdot L/t^3 \cdot T$

7-124

The primary dimensions of the gas constant over the universal gas constant R/R_u is

(*a*) $L^2/t^2 \cdot T$ (*b*) $m \cdot L/N$ (*c*) $m/t \cdot N \cdot T$ (*d*) m/L^3 (*e*) N/m

Answer (*e*) N/m

7-125

The primary dimensions of the universal gas constant R_u is

(*a*) m·L/t^2·T (*b*) m^2·L/N (*c*) m·L^2/t^2·N·T (*d*) L^2/t^2·T (*e*) N/m·t

Answer (*c*) m·L^2/t^2·N·T

7-126

There are four additive terms in an equation, and their units are given below. Which one is not a unit of this equation?

(*a*) J (*b*) W/m (*c*) kg·m^2/s^2 (*d*) Pa·m^3 (*e*) N·m

Answer (*b*) W/m

7-127

Heat transfer coefficient is a function of a nondimensional parameter, which is functions of viscosity μ, specific heat c_p (kJ/kg·K), and thermal conductivity k (W/m·K). This nondimensional parameter is expressed as

(*a*) $c_p/\mu k$ (*b*) $k/\mu c_p$ (*c*) $\mu/c_p k$ (*d*) $\mu c_p/k$ (*e*) $c_p k/\mu$

Answer (*d*) $\mu c_p/k$

7-128

Nondimensional heat transfer coefficient is functions of functions of convection coefficient h (W/m^2·K) thermal conductivity k (W/m·K), and characteristic length L. This nondimensional parameter is expressed as

(*a*) hL/k (*b*) h/kL (*c*) L/hk (*d*) hk/L (*e*) kL/h

Answer (*a*) hL/k

7-129

The drag coefficient C_D is a nondimensional parameter and is functions of drag force F_D, density ρ, velocity V, and area A. The drag coefficient is expressed as

(a) $\dfrac{F_D V^2}{2\rho A}$ (b) $\dfrac{2F_D}{\rho VA}$ (c) $\dfrac{\rho VA^2}{F_D}$ (d) $\dfrac{F_D A}{\rho V}$ (e) $\dfrac{2F_D}{\rho V^2 A}$

Answer (e) $\dfrac{2F_D}{\rho V^2 A}$

7-130

Which similarity condition is related to force-scale equivalence?

(a) Geometric (b) Kinematic (c) Dynamic (d) Kinematic and dynamic

(e) Geometric and kinematic

Answer (c) Dynamic

7-131

A one-third scale model of a car is to be tested in a wind tunnel. The conditions of the actual car are V = 75 km/h and T = 0°C and the air temperature in the wind tunnel is 20°C. In order to achieve similarity between the model and the prototype, the wind tunnel velocity should be

The properties of air at 1 atm and 0°C: ρ = 1.292 kg/m^3, ν = 1.338×10^{-5} m^2/s

The properties of air at 1 atm and 20°C: ρ = 1.204 kg/m^3, ν = 1.516×10^{-5} m^2/s

(a) 255 km/h (b) 225 km/h (c) 147 km/h (d) 75 km/h (e) 25 km/h

Answer (a) 255 km/h

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
L_m=1 [m]
L_p=3 [m]
V_p=75 [km/h]*Convert(km/h, m/s)
T_air_p=0 [C]
T_air_m=20 [C]
rho_p=1.292 [kg/m^3]
nu_p=1.338E-5 [m^2/s]
rho_m=1.204 [kg/m^3]
nu_m=1.516E-5 [m^2/s]
Re_p=V_p*L_p/nu_p
Re_m=V_m*L_m/nu_m
Re_p=Re_m
V_m_km\h=V_m*Convert(m/s, km/h)
```

7-132

A one-fourth scale model of a car is to be tested in a wind tunnel. The conditions of the actual car are V = 45 km/h and T = 0°C and the air temperature in the wind tunnel is 20°C. In order to achieve similarity between the model and the prototype, the wind tunnel is run at 204 km/h. If the average drag force on the model is measured to be 70 N, the drag force on the prototype is

The properties of air at 1 atm and 0°C: ρ = 1.292 kg/m^3, ν = 1.338×10^{-5} m^2/s

The properties of air at 1 atm and 20°C: ρ = 1.204 kg/m^3, ν = 1.516×10^{-5} m^2/s

(*a*) 17.5 N (*b*) 58.5 N (*c*) 70 N (*d*) 93.2 N (*e*) 280 N

Answer (*b*) 58.5 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
L_m=1 [m]
L_p=4 [m]
V_p=45 [km/h]*Convert(km/h, m/s)
T_air_p=0 [C]
T_air_m=20 [C]
rho_p=1.292 [kg/m^3]
nu_p=1.338E-5 [m^2/s]
rho_m=1.204 [kg/m^3]
nu_m=1.516E-5 [m^2/s]
V_m=204 [km/h]*Convert(km/h, m/s)
F_D_m=70 [N]
PI_m=F_D_m/(rho_m*V_m^2*L_m^2)
PI_p=F_D_p/(rho_p*V_p^2*L_p^2)
PI_p=PI_m
```

7-133

A one-third scale model of an airplane is to be tested in water. The airplane has a velocity of 900 km/h in air at –50°C. The water temperature in the test section is 10°C. In order to achieve similarity between the model and the prototype, the water velocity on the model should be

The properties of air at 1 atm and –50°C: ρ = 1.582 kg/m^3, μ = 1.474×10^{-5} kg/m·s

The properties of water at 1 atm and 10°C: ρ = 999.7 kg/m^3, μ = 1.307×10^{-3} kg/m·s

(*a*) 97 km/h (*b*) 186 km/h (*c*) 263 km/h (*d*) 379 km/h (*e*) 450 km/h

Answer (*d*) 379 km/h

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
L_m=1 [m]
L_p=3 [m]
V_p=900 [km/h]*Convert(km/h, m/s)
T_p=-50 [C]
T_m=10 [C]
rho_p=1.582 [kg/m^3]
mu_p=1.474E-5 [kg/m-s]
rho_m=999.7 [kg/m^3]
mu_m=1.307E-3 [kg/m-s]
Re_p=rho_p*V_p*L_p/mu_p
Re_m=rho_m*V_m*L_m/mu_m
Re_p=Re_m
V_m_km\h=V_m*Convert(m/s, km/h)
```

7-134

A one-fourth scale model of an airplane is to be tested in water. The airplane has a velocity of 700 km/h in air at –50°C. The water temperature in the test section is 10°C. In order to achieve similarity between the model and the prototype, the test is done at a water velocity of 393 km/h. If the average drag force on the model is measured to be 13,800 N, the drag force on the prototype is

The properties of air at 1 atm and –50°C: ρ = 1.582 kg/m^3, μ = 1.474×10^{-5} kg/m·s

The properties of water at 1 atm and 10°C: ρ = 999.7 kg/m^3, μ = 1.307×10^{-3} kg/m·s

(*a*) 590 N (*b*) 862 N (*c*) 1109 N (*d*) 4655 N (*e*) 3450 N

Answer (*c*) 1109 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
L_m=1 [m]
L_p=4 [m]
V_p=700 [km/h]*Convert(km/h, m/s)
T_p=-50 [C]
T_m=10 [C]
rho_p=1.582 [kg/m^3]
mu_p=1.474E-5 [kg/m-s]
rho_m=999.7 [kg/m^3]
mu_m=1.307E-3 [kg/m-s]
V_m=393 [km/h]*Convert(km/h, m/s)
F_D_m=13800 [N]
PI_m=F_D_m/(rho_m*V_m^2*L_m^2)
PI_p=F_D_p/(rho_p*V_p^2*L_p^2)
PI_p=PI_m
```

7-135

Consider a boundary layer growing along a thin flat plate. This problem involves the following parameters: boundary layer thickness δ, downstream distance *x*, free-stream velocity *V*, fluid density ρ, and fluid viscosity μ. The number of expected nondimensional parameters Πs of this problem is

(*a*) 5 (*b*) 4 (*c*) 3 (*d*) 2 (*e*) 1

Answer (*d*) 2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n=5 "number of parameters"
j=3 "reduction: number of primary dimensions in the problem"
k=n-j
```

7-136

Consider unsteady fully developed Coutte flow-flow between two infinite parallel plates. This problem involves the following parameters: velocity component *u*, distance between the plates *h*, vertical distance *y*, top plate speed *V*, fluid density ρ, fluid viscosity μ, and time *t*. The number of expected nondimensional parameters Πs of this problem is

(*a*) 6 (*b*) 5 (*c*) 4 (*d*) 3 (*e*) 2

Answer (*c*) 4

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n=7 "number of parameters"
j=3 "reduction: number of primary dimensions in the problem"
k=n-j
```

7-137

Consider a boundary layer growing along a thin flat plate. This problem involves the following parameters: boundary layer thickness δ, downstream distance *x*, free-stream velocity *V*, fluid density ρ, and fluid viscosity μ. The number of primary dimensions represented in this problem is

(*a*) 1 (*b*) 2 (*c*) 3 (*d*) 4 (*e*) 5

Answer (*c*) 3

7-138

Consider a boundary layer growing along a thin flat plate. This problem involves the following parameters: boundary layer thickness δ, downstream distance *x*, free-stream velocity *V*, fluid density ρ, and fluid viscosity μ. The dependent parameter is δ. If we choose three repeating parameters as *x*, ρ, and *V*, the dependent Π is

(*a*) $\delta x^2/V$ (*b*) $\delta V^2/x\rho$ (*c*) $\delta\rho/xV$ (*d*) $x/\delta V$ (*e*) δ/x

Answer (*e*) δ/x

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

Third Edition

Yunus A. Çengel & John M. Cimbala

McGraw-Hill, 2013

Chapter 8
Internal Flow

Laminar and Turbulent Flow

8-1C

Solution We are to compare pipe flow in air and water.

Analysis Reynolds number is inversely proportional to kinematic viscosity, which is much smaller for water than for air (at 25°C, ν_{air} = 1.562×10^{-5} m^2/s and ν_{water} = μ/ρ = 0.891×10^{-3}/997 = 8.9×10^{-7} m^2/s). Therefore, for the same diameter and speed, the Reynolds number will be higher for water flow, and thus **the flow is more likely to be turbulent for water**.

Discussion The actual viscosity (dynamic viscosity) μ is larger for water than for air, but the density of water is so much greater than that of air that the kinematic viscosity of water ends up being smaller than that of air.

8-2C

Solution We are to compare the wall shear stress at the inlet and outlet of a pipe.

Analysis The wall shear stress τ_w is **highest at the tube inlet where the thickness of the boundary layer is nearly zero, and decreases gradually to the fully developed value**. The same is true for turbulent flow.

Discussion We are assuming that the entrance is well-rounded so that the inlet flow is nearly uniform.

8-3C

Solution We are to define and discuss hydraulic diameter.

Analysis For flow through non-circular tubes, the Reynolds number and the friction factor are based on the *hydraulic diameter* D_h defined as $\boxed{D_h = \frac{4A_c}{p}}$ where A_c is the cross-sectional area of the tube and p is its perimeter. The hydraulic diameter is defined such that it **reduces to ordinary diameter *D* for circular tubes** since $\boxed{D_h = \frac{4A_c}{p} = \frac{4\pi D^2/4}{\pi D} = D}$.

Discussion Hydraulic diameter is a useful tool for dealing with non-circular pipes (e.g., air conditioning and heating ducts in buildings).

8-4C

Solution We are to define and discuss hydrodynamic entry length.

Analysis The **region from the tube inlet to the point at which the boundary layer merges at the centerline** is called the *hydrodynamic entrance region*, and the length of this region is called *hydrodynamic entry length*. **The entry length is much longer in laminar flow than it is in turbulent flow**. But at very low Reynolds numbers, L_h is very small (e.g., $L_h = 1.2D$ at Re = 20).

Discussion The entry length increases with increasing Reynolds number, but there is a significant change in entry length when the flow changes from laminar to turbulent.

8-5C

Solution We are to discuss why pipes are usually circular in cross section.

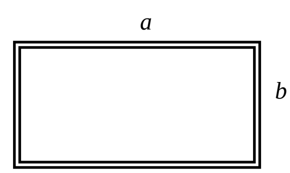
Analysis Liquids are usually transported in circular pipes because pipes with a circular cross section **can withstand large pressure differences between the inside and the outside without undergoing any significant distortion**.

Discussion Piping for gases at low pressure are often non-circular (e.g., air conditioning and heating ducts in buildings).

8-6C

Solution We are to define and discuss Reynolds number for pipe and duct flow.

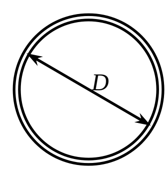
Analysis *Reynolds number* is the **ratio of the inertial forces to viscous forces**, and it serves as a criterion for determining the flow regime. At *large* Reynolds numbers, for example, the flow is turbulent since the inertia forces are large relative to the viscous forces, and thus the viscous forces cannot prevent the random and rapid fluctuations of the fluid. It is defined as follows:



(*a*) For flow in a circular tube of inner diameter *D*: $\boxed{\mathrm{Re} = \dfrac{VD}{\nu}}$

(*b*) For flow in a rectangular duct of cross-section $a \times b$: $\boxed{\mathrm{Re} = \dfrac{VD_h}{\nu}}$

where $D_h = \dfrac{4A_c}{p} = \dfrac{4ab}{2(a+b)} = \dfrac{2ab}{(a+b)}$ is the hydraulic diameter.



Discussion Since pipe flows become fully developed far enough downstream, diameter is the appropriate length scale for the Reynolds number. In boundary layer flows, however, the boundary layer grows continually downstream, and therefore downstream distance is a more appropriate length scale.

8-7C

Solution We are to compare the Reynolds number in air and water.

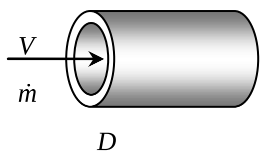
Analysis Reynolds number is inversely proportional to kinematic viscosity, which is much smaller for water than for air (at 25°C, ν_{air} = 1.562×10^{-5} m^2/s and ν_{water} = μ/ρ = 0.891×10^{-3}/997 = 8.9×10^{-7} m^2/s). Therefore, noting that Re = VD/ν, the **Reynolds number is higher for motion in water for the same diameter and speed**.

Discussion Of course, it is not possible to walk as fast in water as in air – try it!

8-8C

Solution We are to express the Reynolds number for a circular pipe in terms of mass flow rate.

Analysis Reynolds number for flow in a circular tube of diameter D is expressed as



$$\mathrm{Re} = \frac{VD}{\nu} \quad \text{where} \quad V = V_{avg} = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho\left(\pi D^2/4\right)} = \frac{4\dot{m}}{\rho \pi D^2} \quad \text{and} \quad \nu = \frac{\mu}{\rho}$$

Substituting,

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{4\dot{m}D}{\rho \pi D^2\left(\mu/\rho\right)} = \frac{4\dot{m}}{\pi D \mu}. \text{ Thus, } \boxed{\mathrm{Re} = \frac{4\dot{m}}{\pi D \mu}}$$

Discussion This result holds only for circular pipes.

8-9C

Solution We are to compare the pumping requirement for water and oil.

Analysis **Engine oil requires a larger pump** because of its much larger viscosity.

Discussion The density of oil is actually 10 to 15% smaller than that of water, and this makes the pumping requirement smaller for oil than water. However, the viscosity of oil is orders of magnitude larger than that of water, and is therefore the dominant factor in this comparison.

8-10C

Solution We are to discuss the Reynolds number for transition from laminar to turbulent flow.

Analysis The generally accepted value of the Reynolds number above which the flow in a smooth pipe is turbulent is **4000**. **In the range 2300 < Re < 4000, the flow is typically transitional between laminar and turbulent**.

Discussion In actual practice, pipe flow may become turbulent at Re lower or higher than this value.

8-11C

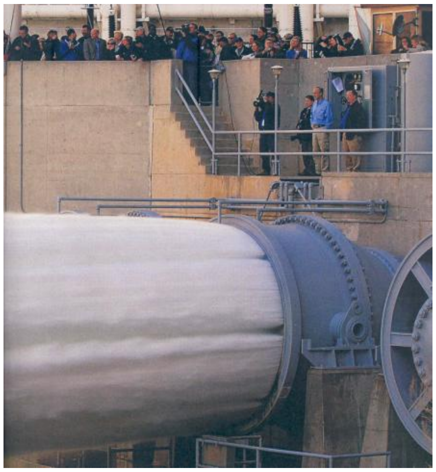
Solution We are to discuss the effect of surface roughness on pressure drop in pipe flow.

Analysis In turbulent flow, tubes with rough surfaces have much higher friction factors than the tubes with smooth surfaces, and thus **surface roughness leads to a much larger pressure drop in turbulent pipe flow**. In the case of laminar flow, the effect of surface roughness on the friction factor and pressure drop is negligible.

Discussion The effect of roughness on pressure drop is significant for turbulent flow, as seen in the Moody chart.

8-12E

Solution We are to estimate the Reynolds number for flow through a pipe, and determine if it is laminar or turbulent.



Assumptions **1** The water is at 20°C. **2** The discharge area is perfectly round (we ignore the rim effects – there appear to be some protrusions around the rim – three of them are visible in the picture).

Properties The density and viscosity of the water are ρ = 62.30 lbm/ft^3, and μ = 6.733 × 10^{-4} lbm/ft·s, respectively.

Analysis We use the people to estimate the diameter of the pipe. Assuming the guy in the blue shirt (who by the way is Secretary of the Interior Dirk Kempthorne) is six feet tall, the pipe diameter is about 13.8 ft. The average velocity is obtained from the given volume flow rate,

$$V_{\text{avg}} = \frac{\dot{V}}{A} = \frac{4\dot{V}}{\pi D^2}$$

and the Reynolds number is estimated as

$$\text{Re} = \frac{\rho D}{\mu} V_{\text{avg}} = \frac{\rho D}{\mu}\frac{4\dot{V}}{\pi D^2} = \frac{4\rho\dot{V}}{\pi\mu D}$$

$$= \frac{4\left(62.30 \text{ lbm/ft}^3\right)\left(300{,}000 \text{ gal/s}\right)}{\pi\left(6.733\times10^{-4} \text{ lbm/ft}\cdot\text{s}\right)\left(13.8 \text{ ft}\right)}\left(\frac{231 \text{ in}^3}{1 \text{ gal}}\right)\left(\frac{1 \text{ ft}}{12 \text{ in}}\right)^3 = 3.424\times10^8 \cong \mathbf{3.4\times10^8}$$

where we give our final result to two significant digits. Since Re > 2300, this flow is definitely **turbulent**.

Discussion There is absolutely no doubt that this flow is turbulent! You can even see the unsteady turbulent fluctuations in the photograph.

Fully Developed Flow in Pipes

8-13C

Solution We are to examine a claim about volume flow rate in laminar pipe flow.

Analysis **Yes**, the volume flow rate in a circular pipe with laminar flow can be determined by measuring the velocity at the centerline in the fully developed region, multiplying it by the cross-sectional area, and dividing the result by 2. This works for fully developed laminar pipe flow in round pipes since $\dot{V} = V_{\text{avg}} A_c = (V_{\max} / 2) A_c$.

Discussion This is *not* true for turbulent flow, so one must be careful that the flow is laminar before trusting this measurement. It is also *not* true if the pipe is not round, even if the flow is fully developed and laminar.

8-14C

Solution We are to examine a claim about volume flow rate in laminar pipe flow.

Analysis **No**, the average velocity in a circular pipe in fully developed laminar flow *cannot* be determined by simply measuring the velocity at $R/2$ (midway between the wall surface and the centerline). The average velocity is $V_{\max}/2$, but the velocity at $R/2$ is

$$V(R/2) = V_{\max}\left(1 - \frac{r^2}{R^2}\right)_{r=R/2} = \frac{3V_{\max}}{4},$$ which is much larger than $V_{\max}/2$.

Discussion There is, of course, a radial location in the pipe at which the local velocity *is* equal to the average velocity. Can you find that location?

8-15C

Solution We are to discuss the value of shear stress at the center of a pipe.

Analysis **The shear stress at the center of a circular tube during fully developed laminar flow is zero** since the shear stress is proportional to the velocity gradient, which is zero at the tube center.

Discussion This result is due to the axisymmetry of the velocity profile.

8-16C

Solution We are to discuss whether the maximum shear stress in a turbulent pipe flow occurs at the wall.

Analysis **Yes, the shear stress at the surface of a tube during fully developed turbulent flow is maximum** since the shear stress is proportional to the velocity gradient, which is maximum at the tube surface.

Discussion This result is also true for laminar flow.

8-17C

Solution We are to discuss how the wall shear stress varies along the flow direction in a pipe.

Analysis The wall shear stress τ_w **remains constant along the flow direction in the fully developed region** in both laminar and turbulent flow.

Discussion However, in the entrance region, τ_w starts out large, and decreases until the flow becomes fully developed.

8-18C

Solution We are to discuss the fluid property responsible for development of a velocity boundary layer.

Analysis The fluid **viscosity** is responsible for the development of the velocity boundary layer.

Discussion You can think of it this way: As the flow moves downstream, more and more of it gets slowed down near the wall due to friction, which is due to viscosity in the fluid.

8-19C

Solution We are to discuss the velocity profile in fully developed pipe flow.

Analysis In the fully developed region of flow in a circular pipe, the velocity profile **does *not* change** in the flow direction.

Discussion This is, in fact, the definition of fully developed – namely, the velocity profile remains of constant shape.

8-20C

Solution We are to discuss the relationship between friction factor and pressure loss in pipe flow.

Analysis The f**riction factor for flow in a tube is proportional to the pressure loss**. Since the pressure loss along the flow is directly related to the power requirements of the pump to maintain flow, the **friction factor is also proportional to the power requirements to overcome friction**. The applicable relations are

$$\boxed{\dot{W}_{\text{pump}} = \frac{\dot{m}\Delta P_L}{\rho}} \quad \text{and} \quad \boxed{\dot{W}_{\text{pump}} = \frac{\dot{m}\Delta P_L}{\rho}}$$

Discussion This type of pressure loss due to friction is an *irreversible* loss. Hence, it is always positive (positive being defined as a pressure drop down the pipe). A negative pressure loss would violate the second law of thermodynamics.

8-21C

Solution We are to discuss whether fully developed pipe flow is one-, two-, or three-dimensional.

Analysis The geometry is axisymmetric, which is two-dimensional. However, since the velocity profile does not change down the pipe axis, *u* is a function only of *r*, and thus **the velocity is one-dimensional with respect to radial coordinate *r***. Pressure, on the other hand, varies only with axial location *x* in fully developed pipe flow (ignoring the hydrostatic pressure component, which acts independently of the flow component). So, **the pressure is one-dimensional with respect to axial coordinate *x***.

Discussion In the developing portion of the flow, *u* varies with *x* as well as with *r*, and thus the flow is two-dimensional in the developing region.

8-22C

Solution We are to discuss the change in head loss when the pipe length is doubled.

Analysis In fully developed flow in a circular pipe with negligible entrance effects, if the length of the pipe is doubled, **the head loss also doubles** (the head loss is proportional to pipe length in the fully developed region of flow).

Discussion If entrance lengths are not negligible, the head loss in the longer pipe would be less than twice that of the shorter pipe, since the shear stress is larger in the entrance region than in the fully developed region.

8-23C

Solution We are to compare the head loss when the pipe diameter is halved.

Analysis In fully developed laminar flow in a circular pipe, the head loss is given by

$$h_L = f\frac{L}{D}\frac{V^2}{2g} = \frac{64}{\mathrm{Re}}\frac{L}{D}\frac{V^2}{2g} = \frac{64}{V\,D/\nu}\frac{L}{D}\frac{V^2}{2g} = \frac{64\nu}{D}\frac{L}{D}\frac{V}{2g}$$

The average velocity can be expressed in terms of the flow rate as $V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4}$. Substituting,

$$h_L = \frac{64\nu}{D^2}\frac{L}{2g}\left(\frac{\dot{V}}{\pi D^2/4}\right) = \frac{64\nu}{D^2}\frac{4L\dot{V}}{2g\pi D^2} = \frac{128\nu L\dot{V}}{g\pi D^4}$$

Therefore, at constant flow rate and pipe length, the head loss is inversely proportional to the 4$^{\text{th}}$ power of diameter, and thus **reducing the pipe diameter by half increases the head loss by a factor of 16**.

Discussion This is a very significant increase in head loss, and shows why larger diameter tubes lead to much smaller pumping power requirements.

8-24C

Solution We are to explain why friction factor is independent of Re at very large Re.

Analysis At very large Reynolds numbers, the flow is fully rough and the friction factor is independent of the Reynolds number. This is because **the thickness of viscous sublayer decreases with increasing Reynolds number, and it be comes so thin that the surface roughness protrudes into the flow**. The viscous effects in this case are produced in the main flow primarily by the protruding roughness elements, and the contribution of the viscous sublayer is negligible.

Discussion This effect is clearly seen in the Moody chart – at large Re, the curves flatten out horizontally.

8-25C

Solution We are to define and discuss turbulent viscosity.

Analysis Turbulent viscosity μ_t is caused by turbulent eddies, and it **accounts for momentum transport by turbulent eddies**. It is expressed as $\boxed{\tau_t = -\rho\overline{u'v'} = \mu_t\frac{\partial\bar{u}}{\partial y}}$ where $\bar{u}$ is the mean value of velocity in the flow direction and u' and u' are the fluctuating components of velocity.

Discussion Turbulent viscosity is a derived, or non-physical quantity. Unlike the viscosity, it is *not* a property of the fluid; rather, it is a property of the *flow*.

8-26C

Solution We are to discuss the dimensions of a constant in a head loss expression.

Analysis We compare the dimensions of the two sides of the equation $h_L = 0.0826\, fL\dfrac{\dot{V}^2}{D^5}$. Using curly brackets to mean "the dimensions of", we have $\{L\} = \{0.0826\}\cdot\{1\}\{L\}\cdot\{L^3t^{-1}\}^2\cdot\{L^{-5}\}$, and the dimensions of the constant are thus $\{0.0826\} = \{L^{-1}t^2\}$. Therefore, **the constant 0.0826 is *not* dimensionless**. **This is *not* a dimensionally homogeneous equation**, and it cannot be used in any consistent set of units.

Discussion Engineers often create dimensionally inhomogeneous equations like this. While they are useful for practicing engineers, they are valid only when the proper units are used for each variable, and this can occasionally lead to mistakes. For this reason, the present authors do not encourage their use.

8-27C

Solution We are to discuss the change in head loss due to a decrease in viscosity by a factor of two.

Analysis In fully developed laminar flow in a circular pipe, the pressure loss and the head loss are given by

$$\Delta P_L = \frac{32\mu LV}{D^2} \quad \text{and} \quad h_L = \frac{\Delta P_L}{\rho g} = \frac{32\mu LV}{\rho g D^2}$$

When the flow rate and thus the average velocity are held constant, the head loss becomes proportional to viscosity. Therefore, **the head loss is reduced by half** when the viscosity of the fluid is reduced by half.

Discussion This result is *not* valid for turbulent flow – only for laminar flow. It is also not valid for laminar flow in situations where the entrance length effects are not negligible.

8-28C

Solution We are to discuss the relationship between head loss and pressure drop in pipe flow.

Analysis The head loss is related to pressure loss by $\boxed{h_L = \Delta P_L / \rho g}$. For a given fluid, the head loss can be converted to pressure loss by multiplying the head loss by the acceleration of gravity and the density of the fluid. Thus, for constant density, **head loss and pressure drop are linearly proportional to each other**.

Discussion This result is true for both laminar and turbulent pipe flow.

8-29C

Solution We are to discuss if the friction factor is zero for laminar pipe flow with a perfectly smooth surface.

Analysis During laminar flow of air in a circular pipe with perfectly smooth surfaces, **the friction factor is *not* zero** because of the no-slip boundary condition, which must hold even for perfectly smooth surfaces.

Discussion If we compare the friction factor for rough and smooth surfaces, roughness has no effect on friction factor for fully developed laminar pipe flow unless the roughness height is very large. For turbulent pipe flow, however, roughness very strongly impacts the friction factor.

8-30C

Solution We are to discuss why the friction factor is higher in turbulent pipe flow compared to laminar pipe flow.

Analysis In turbulent flow, it is the **turbulent eddies due to enhanced mixing** that cause the friction factor to be larger. This turbulent mixing leads to a much larger wall shear stress, which translates into larger friction factor.

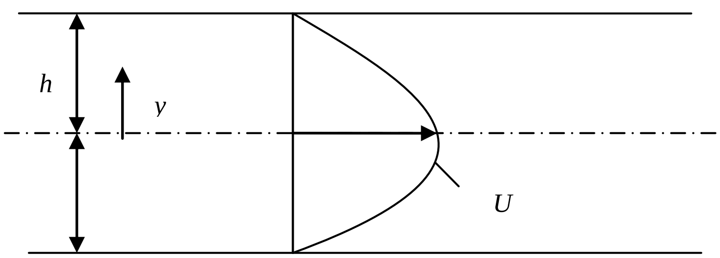
Discussion Another way to think of it is that the turbulent eddies cause the turbulent velocity profile to be much *fuller* (closer to uniform flow) than the laminar velocity profile.

8-31

Solution The velocity profile for the flow of a fluid between two large parallel plates is given. A relation for the flow rate through the plates is to be determined.

Assumptions **1** The flow is steady and incompressible.

Analysis



$$\dot{V} = \int U(y)dA = \int_{-h}^{+h} U(y)bdy = 2\int_0^h U(y)bdy = 2\frac{3Ub}{2}\int_0^h \left[1-\left(\frac{y}{h}\right)^2\right]dy = 3Ubh\int_0^1 \left[1-\left(\frac{y}{h}\right)^2\right]d\left(\frac{y}{h}\right)$$

$$\dot{V} = 3Ubh\left[\frac{y}{h}-\left(\frac{y}{h}\right)^3\frac{1}{3}\right]\Bigg|_{0=y/h}^{1=y/h}$$

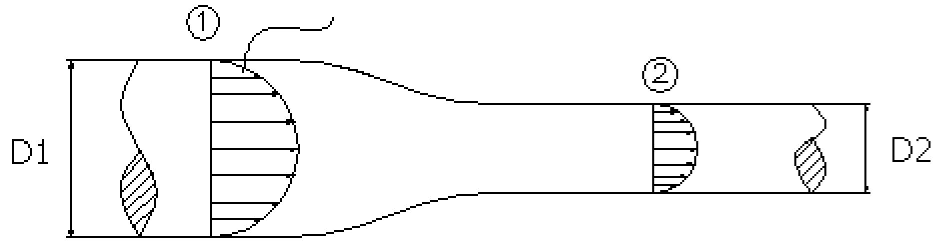
$$\dot{V} = Ubh\left[\left(1-\frac{1}{3}\right)-0\right] = \frac{2}{3}3Ubh = 2Ubh$$

8-32

Solution Water flows in a reducing pipe section. The flow upstream is laminar and the flow downstream is turbulent. The ratio of centerline velocities is to be determined.

Assumptions **1** The flow is steady and incompressible.

Analysis



$$\dot{V}_1 = \dot{V}_2 \quad , \quad \int_1 u_1 dA_1 = \int u_2 dA_2$$

$$\int_0^{R_1} u_1 (1 - \frac{r_1^{\,2}}{R_1^{\,2}}) 2\pi r_1 dr_1 = \int_0^{R_2} u_2 (1 - \frac{r_2}{R_2})^{1/7} 2\pi r_2 dr_2$$

$$u_1 \int_0^1 \left[1 - (\frac{r_1}{R_1})^2\right]\left(\frac{r_1}{R_1}\right) d\left(\frac{r_1}{R_1}\right) = u_2 \int \left(1 - \frac{r_2}{R_2}\right)^{1/7} \frac{r_2}{R_2} d\left(\frac{r_2}{R_2}\right)$$

$$\frac{r_1}{R_2} \longrightarrow x \qquad , \qquad \frac{r_2}{R_2} \longrightarrow y$$

$$u_u \int \left(x - x^3\right) dx = u_2 \int (1 - y)^{1/7} y\, dy$$

$$\dot{V}_1 = \int_0^{R_1} u_1 \left(1 - \frac{r^2}{R_1^{\,2}}\right) 2\pi\, r\, dr = \pi\, R_1^{\,2} \frac{u_1}{2}$$

$$\dot{V}_1 = \int_0^{R_2} u_2 \left(1 - \frac{r}{R_2}\right)^{1/7} 2\,\pi\, r\, dr = u_2 \frac{49\pi}{60} R_2^{\,2}$$

$$\dot{V}_1 = \dot{V}_2$$

$$\pi\, R_1^{\,2} \frac{u_1}{2} = u_2 \frac{49\pi}{60} R_2$$

$$\frac{u_1}{u_2} = \left(\frac{R_2}{R_1}\right)^2 . \frac{49}{30} \Rightarrow$$

$$\frac{u_1}{u_2} = \left(\frac{4}{7}\right)^2 . \frac{49}{30} = \frac{16}{30} = \frac{8}{15}$$

8-33

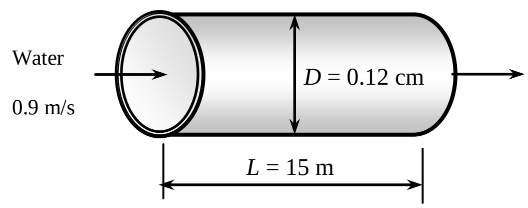
Solution The average flow velocity in a pipe is given. The pressure drop, the head loss, and the pumping power are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water are given to be ρ = 999.7 kg/m^3 and μ = 1.307×10^{-3} kg/m·s, respectively.

Analysis (*a*) First we need to determine the flow regime. The Reynolds number of the flow is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(999.7\ \text{kg/m}^3)(0.9\ \text{m/s})(1.2\times10^{-3}\ \text{m})}{1.307\times10^{-3}\ \text{kg/m}\cdot\text{s}} = 826.1$$



which is less than 2300. Therefore, the flow is laminar. Then the friction factor and the pressure drop become

$$f = \frac{64}{\text{Re}} = \frac{64}{826.1} = 0.07748$$

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.07748\frac{15\ \text{m}}{0.0012\ \text{m}}\frac{(999.7\ \text{kg/m}^3)(0.9\ \text{m/s})^2}{2}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) = \mathbf{392\ kPa}$$

(*b*) The head loss in the pipe is determined from

$$h_L = \frac{\Delta P_L}{\rho g} = f\frac{L}{D}\frac{V^2}{2g} = 0.07748\frac{15\ \text{m}}{0.0012\ \text{m}}\frac{(0.9\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = \mathbf{40.0\ m}$$

(*c*) The volume flow rate and the pumping power requirements are

$$\dot{V} = VA_c = V(\pi D^2/4) = (0.9\ \text{m/s})\pi(0.0012\ \text{m})^2/4 = 1.018\times10^{-6}\ \text{m}^3/\text{s}$$

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (1.018\times10^{-6}\ \text{m}^3/\text{s})(392\ \text{kPa})\left(\frac{1000\ \text{W}}{1\ \text{kPa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{0.399\ W}$$

Therefore, power input in the amount of 0.399 W is needed to overcome the frictional losses in the flow due to viscosity.

Discussion If the flow were instead *turbulent*, the pumping power would be much greater since the head loss in the pipe would be much greater.

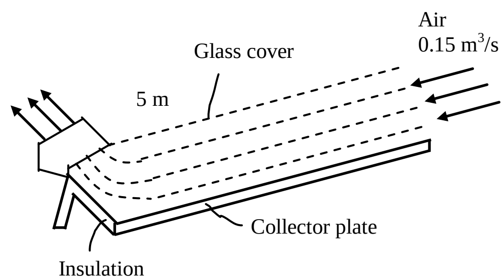
8-34

Solution Air enters the constant spacing between the glass cover and the plate of a solar collector. The pressure drop of air in the collector is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The roughness effects are negligible, and thus the inner surfaces are considered to be smooth, $\varepsilon \approx 0$. **4** Air is an ideal gas. **5** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and 45° are ρ = 1.109 kg/m^3, μ = 1.941×10^{-5} kg/m·s, and ν = 1.750×10^{-5} m^2/s.

Analysis Mass flow rate, cross-sectional area, hydraulic diameter, average velocity, and the Reynolds number are



$$\dot{m} = \rho \dot{V} = (1.11\ \text{kg/m}^3)(0.15\ \text{m}^3/\text{s}) = 0.1665\ \text{kg/s}$$

$$A_c = a \times b = (1\ \text{m})(0.03\ \text{m}) = 0.03\ \text{m}^2$$

$$D_h = \frac{4A_c}{p} = \frac{4(0.03\ \text{m}^2)}{2(1+0.03)\ \text{m}} = 0.05825\ \text{m}$$

$$V = \frac{\dot{V}}{A_c} = \frac{0.15\ \text{m}^3/s}{0.03\ \text{m}^2} = 5\ \text{m}/s$$

$$\text{Re} = \frac{VD_h}{\upsilon} = \frac{(5\ \text{m/s})(0.05825\ \text{m})}{1.750\times 10^{-5}\ \text{m}^2/\text{s}} = 1.664\times 10^4$$

Since Re is greater than 4000, the flow is turbulent. The friction factor corresponding to this Reynolds number for a smooth flow section (ε/D = 0) can be obtained from the Moody chart. But to avoid reading error, we use the Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(0 + \frac{2.51}{16{,}640\sqrt{f}}\right)$$

which gives f = 0.0271. Then the pressure drop becomes

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.0271\frac{5\ \text{m}}{0.05825\ \text{m}}\frac{(1.11\ \text{kg/m}^3)(5\ \text{m/s})^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{Pa}}{1\ \text{N/m}^2}\right) = \mathbf{32.3\ Pa}$$

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.0270, which is sufficiently close to 0.0271.

8-35E

Solution The flow rate and the head loss in an air duct is given. The minimum diameter of the duct is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The duct involves no components such as bends, valves, and connectors. **4** Air is an ideal gas. **5** The duct is smooth since it is made of plastic, $\varepsilon \approx 0$. **6** The flow is turbulent (to be verified).

Properties The density, dynamic viscosity, and kinematic viscosity of air at 100°F are ρ = 0.07088 lbm/ft^3, μ = 0.04615 lbm/ft·h, and ν = 0.6512 ft^2/s = 1.809×10^{-4} ft^2/s.

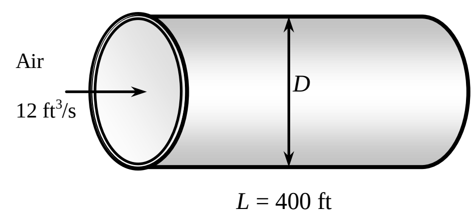
Analysis The average velocity, Reynolds number, friction factor, and the head loss relations can be expressed as (D is in ft, V is in ft/s, Re and f are dimensionless)

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{12\ \text{ft}^3/s}{\pi D^2/4}$$

$$\text{Re} = \frac{VD}{\nu} = \frac{VD}{1.809\times 10^{-4}\ \text{ft}^2/s}$$

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) = -2.0\log\left(\frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$h_L = f\frac{L}{D}\frac{V^2}{2g} \quad\rightarrow\quad 50 = f\frac{L}{D}\frac{V^2}{2g} = f\frac{400\ \text{ft}}{D}\frac{V^2}{2(32.2\ \text{ft/s}^2)}$$



This is a set of 4 equations in 4 unknowns, and solving them with an equation solver gives

D = **0.88 ft**, f = 0.0181, V = 19.8 ft/s, and Re = 96,040

Therefore, the diameter of the duct should be more than 0.88 ft if the head loss is not to exceed 50 ft. Note that Re > 4000, and thus the turbulent flow assumption is verified.

The diameter can also be determined directly from the third Swamee-Jain formula to be

$$D = 0.66\left[\varepsilon^{1.25}\left(\frac{L\dot{V}^2}{gh_L}\right)^{4.75} + \nu\dot{V}^{9.4}\left(\frac{L}{gh_L}\right)^{5.2}\right]^{0.04}$$

$$= 0.66\left[0 + (0.180\times 10^{-3}\ \text{ft}^2/s)(12\ \text{ft}^3/s)^{9.4}\left(\frac{400\ \text{ft}}{(32.2\ \text{ft/s}^2)(50\ \text{ft})}\right)^{5.2}\right]^{0.04}$$

$$= 0.89\ \text{ft}$$

Discussion Note that the difference between the two results is less than 2%. Therefore, the simple Swamee-Jain relation can be used with confidence.

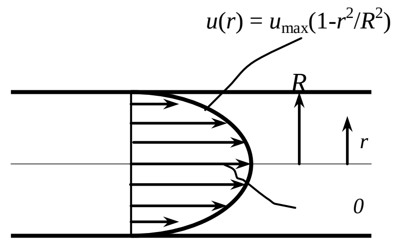
8-36

Solution In fully developed laminar flow in a circular pipe, the velocity at $r = R/2$ is measured. The velocity at the center of the pipe ($r = 0$) is to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is given by

$$u(r) = u_{\max}\left(1 - \frac{r^2}{R^2}\right)$$



where u_{max} is the maximum velocity which occurs at pipe center, $r = 0$. At $r = R/2$,

$$u(R/2) = u_{\max}\left(1 - \frac{(R/2)^2}{R^2}\right) = u_{\max}\left(1 - \frac{1}{4}\right) = \frac{3u_{\max}}{4}$$

Solving for u_{max} and substituting,

$$u_{\max} = \frac{4u(R/2)}{3} = \frac{4(11\text{ m/s})}{3} = \mathbf{14.7\ m/s}$$

which is the velocity at the pipe center.

Discussion The relationship used here is valid only for fully developed laminar flow. The result would be much different if the flow were turbulent.

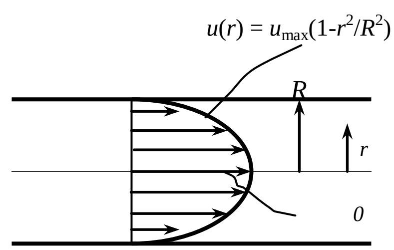
8-37

Solution The velocity profile in fully developed laminar flow in a circular pipe is given. The average and maximum velocities as well as the flow rate are to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is given by

$$u(r) = u_{\max}\left(1 - \frac{r^2}{R^2}\right)$$



The velocity profile in this case is given by

$$u(r) = 4(1 - r^2/R^2)$$

Comparing the two relations above gives the maximum velocity to be $\boldsymbol{u_{max}}$ **= 4.00 m/s**. Then the average velocity and volume flow rate become

$$V_{avg} = \frac{u_{max}}{2} = \frac{4\text{ m/s}}{2} = \mathbf{2.00\ m/s}$$

$$\dot{V} = V_{avg}A_c = V_{avg}(\pi R^2) = (2\text{ m/s})[\pi(0.02\text{ m})^2] = \mathbf{0.00251\ m^3/s}$$

Discussion A unique feature of fully developed laminar pipe flow is that the maximum velocity is exactly twice the average velocity. This is *not* the case for turbulent pipe flow, since the velocity profile is much fuller.

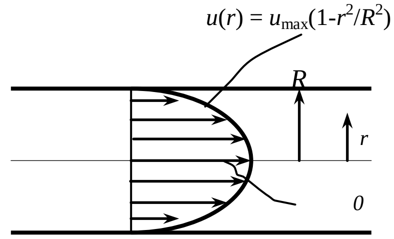
8-38

Solution The velocity profile in fully developed laminar flow in a circular pipe is given. The average and maximum velocities as well as the flow rate are to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is given by

$$u(r) = u_{\max}\left(1 - \frac{r^2}{R^2}\right)$$



The velocity profile in this case is given by

$$u(r) = 4(1 - r^2/R^2)$$

Comparing the two relations above gives the maximum velocity to be ***u*$_{\max}$ = 4.00 m/s**. Then the average velocity and volume flow rate become

$$V_{avg} = \frac{u_{max}}{2} = \frac{4 \text{ m/s}}{2} = \mathbf{2.00\ m/s}$$

$$\dot{V} = V_{avg}A_c = V_{avg}(\pi R^2) = (2 \text{ m/s})[\pi(0.07 \text{ m})^2] = \mathbf{0.0308\ m^3/s}$$

Discussion Compared to the previous problem, the average velocity remains the same since the maximum velocity (at the centerline) remains the same, but the volume flow rate increases as the diameter increases.

8-39

Solution The flow rate through a specified water pipe is given. The pressure drop, the head loss, and the pumping power requirements are to be determined.

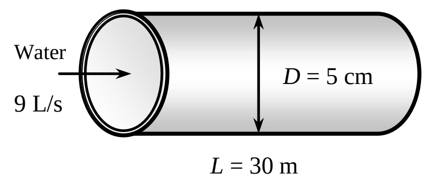
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water are given to be ρ = 999.1 kg/m^3 and μ = 1.138×10^{-3} kg/m·s, respectively. The roughness of stainless steel is 0.002 mm.

Analysis First we calculate the average velocity and the Reynolds number to determine the flow regime:

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.009\,\text{m}^3/s}{\pi(0.05\,\text{m})^2/4} = 4.584\,\text{m}/s$$

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(999.1\,\text{kg/m}^3)(4.584\,\text{m/s})(0.05\,\text{m})}{1.138\times10^{-3}\,\text{kg/m}\cdot\text{s}} = 2.012\times10^5$$



which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{2\times10^{-6}\,\text{m}}{0.05\,\text{m}} = 4\times10^{-5}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{4\times10^{-5}}{3.7} + \frac{2.51}{2.012\times10^5\sqrt{f}}\right)$$

It gives f = 0.01594. Then the pressure drop, head loss, and the required power input become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.01594\frac{30\,\text{m}}{0.05\,\text{m}}\frac{(999.1\,\text{kg/m}^3)(4.584\,\text{m/s})^2}{2}\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1\,\text{kN/m}^2}\right) = 100.4\,\text{kPa} \cong \mathbf{100\,kPa}$$

$$h_L = \frac{\Delta P_L}{\rho g} = f\frac{L}{D}\frac{V^2}{2g} = 0.01594\frac{30\,\text{m}}{0.05\,\text{m}}\frac{(4.584\,\text{m/s})^2}{2(9.81\,\text{m/s}^2)} = \mathbf{10.2\,m}$$

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (0.009\,\text{m}^3/s)(100.4\,\text{kPa})\left(\frac{1\,\text{kW}}{1\,\text{kPa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{0.904\,kW}$$

Therefore, useful power input in the amount of 0.904 kW is needed to overcome the frictional losses in the pipe.

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.01574, which is sufficiently close to 0.01594. Also, the friction factor corresponding to ε = 0 in this case is 0.01562, which indicates that stainless steel pipes in this case can be assumed to be smooth with an error of about 2%. Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

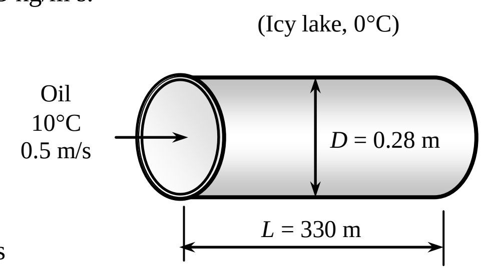
8-40

Solution Oil flows through a pipeline that passes through icy waters of a lake. The pumping power needed to overcome pressure losses is to be determined.

Assumptions The flow is steady and incompressible. **2** The flow section considered is away from the entrance, and thus the flow is fully developed. **3** The roughness effects are negligible, and thus the inner surfaces are considered to be smooth, $\varepsilon \approx 0$.

Properties The properties of oil are given to be ρ = 894 kg/m^3 and μ = 2.33 kg/m·s.

Analysis The volume flow rate and the Reynolds number in this case are



$$\dot{V} = VA_c = V\frac{\pi D^2}{4} = (0.5\text{ m/s})\frac{\pi(0.28\text{ m})^2}{4} = 0.03079\text{ m}^3/\text{s}$$

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(894\text{ kg/m}^3)(0.5\text{ m/s})(0.28\text{ m})}{2.33\text{ kg/m}\cdot\text{s}} = 53.72$$

which is less than 2300. Therefore, the flow is laminar, and the friction factor is

$$f = \frac{64}{\text{Re}} = \frac{64}{53.72} = 1.191$$

Then the pressure drop in the pipe and the required pumping power become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 1.191\frac{330\text{ m}}{0.28\text{ m}}\frac{(894\text{ kg/m}^3)(0.5\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = 156.9\text{ kPa}$$

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (0.03079\text{ m}^3/\text{s})(156.9\text{ kPa})\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{4.83\text{ kW}}$$

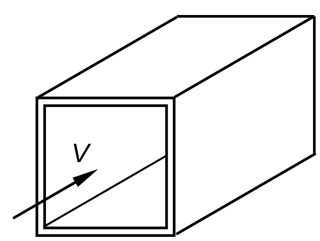
Discussion The power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be much more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

8-41

Solution Laminar flow through a square channel is considered. The change in the head loss is to be determined when the average velocity is doubled.

Assumptions **1** The flow remains laminar at all times. **2** The entrance effects are negligible, and thus the flow is fully developed.

Analysis The friction factor for fully developed laminar flow in a square channel is



$$f = \frac{56.92}{\text{Re}} \quad \text{where} \quad \text{Re} = \frac{\rho VD}{\mu}$$

Then the head loss for laminar flow can be expressed as

$$h_{L,1} = f\frac{L}{D}\frac{V^2}{2g} = \frac{56.92}{\text{Re}}\frac{L}{D}\frac{V^2}{2g} = \frac{56.92\mu}{\rho VD}\frac{L}{D}\frac{V^2}{2g} = 28.46V\frac{\mu L}{\rho g D^2}$$

which shows that the head loss is proportional to the average velocity. Therefore, **the head loss doubles when the average velocity is doubled**. This can also be shown as

$$h_{L,2} = 28.46V_2\frac{\mu L}{\rho g D^2} = 28.46(2V)\frac{\mu L}{\rho g D^2} = 2\left(28.46V\frac{\mu L}{\rho g D^2}\right) = 2h_{L,1}$$

Discussion The conclusion above is also valid for laminar flow in channels of different cross-sections.

8-42

Solution Turbulent flow through a smooth pipe is considered. The change in the head loss is to be determined when the average velocity is doubled.

Assumptions **1** The flow remains turbulent at all times. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The inner surface of the pipe is smooth.

Analysis The friction factor for the turbulent flow in smooth pipes is given as

$$f = 0.184\,\mathrm{Re}^{-0.2} \quad \text{where} \quad \mathrm{Re} = \frac{\rho VD}{\mu}$$

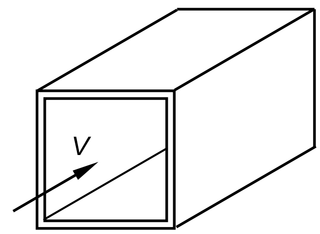
Then the head loss of the fluid for turbulent flow can be expressed as

$$h_{L,1} = f\frac{L}{D}\frac{V^2}{2g} = 0.184\,\mathrm{Re}^{-0.2}\frac{L}{D}\frac{V^2}{2g} = 0.184\left(\frac{\rho VD}{\mu}\right)^{-0.2}\frac{L}{D}\frac{V^2}{2g} = 0.184\left(\frac{\rho D}{\mu}\right)^{-0.2}\frac{L}{D}\frac{V^{1.8}}{2g}$$

which shows that the head loss is proportional to the 1.8$^{\text{th}}$ power of the average velocity. Therefore, the head loss increases by a factor of $2^{1.8} = 3.48$ when the average velocity is doubled. This can also be shown as

$$h_{L,2} = 0.184\left(\frac{\rho D}{\mu}\right)^{-0.2}\frac{L}{D}\frac{V_2^{1.8}}{2g} = 0.184\left(\frac{\rho D}{\mu}\right)^{-0.2}\frac{L}{D}\frac{(2V)^{1.8}}{2g}$$

$$= 2^{1.8}\left[0.184\left(\frac{\rho D}{\mu}\right)^{-0.2}\frac{L}{D}\frac{V^{1.8}}{2g}\right] = 2^{1.8}h_{L,1} = 3.48h_{L,1}$$



For fully rough flow in a rough pipe, the friction factor is independent of the Reynolds number and thus the flow velocity. Therefore, **the head loss increases by a factor of 4** in this case since

$$h_{L,1} = f\frac{L}{D}\frac{V^2}{2g}$$

and thus the **head loss is proportional to the square of the average velocity** when *f*, *L*, and *D* are constant.

Discussion Most flows in practice are in the fully rough regime, and thus the head loss is generally assumed to be proportional to the square of the average velocity for all kinds of turbulent flow. Note that we use diameter *D* here in place of hydraulic diameter D_h. For a square duct, it turns out that $D_h = D$, so this is a valid approximation.

8-43

Solution Air enters a rectangular duct. The fan power needed to overcome the pressure losses is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** Air is an ideal gas. **4** The duct involves no components such as bends, valves, and connectors. **5** The flow section involves no work devices such as fans or turbines

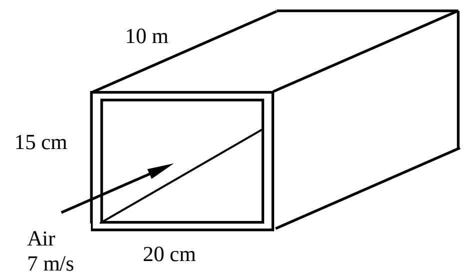
Properties The properties of air at 1 atm and 35°C are ρ = 1.145 kg/m^3, μ = 1.895×10^{-5} kg/m·s, and ν = 1.655×10^{-5} m^2/s. The roughness of commercial steel surfaces is ε = 0.000045 m (Table 8-2).

Analysis The hydraulic diameter, the volume flow rate, and the Reynolds number in this case are

$$D_h = \frac{4A_c}{p} = \frac{4ab}{2(a+b)} = \frac{4(0.15\text{ m})(0.20\text{ m})}{2(0.15+0.20)\text{ m}} = 0.17143\text{ m}$$

$$\dot{V} = VA_c = V(a\times b) = (7\text{ m/s})(0.15\times 0.20\text{ m}^2) = 0.21\text{ m}^3\text{/s}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{(1.145\text{ kg/m}^3)(7\text{ m/s})(0.17143\text{ m})}{1.895\times 10^{-5}\text{ kg/m}\cdot\text{s}} = 72{,}506$$



which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon / D_h = \frac{4.5\times 10^{-5}\text{ m}}{0.17143\text{ m}} = 2.625\times 10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon / D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \quad \rightarrow \quad \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{2.625\times 10^{-4}}{3.7} + \frac{2.51}{72{,}506\sqrt{f}}\right)$$

It gives f = 0.02036. Then the pressure drop in the duct and the required pumping power become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.02036\frac{10\text{ m}}{0.17143\text{ m}}\frac{(1.145\text{ kg/m}^3)(7\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ Pa}}{1\text{ N/m}^2}\right) = 33.317\text{ Pa}$$

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (0.21\text{ m}^3\text{/s})(33.317\text{ Pa})\left(\frac{1\text{ W}}{1\text{ Pa}\cdot\text{m}^3\text{/s}}\right) = 6.9965\text{ W} \cong \mathbf{7.00\ W}$$

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.02008, which is reasonably close to 0.02037. Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be much more than this due to fan inefficiency; the electrical power input will be even more due to motor inefficiency.

8-44E

Solution Water passes through copper tubes at a specified rate. The pumping power required per ft length to maintain flow is to be determined.

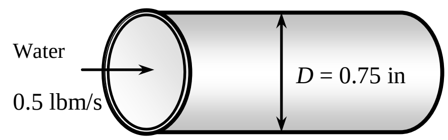
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water at 70°F are ρ = 62.30 lbm/ft^3 and μ = 6.556×10^{-4} lbm/ft·s. The roughness of copper tubing is 5×10^{-6} ft.

Analysis First we calculate the average velocity and the Reynolds number to determine the flow regime:

$$V = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho(\pi D^2/4)} = \frac{0.5\,\text{lbm}/s}{(62.30\,\text{lbm/ft}^3)[\pi(0.75/12\,\text{ft})^2/4]} = 2.616\,\text{ft/s}$$

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(62.30\,\text{lbm/ft}^3)(2.616\,\text{ft/s})(0.75/12\,\text{ft})}{6.556\times10^{-4}\,\text{lbm/ft}\cdot\text{s}} = 15{,}540$$



which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{5\times10^{-6}\,\text{ft}}{0.75/12\,\text{ft}} = 8\times10^{-5}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{8\times10^{-5}}{3.7} + \frac{2.51}{15{,}540\sqrt{f}}\right)$$

It gives f = 0.02771. Then the pressure drop and the required power input become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.02771\frac{1\,\text{ft}}{0.75/12\,\text{ft}}\frac{(62.30\,\text{lbm/ft}^3)(2.616\,\text{ft/s})^2}{2}\left(\frac{1\,\text{lbf}}{32.2\,\text{lbm}\cdot\text{ft/s}^2}\right) = 2.935\,\text{lbf/ft}^2$$

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = \frac{\dot{m}\Delta P}{\rho} = \frac{(0.5\,\text{lbm/s})(2.935\,\text{lbf/ft}^2)}{62.30\,\text{lbm/ft}^3}\left(\frac{1\,\text{W}}{0.737\,\text{lbf}\cdot\text{ft/s}}\right) = \mathbf{0.0320\,W}\ \text{(per ft length)}$$

Therefore, useful power input in the amount of 0.0320 W is needed per ft of tube length to overcome the frictional losses in the pipe.

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.02757, which is sufficiently close to 0.02771. Also, the friction factor corresponding to ε = 0 in this case is 0.02756, which indicates that copper pipes can be assumed to be smooth with a negligible error. Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

8-45

Solution The pressure of oil in a pipe which discharges into the atmosphere is measured at a certain location. The flow rates are to be determined for 3 different orientations.

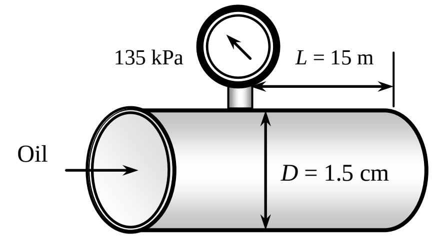
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is laminar (to be verified). **4** The pipe involves no components such as bends, valves, and connectors. **5** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of oil are given to be $\rho = 876$ kg/m^3 and $\mu = 0.24$ kg/m·s.

Analysis The pressure drop across the pipe and the cross-sectional area are

$$\Delta P = P_1 - P_2 = 135 - 88 = 47 \text{ kPa}$$

$$A_c = \pi D^2 / 4 = \pi (0.015 \text{ m})^2 / 4 = 1.767 \times 10^{-4} \text{ m}^2$$



(*a*) The flow rate for all three cases can be determined from,

$$\dot{V} = \frac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128 \mu L}$$

where θ is the angle the pipe makes with the horizontal. For the horizontal case, θ = 0 and thus sin θ = 0. Therefore,

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128 \mu L} = \frac{(47 \text{ kPa})\pi (0.015 \text{ m})^4}{128(0.24 \text{ kg/m}\cdot\text{s})(15 \text{ m})}\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ N}}\right)\left(\frac{1000 \text{ N/m}^2}{1 \text{ kPa}}\right) = \mathbf{1.62 \times 10^{-5}\ m^3/s}$$

(*b*) For uphill flow with an inclination of 8°, we have θ = +8°, and

$$\dot{V}_{\text{uphill}} = \frac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128 \mu L}$$

$$= \frac{[(47{,}000 \text{ Pa} - (876 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(15 \text{ m}) \sin 8°]\pi (0.015 \text{ m})^4}{128(0.24 \text{ kg/m}\cdot\text{s})(15 \text{ m})}\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ Pa}\cdot\text{m}^2}\right)$$

$$= \mathbf{1.00 \times 10^{-5}\ m^3/s}$$

(*c*) For downhill flow with an inclination of 8°, we have θ = -8°, and

$$\dot{V}_{\text{downhill}} = \frac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128 \mu L}$$

$$= \frac{[(47{,}000 \text{ Pa} - (876 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(15 \text{ m}) \sin(-8°)]\pi (0.015 \text{ m})^4}{128(0.24 \text{ kg/m}\cdot\text{s})(15 \text{ m})}\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ Pa}\cdot\text{m}^2}\right)$$

$$= \mathbf{2.24 \times 10^{-5}\ m^3/s}$$

The flow rate is the highest for downhill flow case, as expected. The average fluid velocity and the Reynolds number in this case are

$$V = \frac{\dot{V}}{A_c} = \frac{2.24 \times 10^{-5} . \text{m}^3/\text{s}}{1.767 \times 10^{-4} \text{ m}^2} = 0.127 \text{ m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(876 \text{ kg/m}^3)(0.127 \text{ m/s})(0.015 \text{ m})}{0.24 \text{ kg/m}\cdot\text{s}} = 7.0$$

which is less than 2300. Therefore, the flow is **laminar** for all three cases, and the analysis above is valid.

Discussion Note that the flow is driven by the combined effect of pressure difference and gravity. As can be seen from the calculated rates above, gravity opposes uphill flow, but helps downhill flow. Gravity has no effect on the flow rate in the horizontal case.

8-46

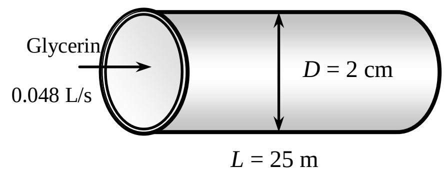
Solution Glycerin is flowing through a horizontal pipe which discharges into the atmosphere at a specified flow rate. The absolute pressure at a specified location in the pipe, and the angle θ that the pipe must be inclined downwards for the pressure in the entire pipe to be atmospheric pressure are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is laminar (to be verified). **4** The pipe involves no components such as bends, valves, and connectors. **5** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of glycerin at 40°C are given to be ρ = 1252 kg/m^3 and μ = 0.27 kg/m·s.

Analysis (*a*) The flow rate for horizontal or inclined pipe can be determined from

$$\dot{V} = \frac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128\mu L} \qquad (1)$$



where θ is the angle the pipe makes with the horizontal. For the horizontal case, θ = 0 and thus sin θ = 0. Therefore,

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128\mu L} \qquad (2)$$

Solving for ΔP and substituting,

$$\Delta P = \frac{128\mu L \dot{V}_{\text{horiz}}}{\pi D^4} = \frac{128(0.27\ \text{kg/m}\cdot\text{s})(25\ \text{m})(0.048\times 10^{-3}\ \text{m}^3/\text{s})}{\pi(0.02\ \text{m})^4}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 82.5\ \text{kN/m}^2 = 82.5\ \text{kPa}$$

Then the pressure 25 m before the pipe exit becomes

$$\Delta P = P_1 - P_2 \quad \rightarrow \quad P_1 = P_2 + \Delta P = 100 + 82.5 = \mathbf{182.5\ kPa}$$

(*b*) When the flow is gravity driven downhill with an inclination θ, and the pressure in the entire pipe is constant at the atmospheric pressure, the hydrostatic pressure rise with depth is equal to pressure drop along the pipe due to frictional effects. Setting $\Delta P = P_1 - P_2 = 0$ in Eq. (1) and substituting, θ is determined to be

$$\dot{V}_{\text{downhill}} = \frac{\rho g \sin\theta \pi D^4}{128\mu}$$

$$0.048\times 10^{-3}\ \text{m}^3/\text{s} = \frac{-(1252\ \text{kg/m}^3)(9.81\ \text{m/s}^2)\sin\theta\pi(0.02\ \text{m})^4}{128(0.27\ \text{kg/m}\cdot\text{s})} \quad \rightarrow \quad \theta = \mathbf{-15.6}°$$

Therefore, the pipe must be inclined 15.6° downwards from the horizontal to maintain flow in the pipe at the same rate.

Verification: The average fluid velocity and the Reynolds number in this case are

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.048\times 10^{-3}\ \text{m}^3/\text{s}}{\pi(0.02\ \text{m})^2/4} = 0.153\ \text{m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(1252\ \text{kg/m}^3)(0.153\ \text{m/s})(0.02\ \text{m})}{0.27\ \text{kg/m}\cdot\text{s}} = 14.2$$

which is less than 2300. Therefore, the flow is **laminar**, as assumed, and the analysis above is valid.

Discussion Note that the flow is driven by the combined effect of pressure difference and gravity. Gravity has no effect on the flow rate in the horizontal case, but it governs the flow alone when there is no pressure difference across the pipe.

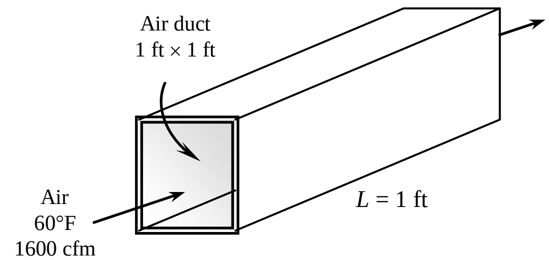
8-47E

Solution Air is flowing through a square duct made of commercial steel at a specified rate. The pressure drop and head loss per ft of duct are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** Air is an ideal gas. **4** The duct involves no components such as bends, valves, and connectors. **5** The flow section involves no work devices such as fans or turbines.

Properties The density and dynamic viscosity of air at 1 atm and 60°F are $\rho = 0.07633$ lbm/ft^3, $\mu = 0.04365$ lbm/ft·h, and $\nu = 0.5718$ ft^2/s $= 1.588\times10^{-4}$ ft^2/s. The roughness of commercial steel surfaces is $\varepsilon = 0.00015$ ft.

Analysis The hydraulic diameter, the average velocity, and the Reynolds number in this case are



$$D_h = \frac{4A_c}{p} = \frac{4a^2}{4a} = a = 1\text{ ft}$$

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{a^2} = \frac{1600\text{ ft}^3/\text{min}}{(1\text{ ft})^2} = 1600\text{ ft/min} = 26.67\text{ ft/s}$$

$$\text{Re} = \frac{VD_h}{\nu} = \frac{(26.67\text{ ft/s})(1\text{ ft})}{1.588\times10^{-4}\text{ ft}^2/\text{s}} = 1.679\times10^5$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the duct is

$$\varepsilon / D_h = \frac{0.00015\text{ ft}}{1\text{ ft}} = 1.5\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{1.5\times10^{-4}}{3.7} + \frac{2.51}{1.679\times10^5\sqrt{f}}\right)$$

It gives $f = 0.01721$. Then the pressure drop in the duct and the head loss become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.01721\frac{1\text{ ft}}{1\text{ ft}}\frac{(0.07633\text{ lbm/ft}^3)(26.67\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}}\right) = \mathbf{0.0145\ lbf/ft^2}$$

$$h_L = \frac{\Delta P_L}{\rho g} = f\frac{L}{D}\frac{V^2}{2g} = 0.01721\frac{1\text{ ft}}{1\text{ ft}}\frac{(26.67\text{ ft/s})^2}{2(32.2\text{ ft/s}^2)} = \mathbf{0.190\ ft}$$

Discussion The required pumping power in this case is

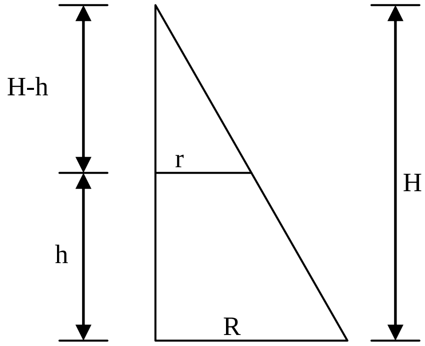
$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (1600/60\text{ ft}^3/\text{s})(0.0145\text{ lbf/ft}^2)\left(\frac{1\text{ W}}{0.737\text{ lbf}\cdot\text{ft/s}}\right) = 0.525\text{ W (per ft length)}$$

Therefore, 0.525 W of mechanical power needs to be imparted to the fluid per ft length of the duct. The shaft power will be more than this due to fan inefficiency; the electrical power input will be even more due to motor inefficiency. Also, the friction factor could be determined easily from the explicit Haaland relation. It would give $f = 0.01697$, which is sufficiently close to 0.01721.

8-48

Solution Water enters into a cone through a small hole at the base. A relation for the variation of water height from the cone base with time is to be obtained.

Analysis From the conservation of mass principle we write



$$\frac{d}{dt}\int_{CV} dV + \dot{V}_{out} - \dot{V}_{in} = 0 \rightarrow \frac{dV_{CV}}{dt} = \dot{V}_{in} = C_d A_h v \qquad (1)$$

The volume of water in the control volume at time t is given by the following relation:

$$V_{CV} = \frac{\pi h}{3}(R^2 + rR + r^2) \qquad (2)$$

From the figure we have the followig linear relation between r and h:

$$r = \frac{R}{H}(H-h) = R - \frac{R}{H}h$$

Substituting in Eq. 2 gives

$$V_{CV} = \frac{\pi R^2}{3H^2}(3hH^2 - 3Hh^2 + h^3)$$

Substituting in Eq. 1 yields

$$\frac{d}{dt}\left(\frac{\pi R^2}{3H^2}(3hH^2 - 3Hh^2 + h^3)\right) = C_d A_h v$$

or

$$\frac{\pi R^2}{3H^2}\frac{d}{dt}(3H^2 h - 3Hh^2 + h^3) = C_d A_h v$$

$$\frac{\pi R^2}{3H^2}\left(3H^2\frac{dh}{dt} - 6Hh\frac{dh}{dt} + 3h^2\frac{dh}{dt}\right) = C_d A_h v$$

$$\frac{\pi R^2}{H^2}(H^2 - 2Hh + h^2)\frac{dh}{dt} = C_d A_h v \rightarrow \frac{\pi R^2}{H^2}(H-h)^2\frac{dh}{dt} = C_d A_h v$$

or

$$(H-h)^2 dh = \frac{C_d A_h v H^2}{\pi R^2}dt$$

where

$$k = \frac{C_d A_h v H^2}{\pi R^2}$$

Integrating

$$-\frac{1}{3}(H-h)^3 = \frac{C_d A_h v H^2}{\pi R^2}t + C$$

Introducing $h(0) = 0$

$$-\frac{H^3}{3} = C$$

Then we obtain

$$-\frac{1}{3}(H-h)^3 = \frac{C_d A_h v H^2}{\pi R^2}t - \frac{H^3}{3}$$

$$h(t) = H - \sqrt[3]{H^3 - kt}$$

where

$$k = 3\frac{C_d A_h v H^2}{\pi R^2}$$

8-49

Solution The velocity profile for incompressible turbulent flow in a pipe is given. An expression for the average velocity in the pipe is to be obtained.

Assumptions **1** The flow is steady and incompressible.

Analysis

$$\overline{V} = \frac{1}{A}\int_A u dA = \frac{1}{\cancel{\pi} R^2}\int_0^R u_{max}\left(1 - \frac{r}{R}\right)^{\frac{1}{7}} 2\cancel{\pi} r dr$$

$$\overline{V} = \frac{2u_{max}}{R^2}\int_0^R \left(1-\frac{r}{R}\right)^{\frac{1}{7}} r dr \quad \text{or}$$

$$\overline{V} = 2u_{max}\int_0^R \left(1-\frac{r}{R}\right)^{\frac{1}{7}} \frac{r}{R} d\left(\frac{r}{R}\right)$$

Letting $1-\frac{r}{R}=x$, $-d\left(\frac{r}{R}\right)=dx$ and $\frac{r}{R}=1-x$

Therefore,

$$\overline{V} = 2u_{max}\int (x)^{\frac{1}{7}}(1-x)(-dx) = 2u_{max}\int\left(x^{\frac{8}{7}} - x^{\frac{1}{7}}\right)dx$$

$$= 2u_{max}\left(\frac{7}{15}x^{\frac{15}{7}} - \frac{7}{8}x^{\frac{8}{7}}\right) = 2u_{max}\left(\frac{7}{15}\left(1-\frac{r}{R}\right)^{\frac{15}{7}} - \frac{7}{8}\left(1-\frac{r}{R}\right)^{\frac{8}{7}}\right)$$

$$\overline{V} = 2u_{max}\left[0 - \left(\underbrace{\frac{7}{15} - \frac{7}{8}}_{-0.408}\right)\right] = 0.816u_{max}$$

8-50

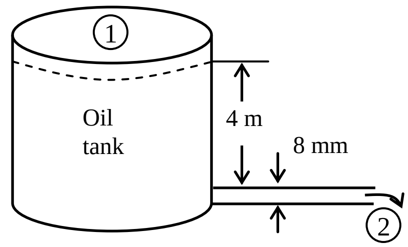
Solution Oil is being discharged by a horizontal pipe from a storage tank open to the atmosphere. The flow rate of oil through the pipe is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The entrance and exit loses are negligible. **4** The flow is laminar (to be verified). **5** The pipe involves no components such as bends, valves, and connectors. **6** The piping section involves no work devices such as pumps and turbines.

Properties The density and kinematic viscosity of oil are given to be ρ = 850 kg/m^3 and ν = 0.00062 m^2/s, respectively. The dynamic viscosity is calculated to be

$$\mu = \rho\nu = (850\ \mathrm{kg/m^3})(0.00062\ \mathrm{m^2}/s) = 0.527\ \mathrm{kg/m\cdot s}$$

Analysis We solve the problem two ways for comparison.



Method 1 – First, the more rigorous way, using the energy equation: We take a control volume with the surface of the oil tank as the inlet (1), and the pipe discharge as the outlet (2), as sketched. The energy equation in head form from 1 to 2 (see Chapter 5) is

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump, u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine, e}} + h_L$$

but for our control volume, $P_1 = P_2 = P_{\text{atm}}$, so the pressure terms cancel. Also, V_1 is negligibly small compared to V_2 since the tank is so large compared to the pipe. Also, there are no turbines or pumps in the flow. Thus, the energy equation reduces to

$$z_1 - z_2 = \alpha_2 \frac{V_2^2}{2g} + h_L \qquad (1)$$

The kinetic energy correction factor and the equation for the head loss term both depend on whether the flow in the pipe is laminar or turbulent. We assume one or the other, and then verify at the end whether our assumption was correct. Since the fluid is very viscous and the diameter is small, *we assume laminar flow*, for which α_2 = 2 if the flow is fully developed at the end of the pipe. Also, for fully developed laminar pipe flow, the Darcy friction factor is 64/Re, and therefore the irreversible head loss is

$$h_L = f\frac{L}{D}\frac{V_{\text{avg}}^{\ 2}}{2g} = \frac{64}{\text{Re}}\frac{L}{D}\frac{V_2^2}{2g} = \frac{64\mu}{\rho D V_2}\frac{L}{D}\frac{V_2^2}{2g} = \frac{32\mu L V_2}{\rho g D^2} \qquad (2)$$

where we have also used the fact that $V_2 = V_{\text{avg}}$. Combining Eqs. 1 and 2, we get

$$\alpha_2 \frac{V_2^2}{2g} + \frac{32\mu L V_2}{\rho g D^2} - (z_1 - z_2) = 0 \qquad (3)$$

Equation 3 is in standard form for a quadratic equation for V_2, which we can easily solve, yielding

$$V_2 = \frac{-\dfrac{32\mu L}{\rho g D^2} \pm \sqrt{\left(\dfrac{32\mu L}{\rho g D^2}\right)^2 + 4\dfrac{\alpha_2}{2g}(z_1 - z_2)}}{\dfrac{\alpha_2}{g}} \qquad (4)$$

The negative root of Eq. 4 makes no physical sense, since the velocity cannot be negative at the outlet, so we take the positive root to calculate V_2,

$$V_2 = \frac{-\dfrac{32(0.527\ \mathrm{kg/m\cdot s})(40\ \mathrm{m})}{(850\ \mathrm{kg/m^3})(9.807\ \mathrm{m/s^2})(0.008\ \mathrm{m})^2} + \sqrt{\left(\dfrac{32(0.527\ \mathrm{kg/m\cdot s})(40\ \mathrm{m})}{(850\ \mathrm{kg/m^3})(9.807\ \mathrm{m/s^2})(0.008\ \mathrm{m})^2}\right)^2 + 4\dfrac{2}{2(9.807\ \mathrm{m/s^2})}(4\ \mathrm{m})}}{\dfrac{2}{9.807\ \mathrm{m/s^2}}}$$

$$= 0.0031632\ \mathrm{m/s}$$

from which we calculate the volume flow rate,

$$\dot{V} = A_c V_2 = (\pi D^2/4)V_2 = \left[\pi(0.008\,\text{m})^2/4\right](0.0031632\,\text{m/s}) = 1.590\times10^{-7}\ \text{m/s}$$

Finally, we verify that the flow is indeed laminar by calculating the Reynolds number,

$$\text{Re} = \frac{\rho V_2 D}{\mu} = \frac{(850\,\text{kg/m}^3)(0.0031632\,\text{m/s})(0.008\,\text{m})}{0.527\,\text{kg/m}\cdot\text{s}} = 0.0408$$

Since this Reynolds number is much lower than 2300, we are confident that the flow is laminar, and thus the analysis is correct.

Method 2 – We re-solve the problem making the assumption that since the velocity through the pipe is so small, the pressure at the pipe entrance is nearly the same as the hydrostatic pressure at that location. The pressure at the bottom of the tank is

$$\begin{aligned} P_{1,\text{gage}} &= \rho g h \\ &= (850\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(4\,\text{m})\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right) \\ &= 33.35\,\text{kN/m}^2 \end{aligned}$$

Disregarding inlet and outlet losses, the pressure drop across the pipe is

$$\Delta P = P_1 - P_2 = P_1 - P_{\text{atm}} = P_{1,\text{gage}} = 33.35\,\text{kN/m}^2 = 33.35\,\text{kPa}$$

The flow rate through a horizontal pipe in laminar flow is determined from

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128\mu L} = \frac{(33.35\,\text{kN/m}^2)\pi(0.008\,\text{m})^4}{128(0.527\,\text{kg/m}\cdot\text{s})(40\text{ m})}\left(\frac{1000\,\text{kg}\cdot\text{m/s}^2}{1\,\text{kN}}\right) = \mathbf{1.590\times10^{-7}\ m^3/s}$$

The average fluid velocity and the Reynolds number in this case are

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{1.590\times10^{-7}\ \text{m}^3/\text{s}}{\pi(0.008\,\text{m})^2/4} = 3.164\times10^{-3}\ \text{m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(850\,\text{kg/m}^3)(3.164\times10^{-3}\ \text{m/s})(0.008\,\text{m})}{0.527\,\text{kg/m}\cdot\text{s}} = 0.0408$$

which is less than 2300. Therefore, the flow is *laminar* and the analysis above is valid.

Discussion The flow rate will be somewhat less when the inlet and outlet losses are considered, especially when the inlet is not well-rounded. The two methods give the same answer to four significant digits. This justifies the assumption made in the second method.

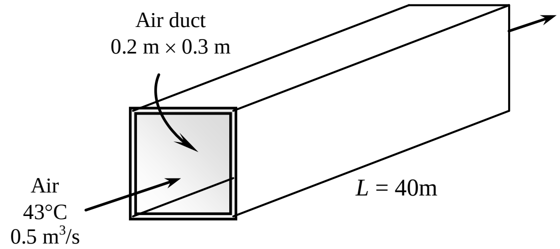
8-51

Solution Air in a heating system is distributed through a rectangular duct made of commercial steel at a specified rate. The pressure drop and head loss through a section of the duct are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** Air is an ideal gas. **4** The duct involves no components such as bends, valves, and connectors. **5** The flow section involves no work devices such as fans or turbines.

Properties The roughness of commercial steel surfaces is ε = 0.000045 m. The dynamic viscosity of air at 40°C is $\mu = 1.918\times10^{-5}$ kg/m·s, and it is independent of pressure. The density of air listed in that table is for 1 atm. The density at 105 kPa and 315 K can be determined from the ideal gas relation to be

$$\rho = \frac{P}{RT} = \frac{105 \text{ kPa}}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg.K})(40+273.15 \text{ K})} = 1.1683 \text{ kg/m}^3$$



Analysis The hydraulic diameter, average velocity, and Reynolds number are

$$D_h = \frac{4A_c}{p} = \frac{4ab}{2(a+b)} = \frac{4(0.3\text{ m})(0.20\text{ m})}{2(0.3+0.20)\text{ m}} = 0.24\text{ m}$$

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{a\times b} = \frac{0.5 \text{ m}^3/\text{s}}{(0.3 \text{ m})(0.2 \text{ m})} = 8.3333 \text{ m/s}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{(1.1683 \text{ kg/m}^3)(8.3333 \text{ m/s})(0.24 \text{ m})}{1.918\times10^{-5} \text{ kg/m}\cdot\text{s}} = 121{,}825$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the duct is

$$\varepsilon / D_h = \frac{4.5\times10^{-5} \text{ m}}{0.24\text{ m}} = 1.875\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \quad \rightarrow \quad \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{1.875\times10^{-4}}{3.7} + \frac{2.51}{121{,}825\sqrt{f}}\right)$$

It gives f = 0.01833. Then the pressure drop in the duct and the head loss become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.01833\frac{40 \text{ m}}{0.24 \text{ m}}\frac{(1.1683 \text{ kg/m}^3)(8.3333 \text{ m/s})^2}{2}\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = 123.93 \text{ N/m}^2 \cong \mathbf{124\ Pa}$$

$$h_L = \frac{\Delta P_L}{\rho g} = f\frac{L}{D}\frac{V^2}{2g} = 0.01833\frac{40 \text{ m}}{0.24 \text{ m}}\frac{(8.3333 \text{ m/s})^2}{2(9.81 \text{ m/s}^2)} = 10.813 \text{ m} \cong \mathbf{10.8\ m}$$

Discussion All final answers are given to three significant digits, but 4 or 5 significant digits are recorded for intermediate steps in order to avoid round-off error. The required pumping power in this case is

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (0.5\text{ m}^3/\text{s})(124\text{ Pa})\left(\frac{1 \text{ W}}{1 \text{ Pa}\cdot\text{m}^3/\text{s}}\right) = 62 \text{ W}$$

Therefore, 62 W of mechanical power needs to be imparted to the fluid. The shaft power will be more than this due to fan inefficiency; the electrical power input will be even more due to motor inefficiency. Also, the friction factor could be determined easily from the explicit Haaland relation. It would give f = 0.0181, which is sufficiently close to 0.0183.

8-52

Solution Glycerin is flowing through a smooth pipe with a specified average velocity. The pressure drop per 10 m of the pipe is to be determined.

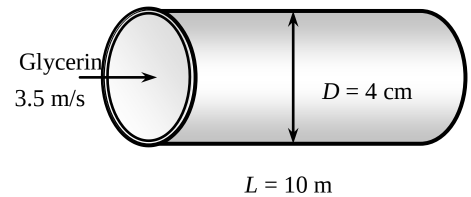
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of glycerin at 40°C are given to be ρ = 1252 kg/m^3 and μ = 0.27 kg/m·s, respectively.

Analysis The volume flow rate and the Reynolds number are

$$\dot{V} = VA_c = V(\pi D^2/4) = (3.5\text{ m/s})[\pi(0.04\text{ m})^2/4] = 0.004398\text{ m}^3/\text{s}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{(1252\text{ kg/m}^3)(3.5\text{ m/s})(0.04\text{ m})}{0.27\text{ kg/m}\cdot\text{s}} = 649.2$$



which is less than 2300. Therefore, the flow is laminar, and the friction factor for this circular pipe is

$$f = \frac{64}{\text{Re}} = \frac{64}{649.2} = 0.09859$$

Then the pressure drop in the pipe becomes

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.09859\frac{10\text{ m}}{0.04\text{ m}}\frac{(1252\text{ kg/m}^3)(3.5\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = \mathbf{189\text{ kPa}}$$

Discussion The required pumping power in this case is

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (0.004398\text{ m}^3/\text{s})(189\text{ kPa})\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{0.831\text{ kW}}$$

Therefore, 0.831 kW of mechanical power needs to be imparted to the fluid. The shaft power will be more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

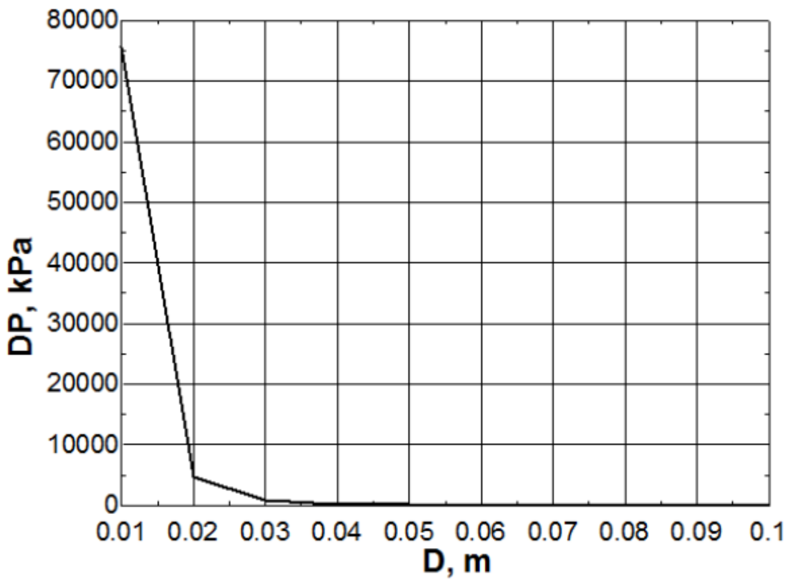
8-53

Solution In the previous problem, the effect of the pipe diameter on the pressure drop for the same constant flow rate is to be investigated by varying the pipe diameter from 1 cm to 10 cm in increments of 1 cm.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81
Vdot=3.5*pi*(0.05)^2/4
Ac=pi*D^2/4
rho= 1252
nu=mu/rho
mu=0.27
L= 10
V=Vdot/Ac
"Reynolds number"
Re=V*D/nu
f=64/Re
DP=f*(L/D)*rho*V^2/2000 "kPa"
W=Vdot*DP "kW"
```

D, m	*ΔP*, kPa	*V*, m/s	Re
0.01	75600	87.5	4057
0.02	4725	21.88	2029
0.03	933.3	9.722	1352
0.04	295.3	5.469	1014
0.05	121	3.5	811.5
0.06	58.33	2.431	676.2
0.07	31.49	1.786	579.6
0.08	18.46	1.367	507.2
0.09	11.52	1.08	450.8
0.1	7.56	0.875	405.7



Discussion The pressure drop decays quite rapidly with increasing diameter – by several orders of magnitude, in fact. We conclude that larger diameter pipes are better when pressure drop is of concern. Of course, bigger pipes cost more and take up more space, so there is typically an optimum pipe size that is a compromise between cost and practicality.

8-54E

Solution The pressure readings across a pipe are given. The flow rates are to be determined for three different orientations of horizontal, uphill, and downhill flow.

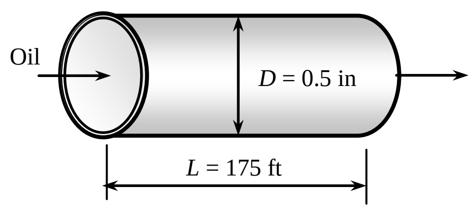
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is laminar (to be verified). **4** The pipe involves no components such as bends, valves, and connectors. **5** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of oil are given to be ρ = 56.8 lbm/ft^3 and μ = 0.0278 lbm/ft·s, respectively.

Analysis The pressure drop across the pipe and the cross-sectional area of the pipe are

$$\Delta P = P_1 - P_2 = 80 - 14 = 66\text{ psi}$$

$$A_c = \pi D^2/4 = \pi(0.5/12\text{ ft})^2/4 = 0.001364\text{ ft}^2$$



(*a*) The flow rate for all three cases can be determined from

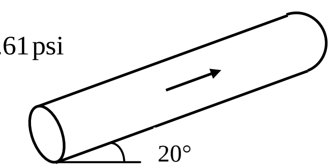
$$\dot{V} = \frac{(\Delta P - \rho g L\sin\theta)\pi D^4}{128\mu L}$$

where θ is the angle the pipe makes with the horizontal. For the horizontal case, θ = 0 and thus sin θ = 0. Therefore,

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128\mu L} = \frac{(66\text{ psi})\pi(0.5/12\text{ ft})^4}{128(0.0278\text{ lbm/ft}\cdot\text{s})(175\text{ ft})}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right) = \mathbf{0.00465\,ft^3/s}$$

(*b*) For uphill flow with an inclination of 20°, we have θ = +20°, and

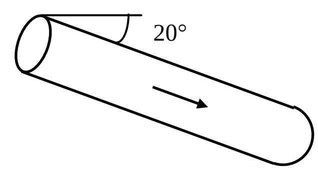
$$\rho g L\sin\theta = (56.8\text{ lbm/ft}^3)(32.2\text{ ft/s}^2)(175\text{ ft})\sin 20°\left(\frac{1\text{ psi}}{144\text{ lbf/ft}^2}\right)\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 23.61\text{ psi}$$



$$\dot{V}_{\text{uphill}} = \frac{(\Delta P - \rho g L\sin\theta)\pi D^4}{128\mu L} =$$

$$= \frac{(66 - 23.61\text{ psi})\pi(0.5/12\text{ ft})^4}{128(0.0278\text{ lbm/ft}\cdot\text{s})(175\text{ ft})}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right) = \mathbf{0.00299\,ft^3/s}$$

(*c*) For downhill flow with an inclination of 20°, we have θ = -20°, and



$$\dot{V}_{\text{downhill}} = \frac{(\Delta P - \rho g L\sin\theta)\pi D^4}{128\mu L}$$

$$= \frac{[66 - (-23.61)\text{ psi}]\pi(0.5/12\text{ ft})^4}{128(0.0278\text{ lbm/ft}\cdot\text{s})(175\text{ ft})}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right) = \mathbf{0.00201\,ft^3/s}$$

The flow rate is the highest for downhill flow case, as expected. The average fluid velocity and the Reynolds number in this case are

$$V = \frac{\dot{V}}{A_c} = \frac{0.00201\text{ ft}^3/\text{s}}{0.001364\text{ ft}^2} = 1.474\text{ ft/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(56.8\text{ lbm/ft}^3)(1.474\text{ ft/s})(0.5/12\text{ ft})}{0.0278\text{ lbm/ft}\cdot\text{s}} = 126$$

which is less than 2300. Therefore, the flow is **laminar** for all three cases, and the analysis above is valid.

Discussion Note that the flow is driven by the combined effect of pressure difference and gravity. As can be seen from the calculated rates above, gravity opposes uphill flow, but helps downhill flow. Gravity has no effect on the flow rate in the horizontal case. Downhill flow can occur even in the absence of an applied pressure difference.

8-55

Solution Liquid ammonia is flowing through a copper tube at a specified mass flow rate. The pressure drop, the head loss, and the pumping power required to overcome the frictional losses in the tube are to be determined.

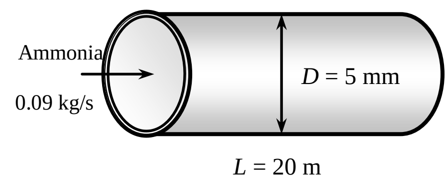
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of liquid ammonia at -20°C are ρ = 665.1 kg/m^3 and μ = 2.361×10^{-4} kg/m·s. The roughness of copper tubing is 1.5×10^{-6} m.

Analysis First we calculate the average velocity and the Reynolds number to determine the flow regime:

$$V = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho(\pi D^2/4)} = \frac{0.09\text{ kg/s}}{(665.1\text{ kg/m}^3)[\pi(0.005\text{ m})^2/4]} = 6.892\text{ m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(665.1\text{ kg/m}^3)(6.892\text{ m/s})(0.005\text{ m})}{2.361\times10^{-4}\text{ kg/m}\cdot\text{s}} = 0.9707\times10^5$$



which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{1.5\times10^{-6}\text{ m}}{0.005\text{ m}} = 3\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{3\times10^{-4}}{3.7} + \frac{2.51}{0.9707\times10^5\sqrt{f}}\right)$$

It gives f = 0.01956. Then the pressure drop, the head loss, and the useful pumping power required become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.01956\frac{20\text{ m}}{0.005\text{ m}}\frac{(665.1\text{ kg/m}^3)(6.892\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = 1237\text{ kPa} \cong \mathbf{1240\ kPa}$$

$$h_L = \frac{\Delta P_L}{\rho g} = f\frac{L}{D}\frac{V^2}{2g} = 0.01956\frac{20\text{ m}}{0.005\text{ m}}\frac{(6.892\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = \mathbf{189\ m}$$

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = \frac{\dot{m}\Delta P}{\rho} = \frac{(0.09\text{ kg/s})(1237\text{ kPa})}{665.1\text{ kg/m}^3}\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3\text{/s}}\right) = \mathbf{0.167\ kW}$$

Therefore, useful power input in the amount of 0.167 kW is needed to overcome the frictional losses in the tube.

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.01928, which is sufficiently close to 0.01956. The friction factor corresponding to ε = 0 in this case is 0.01810, which is about 8% lower. Therefore, the copper tubes in this case are nearly "smooth".

Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

Minor Losses

8-56C

Solution We are to compare two different ways to reduce the minor loss in pipe bends.

Analysis The loss coefficient is lower for flow through a 90° miter elbow with well-designed vanes ($K_L \approx 0.2$) than it is for flow through a smooth curved bend ($K_L \approx 0.9$). Therefore, **using miter elbows with vanes results in a greater reduction in pumping power requirements**.

Discussion Both values are for threaded elbows. The loss coefficients for flanged elbows are much lower.

8-57C

Solution We are to define equivalent length and its relationship to the minor loss coefficient.

Analysis *Equivalent length* is **the length of a straight pipe which would give the same head loss as the minor loss component**. It is related to the minor loss coefficient by

$$L_{\text{equiv}} = \frac{D}{f} K_L$$

Discussion Equivalent length is not as universal as minor loss coefficient because it depends on the roughness and Reynolds number of the equivalent straight section of pipe.

8-58C

Solution We are to discuss the effect of rounding a pipe inlet.

Analysis The effect of rounding of a pipe inlet on the loss coefficient is **(*c*) very significant**.

Discussion In fact, the minor loss coefficient changes from 0.8 for a reentrant pipe inlet to about 0.03 for a well-rounded pipe inlet – quite a significant improvement.

8-59C

Solution We are to discuss the effect of rounding on a pipe outlet.

Analysis The effect of rounding of a pipe exit on the loss coefficient is **(*a*) negligible**.

Discussion At any pipe outlet, all the kinetic energy is wasted, and the minor loss coefficient is equal to α, which is about 1.05 for fully developed turbulent pipe flow. Rounding of the outlet does not help.

8-60C

Solution We are to compare the minor losses of a gradual expansion and a gradual contraction.

Analysis **A gradual expansion, in general, has a greater minor loss coefficient than a gradual contraction** in pipe flow. This is due to the adverse pressure gradient in the boundary layer, which may lead to flow separation.

Discussion Note, however, that pressure is "recovered" in a gradual expansion. In other words, the pressure rises in the direction of flow. Such a device is called a *diffuser*.

8-61C

Solution We are to discuss ways to reduce the head loss in a pipe flow with bends.

Analysis Another way of reducing the head loss associated with turns is to **install turning vanes inside the elbows**.

Discussion There are many other possible answers, such as: **reduce the inside wall roughness of the pipe**, **use a larger diameter pipe**, **shorten the length of pipe as much as possible**, etc.

8-62C

Solution We are to define minor loss and minor loss coefficient.

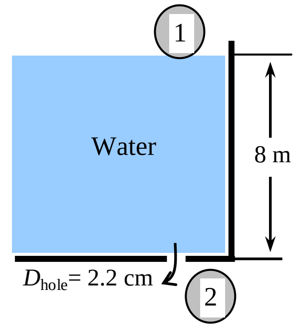
Analysis The **head losses associated with the flow of a fluid through fittings, valves, bends, elbows, tees, inlets, exits, enlargements, contractions, etc.** are called *minor losses*, and are expressed in terms of the *minor loss coefficient* as

$$K_L = \frac{h_L}{V^2/(2g)}$$

Discussion Basically, any irreversible loss that is not due to friction in long, straight sections of pipe is a minor loss.

8-63

Solution Water is to be withdrawn from a water reservoir by drilling a hole at the bottom surface. The flow rate of water through the hole is to be determined for the well-rounded and sharp-edged entrance cases.



Assumptions **1** The flow is steady and incompressible. **2** The reservoir is open to the atmosphere so that the pressure is atmospheric pressure at the free surface. **3** The effect of the kinetic energy correction factor is disregarded, and thus $\alpha = 1$.

Analysis The loss coefficient is K_L = 0.5 for the sharp-edged entrance, and K_L = 0.03 for the well-rounded entrance. We take point 1 at the free surface of the reservoir and point 2 at the exit of the hole. We also take the reference level at the exit of the hole (z_2 = 0). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is zero (V_1 = 0), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

where the head loss is expressed as $h_L = K_L\dfrac{V^2}{2g}$. Substituting and solving for V_2 gives

$$z_1=\alpha_2\frac{V_2^2}{2g}+K_L\frac{V_2^2}{2g} \quad\rightarrow\quad 2gz_1=V_2^2(\alpha_2+K_L) \quad\rightarrow\quad V_2=\sqrt{\frac{2gz_1}{\alpha_2+K_L}}=\sqrt{\frac{2gz_1}{1+K_L}}$$

since α_2 = 1. Note that in the special case of K_L = 0, it reduces to the Toricelli equation $V_2=\sqrt{2gz_1}$, as expected. Then the volume flow rate becomes

$$\dot{V}=A_cV_2=\frac{\pi D_{hole}^2}{4}\sqrt{\frac{2gz_1}{1+K_L}}$$

Substituting the numerical values, the flow rate for both cases are determined to be

Well-rounded entrance: $$\dot{V}=\frac{\pi D_{hole}^2}{4}\sqrt{\frac{2gz_1}{1+K_L}}=\frac{\pi(0.022\text{ m})^2}{4}\sqrt{\frac{2(9.81\text{ m/s}^2)(8\text{ m})}{1+0.03}}=\mathbf{4.69\times10^{-3}\ m^3/s}$$

Sharp-edged entrance: $$\dot{V}=\frac{\pi D_{hole}^2}{4}\sqrt{\frac{2gz_1}{1+K_L}}=\frac{\pi(0.022\text{m})^2}{4}\sqrt{\frac{2(9.81\text{ m/s}^2)(8\text{ m})}{1+0.5}}=\mathbf{3.89\times10^{-3}\ m^3/s}$$

Discussion The flow rate in the case of frictionless flow (K_L = 0) is 4.76×10^{-3} m^3/s. Note that the frictional losses cause the flow rate to decrease by 1.5% for well-rounded entrance, and 18.3% for the sharp-edged entrance.

8-64

Solution Water is discharged from a water reservoir through a circular hole of diameter D at the side wall at a vertical distance H from the free surface. A relation for the "equivalent diameter" of the sharp-edged hole for use in frictionless flow relations is to be obtained.

Assumptions **1** The flow is steady and incompressible. **2** The reservoir is open to the atmosphere so that the pressure is atmospheric pressure at the free surface. **3** The effect of the kinetic energy correction factor is disregarded, and thus $\alpha = 1$.

Analysis The loss coefficient is $K_L = 0.5$ for the sharp-edged entrance, and $K_L = 0$ for the "frictionless" flow. We take point 1 at the free surface of the reservoir and point 2 at the exit of the hole, which is also taken to be the reference level ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is zero ($V_1 = 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine, e}}+h_L \quad\rightarrow\quad H=\alpha_2\frac{V_2^2}{2g}+h_L$$

where the head loss is expressed as $h_L = K_L\frac{V^2}{2g}$. Substituting and solving for V_2 gives

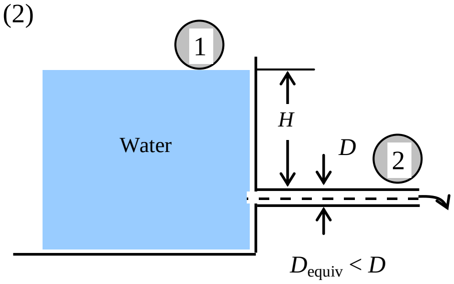
$$H=\alpha_2\frac{V_2^2}{2g}+K_L\frac{V_2^2}{2g} \quad\rightarrow\quad 2gH=V_2^2(\alpha_2+K_L) \quad\rightarrow\quad V_2=\sqrt{\frac{2gH}{\alpha_2+K_L}}=\sqrt{\frac{2gH}{1+K_L}}$$

since $\alpha_2 = 1$. Then the volume flow rate becomes

$$\dot{V}=A_cV_2=\frac{\pi D^2}{4}\sqrt{\frac{2gH}{1+K_L}} \qquad (1)$$

Note that in the special case of $K_L = 0$ (frictionless flow), the velocity relation reduces to the Toricelli equation, $V_{2,\text{frictionless}}=\sqrt{2gH}$. The flow rate in this case through a hole of D_e (equivalent diameter) is

$$\dot{V}=A_{c,\text{equiv}}V_{2,\text{frictionless}}=\frac{\pi D_{\text{equiv}}^2}{4}\sqrt{2gH} \qquad (2)$$



Setting Eqs. (1) and (2) equal to each other gives the desired relation for the equivalent diameter,

$$\frac{\pi D_{\text{equiv}}^2}{4}\sqrt{2gH}=\frac{\pi D^2}{4}\sqrt{\frac{2gH}{1+K_L}}$$

which gives

$$D_{\text{equiv}}=\frac{D}{(1+K_L)^{1/4}}=\frac{D}{(1+0.5)^{1/4}}=\mathbf{0.904\,D}$$

Discussion Note that the effect of frictional losses of a sharp-edged entrance is to reduce the diameter by about 10%. Also, noting that the flow rate is proportional to the square of the diameter, we have $\dot{V}\propto D_{\text{equiv}}^2=(0.904D)^2=0.82D^2$. Therefore, the flow rate through a sharp-edged entrance is about 18% less compared to the frictionless entrance case.

8-65

Solution Water is discharged from a water reservoir through a circular hole of diameter D at the side wall at a vertical distance H from the free surface. A relation for the "equivalent diameter" of the slightly rounded hole for use in frictionless flow relations is to be obtained.

Assumptions **1** The flow is steady and incompressible. **2** The reservoir is open to the atmosphere so that the pressure is atmospheric pressure at the free surface. **3** The effect of the kinetic energy correction factor is disregarded, and thus $\alpha = 1$.

Analysis The loss coefficient is $K_L = 0.12$ for the slightly rounded entrance, and $K_L = 0$ for the "frictionless" flow.

We take point 1 at the free surface of the reservoir and point 2 at the exit of the hole, which is also taken to be the reference level ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is zero ($V_1 = 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine, e}} + h_L \quad \rightarrow \quad H = \alpha_2 \frac{V_2^2}{2g} + h_L$$

where the head loss is expressed as $h_L = K_L \dfrac{V^2}{2g}$. Substituting and solving for V_2 gives

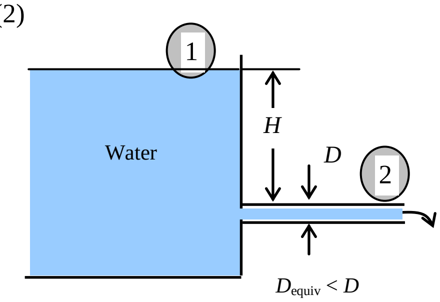
$$H = \alpha_2 \frac{V_2^2}{2g} + K_L \frac{V_2^2}{2g} \quad \rightarrow \quad 2gH = V_2^2(\alpha_2 + K_L) \quad \rightarrow \quad V_2 = \sqrt{\frac{2gH}{\alpha_2 + K_L}} = \sqrt{\frac{2gH}{1 + K_L}}$$

since $\alpha_2 = 1$. Then the volume flow rate becomes

$$\dot{V} = A_c V_2 = \frac{\pi D^2}{4} \sqrt{\frac{2gH}{1 + K_L}} \qquad (1)$$

Note that in the special case of $K_L = 0$ (frictionless flow), the velocity relation reduces to the Toricelli equation, $V_{2,\text{frictionless}} = \sqrt{2gz_1}$. The flow rate in this case through a hole of D_e (equivalent diameter) is

$$\dot{V} = A_{c,\text{equiv}} V_{2,\text{frictionless}} = \frac{\pi D_{\text{equiv}}^2}{4} \sqrt{2gH} \qquad (2)$$



Setting Eqs. (1) and (2) equal to each other gives the desired relation for the equivalent diameter,

$$\frac{\pi D_{\text{equiv}}^2}{4} \sqrt{2gH} = \frac{\pi D^2}{4} \sqrt{\frac{2gH}{1 + K_L}}$$

which gives

$$D_{\text{equiv}} = \frac{D}{(1 + K_L)^{1/4}} = \frac{D}{(1 + 0.12)^{1/4}} = \mathbf{0.972\,D}$$

Discussion Note that the effect of frictional losses of a slightly rounded entrance is to reduce the diameter by about 3%. Also, noting that the flow rate is proportional to the square of the diameter, we have $\dot{V} \propto D_{\text{equiv}}^2 = (0.972D)^2 = 0.945D^2$. Therefore, the flow rate through a slightly rounded entrance is about 5% less compared to the frictionless entrance case.

8-66

Solution A horizontal water pipe has an abrupt expansion. The water velocity and pressure in the smaller diameter pipe are given. The pressure after the expansion and the error that would have occurred if the Bernoulli Equation had been used are to be determined.

Assumptions **1** The flow is steady, horizontal, and incompressible. **2** The flow at both the inlet and the outlet is fully developed and turbulent with kinetic energy corrections factors of $\alpha_1 = \alpha_2 = 1.06$ (given).

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

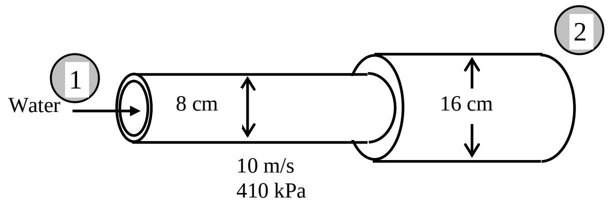
Analysis Noting that ρ = const. (incompressible flow), the downstream velocity of water is

$$\dot m_1 = \dot m_2 \;\rightarrow\; \rho V_1 A_1 = \rho V_2 A_2 \;\rightarrow\; V_2 = \frac{A_1}{A_2}V_1 = \frac{\pi D_1^2/4}{\pi D_2^2/4}V_1 = \frac{D_1^2}{D_2^2}V_1 = \frac{(0.08\text{ m})^2}{(0.16\text{ m})^2}(10\text{ m/s}) = 2.5\text{ m/s}$$

The loss coefficient for sudden expansion and the head loss can be calculated from

$$K_L = \left(1 - \frac{A_{\text{small}}}{A_{\text{large}}}\right)^2 = \left(1 - \frac{D_1^2}{D_2^2}\right)^2 = \left(1 - \frac{0.08^2}{0.16^2}\right)^2 = 0.5625$$

$$h_L = K_L\frac{V_1^2}{2g} = (0.5625)\frac{(10\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 2.87\text{ m}$$



Noting that $z_1 = z_2$ and there are no pumps or turbines involved, the energy equation for the expansion section can be expressed in terms of heads as

$$\frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad\rightarrow\quad \frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} = \frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + h_L$$

Solving for P_2 and substituting,

$$P_2 = P_1 + \rho\left\{\frac{\alpha_1 V_1^2 - \alpha_2 V_2^2}{2} - gh_L\right\}$$

$$= (410\text{ kPa}) + (1000\text{ kg/m}^3)\left\{\frac{1.06(10\text{ m/s})^2 - 1.06(2.5\text{ m/s})^2}{2} - (9.81\text{ m/s}^2)(2.87\text{ m})\right\}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)$$

$$= 431.5\text{ kPa} \cong \mathbf{432\ kPa}$$

Therefore, despite the head (and pressure) loss, the pressure increases from 300 kPa to 321 kPa after the expansion. This is due to the conversion of dynamic pressure to static pressure when the velocity is decreased.

When the head loss is disregarded, the downstream pressure is determined from the Bernoulli equation to be

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad\rightarrow\quad \frac{P_1}{\rho g} + \frac{V_1^2}{2g} = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} \quad\rightarrow\quad P_1 = P_1 + \rho\frac{V_1^2 - V_2^2}{2}$$

Substituting,

$$P_2 = (410\text{ kPa}) + (1000\text{ kg/m}^3)\frac{(10\text{ m/s})^2 - (2.5\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = 456.9\text{ kPa}$$

Therefore, the error in the Bernoulli equation is

Error = $P_{2,\text{ Bernoulli}} - P_2$ = 456.9 – 431.5 = **25.4 kPa**

Note that the use of the Bernoulli equation results in an error of (456.9 – 431.5) / 431.5 = 0.059 or 5.9%.

Discussion It is common knowledge that higher pressure upstream is necessary to cause flow, and it may come as a surprise that the downstream pressure has *increased* after the abrupt expansion, despite the loss. This is because the sum of the three Bernoulli terms which comprise the total head, consisting of pressure head, velocity head, and elevation head, namely $[P/\rho g + ½V^2/g + z]$, drives the flow. With a geometric flow expansion, initially higher velocity head is converted to downstream pressure head, and this increase outweighs the non-convertible and non-recoverable head loss term.

Piping Systems and Pump Selection

8-67C

Solution We are to discuss whether the required pump head is equal to the elevation difference when irreversible head losses are negligible.

Analysis **Yes**, when the head loss is negligible, the required pump head is equal to the elevation difference between the free surfaces of the two reservoirs.

Discussion A pump in a piping system may: (1) raise the fluid's elevation, and/or (2) increase the fluid's kinetic energy, and/or (3) increase the fluid's pressure, and/or (4) overcome irreversible losses. In this case, (2), (3), and (4) are zero or negligible; thus only (1) remains.

8-68C

Solution We are to explain how the operating point of a pipe/pump system is determined.

Analysis The pump installed in a piping system **operates at the point where the *system curve* (required demand) and the *characteristic curve* (available supply) intersect**. This **point of intersection** is called the ***operating point***.

Discussion The volume flow rate "automatically" adjusts itself to reach the operating point.

8-69C

Solution We are to discuss whether attaching a nozzle to a garden hose will increase or decrease the filling time when the hose is used to fill a bucket.

Analysis Any additional component in a flow line introduces additional resistance to flow and thus head loss, which results in a reduction of flow rate for a given pressure difference. Therefore, **the filling time of the bucket will increase**.

Discussion If he used a *diffuser* instead (diverging rather than converging section of pipe), the filling time would decrease somewhat (larger volume flow rate) due to the pressure recovery created by the diffuser.

8-70C

Solution We are to compare discharge times of two water tanks for two cases.

Analysis If we ignore frictional losses in the hose, **the tank with the hose attached will empty first** since it has an available head of 3-m at the start while the other tank has an available head of just 2 m. The water discharging directly to the table continues to accelerate during its free fall from the table to the ground, but this does not affect the flow velocity (and thus flow rate) at the discharge.

Discussion In real life, there are irreversible head losses in the hose, and the answer may change, depending on how significant these losses are.

8-71C

Solution We are to compare the flow rate and pressure drop in two pipes of different diameters in series.

Analysis For a piping system that involves two pipes of different diameters (but of identical length, material, and roughness) connected in *series*, (*a*) **the flow rate through both pipes is the same** and (*b*) **the pressure drop through the smaller diameter pipe is larger**.

Discussion The wall shear stress on the smaller pipe is larger, friction factor f is larger, and thus the head loss is higher.

8-72C

Solution We are to compare the flow rate and pressure drop in two pipes of different diameters in parallel.

Analysis For a piping system that involves two pipes of different diameters (but of identical length, material, and roughness) connected in *parallel*, (*a*) **the flow rate through the larger diameter pipe is larger** and (*b*) **the pressure drop through both pipes is the same**.

Discussion Since the two pipes separate from each other but then later re-join, the pressure drop between the two junctions must be the same, regardless of which pipe segment is under consideration.

8-73C

Solution We are to compare the pressure drop of two different-length pipes in parallel.

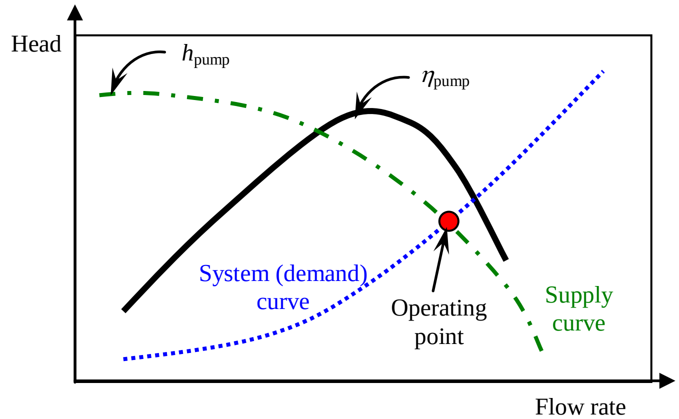
Analysis **The pressure drop through both pipes is the *same*** since the pressure at a point has a single value, and the inlet and exits of these the pipes connected in parallel coincide.

Discussion The length, diameter, roughness, and number and type of minor losses are all irrelevant – for any two pipes in parallel, both have the same pressure drop.

8-74C

Solution We are to draw a pump head versus flow rate chart and identify several parameters.

Analysis The **plot of the head loss versus the flow rate** is called the *system curve* **(required demand)**. The experimentally determined **pump head and pump efficiency versus the flow rate curves** are called *characteristic curves* **(available supply)**. **The pump installed in a piping system operates at the point where the *system curve* and the *characteristic curve* intersect**. This point of intersection is called the *operating point*.



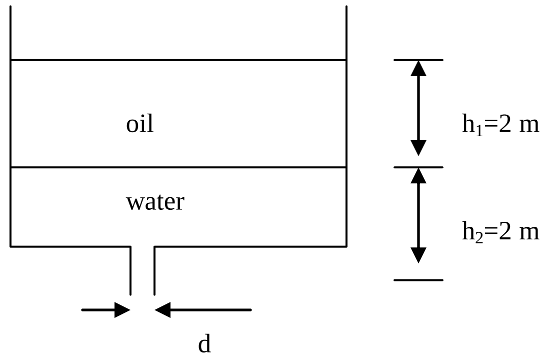
Discussion The volume flow rate "automatically" adjusts itself to reach the operating point.

8-75

Solution Water is withdrawn from a hole at the bottom of a cylindrical tank. The time it will take for the water in the tank to empty completely is to be determined.

Assumptions **1** The flow is steady and incompressible. **1** The Bernoulli equation is applicable.

Analysis



At the water/oil interface, the pressure $p_1 = \gamma_{oil} h_1$

Applying Bernolli Equation between 1-2,

$$\frac{\gamma_{oil} h_1}{\gamma_{water}} + h_2 + 0 = 0 + 0 + \frac{V_2^{\ 2}}{2g} \text{ therefore}$$

$$V_2 = \sqrt{2g\left(h_2 + \frac{\gamma_{oil}}{\gamma_w} h_1\right)}$$

The flow rate from the tank is then

$$Q = C\ A_{hole}\ V_2 = C\ A_h\ V_2 = C\ A_h \sqrt{2g(h_2 + SG_{oil} h_1)}$$

This flow would cause a decrease in the water level by the flow rate of $-A_T \dfrac{dh_2}{dt}$

Therefore we can write

$$-A_T \frac{dh_2}{dt} = C\ A_h \sqrt{2g(h_2 + SGh_1)} = C\ A_h \sqrt{2g}\ \sqrt{h_2 + SGh_1}$$

$$\int_2^0 \frac{dh_2}{\sqrt{h_2 + SGh_1}} = -\frac{CA_h\sqrt{2g}}{A_T} \int_0^t dt$$

$$2\sqrt{h_2 + SGh_1}\Big|_2^0 = -\frac{CA_h\sqrt{2g}}{A_T} t$$

$$2[\sqrt{0.75\times 2} - \sqrt{2 + 0.75\times 2}] = -\frac{0.85\pi\frac{0.01^2}{4}\sqrt{2g}}{1.5} t$$

$$-1.292 = -1.971\times 10^{-4} t$$

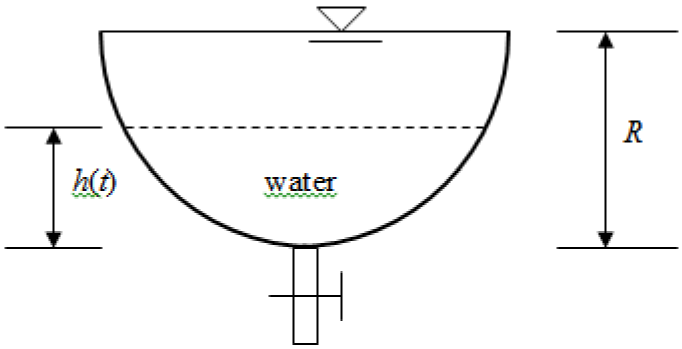
$$t = 6555\ s \cong \mathbf{1.82\ h}$$

8-76

Solution Water is withdrawn from a hole at the bottom of a semi-spherical tank. An expression for the time needed to empty the tank completely is to be determined.

Assumptions **1** The flow is steady and incompressible.

Analysis



The instantaneous flow rate through the hole is

$$Q = CA_h\sqrt{2gh}$$

From the conservation of mass principle, we have

$$CA_h\sqrt{2gh} = -A_T dh = -\pi x^2 dh$$

However x depends on h, such that

$$x^2 = R^2 - (R-h)^2 = 2Rh - h^2$$

Therefore we obtain

$$CA_h\sqrt{2gh}dt = -\pi\left(2Rh - h^2\right)dh$$

Separating variables would yield

$$dt = -\frac{\pi\left(2Rh - h^2\right)dh}{CA_h\sqrt{2g}\sqrt{h}} = -\frac{\pi}{CA_h\sqrt{2g}}\int_{h_1}^{h_2}\left(2Rh^{1/2} - h^{3/2}\right)dh$$

$$t = \frac{\pi}{CA_h\sqrt{2g}}\left[\frac{4}{3}Rh^{3/2} - \frac{2}{5}h^{5/2}\right]_{h_2}^{h_1}$$

Taking h_1=R, and h_2=0, we obtain

$$t = \frac{14\pi R^{5/2}}{15CA_h\sqrt{2g}}$$

8-77

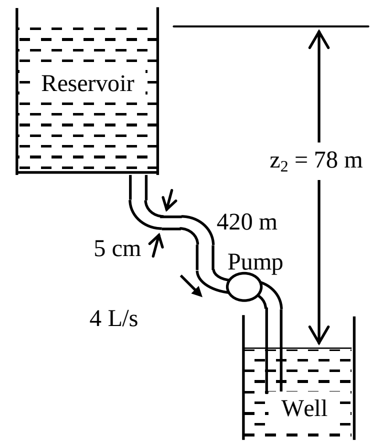
Solution Underground water is to be pumped to a reservoir at a much higher elevation using plastic pipes. The required power input to the pump is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The total minor loss coefficient due to the use elbows, vanes, etc is estimated to be 12. **4** Water level in the well remains constant. **5** Both the well and the reservoir are open to the atmosphere. **6** The kinetic energy correction factors are the same at both the inlet and exit.

Properties The density and dynamic viscosity of water are given to be ρ = 1000 kg/m^3 and μ = 0.00131 kg/m·s. The plastic pipe is smooth and thus ε = 0.

Analysis We choose point 1 at the free surface of well water and point 2 at the free surface of the reservoir in the farm. We note the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and the fluid velocities at both points are low ($V_1 \cong V_2 \cong 0$). We take the free surface of the well as the reference level (z_1 = 0). Then the energy equation for a control volume between these two points simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=z_2+h_L$$



where z_2 = 58+20 = 78 m and $h_L = h_{L,\text{total}} = h_{L,\text{major}} + h_{L,\text{minor}} = \left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$

The average velocity in the pipe and the Reynolds number are

$$V=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4}=\frac{0.004\text{ m}^3/\text{s}}{\pi(0.05\text{ m})^2/4}=2.037\text{ m/s}$$

$$\text{Re}=\frac{\rho VD}{\mu}=\frac{(1000\text{ kg/m}^3)(2.037\text{ m/s})(0.05\text{ m})}{0.00131\text{ kg/m}\cdot\text{s}}=77{,}750$$

which is greater than 4000. Therefore, the flow is turbulent. The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation,

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\ \rightarrow\ \frac{1}{\sqrt{f}}=-2.0\log\left(0+\frac{2.51}{77{,}750\sqrt{f}}\right)$$

It gives f = 0.01897. The sum of the loss coefficients is given to be 12. Then the total head loss, the useful pump head, and the required pumping power become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.01897)\frac{420\text{ m}}{0.05\text{ m}}+12\right)\frac{(2.037\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=36.25\text{ m}$$

$$h_{\text{pump,u}}=z_2+h_L=78+36.25\text{ m}=114.25\text{ m}$$

$$\dot{W}_{\text{pump}}=\frac{\dot{V}\rho g h_{\text{pump,u}}}{\eta_{\text{pump}}}=\frac{(0.004\text{ m}^3/\text{s})(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(114.25\text{ m})}{0.75}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right)=\mathbf{6.0\ kW}$$

Discussion Pumps have different characteristics, depending on their design, and an ordinary 6 kW pump will not do the job. Some pumps are designed for low head and high flow rate, and others for high head and low flow rate. The proper pump model to be purchased must be selected after examining the performance curves supplied by the manufacturer.

8-78E

Solution The flow rate through a piping system connecting two reservoirs is given. The elevation of the source is to be determined.

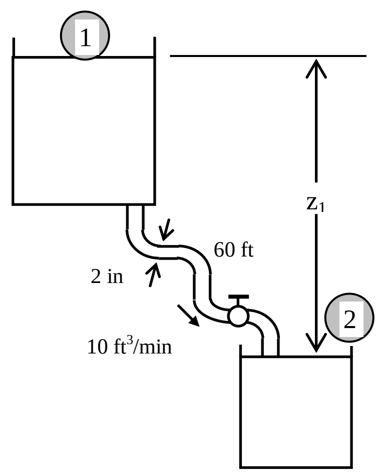
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The elevations of the reservoirs remain constant. **4** There are no pumps or turbines in the piping system.

Properties The density and dynamic viscosity of water at 70°F are ρ = 62.30 lbm/ft^3 and μ = 2.360 lbm/ft·h = 6.556×10^{-4} lbm/ft·s. The roughness of cast iron pipe is ε = 0.00085 ft.

Analysis The piping system involves 60 ft of 2-in diameter piping, a well-rounded entrance (K_L = 0.03), 4 standard flanged elbows (K_L = 0.3 each), a fully open gate valve (K_L = 0.2), and a sharp-edged exit (K_L = 1.0). We choose points 1 and 2 at the free surfaces of the two reservoirs. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$), the fluid velocities at both points are zero ($V_1 = V_2 = 0$), the free surface of the lower reservoir is the reference level ($z_2 = 0$), and that there is no pump or turbine ($h_{pump,u} = h_{turbine} = 0$), the energy equation for a control volume between these two points simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=h_L$$

where $h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\dfrac{L}{D}+\sum K_L\right)\dfrac{V^2}{2g}$



since the diameter of the piping system is constant. The average velocity in the pipe and the Reynolds number are

$$V=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4}=\frac{10/60\ \text{ft}^3/\text{s}}{\pi(2/12\ \text{ft})^2/4}=7.64\ \text{ft/s}$$

$$\text{Re}=\frac{\rho VD}{\mu}=\frac{(62.3\ \text{lbm/ft}^3)(7.64\ \text{ft/s})(2/12\ \text{ft})}{1.307\times10^{-3}\ \text{lbm/ft}\cdot\text{s}}=60{,}700$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{0.00085\ \text{ft}}{2/12\ \text{ft}}=0.0051$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt f}\right)\ \rightarrow\ \frac{1}{\sqrt f}=-2.0\log\left(\frac{0.0051}{3.7}+\frac{2.51}{60{,}700\sqrt f}\right)$$

It gives f = 0.0320. The sum of the loss coefficients is

$$\sum K_L=K_{L,\text{entrance}}+4K_{L,\text{elbow}}+K_{L,\text{valve}}+K_{L,\text{exit}}=0.03+4\times0.3+0.2+1.0=2.43$$

Then the total head loss and the elevation of the source become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g}=\left((0.0320)\frac{60\ \text{ft}}{2/12\ \text{ft}}+2.43\right)\frac{(7.64\ \text{ft/s})^2}{2(32.2\ \text{ft/s}^2)}=12.6\ \text{ft}$$

$$z_1=h_L=\mathbf{12.6\ ft}$$

Therefore, the free surface of the first reservoir must be 12.6 ft above the free surface of the lower reservoir to ensure water flow between the two reservoirs at the specified rate.

Discussion Note that fL/D = 11.5 in this case, which is about 5 folds of the total minor loss coefficient. Therefore, ignoring the sources of minor losses in this case would result in an error of about 20%.

8-79

Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere. The initial velocity from the tank and the time required to empty the tank are to be determined.

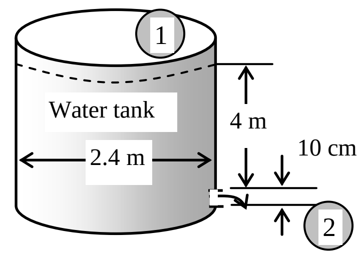
Assumptions **1** The flow is uniform and incompressible. **2** The flow is turbulent so that the tabulated value of the loss coefficient can be used. **3** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The loss coefficient is $K_L = 0.5$ for a sharp-edged entrance.

Analysis (*a*) We take point 1 at the free surface of the tank, and point 2 at the exit of the orifice. We also take the reference level at the centerline of the orifice ($z_2 = 0$), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad z_1 = \alpha_2 \frac{V_2^2}{2g} + h_L$$

where the head loss is expressed as $h_L = K_L \dfrac{V^2}{2g}$. Substituting and solving for V_2 gives



$$z_1 = \alpha_2 \frac{V_2^2}{2g} + K_L \frac{V_2^2}{2g} \quad \rightarrow \quad 2gz_1 = V_2^2(\alpha_2 + K_L) \quad \rightarrow \quad V_2 = \sqrt{\frac{2gz_1}{\alpha_2 + K_L}}$$

where $\alpha_2 = 1$. Noting that initially $z_1 = 2$ m, the initial velocity is determined to be

$$V_2 = \sqrt{\frac{2gz_1}{1+K_L}} = \sqrt{\frac{2(9.81\text{ m/s}^2)(4\text{ m})}{1+0.5}} = \mathbf{7.23\ m/s}$$

The average discharge velocity through the orifice at any given time, in general, can be expressed as

$$V_2 = \sqrt{\frac{2gz}{1+K_L}}$$

where z is the water height relative to the center of the orifice at that time.

(*b*) We denote the diameter of the orifice by D, and the diameter of the tank by D_0. The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the orifice area,

$$\dot{V} = A_{\text{orifice}} V_2 = \frac{\pi D^2}{4} \sqrt{\frac{2gz}{1+K_L}}$$

Then the amount of water that flows through the orifice during a differential time interval dt is

$$d\boldsymbol{V} = \dot{\boldsymbol{V}} dt = \frac{\pi D^2}{4} \sqrt{\frac{2gz}{1+K_L}}\, dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV = A_{\text{tank}}(-dz) = -\frac{\pi D_0^2}{4} dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4} \sqrt{\frac{2gz}{1+K_L}}\, dt = -\frac{\pi D_0^2}{4} dz \quad \rightarrow \quad dt = -\frac{D_0^2}{D^2} \sqrt{\frac{1+K_L}{2gz}}\, dz \quad \rightarrow \quad dt = -\frac{D_0^2}{D^2} \sqrt{\frac{1+K_L}{2g}}\, z^{-1/2} dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained tank) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1+K_L}{2g}}\int_{z=z_1}^{0} z^{-1/2}dz \quad \rightarrow \quad t_f = -\frac{D_0^2}{D^2}\sqrt{\frac{1+K_L}{2g}}\left|\frac{z^{-\frac{1}{2}+1}}{-\frac{1}{2}+1}\right|_{z_1}^{0} = \frac{2D_0^2}{D^2}\sqrt{\frac{1+K_L}{2g}}z_1^{1/2}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{D_0^2}{D^2}\sqrt{\frac{2z_1(1+K_L)}{g}} = \frac{(2.4\text{ m})^2}{(0.1\text{ m})^2}\sqrt{\frac{2(4\text{ m})(1+0.5)}{9.81\text{ m/s}^2}} = \mathbf{637\ s} = \mathbf{10.6\ min}$$

Discussion The effect of the loss coefficient K_L on the draining time can be assessed by setting it equal to zero in the draining time relation. It gives

$$t_{f,\text{zero loss}} = \frac{D_0^2}{D^2}\sqrt{\frac{2z_1}{g}} = \frac{(2.4\text{ m})^2}{(0.1\text{ m})^2}\sqrt{\frac{2(4\text{ m})}{9.81\text{ m/s}^2}} = 520\text{ s} = 8.7\text{ min}$$

Note that the loss coefficient causes the draining time of the tank to increase by (10.6 - 8.7)/10.6 = 0.18 or 18%, which is quite **significant**. Therefore, the loss coefficient should always be considered in draining processes.

8-80

Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere through a long pipe. The initial velocity from the tank and the time required to empty the tank are to be determined.

Assumptions **1** The flow is uniform and incompressible. **2** The draining pipe is horizontal. **3** The flow is turbulent so that the tabulated value of the loss coefficient can be used. **4** The friction factor remains constant (in reality, it changes since the flow velocity and thus the Reynolds number changes). **5** The effect of the kinetic energy correction factor is negligible, so we set $\alpha = 1$.

Properties The loss coefficient is K_L = 0.5 for a sharp-edged entrance. The friction factor of the pipe is given to be 0.015.

Analysis (*a*) We take point 1 at the free surface of the tank, and point 2 at the exit of the pipe. We take the reference level at the centerline of the pipe (z_2 = 0), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

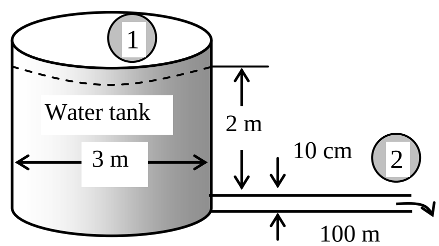
where

$$h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g}=\left(f\frac{L}{D}+K_L\right)\frac{V^2}{2g}$$

since the diameter of the piping system is constant. Substituting and solving for V_2 gives

$$z_1=\alpha_2\frac{V_2^2}{2g}+\left(f\frac{L}{D}+K_L\right)\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{\frac{2gz_1}{\alpha_2+fL/D+K_L}}$$

where α_2 = 1. Noting that initially z_1 = 2 m, the initial velocity is determined to be



$$V_{2,i}=\sqrt{\frac{2gz_1}{1+fL/D+K_L}}=\sqrt{\frac{2(9.81\,\text{m/s}^2)(2\,\text{m})}{1+0.015(100\,\text{m})/(0.1\,\text{m})+0.5}}=\mathbf{1.54\,m/s}$$

The average discharge velocity at any given time, in general, can be expressed as

$$V_2=\sqrt{\frac{2gz}{1+fL/D+K_L}}$$

where z is the water height relative to the center of the orifice at that time.

(*b*) We denote the diameter of the pipe by D, and the diameter of the tank by D_o. The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{\boldsymbol{V}}=A_{pipe}V_2=\frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D+K_L}}$$

Then the amount of water that flows through the pipe during a differential time interval dt is

$$d\boldsymbol{V}=\dot{\boldsymbol{V}}dt=\frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D+K_L}}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV=A_{\tan k}(-dz)=-\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D+K_L}}dt = -\frac{\pi D_0^2}{4}dz \;\rightarrow\; dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2gz}}dz = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained tank) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2g}}\int_{z=z_1}^{0} z^{-1/2}dz \;\rightarrow t_f = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2g}}\left|\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0} = \frac{2D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2g}}z_1^{\frac{1}{2}}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{D_0^2}{D^2}\sqrt{\frac{2z_1(1+fL/D+K_L)}{g}} = \frac{(3\,\text{m})^2}{(0.1\,\text{m})^2}\sqrt{\frac{2(2\,\text{m})[1+(0.015)(100\,\text{m})/(0.1\,\text{m})+0.5]}{9.81\,\text{m/s}^2}} = 2334\,\text{s} = \mathbf{38.9\,min}$$

Discussion It can be shown by setting $L = 0$ that the draining time without the pipe is only 11.7 min. Therefore, the pipe in this case increases the draining time by more than 3 folds.

8-81

Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere through a long pipe equipped with a pump. For a specified initial velocity, the required useful pumping power and the time required to empty the tank are to be determined.

Assumptions **1** The flow is uniform and incompressible. **2** The draining pipe is horizontal. **3** The flow is turbulent so that the tabulated value of the loss coefficient can be used. **4** The friction factor remains constant. **5** The effect of the kinetic energy correction factor is negligible, so we set $\alpha = 1$.

Properties The loss coefficient is K_L = 0.5 for a sharp-edged entrance. The friction factor of the pipe is given to be 0.015. The density of water at 30°C is $\rho = 996$ kg/m^3.

Analysis (*a*) We take point 1 at the free surface of the tank, and point 2 at the exit of the pipe. We take the reference level at the centerline of the orifice ($z_2 = 0$), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{\text{atm}}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1+h_{\text{pump,u}}=\alpha_2\frac{V_2^2}{2g}+h_L$$

where α_2 = 1 and

$$h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g}=\left(f\frac{L}{D}+K_L\right)\frac{V^2}{2g}$$

since the diameter of the piping system is constant. Substituting and noting that the initial discharge velocity is 4 m/s, the required useful pumping head and power are determined to be

$$\dot m=\rho A_c V_2=\rho(\pi D^2/4)V_2=(996\text{ kg/m}^3)[\pi(0.1\text{ m})^2/4](4\text{ m/s})=31.3\text{ kg/m}^3$$

$$h_{\text{pump,u}}=\left(1+f\frac{L}{D}+K_L\right)\frac{V_2^2}{2g}-z_1=\left(1+(0.015)\frac{100\text{ m}}{0.1\text{m}}+0.5\right)\frac{(4\text{ m/s})^2}{2(9.81\text{ m/s}^2)}-(2\text{ m})=11.46\text{ m}$$

$$\dot W_{\text{pump,u}}=\dot V\Delta P=\dot m g h_{\text{pump,u}}=(31.3\text{ kg/s})(9.81\text{ m/s}^2)(11.46\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right)=\mathbf{3.52\ kW}$$

Therefore, the pump must supply 3.52 kW of mechanical energy to water. Note that the shaft power of the pump must be greater than this to account for the pump inefficiency.

(*b*) When the discharge velocity remains constant, the flow rate of water becomes

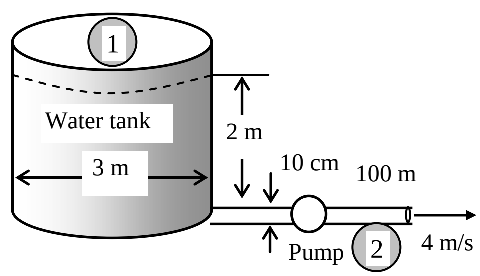
$$\dot V=A_cV_2=(\pi D^2/4)V_2=[\pi(0.1\text{ m})^2/4](4\text{ m/s})=0.03142\text{ m}^3/\text{s}$$

The volume of water in the tank is

$$V=A_{\text{tank}}z_1=(\pi D_0^2/4)z_1=[\pi(3\text{ m})^2/4](2\text{ m})=14.14\text{ m}^3$$

Then the discharge time becomes

$$\Delta t=\frac{V}{\dot V}=\frac{14.14\text{ m}^3}{0.03142\text{ m}^3/\text{s}}=450\text{ s}=\mathbf{7.5\,min}$$



Discussion

1 Note that the pump reduces the discharging time from 38.9 min to 7.5 min. The assumption of constant discharge velocity can be justified on the basis of the pump head being much larger than the elevation head (therefore, the pump will dominate the discharging process). The answer obtained assumes that the elevation head remains constant at 2 m (rather than decreasing to zero eventually), and thus it under predicts the actual discharge time. By an exact analysis, it can be shown that when the effect of the decrease in elevation is considered, the discharge time becomes 468 s = 7.8 min. This is demonstrated below.

2 The required pump head (of water) is 11.46 m, which is more than 10.3 m of water column which corresponds to the atmospheric pressure at sea level. If the pump exit is at 1 atm, then the absolute pressure at pump inlet must be negative (= -1.16 m or – 11.4 kPa), which is impossible. Therefore, the system cannot work if the pump is installed near the pipe exit, and cavitation will occur long before the pipe exit where the pressure drops to 4.2 kPa and thus the pump must be installed close to the pipe entrance. A detailed analysis is given below.

Demonstration 1 for Prob. 8-84 (extra) (the effect of drop in water level on discharge time)

Noting that the water height z in the tank is variable, the average discharge velocity through the pipe at any given time, in general, can be expressed as

$$h_{\text{pump,u}} = \left(1 + f\frac{L}{D} + K_L\right)\frac{V_2^2}{2g} - z \quad \to \quad V_2 = \sqrt{\frac{2g(z + h_{\text{pump,u}})}{1 + fL/D + K_L}}$$

where z is the water height relative to the center of the orifice at that time. We denote the diameter of the pipe by D, and the diameter of the tank by D_0. The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the cross-sectional area of the pipe,

$$\dot{V} = A_{pipe}V_2 = \frac{\pi D^2}{4}\sqrt{\frac{2g(z + h_{\text{pump,u}})}{1 + fL/D + K_L}}$$

Then the amount of water that flows through the orifice during a differential time interval dt is

$$dV = \dot{V}dt = \frac{\pi D^2}{4}\sqrt{\frac{2g(z + h_{\text{pump,u}})}{1 + fL/D + K_L}}dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV = A_{\tan k}(-dz) = -\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{\frac{2g(z + h_{\text{pump,u}})}{1 + fL/D + K_L}}dt = -\frac{\pi D_0^2}{4}dz \quad \to dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1 + fL/D + K_L}{2g}}(z + h_{\text{pump,u}})^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained tank) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1 + fL/D + K_L}{2g}}\int_{z=z_1}^{0}(z + h_{\text{pump,u}})^{-1/2}dz$$

Performing the integration gives

$$t_f = -\frac{D_0^2}{D^2}\sqrt{\frac{1 + fL/D + K_L}{2g}}\left|\frac{(z + h_{\text{pump}})^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0} = \frac{D_0^2}{D^2}\left(\sqrt{\frac{2(z_1 + h_{\text{pump}})(1 + fL/D + K_L)}{g}} - \sqrt{\frac{2h_{\text{pump}}(1 + fL/D + K_L)}{g}}\right)$$

Substituting the values given, the draining time is determined to be

$$t_f = \frac{(3\,\text{m})^2}{(0.1\,\text{m})^2}\left(\sqrt{\frac{2(2 + 11.46\,\text{m})[1 + 0.015 \times 100/0.1 + 0.5]}{9.81\,\text{m/s}^2}} - \sqrt{\frac{2(11.46\,\text{m})[1 + 0.015 \times 100/0.1 + 0.5]}{9.81\,\text{m/s}^2}}\right)$$
$$= 468\,\text{s} = \mathbf{7.8\,min}$$

Demonstration 2 for Prob. 8-84 (on cavitation)

We take the pump as the control volume, with point 1 at the inlet and point 2 at the exit. We assume the pump inlet and outlet diameters to be the same and the elevation difference between the pump inlet and the exit to be negligible. Then we have $z_1 \cong z_2$ and $V_1 \cong V_2$. The pump is located near the pipe exit, and thus the pump exit pressure is equal to the pressure at the pipe exit, which is the atmospheric pressure, $P_2 = P_{atm}$. Also, the can take $h_L = 0$ since the frictional effects and loses in the pump are accounted for by the pump efficiency. Then the energy equation for the pump (in terms of heads) reduces to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad \frac{P_{1,\text{abs}}}{\rho g}+h_{\text{pump,u}}=\frac{P_{\text{atm}}}{\rho g}$$

Solving for P_1 and substituting,

$$P_{1,\text{abs}}=P_{\text{atm}}-\rho g h_{\text{pump,u}}$$

$$=(101.3\text{ kPa})-(996\text{ kg/m}^3)(9.81\text{ m/s}^2)(11.46\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)=\mathbf{-10.7 kPa}$$

which is impossible (absolute pressure cannot be negative). The technical answer to the question is that cavitation **will occur** since the pressure drops below the vapor pressure of 4.246 kPa. The practical answer is that the question is invalid (void) since the system will not work anyway. Therefore, we conclude that the pump must be located **near the beginning**, not the end of the pipe. Note that when doing a cavitation analysis, we must work with the absolute pressures. (If the system were installed as indicated, a water velocity of V = 4 m/s could not be established regardless of how much pump power were applied. This is because the atmospheric air and water elevation heads alone are not sufficient to drive such flow, with the pump restoring pressure after the flow.)

To determine the furthest distance from the tank the pump can be located without allowing cavitation, we assume the pump is located at a distance L^* from the exit, and choose the pump and the discharge portion of the pipe (from the pump to the exit) as the system, and write the energy equation. The energy equation this time will be as above, except that h_L (the pipe losses) must be considered and the pressure at 1 (pipe inlet) is the cavitation pressure, P_1 = 4.246 kPa:

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad \frac{P_{1,\text{abs}}}{\rho g}+h_{\text{pump,u}}=\frac{P_{\text{atm}}}{\rho g}+f\frac{L^*}{D}\frac{V^2}{2g}$$

or

$$f\frac{L^*}{D}\frac{V^2}{2g}=\frac{P_{1,\text{abs}}-P_{\text{atm}}}{\rho g}+h_{\text{pump,u}}$$

Substituting the given values and solving for L^* gives

$$(0.015)\frac{L^*}{0.1\text{ m}}\frac{(4\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=\frac{(4.246-101.3)\text{ kN/m}^2}{(996\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kN}}\right)+(11.46\text{ m}) \quad\rightarrow\quad L^*=12.5\text{ m}$$

Therefore, the pump must be at least 12.5 m from the pipe exit to avoid cavitation at the pump inlet (this is where the lowest pressure occurs in the piping system, and where the cavitation is most likely to occur).

Cavitation onset places an upper limit to the length of the pipe on the suction side. A pipe slightly longer would become vapor bound, and the pump could not pull the suction necessary to sustain the flow. Even if the pipe on the suction side were slightly shorter than 100 – 12.5 = 87.5 m, cavitation can still occur in the pump since the liquid in the pump is usually accelerated at the expense of pressure, and cavitation in the pump could erode and destroy the pump.

Also, over time, scale and other buildup inside the pipe can and will increase the pipe roughness, increasing the friction factor f, and therefore the losses. Buildup also decreases the pipe diameter, which increases pressure drop. Therefore, flow conditions and system performance may change (generally decrease) as the system ages. A new system that marginally misses cavitation may degrade to where cavitation becomes a problem. Proper design avoids these problems, or where cavitation cannot be avoided for some reason, it can at least be anticipated.

8-82

Solution Water is transported to a residential area through concrete pipes, and the idea of lining the interior surfaces of the pipe is being evaluated to reduce frictional losses. The percent increase or decrease in the pumping power requirements as a result of lining the concrete pipes is to be determined.

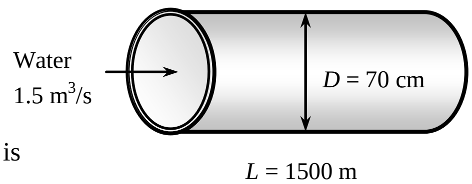
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe line involves no components such as bends, valves, and connectors, and thus no minor losses. **4** The flow is turbulent (to be verified).

Properties The density and kinematic viscosity of water are given to be ρ = 1000 kg/m^3 and ν = 1×10^{-6} m^2/s. The surface roughness is 3 mm for concrete and 0.04 mm for the lining.

Analysis **CASE 1** (concrete pipe, D = 0.70 m). The average velocity and the Reynolds number are:

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{1.5\ \text{m}^3/\text{s}}{\pi (0.70\ \text{m})^2/4} = 3.898\ \text{m/s}$$

$$\text{Re} = \frac{VD}{\nu} = \frac{(3.898\ \text{m/s})(0.70\ \text{m})}{1\times 10^{-6}\ \text{m}^2/s} = 2.728\times 10^6$$



Since Re > 4000, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon / D = \frac{0.003\ \text{m}}{0.7\ \text{m}} = 0.004286$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{0.004286}{3.7} + \frac{2.51}{2.728\times 10^6\sqrt{f}}\right)$$

It gives f = 0.02904. Then the head loss and the required useful power input become

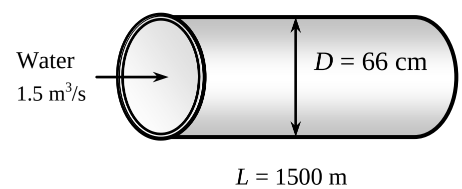
$$h_L = f\frac{L}{D}\frac{V_2^2}{2g} = \left((0.02904)\frac{1500\ \text{m}}{0.70\ \text{m}}\right)\frac{(3.898\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = 48.19\ \text{m} \cong \mathbf{48.2\ m}$$

$$\dot{W}_{\text{pump,u}} = \dot{V}\rho g h_L = (1.5\ \text{m}^3/\text{s})(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(48.19\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1\ \text{kN}\cdot\text{m/s}}\right) = 709.1\ \text{kW} \cong \mathbf{709\ kW}$$

CASE 2 For the case of pipe with lining, D = 0.66 m. Then the average velocity and the Reynolds number are:

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{1.5\ \text{m}^3/\text{s}}{\pi (0.66\ \text{m})^2/4} = 4.384\ \text{m/s}$$

$$\text{Re} = \frac{VD}{\nu} = \frac{(4.384\ \text{m/s})(0.66\ \text{m})}{1\times 10^{-6}\ \text{m}^2/s} = 2.893\times 10^6$$



Since Re > 4000, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon / D = \frac{4\times 10^{-5}\ \text{m}}{0.66\ \text{m}} = 6.061\times 10^{-5}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{6.061\times 10^{-5}}{3.7} + \frac{2.51}{2.893\times 10^6\sqrt{f}}\right)$$

which yields f = 0.01175. Then the head loss and the required useful power input become

$$h_L = f\frac{L}{D}\frac{V_2^2}{2g} = \left((0.01175)\frac{1500\ \text{m}}{0.66\ \text{m}}\right)\frac{(4.384\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = 26.16\ \text{m} \cong \mathbf{26.2\ m}$$

$$\dot{W}_{\text{pump,u}} = \dot{V}\rho g h_L = (1.5\ \text{m}^3/\text{s})(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(26.16\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1\ \text{kN}\cdot\text{m/s}}\right) = 384.9\ \text{kW} \cong \mathbf{385\ kW}$$

The required pumping power changes by (385-709)709 = -0.457; a decrease of 45.7%.

Discussion Note that the pipe head losses and the required pumping power are reduced almost by almost half as result of lining the inner surface of the concrete pipes, despite the reduction in pipe diameter. This confirms the importance of having smooth surfaces in piping.

8-83E

Solution The air discharge rate of a clothes drier with no ducts is given. The flow rate when duct work is attached is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects in the duct are negligible, and thus the flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used. **4** The losses at the vent and its proximity are negligible. **5** The effect of the kinetic energy correction factor on discharge stream is negligible, $\alpha = 1$.

Properties The density of air at 1 atm and 120°F is $\rho = 0.06843$ lbm/ft^3. The roughness of galvanized iron pipe is $\varepsilon = 0.0005$ ft. The loss coefficient is $K_L \approx 0$ for a well-rounded entrance with negligible loss, $K_L = 0.3$ for a flanged 90° smooth bend, and $K_L = 1.0$ for an exit. The friction factor of the duct is given to be 0.019.

Analysis To determine the useful fan power input, we choose point 1 inside the drier sufficiently far from the vent, and point 2 at the exit on the same horizontal level so that $z_1 = z_2$ and $P_1 = P_2$, and the flow velocity at point 1 is negligible ($V_1 = 0$) since it is far from the inlet of the fan. Also, the frictional piping losses between 1 and 2 are negligible, and the only loss involved is due to fan inefficiency. Then the energy equation for a control volume between 1 and 2 reduces to

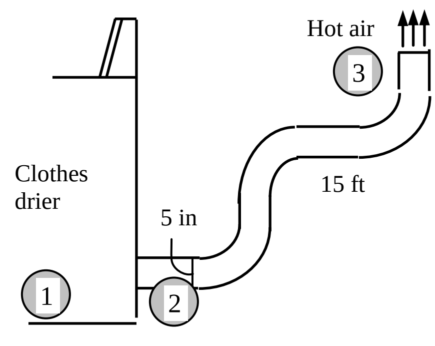
$$\dot{m}\left(\frac{P_1}{\rho}+\alpha_1\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{\text{fan}}=\dot{m}\left(\frac{P_2}{\rho}+\alpha_2\frac{V_2^2}{2}+gz_2\right)+\dot{W}_{\text{turbine}}+\dot{E}_{\text{mech,loss}} \quad\rightarrow\quad \dot{W}_{\text{fan, u}}=\dot{m}\frac{V_2^2}{2} \qquad (1)$$

since $\alpha = 1$ and $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss,fan}} + \dot{E}_{\text{mech loss,piping}}$ and $\dot{W}_{\text{fan, u}} = \dot{W}_{\text{fan}} - \dot{E}_{\text{mech loss,fan}}$.

The average velocity is $V_2 = \dfrac{\dot{V}}{A_c} = \dfrac{\dot{V}}{\pi D^2/4} = \dfrac{1.2\ \text{ft}^3/\text{s}}{\pi(5/12\ \text{ft})^2/4} = 8.80\ \text{ft/s}$

Now we attach the ductwork, and take point 3 to be at the duct exit so that the duct is included in the control volume. The energy equation for this control volume simplifies to

$$\dot{W}_{\text{fan, u}} = \dot{m}\frac{V_3^2}{2}+\dot{m}gh_L \qquad (2)$$



Combining (1) and (2),

$$\rho\dot{V}_2\frac{V_2^2}{2}=\rho\dot{V}_3\frac{V_3^2}{2}+\rho\dot{V}_3 gh_L \quad\rightarrow\quad \dot{V}_2\frac{V_2^2}{2}=\dot{V}_3\frac{V_3^2}{2}+\dot{V}_3 gh_L \qquad (3)$$

where

$$V_3=\frac{\dot{V}_3}{A_c}=\frac{\dot{V}_3}{\pi D^2/4}=\frac{\dot{V}_3\ \text{ft}^3/\text{s}}{\pi(5/12\ \text{ft})^2/4}=7.33\dot{V}_3\ \text{ft/s}$$

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_3^2}{2g}=\left(0.019\frac{15\ \text{ft}}{5/12\ \text{ft}}+3\times0.3+1\right)\frac{V_3^2}{2g}=2.58\frac{V_3^2}{2g}$$

Substituting into Eq. (3),

$$\dot{V}_2\frac{V_2^2}{2}=\dot{V}_3\frac{V_3^2}{2}+\dot{V}_3 g\times2.58\frac{V_3^2}{2g}=\dot{V}_3\frac{(7.33\dot{V}_3)^2}{2}+\dot{V}_3\times2.58\frac{(7.33\dot{V}_3)^2}{2}=96.2\dot{V}_3^3$$

Solving for $\dot{V}_3$ and substituting the numerical values gives

$$\dot{V}_3=\left(\dot{V}_2\frac{V_2^2}{2\times96.2}\right)^{1/3}=\left(1.2\frac{8.80^2}{2\times96.2}\right)^{1/3}=\mathbf{0.78\ ft^3/s}$$

Discussion Note that the flow rate decreased considerably for the same fan power input, as expected. We could also solve this problem by solving for the useful fan power first,

$$\dot{W}_{\text{fan, u}}=\rho\dot{V}_2\frac{V_2^2}{2}=(0.06843\ \text{lbm/ft}^3)(1.2\ \text{ft}^3/\text{s})\frac{(8.80\ \text{ft/s})^2}{2}\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{W}}{0.737\ \text{lbf}\cdot\text{ft/s}}\right)=0.13\ \text{W}$$

Therefore, the fan supplies 0.13 W of useful mechanical power when the drier is running.

8-84

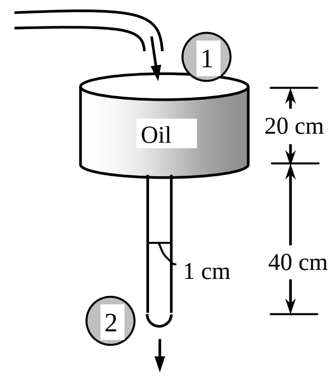
Solution Oil is flowing through a vertical glass funnel which is always maintained full. The flow rate of oil through the funnel and the funnel effectiveness are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed (to be verified). **3** The frictional loses in the cylindrical reservoir are negligible since its diameter is very large and thus the oil velocity is very low.

Properties The density and viscosity of oil at 20°C are ρ = 888.1 kg/m^3 and μ = 0.8374 kg/m·s.

Analysis We take point 1 at the free surface of the oil in the cylindrical reservoir, and point 2 at the exit of the funnel pipe which is also taken as the reference level ($z_2 = 0$). The fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is negligible ($V_1 \cong 0$). For the ideal case of "frictionless flow," the exit velocity is determined from the Bernoulli equation to be

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \qquad \rightarrow \qquad V_2 = V_{2,\max} = \sqrt{2gz_1}$$



Substituting,

$$V_{2,\max} = \sqrt{2gz_1} = \sqrt{2(9.81\,\text{m/s}^2)(0.60\,\text{m})} = 3.431\,\text{m/s}$$

This is the flow velocity for the frictionless case, and thus it is the maximum flow velocity. Then the maximum flow rate and the Reynolds number become

$$\dot{V}_{\max} = V_{2,\max} A_2 = V_{2,\max}(\pi D_2^2/4)$$
$$= (3.431\,\text{m/s})[\pi(0.01\,\text{m})^2/4] = 2.695\times10^{-4}\ \text{m}^3\text{/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(888.1\,\text{kg/m}^3)(3.431\,\text{m/s})(0.01\,\text{m})}{0.8374\,\text{kg/m}\cdot\text{s}} = 36.39$$

which is less than 2300. Therefore, the flow is laminar, as postulated. (Note that in the actual case the velocity and thus the Reynolds number will be even smaller, verifying the flow is always laminar). The entry length in this case is

$$L_h = 0.05\,\text{Re}\,D = 0.05\times36.39\times(0.01\,\text{m}) = 0.018\,\text{m}$$

which is much less than the 0.40 m pipe length. Therefore, the entrance effects can be neglected as postulated.

Noting that the flow through the pipe is laminar and can be assumed to be fully developed, the flow rate can be determined from the appropriate relation with θ = -90° since the flow is downwards in the vertical direction,

$$\dot{V} = \frac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128\mu L}$$

where $\Delta P = P_{\text{pipeinlet}} - P_{\text{pipeexit}} = (P_{\text{atm}} + \rho g h_{\text{cylinder}}) - P_{\text{atm}} = \rho g h_{\text{cylinder}}$ is the pressure difference across the pipe, $L = h_{\text{pipe}}$, and sin θ = sin (-90°) = -1. Substituting, the flow rate is determined to be

$$\dot{V} = \frac{\rho g(h_{\text{cylinder}} + h_{\text{pipe}})\pi D^4}{128\mu L} = \frac{(888.1\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(0.20+0.40\,\text{m})\pi(0.01\,\text{m})^4}{128(0.8374\,\text{kg/m}\cdot\text{s})(0.40\,\text{m})}$$
$$= 3.830\times10^{-6}\ \text{m}^3\text{/s} \cong \mathbf{3.83\times10^{-6}\ m^3/s}$$

Then the "funnel effectiveness" becomes

$$\text{Eff} = \frac{\dot{V}}{\dot{V}_{\max}} = \frac{3.830\times10^{-6}\ \text{m}^3\text{/s}}{2.695\times10^{-4}\ \text{m}^3\text{/s}} = 0.0142 \ \text{ or approximately } \ \mathbf{1.4\%}$$

Discussion Note that the flow is driven by gravity alone, and the actual flow rate is a small fraction of the flow rate that would have occurred if the flow were frictionless.

8-85

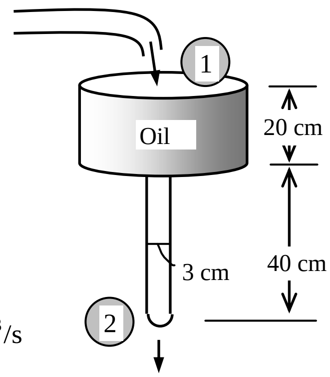
Solution Oil is flowing through a vertical glass funnel which is always maintained full. The flow rate of oil through the funnel and the funnel effectiveness are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed (to be verified). **3** The frictional loses in the cylindrical reservoir are negligible since its diameter is very large and thus the oil velocity is very low.

Properties The density and viscosity of oil at 20°C are ρ = 888.1 kg/m^3 and μ = 0.8374 kg/m·s.

Analysis We take point 1 at the free surface of the oil in the cylindrical reservoir, and point 2 at the exit of the funnel pipe, which is also taken as the reference level ($z_2 = 0$). The fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is negligible ($V_1 \cong 0$). For the ideal case of "frictionless flow," the exit velocity is determined from the Bernoulli equation to be

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad V_2=V_{2,\max}=\sqrt{2gz_1}$$



(*a*) **Case 1**: *Pipe length remains constant at 25 cm, but the pipe diameter is tripled* to D_2 = 3 cm:

Substitution gives

$$V_{2,\max}=\sqrt{2gz_1}=\sqrt{2(9.81\,\text{m/s}^2)(0.60\,\text{m})}=3.431\,\text{m/s}$$

This is the flow velocity for the frictionless case, and thus it is the maximum flow velocity. Then the maximum flow rate and the Reynolds number become

$$\dot{V}_{\max}=V_{2,\max}A_2=V_{2,\max}(\pi D_2^2/4)=(3.431\,\text{m/s})[\pi(0.03\,\text{m})^2/4]=2.425\times10^{-3}\ \text{m}^3/\text{s}$$

$$\text{Re}=\frac{\rho VD}{\mu}=\frac{(888.1\,\text{kg/m}^3)(3.431\,\text{m/s})(0.03\,\text{m})}{0.8374\,\text{kg/m}\cdot\text{s}}=109.2$$

which is less than 2300. Therefore, the flow is laminar. (Note that in the actual case the velocity and thus the Reynolds number will be even smaller, verifying the flow is always laminar). The entry length is

$$L_h=0.05\,\text{Re}\,D=0.05\times109.2\times(0.03\,\text{m})=0.164\,\text{m}$$

which is considerably less than the 0.40 m pipe length. Therefore, the entrance effects can be neglected (with reservation).

Noting that the flow through the pipe is laminar and can be assumed to be fully developed, the flow rate can be determined from the appropriate relation with θ = -90° since the flow is downwards in the vertical direction,

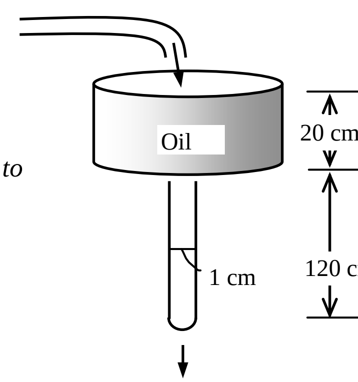
$$\dot{V}=\frac{(\Delta P-\rho gL\sin\theta)\pi D^4}{128\mu L}$$

where $\Delta P=P_{\text{pipeinlet}}-P_{\text{pipeexit}}=(P_{\text{atm}}+\rho gh_{\text{cylinder}})-P_{\text{atm}}=\rho gh_{\text{cylinder}}$ is the pressure difference across the pipe, $L = h_{pipe}$, and sin θ = sin (-90°) = -1. Substituting, the flow rate is determined to be

$$\dot{V}=\frac{\rho g(h_{\text{cylinder}}+h_{\text{pipe}})\pi D^4}{128\mu L}=\frac{(888.1\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(0.20+0.40\,\text{m})\pi(0.03\,\text{m})^4}{128(0.8374\,\text{kg/m}\cdot\text{s})(0.40\,\text{m})}=\mathbf{3.103\times10^{-4}\ m^3/s}$$

Then the "funnel effectiveness" becomes

$$\text{Eff}=\frac{\dot{V}}{\dot{V}_{\max}}=\frac{3.103\times10^{-4}\ \text{m}^3/\text{s}}{2.425\times10^{-3}\ \text{m}^3/\text{s}}=0.1279\quad\text{or}\quad \mathbf{12.8\%}$$



(*b*) **Case 2**: *Pipe diameter remains constant at 1 cm, but the pipe length is tripled to L = 120 cm*:

Substitution gives

$$V_{2,\max}=\sqrt{2gz_1}=\sqrt{2(9.81\,\text{m/s}^2)(1.40\,\text{m})}=5.241\,\text{m/s}$$

This is the flow velocity for the frictionless case, and thus it is the maximum flow velocity. Then the maximum flow rate and the Reynolds number become

$$\dot{V}_{max} = V_{2,max} A_2 = V_{2,max}(\pi D_2^2/4) = (5.241\,\text{m/s})[\pi(0.01\,\text{m})^2/4] = 4.116\times10^{-4}\ \text{m}^3/s$$

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(888.1\,\text{kg/m}^3)(5.241\,\text{m/s})(0.01\,\text{m})}{0.8374\,\text{kg/m}\cdot\text{s}} = 55.58$$

which is less than 2300. Therefore, the flow is laminar. (Note that in the actual case the velocity and thus the Reynolds number will be even smaller, verifying the flow is always laminar). The entry length is

$$L_h = 0.05\,\text{Re}\,D = 0.05\times55.58\times(0.01\,\text{m}) = 0.028\,\text{m}$$

which is much less than the 1.20 m pipe length. Therefore, the entrance effects can be neglected.

Noting that the flow through the pipe is laminar and can be assumed to be fully developed, the flow rate can be determined from the appropriate relation with θ = -90° since the flow is downwards in the vertical direction,

$$\dot{V} = \frac{(\Delta P - \rho g L\sin\theta)\pi D^4}{128\mu L}$$

where $\Delta P = P_{\text{pipeinlet}} - P_{\text{pipeexit}} = (P_{\text{atm}} + \rho g h_{\text{cylinder}}) - P_{\text{atm}} = \rho g h_{\text{cylinder}}$ is the pressure difference across the pipe, $L = h_{\text{pipe}}$, and sin θ = sin (-90°) = -1. Substituting, the flow rate is determined to be

$$\dot{V} = \frac{\rho g(h_{\text{cylinder}} + h_{\text{pipe}})\pi D^4}{128\mu L} = \frac{(888.1\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(0.20+1.20\,\text{m})\pi(0.01\,\text{m})^4}{128(0.8374\,\text{kg/m}\cdot\text{s})(1.20\,\text{m})}$$

$$= 2.979\times10^{-6}\ \text{m}^3/\text{s} \cong \mathbf{2.98\times10^{-6}\ m^3/s}$$

Then the "funnel effectiveness" becomes

$$\text{Eff} = \frac{\dot{V}}{\dot{V}_{max}} = \frac{2.979\times10^{-6}\ \text{m}^3/\text{s}}{4.116\times10^{-4}\ \text{m}^3/\text{s}} = 0.00724 \quad \text{or} \quad \mathbf{0.72\%}$$

Discussion Note that the funnel effectiveness increases as the pipe diameter is increased, and decreases as the pipe length is increased. This is because the frictional losses are proportional to the length but inversely proportional to the diameter of the flow sections.

8-86

Solution Water is drained from a large reservoir through two pipes connected in series. The discharge rate of water from the reservoir is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The pipes are horizontal. **3** The entrance effects are negligible, and thus the flow is fully developed. **4** The flow is turbulent so that the tabulated value of the loss coefficients can be used. **5** The pipes involve no components such as bends, valves, and other connectors. **6** The piping section involves no work devices such as pumps and turbines. **7** The reservoir is open to the atmosphere so that the pressure is atmospheric pressure at the free surface. **8** The water level in the reservoir remains constant. **9** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

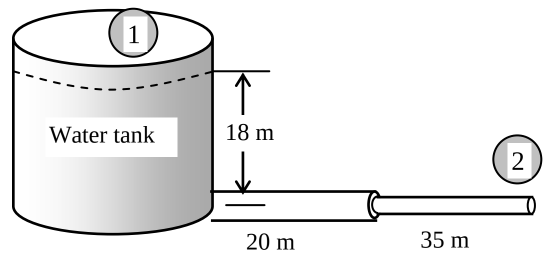
Properties The density and dynamic viscosity of water at 15°C are ρ = 999.1 kg/m^3 and μ = 1.138×10^{-3} kg/m·s, respectively. The loss coefficient is K_L = 0.5 for a sharp-edged entrance, and it is 0.46 for the sudden contraction, corresponding to $d^2/D^2 = 4^2/10^2 = 0.16$. The pipes are made of plastic and thus they are smooth, $\varepsilon = 0$.

Analysis We take point 1 at the free surface of the reservoir, and point 2 at the exit of the pipe, which is also taken to be the reference level (z_2 = 0). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$), the fluid level in the reservoir is constant (V_1 = 0), and that there are no work devices such as pumps and turbines, the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

where α_2 = 1. Substituting,

$$18\text{ m}=\frac{V_2^2}{2(9.81\text{m/s}^2)}+h_L \qquad (1)$$



where

$$h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\sum\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g}$$

Note that the diameters of the two pipes, and thus the flow velocities through them are different. Denoting the first pipe by 1 and the second pipe by 2, and using conservation of mass, the velocity in the first pipe can be expressed in terms of V_2 as

$$\dot m_1=\dot m_2 \quad\rightarrow\quad \rho V_1A_1=\rho V_2A_2 \quad\rightarrow\quad V_1=\frac{A_2}{A_1}V_2=\frac{D_2^2}{D_1^2}V_2=\frac{(4\text{ cm})^2}{(10\text{ cm})^2}V_2 \quad\rightarrow\quad V_1=0.16V_2 \qquad (2)$$

Then the head loss can be expressed as

$$h_L=\left(f_1\frac{L_1}{D_1}+K_{L,\text{entrance}}\right)\frac{V_1^2}{2g}+\left(f_2\frac{L_2}{D_2}+K_{L,\text{contraction}}\right)\frac{V_2^2}{2g}$$

or

$$h_L=\left(f_1\frac{20\text{ m}}{0.10\text{ m}}+0.5\right)\frac{V_1^2}{2(9.81\text{m/s}^2)}+\left(f_2\frac{35\text{ m}}{0.04\text{ m}}+0.46\right)\frac{V_2^2}{2(9.81\text{m/s}^2)} \qquad (3)$$

The flow rate, the Reynolds number, and the friction factor are expressed as

$$\dot V=V_2A_2=V_2(\pi D_2^2/4) \quad\rightarrow\quad \dot V=V_2[\pi(0.04\text{ m})^2/4] \qquad (4)$$

$$\text{Re}_1=\frac{\rho V_1D_1}{\mu} \quad\rightarrow\quad \text{Re}_1=\frac{(999.1\text{ kg/m}^3)\mathbf{V}_1(0.10\text{ m})}{1.138\times10^{-3}\text{ kg/m}\cdot s} \qquad (5)$$

$$\text{Re}_2=\frac{\rho V_2D_2}{\mu} \quad\rightarrow\quad \text{Re}_2=\frac{(999.1\text{ kg/m}^3)\mathbf{V}_2(0.04\text{ m})}{1.138\times10^{-3}\text{ kg/m}\cdot s} \qquad (6)$$

$$\frac{1}{\sqrt{f_1}}=-2.0\log\left(\frac{\varepsilon/D_1}{3.7}+\frac{2.51}{\text{Re}_1\sqrt{f_1}}\right) \quad\rightarrow\quad \frac{1}{\sqrt{f_1}}=-2.0\log\left(0+\frac{2.51}{\text{Re}_1\sqrt{f_1}}\right) \qquad (7)$$

$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{\varepsilon / D_2}{3.7} + \frac{2.51}{\mathrm{Re}_2\sqrt{f_2}}\right) \rightarrow \frac{1}{\sqrt{f_2}} = -2.0\log\left(0 + \frac{2.51}{\mathrm{Re}_2\sqrt{f_2}}\right) \qquad (8)$$

This is a system of 8 equations in 8 unknowns, and their simultaneous solution by an equation solver gives

$\dot{V} = \mathbf{0.00595\ m^3/s}$, V_1 = 0.757 m/s, V_2 = 4.73 m/s, $h_L = h_{L1} + h_{L2}$ = 0.13 + 16.73 = 16.86 m,

Re_1 = 66,500, Re_2 = 166,200, f_1 = 0.0196, f_2 = 0.0162

Note that Re > 4000 for both pipes, and thus the assumption of turbulent flow is valid.

Discussion This problem can also be solved by using an iterative approach by assuming an exit velocity, but it will be very time consuming. Equation solvers such as EES are invaluable for this kind of problems.

8-87E

Solution The flow rate through a piping system between a river and a storage tank is given. The power input to the pump is to be determined.

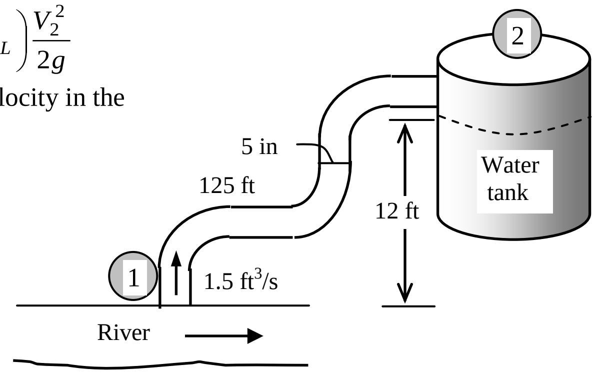
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** The elevation difference between the free surfaces of the tank and the river remains constant. **5** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 70°F are ρ = 62.30 lbm/ft^3 and μ = 2.360 lbm/ft·h = 6.556×10^{-4} lbm/ft·s. The roughness of galvanized iron pipe is ε = 0.0005 ft.

Analysis The piping system involves 125 ft of 5-in diameter piping, an entrance with negligible loses, 3 standard flanged 90° smooth elbows (K_L = 0.3 each), and a sharp-edged exit (K_L = 1.0). We choose points 1 and 2 at the free surfaces of the river and the tank, respectively. We note that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{\text{atm}}$), and the fluid velocity is 6 ft/s at point 1 and zero at point 2 (V_1 = 6 ft/s and V_2 =0). We take the free surface of the river as the reference level (z_1 = 0). Then the energy equation for a control volume between these two points simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad \alpha_1\frac{V_1^2}{2g}+h_{\text{pump,u}}=z_2+h_L$$

where α_1 = 1 and $h_L = h_{L,\text{total}} = h_{L,\text{major}} + h_{L,\text{minor}} = \left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$



since the diameter of the piping system is constant. The average velocity in the pipe and the Reynolds number are

$$V=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4}=\frac{1.5\text{ ft}^3/\text{s}}{\pi(5/12\text{ ft})^2/4}=11.0\text{ ft/s}$$

$$\text{Re}=\frac{\rho VD}{\mu}=\frac{(62.3\text{ lbm/ft}^3)(11.0\text{ ft/s})(5/12\text{ ft})}{6.556\times10^{-4}\text{ lbm/ft}\cdot\text{s}}=435{,}500$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{0.0005\text{ ft}}{5/12\text{ ft}}=0.0012$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt f}\right)\quad\rightarrow\quad\frac{1}{\sqrt f}=-2.0\log\left(\frac{0.0012}{3.7}+\frac{2.51}{435{,}500\sqrt f}\right)$$

It gives f = 0.0211. The sum of the loss coefficients is

$$\sum K_L=K_{L,\text{entrance}}+3K_{L,\text{elbow}}+K_{L,\text{exit}}=0+3\times0.3+1.0=1.9$$

Then the total head loss becomes

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g}=\left((0.0211)\frac{125\text{ ft}}{5/12\text{ ft}}+1.90\right)\frac{(11.0\text{ ft/s})^2}{2(32.2\text{ ft/s}^2)}=15.5\text{ ft}$$

The useful pump head input and the required power input to the pump are

$$h_{\text{pump,u}}=z_2+h_L-\frac{V_1^2}{2g}=12+15.5-\frac{(6\text{ ft/s})^2}{2(32.2\text{ ft/s}^2)}=26.9\text{ ft}$$

$$\dot W_{\text{pump}}=\frac{\dot W_{\text{pump, u}}}{\eta_{\text{pump}}}=\frac{\dot V\rho g h_{\text{pump, u}}}{\eta_{\text{pump}}}$$

$$=\frac{(1.5\text{ ft}^3/\text{s})(62.30\text{ lbm/ft}^3)(32.2\text{ ft/s}^2)(26.9\text{ ft})}{0.70}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\text{ kW}}{737\text{ lbf}\cdot\text{ft/s}}\right)=\mathbf{4.87\text{ kW}}$$

Therefore, 4.87 kW of electric power must be supplied to the pump.

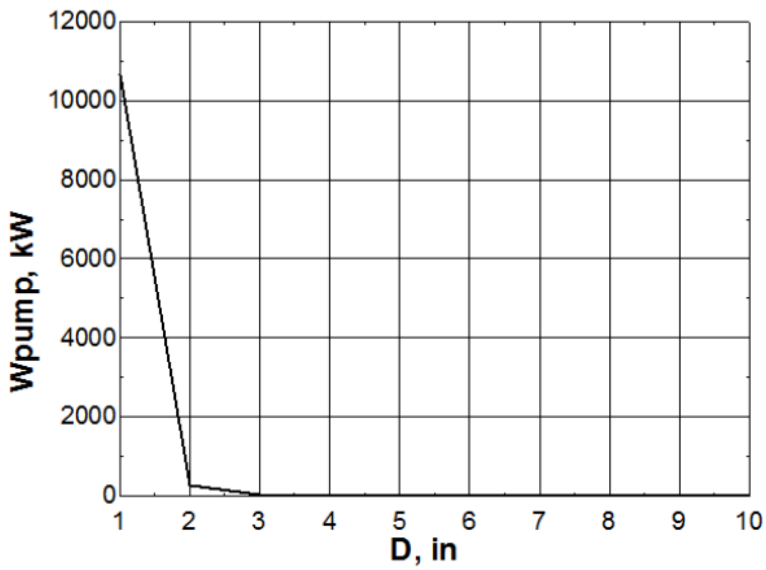
Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.0211, which is identical to the calculated value. The friction coefficient would drop to 0.0135 if smooth pipes were used. Note that fL/D = 6.3 in this case, which is about 3 times the total minor loss coefficient of 1.9. Therefore, the frictional losses in the pipe dominate the minor losses, but the minor losses are still significant.

8-88E

Solution In the previous problem, the effect of the pipe diameter on pumping power for the same constant flow rate is to be investigated by varying the pipe diameter from 1 in to 10 in in increments of 1 in.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
g=32.2
L= 125
D=Dinch/12
z2= 12
rho=62.30
nu=mu/rho
mu=0.0006556
eff=0.70
Re=V2*D/nu
A=pi*(D^2)/4
V2=Vdot/A
Vdot= 1.5
V1=6
eps1=0.0005
rf1=eps1/D
1/sqrt(f1)=-2*log10(rf1/3.7+2.51/(Re*sqrt(f1)))
KL= 1.9
HL=(f1*(L/D)+KL)*(V2^2/(2*g))
hpump=z2+HL-V1^2/(2*32.2)
Wpump=(Vdot*rho*hpump)/eff/737
```



EES Hint: You may need to set the initial guess for variable f1 as 0.02 and Re as 1000 or something reasonable in order to avoid a square root of a negative number in the EES iterations.

D, in	W_{pump}, kW	V, ft/s	Re
1	2.178E+06	275.02	10667.48
2	1.089E+06	68.75	289.54
3	7.260E+05	30.56	38.15
4	5.445E+05	17.19	10.55
5	4.356E+05	11.00	4.88
6	3.630E+05	7.64	3.22
7	3.111E+05	5.61	2.62
8	2.722E+05	4.30	2.36
9	2.420E+05	3.40	2.24
10	2.178E+05	2.75	2.17

Discussion We see that the required pump power decreases very rapidly as pipe diameter increases. This is due to the significant decrease in irreversible head loss in larger diameter pipes. The Reynolds number drops into the laminar region, which makes these calculations invalid.

8-89

Solution A solar heated water tank is to be used for showers using gravity driven flow. For a specified flow rate, the elevation of the water level in the tank relative to showerhead is to be determined.

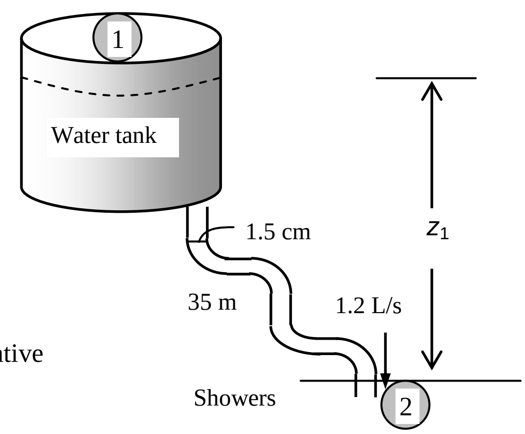
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** The elevation difference between the free surface of water in the tank and the shower head remains constant. **5** There are no pumps or turbines in the piping system. **6** The losses at the entrance and at the showerhead are said to be negligible. **7** The water tank is open to the atmosphere. **8** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 40°C are ρ = 992.1 kg/m^3 and μ = 0.653×10^{-3} kg/m·s, respectively. The loss coefficient is K_L = 0.5 for a sharp-edged entrance. The roughness of galvanized iron pipe is ε = 0.00015 m.

Analysis The piping system involves 20 m of 1.5-cm diameter piping, an entrance with negligible loss, 4 miter bends (90°) without vanes (K_L = 1.1 each), and a wide open globe valve (K_L = 10). We choose point 1 at the free surface of water in the tank, and point 2 at the shower exit, which is also taken to be the reference level (z_2 = 0). The fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$), and V_1 = 0. Then the energy equation for a control volume between these two points simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

where $h_L = h_{L,\text{total}} = h_{L,\text{major}} + h_{L,\text{minor}} = \left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$



since the diameter of the piping system is constant. The average velocity in the pipe and the Reynolds number are

$$V_2=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4}=\frac{0.0012\,\text{m}^3/\text{s}}{\pi(0.015\,\text{m})^2/4}=6.791\,\text{m/s}$$

$$\text{Re}=\frac{\rho V_2 D}{\mu}=\frac{(992.1\,\text{kg/m}^3)(6.791\,\text{m/s})(0.015\,\text{m})}{0.653\times10^{-3}\,\text{kg/m}\cdot\text{s}}=154{,}750$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{0.00015\,\text{m}}{0.015\,\text{m}}=0.01$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}}=-2.0\log\left(\frac{0.01}{3.7}+\frac{2.51}{154{,}750\sqrt{f}}\right)$$

It gives f = 0.03829. The sum of the loss coefficients is

$$\sum K_L = K_{L,\text{entrance}}+4K_{L,\text{elbow}}+K_{L,\text{valve}}+K_{L,\text{exit}}=0+4\times1.1+10+0=14.4$$

Note that we do not consider the exit loss unless the exit velocity is dissipated within the system considered (in this case it is not). Then the total head loss and the elevation of the source become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.03829)\frac{35\,\text{m}}{0.015\,\text{m}}+14.4\right)\frac{(6.791\,\text{m/s})^2}{2(9.81\,\text{m/s}^2)}=243.9\,\text{m}$$

$$z_1=\alpha_2\frac{V_2^2}{2g}+h_L=(1)\frac{(6.791\,\text{m/s})^2}{2(9.81\,\text{m/s}^2)}+243.9\,\text{m}=\mathbf{246\,m}$$

since α_2 = 1. Therefore, the free surface of the tank must be 246 m above the shower exit to ensure water flow at the specified rate.

Discussion We neglected the minor loss associated with the shower head. In reality, this loss is most likely significant.

8-90

Solution The flow rate through a piping system connecting two water reservoirs with the same water level is given. The absolute pressure in the pressurized reservoir is to be determined.

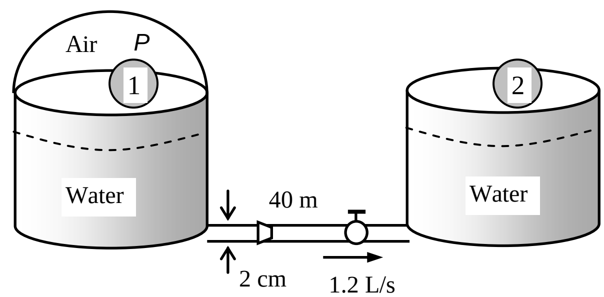
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** There are no pumps or turbines in the piping system.

Properties The density and dynamic viscosity of water at 10°C are ρ = 999.7 kg/m^3 and μ = 1.307×10^{-3} kg/m·s. The loss coefficient is K_L = 0.5 for a sharp-edged entrance, K_L = 2 for swing check valve, K_L = 0.2 for the fully open gate valve, and K_L = 1 for the exit. The roughness of cast iron pipe is ε = 0.00026 m.

Analysis We choose points 1 and 2 at the free surfaces of the two reservoirs. We note that the fluid velocities at both points are zero ($V_1 = V_2 = 0$), the fluid at point 2 is open to the atmosphere (and thus $P_2 = P_{atm}$), both points are at the same level ($z_1 = z_2$). Then the energy equation for a control volume between these two points simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad \frac{P_1}{\rho g}=\frac{P_{\text{atm}}}{\rho g}+h_L \quad\rightarrow\quad P_1=P_{\text{atm}}+\rho g h_L$$

where $h_L = h_{L,\text{total}} = h_{L,\text{major}} + h_{L,\text{minor}} = \left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$



since the diameter of the piping system is constant. The average flow velocity and the Reynolds number are

$$V_2=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4}=\frac{0.0012\ \text{m}^3/\text{s}}{\pi(0.02\ \text{m})^2/4}=3.82\ \text{m/s}$$

$$\text{Re}=\frac{\rho V_2 D}{\mu}=\frac{(999.7\ \text{kg/m}^3)(3.82\ \text{m/s})(0.02\ \text{m})}{1.307\times10^{-3}\ \text{kg/m}\cdot\text{s}}=58{,}400$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{0.00026\ \text{m}}{0.02\ \text{m}}=0.013$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt f}\right)\quad\rightarrow\quad\frac{1}{\sqrt f}=-2.0\log\left(\frac{0.013}{3.7}+\frac{2.51}{58{,}400\sqrt f}\right)$$

It gives f = 0.0424. The sum of the loss coefficients is

$$\sum K_L = K_{L,\text{entrance}}+K_{L,\text{check valve}}+K_{L,\text{gate valve}}+K_{L,\text{exit}}=0.5+2+0.2+1=3.7$$

Then the total head loss becomes

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.0424)\frac{40\ \text{m}}{0.02\ \text{m}}+3.7\right)\frac{(3.82\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)}=65.8\ \text{m}$$

Substituting,

$$P_1=P_{\text{atm}}+\rho g h_L=(88\ \text{kPa})+(999.7\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(65.8\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right)=\mathbf{733\ kPa}$$

Discussion The absolute pressure above the first reservoir must be 734 kPa, which is quite high. Note that the minor losses in this case are negligible (about 4% of total losses). Also, the friction factor could be determined easily from the explicit Haaland relation (it gives the same result, 0.0424). The friction coefficient would drop to 0.0202 if smooth pipes were used.

8-91

Solution A tanker is to be filled with fuel oil from an underground reservoir using a plastic hose. The required power input to the pump is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** Fuel oil level remains constant. **5** Reservoir is open to the atmosphere.

Properties The density and dynamic viscosity of fuel oil are given to be ρ = 920 kg/m^3 and μ = 0.045 kg/m·s. The loss coefficient is K_L = 0.12 for a slightly-rounded entrance and K_L = 0.3 for a 90° smooth bend (flanged). The plastic pipe is smooth and thus ε = 0. The kinetic energy correction factor at hose discharge is given to be α = 1.05.

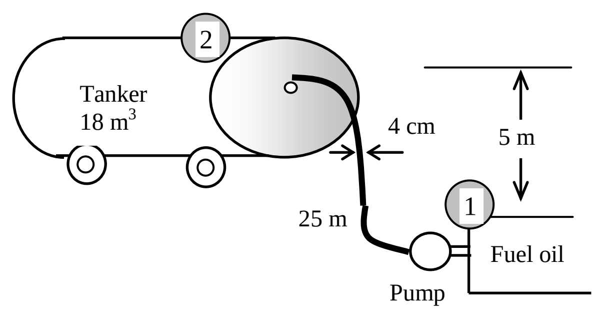
Analysis We choose point 1 at the free surface of oil in the reservoir and point 2 at the exit of the hose in the tanker. We note the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and the fluid velocity at point 1 is zero (V_1 = 0). We take the free surface of the reservoir as the reference level (z_1 = 0). Then the energy equation for a control volume between these two points simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=\alpha_2\frac{V_2^2}{2g}+z_2+h_L$$

where

$$h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$$

since the diameter of the piping system is constant. The flow rate is determined from the requirement that the tanker must be filled in 30 min,



$$\dot{V}=\frac{V_{\text{tanker}}}{\Delta t}=\frac{18\text{ m}^3}{(30\times 60\text{ s})}=0.01\text{ m}^3/\text{s}$$

Then the average velocity in the pipe and the Reynolds number become

$$V_2=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4}=\frac{0.01\text{ m}^3/\text{s}}{\pi(0.04\text{ m})^2/4}=7.958\text{ m/s}$$

$$\text{Re}=\frac{\rho V_2 D}{\mu}=\frac{(920\text{ kg/m}^3)(7.958\text{ m/s})(0.04\text{ m})}{0.045\text{ kg/m}\cdot\text{s}}=6508$$

which is greater than 4000. Therefore, the flow is turbulent. The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation,

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\quad\rightarrow\quad\frac{1}{\sqrt{f}}=-2.0\log\left(0+\frac{2.51}{6508\sqrt{f}}\right)$$

It gives f = 0.0347. The sum of the loss coefficients is

$$\sum K_L=K_{L,\text{entrance}}+2K_{L,\text{bend}}=0.12+2\times 0.3=0.72$$

Note that we do not consider the exit loss unless the exit velocity is dissipated within the system (in this case it is not). Then the total head loss, the useful pump head, and the required pumping power become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.0347)\frac{25\text{ m}}{0.04\text{ m}}+0.72\right)\frac{(7.958\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=72.33\text{ m}$$

$$h_{\text{pump,u}}=\frac{V_2^2}{2g}+z_2+h_L=1.05\frac{(7.958\text{ m/s})^2}{2(9.81\text{ m/s}^2)}+5\text{ m}+73.33\text{ m}=80.72\text{ m}$$

$$\dot{W}_{\text{pump}}=\frac{\dot{V}\rho g h_{\text{pump,u}}}{\eta_{\text{pump}}}=\frac{(0.01\text{ m}^3/\text{s})(920\text{ kg/m}^3)(9.81\text{ m/s}^2)(80.72\text{ m})}{0.82}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right)=\mathbf{8.88\,kW}$$

Discussion Note that the minor losses in this case are negligible (0.72/21.69 = 0.033 or about 3% of total losses). Also, the friction factor could be determined easily from the Haaland relation (it gives 0.0349).

8-92

Solution Two pipes of identical length and material are connected in parallel. The diameter of one of the pipes is twice the diameter of the other. The ratio of the flow rates in the two pipes is to be determined

Assumptions **1** The flow is steady and incompressible. **2** The friction factor is given to be the same for both pipes. **3** The minor losses are negligible.

Analysis When two pipes are parallel in a piping system, the head loss for each pipe must be same. When the minor losses are disregarded, the head loss for fully developed flow in a pipe of length L and diameter D can be expressed as

$$h_L = f\frac{L}{D}\frac{V^2}{2g} = f\frac{L}{D}\frac{1}{2g}\left(\frac{\dot{V}}{A_c}\right)^2 = f\frac{L}{D}\frac{1}{2g}\left(\frac{\dot{V}}{\pi D^2/4}\right)^2 = 8f\frac{L}{D}\frac{1}{g}\frac{\dot{V}^2}{\pi^2 D^4} = 8f\frac{L}{g\pi^2}\frac{\dot{V}^2}{D^5}$$

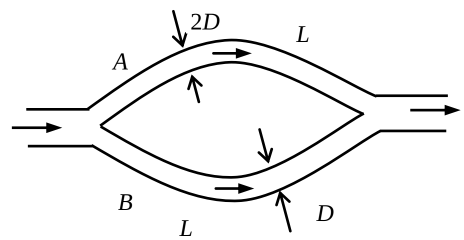
Solving for the flow rate gives

$$\dot{V} = \sqrt{\frac{\pi^2 h_L g}{8fL}}D^{2.5} = kD^{2.5} \qquad (k = \text{constant of proportionality})$$

When the pipe length, friction factor, and the head loss is constant, which is the case here for parallel connection, the flow rate becomes proportional to the 2.5th power of diameter. Therefore, when the diameter is doubled, the flow rate will increase by a factor of $2^{2.5} = 5.66$ since

If $\qquad \dot{V}_A = kD_A^{2.5}$

Then $\qquad \dot{V}_B = kD_B^{2.5} = k(2D_A)^{2.5} = 2^{2.5}kD_A^{2.5} = 2^{2.5}\dot{V}_A = 5.66\dot{V}_A$



Therefore, the ratio of the flow rates in the two pipes is **5.66**.

Discussion The relationship of flow rate to pipe diameter is not linear or even quadratic.

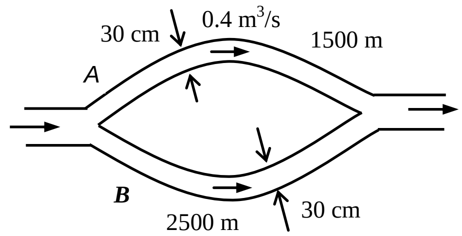
8-93

Solution Cast iron piping of a water distribution system involves a parallel section with identical diameters but different lengths. The flow rate through one of the pipes is given, and the flow rate through the other pipe is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible. **4** The flow is fully turbulent and thus the friction factor is independent of the Reynolds number (to be verified).

Properties The density and dynamic viscosity of water at 15°C are ρ = 999.1 kg/m^3 and μ = 1.138×10^{-3} kg/m·s. The roughness of cast iron pipe is ε = 0.00026 m.

Analysis The average velocity in pipe A is



$$V_A = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.4\ \text{m}^3/\text{s}}{\pi(0.30\ \text{m})^2/4} = 5.659\ \text{m/s}$$

When two pipes are parallel in a piping system, the head loss for each pipe must be same. When the minor losses are disregarded, the head loss for fully developed flow in a pipe of length L and diameter D is

$$h_L = f\frac{L}{D}\frac{V^2}{2g}$$

Writing this for both pipes and setting them equal to each other, and noting that $D_A = D_B$ (given) and $f_A = f_B$ (to be verified) gives

$$f_A\frac{L_A}{D_A}\frac{V_A^2}{2g} = f_B\frac{L_B}{D_B}\frac{V_B^2}{2g} \quad\rightarrow\quad V_B = V_A\sqrt{\frac{L_A}{L_B}} = (5.659\ \text{m/s})\sqrt{\frac{1500\ \text{m}}{2500\ \text{m}}} = 4.383\ \text{m/s}$$

Then the flow rate in pipe B becomes

$$\dot{V}_B = A_c V_B = [\pi D^2/4]V_B = [\pi(0.30\ \text{m})^2/4](4.383\ \text{m/s}) = \mathbf{0.310\ m^3/s}$$

***Proof** that flow is fully turbulent and thus friction factor is independent of Reynolds number:*

The velocity in pipe B is lower. Therefore, if the flow is fully turbulent in pipe B, then it is also fully turbulent in pipe A. The Reynolds number in pipe B is

$$\text{Re}_B = \frac{\rho V_B D}{\mu} = \frac{(999.1\ \text{kg/m}^3)(3.267\ \text{m/s})(0.30\ \text{m})}{1.138\times10^{-3}\ \text{kg/m}\cdot\text{s}} = 0.860\times10^6$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{0.00026\ \text{m}}{0.30\ \text{m}} = 0.00087$$

From Moody's chart, we observe that for a relative roughness of 0.00087, the flow is fully turbulent for Reynolds number greater than about 10^6. Therefore, the flow in both pipes is fully turbulent, and thus the assumption that the friction factor is the same for both pipes is valid.

Discussion Note that the flow rate in pipe B is less than the flow rate in pipe A because of the larger losses due to the larger length.

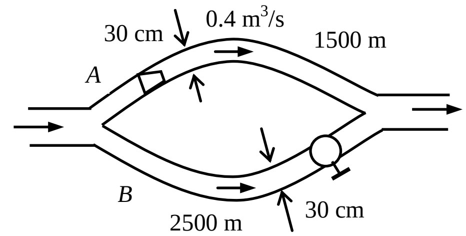
8-94

Solution Cast iron piping of a water distribution system involves a parallel section with identical diameters but different lengths and different valves. The flow rate through one of the pipes is given, and the flow rate through the other pipe is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses other than those for the valves are negligible.

Properties The density and dynamic viscosity of water at 15°C are ρ = 999.1 kg/m^3 and μ = 1.138×10^{-3} kg/m·s. The roughness of cast iron pipe is ε = 0.00026 m.

Analysis For pipe *A*, the average velocity and the Reynolds number are



$$V_A = \frac{\dot{V}_A}{A_c} = \frac{\dot{V}_A}{\pi D^2/4} = \frac{0.4\ \text{m}^3/\text{s}}{\pi(0.30\ \text{m})^2/4} = 5.659\ \text{m/s}$$

$$\text{Re}_A = \frac{\rho V_A D}{\mu} = \frac{(999.1\ \text{kg/m}^3)(5.659\ \text{m/s})(0.30\ \text{m})}{1.138\times10^{-3}\ \text{kg/m}\cdot\text{s}} = 1.49\times10^6$$

The relative roughness of the pipe is

$$\varepsilon/D = \frac{0.00026\ \text{m}}{0.30\ \text{m}} = 8.667\times10^{-4}$$

The friction factor corresponding to this relative roughness and the Reynolds number can simply be determined from the Moody chart. To avoid the reading error, we determine it from the Colebrook equation

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{8.667\times10^{-4}}{3.7} + \frac{2.51}{1.49\times10^6\sqrt{f}}\right)$$

It gives f = 0.0192. Then the total head loss in pipe *A* becomes

$$h_{L,A} = \left(f\frac{L_A}{D} + K_L\right)\frac{V_A^2}{2g} = \left((0.0192)\frac{1500\ \text{m}}{0.30\ \text{m}} + 2.1\right)\frac{(5.659\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = 160.1\ \text{m}$$

When two pipes are parallel in a piping system, the head loss for each pipe must be same. Therefore, the head loss for pipe *B* must also be 160.1 m. Then the average velocity in pipe *B* and the flow rate become

$$h_{L,B} = \left(f\frac{L_B}{D} + K_L\right)\frac{V_B^2}{2g} \rightarrow 160.1\ \text{m} = \left((0.0192)\frac{2500\ \text{m}}{0.30\ \text{m}} + 10\right)\frac{V_B^2}{2(9.81\ \text{m/s}^2)} \rightarrow V_B = 4.299\ \text{m/s}$$

$$\dot{V}_B = A_c V_B = [\pi D^2/4]V_B = [\pi(0.30\ \text{m})^2/4](4.299\ \text{m/s}) = \mathbf{0.304\ m^3/s}$$

Discussion Note that the flow rate in pipe *B* decreases slightly (from 0.310 to 0.304 m^3/s) due to the larger minor loss in that pipe. Also, minor losses constitute just a few percent of the total loss, and they can be neglected if great accuracy is not required.

8-95

Solution Geothermal water is supplied to a city through stainless steel pipes at a specified rate. The electric power consumption and its daily cost are to be determined, and it is to be assessed if the frictional heating during flow can make up for the temperature drop caused by heat loss.

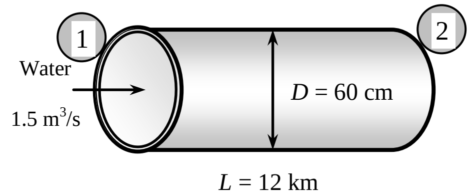
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses. **4** The geothermal well and the city are at about the same elevation. **5** The properties of geothermal water are the same as fresh water. **6** The fluid pressures at the wellhead and the arrival point in the city are the same.

Properties The properties of water at 110°C are $\rho = 950.6$ kg/m^3, $\mu = 0.255\times10^{-3}$ kg/m·s, and $C_p = 4.229$ kJ/kg·°C. The roughness of stainless steel pipes is 2×10^{-6} m.

Analysis (*a*) We take point 1 at the well-head of geothermal resource and point 2 at the final point of delivery at the city, and the entire piping system as the control volume. Both points are at the same elevation ($z_2 = z_2$) and the same velocity ($V_1 = V_2$) since the pipe diameter is constant, and the same pressure ($P_1 = P_2$). Then the energy equation for this control volume simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=h_L$$

That is, the pumping power is to be used to overcome the head losses due to friction in flow. The average velocity and the Reynolds number are



$$V=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4}=\frac{1.5\text{ m}^3/\text{s}}{\pi(0.60\text{ m})^2/4}=5.305\text{ m/s}$$

$$\text{Re}=\frac{\rho VD}{\mu}=\frac{(950.6\text{ kg/m}^3)(5.305\text{ m/s})(0.60\text{ m})}{0.255\times10^{-3}\text{ kg/m}\cdot\text{s}}=1.187\times10^7$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{2\times10^{-6}\text{ m}}{0.60\text{ m}}=3.33\times10^{-6}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\quad\rightarrow\quad\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{3.33\times10^{-6}}{3.7}+\frac{2.51}{1.187\times10^7\sqrt{f}}\right)$$

It gives $f = 0.00829$. Then the pressure drop, the head loss, and the required power input become

$$\Delta P=\Delta P_L=f\frac{L}{D}\frac{\rho V^2}{2}=0.00829\frac{12{,}000\text{ m}}{0.60\text{ m}}\frac{(950.6\text{ kg/m}^3)(5.305\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)=2218\text{ kPa}$$

$$h_L=\frac{\Delta P_L}{\rho g}=f\frac{L}{D}\frac{V^2}{2g}=(0.00829)\frac{12{,}000\text{ m}}{0.60\text{ m}}\frac{(5.305\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=238\text{ m}$$

$$\dot{W}_{\text{electric, in}}=\frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}}=\frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}}=\frac{(1.5\text{ m}^3/\text{s})(2218\text{ kPa})}{0.80}\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right)=4159\text{ kW}\cong\mathbf{4160\ kW}$$

Therefore, the pumps will consume 4160 kW of electric power to overcome friction and maintain flow. The pumps must raise the pressure of the geothermal water by 2218 kPa. Providing a pressure rise of this magnitude at one location may create excessive stress in piping at that location. Therefore, it is more desirable to raise the pressure by smaller amounts at a several locations along the flow. This will keep the maximum pressure in the system and the stress in piping at a safer level.

(*b*) The daily cost of electric power consumption is determined by multiplying the amount of power used per day by the unit cost of electricity,

$$\text{Amount} = \dot{W}_{\text{elect,in}}\Delta t = (4159\ \text{kW})(24\ \text{h/day}) = 99{,}816\ \text{kWh/day}$$

$$\text{Cost} = \text{Amount} \times \text{Unit cost} = (99{,}816\ \text{kWh/day})(\$0.06/\text{kWh}) = \$5989/\text{day} \cong \mathbf{\$5990/day}$$

(*c*) The energy consumed by the pump (except the heat dissipated by the motor to the air) is eventually dissipated as heat due to the frictional effects. Therefore, this problem is equivalent to heating the water by a 4159 kW of resistance heater (again except the heat dissipated by the motor). To be conservative, we consider only the useful mechanical energy supplied to the water by the pump. The temperature rise of water due to this addition of energy is

$$\dot{W}_{\text{mech}} = \rho\dot{V}c_p\Delta T \ \rightarrow\ \Delta T = \frac{\eta_{\text{pump-motor}}\dot{W}_{\text{elect,in}}}{\rho\dot{V}c_p} = \frac{0.80 \times (4159\ \text{kJ/s})}{(950.6\ \text{kg/m}^3)(1.5\ \text{m}^3/\text{s})(4.229\ \text{kJ/kg}\cdot°\text{C})} = \mathbf{0.55°C}$$

Therefore, the temperature of water will rise at least 0.55°C, which is more than the 0.5°C drop in temperature (in reality, the temperature rise will be more since the energy dissipation due to pump inefficiency will also appear as temperature rise of water). Thus we conclude that the frictional heating during flow can more than make up for the temperature drop caused by heat loss.

Discussion The pumping power requirement and the associated cost can be reduced by using a larger diameter pipe. But the cost savings should be compared to the increased cost of larger diameter pipe.

8-96

Solution Geothermal water is supplied to a city through cast iron pipes at a specified rate. The electric power consumption and its daily cost are to be determined, and it is to be assessed if the frictional heating during flow can make up for the temperature drop caused by heat loss.

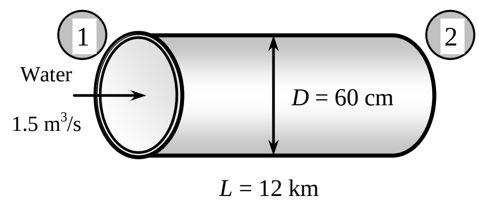
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses. **4** The geothermal well and the city are at about the same elevation. **5** The properties of geothermal water are the same as fresh water. **6** The fluid pressures at the wellhead and the arrival point in the city are the same.

Properties The properties of water at 110°C are $\rho = 950.6$ kg/m^3, $\mu = 0.255\times10^{-3}$ kg/m·s, and $C_p = 4.229$ kJ/kg·°C. The roughness of cast iron pipes is 0.00026 m.

Analysis (*a*) We take point 1 at the well-head of geothermal resource and point 2 at the final point of delivery at the city, and the entire piping system as the control volume. Both points are at the same elevation ($z_2 = z_2$) and the same velocity ($V_1 = V_2$) since the pipe diameter is constant, and the same pressure ($P_1 = P_2$). Then the energy equation for this control volume simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad h_{\text{pump,u}} = h_L$$

That is, the pumping power is to be used to overcome the head losses due to friction in flow. The average velocity and the Reynolds number are



$$V = \frac{\dot V}{A_c} = \frac{\dot V}{\pi D^2/4} = \frac{1.5\,\text{m}^3/\text{s}}{\pi(0.60\,\text{m})^2/4} = 5.305\,\text{m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(950.6\,\text{kg/m}^3)(5.305\,\text{m/s})(0.60\,\text{m})}{0.255\times10^{-3}\,\text{kg/m}\cdot\text{s}} = 1.187\times10^7$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{0.00026\,\text{m}}{0.60\,\text{m}} = 4.33\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{4.33\times10^{-4}}{3.7} + \frac{2.51}{1.187\times10^7\sqrt{f}}\right)$$

It gives $f = 0.0162$. Then the pressure drop, the head loss, and the required power input become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.0162\frac{12{,}000\,\text{m}}{0.60\,\text{m}}\frac{(950.6\,\text{kg/m}^3)(5.305\,\text{m/s})^2}{2}\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}}\right)\left(\frac{1\,\text{kPa}}{1\,\text{kN/m}^2}\right) = 4334\ \text{kPa}$$

$$h_L = \frac{\Delta P_L}{\rho g} = f\frac{L}{D}\frac{V^2}{2g} = (0.0162)\frac{12{,}000\,\text{m}}{0.60\,\text{m}}\frac{(5.305\,\text{m/s})^2}{2(9.81\,\text{m/s}^2)} = 465\,\text{m}$$

$$\dot W_{\text{elect, in}} = \frac{\dot W_{\text{pump, u}}}{\eta_{\text{pump-motor}}} = \frac{\dot V\Delta P}{\eta_{\text{pump-motor}}} = \frac{(1.5\text{ m}^3/\text{s})(4334\text{ kPa})}{0.74}\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right) = 8785\text{ kW} \cong \mathbf{8790\ kW}$$

Therefore, the pumps will consume 8785 kW of electric power to overcome friction and maintain flow. The pumps must raise the pressure of the geothermal water by 4334 kPa. Providing a pressure rise of this magnitude at one location may create excessive stress in piping at that location. Therefore, it is more desirable to raise the pressure by smaller amounts at a several locations along the flow. This will keep the maximum pressure in the system and the stress in piping at a safer level.

(*b*) The daily cost of electric power consumption is determined by multiplying the amount of power used per day by the unit cost of electricity,

$$\text{Amount} = \dot W_{\text{elect,in}}\Delta t = (8785\,\text{kW})(24\,\text{h/day}) = 210{,}800\,\text{kWh/day}$$

$$\text{Cost} = \text{Amount} \times \text{Unit cost} = (210{,}800 \text{ kWh/day})(\$0.06/\text{kWh}) = \$12{,}650/\text{day} \cong \mathbf{\$12{,}700/day}$$

(*c*) The energy consumed by the pump (except the heat dissipated by the motor to the air) is eventually dissipated as heat due to the frictional effects. Therefore, this problem is equivalent to heating the water by a 8785 kW of resistance heater (again except the heat dissipated by the motor). To be conservative, we consider only the useful mechanical energy supplied to the water by the pump. The temperature rise of water due to this addition of energy is

$$\dot{W}_{\text{mech}} = \rho \dot{V} c_p \Delta T \;\rightarrow\; \Delta T = \frac{\eta_{\text{pump-motor}} \dot{W}_{\text{elect,in}}}{\rho \dot{V} c_p} = \frac{0.74 \times (8785 \text{ kJ/s})}{(950.6 \text{ kg/m}^3)(1.5 \text{ m}^3/\text{s})(4.229 \text{ kJ/kg} \cdot {}^\circ\text{C})} = \mathbf{1.1^\circ C}$$

Therefore, the temperature of water will rise at least 1.1°C, which is more than the 0.5°C drop in temperature (in reality, the temperature rise will be more since the energy dissipation due to pump inefficiency will also appear as temperature rise of water). Thus we conclude that the frictional heating during flow can more than make up for the temperature drop caused by heat loss.

Discussion The pumping power requirement and the associated cost can be reduced by using a larger diameter pipe. But the cost savings should be compared to the increased cost of larger diameter pipe.

8-97

Solution Water is transported through a plastic pipe by gravity. The flow rate of water and the power requirement to maintain this flow rate if the pipe were horizontal are to be determined.

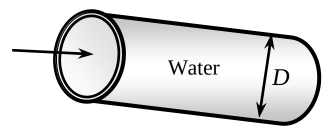
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors, and thus no minor losses. **4** Water level in the well remains constant. **5** Both ends of the pipe are open to the atmosphere. **6** The kinetic energy correction factors are the same at both the inlet and exit.

Properties The density and kinematic viscosity of water are given to be ρ = 1000 kg/m^3 and $\nu = 1\times10^{-6}$ m^2/s. The plastic pipe is smooth and thus $\varepsilon = 0$.

Analysis We choose point 1 at the inlet and point 2 at the outlet of the pipe. We note the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and the fluid velocities at both points are equal ($V_1 = V_2$). There are no pumps and turbines involved. Then the energy equation for a control volume that consists of the entire pipe simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_L=z_1-z_2$$

where $h_L = z_1 - z_2$ = Gradient×Length = 0.01×800 m = 8 m. Note that flow rate is established at the point where the head loss equals the elevation drop. The average velocity, Reynolds number, friction factor, and the head loss relations can be expressed as (D is in m, V is in m/s, Re and f are dimensionless)



$$V=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4}=\frac{\dot{V}}{\pi(0.12\text{ m})^2/4}$$

$$\text{Re}=\frac{VD}{\nu}=\frac{V(0.12\text{ m})}{1\times10^{-6}\text{ m}^2/s}$$

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\quad\rightarrow\quad\frac{1}{\sqrt{f}}=-2.0\log\left(0+\frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$h_L=f\frac{L}{D}\frac{V^2}{2g}\quad\rightarrow\quad 8\text{ m}=f\frac{800\text{ m}}{0.12\text{ m}}\frac{V^2}{2(9.81\text{ m/s}^2)}$$

This is a set of 4 equations in 4 unknowns, and solving them with an equation solver gives

$\dot{V}$ = **0.0134 m^3/s**, f = 0.0167, V = 1.186 m/s, and Re = 1.42×10^5

Therefore, the 8-m elevation drop will cause a flow rate of 13.4 L/s. Note that Re > 4000, and thus the turbulent flow assumption is verified.

If the pipe were horizontal, the required useful pumping power to achieve the same flow rate is determined as follows. If the flow rate is the same, then the average velocity, the friction factor, the Reynolds number, and the head loss will also be the same. The energy equation in this horizontal pipe case reduces to $h_{\text{pump,u}}=h_L=8\text{ m}$. To overcome this head loss, the required useful pumping power must be

$$\dot{W}_{\text{pump,u}}=\dot{V}\rho g h_{\text{pump,u}}=(0.0134\text{ m}^3/\text{s})(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(8\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right)=\mathbf{1.05\text{ kW}}$$

Discussion Note that the effect of gravity in this case is equivalent to a pump that delivers 1.05 kW of useful power.

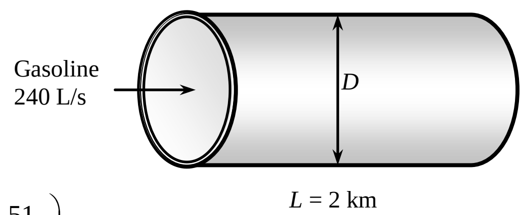
8-98

Solution The flow rate and the maximum head loss in a gasoline pipeline are given. The required minimum diameter of the pipe is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe line involves no components such as bends, valves, and connectors, and thus no minor losses. **4** The flow is turbulent (to be verified).

Properties The density and kinematic viscosity of gasoline are given to be $\rho = 680$ kg/m^3 and $\nu = 4.29\times10^{-7}$ m^2/s.

Analysis The average velocity, Reynolds number, friction factor, and the head loss relations can be expressed as (D is in m, V is in m/s, Re and f are dimensionless)



$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.240\text{ m}^3/s}{\pi D^2/4}$$

$$\text{Re} = \frac{VD}{\nu} = \frac{VD}{4.29\times10^{-7}\text{ m}^2/s}$$

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) = -2.0\log\left(\frac{(3\times10^{-5}\text{ m})/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$h_L = f\frac{L}{D}\frac{V^2}{2g} \quad \rightarrow \quad 10\text{ m} = f\frac{2000\text{ m}}{D}\frac{V^2}{2(9.81\text{ m/s}^2)}$$

This is a set of 4 equations in 4 unknowns, and solving them with an equation solver gives

D = **0.412 m**, $f = 0.0124$, $V = 1.80$ m/s, and Re $= 1.73\times10^6$

Therefore, the diameter of the duct should be more than 41.2 cm if the head loss is not to exceed 10 m. Note that Re > 4000, and thus the turbulent flow assumption is verified.

The diameter can also be determined directly from the third Swamee-Jain formula to be

$$D = 0.66\left[\varepsilon^{1.25}\left(\frac{L\dot{V}^2}{gh_L}\right)^{4.75} + \nu\dot{V}^{9.4}\left(\frac{L}{gh_L}\right)^{5.2}\right]^{0.04}$$

$$= 0.66\left[(3\times10^{-5}\text{ m})^{1.25}\left(\frac{(2000\text{ m})(0.240\text{ m}^3/\text{s})^2}{(9.81\text{ m/s}^2)(10\text{ m})}\right)^{4.75} + (4.29\times10^{-7}\text{ m}^2/\text{s})(0.240\text{ m}^3/\text{s})^{9.4}\left(\frac{2000\text{ m}}{(9.81\text{ m/s}^2)(10\text{ m})}\right)^{5.2}\right]^{0.04}$$

$$= 0.414\text{ m}$$

Discussion Note that the difference between the two results is less than 1%. Therefore, the simple Swamee-Jain relation can be used with confidence.

8-99

Solution Hot water in a water tank is circulated through a loop made of cast iron pipes at a specified average velocity. The required power input for the recirculating pump is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** Minor losses other than those for elbows and valves are negligible.

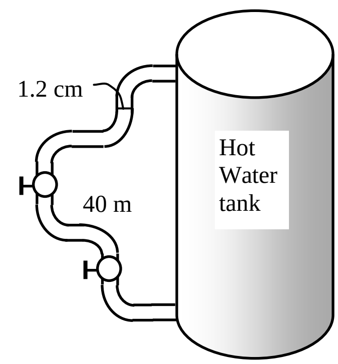
Properties The density and dynamic viscosity of water at 60°C are ρ = 983.3 kg/m^3, μ = 0.467×10^{-3} kg/m·s. The roughness of cast iron pipes is 0.00026 m. The loss coefficient is K_L = 0.9 for a threaded 90° smooth bend and K_L = 0.2 for a fully open gate valve.

Analysis Since the water circulates continually and undergoes a cycle, we can take the entire recirculating system as the control volume, and choose points 1 and 2 at any location at the same point. Then the properties (pressure, elevation, and velocity) at 1 and 2 will be identical, and the energy equation will simplify to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=h_L$$

where

$$h_L=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$$



since the diameter of the piping system is constant. Therefore, the pumping power is to be used to overcome the head losses in the flow. The flow rate and the Reynolds number are

$$\dot V=VA_c=V(\pi D^2/4)=(2\text{ m/s})[\pi(0.012\text{ m})^2/4]=2.262\times10^{-4}\text{ m}^3/\text{s}$$

$$\text{Re}=\frac{\rho VD}{\mu}=\frac{(983.3\text{ kg/m}^3)(2\text{ m/s})(0.012\text{ m})}{0.467\times10^{-3}\text{ kg/m}\cdot\text{s}}=50{,}534$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{0.00026\text{ m}}{0.012\text{ m}}=0.0217$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt f}\right)\ \rightarrow\ \frac{1}{\sqrt f}=-2.0\log\left(\frac{0.0217}{3.7}+\frac{2.51}{50{,}534\sqrt f}\right)$$

It gives f = 0.05089. Then the total head loss, pressure drop, and the required pumping power input become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.05089)\frac{40\text{ m}}{0.012\text{ m}}+6\times0.9+2\times0.2\right)\frac{(2\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=35.77\text{ m}$$

$$\Delta P=\Delta P_L=\rho g h_L=(983.3\text{ kg/m}^3)(9.81\text{ m/s}^2)(35.77\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)=345.0\text{ kPa}$$

$$\dot W_{\text{elect}}=\frac{\dot W_{\text{pump,u}}}{\eta_{\text{pump-motor}}}=\frac{\dot V\Delta P}{\eta_{\text{pump-motor}}}=\frac{(2.262\times10^{-4}\text{ m}^3/\text{s})(345.0\text{ kPa})}{0.70}\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right)=\mathbf{0.111\text{ kW}}$$

Therefore, **the required power input of the recirculating pump is 111 W**.

Discussion It can be shown that the required pumping power input for the recirculating pump is 0.108 kW when the minor losses are not considered. Therefore, the minor losses can be neglected in this case without a major loss in accuracy.

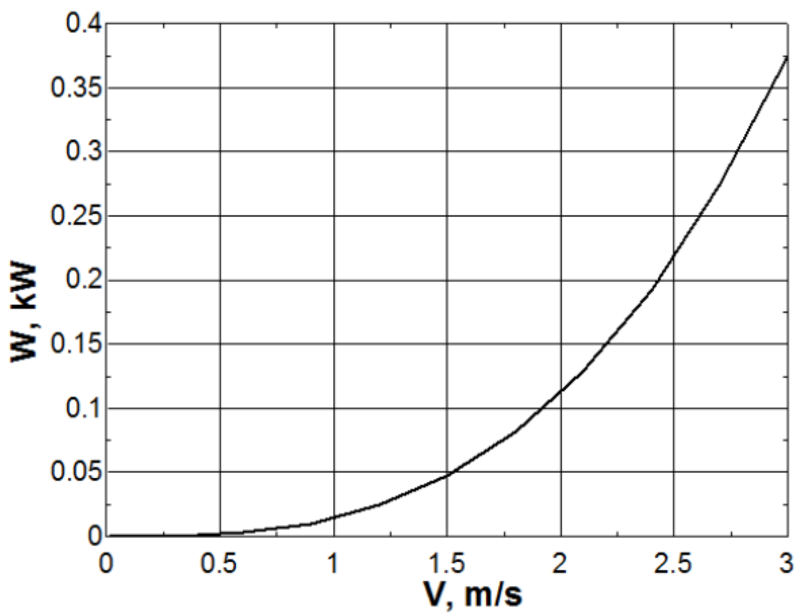
8-100

Solution In the previous problem, the effect of average flow velocity on the power input to the recirculating pump for the same constant flow rate is to be investigated by varying the velocity from 0 to 3 m/s in increments of 0.3 m/s.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81
rho=983.3
nu=mu/rho
mu=0.000467
D=0.012
L=40
KL=6*0.9+2*0.2
Eff=0.7
Ac=pi*D^2/4
Vdot=V*Ac
eps=0.00026
rf=eps/D
"Reynolds number"
Re=V*D/nu
1/sqrt(f)=-2*log10(rf/3.7+2.51/(Re*sqrt(f)))
DP=(f*L/D+KL)*rho*V^2/2000 "kPa"
W=Vdot*DP/Eff "kW"
HL=(f*L/D+KL)*(V^2/(2*g))
```

V, m/s	W_{pump}, kW	ΔP_L, kPa	Re
0.0	0.0000	0.0	0
0.3	0.0004	8.3	7580
0.6	0.0031	32.0	15160
0.9	0.0103	71.0	22740
1.2	0.0243	125.3	30320
1.5	0.0472	195.0	37900
1.8	0.0814	279.9	45480
2.1	0.1290	380.1	53060
2.4	0.1922	495.7	60640
2.7	0.2733	626.6	68220
3.0	0.3746	772.8	75800



Discussion As you might have suspected, the required power does not increase linearly with average velocity. Rather, the relationship is nearly quadratic. A larger diameter pipe would cut reduce the required pumping power considerably.

8-101

Solution Hot water in a water tank is circulated through a loop made of plastic pipes at a specified average velocity. The required power input for the recirculating pump is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** Minor losses other than those for elbows and valves are negligible.

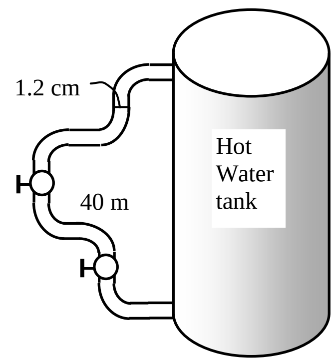
Properties The density and viscosity of water at 60°C are ρ = 983.3 kg/m^3, $\mu = 0.467\times10^{-3}$ kg/m·s. Plastic pipes are smooth, and thus their roughness is very close to zero, ε = 0. The loss coefficient is K_L = 0.9 for a threaded 90° smooth bend and K_L = 0.2 for a fully open gate valve.

Analysis Since the water circulates continually and undergoes a cycle, we can take the entire recirculating system as the control volume, and choose points 1 and 2 at any location at the same point. Then the properties (pressure, elevation, and velocity) at 1 and 2 will be identical, and the energy equation will simplify to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=h_L$$

where

$$h_L=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$$



since the diameter of the piping system is constant. Therefore, the pumping power is to be used to overcome the head losses in the flow. The flow rate and the Reynolds number are

$$\dot{V}=VA_c=V(\pi D^2/4)=(2.5\text{ m/s})[\pi(0.012\text{ m})^2/4]=2.827\times10^{-4}\text{ m}^3\text{/s}$$

$$\text{Re}=\frac{\rho VD}{\mu}=\frac{(983.3\text{ kg/m}^3)(2.5\text{ m/s})(0.012\text{ m})}{0.467\times10^{-3}\text{ kg/m}\cdot\text{s}}=63{,}200$$

which is greater than 4000. Therefore, the flow is turbulent. The friction factor corresponding to the relative roughness of zero and this Reynolds number can simply be determined from the Moody chart. To avoid the reading error, we determine it from the Colebrook equation,

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\quad\rightarrow\quad\frac{1}{\sqrt{f}}=-2.0\log\left(0+\frac{2.51}{63200\sqrt{f}}\right)$$

It gives f = 0.0198. Then the total head loss, pressure drop, and the required pumping power input become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.0198)\frac{40\text{ m}}{0.012\text{ m}}+6\times0.9+2\times0.2\right)\frac{(2.5\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=22.9\text{ m}$$

$$\Delta P=\rho g h_L=(983.3\text{ kg/m}^3)(9.81\text{ m/s}^2)(22.9\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)=221\text{ kPa}$$

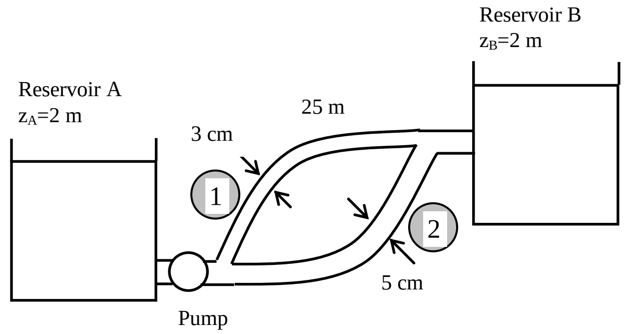
$$\dot{W}_{\text{elect}}=\frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}}=\frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}}=\frac{(2.827\times10^{-4}\text{ m}^3\text{/s})(221\text{ kPa})}{0.70}\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3\text{/s}}\right)=\mathbf{0.0893\,kW}$$

Therefore, **the required power input of the recirculating pump is 89.3 W**.

Discussion It can be shown that the required pumping power input for the recirculating pump is 82.1 W when the minor losses are not considered. Therefore, the minor losses can be neglected in this case without a major loss in accuracy. Compared to the cast iron pipes of the previous problem, the plastic pipes reduced the required power by more than 50%, from 217 to 89.3 W. Furthermore, plastic pipes are lighter and easier to install, and they don't rust.

8-102

Solution The pumping power input to a piping system with two parallel pipes between two reservoirs is given. The flow rates are to be determined.



Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The elevations of the reservoirs remain constant. **4** The minor losses and the head loss in pipes other than the parallel pipes are said to be negligible. **5** The flows through both pipes are turbulent (to be verified).

Properties The density and dynamic viscosity of water at 20°C are ρ = 998 kg/m^3 and μ = 1.002×10^{-3} kg/m·s. Plastic pipes are smooth, and their roughness is zero, ε = 0.

Analysis This problem cannot be solved directly since the velocities (or flow rates) in the pipes are not known. Therefore, we would normally use a trial-and-error approach here. However, nowadays the equation solvers such as EES are widely available, and thus below we will simply set up the equations to be solved by an equation solver. The head supplied by the pump to the fluid is determined from

$$\dot{W}_{\text{elect,in}} = \frac{\rho \dot{V} g h_{\text{pump,u}}}{\eta_{\text{pump-motor}}} \quad \rightarrow \quad 7000 \text{ W} = \frac{(998 \text{ kg/m}^3)\dot{V}(9.81 \text{ m/s}^2)h_{\text{pump,u}}}{0.68} \qquad (1)$$

We choose points A and B at the free surfaces of the two reservoirs. Noting that the fluid at both points is open to the atmosphere (and thus $P_A = P_B = P_{\text{atm}}$) and that the fluid velocities at both points are zero ($V_A = V_B$ =0), the energy equation for a control volume between these two points simplifies to

$$\frac{P_A}{\rho g} + \alpha_A \frac{V_A^2}{2g} + z_A + h_{\text{pump,u}} = \frac{P_B}{\rho g} + \alpha_B \frac{V_B^2}{2g} + z_B + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad h_{\text{pump,u}} = (z_B - z_A) + h_L$$

or

$$h_{\text{pump,u}} = (9-2) + h_L \quad (2)$$

where

$$h_L = h_{L,1} = h_{L,2} \qquad (3)\,(4)$$

We designate the 3-cm diameter pipe by 1 and the 5-cm diameter pipe by 2. The average velocity, Reynolds number, friction factor, and the head loss in each pipe are expressed as

$$V_1 = \frac{\dot{V}_1}{A_{c,1}} = \frac{\dot{V}_1}{\pi D_1^2/4} \quad \rightarrow \quad V_1 = \frac{\dot{V}_1}{\pi (0.03\text{m})^2/4} \qquad (5)$$

$$V_2 = \frac{\dot{V}_2}{A_{c,2}} = \frac{\dot{V}_2}{\pi D_2^2/4} \quad \rightarrow \quad V_2 = \frac{\dot{V}_2}{\pi (0.05\text{m})^2/4} \qquad (6)$$

$$\text{Re}_1 = \frac{\rho V_1 D_1}{\mu} \quad \rightarrow \quad \text{Re}_1 = \frac{(998 \text{ kg/m}^3)V_1(0.03 \text{ m})}{1.002\times10^{-3} \text{ kg/m}\cdot s} \qquad (7)$$

$$\text{Re}_2 = \frac{\rho V_2 D_2}{\mu} \quad \rightarrow \quad \text{Re}_2 = \frac{(998 \text{ kg/m}^3)V_2(0.05 \text{ m})}{1.002\times10^{-3} \text{ kg/m}\cdot s} \qquad (8)$$

$$\frac{1}{\sqrt{f_1}} = -2.0\log\left(\frac{\varepsilon/D_1}{3.7} + \frac{2.51}{\text{Re}_1\sqrt{f_1}}\right) \quad \rightarrow \quad \frac{1}{\sqrt{f_1}} = -2.0\log\left(0 + \frac{2.51}{\text{Re}_1\sqrt{f_1}}\right) \qquad (9)$$

$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{\varepsilon/D_2}{3.7} + \frac{2.51}{\text{Re}_2\sqrt{f_2}}\right) \rightarrow \frac{1}{\sqrt{f_2}} = -2.0\log\left(0 + \frac{2.51}{\text{Re}_2\sqrt{f_2}}\right) \quad (10$$

$$h_{L,1} = f_1 \frac{L_1}{D_1}\frac{V_1^2}{2g} \quad \rightarrow \quad h_{L,1} = f_1 \frac{25\,\text{m}}{0.03\,\text{m}}\frac{V_1^2}{2(9.81\,\text{m/s}^2)} \qquad (11)$$

$$h_{L,2} = f_2 \frac{L_2}{D_2}\frac{V_2^2}{2g} \quad \rightarrow \quad h_{L,2} = f_2 \frac{25\,\text{m}}{0.05\,\text{m}}\frac{V_2^2}{2(9.81\,\text{m/s}^2)} \qquad (12)$$

$$\dot{V} = \dot{V}_1 + \dot{V}_2 \qquad (13)$$

This is a system of 13 equations in 13 unknowns, and their simultaneous solution by an equation solver gives

$$\dot{V} = \mathbf{0.0183\,m^3/s}, \quad \dot{V}_1 = \mathbf{0.0037\,m^3/s}, \quad \dot{V}_2 = \mathbf{0.0146\,m^3/s},$$

V_1 = 5.30 m/s, V_2 = 7.42 m/s, $h_L = h_{L,1} = h_{L,2} = 19.5\,\text{m}$, $h_{\text{pump,u}}$ = 26.5 m

Re_1 = 158,300, Re_2 = 369,700, f_1 = 0.0164, f_2 = 0.0139

Note that Re > 4000 for both pipes, and thus the assumption of turbulent flow is verified.

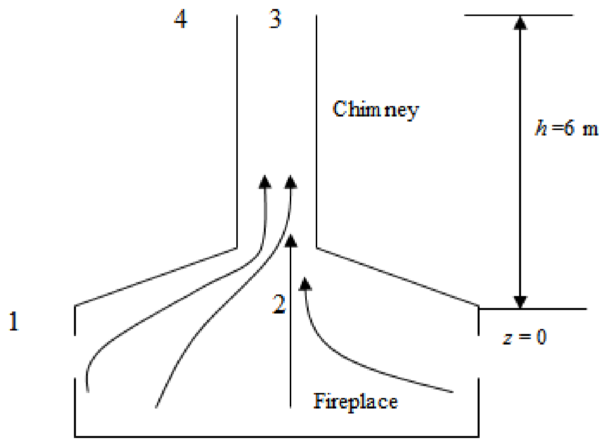
Discussion This problem can also be solved by using an iterative approach, but it will be very time consuming. Equation solvers such as EES are invaluable for this kind of problems.

8-103

Solution A chimney is to be designed to discharge hot gases from a fireplace. The chimney diameter that would discharge the hot gases at the desired rate is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The flow is fully developed.

Analysis



We have denoted various points by 1, 2, 3, and 4 as shown.

Writing Energy equation between 2 and 3 we get;

$$P_2+\frac{1}{2}\rho_G V_2^{\,2}+\gamma_G Z_2=P_3+\frac{1}{2}\rho_G V_3^{\,2}+\gamma_G Z_3+K.\frac{1}{2}\rho_G V_3^{\,2}+\lambda\frac{L}{d_c}\rho_G\frac{V_3^{\,2}}{2}$$

Since $V_2\cong 0$, $Z_2=0$, $P_1=P_4+\gamma_{air}.h$, $P_3=P_4=P_{atm}$, $P_1=P_2$, and $Z_3=h$

Bernoulli equation reduces to

$$P_{atm}+\gamma_{air}.h=P_{atm}+\gamma_G h+\frac{1}{2}\rho_G V_3^{\,2}\left(1+K+\lambda\frac{L}{d_c}\right)$$

or

$$2gh\left(\frac{\rho_{air}}{\rho_G}-1\right)=V_3^{\,2}\left(1+K+\lambda\frac{L}{dc}\right)=\frac{16Q^2}{\pi^2 d_c^{\,4}}\left(1+K+\lambda\frac{L}{dc}\right)$$

Assuming an isothermal flow,

$$\frac{\rho_{air}}{\rho_G}=\frac{T_{air}}{T_G}=\frac{180+273}{20+273} \qquad \frac{\rho_{air}}{\rho_G}=1.546$$

$$2\times 9.81\times 6(1.546-1)=\frac{16\times 0.15^2}{\pi^2\times d_c^{\,4}}\left(1+1.5+0.02\frac{6}{d_c}\right)$$

$$1762.14=\left(\frac{2.5}{d_c^{\,4}}+\frac{0.12}{d_c}\right)$$

This equation can be solved by try and error method, which would give

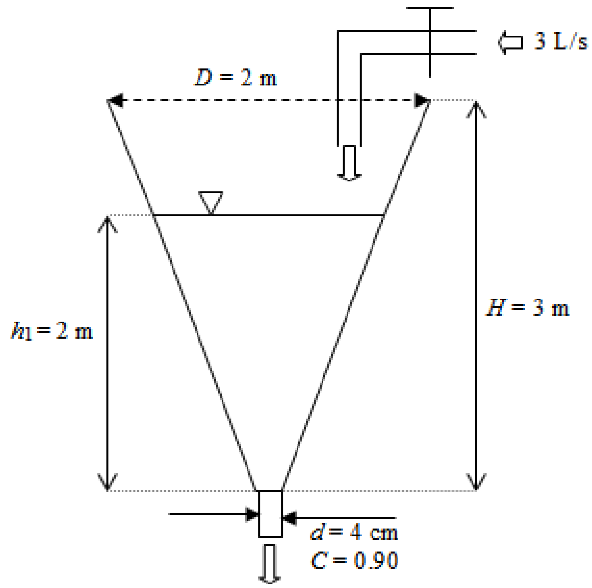
$d_c\cong$ **0.194 m**

8-104

Solution An inverted conical container is filled with water. A faucet supply water into the container and water is withdrawn from a hole at the bottom. The time it will take for the water level in the tank to drop to a specified level is to be determined.

Assumptions **1** The flow is steady and incompressible.

Analysis



From the continuity

$$C.A_h.\sqrt{2gh}dt - qdt = -A_T.dh \text{ where}$$

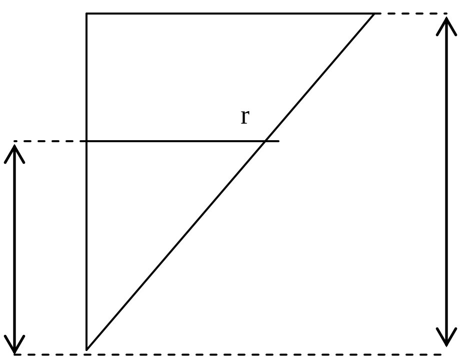
A_h=hole cross-sectional area, A_T=container cross-sectional area.

Therefore ; $$dt = \frac{-A_T.dh}{C.A_h\sqrt{2gh} - q}$$

In other hand, $A_T=\pi r^2$ at any h depth of water; that is $A_T=f(h)$

From the similarity;

D/2=1 m



$$\frac{h}{3} = \frac{r}{1} \rightarrow r = \frac{h}{3}$$

3m

h

$A_T=\pi r^2=\pi/9.h^2$

Thus ,

$$dt = \frac{\frac{\pi}{9}h^2dh}{q - C.A_h\sqrt{2g}\sqrt{h}} = \frac{\pi}{9}\frac{h^2dh}{q - K\sqrt{h}} \quad \text{where } K = CA_h\sqrt{2g}$$

Substituting the given values we obtain

$$\int_0^T dt = \int_{h_1=2}^{h_2=1} \frac{0.349h^2}{0.003 - 0.005009\sqrt{h}}$$

However it may be difficult to take the given integral. A numerical approach such as Simpson's Rule or Trapezoidal Rule can be effectively used for integration. Using Simpson's rule we obtained T = **252.9 seconds**. The exact solution is T=252.848 seconds which is very close to the numerically obtained value. If we take the limit for $T \rightarrow \infty$ we would see that the tank would never be emptied completely and h_2 cannot be lower than approximately 0.36cm.

Flow Rate and Velocity Measurements

8-105C

Solution We are to compare thermal and laser Doppler anemometers.

Analysis A *thermal anemometer* involves a **very small electrically heated sensor (hot wire) which loses heat to the fluid, and the flow velocity is related to the electric current needed to maintain the sensor at a constant temperature**. The flow velocity is determined by measuring the voltage applied or the electric current passing through the sensor. A *laser Doppler anemometer (LDA)* does not have a sensor that intrudes into flow. Instead, it **uses two laser beams that intersect at the point where the flow velocity is to be measured, and it makes use of the frequency shift (the Doppler effect) due to fluid flow to measure velocity**.

Discussion Both of these devices measure the flow velocity at a point in the flow. Of the two, the hot wire system is much less expensive and has higher frequency resolution, but may interfere with the flow being measured.

8-106C

Solution We are to compare LDV and PIV.

Analysis *Laser Doppler velocimetry (LDV)* **measures velocity at a point**, but *particle image velocimetry (PIV)* **provides velocity values simultaneously throughout an entire cross-section and thus it is a whole-field technique**. PIV combines the accuracy of LDV with the capability of flow visualization, and provides instantaneous flow field mapping. Both methods are non-intrusive, and both utilize laser light beams.

Discussion In both cases, optical access is required – a hot-wire system does not require optical access, but, like the LDV system, measures velocity only at a single point.

8-107C

Solution We are to discus the primary considerations when choosing a flowmeter.

Analysis The primary considerations when selecting a *flowmeter* are **cost, size, pressure drop, capacity, accuracy, and reliability**.

Discussion As with just about everything you purchase, you usually get what you pay for.

8-108C

Solution We are to explain how a Pitot-static tube works and discuss its application.

Analysis A *Pitot-static tube* **measures the difference between the stagnation and static pressure**, which is the *dynamic pressure*, which is related to flow velocity by $\boxed{V = \sqrt{2(P_1 - P_2)/\rho}}$. Once the average flow velocity is determined, the flow rate is calculated from $\dot{V} = VA_c$. The Pitot tube is inexpensive, highly reliable since it has no moving parts, it has very small pressure drop, and its accuracy (which is about 3%) is acceptable for most engineering applications.

Discussion The term "Pitot tube" or "Pitot probe" is often used in place of "Pitot-static probe". Technically, however, a Pitot probe measures only stagnation pressure, while a Pitot-static probe measures both stagnation *and* static pressures.

8-109C

Solution We are to discuss the operation of obstruction flowmeters.

Analysis An *obstruction flowmeter* measures the flow rate through a pipe by **constricting the flow, and measuring the decrease in pressure due to the increase in velocity at (or downstream of) the constriction site**. The flow rate for obstruction flowmeters is expressed as $\boxed{\dot{V} = A_o C_0 \sqrt{2(P_1 - P_2)/[\rho(1-\beta^4)]}}$ where $A_0 = \pi\, d^2 / 4$ is the cross-sectional area of the obstruction and $\beta = d/D$ is the ratio of obstruction diameter to the pipe diameter. Of the three types of obstruction flow meters, the orifice meter is the cheapest, smallest, and least accurate, and it causes the greatest head loss. The Venturi meter is the most expensive, the largest, the most accurate, and it causes the smallest head loss. The nozzle meter is between the orifice and Venturi meters in all aspects.

Discussion As diameter ratio β decreases, the pressure drop across the flowmeter increases, leading to a larger minor head loss associated with the flowmeter, but increasing the sensitivity of the measurement.

8-110C

Solution We are to discuss the operation of positive displacement flowmeters.

Analysis A **positive displacement flowmeter** operates by **trapping a certain amount of incoming fluid, displacing it to the discharge side of the meter, and counting the number of such discharge-recharge cycles to determine the total amount of fluid displaced**. Positive displacement flowmeters are commonly used to meter gasoline, water, and natural gas because they are simple, reliable, inexpensive, and highly accurate even when the flow is unsteady.

Discussion In applications such as a gasoline meter, it is not the flow *rate* that is measured, but the flow *volume*.

8-111C

Solution We are to discuss the operation of a turbine flowmeter.

Analysis A *turbine flowmeter* consists of a **cylindrical flow section that houses a turbine that is free to rotate, and a sensor that generates a pulse each time a marked point on the turbine passes by to determine the rate of rotation**. Turbine flowmeters are relatively inexpensive, give highly accurate results (as accurate as 0.25%) over a wide range of flow rates, and cause a very small head loss.

Discussion Turbine flowmeters must be calibrated so that a reading of the rpm of the turbine is translated into average velocity in the pipe or volume flow rate through the pipe.

8-112C

Solution We are to discuss the operation of rotameters.

Analysis A *variable-area flowmeter* (or *rotameter*) consists of **a tapered conical transparent tube made of glass or plastic with a float inside that is free to move**. As fluid flows through the tapered tube, the float rises within the tube to a location where the float weight, drag force, and buoyancy force balance each other. Variable-area flowmeters are very simple devices with no moving parts except for the float (but even the float remains stationary during steady operation), and thus they are very reliable. They are also very inexpensive, and they cause a relatively small head loss.

Discussion There are also some disadvantages. For example, they must be mounted vertically, and most of them require a visual reading, and so cannot be automated or connected to a computer system.

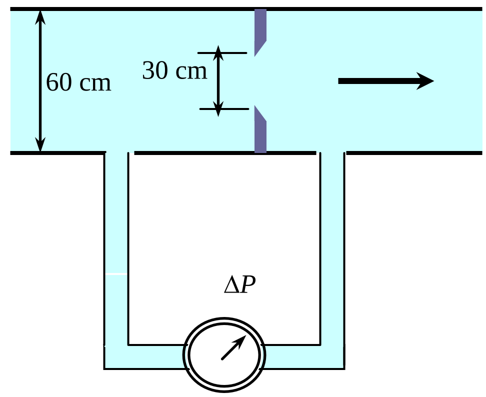
8-113

Solution The flow rate of water is measured with an orifice meter. The pressure difference indicated by the orifice meter and the head loss are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The discharge coefficient of the orifice meter is C_d = 0.61.

Properties The density and dynamic viscosity of water are given to be ρ = 998 kg/m^3 and μ = 1.002×10^{-4} kg/m·s, respectively.

Analysis The diameter ratio and the throat area of the orifice are



$$\beta = d/D = 30/60 = 0.50$$

$$A_0 = \pi d^2/4 = \pi(0.30\,\text{m})^2/4 = 0.07069\,\text{m}^2$$

For a pressure drop of $\Delta P = P_1 - P_2$ across the orifice plate, the flow rate is expressed as

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}}$$

Substituting,

$$0.40\,\text{m}^3/s = (0.07069\,\text{m}^2)(0.61)\sqrt{\frac{2\Delta P}{(998\,\text{kg/m}^3)((1-0.50^4)}}$$

which gives the pressure drop across the orifice plate to be

$$\Delta P = 40{,}250\,\text{kg}\cdot\text{m/s}^2 \cong \mathbf{40.3\,kPa}$$

It corresponds to a water column height of

$$h_w = \frac{\Delta P}{\rho_w g} = \frac{40{,}250\,\text{kg}\cdot\text{m/s}^2}{(998\,\text{kg/m}^3)(9.81\,\text{m/s}^2)} = 4.11\,\text{m}$$

The head loss between the two measurement sections can be estimated from the energy equation. Since $z_1 = z_2$, the head form of the energy equation simplifies to

$$h_L \approx \frac{P_1 - P_2}{\rho_f g} - \frac{V_2^2 - V_1^2}{2g} = h_w - \frac{\left[(D/d)^4 - 1\right]V_1^2}{2g} = 4.11\,\text{m} - \frac{\left[(60/30)^4 - 1\right](1.415\,\text{m/s})^2}{2(9.81\,\text{m/s}^2)} = \mathbf{2.58\,m\,H_2O}$$

where $V = \dfrac{\dot{V}}{A_c} = \dfrac{\dot{V}}{\pi D^2/4} = \dfrac{0.400\,\text{m}^3/\text{s}}{\pi(0.60\,\text{m})^2/4} = 1.415\,\text{m/s}$

Discussion The Reynolds number of flow through the pipe is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(998\,\text{kg/m}^3)(1.415\,\text{m/s})(0.60\,\text{m})}{1.002\times10^{-3}\,\text{kg/m}\cdot\text{s}} = 8.46\times10^5$$

Substituting β and Re values into the orifice discharge coefficient relation

$$C_d = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 + \frac{91.71\beta^{2.5}}{\text{Re}^{0.75}}$$

gives C_d = 0.603, which is very close to the assumed value of 0.61.

8-114

Solution We are to calculate the volume flow rate of oil through a pipe.

Assumptions **1** The Pitot-static probe is located at a location in the pipe where the local $V = V_{avg}$.

Properties The density of the oil is given as $\rho = 860$ kg/m^3, and the viscosity is given as $\mu = 0.0103$ kg/m·s.

Analysis Volume flow rate is equal to average velocity times cross-sectional area, and the average velocity is assumed to equal the measured local velocity, as obtained from the Pitot formula. Thus,

$$\dot{V} = \pi R^2 V_{avg} = \pi R^2 \sqrt{\frac{2\Delta P}{\rho}} = \pi (0.025\text{ m})^2 \sqrt{\frac{2(95.8\text{ Pa})}{860\text{ kg/m}^3}\left(\frac{1\text{ N/m}^2}{1\text{ Pa}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}$$

$$= 9.2678\times10^{-4}\ \frac{\text{m}^3}{\text{s}} \cong \mathbf{9.27\times10^{-4}\ \frac{m^3}{s}}$$

where we give our final result to three significant digits.

Discussion Since the velocity profile is not uniform throughout the pipe, the result is dependent on the location of the probe. If the probe is not located properly, the measured volume flow rate will be in error. A more sophisticated way to measure the volume flow rate would be to traverse the probe across the entire pipe (or half of it, assuming axial symmetry) and then integrate.

8-115

Solution We are to calculate the Reynolds number for flow through a pipe, and determine if it is laminar or turbulent.

Assumptions **1** The Pitot-static probe is located at a location in the pipe where the local $V = V_{avg}$.

Properties The density of the oil is given as $\rho = 860$ kg/m^3, and the viscosity is given as $\mu = 0.0103$ kg/m·s.

Analysis The average velocity is obtained from the Pitot formula. Thus,

$$\text{Re} = \frac{\rho D V_{avg}}{\mu} = \frac{\rho 2R\sqrt{2\Delta P/\rho}}{\mu} = \frac{2R\sqrt{2\rho\Delta P}}{\mu}$$

$$= \frac{2(0.025\text{ m})\sqrt{2(860\text{ kg/m}^3)(95.8\text{ Pa})\left(\frac{1\text{ N/m}^2}{1\text{ Pa}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}}{0.0103\text{ kg/m}\cdot\text{s}} = \mathbf{1970}$$

where we give our final result to three significant digits. Since Re < 2300, this flow is most likely **laminar**.

Discussion If the pipe were very rough or there were severe disturbances, this flow might be transitional.

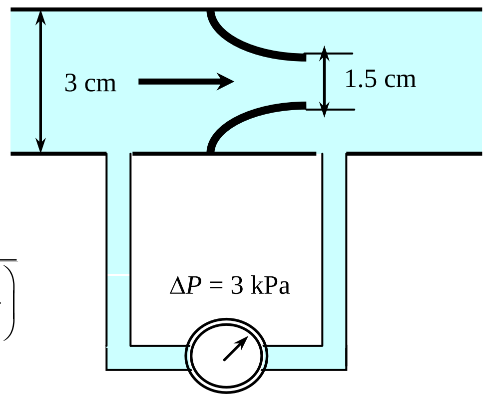
8-116

Solution The flow rate of water is to be measured with flow nozzle equipped with a differential pressure gage. For a given pressure drop, the flow rate, the average flow velocity, and head loss are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The discharge coefficient of the nozzle meter is C_d = 0.96.

Properties The density and dynamic viscosity of water are given to be ρ = 999.7 kg/m^3 and μ = 1.307×10^{-3} kg/m·s, respectively.

Analysis The diameter ratio and the throat area of the meter are



$$\beta = d/D = 1.5/3 = 0.50$$

$$A_0 = \pi d^2/4 = \pi(0.015\text{ m})^2/4 = 1.767\times10^{-4}\text{ m}^2$$

Noting that ΔP = 4 kPa = 4000 N/m^2, the flow rate becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1-P_2)}{\rho(1-\beta^4)}}$$

$$= (1.767\times10^{-4}\text{ m}^2)(0.96)\sqrt{\frac{2\times3000\text{ N/m}^2}{(999.7\text{ kg/m}^3)((1-0.50^4)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}$$

$$= 0.429\times10^{-3}\text{ m}^3\text{/s}$$

which is equivalent to 0.429 L/s. The average flow velocity in the pipe is determined by dividing the flow rate by the cross-sectional area of the pipe,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.429\times10^{-3}\text{ m}^3\text{/s}}{\pi(0.03\text{ m})^2/4} = 0.607\text{ m/s}$$

The Reynolds number of flow through the pipe is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(999.7\text{ kg/m}^3)(0.607\text{ m/s})(0.03\text{ m})}{1.307\times10^{-3}\text{ kg/m}\cdot\text{s}} = 1.39\times10^4$$

Substituting the β and Re values into the orifice discharge coefficient relation gives

$$C_d = 0.9975 - \frac{6.53\beta^{0.5}}{\text{Re}^{0.5}} = 0.9975 - \frac{6.53(0.50)^{0.5}}{(1.39\times10^4)^{0.5}} = 0.958$$

which is close to the assumed value of 0.96. We do some iteration to obtain a more precise answer: Using this revised value of C_d we obtain $\dot{V} = 0.4284\times10^{-3}\text{ m}^3\text{/s}$, leading to a revised V and Re of 0.60602 m/s and 13906 respectively. This value of Re yields C_d = 0.9583. Another iteration yields $\dot{V} = 0.4285\times10^{-3}\text{ m}^3\text{/s}$, V = 0.60624 m/s, Re = 13911, and C_d = 0.95835. You can see that the convergence is rapid. After one final iteration to make sure we have converged enough, we give the final results: $\dot{V} = 0.42853\times10^{-3}\text{ m}^3\text{/s}$, V = 0.60625 m/s, Re = 13911, and C_d = 0.95835. Thus we have converged to 5 significant digits – way more than we need. The final answers to 3 significant digits are $\dot{V} = \mathbf{0.429\times10^{-3}\text{ m}^3\text{/s}}$, ***V* = 0.606 m/s**, Re = 139001, and C_d = 0.958.

Discussion The water column height corresponding to a pressure drop of 3 kPa is

$$h_w = \frac{\Delta P}{\rho_w g} = \frac{3000\text{ kg}\cdot\text{m/s}^2}{(999.7\text{ kg/m}^3)(9.81\text{ m/s}^2)} = 0.306\text{ m}$$

The head loss between the two measurement sections is determined from the energy equation, which for $z_1 = z_2$ simplifies to

$$h_L = \frac{P_1-P_2}{\rho_f g} - \frac{V_2^2-V_1^2}{2g} = h_w - \frac{[(D/d)^4-1]V_1^2}{2g} = 0.306\text{ m} - \frac{[(3/1.5)^4-1](0.607\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 0.024\text{ m H2O}$$

Note, however, that this is not the same as the total irreversible head loss through the entire flow nozzle (which can be thought of as a type of minor loss in the piping system). The total irreversible head loss would be much higher than that calculated here because losses downstream of the nozzle exit plane, where there is turbulent mixing and flow separation.

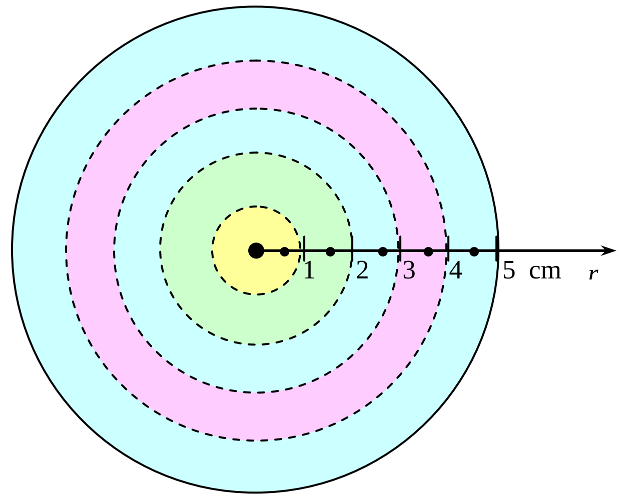
8-117

Solution The flow rate of water through a circular pipe is to be determined by measuring the water velocity at several locations along a cross-section. For a given set of measurements, the flow rate is to be determined.

Assumptions The points of measurements are sufficiently close so that the variation of velocity between points can be assumed to be linear.

Analysis The velocity measurements are given to be

R, cm	*V*, m/s
0	6.4
1	6.1
2	5.2
3	4.4
4	2.0
5	0.0



The divide the cross-section of the pipe into 1-cm thick annual regions, as shown in the figure. Using midpoint velocity values for each section, the flow rate is determined to be

$$\dot{V} = \int_{A_c} V dA_c \cong \sum V\pi(r_{out}^2 - r_{in}^2)$$

$$= \pi\left(\frac{6.4+6.1}{2}\right)(0.01^2 - 0) + \pi\left(\frac{6.1+5.2}{2}\right)(0.02^2 - 0.01^2) + \pi\left(\frac{5.2+4.4}{2}\right)(0.03^2 - 0.02^2)$$

$$+\pi\left(\frac{4.4+2.0}{2}\right)(0.04^2 - 0.02^2) + \pi\left(\frac{2.0+0}{2}\right)(0.05^2 - 0.04^2)$$

$$= \mathbf{0.0297\ m^3/s}$$

Discussion We can also solve this problem by curve-fitting the given data using a second-degree polynomial, and then performing the integration.

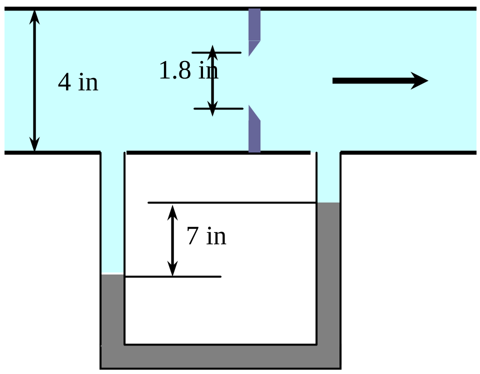
8-118E

Solution The flow rate of water is to be measured with an orifice meter. For a given pressure drop across the orifice plate, the flow rate, the average velocity, and the head loss are to be determined.

Assumptions **1** The flow is steady and incompressible.

Properties The density and dynamic viscosity of water are given to be ρ = 62.36 kg/m^3 and μ = 7.536×10^{-4} lbm/ft·s, respectively. We take the density of mercury to be 847 lbm/ft^3.

Analysis The diameter ratio and the throat area of the orifice are



$$\beta = d/D = 1.8/4 = 0.45$$

$$A_0 = \pi d^2/4 = \pi(1.8/12\text{ ft})^2/4 = 0.017671\text{ ft}^2$$

The pressure drop across the orifice plate can be expressed as

$$\Delta P = P_1 - P_2 = (\rho_{\text{Hg}} - \rho_{\text{f}})gh$$

Then the flow rate relation for obstruction meters becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}} = A_o C_d \sqrt{\frac{2(\rho_{\text{Hg}} - \rho_{\text{f}})gh}{\rho_{\text{f}}(1-\beta^4)}} = A_o C_d \sqrt{\frac{2(\rho_{\text{Hg}}/\rho_{\text{f}} - 1)gh}{1-\beta^4}}$$

As a first quess, the discharge coefficient of the orifice meter is approximated as C_d = 0.61. Substituting, the flow rate is determined to be

$$\dot{V} = (0.017671\text{ ft}^2)(0.61)\sqrt{\frac{2(847/62.36-1)(32.2\text{ ft/s}^2)(7/12\text{ ft})}{1-0.45^4}} = 0.2393\text{ ft}^3\text{/s}$$

The average velocity in the pipe is determined by dividing the flow rate by the cross-sectional area of the pipe,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.2393\text{ ft}^3\text{/s}}{\pi(4/12\text{ ft})^2/4} = 2.742\text{ ft/s}$$

The head loss between the two measurement sections can be estimated from the energy equation. Since $z_1 = z_2$, the head form of the energy equation simplifies to

$$h_L \approx \frac{P_1 - P_2}{\rho_f g} - \frac{V_2^2 - V_1^2}{2g} = \frac{\rho_{Hg}\cancel{g}h_{Hg}}{\rho_f \cancel{g}} - \frac{[(D/d)^4 - 1]V_1^2}{2g} = 13.6 \times \frac{7}{12}\text{ ft} - \frac{[(4/1.8)^4 - 1](2.742\text{ ft/s})^2}{2(32.2\text{ ft/s}^2)} = 5.20\text{ ft H}_2\text{O}$$

The Reynolds number of flow through the pipe is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(62.36\text{ lbm/ft}^3)(2.742\text{ ft/s})(4/12\text{ ft})}{7.536\times10^{-4}\text{ lbm/ft}\cdot\text{s}} = 7.56\times10^4$$

The values calculated above are reasonable approximations, but are not exact since we guessed the value of the discharge coefficient. We correct by substituting β and Re values into the orifice discharge coefficient relation,

$$C_d = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 + \frac{91.71\beta^{2.5}}{\text{Re}^{0.75}} = 0.6042$$

Using this refined value of C_d, the flow rate becomes 0.2370 ft^3/s, which differs from our original result by less than 1%. Plugging in this volume flow rate, we get V = 2.716 ft/s, and h_L = 5.255 ft H_2O, Re = 74920, and the new C_d = 0.6042. our final answers to three digits are thus $\dot{V}$ = **0.237 ft^3/s**, ***V*** **= 2.72 ft/s**, and h_L **= 5.26 ft H_2O.**

Discussion If we do not iterate, and just assume that C_d = 0.61 for the discharge coefficient, the results differ by about 1%. In some engineering analyses, this is a negligible error, but to be consistent and as accurate as possible, we should iterate as shown here.

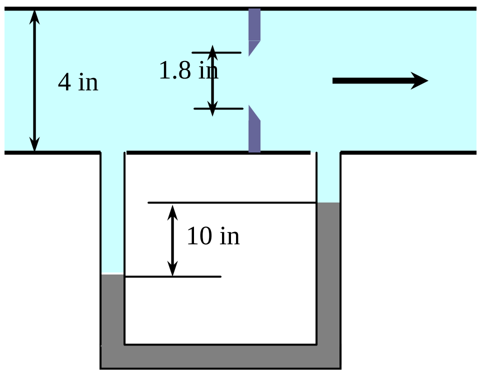
8-119E

Solution The flow rate of water is to be measured with an orifice meter. For a given pressure drop across the orifice plate, the flow rate, the average velocity, and the head loss are to be determined.

Assumptions **1** The flow is steady and incompressible.

Properties The density and dynamic viscosity of water are given to be ρ = 62.36 kg/m^3 and μ = 7.536×10^{-4} lbm/ft·s, respectively. We take the density of mercury to be 847 lbm/ft^3.

Analysis The diameter ratio and the throat area of the orifice are



$$\beta = d/D = 1.8/4 = 0.45$$

$$A_0 = \pi d^2/4 = \pi(1.8/12\text{ ft})^2/4 = 0.01767\text{ ft}^2$$

$$A_c = \pi D^2/4 = \pi(4/12\text{ ft})^2/4 = 0.087266\text{ ft}^2$$

The pressure drop across the orifice plate can be expressed as

$$\Delta P = P_1 - P_2 = (\rho_{\text{Hg}} - \rho_{\text{f}})gh$$

Then the flow rate relation for obstruction meters becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}} = A_o C_d \sqrt{\frac{2(\rho_{\text{Hg}} - \rho_{\text{f}})gh}{\rho_{\text{f}}(1-\beta^4)}} = A_o C_d \sqrt{\frac{2(\rho_{\text{Hg}}/\rho_{\text{f}} - 1)gh}{1-\beta^4}}$$

As a first quess, the discharge coefficient of the orifice meter is approximated as C_d = 0.61. Substituting, the flow rate is determined to be

$$\dot{V} = (0.01767\text{ ft}^2)(0.61)\sqrt{\frac{2(847/62.36 - 1)(32.2\text{ ft/s}^2)(10/12\text{ ft})}{1-0.45^4}} = 0.2860\text{ ft}^3\text{/s}$$

The average velocity in the pipe is determined by dividing the flow rate by the cross-sectional area of the pipe,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.2860\text{ ft}^3\text{/s}}{\pi(4/12\text{ ft})^2/4} = 3.278\text{ ft/s}$$

The head loss between the two measurement sections can be estimated from the energy equation. Since $z_1 = z_2$, the head form of the energy equation simplifies to

$$h_L \approx \frac{P_1 - P_2}{\rho_f g} - \frac{V_2^2 - V_1^2}{2g} = \frac{\rho_{Hg} g h_{Hg}}{\rho_f g} - \frac{[(D/d)^4 - 1]V_1^2}{2g} = 13.6 \times \left(\frac{10}{12}\text{ft}\right) - \frac{[(4/1.8)^4 - 1](3.278\text{ ft/s})^2}{2(32.2\text{ ft/s}^2)} = 7.431\text{ ft H}_2\text{O}$$

The Reynolds number of flow through the pipe is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(62.36\text{ lbm/ft}^3)(3.278\text{ ft/s})(4/12\text{ ft})}{7.536\times10^{-4}\text{ lbm/ft}\cdot\text{s}} = 9.042\times10^4$$

The values calculated above are reasonable approximations, but are not exact since we guessed the value of the discharge coefficient. We correct by substituting β and Re values into the orifice discharge coefficient relation,

$$C_d = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 + \frac{91.71\beta^{2.5}}{\text{Re}^{0.75}} = 0.6038$$

Using this refined value of C_d, the flow rate becomes 0.2831 ft^3/s, which differs from our original result by less than 1%. Plugging in this volume flow rate, we get V = 3.244 ft/s, and h_L = 7.511 ft H_2O, Re = 89490, and the new C_d = 0.6038. our final answers to three digits are thus $\dot{V}$ = **0.283 ft^3/s**, ***V*** **= 3.24 ft/s**, and ***h***$_L$ **= 7.51 ft H$_2$O.**

Discussion If we do not iterate, and just assume that C_d = 0.61 for the discharge coefficient, the results differ by about 1%. In some engineering analyses, this is a negligible error, but to be consistent and as accurate as possible, we should iterate as shown here.

8-120

Solution We are to calculate the velocity measured by a Pitot-static probe.

Assumptions **1** The flow is steady and incompressible. **2** For the analytical analysis, we neglect irreversibilities such as friction so that the Bernoulli approximation can be used.

Analysis We apply the equation given in the text for a Pitot-static probe,

$$V = \sqrt{\frac{2\left(P_{\text{stag}} - P\right)}{\rho}}$$

For the given conditions we get

$$V = \sqrt{\frac{2(P_1 - P_2)}{\rho}} = \sqrt{\frac{2(472.6 - 15.43)\ \text{N/m}^2}{1.225\ \text{kg/m}^3}\left(\frac{1\ \text{kg}\cdot\text{m/s}^3}{1\ \text{N}}\right)} = 19.32\ \text{m/s} \cong \mathbf{19.3\ m/s}$$

Discussion We give the final result to three significant digits. We must assume that the probe is aligned directly into the oncoming flow; otherwise the measured pressures would be incorrect.

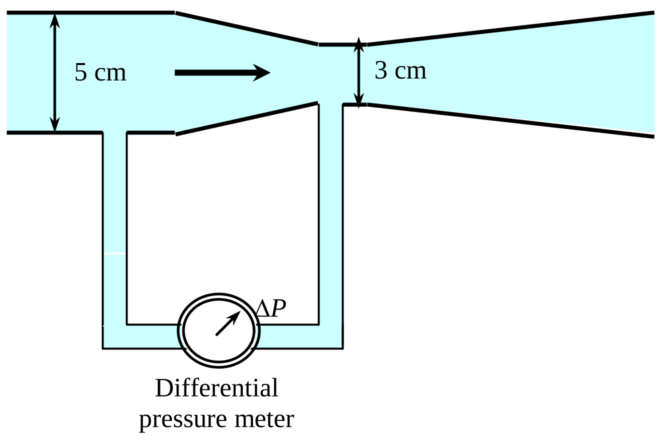
8-121

Solution A Venturi meter equipped with a differential pressure gage is used to measure to flow rate of water through a horizontal pipe. For a given pressure drop, the volume flow rate of water and the average velocity through the pipe are to be determined.

Assumptions The flow is steady and incompressible.

Properties The density of water is given to be ρ = 999.1 kg/m^3. The discharge coefficient of Venturi meter is given to be C_d = 0.98.

Analysis The diameter ratio and the throat area of the meter are



$$\beta = d/D = 3/5 = 0.60$$

$$A_0 = \pi d^2/4 = \pi(0.03\ \text{m})^2/4 = 7.069\times10^{-4}\ \text{m}^2$$

Noting that ΔP = 5 kPa = 5000 N/m^2, the flow rate becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}}$$

$$= (7.069\times10^{-4}\ \text{m}^2)(0.98)\sqrt{\frac{2\times5000\ \text{N/m}^2}{(999.1\ \text{kg/m}^3)((1-0.60^4)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)}$$

$$= \mathbf{0.00235\ m^3/s} = \mathbf{2.35\ L/s}$$

which is equivalent to 2.35 L/s. The average flow velocity in the pipe is determined by dividing the flow rate by the cross-sectional area of the pipe,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.00235\ \text{m}^3/\text{s}}{\pi(0.05\ \text{m})^2/4} = \mathbf{1.20\ m/s}$$

Discussion Note that the flow rate is proportional to the square root of pressure difference across the Venturi meter.

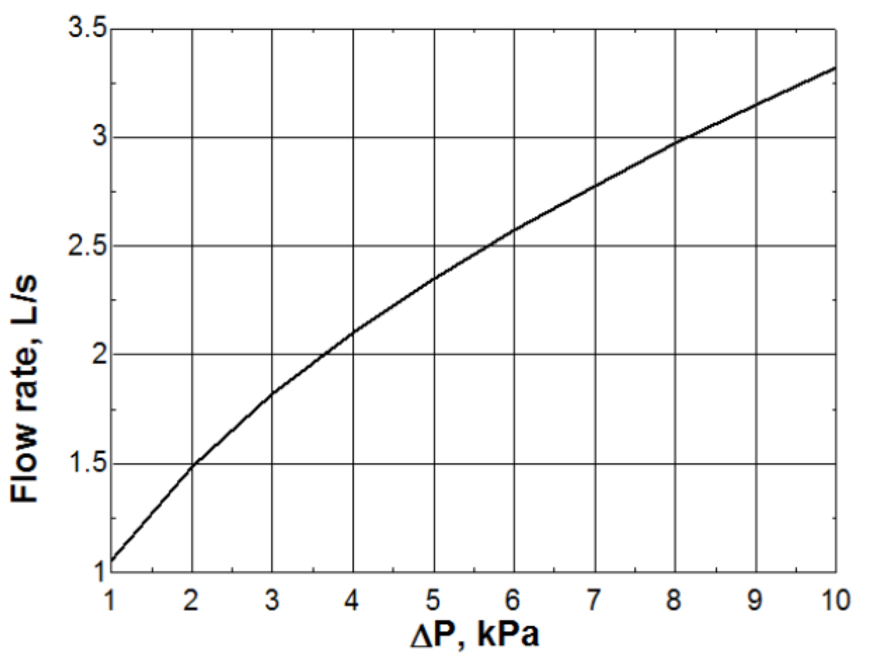
8-122

Solution The previous problem is reconsidered. The variation of flow rate as the pressure drop varies from 1 kPa to 10 kPa at intervals of 1 kPa is to be investigated, and the results are to be plotted.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
rho=999.1 "kg/m3"
D=0.05 "m"
d0=0.03 "m"
beta=d0/D
A0=pi*d0^2/4
Cd=0.98
Vol=A0*Cd*SQRT(2*DeltaP*1000/(rho*(1-beta^4)))*1000 "L/s"
```

Pressure Drop ΔP, kPa	Flow rate L/s
1	1.05
2	1.49
3	1.82
4	2.10
5	2.35
6	2.57
7	2.78
8	2.97
9	3.15
10	3.32

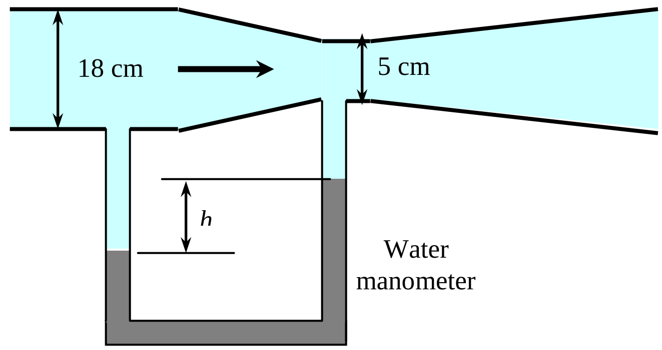


Discussion This type of plot can be thought of as a calibration plot for the flowmeter, although a real calibration plot would use actual experimental data rather than data from equations. It would be interesting to compare the above plot to experimental data to see how close the predictions are.

8-123

Solution A Venturi meter equipped with a water manometer is used to measure to flow rate of air through a duct. For a specified maximum differential height for the manometer, the maximum mass flow rate of air that can be measured is to be determined.

Assumptions The flow is steady and incompressible.



Properties The density of air is given to be ρ_{air} = 1.204 kg/m^3. We take the density of water to be ρ_w = 1000 kg/m^3. The discharge coefficient of Venturi meter is given to be C_d = 0.98.

Analysis The diameter ratio and the throat area of the meter are

$$\beta = d/D = 5/18 = 0.2778$$

$$A_0 = \pi d^2/4 = \pi(0.05\,\text{m})^2/4 = 0.001963\,\text{m}^2$$

The pressure drop across the Venturi meter can be expressed as

$$\Delta P = P_1 - P_2 = (\rho_w - \rho_f)gh$$

Then the flow rate relation for obstruction meters becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}} = A_o C_d \sqrt{\frac{2(\rho_w - \rho_f)gh}{\rho_f(1-\beta^4)}} = A_o C_d \sqrt{\frac{2(\rho_w/\rho_{air} - 1)gh}{1-\beta^4}}$$

Substituting and using h = 0.40 m, the maximum volume flow rate is determined to be

$$\dot{V} = (0.001963\,\text{m}^2)(0.98)\sqrt{\frac{2(1000/1.204 - 1)(9.81\,\text{m/s}^2)(0.40\,\text{m})}{1-0.2778^4}} = 0.1557\,\text{m}^3/s$$

Then the maximum mass flow rate this Venturi meter can measure is

$$\dot{m} = \rho\dot{V} = (1.204\,\text{kg/m}^3)(0.1557\,\text{m}^3/\text{s}) = 0.1875\,\text{kg/s} \cong \mathbf{0.188\,kg/s}$$

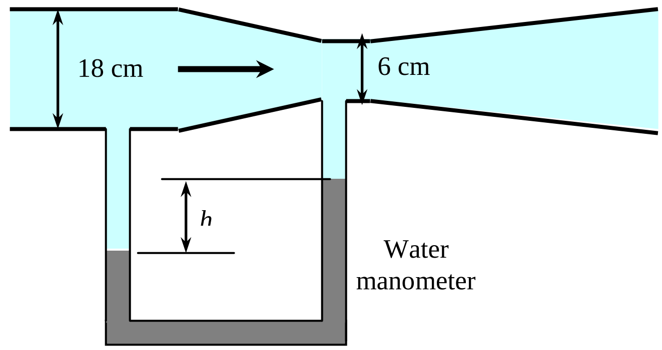
Also, the average flow velocity in the duct is

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.1557\,\text{m}^3/\text{s}}{\pi(0.18\,\text{m})^2/4} = 6.12\,\text{m/s}$$

Discussion Note that the maximum available differential height limits the flow rates that can be measured with a manometer.

8-124

Solution A Venturi meter equipped with a water manometer is used to measure to flow rate of air through a duct. For a specified maximum differential height for the manometer, the maximum mass flow rate of air that can be measured is to be determined.



Assumptions The flow is steady and incompressible.

Properties The density of air is given to be ρ_{air} = 1.204 kg/m^3. We take the density of water to be ρ_w = 1000 kg/m^3. The discharge coefficient of Venturi meter is given to be C_d = 0.98.

Analysis The diameter ratio and the throat area of the meter are

$$\beta = d/D = 6/18 = 0.3333$$

$$A_0 = \pi d^2/4 = \pi(0.06\text{ m})^2/4 = 0.002827\text{ m}^2$$

The pressure drop across the Venturi meter can be expressed as

$$\Delta P = P_1 - P_2 = (\rho_w - \rho_f)gh$$

Then the flow rate relation for obstruction meters becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1-P_2)}{\rho(1-\beta^4)}} = A_o C_d \sqrt{\frac{2(\rho_w-\rho_f)gh}{\rho_f(1-\beta^4)}} = A_o C_d \sqrt{\frac{2(\rho_w/\rho_{air}-1)gh}{1-\beta^4}}$$

Substituting and using h = 0.40 m, the maximum volume flow rate is determined to be

$$\dot{V} = (0.002827\text{ m}^2)(0.98)\sqrt{\frac{2(1000/1.204-1)(9.81\text{ m/s}^2)(0.40\text{ m})}{1-0.3333^4}} = 0.2250\text{ m}^3/s$$

Then the maximum mass flow rate this Venturi meter can measure is

$$\dot{m} = \rho\dot{V} = (1.204\text{ kg/m}^3)(0.2250\text{ m}^3\text{/s}) = 0.2709\text{ kg/s} \cong \mathbf{0.271\ kg/s}$$

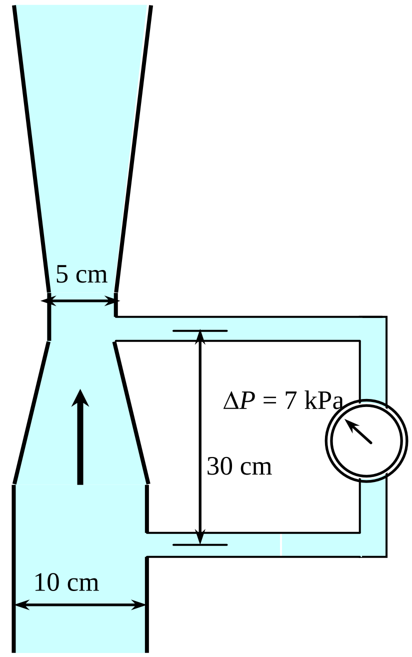
Also, the average flow velocity in the duct is

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.2250\text{ m}^3/\text{s}}{\pi(0.18\text{ m})^2/4} = 8.84\text{ m/s}$$

Discussion Note that the maximum available differential height limits the flow rates that can be measured with a manometer.

8-125

Solution A Venturi meter equipped with a differential pressure gage is used to measure the flow rate of liquid propane through a vertical pipe. For a given pressure drop, the volume flow rate is to be determined.



Assumptions The flow is steady and incompressible.

Properties The density of propane is given to be ρ = 514.7 kg/m^3. The discharge coefficient of Venturi meter is given to be C_d = 0.98.

Analysis The diameter ratio and the throat area of the meter are

$$\beta = d/D = 5/10 = 0.5$$

$$A_0 = \pi d^2/4 = \pi(0.05\,\text{m})^2/4 = 0.001963\,\text{m}^2$$

Noting that ΔP = 7 kPa = 7000 N/m^2, the flow rate becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}}$$

$$= (0.001963\,\text{m}^2)(0.98)\sqrt{\frac{2\times 7000\,\text{N/m}^2}{(514.7\,\text{kg/m}^3)((1-0.5^4)}\left(\frac{1\,\text{kg}\cdot\text{m/s}^2}{1\,\text{N}}\right)}$$

$$= \mathbf{0.0104\,m^3/s}$$

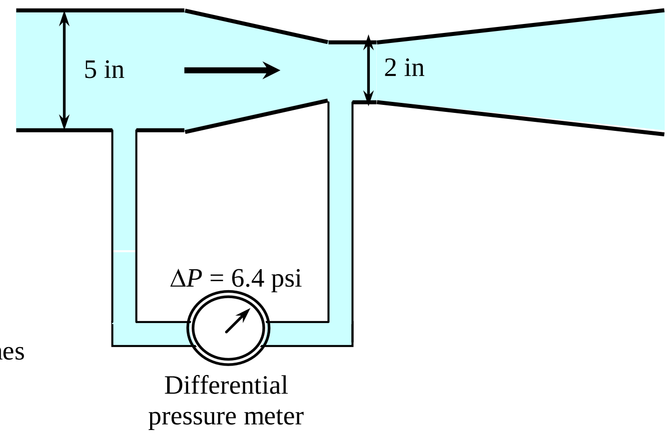
which is equivalent to 10.9 L/s. Also, the average flow velocity in the pipe is

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.0104\,\text{m}^3/\text{s}}{\pi(0.10\,\text{m})^2/4} = 1.32\,\text{m/s}$$

Discussion Note that the elevation difference between the locations of the two probes does not enter the analysis since the pressure gage measures the pressure differential at a specified location. When there is no flow through the Venturi meter, for example, the pressure gage would read zero.

8-126E

Solution A Venturi meter equipped with a differential pressure meter is used to measure to flow rate of refrigerant-134a through a horizontal pipe. For a measured pressure drop, the volume flow rate is to be determined.



Assumptions The flow is steady and incompressible.

Properties The density of R-134a is given to be ρ = 83.31 lbm/ft^3. The discharge coefficient of Venturi meter is given to be C_d = 0.98.

Analysis The diameter ratio and the throat area of the meter are

$$\beta = d/D = 2/5 = 0.40$$

$$A_0 = \pi d^2/4 = \pi(2/12\text{ ft})^2/4 = 0.02182\text{ ft}^2$$

Noting that ΔP = 6.4 psi = 6.4×144 lbf/ft^2, the flow rate becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}}$$

$$= (0.02182\text{ ft}^2)(0.98)\sqrt{\frac{2\times 6.4\times 144\text{ lbf/ft}^2}{(83.31\text{ lbm/ft}^3)((1-0.40^4)}\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)}$$

$$= \mathbf{0.578\ ft^3/s}$$

Also, the average flow velocity in the pipe is determined by dividing the flow rate by the cross-sectional area of the pipe,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.578\text{ ft}^3/\text{s}}{\pi(5/12\text{ ft})^2/4} = \mathbf{4.24\ ft/s}$$

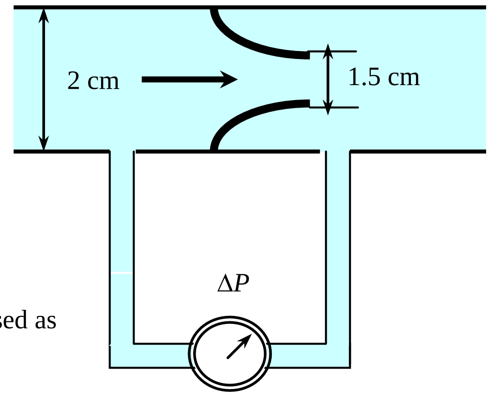
Discussion Note that the flow rate is proportional to the square root of pressure difference across the Venturi meter.

8-127

Solution A kerosene tank is filled with a hose equipped with a nozzle meter. For a specified filling time, the pressure difference indicated by the nozzle meter is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The discharge coefficient of the nozzle meter is C_d = 0.96.

Properties The density of kerosene is given to be ρ = 820 kg/m^3.



Analysis The diameter ratio and the throat area of the meter are

$$\beta = d/D = 1.5/2 = 0.75$$

$$A_0 = \pi d^2/4 = \pi(0.015\text{ m})^2/4 = 1.767\times10^{-4}\text{ m}^2$$

To fill a 16-L tank in 20 s, the flow rate must be

$$\dot{V} = \frac{V_{\text{tank}}}{\Delta t} = \frac{22\text{ L}}{20\text{ s}} = 1.1\text{ L/s}$$

For a pressure drop of $\Delta P = P_1 - P_2$ across the meter, the flow rate is expressed as

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}}$$

Substituting,

$$0.0011\text{m}^3/s = (1.767\times10^{-4}\text{ m}^2)(0.96)\sqrt{\frac{2\Delta P}{(820\text{ kg/m}^3)((1-0.75^4)}}$$

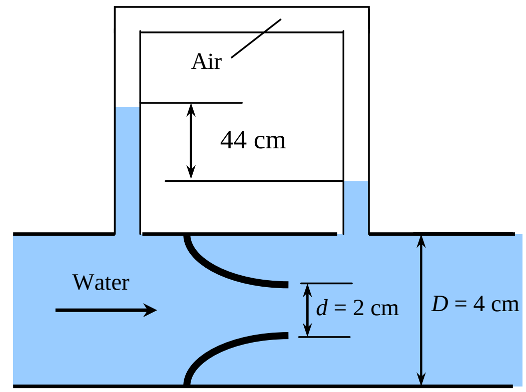
which gives the pressure drop across the meter to be

$$\Delta P = 11{,}780\text{ kg}\cdot\text{m/s}^2 = \mathbf{11.8kPa}$$

Discussion Note that the flow rate is proportional to the square root of pressure difference across the nozzle meter.

8-128

Solution The flow rate of water is to be measured with flow nozzle equipped with an inverted air-water manometer. For a given differential height, the flow rate and head loss caused by the nozzle meter are to be determined.



Assumptions **1** The flow is steady and incompressible. **2** The discharge coefficient of the nozzle meter is C_d = 0.96.

Properties The density and dynamic viscosity of water are given to be ρ = 998 kg/m^3 and μ = 1.002×10^{-3} kg/m·s, respectively. The density of the air in the manometer is assumed to by 1.20 kg/m^3.

Analysis The diameter ratio and the throat area of the meter are

$$\beta = d / D = 2/4 = 0.50$$

$$A_0 = \pi d^2/4 = \pi(0.02\text{ m})^2/4 = 3.142\times10^{-4}\text{ m}^2$$

We use manometry to calculate ΔP, and we need to iterate since we do not know the exact value of discharge coefficient. As a first guess, we set C_d = 0.96, and the flow rate becomes

$$\dot{V} = A_o C_d\sqrt{\frac{2(P_1-P_2)}{\rho(1-\beta^4)}} = A_o C_d\sqrt{\frac{2(\rho_w-\rho_{air})}{\rho_w(1-\beta^4)}}$$

$$= \frac{\pi(0.020\text{ m})^2}{4}(0.96)\sqrt{\frac{2(998-1.20)\text{ kg/m}^3(9.81\text{ m/s}^2)(0.44\text{ m})}{(998\text{ kg/m}^3)((1-0.50^4)}}$$

$$= 0.0009146\text{ m}^3/\text{s}$$

which is equivalent to about 0.915 L/s. The average flow velocity in the pipe is

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.0009146\text{ m}^3/\text{s}}{\pi(0.040\text{ m})^2/4} = 0.7278\text{ m/s}$$

The Reynolds number of flow through the pipe is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(998\text{ kg/m}^3)(0.7278\text{ m/s})(0.04\text{ m})}{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}} = 2.900\times10^4$$

Substituting the β and Re values into the orifice discharge coefficient relation gives

$$C_d = 0.9975 - \frac{6.53\beta^{0.5}}{\text{Re}^{0.5}} = 0.9975 - \frac{6.53(0.50)^{0.5}}{(2.900\times10^4)^{0.5}} = 0.9704$$

This is a corrected value of discharge coefficient. It is close to our original guess of 0.96, but is more accurate. We iterate another time using this corrected value of C_d, yielding $\dot{V}$ = 0.0009245 m^3/s from which V = 0.7357 m/s, and Re = 2.931×10^4. Using this corrected value of Re, we get C_d = 0.9705. One further iteration reveals that the solution has converged. The final results are V = 0.736 m/s, Re = 2.93×10^4, C_d = 0.971, and $\dot{V}$ = **0.000925 m^3/s** (to 3 significant digits each).

The head loss between the two measurement sections can be determined from the energy equation. Since $z_1 = z_2$, the energy equation simplifies to

$$h_L = \frac{P_1-P_2}{\rho_f g} - \frac{V_2^2 - V_1^2}{2g} = h_w - \frac{[(D/d)^4 - 1]V_1^2}{2g} = 0.44\text{ m} - \frac{[(4/2)^4-1](0.736\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 0.0259\text{ m H2O}$$

So, the head loss is $\boldsymbol{h_L}$ **= 0.0259 m of water** (to three significant digits).

Discussion You could save some calculations by ignoring the air density compared to water and by not iterating, but the results would not be as accurate.

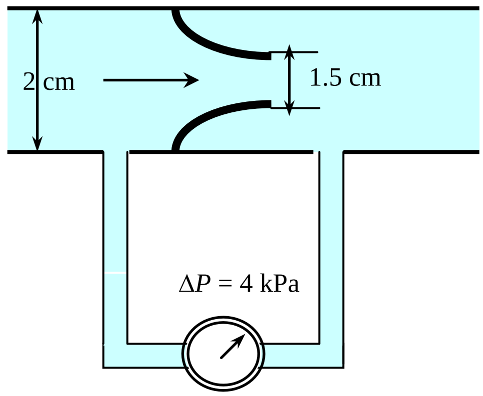
8-129

Solution The flow rate of ammonia is to be measured with flow nozzle equipped with a differential pressure gage. For a given pressure drop, the flow rate and the average flow velocity are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The discharge coefficient of the nozzle meter is C_d = 0.96.

Properties The density and dynamic viscosity of ammonia are given to be ρ = 624.6 kg/m^3 and $\mu = 1.697\times10^{-4}$ kg/m·s, respectively.

Analysis The diameter ratio and the throat area of the meter are



$$\beta = d/D = 1.5/2 = 0.75$$

$$A_0 = \pi d^2/4 = \pi(0.015\text{ m})^2/4 = 1.767\times10^{-4}\text{ m}^2$$

Noting that ΔP = 4 kPa = 4000 N/m^2, the flow rate becomes

$$\dot{V} = A_o C_d\sqrt{\frac{2(P_1-P_2)}{\rho(1-\beta^4)}}$$

$$= (1.767\times10^{-4}\text{ m}^2)(0.96)\sqrt{\frac{2\times4000\text{ N/m}^2}{(624.6\text{ kg/m}^3)(1-0.75^4)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}$$

$$= \mathbf{0.734\times10^{-3}\ m^3/s}$$

which is equivalent to 0.734 L/s. The average flow velocity in the pipe is determined by dividing the flow rate by the cross-sectional area of the pipe,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.734\times10^{-3}\text{ m}^3/\text{s}}{\pi(0.02\text{ m})^2/4} = \mathbf{2.34\ m/s}$$

Discussion The Reynolds number of flow through the pipe is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(624.6\text{ kg/m}^3)(2.34\text{ m/s})(0.02\text{ m})}{1.697\times10^{-4}\text{ kg/m}\cdot\text{s}} = 1.72\times10^5$$

Substituting the β and Re values into the orifice discharge coefficient relation gives

$$C_d = 0.9975 - \frac{6.53\beta^{0.5}}{\text{Re}^{0.5}} = 0.9975 - \frac{6.53(0.75)^{0.5}}{(1.72\times10^5)^{0.5}} = 0.984$$

which is about 2% different than the assumed value of 0.96. Using this refined value of C_d, the flow rate becomes 0.752 L/s, which differs from our original result by only 2.4%. If the problem is solved using an equation solver such as EES, then the problem can be formulated using the curve-fit formula for C_d (which depends on Reynolds number), and all equations can be solved simultaneously by letting the equation solver perform the iterations as necessary.

Review Problems

8-130

Solution The velocity and temperature profiles at a cross-section are given. A relation for the bulk fluid temperature at that cross section is to be obtained.

Analysis

$$T_m = \frac{\int_0^R U_{\max}\left(1 - r^2/R^2\right)\left(A + Br^2 - Cr^4\right)2\pi r dr}{\int_0^R U_{\max}\left(1 - r^2/R^2\right)2\pi r dr}$$

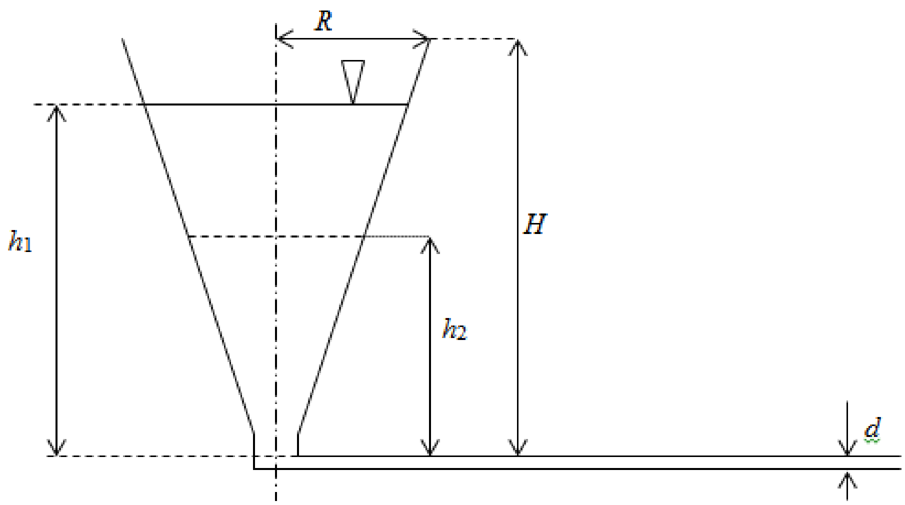
$$T_m = \frac{\dfrac{R^6 C}{8} + \dfrac{\left(-C - \dfrac{B}{R^2}\right)R^6}{6} + \dfrac{\left(B - \dfrac{A}{R^2}\right)R^4}{4} + \dfrac{AR^2}{2}}{\dfrac{R^2}{4}} = \frac{-\dfrac{R^2\left(R^4 C - 2BR^2 - 6A\right)}{24}}{\dfrac{R^2}{4}}$$

$$T_m = A + \frac{BR^2}{3} - \frac{CR^4}{6}$$

8-131

Solution A conical container is used to measure the viscosity of oil. An expression for the viscosity of oil in the container as a function of the discharge time is to be obtained.

Analysis



Bernoulli equation from the bottom of the container to the end of the pipe gives,

$$P_1 + \frac{1}{2}\rho V_1^2 + \gamma z_1 = P_2 + \frac{1}{2}\rho V_2^2 + \gamma z_2 + \lambda \frac{L}{d}\rho \frac{V^2}{2g}$$

$P_2=0$, $P_1=\gamma h$, $V_1=0$, $z_1=z_2$, and $V_1=V_2$.

Therefore; $\gamma h = \lambda \frac{L}{d}\rho \frac{V^2}{2g}$, $\lambda = \frac{64}{\mathrm{Re}} = \frac{64\mu}{\rho v}$

$$h = \frac{64\mu}{\rho v d}\frac{L}{d}\rho \frac{V^2}{2g} \rightarrow V = \frac{\gamma d^2 h}{32\mu L}$$

From the continuity;

$$-A_T dh = A_h V dt = A_h \frac{\gamma d^2 h}{32\mu L}$$

$$\frac{h}{H} = \frac{r}{R} \quad ; \quad r = \frac{R}{H}h \quad , \quad A_T = \pi r^2 \quad ; \quad A_T = \pi\left(\frac{R}{H}\right)^2 h^2$$

$$-\pi\left(\frac{R}{H}\right)^2 h^2 dh = A_h \frac{\gamma d^2 h}{32\mu L} dt$$

$$\int_{h_1}^{h_2} h dh = -A_h \frac{\gamma d^2 h}{32\pi\mu L (R/H)^2} dt = -\int_0^T K dt \quad \frac{1}{2}h^2 \Big|_{h_1}^{h_2} = -KT \quad or \quad \frac{1}{2}\left(h_1^2 - h_2^2\right) = KT$$

$$K = \frac{A_h \gamma d^2}{32\pi L (R/H)^2}\frac{1}{\mu} = \frac{h_1^2 - h_2^2}{2T}$$

Therefore;

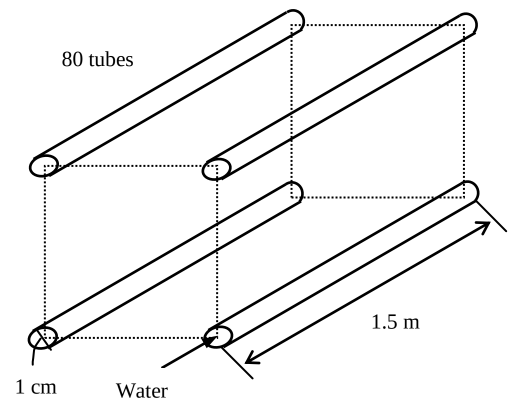
$$\mu = \frac{A_h \gamma d^2 T}{16\pi L (R/H)^2}\frac{1}{h_1^2 - h_2^2}$$

8-132

Solution Water is flowing through a brass tube bank of a heat exchanger at a specified flow rate. The pressure drop and the pumping power required are to be determined. Also, the percent reduction in the flow rate of water through the tubes is to be determined after scale build-up on the inner surfaces of the tubes.

Assumptions **1** The flow is steady, horizontal, and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed (this is a questionable assumption since the tubes are short, and it will be verified). **3** The inlet, exit, and header losses are negligible, and the tubes involve no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water at 20°C are ρ = 983.3 kg/m^3 and μ = 0.467×10^{-3} kg/m·s, respectively. The roughness of brass tubing is 1.5×10^{-6} m.



Analysis First we calculate the average velocity and the Reynolds number to determine the flow regime:

$$V=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{N_{\text{tune}}(\pi D^2/4)}=\frac{0.015\ \text{m}^3/\text{s}}{80[\pi(0.01\ \text{m})^2/4]}=2.387\ \text{m/s}$$

$$\text{Re}=\frac{\rho V D_h}{\mu}=\frac{(983.3\ \text{kg/m}^3)(2.387\ \text{m/s})(0.01\ \text{m})}{0.467\times10^{-3}\ \text{kg/m}\cdot\text{s}}=50{,}270$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{1.5\times10^{-6}\ \text{m}}{0.01\ \text{m}}=1.5\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\ \rightarrow\ \frac{1}{\sqrt{f}}=-2.0\log\left(\frac{1.5\times10^{-4}}{3.7}+\frac{2.51}{50{,}270\sqrt{f}}\right)$$

It gives f = 0.0214. Then the pressure drop, the head loss, and the useful pumping power required become

$$\Delta P=\Delta P_L=f\frac{L}{D}\frac{\rho V^2}{2}=0.0214\frac{1.5\ \text{m}}{0.01\ \text{m}}\frac{(983.3\ \text{kg/m}^3)(2.387\ \text{m/s})^2}{2}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right)=\mathbf{8.99\ kPa}$$

$$\dot{W}_{\text{pump}}=\dot{V}\Delta P=(0.015\ \text{m}^3/\text{s})(8.99\ \text{kPa})\left(\frac{1\ \text{kW}}{1\ \text{kPa}\cdot\text{m}^3/\text{s}}\right)=\mathbf{0.135\ kW}$$

Therefore, useful power input in the amount of 0.135 kW is needed to overcome the frictional losses in the tube. The hydrodynamic entry length in this case is

$$L_{\text{h,turbulent}}\approx10D=10(0.01\ \text{m})=0.1\ \text{m}$$

which is much less than 1.5 m. Therefore, the assumption of fully developed flow is valid. (The effect of the entry region is to increase the friction factor, and thus the pressure drop and pumping power).

After scale buildup: When 1-mm thick scale builds up on the inner surfaces (and thus the diameter is reduced to 0.8 cm from 1 cm) with an equivalent roughness of 0.4 mm, and the useful power input is fixed at 0.135 kW, the problem can be formulated as follows (note that the flow rate and thus the average velocity are unknown in this case):

$$V=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{N_{\text{tune}}(\pi D^2/4)}\ \rightarrow\ V=\frac{\dot{V}}{80[\pi(0.008\ \text{m})^2/4]}\qquad(1)$$

$$\text{Re}=\frac{\rho VD}{\mu}\ \rightarrow\ \text{Re}=\frac{(983.3\ \text{kg/m}^3)V(0.008\ \text{m})}{0.467\times10^{-3}\ \text{kg/m}\cdot\text{s}}\qquad(2)$$

$$\varepsilon / D = \frac{0.0004\text{ m}}{0.008\text{ m}} = 0.05$$

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{0.05}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \quad (3)$$

$$\Delta P = f\,\frac{L}{D}\frac{\rho V^2}{2} \quad \rightarrow \quad \Delta P = f\,\frac{1.5\text{ m}}{0.008\text{ m}}\frac{(983.3\text{ kg/m}^3)V^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) \quad (4)$$

$$\dot{W}_{\text{pump}} = 0.135\text{ kW} \rightarrow \dot{V}\Delta P = 0.135 \quad (5)$$

Solving this system of 5 equations in 5 unknown (f, Re, V, ΔP, and $\dot{V}$) using an equation solver (or a trial-and-error approach, by assuming a velocity value) gives

f = 0.0723, Re = 28,870, V = 1.714 m/s, ΔP = 19.6 kPa, and $\dot{V}$ = 0.00689 m^3/s = 6.89 L/s

Then the percent reduction in the flow rate becomes

$$\text{Reduction ratio} = \frac{\dot{V}_{\text{clean}} - \dot{V}_{\text{dirty}}}{\dot{V}_{\text{clean}}} = \frac{15 - 6.89}{15} = 0.54 = \mathbf{54\%}$$

Therefore, for the same pump input, the flow rate will be reduced to less than half of the original flow rate when the pipes were new and clean.

Discussion The friction factor could also be determined easily from the explicit Haaland relation. Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

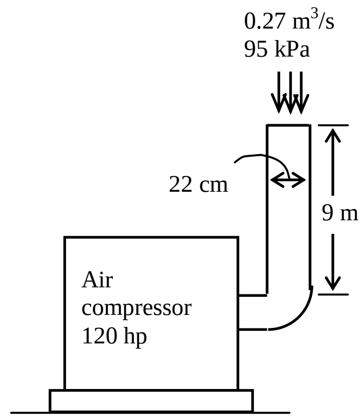
8-133

Solution A compressor takes in air at a specified rate at the outdoor conditions. The useful power used by the compressor to overcome the frictional losses in the duct is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** Air is an ideal gas. **4** The duct involves no components such as bends, valves, and connectors, and thus minor losses are negligible. **5** The flow section involves no work devices such as fans or turbines.

Properties The properties of air at 1 atm = 101.3 kPa and 15°C are ρ_0 = 1.225 kg/m^3 and μ = 1.802×10^{-5} kg/m·s. The roughness of galvanized iron surfaces is ε = 0.00015 m. The dynamic viscosity is independent of pressure, but density of an ideal gas is proportional to pressure. The density of air at 95 kPa is

$$\rho = (P/P_0)\rho_0 = (95/101.3)(1.225\ \text{kg/m}^3) = 1.149\ \text{kg/m}^3.$$



Analysis The average velocity and the Reynolds number are

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.27\ \text{m}^3/\text{s}}{\pi(0.22\ \text{m})^2/4} = 7.103\ \text{m/s}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{(1.149\ \text{kg/m}^3)(7.103\ \text{m/s})(0.22\ \text{m})}{1.802\times10^{-5}\ \text{kg/m}\cdot\text{s}} = 0.9964\times10^5$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{1.5\times10^{-4}\ \text{m}}{0.22\ \text{m}} = 6.818\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{6.818\times10^{-4}}{3.7} + \frac{2.51}{0.9964\times10^5\sqrt{f}}\right)$$

It gives f = 0.02105. Then the pressure drop in the duct and the required pumping power become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.02105\frac{9\ \text{m}}{0.22\ \text{m}}\frac{(1.149\ \text{kg/m}^3)(7.103\ \text{m/s})^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{Pa}}{1\ \text{N/m}^2}\right) = 24.96\ \text{Pa}$$

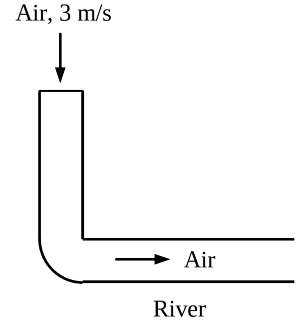
$$\dot{W}_{\text{pump},u} = \dot{V}\Delta P = (0.27\ \text{m}^3/\text{s})(24.96\ \text{Pa})\left(\frac{1\ \text{W}}{1\ \text{Pa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{6.74\ W}$$

Discussion Note hat the pressure drop in the duct and the power needed to overcome it is very small (relative to 120 hp), and can be disregarded. The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.02081, which is very close to the Colebrook value. Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be more than this due to fan inefficiency; the electrical power input will be even more due to motor inefficiency.

8-134

Solution Air enters the underwater section of a circular duct. The fan power needed to overcome the flow resistance in this section of the duct is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** Air is an ideal gas. **4** The duct involves no components such as bends, valves, and connectors. **5** The flow section involves no work devices such as fans or turbines. **6** The pressure of air is 1 atm.



Properties The properties of air at 1 atm and 15°C are ρ_0 = 1.225 kg/m^3 and μ = 1.802×10^{-5} kg/m·s. The roughness of stainless steel pipes is ε = 0.000005 m.

Analysis The volume flow rate and the Reynolds number are

$$\dot{V} = VA_c = V(\pi D^2/4) = (3\,\text{m/s})[\pi(0.20\,\text{m})^2/4] = 0.0942\,\text{m}^3/\text{s}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{(1.225\,\text{kg/m}^3)(3\,\text{m/s})(0.20\,\text{m})}{1.802\times10^{-5}\,\text{kg/m}\cdot\text{s}} = 4.079\times10^4$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{5\times10^{-6}\,\text{m}}{0.20\,\text{m}} = 2.5\times10^{-5}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{2.5\times10^{-5}}{3.7} + \frac{2.51}{4.079\times10^4\sqrt{f}}\right)$$

It gives f = 0.02195. Then the pressure drop in the duct and the required pumping power become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.02195\frac{15\,\text{m}}{0.2\,\text{m}}\frac{(1.225\,\text{kg/m}^3)(3\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{Pa}}{1\,\text{N/m}^2}\right) = 9.07\,\text{Pa}$$

$$\dot{W}_{\text{electric}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}} = \frac{(0.0942\,\text{m}^3/\text{s})(9.07\,\text{Pa})}{0.62}\left(\frac{1\,\text{W}}{1\,\text{Pa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{1.4\,W}$$

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.02175, which is sufficiently close to 0.02195. Assuming the pipe to be smooth would give 0.02187 for the friction factor, which is almost identical to the f value obtained from the Colebrook relation. Therefore, the duct can be treated as being smooth with negligible error.

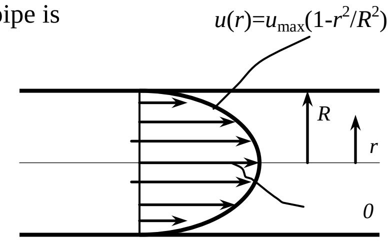
8-135

Solution The velocity profile in fully developed laminar flow in a circular pipe is given. The radius of the pipe, the average velocity, and the maximum velocity are to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is

$$u(r) = u_{\max}\left(1 - \frac{r^2}{R^2}\right)$$



The velocity profile in this case is given by $u(r) = 6(1 - 0.01r^2)$. Comparing the two relations above gives the pipe radius, the maximum velocity, and the average velocity to be

$$R^2 = \frac{1}{100} \quad \rightarrow \quad R = \mathbf{0.10\ m}$$

$$u_{\max} = \mathbf{6\ m/s} \quad \rightarrow \quad V_{avg} = \frac{u_{\max}}{2} = \frac{6\text{ m/s}}{2} = \mathbf{3\ m/s}$$

Discussion In fully developed laminar pipe flow, average velocity is exactly half of maximum (centerline) velocity.

8-136E

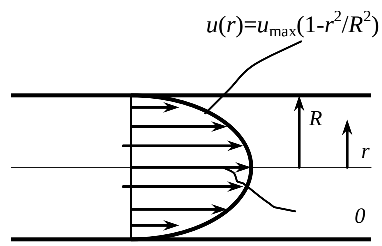
Solution The velocity profile in fully developed laminar flow in a circular pipe is given. The volume flow rate, the pressure drop, and the useful pumping power required to overcome this pressure drop are to be determined.

Assumptions **1** The flow is steady, laminar, and fully developed. **2** The pipe is horizontal.

Properties The density and dynamic viscosity of water at 40°F are ρ = 62.42 lbm/ft^3 and μ = 3.74 lbm/ft·h = 1.039×10^{-3} lbm/ft·s, respectively.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is

$$u(r) = u_{\max}\left(1 - \frac{r^2}{R^2}\right)$$



The velocity profile in this case is given by

$$u(r) = 0.8(1 - 625r^2)$$

Comparing the two relations above gives the pipe radius, the maximum velocity, and the average velocity to be

$$R^2 = \frac{1}{625} \quad \rightarrow \quad R = 0.04 \text{ ft}$$

u_{max} = 0.8 ft/s

$$V = V_{avg} = \frac{u_{\max}}{2} = \frac{0.8 \text{ ft/s}}{2} = 0.4 \text{ ft/s}$$

Then the volume flow rate and the pressure drop become

$$\dot{V} = VA_c = V(\pi R^2) = (0.4 \text{ ft/s})[\pi(0.04 \text{ ft})^2] = \mathbf{0.00201\ ft^3/s}$$

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128\mu L} \quad \rightarrow \quad 0.00201 \text{ ft}^3/\text{s} = \frac{(\Delta P)\pi(0.08 \text{ ft})^4}{128(1.039\times10^{-3} \text{ lbm/ft}\cdot\text{s})(250 \text{ ft})}\left(\frac{32.2 \text{ lbm}\cdot\text{ft/s}^2}{1 \text{ lbf}}\right)$$

It gives

$$\Delta P = 16.13 \text{ lbf/ft}^2 = \mathbf{0.112\ psi}$$

Then the useful pumping power requirement becomes

$$\dot{W}_{\text{pump,u}} = \dot{V}\Delta P = (0.00201 \text{ ft}^3/\text{s})(16.13 \text{ lbf/ft}^2)\left(\frac{1 \text{ W}}{0.737 \text{ lbf}\cdot\text{ft/s}}\right) = \mathbf{0.0440\ W}$$

Checking The flow was assumed to be laminar. To verify this assumption, we determine the Reynolds number:

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(62.42 \text{ lbm/ft}^3)(0.4 \text{ ft/s})(0.08 \text{ ft})}{1.039\times10^{-3} \text{ lbm/ft}\cdot\text{s}} = 1922$$

which is less than 2300. Therefore, the flow is laminar.

Discussion Note that the pressure drop across the water pipe and the required power input to maintain flow is negligible. This is due to the very low flow velocity. Such water flows are the exception in practice rather than the rule.

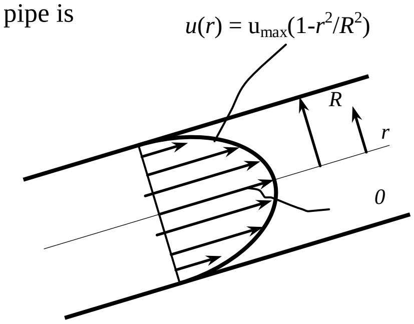
8-137E

Solution The velocity profile in fully developed laminar flow in a circular pipe is given. The volume flow rate, the pressure drop, and the useful pumping power required to overcome this pressure drop are to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Properties The density and dynamic viscosity of water at 40°F are ρ = 62.42 lbm/ft^3 and μ = 3.74 lbm/ft·h = 1.039×10^{-3} lbm/ft·s, respectively.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is



$$u(r) = u_{\max}\left(1 - \frac{r^2}{R^2}\right)$$

The velocity profile in this case is given by

$$u(r) = 0.8(1 - 625r^2)$$

Comparing the two relations above gives the pipe radius, the maximum velocity, the average velocity, and the volume flow rate to be

$$R^2 = \frac{1}{625} \quad \rightarrow \quad R = 0.04\text{ ft}$$

$u_{\max}$ = 0.8 ft/s

$$V = V_{avg} = \frac{u_{\max}}{2} = \frac{0.8\text{ ft/s}}{2} = 0.4\text{ ft/s}$$

$$\dot{V} = VA_c = V(\pi R^2) = (0.4\text{ ft/s})[\pi(0.04\text{ ft})^2] = \mathbf{0.00201\ ft^3/s}$$

For uphill flow with an inclination of 12°, we have θ = +12°, and

$$\rho gL\sin\theta = (62.42\text{ lbm/ft}^3)(32.2\text{ ft/s}^2)(250\text{ ft})\sin 12°\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 3244\text{ lbf/ft}^2$$

$$\dot{V}_{\text{uphill}} = \frac{(\Delta P - \rho gL\sin\theta)\pi D^4}{128\mu L} \quad \rightarrow \quad 0.00201\text{ ft}^3\text{/s} = \frac{(\Delta P - 3244)\pi(0.08\text{ ft})^4}{128(1.039\times10^{-3}\text{ lbm/ft}\cdot\text{s})(250\text{ ft})}\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)$$

It gives

$$\Delta P = 3260\text{ lbf/ft}^2 = 22.64\text{ psi}$$

Then the useful pumping power requirement becomes

$$\dot{W}_{\text{pump,u}} = \dot{V}\Delta P = (0.00201\text{ ft}^3\text{/s})(3260\text{ lbf/ft}^2)\left(\frac{1\text{ W}}{0.737\text{ lbf}\cdot\text{ft/s}}\right) = \mathbf{8.89\ W}$$

Checking The flow was assumed to be laminar. To verify this assumption, we determine the Reynolds number:

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(62.42\text{ lbm/ft}^3)(0.4\text{ ft/s})(0.08\text{ ft})}{1.039\times10^{-3}\text{ lbm/ft}\cdot\text{s}} = 1922$$

which is less than 2300. Therefore, the flow is laminar.

Discussion Note that the pressure drop across the water pipe and the required power input to maintain flow is negligible. This is due to the very low flow velocity. Such water flows are the exception in practice rather than the rule.

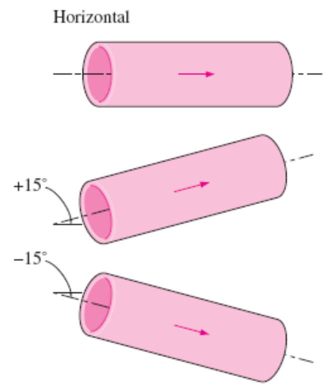
8-138

Solution The pressure readings at the inlet and the outlet of a pipe are given. The flow rates are to be determined for three different orientations, and the flow is to be shown to be laminar.

Assumptions **1** The flow is steady, laminar, and fully developed. **2** There are no pumps or turbines in the flow section. **3** There are no valves, elbows, or other devices that may cause local losses.

Properties The density and dynamic viscosity of oil at 20°C are $\rho = 888$ kg/m^3 and $\mu = 0.800$ kg/m·s, respectively.

Analysis The pressure drop across the pipe and the pipe cross-sectional area are



$$\Delta P = P_1 - P_2 = 745 - 97 = 648 \text{ kPa}$$

$$A_c = \pi D^2/4 = \pi(0.05 \text{ m})^2/4 = 0.001963 \text{ m}^2$$

The flow rate for all three cases can be determined from

$$\dot{V} = \frac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128\mu L}$$

where θ is the angle the pipe makes with the horizontal. For the horizontal case, $\theta = 0$ and thus $\sin\theta = 0$. Therefore,

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128\mu L} = \frac{(648 \text{ kPa})\pi(0.05 \text{ m})^4}{128(0.8 \text{ kg/m}\cdot\text{s})(40 \text{ m})}\left(\frac{1000 \text{ N/m}^2}{1 \text{ kPa}}\right)\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ N}}\right) = \mathbf{0.00311 \ m^3/s}$$

(*b*) For uphill flow with an inclination of 15°, we have θ = +15°, and

$$\dot{V}_{\text{uphill}} = \frac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128\mu L}$$

$$= \frac{[(648{,}000 \text{ Pa} - (888 \text{ kg/m}^3)(9.8 \text{ m/s}^2)(40 \text{ m})\sin 15°]\pi(0.05 \text{ m})^4}{128(0.8 \text{ kg/m}\cdot\text{s})(40 \text{ m})}\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ Pa}\cdot\text{m}^2}\right)$$

$$= \mathbf{0.00267 \ m^3/s}$$

(*c*) For downhill flow with an inclination of 15°, we have $\theta = -15°$, and

$$\dot{V}_{\text{downhill}} = \frac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128\mu L}$$

$$= \frac{[(648{,}000 \text{ Pa} - (888 \text{ kg/m}^3)(9.8 \text{ m/s}^2)(40 \text{ m})\sin(-15°)]\pi(0.05 \text{ m})^4}{128(0.8 \text{ kg/m}\cdot\text{s})(40 \text{ m})}\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ Pa}\cdot\text{m}^2}\right)$$

$$= \mathbf{0.00354 \ m^3/s}$$

The flow rate is the highest for downhill flow case, as expected. The average fluid velocity and the Reynolds number in this case are

$$V = \frac{\dot{V}}{A_c} = \frac{0.00354 \text{ m}^3/\text{s}}{0.001963 \text{ m}^2} = 1.80 \text{ m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(888 \text{ kg/m}^3)(1.80 \text{ m/s})(0.05 \text{ m})}{0.8 \text{ kg/m}\cdot\text{s}} = 100$$

which is less than 2300. Therefore, the flow is **laminar** for all three cases, and the analysis above is valid.

Discussion Note that the flow is driven by the combined effect of pumping power and gravity. As expected, gravity opposes uphill flow, enhances downhill flow, and has no effect on horizontal flow. Downhill flow can occur even in the absence of a pressure difference applied by a pump. For the case of $P_1 = P_2$ (i.e., no applied pressure difference), the pressure throughout the entire pipe would remain constant, and the fluid would flow through the pipe under the influence of gravity at a rate that depends on the angle of inclination, reaching its maximum value when the pipe is vertical. When solving pipe flow problems, it is always a good idea to calculate the Reynolds number to verify the flow regime – laminar or turbulent.

8-139

Solution Water is discharged from a water reservoir through a circular pipe of diameter D at the side wall at a vertical distance H from the free surface with a reentrant section. A relation for the "equivalent diameter" of the reentrant pipe for use in relations for frictionless flow through a hole is to be obtained.

Assumptions **1** The flow is steady and incompressible. **2** The reservoir is open to the atmosphere so that the pressure is atmospheric pressure at the free surface. **3** The water level in the reservoir remains constant. **4** The pipe is horizontal. **5** The entrance effects are negligible, and thus the flow is fully developed and the friction factor f is constant. **6** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The loss coefficient is $K_L = 0.8$ for the reentrant section, and $K_L = 0$ for the "frictionless" flow.

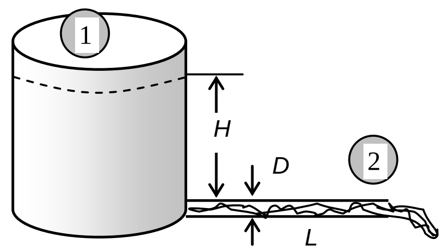
Analysis We take point 1 at the free surface of the reservoir and point 2 at the exit of the pipe, which is also taken as the reference level ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface of the reservoir is zero ($V_1 = 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad H=\alpha_2\frac{V_2^2}{2g}+h_L$$

where $\alpha_2 = 1$ and

$$h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left(f\frac{L}{D}+K_L\right)\frac{V_2^2}{2g}$$

since the diameter of the pipe is constant. Substituting and solving for V_2 gives



$$H=\frac{V_2^2}{2g}+\left(f\frac{L}{D}+K_L\right)\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{\frac{2gH}{1+fL/D+K_L}}$$

Then the volume flow rate becomes

$$\dot{V}=A_cV_2=\frac{\pi D^2}{4}\sqrt{\frac{2gH}{1+fL/D+K_L}} \qquad (1)$$

Note that in the special case of $K_L = 0$ and $f = 0$ (frictionless flow), the velocity relation reduces to the Toricelli equation, $V_{2,\text{frictionless}}=\sqrt{2gz_1}$. The flow rate in this case through a hole of D_e (equivalent diameter) is

$$\dot{V}=A_{c,\text{equiv}}V_{2,\text{frictionless}}=\frac{\pi D_{\text{equiv}}^2}{4}\sqrt{2gH} \qquad (2)$$

Setting Eqs. (1) and (2) equal to each other gives the desired relation for the equivalent diameter,

$$\frac{\pi D_{\text{equiv}}^2}{4}\sqrt{2gH}=\frac{\pi D^2}{4}\sqrt{\frac{2gH}{1+fL/D+K_L}}$$

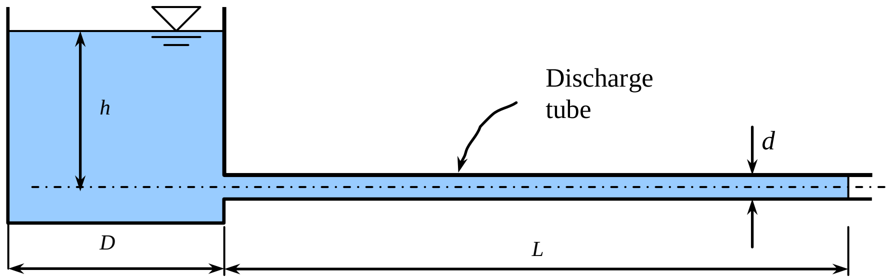
which gives

$$D_{\text{equiv}}=\frac{D}{(1+fL/D+K_L)^{1/4}}=\frac{D}{(1+0.018\times 10/0.04+0.8)^{1/4}}=\mathbf{0.63D=0.025\ m}$$

Discussion Note that the effect of frictional losses of a pipe with a reentrant section is to reduce the diameter by about 40% in this case. Also, noting that the flow rate is proportional to the square of the diameter, we have $\dot{V}\propto D_{\text{equiv}}^2=(0.63D)^2=0.40D^2$. Therefore, the flow rate through a sharp-edged entrance is about two-thirds less compared to the frictionless flow case.

8-140

Solution A highly viscous liquid discharges from a large container through a small diameter tube in laminar flow. A relation is to be obtained for the variation of fluid depth in the tank with time.



Assumptions **1** The fluid is incompressible. **2** The discharge tube is horizontal, and the flow is laminar. **3** Entrance effects and the velocity heads are negligible.

Analysis We take point 1 at the free surface of water in the tank, and point 2 at the exit of the pipe. We take the centerline of the pipe as the reference level ($z_1 = h$ and $z_2 = 0$). Noting that the fluid at both points 1 and 2 are open to the atmosphere (and thus $P_1 = P_2 = P_{\text{atm}}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$) and the velocity heads are negligible, the energy equation for a control volume between these two points gives

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_L \quad \rightarrow \quad \frac{P_{atm}}{\rho g} + h = \frac{P_{atm}}{\rho g} + h_L \quad \rightarrow \quad h_L = h \qquad (1)$$

where h is the liquid height in the tank at any time t. The total head loss through the pipe consists of major losses in the pipe since the minor losses are negligible. Also, the entrance effects are negligible and thus the friction factor for the entire tube is constant at the fully developed value. Noting that $f = \text{Re}/64$ for fully developed laminar flow in a circular pipe of diameter d, the head loss can be expressed as

$$h_L = f \frac{L}{d} \frac{V^2}{2g} = \frac{64}{\text{Re}} \frac{L}{d} \frac{V^2}{2g} = \frac{64}{Vd/\nu} \frac{L}{d} \frac{V^2}{2g} = \frac{64\nu L}{d^2} \frac{V}{2g} \qquad (2)$$

The average velocity can be expressed in terms of the flow rate as $V = \dfrac{\dot{V}}{A_c} = \dfrac{\dot{V}}{\pi d^2/4}$. Substituting into (2),

$$h_L = \frac{64\nu L}{d^2} \frac{1}{2g} \left(\frac{\dot{V}}{\pi d^2/4} \right) = \frac{64\nu L}{d^2} \frac{4\dot{V}}{2g\pi d^2} = \frac{128\nu L \dot{V}}{g\pi d^4} \qquad (3)$$

Combining Eqs. (1) and (3): $$h = \frac{128\nu L \dot{V}}{g\pi d^4} \qquad (4)$$

Noting that the liquid height h in the tank decreases during flow, the flow rate can also be expressed in terms of the rate of change of liquid height in the tank as

$$\dot{V} = -A_{\text{tank}} \frac{dh}{dt} = -\frac{\pi D^2}{4} \frac{dh}{dt} \qquad (5)$$

Substituting Eq. (5) into (4): $h = -\dfrac{128\nu L}{g\pi d^4} \dfrac{\pi D^2}{4} \dfrac{dh}{dt} = -\dfrac{32\nu L D^2}{g d^4} \dfrac{dh}{dt}$ (6)

To separate variables, it can be rearranged as $dt = -\dfrac{32\nu L D^2}{g d^4} \dfrac{dh}{h}$

Integrating from $t = 0$ (at which $h = H$) to $t = t$ (at which $h = h$) gives

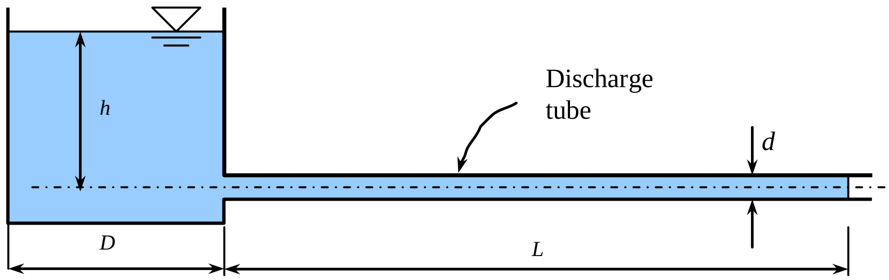
$$\boxed{t = \frac{32\nu L D^2}{g d^4} \ln(H/h)}$$

which is the desired relation for the variation of fluid depth h in the tank with time t.

Discussion If the entrance effects and the outlet kinetic energy were included in the analysis, the time would be slower.

8-141

Solution Using the setup described in the previous problem, the viscosity of an oil is to be determined for a given set of data.



Assumptions **1** The oil is incompressible. **2** The discharge tube is horizontal, and the flow is laminar. **3** Entrance effects and the inlet and the exit velocity heads are negligible.

Analysis The variation of fluid depth h in the tank with time t was determined in the previous problem to be

$$t = \frac{32\nu LD^2}{gd^4}\ln(H/h)$$

Solving for ν and substituting the given values, the kinematic viscosity of the oil is determined to be

$$\nu = \frac{gd^4}{32LD^2\ln(H/h)}t = \frac{(9.81\,\text{m/s}^2)(0.006\,\text{m})^4}{32(0.65\,\text{m})(0.63\,\text{m})^2\ln(0.4/0.34)}(1400\,\text{s}) = \mathbf{1.33\times10^{-5}\ m^2/s}$$

Discussion Note that the entrance effects are not considered, and the velocity heads are disregarded. Also, the value of the viscosity strongly depends on temperature, and thus the oil temperature should be maintained constant during the test.

8-142

Solution A water pipe has an abrupt expansion from diameter D_1 to D_2. It is to be shown that the loss coefficient is $K_L = (1 - D_1^2 / D_2^2)^2$, and K_L and P_2 are to be calculated.

Assumptions **1** The flow is steady and incompressible. **2** The pressure is uniform at the cross-section where expansion occurs, and is equal to the upstream pressure P_1. **3** The flow section is horizontal (or the elevation difference across the expansion section is negligible). **4** The flow is turbulent, and the effects of kinetic energy and momentum-flux correction factors are negligible, $\beta \approx 1$ and $\alpha \approx 1$.

Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis We designate the cross-section where expansion occurs by x. We choose cross-section 1 in the smaller diameter pipe shortly before x, and section 2 in the larger diameter pipe shortly after x. We take the region occupied by the fluid between cross-sections 1 and 2 as the control volume, with an inlet at 1 and exit at 2. The velocity, pressure, and cross-sectional area are V_1, P_1, and A_1 at cross-section 1, and V_2, P_2, and A_2 at cross-section 2. We assume the pressure along the cross-section x to be P_1 so that $P_x = P_1$. Then the continuity, momentum, and energy equations applied to the control volume become

(1) *Continuity*: $\dot m_1 = \dot m_2 \;\rightarrow\; \rho V_1 A_1 = \rho V_2 A_2 \;\rightarrow\; V_2 = \dfrac{A_1}{A_2} V_1$ (1)

(2) *Momentum*: $\sum \vec F = \sum_{\text{out}} \beta \dot m V - \sum_{\text{in}} \beta \dot m V \;\rightarrow\; P_1 A_1 + P_1 (A_x - A_1) - P_2 A_2 = \dot m (V_2 - V_1)$

But $P_1 A_1 + P_1 (A_x - A_1) = P_1 A_x = P_1 A_2$

$$\dot m (V_2 - V_1) = \rho A_2 V_2 (V_2 - V_1) = \rho A_2 \frac{A_1}{A_2} V_1 \left(\frac{A_1}{A_2} V_1 - V_1 \right) = \rho A_2 \frac{A_1}{A_2} \left(\frac{A_1}{A_2} - 1 \right) V_1^2$$

Therefore, $P_1 A_2 - P_2 A_2 = \rho A_2 \dfrac{A_1}{A_2} \left(\dfrac{A_1}{A_2} - 1 \right) V_1^2 \;\rightarrow\; \dfrac{P_1 - P_2}{\rho} = \dfrac{A_1}{A_2} \left(\dfrac{A_1}{A_2} - 1 \right) V_1^2$ (2)

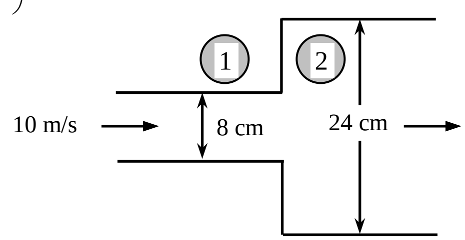
(3) *Energy*: $\dfrac{P_1}{\rho g} + \alpha_1 \dfrac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \dfrac{P_2}{\rho g} + \alpha_2 \dfrac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \;\rightarrow\; h_L = \dfrac{P_1 - P_2}{\rho g} + \dfrac{V_1^2 - V_2^2}{2g}$ (3)

Substituting Eqs. (1) and (2) and $h_L = K_L \dfrac{V_1^2}{2g}$ into Eq. (3) gives

$$K_L \frac{V_1^2}{2g} = \frac{A_1}{A_2} \left(\frac{A_1}{A_2} - 1 \right) \frac{V_1^2}{g} + \frac{V_1^2 - (A_1^2 / A_2^2) V_1^2}{2g} \;\rightarrow\; K_L = \frac{2A_1}{A_2} \left(\frac{A_1}{A_2} - 1 \right) + \left(1 - \frac{A_1^2}{A_2^2} \right)$$

Simplifying and substituting $A = \pi D^2/4$ gives the desired relation and its value,

$$K_L = \left(1 - \frac{A_1}{A_1} \right)^2 = \left(1 - \frac{\pi D_1^2 / 4}{\pi D_2^2 / 4} \right)^2 = \left(1 - \frac{D_1^2}{D_2^2} \right)^2 = \left(1 - \frac{(0.08 \text{ m})^2}{(0.24 \text{ m})^2} \right)^2 = 0.7901$$



Also, $h_L = K_L \dfrac{V_1^2}{2g} = (0.7901) \dfrac{(10 \text{ m/s})^2}{2(9.81 \text{ m/s}^2)} = 4.027 \text{ m}$

$$V_2 = \frac{A_1}{A_2} V_1 = \frac{D_1^2}{D_2^2} V_1 = \frac{(0.08 \text{ m})^2}{(0.24 \text{ m})^2} (10 \text{ m/s}) = 1.1111 \text{ m/s}$$

Solving for P_2 from Eq. (3) and substituting,

$$P_2 = P_1 + \rho \{ (V_1^2 - V_2^2)/2 - g h_L \}$$

$$= (135 \text{ kPa}) + (1000 \text{ kg/m}^3) \left\{ \frac{(10 \text{ m/s})^2 - (1.1111 \text{ m/s})^2}{2} - (9.81 \text{ m/s}^2)(4.027 \text{ m}) \right\} \left(\frac{1 \text{ kPa} \cdot \text{m}^2}{1000 \text{ kg} \cdot \text{m/s}^2} \right)$$

$$= 144.9 \text{ kPa} \cong \mathbf{145 \text{ kPa}}$$

Note that the pressure increases by 25 kPa after the expansion due to the conversion of dynamic pressure to static pressure when the velocity is decreased. Also, $K_L \cong 1$ (actually, $K_L = \alpha$) when $D_2 >> D_1$ (discharging into a reservoir).

Discussion At a discharge into a large reservoir, *all* the kinetic energy is wasted as heat.

8-143

Solution The piping system of a geothermal district heating system is being designed. The pipe diameter that will optimize the initial system cost and the energy cost is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses, the only significant energy loss arises from pipe friction. **4** The piping system is horizontal. **5** The properties of geothermal water are the same as fresh water. **6** The friction factor is constant at the given value. **7** The interest rate, the inflation rate, and the salvage value of the system are all zero. **8** The flow rate through the system remains constant.

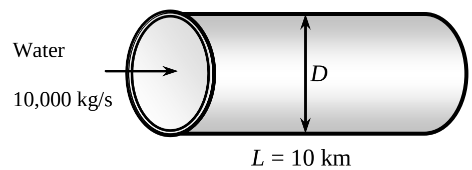
Properties We take the density of water to be $\rho = 1000$ kg/m^3. The friction factor is given to be $f = 0.015$.

Analysis The system operates in a loop, and thus we can take any point in the system as points 1 and 2 (the same point), and thus $z_1 = z_2$, $V_1 = V_2$, and $P_1 = P_2$. Then the energy equation for this piping system simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_1\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=h_L$$

That is, the pumping power is to be used to overcome the head losses due to friction in flow. When the minor losses are disregarded, the head loss for fully developed flow in a pipe of length L and diameter D can be expressed as

$$\Delta P = f\frac{L}{D}\frac{\rho V^2}{2}$$



The flow rate of geothermal water is

$$\dot{V}=\frac{\dot{m}}{\rho}=\frac{10{,}000\text{ kg/s}}{1000\text{ kg/m}^3}=10\text{ m}^3/\text{s}$$

To expose the dependence of pressure drop on diameter, we express it in terms of the flow rate as

$$\Delta P = f\frac{L}{D}\frac{\rho V^2}{2}=f\frac{L}{D}\frac{\rho}{2}\left(\frac{\dot{V}}{\pi D^2/4}\right)^2=f\frac{16L}{D}\frac{\rho\dot{V}^2}{2\pi^2D^4}=f\frac{8L}{D^5}\frac{\rho\dot{V}^2}{\pi^2}$$

Then the required pumping power can be expressed as

$$\dot{W}_{\text{pump}}=\frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}}=\frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}}=\frac{\dot{V}}{\eta_{\text{pump-motor}}}f\frac{8L}{D^5}\frac{\rho\dot{V}^2}{\pi^2}=f\frac{8L}{D^5}\frac{\rho\dot{V}^3}{\eta_{\text{pump-motor}}\pi^2}$$

Note that the pumping power requirement is proportional to f and L, consistent with our intuitive expectation. Perhaps not so obvious is that power is proportional to the *cube* of flow rate. The fact that the *power is inversely proportional to pipe diameter D to the fifth power* averages that a slight increase in pipe diameter will manifest as a tremendous reduction in power dissipation due to friction in a long pipeline. Substituting the given values and expressing the diameter D in meters,

$$\dot{W}_{\text{pump}}=(0.015)\frac{8(10{,}000\text{ m})}{D^5\text{ m}^5}\frac{(1000\text{ kg/m}^3)(10\text{ m}^3/\text{s})^3}{\pi^2(0.80)}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right)=\frac{1.52\times10^5}{D^5}\text{ kW}$$

The number of hours in one year are 24×365 = 8760 h. Then the total amount of electric power used and its cost per year are

$$E_{\text{pump}}=\dot{W}_{\text{pump}}\Delta t=\frac{1.52\times10^5}{D^5}(8760\text{ h})=\frac{1.332\times10^9}{D^5}\text{ kWh/yr}$$

$$\text{Energy cost}=E_{\text{pump}}\times\text{Unit cost}=\left(\frac{1.332\times10^9}{D^5}\text{ kWh/y}\right)(\$0.06/\text{kWh})=\frac{7.99\times10^7}{D^5}\text{ \$/yr}$$

The installation cost of the system with a 30-year lifetime is given to be Cost = $\$10^6 D^2$ where D is in meters. The annual cost of the system is then 1/30$^{\text{th}}$ of it, which is

$$\text{System cost} = \frac{\text{Total cost}}{\text{Life time}} = \frac{\$10^6 D^2}{30 \text{ yr}} = \$3.33\times10^4 D^2 \text{ (per year)}$$

Then the total annual cost of the system (installation + operating) becomes

$$\text{Total cost = Energy cost + System cost} = \frac{7.99\times10^7}{D^5} + 3.33\times10^4 D^2 \quad \$/\text{yr}$$

The optimum pipe diameter is the value that minimizes this total, and it is determined by taking the derivative of the total cost with respect to *D* and setting it equal to zero,

$$\frac{\partial(\text{Total cost})}{\partial D} = -5\times\frac{7.99\times10^7}{D^6} + 2\times3.33\times10^4 D = 0$$

Simplifying gives D^7 = 5998 whose solution is

D = **3.5 m**

This is the optimum pipe diameter that minimizes the total cost of the system under stated assumptions. A larger diameter pipe will increase the system cost more than it decreases the energy cost, and a smaller diameter pipe will increase the system cost more than it decreases the energy cost.

Discussion The assumptions of zero interest and zero inflation are not realistic, and an actual economic analysis must consider these factors as they have a major effect on the pipe diameter. This is done by considering the time value of money, and expressing all the costs at the same time. Pipe purchase is a present cost, and energy expenditures are future annual costs spread out over the project lifetime. Thus, to provide consistent dollar comparisons between initial and future costs, all future energy costs must be expressed as a single present lump sum to reflect the time-value of money. Then we can compare pipe and energy costs on a consistent basis. Economists call the necessary factor the "Annuity Present Value Factor", F. If interest rate is 10% per year with n = 30 years, then F = 9.427. Thus, if power costs $1,000,000/year for the next 30 years, then the present value of those future payments is $9,427,000 (and not $30,000,000!) if money is worth 10%. Alternatively, if you must pay $1,000,000 every year for 30 years, and you can today invest $9,437,000 at 10%, then you can meet 30 years of payments at the end of each year. The energy cost in this case can be determined by dividing the energy cost expression developed above by 9.427. This will result in a pipe diameter of *D* = 2.5 m. In an actual design, we also need to calculate the average flow velocity and the pressure head to make sure that they are reasonable. For a pipe diameter of 2.5 m, for example, the average flow velocity is 1.47 m/s and the pump pressure head is 5.6 m.

8-144

Solution Water is drained from a large reservoir through two pipes connected in series at a specified rate using a pump. The required pumping head and the minimum pumping power are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The pipes are horizontal. **3** The entrance effects are negligible, and thus the flow is fully developed. **4** The flow is turbulent so that the tabulated value of the loss coefficients can be used. **5** The pipes involve no components such as bends, valves, and other connectors that cause additional minor losses. **6** The reservoir is open to the atmosphere so that the pressure is atmospheric pressure at the free surface. **7** The water level in the reservoir remains constant. **8** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 15°C are ρ = 999.1 kg/m^3 and μ = 1.138×10^{-3} kg/m·s. The loss coefficient is K_L = 0.5 for a sharp-edged entrance. The roughness of cast iron pipes is ε = 0.00026 m.

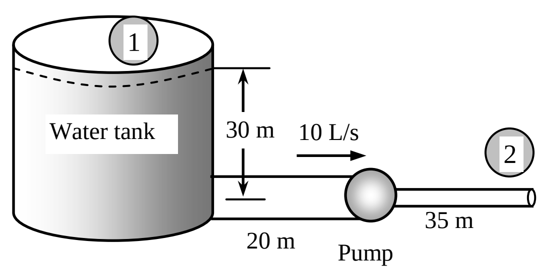
Analysis We take point 1 at the free surface of the tank, and point 2 and the reference level at the centerline of the pipe (z_2 = 0). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{\text{atm}}$) and that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1+h_{\text{pump,u}}=\alpha_2\frac{V_2^2}{2g}+h_L$$

where α_2 = 1 and

$$h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\sum\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g}$$

and the summation is over two pipes. Noting that the two pipes are connected in series and thus the flow rate through each of them is the same, the head loss for each pipe is determined as follows (we designate the first pipe by 1 and the second one by 2):



Pipe 1: $V_1=\dfrac{\dot V}{A_{c1}}=\dfrac{\dot V}{\pi D_1^2/4}=\dfrac{0.018\,\text{m}^3/\text{s}}{\pi(0.06\,\text{m})^2/4}=6.366\ \text{m/s}$

$$\text{Re}_1=\frac{\rho V_1 D_1}{\mu}=\frac{(999.1\,\text{kg/m}^3)(6.366\,\text{m/s})(0.06\,\text{m})}{1.138\times10^{-3}\ \text{kg/m}\cdot\text{s}}=335{,}300$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D_1=\frac{0.00026\,\text{m}}{0.06\,\text{m}}=0.00433$$

The friction factor corresponding to this relative roughness and the Reynolds number is, from the Colebrook equation,

$$\frac{1}{\sqrt{f_1}}=-2.0\log\left(\frac{\varepsilon/D_1}{3.7}+\frac{2.51}{\text{Re}_1\sqrt{f_1}}\right)\ \rightarrow\ \frac{1}{\sqrt{f_1}}=-2.0\log\left(\frac{0.00433}{3.7}+\frac{2.51}{335{,}300\sqrt{f_1}}\right)$$

It gives f_1 = 0.02941. The only minor loss is the entrance loss, which is K_L = 0.5. Then the total head loss of the first pipe becomes

$$h_{L1}=\left(f_1\frac{L_1}{D_1}+\sum K_L\right)\frac{V_1^2}{2g}=\left((0.02941)\frac{20\,\text{m}}{0.06\,\text{m}}+0.5\right)\frac{(6.366\,\text{m/s})^2}{2(9.81\,\text{m/s}^2)}=21.3\,\text{m}$$

Pipe 2: $V_2=\dfrac{\dot V}{A_{c2}}=\dfrac{\dot V}{\pi D_2^2/4}=\dfrac{0.018\,\text{m}^3/\text{s}}{\pi(0.04\,\text{m})^2/4}=14.32\ \text{m/s}$

$$\text{Re}_2=\frac{\rho V_2 D_2}{\mu}=\frac{(999.1\,\text{kg/m}^3)(14.32\,\text{m/s})(0.04\,\text{m})}{1.138\times10^{-3}\ \text{kg/m}\cdot\text{s}}=502{,}900$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon / D_2 = \frac{0.00026\text{ m}}{0.04\text{ m}} = 0.0065$$

The friction factor corresponding to this relative roughness and the Reynolds number is, from the Colebrook equation,

$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{\varepsilon / D_2}{3.7} + \frac{2.51}{\text{Re}_2\sqrt{f_2}}\right) \rightarrow \frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{0.0065}{3.7} + \frac{2.51}{502{,}900\sqrt{f_2}}\right)$$

It gives f_2 = 0.03309. The second pipe involves no minor losses. Note that we do not consider the exit loss unless the exit velocity is dissipated within the system considered (in this case it is not). Then the head loss for the second pipe becomes

$$h_{L2} = f_2\frac{L_2}{D_2}\frac{V_2^2}{2g} = (0.03309)\frac{35\text{ m}}{0.04\text{ m}}\frac{(14.32\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 302.6\text{ m}$$

The total head loss for two pipes connected in series is the sum of the head losses of the two pipes,

$$h_L = h_{L,\text{total}} = h_{L1} + h_{L2} = 21.3 + 302.6 = 323.9\text{ m}$$

Then the pumping head and the minimum pumping power required (the pumping power in the absence of any inefficiencies of the pump and the motor, which is equivalent to the useful pumping power) become

$$h_{\text{pump,u}} = \alpha_2\frac{V_2^2}{2g} + h_L - z_1 = (1)\frac{(14.32\text{ m/s})^2}{2(9.81\text{ m/s}^2)} + 323.9 - 30 = 304.4\text{ m}$$

$$\dot{W}_{\text{pump,u}} = \dot{V}\Delta P = \rho\dot{V}gh_{\text{pump,u}}$$

$$= (999.1\text{ kg/m}^3)(0.018\text{ m}^3\text{/s})(9.81\text{ m/s}^2)(304.4\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right) = \mathbf{53.7\ kW}$$

Therefore, the pump must supply a minimum of 53.7 kW of useful mechanical energy to water.

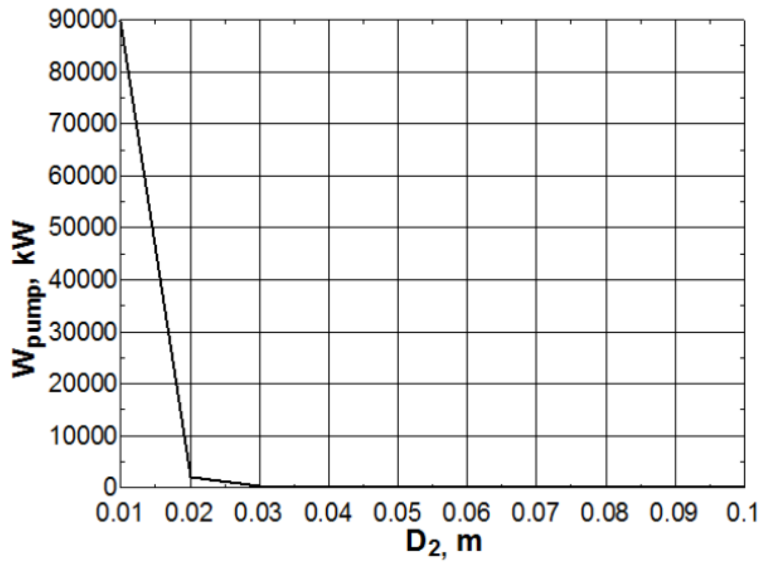
Discussion Note that the shaft power of the pump must be greater than this to account for the pump inefficiency, and the electrical power supplied must be even greater to account for the motor inefficiency.

8-145

Solution In the previous problem, the effect of second pipe diameter on required pumping head for the same flow rate is to be investigated by varying the pipe diameter from 1 cm to 10 cm in increments of 1 cm.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
rho=999.1
mu=0.001138
nu=mu/rho
Vdot=0.018 "m3/s"
g=9.81 "m/s2"
z1=30 "m"
L1=20 "m"
D1=0.06 "m"
Ac1=pi*D1^2/4
Re1=V1*D1/nu
V1=Vdot/Ac1
eps1=0.00026
rf1=eps1/D1
1/sqrt(f1)=-2*log10(rf1/3.7+2.51/(Re1*sqrt(f1)))
KL1=0.5
HL1=(f1*L1/D1+KL1)*V1^2/(2*g)
L2=35
Re2=V2*D2/nu
V2=Vdot/(pi*D2^2/4)
eps2=0.00026
rf2=eps2/D2
1/sqrt(f2)=-2*log10(rf2/3.7+2.51/(Re2*sqrt(f2)))
HL2=f2*(L2/D2)*V2^2/(2*g)
HL=HL1+HL2
hpump=V2^2/(2*g)+HL-z1
Wpump=rho*Vdot*g*hpump/1000 "kW"
```



D_2, m	W_{pump}, kW	h_{L2}, m	Re
0.01	89632.5	505391.6	2.012E+06
0.02	2174.7	12168.0	1.006E+06
0.03	250.8	1397.1	6.707E+05
0.04	53.7	302.8	5.030E+05
0.05	15.6	92.8	4.024E+05
0.06	5.1	35.4	3.353E+05
0.07	1.4	15.7	2.874E+05
0.08	-0.0	7.8	2.515E+05
0.09	-0.7	4.2	2.236E+05
0.10	-1.1	2.4	2.012E+05

Discussion Clearly, the power decreases quite rapidly with increasing diameter. This is not surprising, since the irreversible frictional head loss (major head loss) decreases significantly with increasing pipe diameter.

8-146

Solution Two pipes of identical diameter and material are connected in parallel. The length of one of the pipes is three times the length of the other. The ratio of the flow rates in the two pipes is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The flow is fully turbulent in both pipes and thus the friction factor is independent of the Reynolds number (it is the same for both pipes since they have the same material and diameter). **3** The minor losses are negligible.

Analysis When two pipes are parallel in a piping system, the head loss for each pipe must be same. When the minor losses are disregarded, the head loss for fully developed flow in a pipe of length L and diameter D can be expressed as

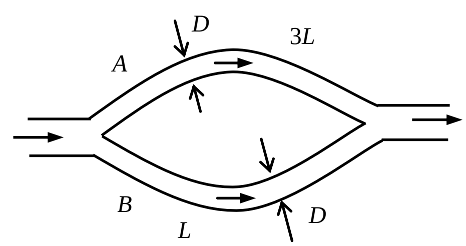
$$h_L = f\frac{L}{D}\frac{V^2}{2g} = f\frac{L}{D}\frac{1}{2g}\left(\frac{\dot{V}}{A_c}\right)^2 = f\frac{L}{D}\frac{1}{2g}\left(\frac{\dot{V}}{\pi D^2/4}\right)^2 = 8f\frac{L}{D}\frac{1}{g}\frac{\dot{V}^2}{\pi^2 D^4} = 8f\frac{L}{g\pi^2}\frac{\dot{V}^2}{D^5}$$

Solving for the flow rate gives

$$\dot{V} = \sqrt{\frac{\pi^2 h_L g D^5}{8fL}} = \frac{k}{\sqrt{L}} \qquad (k \text{ is a constant})$$

When the pipe diameter, friction factor, and the head loss is constant, which is the case here for parallel connection, the flow rate becomes inversely proportional to the square root of length L. Therefore, when the length is tripled, the flow rate will decrease by a factor of $3^{0.5} = 1.73$ since

If $$\dot{V}_A = \frac{k}{\sqrt{L_A}}$$



Then $$\dot{V}_B = \frac{k}{\sqrt{L_B}} = \frac{k}{\sqrt{3L_A}} = \frac{k}{\sqrt{3}\sqrt{L_A}} = \frac{\dot{V}_A}{\sqrt{5}} = 0.447\dot{V}_A$$

Therefore, the ratio of the flow rates in the two pipes is **0.447**.

Discussion Even though one pipe is three times as long as the other, the volume flow rate in the shorter pipe is *not* three times as much – the relationship is nonlinear.

8-147

Solution A pipeline that transports oil at a specified rate branches out into two parallel pipes made of commercial steel that reconnects downstream. The flow rates through each of the parallel pipes are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Entrance effects are negligible, and thus the flow is fully developed. **3** Minor losses are disregarded. **4** Flows through both pipes are turbulent (to be verified).

Properties The density and dynamic viscosity of oil at 40°C are ρ = 876 kg/m^3 and μ = 0.2177 kg/m·s. The roughness of commercial steel pipes is ε = 0.000045 m.

Analysis This problem cannot be solved directly since the velocities (or flow rates) in the pipes are not known. Below we will set up the equations to be solved by an equation solver. The head loss in two parallel branches must be the same, and the total flow rate must the sum of the flow rates in the parallel branches. Therefore,

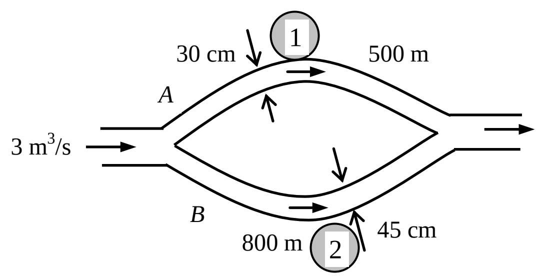
$$h_{L,1} = h_{L,2} \qquad (1)$$

$$\dot{V} = \dot{V}_1 + \dot{V}_2 \;\rightarrow\; \dot{V}_1 + \dot{V}_2 = 3 \qquad (2)$$

We designate the 30-cm diameter pipe by 1 and the 45-cm diameter pipe by 2. The average velocity, the relative roughness, the Reynolds number, friction factor, and the head loss in each pipe are expressed as

$$V_1 = \frac{\dot{V}_1}{A_{c,1}} = \frac{\dot{V}_1}{\pi D_1^2/4} \;\rightarrow\; V_1 = \frac{\dot{V}_1}{\pi(0.30\text{m})^2/4} \qquad (3)$$

$$V_2 = \frac{\dot{V}_2}{A_{c,2}} = \frac{\dot{V}_2}{\pi D_2^2/4} \;\rightarrow\; V_2 = \frac{\dot{V}_2}{\pi(0.45\text{m})^2/4} \qquad (4)$$



$$\text{rf}_1 = \frac{\varepsilon_1}{D_1} = \frac{4.5\times10^{-5}\text{ m}}{0.30\text{ m}} = 1.5\times10^{-4}$$

$$\text{rf}_2 = \frac{\varepsilon_2}{D_2} = \frac{4.5\times10^{-5}\text{ m}}{0.45\text{ m}} = 1\times10^{-4}$$

$$\text{Re}_1 = \frac{\rho V_1 D_1}{\mu} \;\rightarrow\; \text{Re}_1 = \frac{(876\text{ kg/m}^3)\mathbf{V}_1(0.30\text{ m})}{0.2177\text{ kg/m}\cdot s} \qquad (5)$$

$$\text{Re}_2 = \frac{\rho V_2 D_2}{\mu} \;\rightarrow\; \text{Re}_2 = \frac{(876\text{ kg/m}^3)\mathbf{V}_2(0.45\text{ m})}{0.2177\text{ kg/m}\cdot s} \qquad (6)$$

$$\frac{1}{\sqrt{f_1}} = -2.0\log\left(\frac{\varepsilon/D_1}{3.7} + \frac{2.51}{\text{Re}_1\sqrt{f_1}}\right) \;\rightarrow\; \frac{1}{\sqrt{f_1}} = -2.0\log\left(\frac{1.5\times10^{-4}}{3.7} + \frac{2.51}{\text{Re}_1\sqrt{f_1}}\right) \quad (7)$$

$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{\varepsilon/D_2}{3.7} + \frac{2.51}{\text{Re}_2\sqrt{f_2}}\right) \;\rightarrow\; \frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{1\times10^{-4}}{3.7} + \frac{2.51}{\text{Re}_2\sqrt{f_2}}\right) \quad (8)$$

$$h_{L,1} = f_1\frac{L_1}{D_1}\frac{V_1^2}{2g} \;\rightarrow\; h_{L,1} = f_1\frac{500\text{ m}}{0.30\text{ m}}\frac{V_1^2}{2(9.81\text{ m/s}^2)} \qquad (9)$$

$$h_{L,2} = f_2\frac{L_2}{D_2}\frac{V_2^2}{2g} \;\rightarrow\; h_{L,2} = f_2\frac{800\text{ m}}{0.45\text{ m}}\frac{V_2^2}{2(9.81\text{ m/s}^2)} \qquad (10)$$

This is a system of 10 equations in 10 unknowns, and solving them simultaneously by an equation solver gives

$$\dot{V}_1 = \mathbf{0.91\,m^3/s}, \quad \dot{V}_2 = \mathbf{2.09\,m^3/s},$$

V_1 = 12.9 m/s, V_2 = 13.1 m/s, $h_{L,1} = h_{L,2} = 392$ m

Re_1 = 15,540, Re_2 = 23,800, f_1 = 0.02785, f_2 = 0.02505

Note that Re > 4000 for both pipes, and thus the assumption of turbulent flow is verified.

Discussion This problem can also be solved by using an iterative approach, but it would be very time consuming. Equation solvers such as EES are invaluable for theses kinds of problems.

8-148

Solution The piping of a district heating system that transports hot water at a specified rate branches out into two parallel pipes made of commercial steel that reconnects downstream. The flow rates through each of the parallel pipes are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are given to be negligible. **4** Flows through both pipes are turbulent (to be verified).

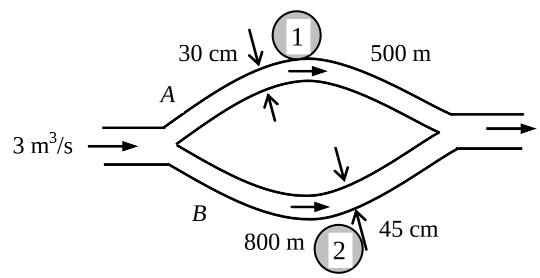
Properties The density and dynamic viscosity of water at 100°C are ρ = 957.9 kg/m^3 and μ = 0.282×10^{-3} kg/m·s. The roughness of commercial steel pipes is ε = 0.000045 m.

Analysis This problem cannot be solved directly since the velocities (or flow rates) in the pipes are not known. Below we will set up the equations to be solved by an equation solver. The head loss in two parallel branches must be the same, and the total flow rate must the sum of the flow rates in the parallel branches. Therefore,

$$h_{L,1} = h_{L,2} \qquad (1)$$

$$\dot{V} = \dot{V}_1 + \dot{V}_2 \quad \rightarrow \quad \dot{V}_1 + \dot{V}_2 = 3 \qquad (2)$$

We designate the 30-cm diameter pipe by 1 and the 45-cm diameter pipe by 2. The average velocity, the relative roughness, the Reynolds number, friction factor, and the head loss in each pipe are expressed as



$$V_1 = \frac{\dot{V}_1}{A_{c,1}} = \frac{\dot{V}_1}{\pi D_1^2/4} \quad \rightarrow \quad V_1 = \frac{\dot{V}_1}{\pi (0.30\text{m})^2/4} \qquad (3)$$

$$V_2 = \frac{\dot{V}_2}{A_{c,2}} = \frac{\dot{V}_2}{\pi D_2^2/4} \quad \rightarrow \quad V_2 = \frac{\dot{V}_2}{\pi (0.45\text{m})^2/4} \qquad (4)$$

$$\text{rf}_1 = \frac{\varepsilon_1}{D_1} = \frac{4.5\times10^{-5}\ \text{m}}{0.30\ \text{m}} = 1.5\times10^{-4}$$

$$\text{rf}_2 = \frac{\varepsilon_2}{D_2} = \frac{4.5\times10^{-5}\ \text{m}}{0.45\ \text{m}} = 1\times10^{-4}$$

$$\text{Re}_1 = \frac{\rho V_1 D_1}{\mu} \quad \rightarrow \quad \text{Re}_1 = \frac{(957.9\ \text{kg/m}^3)V_1(0.30\ \text{m})}{0.282\times10^{-3}\ \text{kg/m}\cdot s} \qquad (5)$$

$$\text{Re}_2 = \frac{\rho V_2 D_2}{\mu} \quad \rightarrow \quad \text{Re}_2 = \frac{(957.9\ \text{kg/m}^3)V_2(0.45\ \text{m})}{0.282\times10^{-3}\ \text{kg/m}\cdot s} \qquad (6)$$

$$\frac{1}{\sqrt{f_1}} = -2.0\log\left(\frac{\varepsilon/D_1}{3.7} + \frac{2.51}{\text{Re}_1\sqrt{f_1}}\right) \quad \rightarrow \quad \frac{1}{\sqrt{f_1}} = -2.0\log\left(\frac{1.5\times10^{-4}}{3.7} + \frac{2.51}{\text{Re}_1\sqrt{f_1}}\right) \qquad (7)$$

$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{\varepsilon/D_2}{3.7} + \frac{2.51}{\text{Re}_2\sqrt{f_2}}\right) \quad \rightarrow \quad \frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{1\times10^{-4}}{3.7} + \frac{2.51}{\text{Re}_2\sqrt{f_2}}\right) \qquad (8)$$

$$h_{L,1} = f_1\frac{L_1}{D_1}\frac{V_1^2}{2g} \quad \rightarrow \quad h_{L,1} = f_1\frac{500\ \text{m}}{0.30\ \text{m}}\frac{V_1^2}{2(9.81\ \text{m/s}^2)} \qquad (9)$$

$$h_{L,2} = f_2\frac{L_2}{D_2}\frac{V_2^2}{2g} \quad \rightarrow \quad h_{L,2} = f_2\frac{800\ \text{m}}{0.45\ \text{m}}\frac{V_2^2}{2(9.81\ \text{m/s}^2)} \qquad (10)$$

This is a system of 10 equations in 10 unknowns, and their simultaneous solution by an equation solver gives

$\dot{V}_1 = \mathbf{0.919\ m^3/s}$, $\dot{V}_2 = \mathbf{2.08\ m^3/s}$,

V_1 = 13.0 m/s, V_2 = 13.1 m/s, $h_{L,1} = h_{L,2} = 187\ \text{m}$

Re_1 = 1.324×10^7, Re_2 = 2.00×10^7, f_1 = 0.0131, f_2 = 0.0121

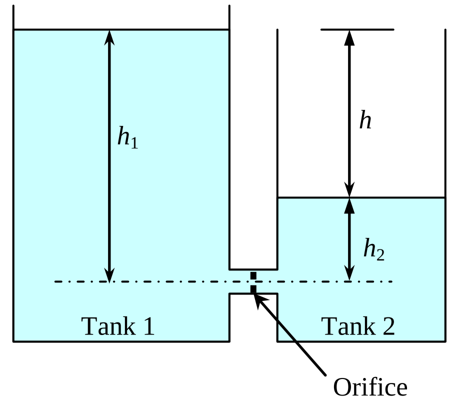
Note that Re > 4000 for both pipes, and thus the assumption of turbulent flow is verified.

Discussion This problem can also be solved by using a trial-and-error approach, but it will be very time consuming. Equation solvers such as EES are invaluable for these kinds of problems.

8-149

Solution A system that concsists of two interconnected cylindrical tanks is used to determine the discharge coefficient of a short 5-mm diameter orifice. For given initial fluid heights and discharge time, the discharge coefficient of the orifice is to be determined.

Assumptions **1** The fluid is incompressible. **2** The entire systems, including the connecting flow section, is horizontal. **3** The discharge coefficient remains constant (in reality, it may change since the flow velocity and thus the Reynolds number changes during flow). **4** Losses other than the ones associated with flow through the orifice are negligible. **5** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.



Analysis We take point 1 at the free surface of water in Tank 1, and point 0 at the exit of the orifice. We take the centerline of the orifice as the reference level ($z_1 = h_1$ and $z_0 = 0$). Noting that the fluid at point 1is open to the atmosphere (and thus $P_1 = P_{atm}$ and $P_0 = P_{atm} + \rho g h_2$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the *Bernoulli equation* between these two points gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_0}{\rho g}+\frac{V_0^2}{2g}+z_0 \;\rightarrow\; \frac{P_{atm}}{\rho g}+h_1=\frac{P_{atm}+\rho g h_2}{\rho g}+\frac{V_0^2}{2g} \;\rightarrow\; V_0=\sqrt{2g(h_1-h_2)}=\sqrt{2gh}$$

where $h = h_1 - h_2$ is the vertical distance between the water levels in the two tanks at any time t. Note that h_1, h_2, h, and V_0 are all variable (h_1 decreases while h_2 and h increase during discharge.

Noting that the fluid is a liquid (ρ = constant) and keeping the conservation of mass in mind and the definition of the discharge coefficient C_d, the flow rate through the orifice can be expressed as

$$\dot{V} = C_d V_o A_o = -A_1\frac{dh_1}{dt} = A_2\frac{dh_2}{dt} \longrightarrow dh_2 = -\frac{A_1}{A_2}dh_1$$

Also, $h = h_1 - h_2 \longrightarrow dh = dh_1 - dh_2 \longrightarrow dh_1 = dh_2 + dh$ (Note that $dh < 0$, $dh_1 < 0$, and $dh_2 > 0$)

Combining the two equations above, $\quad dh_1 = \dfrac{dh}{1 + A_1/A_2}$

Then, $\quad \dot{V} = C_d V_o A_o = -A_1\dfrac{dh_1}{dt} \longrightarrow C_d A_o\sqrt{2gh} = -A_1\dfrac{1}{1 + A_1/A_2}\dfrac{dh}{dt}$

which can be rearranged as $\quad -dt = \dfrac{A_1 A_2}{A_1 + A_2}\dfrac{1}{C_d A_o\sqrt{2g}}\dfrac{dh}{\sqrt{h}}$

Integrating $\quad -\displaystyle\int_0^t dt = \dfrac{A_1 A_2}{A_1 + A_2}\dfrac{1}{C_d A_o\sqrt{2g}}\int_{h_1}^{h}\dfrac{dh}{\sqrt{h}}$

Performing the integration $\quad t = -\dfrac{A_1 A_2}{A_1 + A_2}\dfrac{2}{C_d A_o\sqrt{2g}}\left[\sqrt{h} - \sqrt{h_1}\right]$

Solving for C_d $\quad C_d = \dfrac{2(\sqrt{h_1} - \sqrt{h})}{(A_0/A_2 + A_0/A_1)t\sqrt{2g}}$

Fluid flow stops when the liquid levels in the two tanks become equal (and thus $h = 0$). Substituign the given values, the discharge coefficient is determined to be

$$\frac{A_0}{A_2}+\frac{A_0}{A_1}=\left(\frac{D_0}{D_2}\right)^2+\left(\frac{D_0}{D_1}\right)^2=\left(\frac{0.5\text{ cm}}{30\text{ cm}}\right)^2+\left(\frac{0.5\text{ cm}}{12\text{ cm}}\right)^2=0.002014,$$

$$C_d=\frac{2\sqrt{0.5\text{ m}}}{(0.002014)(170\text{ s})\sqrt{2\times 9.81\text{m/s}^2}}=\mathbf{0.933}$$

Discussion We could add the minor losses at the pipe inlet and outlet without much extra effort.

8-150

Solution Air compressed by a large compressor is transported through a galvanized steel pipe. The pressure drop and the power wasted in the compressed air line are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** Minor losses other than those associated with elbows are disregarded. **4** There are no pumps or turbines in the compressed air line.

Properties The roughness of galvanized steel pipe is given to be ε = 0.00015 m. The dynamic viscosity of air at 60°C is $\mu = 2.008\times10^{-5}$ kg/m·s, and it is independent of pressure. The density of air listed in that table is for 1 atm. The density at 20°C, 100 kPa and 60°C, 900 kPa can be determined from the ideal gas relation to be

$$\rho_{\text{in}} = \frac{P_{\text{in}}}{RT_{\text{in}}} = \frac{100 \text{ kPa}}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg.K})(20+273\text{ K})} = 1.189 \text{ kg/m}^3$$

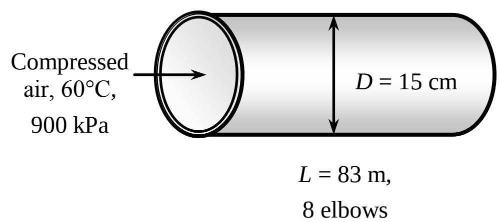
$$\rho = \rho_{\text{line}} = \frac{P_{\text{line}}}{RT_{\text{line}}} = \frac{900 \text{ kPa}}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg.K})(60+273\text{ K})} = 9.417 \text{ kg/m}^3$$

Analysis From the conservation of mass we have $\dot m_{\text{in}} = \dot m_{\text{line}}$ or $\rho_{\text{in}}\dot V = \rho_{\text{line}}\dot V_{\text{line}}$. Then the volume flow rate, the average velocity, and the Reynolds number in the compressed air line become

$$\dot V = \dot V_{\text{line}} = \frac{\rho_{\text{in}}}{\rho_{\text{line}}}\dot V_{\text{in}} = \frac{1.189 \text{ kg/m}^3}{9.417 \text{ kg/m}^3}(0.6 \text{ m}^3/\text{s}) = 0.07577 \text{ m}^3/\text{s}$$

$$V = \frac{\dot V}{A_c} = \frac{\dot V}{\pi D^2/4} = \frac{0.07577 \text{ m}^3/s}{\pi(0.15\text{ m})^2/4} = 4.288 \text{ m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(9.417 \text{ kg/m}^3)(4.288 \text{ m/s})(0.15\text{ m})}{2.008\times10^{-5} \text{ kg/m}\cdot\text{s}} = 3.016\times10^5$$



which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{0.15\text{ mm}}{150\text{ mm}} = 0.001$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt f}\right) \rightarrow \frac{1}{\sqrt f} = -2.0\log\left(\frac{0.001}{3.7} + \frac{2.51}{3.016\times10^5\sqrt f}\right)$$

It gives f = 0.0206. The sum of the loss coefficients is

$$\sum K_L = K_{L,\text{entrance}} + 11K_{L,\text{elbow}} + K_{L,\text{exit}} = 0 + 8\times0.6 + 0 = 4.8$$

Then the pressure drop, the required useful power input to overcome it, and the actual power input required become

$$\Delta P = \Delta P_L = \left(f\frac{L}{D} + \sum K_L\right)\frac{\rho V^2}{2}$$

$$= \left(0.0206\frac{83\text{ m}}{0.15\text{ m}} + 4.8\right)\frac{(9.417 \text{ kg/m}^3)(4.288 \text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = \mathbf{1.40\,kPa}$$

$$\dot W_{\text{pump,u}} = \dot V\Delta P = (0.07577 \text{ m}^3/\text{s})(1.40 \text{ kPa})\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right) = 0.106 \text{ kW}$$

$$\dot W_{\text{pump,act}} = \dot W_{\text{pump,u}}/\eta_{\text{comp}} = (0.106 \text{ kW})/0.85 = \mathbf{0.125\,kW}$$

Therefore, 0.125 kW of the 300 kW consumed by the compressor is expended in the compressed air line.

Discussion Despite the long transportation line and several turns, the power losses are very small in this case because of the large diameter of the pipe. It can be shown that the power wasted in the compressed air line would be 3.83 kW – a nearly 30-fold increase – if the pipe diameter was reduced by half.

8-151

Solution Air compressed by a large compressor is transported through a galvanized steel pipe. It is to be investigated if it is worthwhile to double the pipe diameter to reduce the pressure and power losses in the compressed air line

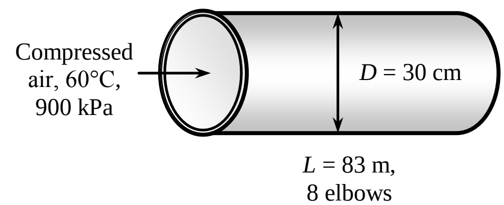
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** Minor losses other than those associated with elbows are disregarded. **4** There are no pumps or turbines in the compressed air line.

Properties The roughness of galvanized steel pipe is given to be ε = 0.00015 m. The dynamic viscosity of air at 60°C is μ = 2.008×10^{-5} kg/m·s, and it is independent of pressure. The density of air listed in that table is for 1 atm. The density at 20°C, 100 kPa and 60°C, 900 kPa can be determined from the ideal gas relation to be

$$\rho_{\text{in}} = \frac{P_{\text{in}}}{RT_{\text{in}}} = \frac{100\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg.K})(20+273\text{ K})} = 1.189\text{ kg/m}^3$$

$$\rho = \rho_{\text{line}} = \frac{P_{\text{line}}}{RT_{\text{line}}} = \frac{900\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg.K})(60+273\text{ K})} = 9.417\text{ kg/m}^3$$

Analysis From the conservation of mass we have $\dot m_{\text{in}} = \dot m_{\text{line}}$ or $\rho_{\text{in}}\dot V = \rho_{\text{line}}\dot V_{\text{line}}$. Then the volume flow rate, the average velocity, and the Reynolds number in the compressed air line become



$$\dot V = \dot V_{\text{line}} = \frac{\rho_{\text{in}}}{\rho_{\text{line}}}\dot V_{\text{in}} = \frac{1.189\text{ kg/m}^3}{9.417\text{ kg/m}^3}(0.6\text{ m}^3/\text{s}) = 0.07577\text{ m}^3/\text{s}$$

$$V = \frac{\dot V}{A_c} = \frac{\dot V}{\pi D^2/4} = \frac{0.07577\text{ m}^3/s}{\pi(0.39\text{ m})^2/4} = 1.072\text{ m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(9.417\text{ kg/m}^3)(1.072\text{ m/s})(0.30\text{ m})}{2.008\times10^{-5}\text{ kg/m}\cdot\text{s}} = 1.508\times10^5$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{0.15\text{ mm}}{300\text{ mm}} = 0.0005$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt f}\right) \rightarrow \frac{1}{\sqrt f} = -2.0\log\left(\frac{0.0005}{3.7} + \frac{2.51}{1.508\times10^5\sqrt f}\right)$$

It gives f = 0.01936. The sum of the loss coefficients is

$$\sum K_L = K_{L,\text{entrance}} + 11K_{L,\text{elbow}} + K_{L,\text{exit}} = 0 + 8\times0.6 + 0 = 4.8$$

Then the pressure drop, the required useful power input to overcome it, and the actual power input required become

$$\Delta P = \Delta P_L = \left(f\frac{L}{D} + \sum K_L\right)\frac{\rho V^2}{2}$$

$$= \left(0.01936\frac{83\text{ m}}{0.30\text{ m}} + 4.8\right)\frac{(9.417\text{ kg/m}^3)(1.072\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = 0.05495\text{ kPa}$$

$$\dot W_{\text{pump,u}} = \dot V\Delta P = (0.07577\text{ m}^3/\text{s})(0.05495\text{ kPa})\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right) = 0.004164\text{ kW}$$

$$\dot W_{\text{pump,act}} = \dot W_{\text{pump,u}}/\eta_{\text{comp}} = (0.004164\text{ kW})/0.85 = \mathbf{0.00490 kW}$$

Therefore, the wasted power goes down from 0.125 kW to 0.0049 kW – a reduction of 0.120 kW. Assuming the extreme case of compressor operating non-stop at full capacity year around and a unit cost of \$0.10/kWh for electricity, the annual power and cost savings would be

Electric power savings = (0.120 kW)(8760 h) = 1050 kWh

Cost savings = (1050 kWh)(\$0.10/kWh) = \$105

Discussion For annual cost savings of about \$100, doubling the pipe diameter would probably **NOT be worthwhile** because of the large cost increases associated with doubling the pipe diameter. This conclusion can be verified using typical pipeline construction costs.

8-152E

Solution A water fountain is to be installed at a remote location by attaching a cast iron pipe directly to a water main. For a specified flow rate, the minimum diameter of the piping system is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed and the friction factor is constant for the entire pipe. **3** The pressure at the water main remains constant. **4** There are no dynamic pressure effects at the pipe-water main connection, and the pressure at the pipe entrance is 60 psia. **5** Elevation difference between the pipe and the fountain is negligible ($z_2 = z_1$) . **6** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 70°F are ρ = 62.30 lbm/ft^3 and μ = 2.360 lbm/ft·h = 6.556×10^{-4} lbm/ft·s. The roughness of cast iron pipe is ε = 0.00085 ft. The minor loss coefficient is K_L = 0.5 for a sharp-edged entrance, K_L = 1.1 for a 90° miter bend without vanes, K_L = 0.2 for a fully open gate valve, and K_L = 5 for an angle valve.

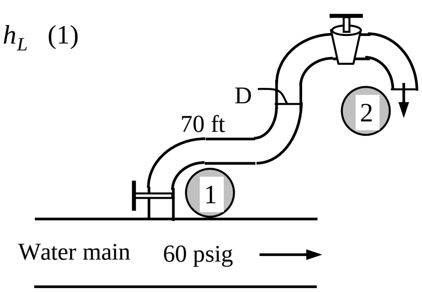
Analysis We choose point 1 in the water main near the entrance where the pressure is 60 psig and the velocity in the pipe to be low. We also take point 1 as the reference level. We take point 2 at the exit of the water fountain where the pressure is the atmospheric pressure ($P_2 = P_{atm}$) and the velocity is the discharge velocity. The energy equation for a control volume between these two points is

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad \frac{P_{1,\text{gage}}}{\rho g}=\alpha_2\frac{V_2^2}{2g}+h_L$$

where α_2 = 1 and $h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$

since the diameter of the piping system is constant. Then the energy equation becomes

$$\frac{60\text{ psi}}{(62.3\text{ lbm/ft}^3)(32.2\text{ ft/s}^2)}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)=\frac{V_2^2}{2(32.2\text{ ft/s}^2)}+h_L \quad (1)$$



The average velocity in the pipe and the Reynolds number are

$$V_2=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4} \quad\rightarrow\quad V_2=\frac{15/60\text{ gal/s}}{\pi D^2/4}\left(\frac{0.1337\text{ ft}^3}{1\text{ gal}}\right) \quad (2)$$

$$\text{Re}=\frac{\rho V_2 D}{\mu} \quad\rightarrow\quad \text{Re}=\frac{(62.3\text{ lbm/ft}^3)V_2 D}{6.556\times10^{-4}\text{ lbm/ft}\cdot\text{s}} \quad (3)$$

The friction factor can be determined from the Colebrook equation,

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D_h}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right) \quad\rightarrow\quad \frac{1}{\sqrt{f}}=-2.0\log\left(\frac{0.00085/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right) \quad (4)$$

The sum of the loss coefficients is

$$\sum K_L=K_{L,\text{entrance}}+3K_{L,\text{elbow}}+K_{L,\text{gate valve}}+K_{L,\text{angle valve}}=0.5+3\times1.1+0.2+5=9$$

Then the total head loss becomes

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g} \quad\rightarrow\quad h_L=\left(f\frac{70\text{ ft}}{D}+9\right)\frac{V_2^2}{2(32.2\text{ ft/s}^2)} \quad (5)$$

These are 5 equations in the 5 unknowns of V_2, h_L, D, Re, and f, and solving them simultaneously using an equation solver such as EES gives

V_2 = 12.1 ft/s, h_L = 136.4 ft, D = 0.0594 ft = **0.713 in**, Re = 68,080, and f = 0.04365

Therefore, **the diameter of the pipe must be at least 0.713 in**.

Discussion The pipe diameter can also be determined approximately by using the Swamee and Jain relation.

8-153E

Solution A water fountain is to be installed at a remote location by attaching a cast iron pipe directly to a water main. For a specified flow rate, the minimum diameter of the piping system is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed and the friction factor is constant for the entire pipe. **3** The pressure at the water main remains constant. **4** There are no dynamic pressure effects at the pipe-water main connection, and the pressure at the pipe entrance is 60 psia. **5** Elevation difference between the pipe and the fountain is negligible ($z_2 = z_1$) . **6** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 70°F are ρ = 62.30 lbm/ft^3 and μ = 2.360 lbm/ft·h = 6.556×10^{-4} lbm/ft·s. The plastic pipes are considered to be smooth, and thus their roughness is ε = 0. The minor loss coefficient is K_L = 0.5 for a sharp-edged entrance, K_L = 1.1 for a 90° miter bend without vanes, K_L = 0.2 for a fully open gate valve, and K_L = 5 for an angle valve.

Analysis We choose point 1 in the water main near the entrance where the pressure is 60 psig and the velocity in the pipe to be low. We also take point 1 as the reference level. We take point 2 at the exit of the water fountain where the pressure is the atmospheric pressure ($P_2 = P_{atm}$) and the velocity is the discharge velocity. The energy equation for a control volume between these two points is

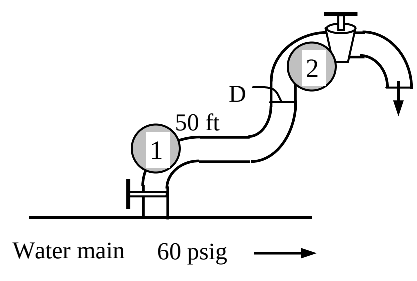
$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad \frac{P_{1,\text{gage}}}{\rho g}=\alpha_2\frac{V_2^2}{2g}+h_L$$

where α_2 = 1 and $h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$

since the diameter of the piping system is constant. Then the energy equation becomes

$$\frac{60\text{ psi}}{(62.3\text{ lbm/ft}^3)(32.2\text{ ft/s}^2)}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)=\frac{V_2^2}{2(32.2\text{ ft/s}^2)}+h_L \quad (1)$$

The average velocity in the pipe and the Reynolds number are



$$V_2=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4} \quad\rightarrow\quad V_2=\frac{20/60\text{ gal/s}}{\pi D^2/4}\left(\frac{0.1337\text{ ft}^3}{1\text{ gal}}\right) \quad (2)$$

$$\text{Re}=\frac{\rho V_2 D}{\mu} \quad\rightarrow\quad \text{Re}=\frac{(62.3\text{ lbm/ft}^3)V_2 D}{6.556\times10^{-4}\text{ lbm/ft}\cdot\text{s}} \quad (3)$$

The friction factor can be determined from the Colebrook equation,

$$\frac{1}{\sqrt f}=-2.0\log\left(\frac{\varepsilon/D_h}{3.7}+\frac{2.51}{\text{Re}\sqrt f}\right) \quad\rightarrow\quad \frac{1}{\sqrt f}=-2.0\log\left(0+\frac{2.51}{\text{Re}\sqrt f}\right) \quad (4)$$

The sum of the loss coefficients is

$$\sum K_L=K_{L,\text{entrance}}+3K_{L,\text{elbow}}+K_{L,\text{gate valve}}+K_{L,\text{angle valve}}=0.5+3\times1.1+0.2+5=9$$

Then the total head loss becomes

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g} \quad\rightarrow\quad h_L=\left(f\frac{50\text{ ft}}{D}+9\right)\frac{V_2^2}{2(32.2\text{ ft/s}^2)} \quad (5)$$

These are 5 equations in the 5 unknowns of V_2, h_L, D, Re, and f, and solving them simultaneously using an equation solver such as EES gives

V_2 = 18.4 ft/s, h_L = 133.4 ft, D = 0.05549 ft = **0.67 in**, Re = 97,170, and f = 0.0181

Therefore, the diameter of the pipe must be at least 0.67 in.

Discussion The pipe diameter can also be determined approximately by using the Swamee and Jain relation. It would give D = 0.62 in, which is within 7% of the result obtained above.

8-154

Solution In a hydroelectric power plant, the flow rate of water, the available elevation head, and the combined turbine-generator efficiency are given. The electric power output of the plant is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed and the friction factor is constant for the entire pipe. **3** The minor losses are given to be negligible. **4** The water level in the reservoir remains constant.

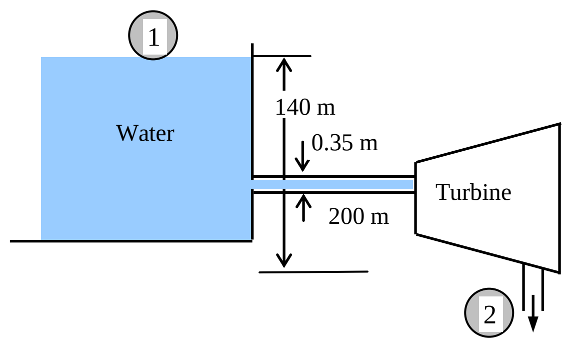
Properties The density and dynamic viscosity of water at 20°C are ρ = 998 kg/m^3 and μ = 1.002×10^{-3} kg/m⋅s. The roughness of cast iron pipes is ε = 0.00026 m.

Analysis We take point 1 at the free surface of the reservoir, and point 2 and the reference level at the free surface of the water leaving the turbine site (z_2 = 0). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocities at both points are very low ($V_1 \cong V_2 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{turbine,e}}=z_1-h_L$$

The average velocity, Reynolds number, friction factor, and head loss in the pipe are

$$V=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4}=\frac{0.6\text{ m}^3/\text{s}}{\pi(0.35\text{ m})^2/4}=6.236\text{ m/s}$$



$$\text{Re}=\frac{\rho VD_h}{\mu}=\frac{(998\text{ kg/m}^3)(6.236\text{ m/s})(0.35\text{ m})}{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}}=2.174\times10^6$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D_h=\frac{0.00026\text{ m}}{0.35\text{ m}}=7.43\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D_h}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\rightarrow\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{7.43\times10^{-4}}{3.7}+\frac{2.51}{2.174\times10^6\sqrt{f}}\right)$$

It gives f = 0.01847. When the minor losses are negligible, the head loss in the pipe and the available turbine head are determined to be

$$h_L=f\frac{L}{D}\frac{V^2}{2g}=0.01847\frac{200\text{ m}}{0.35\text{ m}}\frac{(6.236\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=20.92\text{ m}$$

$$h_{\text{turbine,e}}=z_1-h_L=140-20.92=119.1\text{ m}$$

Then the extracted power from water and the actual power output of the turbine become

$$\dot{W}_{\text{turbine,e}}=\dot{m}gh_{\text{turbine,e}}=\rho\dot{V}gh_{\text{turbine,e}}$$

$$=(998\text{ kg/m}^3)(0.6\text{ m}^3/\text{s})(9.81\text{ m/s}^2)(119.1\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right)=699.5\text{ kW}$$

$$\dot{W}_{\text{turbine-gen}}=\eta_{\text{turbine-gen}}\dot{W}_{\text{turbine,e}}=(0.80)(699.5\text{ kW})=\mathbf{560\ kW}$$

Discussion Note that a perfect turbine-generator would generate 700 kW of electricity from this resource. The power generated by the actual unit is only 560 kW because of the inefficiencies of the turbine and the generator. Also note that about 15 percent of the elevation head is lost in piping due to pipe friction.

8-155

Solution In a hydroelectric power plant, the flow rate of water, the available elevation head, and the combined turbine-generator efficiency are given. The percent increase in the electric power output of the plant is to be determined when the pipe diameter is tripled.

Assumptions **1** The flow is steady and incompressible. **2** Entrance effects are negligible, and thus the flow is fully developed and friction factor is constant. **3** Minor losses are negligible. **4** Water level is constant.

Properties The density and dynamic viscosity of water at 20°C are ρ = 998 kg/m^3 and μ = 1.002×10^{-3} kg/m·s. The roughness of cast iron pipes is ε = 0.00026 m.

Analysis We take point 1 at the free surface of the reservoir, and point 2 and the reference level at the free surface of the water leaving the turbine site (z_2 = 0). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocities at both points are very low ($V_1 \cong V_2 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad h_{\text{turbine,e}}=z_1-h_L$$

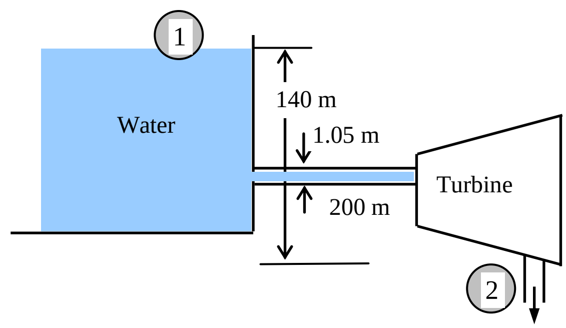
The average velocity, Reynolds number, friction factor, and head loss in the pipe for both cases (pipe diameter being 0.35 m and 1.05 m) are

$$V_1=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4}=\frac{0.6\text{ m}^3/\text{s}}{\pi(0.35\text{ m})^2/4}=6.236\text{ m/s},$$

$$V_2=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4}=\frac{0.6\text{ m}^3/\text{s}}{\pi(1.05\text{ m})^2/4}=0.6929\text{ m/s}$$

$$\text{Re}_1=\frac{\rho V D_h}{\mu}=\frac{(998\text{ kg/m}^3)(6.236\text{ m/s})(0.35\text{ m})}{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}}=2.174\times10^6$$

$$\text{Re}_2=\frac{\rho V D_h}{\mu}=\frac{(998\text{ kg/m}^3)(0.6929\text{ m/s})(1.05\text{ m})}{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}}=0.7247\times10^6$$



which are greater than 4000. Therefore, the flow is turbulent for both cases. The relative roughness of the pipe is

$$\varepsilon/D_1=\frac{0.00026\text{ m}}{0.35\text{ m}}=7.43\times10^{-4} \quad\text{and}\quad \varepsilon/D_2=\frac{0.00026\text{ m}}{1.05\text{ m}}=2.476\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f_1}}=-2.0\log\left(\frac{7.43\times10^{-4}}{3.7}+\frac{2.51}{4.348\times10^6\sqrt{f_1}}\right) \quad\text{and}\quad \frac{1}{\sqrt{f_2}}=-2.0\log\left(\frac{2.476\times10^{-4}}{3.7}+\frac{2.51}{1.449\times10^6\sqrt{f_2}}\right)$$

The friction factors are determined to be f_1 = 0.01847 and f_2 = 0.01545. When the minor losses are negligible, the head losses in the pipes and the head extracted by the turbine are determined to be

$$h_{L1}=f\frac{L}{D}\frac{V^2}{2g}=0.01847\frac{200\text{ m}}{0.35\text{ m}}\frac{(6.236\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=20.92\text{ m},\quad h_{\text{turbine},1}=z_1-h_L=140-20.92=119.1\text{ m}$$

$$h_{L2}=f\frac{L}{D}\frac{V^2}{2g}=0.01545\frac{200\text{ m}}{1.05\text{ m}}\frac{(0.6929\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=0.0720\text{ m},\quad h_{\text{turbine},2}=z_1-h_L=140-0.07=139.9\text{ m}$$

The available or actual power output is proportional to the turbine head. Therefore, the increase in the power output when the diameter is doubled becomes

$$\text{Increase in power output}=\frac{h_{\text{turbine},2}-h_{\text{turbine},1}}{h_{\text{turbine},1}}=\frac{139.9-119.1}{119.1}=\mathbf{0.175}\text{ or }\mathbf{17.5\%}$$

Discussion Note that the power generation of the turbine increases by 17.5 percent when the pipe diameter is tripled at the same flow rate and elevation.

8-156E

Solution The drinking water needs of an office are met by siphoning water through a plastic hose inserted into a large water bottle. The time it takes to fill a glass when the bottle is first opened and when it is empty are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed and the friction factor is constant for the entire pipe. **3** The on/off switch is fully open during filing. **4** The water level in the bottle remains nearly constant during filling. **5** The flow is turbulent (to be verified). **6** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 70°F are ρ = 62.30 lbm/ft^3 and μ = 2.360 lbm/ft·h = 6.556×10^{-4} lbm/ft·s. The plastic pipes are considered to be smooth, and thus their roughness is $\varepsilon = 0$. The total minor loss coefficient is given to be 2.8.

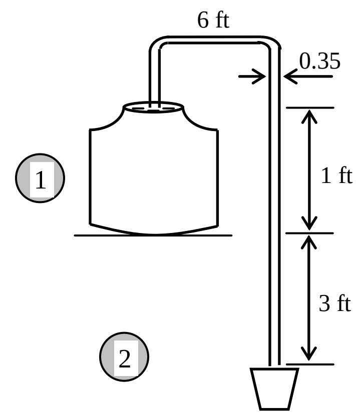
Analysis We take point 1 to be at the free surface of water in the bottle, and point 2 at the exit of the hose., which is also taken to be the reference level ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad z_1 = \alpha_2 \frac{V_2^2}{2g} + h_L$$

where $\alpha_2 = 1$ and $$h_L = \left(f\frac{L}{D} + \sum K_L \right)\frac{V_2^2}{2g} \rightarrow h_L = \left(f\frac{6 \text{ ft}}{0.35/12 \text{ ft}} + 2.8 \right)\frac{V_2^2}{2(32.2 \text{ ft/s}^2)} \qquad (1)$$

since the diameter of the piping system is constant. Then the energy equation becomes

$$z_1 = (1)\frac{V_2^2}{2(32.2 \text{ ft/s}^2)} + h_L \qquad (2)$$



The average velocity in the pipe and the Reynolds number are

$$V_2 = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} \rightarrow V_2 = \frac{\dot{V} \text{ ft}^3\text{/s}}{\pi(0.35/12 \text{ ft})^2/4} \qquad (3)$$

$$\text{Re} = \frac{\rho V_2 D}{\mu} \rightarrow \text{Re} = \frac{(62.3 \text{ lbm/ft}^3)V_2(0.35/12 \text{ ft})}{6.556\times10^{-4} \text{ lbm/ft}\cdot\text{s}} \qquad (4)$$

The friction factor can be determined from the Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}} \right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(0 + \frac{2.51}{\text{Re}\sqrt{f}} \right) \qquad (5)$$

Finally, the filling time of the glass is

$$\Delta t = \frac{V_{\text{glass}}}{\dot{V}} = \frac{0.00835 \text{ ft}^3}{\dot{V} \text{ ft}^3\text{/s}} \qquad (6)$$

These are 6 equations in the 6 unknowns of V_2, $\dot{V}$, h_L, Re, f, and Δt, and solving them simultaneously using an equation solver such as EES with the appropriate z_1 value gives

Case (*a*): The bottle is full and thus $z_1 = 3+1 = 4$ ft:

$V_2 = 5.185$ ft/s, $h_L = 3.58$ ft, $\dot{V} = 0.00346$ ft^3/s, Re = 14,370, $f = 0.02811$, and Δt = **2.4 s**

Case (*b*): The bottle is almost empty and thus $z_1 = 3$ ft:

$V_2 = 4.436$ ft/s, $h_L = 2.69$ ft, $\dot{V} = 0.00296$ ft^3/s, Re = 12,290, $f = 0.02926$, and Δt = **2.8 s**

Note that the flow is turbulent for both cases since Re > 4000.

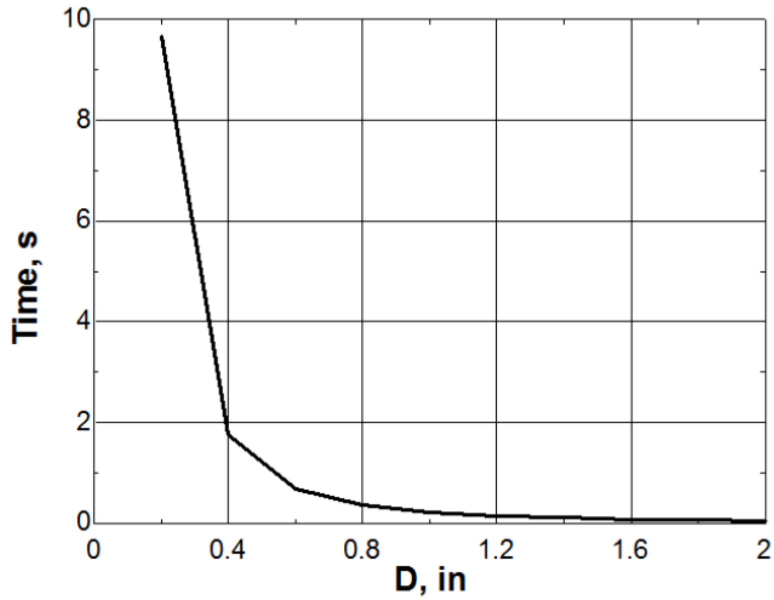
Discussion The filling time of the glass increases as the water level in the bottle drops, as expected.

8-157E

Solution In the previous problem, the effect of the hose diameter on the time required to fill a glass when the bottle is full is to be investigated by varying the pipe diameter from 0.2 to 2 in. in increments of 0.2 in.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
rho=62.3
mu=2.36/3600
nu=mu/rho
g=32.2
z1=4
Volume=0.00835
D=Din/12
Ac=pi*D^2/4
L=6
KL=2.8
eps=0
rf=eps/D
V=Vdot/Ac
"Reynolds number"
Re=V*D/nu
1/sqrt(f)=-2*log10(rf/3.7+2.51/(Re*sqrt(f)))
HL=(f*L/D+KL)*(V^2/(2*g))
z1=V^2/(2*g)+HL
Time=Volume/Vdot
```



D, in	*Time*, s	h_L, ft	Re
0.2	9.66	3.76	6273
0.4	1.75	3.54	17309
0.6	0.68	3.40	29627
0.8	0.36	3.30	42401
1.0	0.22	3.24	55366
1.2	0.15	3.20	68418
1.4	0.11	3.16	81513
1.6	0.08	3.13	94628
1.8	0.06	3.11	107752
2.0	0.05	3.10	120880

Discussion The required time decreases considerably as the tube diameter increases. This is because the irreversible frictional head loss (major loss) in the tube decreases greatly as tube diameter increases. In addition, the minor loss is proportional to V^2. Thus, as tube diameter increases, V decreases, and even the minor losses decrease.

8-158E

Solution The drinking water needs of an office are met by siphoning water through a plastic hose inserted into a large water bottle. The time it takes to fill a glass when the bottle is first opened is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed and the friction factor is constant for the entire pipe. **3** The on/off switch is fully open during filing. **4** The water level in the bottle remains constant during filling. **5** The flow is turbulent (to be verified). **6** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 70°F are ρ = 62.30 lbm/ft^3 and μ = 2.360 lbm/ft·h = 6.556×10^{-4} lbm/ft·s. The plastic pipes are considered to be smooth, and thus their roughness is $\varepsilon = 0$. The total minor loss coefficient is given to be 2.8 during filling.

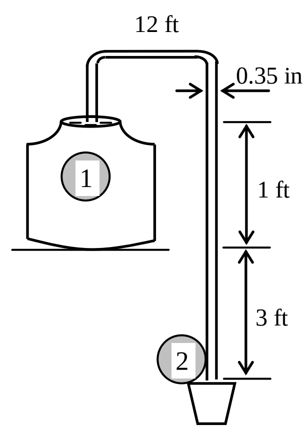
Analysis We take point 1 to be at the free surface of water in the bottle, and point 2 at the exit of the hose, which is also taken to be the reference level ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

where $\alpha_2 = 1$ and $h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g} \rightarrow h_L=\left(f\frac{12\text{ ft}}{0.35/12\text{ ft}}+2.8\right)\frac{V_2^2}{2(32.2\text{ ft/s}^2)}$ (1)

since the diameter of the piping system is constant. Then the energy equation becomes

$$z_1=(1)\frac{V_2^2}{2(32.2\text{ ft/s}^2)}+h_L \qquad (2)$$



The average velocity in the pipe and the Reynolds number are

$$V_2=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4} \rightarrow V_2=\frac{\dot{V}\text{ ft}^3\text{/s}}{\pi(0.35/12\text{ ft})^2/4} \qquad (3)$$

$$\text{Re}=\frac{\rho V_2 D}{\mu} \rightarrow \text{Re}=\frac{(62.3\text{ lbm/ft}^3)V_2(0.35/12\text{ ft})}{1.307\times10^{-3}\text{ lbm/ft}\cdot\text{s}} \qquad (4)$$

The friction factor can be determined from the Colebrook equation,

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D_h}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}}=-2.0\log\left(0+\frac{2.51}{\text{Re}\sqrt{f}}\right) \qquad (5)$$

Finally, the filling time of the glass is

$$\Delta t=\frac{V_{\text{glass}}}{\dot{V}}=\frac{0.00835\text{ ft}^3}{\dot{V}\text{ ft}^3\text{/s}} \qquad (6)$$

These are 6 equations in the 6 unknowns of V_2, $\dot{V}$, h_L, Re, f, and Δt, and solving them simultaneously using an equation solver such as EES with the appropriate z_1 value gives

Case (*a*): The bottle is full and thus z_1 = 3+1 = 4 ft:

V_2 =3.99 ft/s, h_L = 3.75 ft, $\dot{V}$ = 0.002667 ft^3/s, Re = 11,060, f = 0.03007, and Δt = **3.1 s**

Case (*b*): The bottle is almost empty and thus z_1 = 3 ft:

V_2 =3.40 ft/s, h_L = 2.82 ft, $\dot{V}$ = 0.002272 ft^3/s, Re = 9426, f =0.03137, and Δt = **3.7 s**

Note that the flow is turbulent for both cases since Re > 4000.

Discussion The filling times in Prob. 8-129E were 2.4 s and 2.8 s, respectively. Therefore, doubling the tube length increases the filling time by 0.7 s when the bottle is full, and by 0.9 s when it is empty.

8-159

Solution A water tank open to the atmosphere is initially filled with water. The tank is drained to the atmosphere through a 90° horizontal bend of negligible length. The flow rate is to be determined for the cases of the bend being a flanged smooth bend and a miter bend without vanes.

Assumptions **1** The flow is steady and incompressible. **2** The flow is turbulent so that the tabulated value of the loss coefficient can be used. **3** The water level in the tank remains constant. **4** The length of the bend and thus the frictional loss associated with its length is negligible. **5** The entrance is well-rounded, and the entrance loss is negligible.

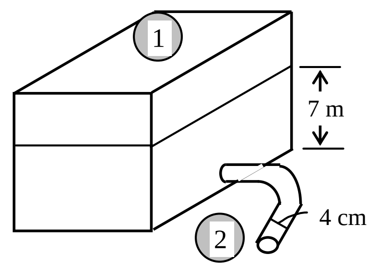
Properties The loss coefficient is $K_L = 0.3$ for a flanged smooth bend and $K_L = 1.1$ for a miter bend without vanes.

Analysis (*a*) We take point 1 at the free surface of the tank, and point 2 at the exit of the bend, which is also taken as the reference level ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

where the head loss is expressed as $h_L = K_L\dfrac{V^2}{2g}$. Substituting and solving for V_2 gives

$$z_1=\alpha_2\frac{V_2^2}{2g}+K_L\frac{V_2^2}{2g} \quad\rightarrow\quad 2gz_1=V_2^2(\alpha_2+K_L) \quad\rightarrow\quad V_2=\sqrt{\frac{2gz_1}{\alpha_2+K_L}}$$



Then the flow rate becomes

$$\dot{V}=A_{\text{pipe}}V_2=\frac{\pi D^2}{4}\sqrt{\frac{2gz_1}{\alpha_2+K_L}}$$

(*a*) Case 1 **Flanged smooth bend (K_L = 0.3):**

$$\dot{V}=A_cV_2=\frac{\pi D^2}{4}\sqrt{\frac{2gz_1}{\alpha_2+K_L}}=\frac{\pi(0.04\text{ m})^2}{4}\sqrt{\frac{2(9.81\text{ m/s}^2)(7\text{ m})}{1.05+0.3}}=\mathbf{0.0127\,m^3/s}=\mathbf{12.7\,L/s}$$

(*b*) Case 2 **Miter bend without vanes (K_L = 1.1):**

$$\dot{V}=A_cV_2=\frac{\pi D^2}{4}\sqrt{\frac{2gz_1}{\alpha_2+K_L}}=\frac{\pi(0.04\text{ m})^2}{4}\sqrt{\frac{2(9.81\text{ m/s}^2)(7\text{ m})}{1.05+1.1}}=\mathbf{0.0100\,m^3/s}=\mathbf{10.0\ L/s}$$

Discussion Note that the type of bend used has a significant effect on the flow rate, and a conscious effort should be made when selecting components in a piping system. If the effect of the kinetic energy correction factor is neglected, $\alpha_2 = 1$ and the flow rates become

(*a*) Case 1 (K_L = 0.3): $$\dot{V}=A_cV_2=\frac{\pi D^2}{4}\sqrt{\frac{2gz_1}{1+K_L}}=\frac{\pi(0.04\text{ m})^2}{4}\sqrt{\frac{2(9.81\text{ m/s}^2)(7\text{ m})}{1+0.3}}=0.0129\ \text{m}^3/\text{s}$$

(*b*) Case 2 (K_L = 1.1): $$\dot{V}=A_cV_2=\frac{\pi D^2}{4}\sqrt{\frac{2gz_1}{1+K_L}}=\frac{\pi(0.04\text{ m})^2}{4}\sqrt{\frac{2(9.81\text{ m/s}^2)(7\text{ m})}{1+1.1}}=0.0102\ \text{m}^3/\text{s}$$

Therefore, the effect of the kinetic energy correction factor is (12.9 - 12.7)/12.7 = 1.6% and (10.2 – 10.0)/10.0 = 2.0%, which is negligible.

8-160

Solution A swimming pool is initially filled with water. A pipe with a well-rounded entrance at the bottom drains the pool to the atmosphere. The initial rate of discharge from the pool and the time required to empty the pool completely are to be determined.

Assumptions **1** The flow is uniform and incompressible. **2** The draining pipe is horizontal. **3** The entrance effects are negligible, and thus the flow is fully developed. **4** The friction factor remains constant (in reality, it changes since the flow velocity and thus the Reynolds number changes during flow). **5** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

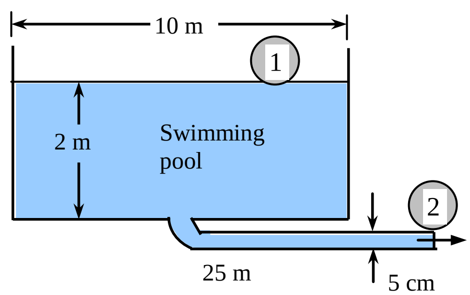
Properties The density and dynamic viscosity of water at 20°C are ρ = 998 kg/m^3 and μ = 1.002×10^{-3} kg/m·s. The friction factor of the pipe is given to be 0.022. Plastic pipes are considered to be smooth, and their surface roughness is ε = 0.

Analysis We take point 1 at the free surface of the pool, and point 2 and the reference level at the exit of the pipe (z_2 = 0), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad z_1 = \alpha_2 \frac{V_2^2}{2g} + h_L$$

where

$$h_L = f\frac{L}{D}\frac{V^2}{2g}$$



since the minor losses are negligible. Substituting and solving for V_2 gives

$$z_1 = \alpha_2 \frac{V_2^2}{2g} + \left(f\frac{L}{D}\right)\frac{V_2^2}{2g} \quad \rightarrow \quad V_2 = \sqrt{\frac{2gz_1}{\alpha_2 + fL/D}}$$

Noting that α_2 = 1 and initially z_1 = 2 m, the initial velocity and flow rate are determined to be

$$V_{2,i} = \sqrt{\frac{2gz_1}{1+fL/D}} = \sqrt{\frac{2(9.81\text{ m/s}^2)(2\text{ m})}{1+0.022(25\text{ m})/(0.05\text{ m})}} = 1.808\text{ m/s}$$

$$\dot{V}_{initial} = V_{2,i}A_c = V_{2,i}(\pi D^2/4) = (1.808\text{ m/s})[\pi(0.05\text{ m})^2/4] = 3.55\times10^{-3}\text{ m}^3\text{/s} = \mathbf{3.55\ L/s}$$

The average discharge velocity at any given time, in general, can be expressed as

$$V_2 = \sqrt{\frac{2gz}{1+fL/D}}$$

where z is the water height relative to the center of the orifice at that time.

We denote the diameter of the pipe by D, and the diameter of the pool by D_o. The flow rate of water from the pool can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{V} = A_c V_2 = \frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D}}$$

Then the amount of water that flows through the pipe during a differential time interval dt is

$$dV = \dot{V}dt = \frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D}}dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the pool,

$$dV = A_{c,\tan k}(-dz) = -\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the pool during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D}}dt = -\frac{\pi D_0^2}{4}dz \rightarrow dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D}{2gz}}dz = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D}{2g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained pool) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D}{2g}}\int_{z=z_1}^{0} z^{-1/2}dz \rightarrow t_f = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D}{2g}}\left.\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0} = \frac{2D_0^2}{D^2}\sqrt{\frac{1+fL/D}{2g}}z_1^{\frac{1}{2}}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{D_0^2}{D^2}\sqrt{\frac{2z_1(1+fL/D)}{g}} = \frac{(10\text{ m})^2}{(0.05\text{ m})^2}\sqrt{\frac{2(2\text{ m})[1+(0.022)(25\text{ m})/(0.05\text{ m})]}{9.81\text{ m/s}^2}} = 88{,}480\text{ s} = \mathbf{24.6\,h}$$

Checking: For plastic pipes, the surface roughness and thus the roughness factor is zero. The Reynolds number at the beginning of draining process is

$$\text{Re} = \frac{\rho V_2 D}{\mu} = \frac{(998\text{ kg/m}^3)(1.808\text{ m/s})(0.05\text{ m})}{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}} = 90{,}040$$

which is greater than 4000. The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(0 + \frac{2.51}{90{,}040\sqrt{f}}\right)$$

It gives $f = 0.0184$. Therefore, the given value of 0.022 is not so accurate. Repeating calculations with this value gives 22.7 h, which is about 8% in error.

Discussion It can be shown by setting $L = 0$ that the draining time without the pipe is only about 7.1 h. Therefore, the pipe in this case increases the draining time by about a factor of 3.5.

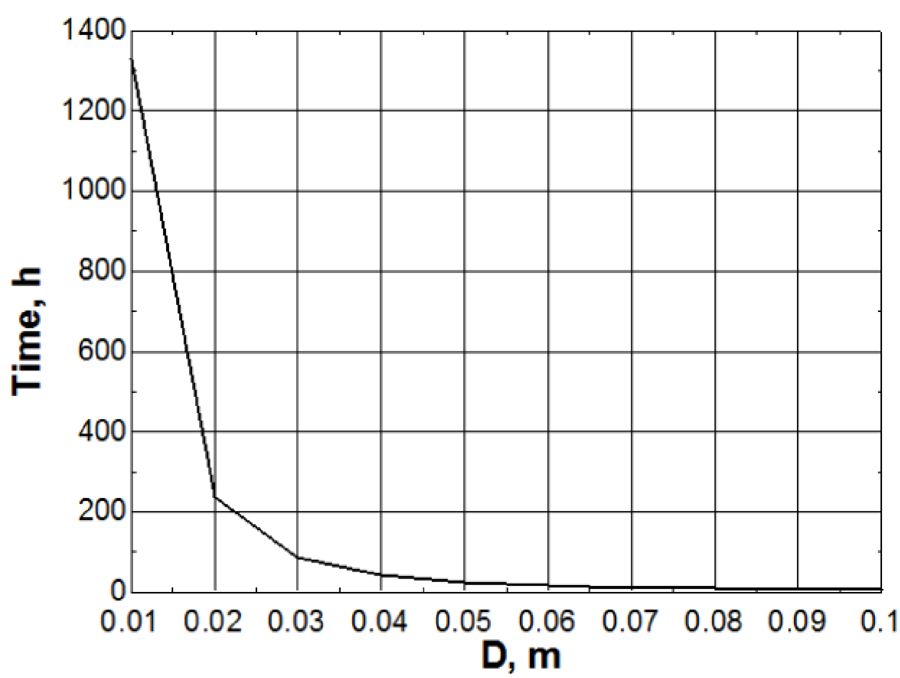
8-161

Solution In the previous problem, the effect of the discharge pipe diameter on the time required to empty the pool completely is to be investigated by varying the pipe diameter from 1 cm to 10 cm in increments of 1 cm.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
rho=998
mu=0.001002
g=9.81
Dtank= 10
Ac=pi*D^2/4
L=25
f=0.022
z1=2
V=(2*g*z1/(1+f*L/D))^0.5
Vdot=V*Ac
Time=(Dtank/D)^2*(2*z1*(1+f*L/D)/g)^0.5/3600
```

D, m	*Time*, h	$V_{initial}$, m/s	Re
0.01	1327.4	0.84	8337
0.02	236.7	1.17	23374
0.03	86.7	1.42	42569
0.04	42.6	1.63	64982
0.05	24.6	1.81	90055
0.06	15.7	1.96	117406
0.07	10.8	2.10	146750
0.08	7.8	2.23	177866
0.09	5.8	2.35	210572
0.10	4.5	2.46	244721



Discussion The required drain time decreases quite rapidly as pipe diameter is increased.

8-162

Solution A swimming pool is initially filled with water. A pipe with a sharp-edged entrance at the bottom drains the pool to the atmosphere. The initial rate of discharge from the pool and the time required to empty the pool completely are to be determined.

Assumptions **1** The flow is uniform and incompressible. **2** The draining pipe is horizontal. **3** The flow is turbulent so that the tabulated value of the loss coefficient can be used. **4** The friction factor remains constant (in reality, it changes since the flow velocity and thus the Reynolds number changes during flow). **5** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 20°C are $\rho = 998$ kg/m^3 and $\mu = 1.002\times10^{-3}$ kg/m·s. The loss coefficient for the sharp-edged entrance is $K_L = 0.5$. Plastic pipes are considered to be smooth, and their surface roughness is $\varepsilon = 0$.

Analysis We take point 1 at the free surface of the pool, and point 2 and the reference level at the exit of the pipe ($z_2 = 0$), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

where $\alpha_2 = 1$ and

$$h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g}=\left(f\frac{L}{D}+K_L\right)\frac{V_2^2}{2g}$$

since the diameter of the piping system is constant. Substituting and solving for V_2 gives

$$z_1=\frac{V_2^2}{2g}+\left(f\frac{L}{D}+K_L\right)\frac{V_2^2}{2g}\quad\rightarrow\quad V_2=\sqrt{\frac{2gz_1}{1+fL/D+K_L}}$$

Noting that initially $z_1 = 2$ m, the initial velocity and flow rate are determined to be

$$V_{2,i}=\sqrt{\frac{2gz_1}{1+fL/D+K_L}}=\sqrt{\frac{2(9.81\text{ m/s}^2)(2\text{ m})}{1+0.022(25\text{ m})/(0.03\text{ m})+0.5}}=1.407\text{ m/s}$$

$$\dot V_{initial}=V_{2,i}A_c=V_{2,i}(\pi D^2/4)=(1.407\text{ m/s})[\pi(0.03\text{ m})^2/4]=9.94\times10^{-4}\text{ m}^3\text{/s}=0.994\text{ L/s}$$

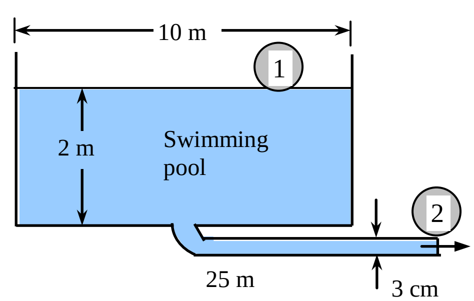
The average discharge velocity at any given time, in general, can be expressed as

$$V_2=\sqrt{\frac{2gz}{1+fL/D+K_L}}$$

where z is the water height relative to the center of the orifice at that time.

We denote the diameter of the pipe by D, and the diameter of the pool by D_o. The flow rate of water from the pool can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{\boldsymbol{V}}=A_cV_2=\frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D+K_L}}$$



Then the amount of water that flows through the pipe during a differential time interval dt is

$$d\boldsymbol{V}=\dot{\boldsymbol{V}}dt=\frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D+K_L}}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the pool,

$$d\boldsymbol{V} = A_{c,\tan k}(-dz) = -\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the pool during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{\frac{2gz}{1+fL/D+K_L}}dt = -\frac{\pi D_0^2}{4}dz \rightarrow dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2gz}}dz = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2g}}z^{-\frac{1}{2}}dz$$ The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained pool) gives

$$\int_{t=0}^{t_f}dt = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2g}}\int_{z=z_1}^{0}z^{-1/2}dz \rightarrow t_f = -\frac{D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2g}}\left|\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0} = \frac{2D_0^2}{D^2}\sqrt{\frac{1+fL/D+K_L}{2g}}z_1^{\frac{1}{2}}$$ Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{D_0^2}{D^2}\sqrt{\frac{2z_1(1+fL/D+K_L)}{g}} = \frac{(10\text{ m})^2}{(0.03\text{ m})^2}\sqrt{\frac{2(2\text{ m})[1+(0.022)(25\text{ m})/(0.03\text{ m})+0.5]}{9.81\text{ m/s}^2}} = 316{,}000\text{ s} = \mathbf{87.8\,h}$$ This is a change of (87.8-86.7)/86.7 = 0.013 or 1.3%. Therefore, the minor loss in this case is **truly minor**.

Checking: For plastic pipes, the surface roughness and thus the roughness factor is zero. The Reynolds number at the beginning of draining process is

$$\text{Re} = \frac{\rho V_2 D}{\mu} = \frac{(998\text{ kg/m}^3)(1.407\text{ m/s})(0.03\text{ m})}{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}} = 42{,}030$$

which is greater than 4000. The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(0+\frac{2.51}{42{,}030\sqrt{f}}\right)$$

It gives $f = 0.022$. Therefore, the given value of 0.022 is accurate.

Discussion It can be shown by setting $L = 0$ that the draining time without the pipe is only about 24 h. Therefore, the pipe in this case increases the draining time more than 3 folds.

8-163

Solution A bypass graft was attached to a coronary artery that was 75% blocked by atherosclerotic plaque. These two "pipes" are effectively connected in parallel. The velocity in the gap created by the plaque in the coronary artery is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** For simplicity, the flow is considered to be turbulent in both "pipes" and thus the friction factor is independent of Reynolds number and the friction factor is considered the same for both "pipes." **3** The material properties for the graft and artery are the same. **4** Any minor losses are negligible. **5** The plaque gap is cylindrical.

Analysis When 2 pipes are parallel in a piping system like the human circulation, the head loss for each pipe must be the same. Without any minor losses, the head losses for fully developed flow can be expressed as:

$$h_L = f\frac{L}{D}\frac{V^2}{2g} = f\frac{L}{D}\frac{1}{2g}\left(\frac{Q}{A_c}\right)^2 = f\frac{L}{D}\frac{1}{2g}\left(\frac{Q}{\pi D^2/4}\right)^2 = 8f\frac{L}{D}\frac{1}{g}\frac{Q^2}{\pi^2 D^4} = 8f\frac{L}{g\pi^2}\frac{Q^2}{D^5}$$

For subscripts to be used in this solution, g is graft, c is coronary artery, and p is plaque. Thus, the following variables are:

$$L_g = 20mm, D_g = 4mm, Q_g = 0.45\ lpm$$
$$L_c = 15mm, D_c = 5mm, Q_c = ?$$
$$D_p = 0.75D_c, Q_p = ?$$

Since the coronary artery and graft are in parallel, the head loss must be equal for both pipes.

$$h_{L,c} = 8f_c\frac{L_c}{g\pi^2}\frac{Q_c^{\,2}}{D_c^{\,5}}$$
$$h_{L,g} = 8f_g\frac{L_g}{g\pi^2}\frac{Q_g^{\,2}}{D_g^{\,5}}$$
$$h_{L,c} = 8f_c\frac{L_c}{g\pi^2}\frac{Q_c^{\,2}}{D_c^{\,5}} = 8f_g\frac{L_g}{g\pi^2}\frac{Q_g^{\,2}}{D_g^{\,5}} = h_{L,g}$$
$$Q_c^{\,2} = \frac{L_g}{L_c}\frac{D_c^{\,5}}{D_g^{\,5}}Q_g^{\,2}$$
$$Q_c^{\,2} = 15.1\ {cm^3}/{s}$$

We know that

$Q_c = Q_p$, so $Q_c = \pi V_p D_p^{\,2}$ and thus, $V_p = \frac{Q_c}{\pi D_p^{\,2}} = 3.4\ x\ 10^{-3}\ {cm}/{s}$

Fundamentals of Engineering (FE) Exam Problems

8-164

The average velocity for fully developed laminar pipe flow is

(*a*) $V_{max}/2$ (*b*) $V_{max}/3$ (*c*) V_{max} (*d*) $2V_{max}/3$ (*e*) $3V_{max}/4$

Answer (*a*) $V_{max}/2$

8-165

The Reynolds number is not a function of

(*a*) Fluid velocity (*b*) Fluid density (*c*) Characteristic length
(*d*) Surface roughness (*e*) Fluid viscosity

Answer (*d*) Surface roughness

8-166

Air flows in a 5 cm × 8 cm cross section rectangular duct at a velocity of 4 m/s at 1 atm and 15°C. The Reynolds number for this flow is

(*a*) 13,605 (*b*) 16,745 (*c*) 17,690 (*d*) 21,770 (*e*) 23,235

Answer (*b*) 16,745

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
a=0.05 [m]
b=0.08 [m]
V=4 [m/s]
T=15 [C]
nu=1.47E-5 [m^2/s] "Table A-9"
A_c=a*b
p=2*(a+b)
D_h=4*A_c/p
Re=V*D_h/nu
```

8-167

Air at 1 atm and 20°C flows in a 4-cm-diameter tube. The maximum velocity of air to keep the flow laminar is

(*a*) 0.872 m/s (*b*) 1.52 m/s (*c*) 2.14 m/s (*d*) 3.11 m/s (*e*) 3.79 m/s

Answer (*a*) 0.872 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=20 [C]
D=0.04 [m]
nu=1.516E-5 [m^2/s] "Table A-9"
Re=V*D/nu
Re=2300
```

8-168

Consider laminar flow of water in a 0.8-cm-diameter pipe at a rate of 1.15 L/min. The velocity of water halfway between the surface and the center of the pipe is

(*a*) 0.381 m/s (*b*) 0.762 m/s (*c*) 1.15 m/s (*d*) 0.874 m/s (*e*) 0.572 m/s

Answer (*e*) 0.572 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.008 [m]
V_dot=1.15 [L/min]*Convert(L/min, m^3/s)
A_c=pi*D^2/4
V_avg=V_dot/A_c
Radius=D/2
r=D/4
u=2*V_avg*(1-r^2/Radius^2)
```

8-169

Consider laminar flow of water at 15°C in a 0.7-cm-diameter pipe at a velocity of 0.4 m/s. The pressure drop of water for a pipe length of 50 m is

(*a*) 6.8 kPa (*b*) 8.7 kPa (*c*) 11.5 kPa (*d*) 14.9 kPa (*e*) 17.3 kPa

Answer (*d*) 14.9 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=15 [C]
D=0.007 [m]
V=0.4 [m/s]
L=50 [m]
mu=1.138E-3 [kg/m-s] "Table A-3"
DELTAP=(32*mu*L*V)/D^2
```

8-170

Engine oil at 40°C [ρ = 876 kg/m^3, μ = 0.2177 kg/m·s] flows in a 20-cm-diameter pipe at a velocity of 1.2 m/s. The pressure drop of oil for a pipe length of 20 m is

(*a*) 4180 Pa (*b*) 5044 Pa (*c*) 6236 Pa (*d*) 7419 Pa (*e*) 8615 Pa

Answer (*a*) 4180 Pa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=20 [C]
D=0.20 [m]
V=1.2 [m/s]
L=20 [m]
rho=876 [kg/m^3] "Table A-7"
mu=0.2177 [kg/m-s] "Table A-7"
Re=rho*V*D/mu
f=64/Re
DELTAP=f*L/D*rho*V^2/2
```

8-171

A fluid flows in a 25-cm-diameter pipe at a velocity of 4.5 m/s. If the pressure drop along the pipe is estimated to be 6400 Pa, the required pumping power to overcome this pressure drop is

(*a*) 452 W (*b*) 640 W (*c*) 923 W (*d*) 1235 W (*e*) 1508 W

Answer (*c*) 923 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.25 [m]
V=4.5 [m/s]
DELTAP=4180 [Pa]
A_c=pi*D^2/4
V_dot=V*A_c
W_dot_pump=V_dot*DELTAP
```

8-172

Water flows in a 15-cm-diameter pipe at a velocity of 1.8 m/s. If the head loss along the pipe is estimated to be 16 m, the required pumping power to overcome this head loss is

(*a*) 3.22 kW (*b*) 3.77 kW (*c*) 4.45 kW (*d*) 4.99 kW (*e*) 5.54 kW

Answer (*d*) 4.99 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=5 [C]
D=0.15 [m]
V=1.8 [m/s]
h_L=16 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
A_c=pi*D^2/4
m_dot=rho*V*A_c
W_dot_pump=m_dot*g*h_L
```

8-173

The pressure drop for a given flow is determined to be 100 Pa. For the same flow rate, if we reduce the diameter of the pipe by half, the pressure drop will be

(*a*) 25 Pa (*b*) 50 Pa (*c*) 200 Pa (*d*) 400 Pa (*e*) 1600 Pa

Answer (*e*) 1600 Pa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D1=1 [m]
DELTAP_1=100 [Pa]
D2=0.5 [m]
DELTAP_2=DELTAP_1*(D1/D2)^4
```

8-174

Air at 1 atm and 25°C [$\nu = 1.562 \times 10^{-5}$ m^2/s] flows in a 9-cm-diameter cast iron pipe at a velocity of 5 m/s. The roughness of the pipe is 0.26 mm. The head loss for a pipe length of 24 m is

(*a*) 8.1 m (*b*) 10.2 m (*c*) 12.9 m (*d*) 15.5 m (*e*) 23.7 m

Answer (*b*) 10.2 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=25 [C]
D=0.09 [m]
V=5 [m/s]
epsilon=0.26E-3 [m]
L=24 [m]
nu=1.562E-5 [m^2/s] "Table A-9"
g=9.81 [m/s^2]
Re=V*D/nu
1/sqrt(f)=-2*log10(epsilon/D*1/3.7+2.51/(Re*sqrt(f)))
h_L=f*L/D*V^2/(2*g)
```

8-175

Consider air flow in a 10-cm-diameter pipe at a high velocity so that the Reynolds number is very large. The roughness of the pipe is 0.002 mm. The friction factor for this flow is

(*a*) 0.0311 (*b*) 0.0290 (*c*) 0.0247 (*d*) 0.0206 (*e*) 0.0163

Answer (*e*) 0.0163

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.10 [m]
epsilon=0.045E-3 [m]
1/sqrt(f)=-2*log10(epsilon/D*1/3.7)
```

8-176

Air at 1 atm and 40°C flows in a 8-cm-diameter pipe at a rate of 2500 L/min. The friction factor is determined from the Moody chart to be 0.027. The required power input to overcome the pressure drop for a pipe length of 150 m is

(*a*) 310 W (*b*) 188 W (*c*) 132 W (*d*) 81.7 W (*e*) 35.9 W

Answer (*d*) 81.7 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=40 [C]
D=0.08 [m]
V_dot=2500 [L/min]*Convert(L/min, m^3/s)
f=0.027
L=150 [m]
rho=1.127 [kg/m^3] "Table A-9"
A_c=pi*D^2/4
V=V_dot/A_c
DELTAP=f*L/D*rho*V^2/2
W_dot=V_dot*DELTAP
```

8-177

Water at 10°C [ρ = 999.7 kg/m^3, μ = 1.307×10^{-3} kg/m·s] is to be transported in a 5-cm-diamater, 30-m-long circular pipe. The roughness of the pipe is 0.22 mm. If the pressure drop in the pipe is not to exceed 19 kPa, the maximum flow rate of water is

(*a*) 324 L/min (*b*) 281 L/min (*c*) 243 L/min (*d*) 195 L/min (*e*) 168 L/min

Answer (*e*) 168 L/min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=10 [C]
D=0.05 [m]
L=30 [m]
epsilon=0.22E-3 [m]
DELTAP=19000 [Pa]
rho=999.7 [kg/m^3] "Table A-3"
mu=1.307E-3 [kg/m-s] "Table A-3"
g=9.81 [m/s^2]
A_c=pi*D^2/4
V=V_dot/A_c
Re=rho*V*D/mu
1/sqrt(f)=-2*log10(epsilon/D*1/3.7+2.51/(Re*sqrt(f)))
DELTAP=f*L/D*rho*V^2/2
V_dot_minh=V_dot*Convert(m^3/s, L/min)
```

8-178

The valve in a piping system causes a 3.1 m head loss. If the velocity of the flow is 6 m/s, the loss coefficient of this valve is

(*a*) 0.87 (*b*) 1.69 (*c*) 1.25 (*d*) 0.54 (*e*) 2.03

Answer (*b*) 1.69

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_L=3.1 [m]
V=6 [m/s]
g=9.81 [m/s^2]
K_L=h_L/(V^2/(2*g))
```

8-179

Consider a sharp-edged pipe exit for fully developed laminar flow of a fluid. The velocity of the flow is 4 m/s. This minor loss is equivalent to a head loss of

(*a*) 0.72 m (*b*) 1.16 m (*c*) 1.63 m (*d*) 2.0 m (*e*) 4.0 m

Answer (*c*) 1.63 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=4 [m/s]
alpha=2
K_L=alpha
g=9.81 [m/s^2]
h_L=K_L*V^2/(2*g)
```

8-180

A water flow system involves a 180° return bend (threaded) and a 90° miter bend (without vanes). The velocity of water is 1.2 m/s. The minor losses due to these bends are equivalent to a pressure loss of

(*a*) 648 Pa (*b*) 933 Pa (*c*) 1255 Pa (*d*) 1872 Pa (*e*) 2600 Pa

Answer (*d*) 1872 Pa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
K_L1=1.5
K_L2=1.1
V=1.2 [m/s]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
K_L_total=K_L1+K_L2
h_L=K_L_total*V^2/(2*g)
DELTAP=h_L*rho*g
```

8-181

A constant-diameter piping system involves multiple flow restrictions with a total loss coefficient of 4.4. The friction factor of piping is 0.025 and the diameter of the pipe is 7 cm. These minor losses are equivalent to the losses in a pipe of length

(*a*) 12.3 m (*b*) 9.1 m (*c*) 7.0 m (*d*) 4.4 m (*e*) 2.5 m

Answer (*a*) 12.3 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
K_L=4.4
f=0.025
D=0.07 [m]
L_equiv=D/f*K_L
```

8-182

Air flows in an 8-cm-diameter, 33-m-long pipe at a velocity of 5.5 m/s. The piping system involves multiple flow restrictions with a total minor loss coefficient of 2.6. The friction factor of pipe is obtained from the Moody chart to be 0.025. The total head loss of this piping system is

(*a*) 13.5 m (*b*) 7.6 m (*c*) 19.9 m (*d*) 24.5 m (*e*) 4.2 m

Answer (*c*) 19.9 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.08 [m]
L=33 [m]
V=5.5 [m/s]
K_L=2.6
f=0.025
g=9.81 [m/s^2]
h_L=(f*L/D+K_L)*V^2/(2*g)
```

8-183

Consider a pipe that branches out into two parallel pipes and then rejoins at a junction downstream. The two parallel pipes have the same lengths and friction factors. The diameters of the pipes are 2 cm and 4 cm. If the flow rate in one pipe is 10 L/min, the flow rate in the other pipe is

(*a*) 10 L/min (*b*) 3.3 L/min (*c*) 100 L/min (*d*) 40 L/min (*e*) 56.6 L/min

Answer (*e*) 56.6 L/min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D1=2 [cm]
D2=4 [cm]
V_dot_1=10 [L/min]
V_dot_2/V_dot_1=(D2/D1)^2.5
```

8-184

Consider a pipe that branches out into two parallel pipes and then rejoins at a junction downstream. The two parallel pipes have the same lengths and friction factors. The diameters of the pipes are 2 cm and 4 cm. If the head loss in one pipe is 0.5 m, the head loss in the other pipe is

(*a*) 0.5 m (*b*) 1 m (*c*) 0.25 m (*d*) 2 m (*e*) 0.125 m

Answer (*a*) 0.5 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D1=2 [cm]
D2=4 [cm]
h_L1=0.5 [m]
h_L2=h_L1
```

8-185

A pump moves water from a reservoir to another reservoir through a piping system at a rate of 0.15 m^3/min. Both reservoirs are open the atmosphere. The elevation difference between the two reservoirs is 35 m and the total head loss is estimated to be 4 m. If the efficiency of the motor-pump unit is 65 percent, the electrical power input to the motor of the pump is

(*a*) 1664 W (*b*) 1472 W (*c*) 1238 W (*d*) 983 W (*e*) 805 W

Answer (*b*) 1472 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_dot=(0.15/60) [m^3/s]
DELTAz=35 [m]
h_L=4 [m]
eta_motor_pump=0.65
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
h_pump_u=DELTAz+h_L
W_dot_elect=rho*V_dot*g*h_pump_u/eta_motor_pump
```

8-186

Consider a pipe that branches out into three parallel pipes and then rejoins at a junction downstream. All three pipes have the same diameters (D = 3 cm) and friction factors (f = 0.018). The lengths of pipe 1 and pipe 2 are 5 m and 8 m, respectively while the velocities of the fluid in pipe 2 and pipe 3 are 2 m/s and 4 m/s, respectively. The length of pipe 3 is

(*a*) 8 m (*b*) 5 m (*c*) 4 m (*d*) 2 m (*e*) 1 m

Answer (*d*) 2 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D1=0.03 [m]
D2=0.03 [m]
D3=0.03 [m]
f1=0.018
f2=0.018
f3=0.018
L1=5 [m]
L2=8 [m]
V2=2 [m/s]
V3=4 [m/s]
g=9.81 [m/s^2]
h_L1=f1*L1/D1*V1^2/(2*g)
h_L2=f2*L2/D2*V2^2/(2*g)
h_L3=f3*L3/D3*V3^2/(2*g)
h_L2=h_L1
h_L3=h_L1
```

Design and Essay Problems

8-187 to 8-190

Solution Students' essays and designs should be unique and will differ from each other.

Solutions Manual for

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Chapter 9

DIFFERENTIAL ANALYSIS OF FLUID FLOW

General and Mathematical Background Problems

9-1C

Solution We are to express the divergence theorem in words.

Analysis For vector $\vec{G}$, **the volume integral of the divergence of $\vec{G}$ over volume *V* is equal to the surface integral of the normal component of $\vec{G}$ taken over the surface *A* that encloses the volume**.

Discussion The divergence theorem is also called *Gauss's theorem*.

9-2C

Solution We are to explain the fundamental differences between a flow domain and a control volume.

Analysis **A *control volume* is used in an integral, control volume solution**. It is a volume over which all mass flow rates, forces, etc. are specified over the entire control surface of the control volume. In a control volume analysis we do not know or care about details *inside* the control volume. Rather, we solve for gross features of the flow such as net force acting on a body. **A *flow domain*, on the other hand, is also a volume, but is used in a differential analysis**. Differential equations of motion are solved everywhere inside the flow domain, and we *are* interested in all the details inside the flow domain.

Discussion Note that we also need to specify what is happening at the boundaries of a flow domain – these are called *boundary conditions*.

9-3C

Solution We are to explain what we mean by coupled differential equations.

Analysis A set of coupled differential equations simply means that **the equations are dependent on each other and must be solved together rather than separately**. For example, the equations of motion for fluid flow involve velocity variables in both the conservation of mass equation and the momentum equation. To solve for these variables, we must solve the coupled set of differential equations together.

Discussion In some very simple fluid flow problems, the equations become uncoupled, and are easier to solve.

9-4C

Solution We are to discuss the number of unknowns and the equations needed to solve for those unknowns for a three-dimensional, unsteady, incompressible flow field.

Analysis There are **four unknowns** (velocity components *u*, *v*, *w*, and pressure *P*) and thus we need to solve **four equations**:

- one from conservation of mass which is a scalar equation
- three from Newton's second law which is a vector equation

Discussion These equations are also coupled in general.

9-5C

Solution We are to discuss the number of unknowns and the equations needed to solve for those unknowns for a two-dimensional, unsteady, compressible flow field with significant variations in both temperature and density.

Analysis There are **five unknowns** (velocity components u and v, and ρ, T, and P) and thus we need to solve **five equations**:

- one from conservation of mass which is a scalar equation
- two from Newton's second law which is a vector equation
- one from the energy equation which is a scalar equation
- one from an equation of state (e.g., ideal gas law) which is a scalar equation

Discussion These equations are also coupled in general.

9-6C

Solution We are to discuss the number of unknowns and the equations needed to solve for those unknowns for a two-dimensional, unsteady, incompressible flow field.

Analysis There are **three unknowns** (velocity components u, and v, and pressure P) and thus we need to solve **three equations**:

- one from conservation of mass which is a scalar equation
- two from Newton's second law which is a vector equation

Discussion These equations are also coupled in general.

9-7

Solution We are to transform a position from Cartesian to cylindrical coordinates.

Analysis We use the coordinate transformations provided in this chapter,

$$r = \sqrt{x^2 + y^2} = \sqrt{(2\text{ m})^2 + (4\text{ m})^2} = 4.47214\text{ m} \tag{1}$$

and

$$\theta = \tan^{-1}\frac{y}{x} = \tan^{-1}\left(\frac{4\text{ m}}{2\text{ m}}\right) = 63.43495° = 1.10715\text{ radians} \tag{2}$$

Coordinate z remains unchanged. Thus, to three significant digits,

Position in cylindrical coordinates: $$\vec{x} = (r, \theta, z) = (\mathbf{4.47\,m, 1.11\,radians, -1\,m}) \tag{3}$$

Discussion Notice that the units of θ are radians – a dimensionless unit.

9-8

Solution We are to transform a position from cylindrical to Cartesian coordinates.

Analysis We use the coordinate transformations provided in this chapter,

$$x = r\cos\theta = (5\text{ m})\cos(\pi/3\text{ radians}) = (5\text{ m})\cos(60^\circ) = 2.5\text{ m} \qquad (1)$$

and

$$y = r\sin\theta = (5\text{ m})\sin(\pi/3\text{ radians}) = (5\text{ m})\sin(60^\circ) = 4.33013\text{ m} \qquad (2)$$

Coordinate z remains unchanged. Thus, to three significant digits,

Position in cylindrical coordinates: $$\vec{x} = (x,y,z) = (\mathbf{2.50\ m, 4.33\ m, 1.27\ m}) \qquad (3)$$

Discussion You can verify your answer by using the reverse equations, as in the previous problem.

9-9

Solution We are to calculate a truncated Taylor series expansion for a given function and compare our result with the exact value.

Analysis The algebra here is simple since $d(e^x)/dx = e^x$. The Taylor series expansion is

Taylor series expansion: $$f(x_0 + dx) = e^{x_0} + e^{x_0}dx + \frac{1}{2}e^{x_0}dx^2 + \frac{1}{3\times 2}e^{x_0}dx^3 + ... \qquad (1)$$

We plug $x_0 = 0$ and $dx = -0.1$ into Eq. 1,

Truncated Taylor series expansion:

$$f(-0.1) \approx 1 + 1\times(-0.1) + \frac{1}{2}\times 1\times(-0.1)^2 + \frac{1}{6}\times 1\times(-0.1)^3 = 0.9048333... \qquad (2)$$

We compare Eq. 2 with the exact value,

Exact value: $$f(-0.1) = e^{-0.1} = 0.904837418... \qquad (3)$$

Comparing Eqs. 2 and 3 we see that **our approximation is good to four or five significant digits**.

Discussion The smaller the value of dx, the better the approximation. You can easily convince yourself of this by trying $dx = 0.01$ instead.

9-10

Solution We are to calculate the divergence of a given vector.

Analysis The divergence of $\vec{G}$ is the dot product of the del operator $\vec{\nabla} = \frac{\partial}{\partial x}\vec{i} + \frac{\partial}{\partial y}\vec{j} + \frac{\partial}{\partial z}\vec{k}$ with $\vec{G}$, which gives

Divergence of $\vec{G}$: $$\vec{\nabla}\cdot\vec{G} = \left(\frac{\partial}{\partial x}\vec{i} + \frac{\partial}{\partial y}\vec{j} + \frac{\partial}{\partial z}\vec{k}\right)\cdot\left(2xz\vec{i} - \frac{1}{2}x^2\vec{j} - z^2\vec{k}\right) = 2z + 0 - 2z = \mathbf{0}$$

It turns out that for this special case, the divergence of $\vec{G}$ is zero.

Discussion If $\vec{G}$ were a velocity vector, this would mean that the flow field is incompressible.

9-11

Solution We are to expand the given equation in Cartesian coordinates and verify it.

Analysis In Cartesian coordinates the del operator is $\vec{\nabla} = \frac{\partial}{\partial x}\vec{i} + \frac{\partial}{\partial y}\vec{j} + \frac{\partial}{\partial z}\vec{k}$ and we let $\vec{F} = F_x\vec{i} + F_y\vec{j} + F_z\vec{k}$ and $\vec{G} = G_x\vec{i} + G_y\vec{j} + G_z\vec{k}$. The left hand side of the equation is thus

$$\vec{\nabla}\cdot\left(\vec{F}\vec{G}\right) = \begin{bmatrix} \frac{\partial}{\partial x} & \frac{\partial}{\partial y} & \frac{\partial}{\partial z} \end{bmatrix} \begin{bmatrix} F_xG_x & F_xG_y & F_xG_z \\ F_yG_x & F_yG_y & F_yG_z \\ F_zG_x & F_zG_y & F_zG_z \end{bmatrix}$$

$$= \left[\frac{\partial}{\partial x}\left(F_xG_x\right) + \frac{\partial}{\partial y}\left(F_yG_x\right) + \frac{\partial}{\partial z}\left(F_zG_x\right)\right]\vec{i} + \left[\frac{\partial}{\partial x}\left(F_xG_y\right) + \frac{\partial}{\partial y}\left(F_yG_y\right) + \frac{\partial}{\partial z}\left(F_zG_y\right)\right]\vec{j} + \left[\frac{\partial}{\partial x}\left(F_xG_z\right) + \frac{\partial}{\partial y}\left(F_yG_z\right) + \frac{\partial}{\partial z}\left(F_zG_z\right)\right]\vec{k} \qquad (1)$$

We use the product rule on each term in Eq. 1 and rearrange to get

Left hand side:

$$\vec{\nabla}\cdot\left(\vec{F}\vec{G}\right) = \left[G_x\left(\frac{\partial F_x}{\partial x} + \frac{\partial F_y}{\partial y} + \frac{\partial F_z}{\partial z}\right) + \left(F_x\frac{\partial}{\partial x} + F_y\frac{\partial}{\partial y} + F_z\frac{\partial}{\partial z}\right)G_x\right]\vec{i} + \left[G_y\left(\frac{\partial F_x}{\partial x} + \frac{\partial F_y}{\partial y} + \frac{\partial F_z}{\partial z}\right) + \left(F_x\frac{\partial}{\partial x} + F_y\frac{\partial}{\partial y} + F_z\frac{\partial}{\partial z}\right)G_y\right]\vec{j} + \left[G_z\left(\frac{\partial F_x}{\partial x} + \frac{\partial F_y}{\partial y} + \frac{\partial F_z}{\partial z}\right) + \left(F_x\frac{\partial}{\partial x} + F_y\frac{\partial}{\partial y} + F_z\frac{\partial}{\partial z}\right)G_z\right]\vec{k} \qquad (2)$$

We recognize that $\frac{\partial F_x}{\partial x} + \frac{\partial F_y}{\partial y} + \frac{\partial F_z}{\partial z} = \vec{\nabla}\cdot\vec{F}$ and $F_x\frac{\partial}{\partial x} + F_y\frac{\partial}{\partial y} + F_z\frac{\partial}{\partial z} = \vec{F}\cdot\vec{\nabla}$. Eq. 2 then becomes

Left hand side:

$$\vec{\nabla}\cdot\left(\vec{F}\vec{G}\right) = \left[G_x\left(\vec{\nabla}\cdot\vec{F}\right) + \left(\vec{F}\cdot\vec{\nabla}\right)G_x\right]\vec{i} + \left[G_y\left(\vec{\nabla}\cdot\vec{F}\right) + \left(\vec{F}\cdot\vec{\nabla}\right)G_y\right]\vec{j} + \left[G_z\left(\vec{\nabla}\cdot\vec{F}\right) + \left(\vec{F}\cdot\vec{\nabla}\right)G_z\right]\vec{k} \qquad (3)$$

After rearrangement, Eq. 3 becomes

Left hand side:

$$\vec{\nabla}\cdot\left(\vec{F}\vec{G}\right) = \left(G_x\vec{i} + G_y\vec{j} + G_z\vec{k}\right)\left(\vec{\nabla}\cdot\vec{F}\right) + \left(\vec{F}\cdot\vec{\nabla}\right)\left(G_x\vec{i} + G_y\vec{j} + G_z\vec{k}\right) \qquad (4)$$

Finally, recognizing vector $\vec{G}$ twice in Eq. 4, we see that the left hand side of the given equation is identical to the right hand side, and **the given equation is verified**.

Discussion It may seem surprising, but $\vec{F}\vec{G} \neq \vec{G}\vec{F}$.

9-12

Solution We are to prove the equation.

Analysis We let $\vec{F} = \rho\vec{V}$ and $\vec{G} = \vec{V}$. Using Eq. 1 of the previous problem, we have

$$\vec{\nabla}\cdot\left(\rho\vec{V}\vec{V}\right) = \vec{V}\vec{\nabla}\cdot\left(\rho\vec{V}\right) + \left(\rho\vec{V}\cdot\vec{\nabla}\right)\vec{V} \qquad (1)$$

However, since the density is not operated on in the second term of Eq. 1, it can be brought outside of the parenthesis, even though it is not a constant in general. Equation 1 can thus be written as

$$\boxed{\vec{\nabla}\cdot\left(\rho\vec{V}\vec{V}\right) = \vec{V}\vec{\nabla}\cdot\left(\rho\vec{V}\right) + \rho\left(\vec{V}\cdot\vec{\nabla}\right)\vec{V}} \qquad (2)$$

Discussion Equation 2 was used in this chapter in the derivation of the alternative form of Cauchy's equation.

9-13

Solution We are to transform cylindrical velocity components to Cartesian velocity components.

Analysis We apply trigonometry, recognizing that the angle between u and u_r is θ, and the angle between v and u_θ is also θ,

x component of velocity: $$\boxed{u = u_r\cos\theta - u_\theta\sin\theta} \qquad (1)$$

Similarly,

y component of velocity: $$\boxed{v = u_r\sin\theta + u_\theta\cos\theta} \qquad (2)$$

The transformation of the z component is trivial,

z component of velocity: $$\boxed{w = u_z} \qquad (3)$$

Discussion These transformations come in handy.

9-14

Solution We are to transform Cartesian velocity components to cylindrical velocity components.

Analysis We apply trigonometry, recognizing that the angle between u and u_r is θ, and the angle between v and u_θ is also θ,

u_r component of velocity: $$\boxed{u_r = u\cos\theta + v\sin\theta} \qquad (1)$$

Similarly,

u_θ component of velocity:: $$\boxed{u_\theta = -u\sin\theta + v\cos\theta} \qquad (2)$$

The transformation of the z component is trivial,

z component of velocity: $$\boxed{u_z = w} \qquad (3)$$

Discussion You can also obtain Eqs. 1 and 2 by solving Eqs. 1 and 2 of the previous problem simultaneously.

9-15

Solution We are to transform a given set of Cartesian coordinates and velocity components into cylindrical coordinates and velocity components.

Analysis First we apply the coordinate transformations given in this chapter,

$$r = \sqrt{x^2 + y^2} = \sqrt{(0.40\text{ m})^2 + (0.20\text{ m})^2} = 0.4472\text{ m} \qquad (1)$$

$$\theta = \tan^{-1}\frac{y}{x} = \tan^{-1}\left(\frac{0.20\text{ m}}{0.40\text{ m}}\right) = 26.565^\circ = 0.4636\text{ radians} \qquad (2)$$

Next we apply the results of the previous problem,

$$u_r = u\cos\theta + v\sin\theta = 10.3\frac{\text{m}}{\text{s}} \times \frac{0.40\text{ m}}{0.4472\text{ m}} - 5.6\frac{\text{m}}{\text{s}} \times \frac{0.20\text{ m}}{0.4472\text{ m}} = 6.708\frac{\text{m}}{\text{s}} \qquad (3)$$

$$u_\theta = -u\sin\theta + v\cos\theta = -10.3\frac{\text{m}}{\text{s}} \times \frac{0.20\text{ m}}{0.4472\text{ m}} - 5.6\frac{\text{m}}{\text{s}} \times \frac{0.40\text{ m}}{0.4472\text{ m}} = -9.615\frac{\text{m}}{\text{s}} \qquad (4)$$

Note that we have used the fact that $x = r\cos\theta$ and $y = r\sin\theta$ for convenience in Eqs. 3 and 4. Our final results are summarized to three significant digits:

Results:

$$\boxed{r = 0.447\text{ m},\ \ \theta = 0.464\text{ radians},\ \ u_r = 6.71\frac{\text{m}}{\text{s}},\ \ u_\theta = -9.62\frac{\text{m}}{\text{s}}} \qquad (5)$$

We verify our result by calculating the square of the speed in both coordinate systems. In Cartesian coordinates,

$$V^2 = u^2 + v^2 = \left(10.3\frac{\text{m}}{\text{s}}\right)^2 + \left(-5.6\frac{\text{m}}{\text{s}}\right)^2 = 137.5\frac{\text{m}^2}{\text{s}^2} \qquad (6)$$

In cylindrical coordinates,

$$V^2 = u_r^{\,2} + u_\theta^{\,2} = \left(6.708\frac{\text{m}}{\text{s}}\right)^2 + \left(-9.615\frac{\text{m}}{\text{s}}\right)^2 = 137.5\frac{\text{m}^2}{\text{s}^2} \qquad (7)$$

Discussion Such checks of our algebra are always wise.

9-16

Solution We are to transform a given set of Cartesian velocity components into cylindrical velocity components, and identify the flow.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x-y* or *r-θ* plane.

Analysis We recognize that $r^2 = x^2 + y^2$. We also know that $y = r\sin\theta$ and $x = r\cos\theta$. Using the results of Problem 9-14, the cylindrical velocity components are

u_r *component of velocity*:

$$\boxed{u_r = u\cos\theta + v\sin\theta = \frac{Cr\sin\theta\cos\theta}{r^2} - \frac{Cr\sin\theta\cos\theta}{r^2} = 0} \qquad (1)$$

u_θ *component of velocity*::

$$\boxed{u_\theta = -u\sin\theta + v\cos\theta = -\frac{Cr\sin^2\theta}{r^2} - \frac{Cr\cos^2\theta}{r^2} = \frac{-C}{r}} \qquad (2)$$

where we have also used the fact that $\cos^2\theta + \sin^2\theta = 1$. We recognize the velocity components of Eqs. 1 and 2 as those of a **line vortex**.

Discussion The negative sign in Eq. 2 indicates that this vortex is in the *clockwise* direction.

9-17

Solution We are to transform a given set of cylindrical velocity components into Cartesian velocity components.

Analysis We apply the coordinate transformations given in this chapter, along with the results of Problem 9-16,

x component of velocity:
$$u = u_r \cos\theta - u_\theta \sin\theta = \frac{m}{2\pi r}\frac{x}{r} - \frac{\Gamma}{2\pi r}\frac{y}{r} \qquad (1)$$

We recognize that $r^2 = x^2 + y^2$. Thus, Eq. 1 becomes

x component of velocity:
$$u = \frac{1}{2\pi\left(x^2+y^2\right)}\left(mx - \Gamma y\right) \qquad (2)$$

Similarly,

y component of velocity:
$$\boxed{v = u_r \sin\theta + u_\theta \cos\theta = \frac{m}{2\pi r}\frac{y}{r} + \frac{\Gamma}{2\pi r}\frac{x}{r}} \qquad (3)$$

Again recognizing that $r^2 = x^2 + y^2$, Eq. 3 becomes

y component of velocity:
$$\boxed{v = \frac{1}{2\pi\left(x^2+y^2\right)}\left(my + \Gamma x\right)} \qquad (4)$$

We verify our result by calculating the square of the speed in both coordinate systems. In Cartesian coordinates,

$$V^2 = u^2 + v^2 = \frac{1}{4\pi^2\left(x^2+y^2\right)^2}\left(m^2x^2 - 2mx\Gamma y + \Gamma^2 y^2\right) + \frac{1}{4\pi^2\left(x^2+y^2\right)^2}\left(m^2y^2 + 2my\Gamma x + \Gamma^2 x^2\right) \qquad (5)$$

Two of the terms in Eq. 5 cancel, and we combine the others. After simplification,

Magnitude of velocity squared:
$$\boxed{V^2 = u^2 + v^2 = \frac{1}{4\pi^2\left(x^2+y^2\right)}\left(m^2 + \Gamma^2\right)} \qquad (6)$$

We calculate V^2 from the components given in cylindrical coordinates as well,

Magnitude of velocity squared:
$$\boxed{V^2 = u_r^{\,2} + u_\theta^{\,2} = \frac{m^2}{4\pi^2 r^2} + \frac{\Gamma^2}{4\pi^2 r^2} = \frac{m^2+\Gamma^2}{4\pi^2 r^2}} \qquad (7)$$

Finally, since $r^2 = x^2 + y^2$, Eqs. 6 and 7 are the same, and **the results are verified**.

Discussion Such checks of our algebra are always wise.

9-18E

Solution We are to transform a given set of Cartesian coordinates and velocity components into cylindrical coordinates and velocity components.

Analysis First we apply the coordinate transformations given in this chapter,

$$x = r\cos\theta = 5.20\ \text{in} \times \cos\left(30.0^\circ\right)\left(\frac{1\ \text{ft}}{12\ \text{in}}\right) = 0.3753\ \text{ft} \qquad (1)$$

and

$$y = r\sin\theta = 5.20\ \text{in} \times \sin\left(30.0^\circ\right)\left(\frac{1\ \text{ft}}{12\ \text{in}}\right) = 0.2167\ \text{ft} \qquad (2)$$

Next we apply the results of a previous problem,

$$u = u_r \cos\theta - u_\theta \sin\theta = (2.06\ \text{ft/s}) \times \cos(30.0°) - (4.66\ \text{ft/s}) \times \sin(30.0°) = -0.5460\ \text{ft/s} \qquad (3)$$

and

$$v = u_r \sin\theta + u_\theta \cos\theta = (2.06\ \text{ft/s}) \times \sin(30.0°) + (4.66\ \text{ft/s}) \times \cos(30.0°) = 5.066\ \text{ft/s} \qquad (4)$$

Our final results are summarized to three significant digits:

Results: $x = \mathbf{0.373\ ft}$, $y = \mathbf{0.217\ ft}$, $u = \mathbf{-0.546\ ft/s}$, $v = \mathbf{5.07\ ft/s}$ (5)

We verify our result by calculating the square of the speed in both coordinate systems. In Cartesian coordinates,

$$V^2 = u^2 + v^2 = (-0.5460\ \text{ft/s})^2 + (5.066\ \text{ft/s})^2 = \mathbf{25.96\ ft^2/s^2} \qquad (6)$$

In cylindrical coordinates,

$$V^2 = u_r^2 + u_\theta^2 = (2.06\ \text{ft/s})^2 + (4.66\ \text{ft/s})^2 = \mathbf{25.96\ ft^2/s^2} \qquad (7)$$

Discussion Such checks of our algebra are always wise.

9-19

Solution We are to perform both integrals of the divergence theorem for a given vector and volume, and verify that they are equal.

Analysis We do the volume integral first:

Volume integral:
$$\int_V \vec{\nabla}\cdot\vec{G}dV = \int_{x=0}^{x=1}\int_{y=0}^{y=1}\int_{z=0}^{z=1}\left(\frac{\partial G_x}{\partial x}+\frac{\partial G_y}{\partial y}+\frac{\partial G_z}{\partial z}\right)dzdydx = \int_{x=0}^{x=1}\int_{y=0}^{y=1}\int_{z=0}^{z=1}\left(4z-2y+y\right)dzdydx \tag{1}$$

The term in parentheses in Eq. 1 reduces to $(4z - y)$, and we integrate this over z first,

$$\int_V \vec{\nabla}\cdot\vec{G}dV = \int_{x=0}^{x=1}\int_{y=0}^{y=1}\left[2z^2 - yz\right]_{z=0}^{z=1} dydx = \int_{x=0}^{x=1}\int_{y=0}^{y=1}\left(2-y\right)dydx$$

Then we integrate over y and then over x,

Volume integral:
$$\int_V \vec{\nabla}\cdot\vec{G}dV = \int_{x=0}^{x=1}\left[2y-\frac{y^2}{2}\right]_{y=0}^{y=1} dx = \int_{x=0}^{x=1}\frac{3}{2}dx = \mathbf{\frac{3}{2}} \tag{2}$$

Next we calculate the surface integral of the divergence theorem. There are six faces of the cube, and unit vector $\vec{n}$ points outward from each face. So, we split the area integral into six parts and sum them. E.g., the right-most face has $\vec{n}$ = (1,0,0), so $\vec{G}\cdot\vec{n}$ = $4xz$ on this face. The bottom face has $\vec{n}$ = (0,–1,0), so $\vec{G}\cdot\vec{n}$ = y^2 on this face. The surface integral is then

Surface integral:
$$\begin{aligned}\oiint_A \vec{G}\cdot\vec{n}dA = &\underbrace{\left[\int_{y=0}^{y=1}\int_{z=0}^{z=1}\left(4xz\right)dzdy\right]_{x=1}}_{\text{Right face}} + \underbrace{\left[\int_{y=0}^{y=1}\int_{z=0}^{z=1}\left(-4xz\right)dzdy\right]_{x=0}}_{\text{Left face}} + \underbrace{\left[\int_{z=0}^{z=1}\int_{x=0}^{x=1}\left(-y^2\right)dxdz\right]_{y=1}}_{\text{Top face}}\\ &+\underbrace{\left[\int_{z=0}^{z=1}\int_{x=0}^{x=1}\left(y^2\right)dxdz\right]_{y=0}}_{\text{Bottom face}} + \underbrace{\left[\int_{x=0}^{x=1}\int_{y=0}^{y=1}\left(yz\right)dydx\right]_{z=1}}_{\text{Front face}} + \underbrace{\left[\int_{x=0}^{x=1}\int_{y=0}^{y=1}\left(-yz\right)dydx\right]_{z=0}}_{\text{Back face}}\end{aligned} \tag{3}$$

The three integrals on the far right of Eq. 3 are obviously zero. The other three integrals can be obtained carefully,

$$\oiint_A \vec{G}\cdot\vec{n}dA = \int_{y=0}^{y=1}\left[2z^2\right]_{z=0}^{z=1}dy + \int_{z=0}^{z=1}\left[-x\right]_{x=0}^{x=1}dz + \int_{x=0}^{x=1}\left[\frac{y^2}{2}\right]_{y=0}^{y=1}dx = \int_{y=0}^{y=1}\left(2\right)dy + \int_{z=0}^{z=1}\left(-1\right)dz + \int_{x=0}^{x=1}\left(\frac{1}{2}\right)dx \tag{4}$$

The last three integrals of Eq. 4 are trivial. The final result is

Surface integral:
$$\oiint_A \vec{G}\cdot\vec{n}dA = 2-1+\frac{1}{2} = \mathbf{\frac{3}{2}} \tag{5}$$

Since Eq. 2 and Eq. 5 are equal, the divergence theorem works for this case.

Discussion The integration is simple in this example since each face is flat and normal to an axis. In the general case in which the surface is curved, integration is much more difficult, but the divergence theorem always works.

9-20

Solution We are to expand a dot product in Cartesian coordinates and verify it.

Analysis In Cartesian coordinates the del operator is $\vec{\nabla} = \frac{\partial}{\partial x}\vec{i} + \frac{\partial}{\partial y}\vec{j} + \frac{\partial}{\partial z}\vec{k}$ and we let $\vec{G} = G_x\vec{i} + G_y\vec{j} + G_z\vec{k}$. The left hand side of the equation is thus

Left hand side:

$$\begin{aligned}\vec{\nabla}\cdot\left(f\vec{G}\right) &= \frac{\partial\left(fG_x\right)}{\partial x} + \frac{\partial\left(fG_y\right)}{\partial y} + \frac{\partial\left(fG_z\right)}{\partial z} \\ &= G_x\frac{\partial f}{\partial x} + f\frac{\partial G_x}{\partial x} + G_y\frac{\partial f}{\partial y} + f\frac{\partial G_y}{\partial y} + G_z\frac{\partial f}{\partial z} + f\frac{\partial G_z}{\partial z}\end{aligned} \tag{1}$$

The right hand side of the equation is

Right hand side:

$$\begin{aligned}&\vec{G}\cdot\vec{\nabla}f + f\vec{\nabla}\cdot\vec{G} \\ &= \left(G_x\vec{i} + G_y\vec{j} + G_z\vec{k}\right)\cdot\left(\frac{\partial f}{\partial x}\vec{i} + \frac{\partial f}{\partial y}\vec{j} + \frac{\partial f}{\partial z}\vec{k}\right) + f\left(\frac{\partial G_x}{\partial x} + \frac{\partial G_y}{\partial y} + \frac{\partial G_z}{\partial z}\right) \\ &= G_x\frac{\partial f}{\partial x} + G_y\frac{\partial f}{\partial y} + G_z\frac{\partial f}{\partial z} + f\frac{\partial G_x}{\partial x} + f\frac{\partial G_y}{\partial y} + f\frac{\partial G_z}{\partial z}\end{aligned} \tag{2}$$

Equations 1 and 2 are the same, and **the given equation is verified**.

Discussion The product rule given in this problem was used in this chapter in the derivation of the alternative form of the continuity equation.

Continuity Equation

9-21C

Solution We are to explain why the derivation of the continuity via the divergence theorem is so much less involved than the derivation of the same equation by summation of mass flow rates through each face of an infinitesimal control volume.

Analysis In the derivation using the divergence theorem, we begin with the control volume form of conservation of mass, and simply apply the divergence theorem. The control volume form was already derived in Chap. 5, so **we begin the derivation in this chapter with an established conservation of mass equation**. On the other hand, the alternative derivation is from "scratch" and therefore requires much more algebra.

Discussion The bottom line is that the divergence theorem enables us to quickly convert the control volume form of the conservation law into the differential form.

9-22C

Solution We are to discuss the material derivative of density for the case of compressible and incompressible flow.

Analysis If the flow field is compressible, we expect that as a fluid particle (a *material element*) moves around in the flow, its density changes. Thus the *material derivative of density* (the rate of change of density following a fluid particle) is **non-zero for compressible flow**. However, if the flow field is incompressible, the density remains constant. As a fluid particle moves around in the flow, the material derivative of density must be **zero for incompressible flow** (no change in density following the fluid particle).

Discussion The material derivative of any property is the rate of change of that property following a fluid particle.

9-23

Solution We are to repeat Example 9-1, but without using continuity.

Assumptions **1** Density varies with time, but not space; in other words, the density is uniform throughout the cylinder at any given time, but changes with time. **2** No mass escapes from the cylinder during the compression.

Analysis The mass inside the cylinder is constant, but the volume decreases linearly as the piston moves up. At $t = 0$ when $L = L_{\text{Bottom}}$ the initial volume of the cylinder is $V(0) = L_{\text{Bottom}}A$, where A is the cross-sectional area of the cylinder. At $t = 0$ the density is $\rho = \rho(0) = m/V(0)$, and thus

Mass in the cylinder:
$$m = \rho(0)V(0) = \rho(0)L_{\text{Bottom}}A \qquad (1)$$

Mass m (Eq. 1) is a constant since no mass escapes during the compression. At some later time t, $L = L_{\text{Bottom}} - V_{\text{P}}t$ and the volume is thus

Cylinder volume at time t:
$$V = (L_{\text{Bottom}} - V_{\text{P}}t)A \qquad (2)$$

The density at time t is

Density at time t:
$$\rho = \frac{m}{V} = \frac{\rho(0)L_{\text{Bottom}}A}{(L_{\text{Bottom}} - V_{\text{P}}t)A} \qquad (3)$$

where we have plugged in Eq. 1 for m and Eq. 2 for V. Equation 3 reduces to

$$\rho = \rho(0)\frac{L_{\text{Bottom}}}{L_{\text{Bottom}} - V_{\text{P}}t} \qquad (4)$$

or, using the nondimensional variables of Example 9-1,

Nondimensional result:
$$\boxed{\frac{\rho}{\rho(0)} = \frac{1}{1 - \dfrac{V_{\text{P}}t}{L_{\text{Bottom}}}} \quad \text{or} \quad \rho^* = \frac{1}{1 - t^*}} \qquad (5)$$

which is identical to Eq. 5 of Example 9-1.

Discussion We see by this exercise that the continuity equation is indeed an equation of conservation of mass.

9-24

Solution We are to expand the continuity equation in Cartesian coordinates.

Analysis We expand the second term by taking the dot product of the del operator $\vec{\nabla} = \left(\frac{\partial}{\partial x}\vec{i} + \frac{\partial}{\partial y}\vec{j} + \frac{\partial}{\partial z}\vec{k}\right)$ with $\rho\vec{V} = (\rho u)\vec{i} + (\rho v)\vec{j} + (\rho w)\vec{k}$, giving

Compressible continuity equation in Cartesian coordinates:
$$\boxed{\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0} \qquad (1)$$

We can further expand Eq. 1 by using the product rule on the spatial derivatives, resulting in 7 terms,

Further expansion:
$$\boxed{\frac{\partial \rho}{\partial t} + \rho\frac{\partial u}{\partial x} + u\frac{\partial \rho}{\partial x} + \rho\frac{\partial v}{\partial y} + v\frac{\partial \rho}{\partial y} + \rho\frac{\partial w}{\partial z} + w\frac{\partial \rho}{\partial z} = 0} \qquad (2)$$

Discussion We can do a similar thing in cylindrical coordinates, but the algebra is somewhat more complicated.

9-25

Solution We are to write the given equation as a word equation and discuss it.

Analysis Here is a word equation: "**The time rate of change of volume of a fluid particle per unit volume is equal to the divergence of the velocity field**." As a fluid particle moves around in a compressible flow, it can distort, rotate, and get larger or smaller. Thus the volume of the fluid element can change with time; this is represented by the left hand side of the equation. The right hand side is identically zero for an incompressible flow, but it is *not* zero for a compressible flow. Thus we can think of the volumetric strain rate as a measure of compressibility of a fluid flow.

Discussion Volumetric strain rate is a kinematic property as discussed in Chap. 4. Nevertheless, it is shown here to be related to the continuity equation (conservation of mass).

9-26

Solution We are to verify that a given flow field satisfies the continuity equation, and we are to discuss conservation of mass at the origin.

Analysis The 2-D cylindrical velocity components (u_r,u_θ) for this flow field are

Cylindrical velocity components:

$$u_r = \frac{m}{2\pi r} \qquad u_\theta = \frac{\Gamma}{2\pi r} \qquad (1)$$

where m and Γ are constants We plug Eq. 1 into the incompressible continuity equation in cylindrical coordinates,

Incompressible continuity:

$$\frac{1}{r}\frac{\partial(ru_r)}{\partial r} + \frac{1}{r}\frac{\partial(u_\theta)}{\partial \theta} + \frac{\partial(u_z)}{\partial z} = 0 \quad \text{or} \quad \frac{1}{r}\underbrace{\cancel{\frac{\partial\left(\frac{m}{2\pi}\right)}{\partial r}}}_{0} + \frac{1}{r}\underbrace{\cancel{\frac{\partial\left(\frac{\Gamma}{2\pi r}\right)}{\partial \theta}}}_{0} + \underbrace{\cancel{\frac{\partial(u_z)}{\partial z}}}_{0} = 0 \qquad (2)$$

The first term is zero because it is the derivative of a constant. The second term is zero because r is not a function of θ. The third term is zero since this is a 2-D flow with $u_z = 0$. Thus, we verify that *the incompressible continuity equation is satisfied for the given velocity field*.

At the origin, both u_r and u_θ go to infinity. Conservation of mass is not affected by u_θ, but the fact that u_r is non-zero at the origin violates conservation of mass. We think of the flow along the z axis as a *line sink* toward which mass approaches from all directions in the plane and then disappears (like a black hole in two dimensions). *Mass is not conserved at the origin*.

Discussion Singularities such as this are unphysical of course, but are nevertheless useful as approximations of real flows, as long as we stay away from the singularity itself.

9-27

Solution We are to verify that a given velocity field satisfies continuity.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The velocity field of Problem 9-16 is

Cartesian velocity components:
$$u = \frac{Cy}{x^2 + y^2} \qquad v = \frac{-Cx}{x^2 + y^2} \qquad (1)$$

We check continuity, staying in Cartesian coordinates,

$$\underbrace{\frac{\partial u}{\partial x}}_{-2xCy\left(x^2+y^2\right)^{-3}} + \underbrace{\frac{\partial v}{\partial y}}_{2yCx\left(x^2+y^2\right)^{-3}} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{0 since 2-D}} = 0$$

So we see that **the incompressible continuity equation is indeed satisfied**.

Discussion The fact that the flow field satisfies continuity does not guarantee that a corresponding pressure field exists that can satisfy the steady conservation of momentum equation. In this case, however, it does.

9-28

Solution We are to verify that a given velocity field is incompressible.

Assumptions **1** The flow is two-dimensional, implying no *z component* of velocity and no variation of *u* or *v* with *z*.

Analysis The components of velocity in the *x* and *y* directions respectively are

$$u = 1.6 + 1.8x \qquad v = 1.5 - 1.8y$$

To check if the flow is incompressible, we see if the incompressible continuity equation is satisfied:

$$\underbrace{\frac{\partial u}{\partial x}}_{2.8} + \underbrace{\frac{\partial v}{\partial y}}_{-2.8} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{0 since 2-D}} = 0 \qquad \text{or} \qquad 1.8 - 1.8 = 0$$

So we see that **the incompressible continuity equation is indeed satisfied**. Hence the flow field is incompressible.

Discussion The fact that the flow field satisfies continuity does not guarantee that a corresponding pressure field exists that can satisfy the steady conservation of momentum equation.

9-29

Solution For a given axial velocity component in an axisymmetric flow field, we are to generate the radial velocity component.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric implying that $u_\theta = 0$ and there is no variation in the θ direction.

Analysis We use the incompressible continuity equation in cylindrical coordinates, simplified as follows for axisymmetric flow,

Incompressible axisymmetric continuity equation:
$$\frac{1}{r}\frac{\partial(ru_r)}{\partial r}+\frac{\partial(u_z)}{\partial z}=0 \qquad (1)$$

We rearrange Eq. 1,

$$\frac{\partial(ru_r)}{\partial r}=-r\frac{\partial(u_z)}{\partial z}=-r\frac{u_{z,\text{exit}}-u_{z,\text{entrance}}}{L} \qquad (2)$$

We integrate Eq. 2 with respect to *r*,

$$ru_r=-\frac{r^2}{2}\frac{u_{z,\text{exit}}-u_{z,\text{entrance}}}{L}+f(z) \qquad (3)$$

Notice that since we performed a *partial* integration with respect to *r*, we add a function of the other variable *z* rather than simply a constant of integration. We divide all terms in Eq. 3 by *r* and recognize that the term with *f*(*z*) will go to infinity at the centerline of the nozzle ($r = 0$) unless $f(z) = 0$. We write our final expression for u_r,

Radial velocity component:
$$\boxed{u_r=-\frac{r}{2}\frac{u_{z,\text{exit}}-u_{z,\text{entrance}}}{L}} \qquad (4)$$

Discussion You should plug the given equation and Eq. 4 into Eq. 1 to verify that the result is correct. (It is.)

9-30

Solution We are to determine a relationship between constants *a*, *b*, *c*, and *d* that ensures incompressibility.

Assumptions **1** The flow is steady. **2** The flow is incompressible (under certain restraints to be determined).

Analysis We plug the given velocity components into the incompressible continuity equation,

Condition for incompressibility:
$$\underbrace{\frac{\partial u}{\partial x}}_{ay^2}+\underbrace{\frac{\partial v}{\partial y}}_{-6cy^2}+\underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0}=0 \qquad ay^2-6cy^2=0$$

Thus to guarantee incompressibility, constants *a* and *c* must satisfy the following relationship:

Condition for incompressibility:
$$\boxed{a=6c} \qquad (1)$$

Discussion If Eq. 1 were not satisfied, the given velocity field might still represent a valid flow field, but density would have to vary with location in the flow field – in other words the flow would be *compressible*.

9-31

Solution We are to determine a relationship between constants *a*, *b*, *c*, and *d* that ensures incompressibility.

Assumptions **1** The flow is steady. **2** The flow is incompressible (under certain restraints to be determined).

Analysis We plug the given velocity components into the incompressible continuity equation,

Condition for incompressibility:
$$\underbrace{\frac{\partial u}{\partial x}}_{2axy}+\underbrace{\frac{\partial v}{\partial y}}_{2cxy}+\underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0}=0 \qquad 2axy+2cxy=0$$

Thus to guarantee incompressibility, constants *a* and *c* must satisfy the following relationship:

Condition for incompressibility:
$$\boxed{a=-c} \qquad (1)$$

Discussion If Eq. 1 were not satisfied, the given velocity field might still represent a valid flow field, but density would have to vary with location in the flow field – in other words the flow would be *compressible*.

9-32

Solution We are to find the *y component* of velocity, *v*, using a given expression for *u*.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane, implying that $w = 0$ and neither *u* nor *v* depend on *z*.

Analysis Since the flow is steady and incompressible, we apply the incompressible continuity in Cartesian coordinates to the flow field, giving

Condition for incompressibility:
$$\frac{\partial v}{\partial y}=-\underbrace{\frac{\partial u}{\partial x}}_{a}-\underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0} \qquad \frac{\partial v}{\partial y}=-a$$

Next we integrate with respect to *y*. Note that since the integration is a *partial* integration, we must add some arbitrary function of *x* instead of simply a constant of integration.

Solution:
$$\boxed{v=-ay+f\left(x\right)}$$

If the flow were three-dimensional, we would add a function of *x* and *z* instead.

Discussion To satisfy the incompressible continuity equation, any function of *x* will work since there are no derivatives of *v* with respect to *x* in the continuity equation. Not all functions of *x* are necessarily physically possible, however, since the flow must also satisfy the steady conservation of momentum equation.

9-33

Solution We are to find the most general form of the tangential velocity component of a purely circular flow that does not violate conservation of mass.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* or *r*-θ plane.

Analysis We use cylindrical coordinates for convenience. We solve for u_θ using the incompressible continuity equation,

$$\underbrace{\frac{1}{\cancel{r}}\frac{\partial\left(\cancel{ru_r}\right)}{\partial r}}_{\text{0 for circular flow}}+\frac{1}{r}\frac{\partial\left(u_\theta\right)}{\partial\theta}+\underbrace{\frac{\partial\left(\cancel{u_z}\right)}{\cancel{\partial z}}}_{\text{0 for 2-D flow}}=0 \quad \text{or} \quad \frac{\partial\left(u_\theta\right)}{\partial\theta}=0 \qquad (1)$$

We integrate Eq. 1 with respect to θ, adding a function of the other variable *r* rather than simply a constant of integration since this is a partial integration,

Result:
$$\boxed{u_\theta = f\left(r\right)} \qquad (2)$$

Discussion Any function of *r* in Eq. 2 will satisfy the continuity equation.

9-34

Solution We are to find the *y component* of velocity, *v*, using a given expression for *u*.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane, implying that *w* = 0 and neither *u* nor *v* depend on *z*.

Analysis We plug the velocity components into the steady incompressible continuity equation,

Condition for incompressibility:
$$\frac{\partial v}{\partial y}=-\underbrace{\frac{\partial u}{\partial x}}_{a}-\underbrace{\frac{\partial \cancel{w}}{\cancel{\partial z}}}_{0} \qquad \frac{\partial v}{\partial y}=-a$$

Next we integrate with respect to *y*. Note that since the integration is a *partial* integration, we must add some arbitrary function of *x* instead of simply a constant of integration.

Solution:
$$\boxed{v=-ay+f\left(x\right)}$$

If the flow were three-dimensional, we would add a function of *x* and *z* instead.

Discussion To satisfy the incompressible continuity equation, any function of *x* will work since there are no derivatives of *v* with respect to *x* in the continuity equation. Not all functions of *x* are necessarily physically possible, however, since the flow may not be able to satisfy the steady conservation of momentum equation.

9-35

Solution We are to find the *y component* of velocity, v, using a given expression for u.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the x-y plane, implying that $w = 0$ and neither u nor v depend on z.

Analysis We plug the velocity components into the steady incompressible continuity equation,

Condition for incompressibility:
$$\frac{\partial v}{\partial y} = -\underbrace{\frac{\partial u}{\partial x}}_{6ax-2by} - \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0} \qquad \frac{\partial v}{\partial y} = -6ax + 2by$$

Next we integrate with respect to y. Note that since the integration is a *partial* integration, we must add some arbitrary function of x instead of simply a constant of integration.

Solution:
$$\boxed{v = -6axy + by^2 + f(x)}$$

If the flow were three-dimensional, we would add a function of x and z instead.

Discussion To satisfy the incompressible continuity equation, any function of x will work since there are no derivatives of v with respect to x in the continuity equation.

9-36

Solution We are to find the most general form of the radial velocity component of a purely radial flow that does not violate conservation of mass.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the x-y or r-θ plane.

Analysis We use cylindrical coordinates for convenience. We solve for u_r using the incompressible continuity equation,

$$\frac{1}{r}\frac{\partial(ru_r)}{\partial r} + \underbrace{\cancel{\frac{1}{r}\frac{\partial(u_\theta)}{\partial\theta}}}_{0 \text{ for radial flow}} + \underbrace{\cancel{\frac{\partial(u_z)}{\partial z}}}_{0 \text{ for 2-D flow}} = 0 \quad \text{or} \quad \frac{\partial(ru_r)}{\partial r} = 0 \qquad (1)$$

We integrate Eq. 1 with respect to r, adding a function of the other variable θ rather than simply a constant of integration since this is a partial integration,

Result:
$$\boxed{ru_r = f(\theta) \quad \text{or} \quad u_r = \frac{f(\theta)}{r}} \qquad (2)$$

Discussion Any function of θ in Eq. 2 will satisfy the continuity equation.

9-37

Solution We are to find the *z component* of velocity using given expressions for *u* and *v*.

Assumptions **1** The flow is steady. **2** The flow is incompressible.

Analysis We apply the steady incompressible continuity equation to the given flow field,

Condition for incompressibility: $$\frac{\partial w}{\partial z} = -\underbrace{\frac{\partial u}{\partial x}}_{2a+by} - \underbrace{\frac{\partial v}{\partial y}}_{-bz^2} \qquad \frac{\partial w}{\partial z} = -2a - by + bz^2$$

Next we integrate with respect to *z*. Note that since the integration is a *partial* integration, we must add some arbitrary function of *x* and *y* instead of simply a constant of integration.

Solution: $$\boxed{w = -2az - byz + \frac{bz^3}{3} + f\left(x, y\right)}$$

Discussion To satisfy the incompressible continuity equation, any function of *x* and *y* will work since there are no derivatives of *w* with respect to *x* or *y* in the continuity equation.

9-38

Solution For given velocity component u and density ρ, we are to predict velocity component v, plot an approximate shape of the duct, and predict its height at section (2).

Assumptions **1** The flow is steady and two-dimensional in the x-y plane, but *compressible*. **2** Friction on the walls is ignored. **3** Axial velocity u and density ρ vary linearly with x. **4** The x axis is a line of top-bottom symmetry.

Properties The fluid is standard air. The speed of sound is about 340 m/s, so the flow is subsonic, but compressible.

Analysis (*a*) We write expressions for u and ρ, forcing them to be linear in x,

$$u = u_1 + C_u x \qquad C_u = \frac{u_2 - u_1}{\Delta x} = \frac{(100-300)\,\frac{\text{m}}{\text{s}}}{2.0\text{ m}} = -100\,\frac{1}{\text{s}} \tag{1}$$

$$\rho = \rho_1 + C_\rho x \qquad C_\rho = \frac{\rho_2 - \rho_1}{\Delta x} = \frac{(1.2-0.85)\,\frac{\text{kg}}{\text{m}^3}}{2.0\text{ m}} = 0.175\,\frac{\text{kg}}{\text{m}^4} \tag{2}$$

where C_u and C_ρ are constants. We use the compressible form of the steady continuity equation, placing the unknown term v on the left hand side, and plugging in Eqs. 1 and 2,

$$\frac{\partial(\rho v)}{\partial y} = -\frac{\partial(\rho u)}{\partial x} = -\frac{\partial\left((\rho_1 + C_\rho x)(u_1 + C_u x)\right)}{\partial x}$$

After some algebra,

$$\frac{\partial(\rho v)}{\partial y} = -(\rho_1 C_u + u_1 C_\rho) - 2C_u C_\rho x \tag{3}$$

We integrate Eq. 3 with respect to y,

$$\rho v = -(\rho_1 C_u + u_1 C_\rho)y - 2C_u C_\rho xy + f(x) \tag{4}$$

Since this is a *partial* integration, we add an arbitrary function of x instead of simply a constant of integration. We now apply boundary conditions. Since the flow is symmetric about the x axis (y = 0), v must equal zero at y = 0 for any x. This is possible only if $f(x)$ is identically zero. Applying $f(x)$ = 0, dividing by ρ to solve for v, and plugging in Eq. 2, Eq. 4 becomes

$$\boxed{v = \frac{-(\rho_1 C_u + u_1 C_\rho)y - 2C_u C_\rho xy}{\rho} = \frac{-(\rho_1 C_u + u_1 C_\rho)y - 2C_u C_\rho xy}{\rho_1 + C_\rho x}} \tag{5}$$

(*b*) For known values of u and v, we can plot streamlines between x = 0 and x = 2.0 m using the technique described in Chap. 4. Several streamlines are shown in Fig. 1. The streamline starting at x = 0, y = 0.8 m is the top wall of the duct.

(*c*) At section (2), the top streamline crosses y = 1.70 m at x = 2.0 m. Thus, *the predicted height of the duct at section* (2) *is* 1.70 m.

Discussion You can verify that the combination of Eqs. 1, 2, and 5 satisfies the steady compressible continuity equation. However, this alone does not guarantee that the density and velocity components will actually *follow* these equations if this diverging duct were to be built. The actual flow depends on the *pressure rise* between sections (1) and (2) – only one unique pressure rise can yield the desired flow deceleration. Temperature may also change considerably in this kind of compressible flow field.

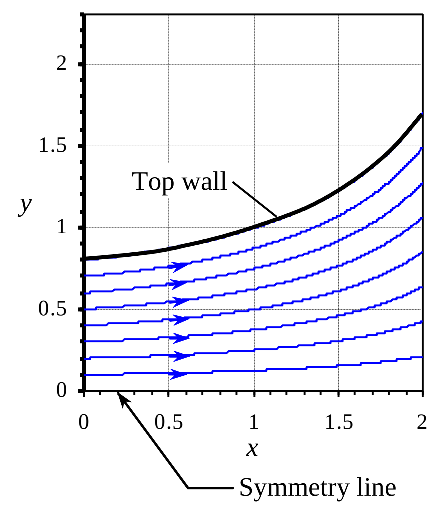


FIGURE 1
Streamlines for a diverging duct.

Stream Function

9-39C

Solution We are to discuss the significance of the difference in value of stream function from one streamline to another.

Analysis **The difference in the value of ψ from one streamline to another is equal to the volume flow rate per unit width between the two streamlines**.

Discussion This fact about the stream function can be used to calculate the volume flow rate in certain applications.

9-40C

Solution We are to discuss why the stream function is called a non-primitive variable in CFD lingo.

Analysis The natural physical variables in a fluid flow problem are the velocity components and the pressure. [If the flow is compressible, density and temperature are also natural physical variables.] These variables can be considered "primitive" because we do not change them in any way – we simply solve for them directly. Stream function, on the other hand, is a contrived or derived variable. **The stream function is not primitive in the sense that it is not one of the original physical variables in the problem**.

Discussion Vorticity is another example of a non-primitive variable. In fact, some 2-D CFD codes use stream function and vorticity as the variables – non-primitive variables.

9-41C

Solution We are to discuss the restrictions on the stream function that cause it to exactly satisfy 2-D incompressible continuity, and why they are necessary.

Analysis **Stream function ψ must be a *smooth function* of x and y (or r and θ)**. These restrictions are necessary so that the second derivatives of ψ with respect to both variables are equal regardless of the order of differentiation. In other words, if $\frac{\partial^2\psi}{\partial x\partial y} = \frac{\partial^2\psi}{\partial y\partial x}$, then the 2-D incompressible continuity equation is satisfied exactly by the definition of ψ.

Discussion If the stream function were not smooth, there would be sudden discontinuities in the velocity field as well – a physical impossibility that would violate conservation of mass.

9-42C

Solution We are to discuss the significance of curves of constant stream function, and why the stream function is useful.

Analysis Curves of constant stream function represent streamlines of a flow. A stream function is useful because by drawing curves of constant ψ, we can visualize the instantaneous velocity field. In addition, the change in the value of ψ from one streamline to another is equal to the volume flow rate per unit width between the two streamlines.

Discussion Streamlines are an instantaneous flow description, as discussed in Chap. 4.

9-43

Solution For a given velocity field we are to generate an expression for ψ, and we are to calculate the volume flow rate per unit width between two streamlines.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis We start by picking one of the two definitions of the stream function (it doesn't matter which part we choose – the solution will be identical).

$$\frac{\partial \psi}{\partial y} = u = V \qquad (1)$$

Next we integrate Eq. 1 with respect to *y*, noting that this is a *partial* integration and we must add an arbitrary function of the other variable, *x*, rather than a simple constant of integration.

$$\psi = Vy + g(x) \qquad (2)$$

Now we choose the other part of the definition of ψ, differentiate Eq. 2, and rearrange as follows:

$$v = -\frac{\partial \psi}{\partial x} = -g'(x) \qquad (3)$$

where $g'(x)$ denotes dg/dx since g is a function of only one variable, x. We now have two expressions for velocity component v, the given equation and Eq. 3. We equate these and integrate with respect to x to find $g(x)$,

$$v = 0 = -g'(x) \qquad g'(x) = 0 \qquad g(x) = C \qquad (4)$$

Note that here we have added an arbitrary constant of integration C since g is a function of x only. Finally, plugging Eq. 4 into Eq. 2 yields the final expression for ψ,

Stream function:
$$\boxed{\psi = Vy + C} \qquad (5)$$

Constant C is arbitrary; it is common to set it to zero, although it can be set to any desired value. Here, $\psi = 0$ along the streamline at $y = 0$, forcing C to equal zero by Eq. 5. For the streamline at $y = 0.5$ m,

Value of ψ_2:
$$\psi_2 = \left(6.94\ \frac{\text{m}}{\text{s}}\right) \times (0.5\ \text{m}) = 3.47\ \frac{\text{m}^2}{\text{s}} \qquad (6)$$

The volume flow rate per unit width between streamlines ψ_2 and ψ_0 is equal to $\psi_2 - \psi_0$,

Volume flow rate per unit width:
$$\frac{\dot{V}}{W} = \psi_2 - \psi_0 = (3.47 - 0)\frac{\text{m}^2}{\text{s}} = \mathbf{3.47}\ \frac{\mathbf{m^2}}{\mathbf{s}} \qquad (7)$$

We verify our result by calculating the volume flow rate per unit width from first principles. Namely, volume flow rate is equal to speed times cross-sectional area,

Volume flow rate per unit width:
$$\frac{\dot{V}}{W} = V(y_2 - y_0) = 6.94\frac{\text{m}}{\text{s}} \times (0.5 - 0)\ \text{m} = \mathbf{3.47}\ \frac{\mathbf{m^2}}{\mathbf{s}} \qquad (8)$$

Discussion If constant C were some value besides zero, we would still get the same result for the volume flow rate since C would cancel out in the subtraction.

9-44

Solution For a given velocity field we are to show that the velocity field satisfies the continuity equation, and we are to determine the stream function corresponding to this velocity field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional.

Analysis For a two-dimensional flow, the continuity equation in cylindrical coordinates is, from Eq. 9–18,

$$\frac{\partial(ru_r)}{\partial r}+\frac{\partial(u_\theta)}{\partial\theta}=0$$

or

$$\frac{\partial}{\partial r}\left(rU_\infty\left(1-\frac{R^2}{r^2}\right)\cos\theta\right)+\frac{\partial}{\partial\theta}\left(-rU_\infty\left(1+\frac{R^2}{r^2}\right)\sin\theta\right)$$
$$=\underbrace{\left[U_\infty\left(1-\frac{R^2}{r^2}\right)\cos\theta+\frac{2U_\infty R^2}{r^2}\cos\theta\right]}_{U_\infty\left(1+\frac{R^2}{r^2}\right)\cos\theta}-U_\infty\left(1+\frac{R^2}{r^2}\right)\cos\theta=0$$

Therefore the velocity field satisfies the continuity equation. The stream function can be determined from Eq. 9–27 as follows:

$$u_r=\frac{1}{r}\frac{\partial\psi}{\partial\theta}\rightarrow\psi(r,\theta)=\int rU_\infty\left(1-\frac{R^2}{r^2}\right)\cos\theta\,d\theta=rU_\infty\left(1-\frac{R^2}{r^2}\right)\sin\theta+f(r)$$
$$u_\theta=-\frac{\partial\psi}{\partial r}\rightarrow\psi(r,\theta)=-\int\left[-U_\infty\left(1+\frac{R^2}{r^2}\right)\sin\theta\right]dr=rU_\infty\left(1-\frac{R^2}{r^2}\right)\sin\theta+g(\theta)$$

Therefore we see that the stream function is

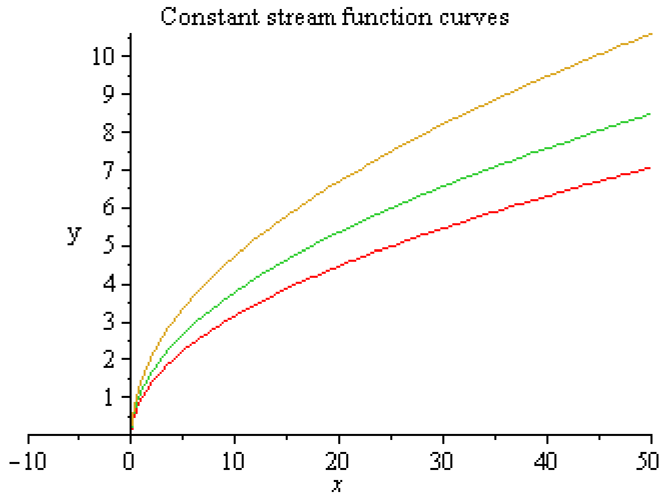
$$\psi(r,\theta)=rU_\infty\left(1-\frac{R^2}{r^2}\right)\sin\theta$$

9-45

Solution For a given stream function we are to sketch stremalines, derive expressions for the velocity components, and determine the pathlines at t = 0.

Assumptions **1** The flow is unsteady. **2** The flow is incompressible. **3** The flow is two-dimensional.

Analysis The streamlines are shown below for different values of stream function.



The velocity component can be found from Eq. 9–20 as follows:

$$u = \frac{\partial \psi}{\partial y} = -\frac{8x}{y^3}t$$

$$v = -\frac{\partial \psi}{\partial x} = -\frac{4}{y^2}t$$

The pathlines are determined from the relations

$$u = \frac{dx}{dt} \text{and} \qquad v = \frac{dy}{dt}$$

from which we obtain

$$-\frac{8x}{y^3}t = \frac{dx}{dt} \rightarrow \int_{x_0}^{x}\frac{dx}{x} = -\frac{8}{y^3}\int_0^t t dt \rightarrow \ln\frac{x}{x_0} = -\frac{4}{y^3}t^2 \rightarrow \boldsymbol{x = x_0 \exp\left(-\frac{4}{y^3}t^2\right)}$$

$$-\frac{4}{y^2}t = \frac{dy}{dt} \rightarrow \int_{y_0}^{y} y^2 dy = -4\int_0^t t dt \rightarrow \frac{1}{3}(y^3 - y_0^3) = -2t^2 \rightarrow \boldsymbol{y = \sqrt[3]{y_0^3 - 6t^2}}$$

9-46

Solution We are to generate an expression for the stream function along a vertical line in a given flow field, and we are to determine ψ at the top wall.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x-y* plane. **4** The flow is fully developed.

Analysis We start by picking one of the two definitions of the stream function (it doesn't matter which part we choose – the solution will be identical).

$$\frac{\partial \psi}{\partial y} = u = \frac{V}{h} y \qquad (1)$$

Next we integrate Eq. 1 with respect to *y*, noting that this is a *partial* integration and we must add an arbitrary function of the other variable, *x*, rather than a simple constant of integration.

$$\psi = \frac{V}{2h} y^2 + g(x) \qquad (2)$$

Now we choose the other part of the definition of ψ, differentiate Eq. 2, and rearrange as follows:

$$v = -\frac{\partial \psi}{\partial x} = -g'(x) \qquad (3)$$

where $g'(x)$ denotes dg/dx since g is a function of only one variable, x. We now have two expressions for velocity component v, the given equation and Eq. 3. We equate these and integrate with respect to x, we find $g(x)$,

$$v = 0 = -g'(x) \qquad g'(x) = 0 \qquad g(x) = C \qquad (4)$$

Note that here we have added an arbitrary constant of integration C since g is a function of x only. Finally, plugging Eq. 4 into Eq. 2 yields the final expression for ψ,

Stream function:
$$\psi = \frac{V}{2h} y^2 + C \qquad (5)$$

We find constant C by employing the boundary condition on ψ. Here, $\psi = 0$ along $y = 0$ (the bottom wall). Thus C is equal to zero by Eq. 5, and

Stream function:
$$\boxed{\psi = \frac{V}{2h} y^2} \qquad (6)$$

Along the top wall, $y = h$, and thus

Stream function along top wall:
$$\boxed{\psi_{\text{top}} = \frac{V}{2h} h^2 = \frac{Vh}{2}} \qquad (7)$$

Discussion The stream function of Eq. 6 is valid not only along the vertical dashed line of the figure provided in the problem statement, but *everywhere* in the flow since the flow is fully developed and there is nothing special about any particular x location.

9-47

Solution We are to generate an expression for the volume flow rate per unit width for Couette flow. We are to compare results from two methods of calculation.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x-y* plane. **4** The flow is fully developed.

Analysis We integrate the *x* component of velocity times cross-sectional area to obtain volume flow rate,

$$\dot{V} = \int_A u dA = \int_{y=0}^{y=h} \frac{V}{h} yW dy = \left[\frac{Vy^2}{2h}W\right]_{y=0}^{y=h} = \frac{Vh}{2}W \qquad (1)$$

where *W* is the width of the channel into the page. On a per unit width basis, we divide Eq. 1 by *W* to get

Volume flow rate per unit width:
$$\boxed{\frac{\dot{V}}{W} = \frac{Vh}{2}} \qquad (2)$$

The volume flow rate per unit width between any two streamlines ψ_2 and ψ_1 is equal to $\psi_2 - \psi_1$. We take the streamlines representing the top wall and the bottom wall of the channel. Using the result from the previous problem,

Volume flow rate per unit width:
$$\boxed{\frac{\dot{V}}{W} = \psi_{\text{top}} - \psi_{\text{bottom}} = \frac{Vh}{2} - 0 = \frac{Vh}{2}} \qquad (3)$$

Equations 2 and 3 agree, as they must.

Discussion The integration of Eq. 1 can be performed at any *x* location in the channel since the flow is fully developed.

9-48E

Solution We are to plot several streamlines using evenly spaced values of ψ and discuss the spacing between the streamlines.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** The flow is fully developed.

Analysis The stream function is obtained from the result of Problem 9-40,

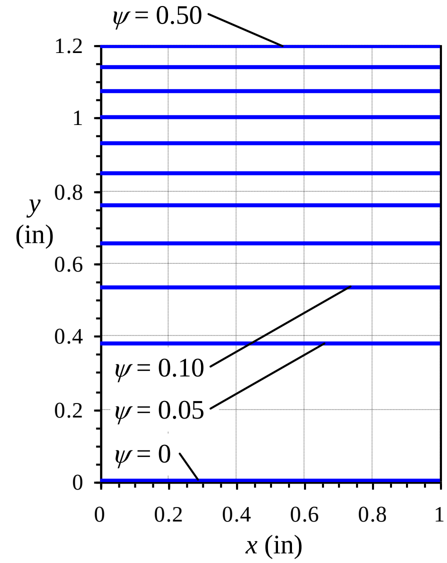
Stream function: $$\psi = \frac{V}{2h}y^2 \quad (1)$$

We solve Eq. 1 for y as a function of ψ so that we can plot streamlines,

Equation for streamlines: $$y = \sqrt{\frac{2h\psi}{V}} \quad (2)$$

We have taken only the positive root in Eq. 2 for obvious reasons. Along the top wall, $y = h$, and thus

$$\psi_{\text{top}} = \frac{Vh}{2} = \frac{10.0\,\frac{\text{ft}}{\text{s}} \times 0.100\text{ ft}}{2} = 0.500\,\frac{\text{ft}^2}{\text{s}} \quad (3)$$



The streamlines themselves are straight, flat horizontal lines as seen by Eq. 1. We divide ψ_{top} by 10 to generate evenly spaced stream functions. We plot 11 streamlines in the figure (counting the streamlines on both walls) by plugging these values of ψ into Eq. 2. **The streamlines are not evenly spaced**. This is because the volume flow rate per unit width between two streamlines ψ_2 and ψ_1 is equal to $\psi_2 - \psi_1$. The flow speeds near the top of the channel are higher than those near the bottom of the channel, so we expect the streamlines to be closer near the top.

Discussion The extent of the *x* axis in the figure is arbitrary since the flow is fully developed. You can immediately see from a streamline plot like Fig. 1 where flow speeds are high and low (relatively speaking).

9-49

Solution We are to generate an expression for the stream function along a vertical line in a given flow field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** The flow is fully developed.

Analysis We start by picking one of the two definitions of the stream function (it doesn't matter which part we choose – the solution will be identical).

$$\frac{\partial \psi}{\partial y} = u = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right) \qquad (1)$$

Next we integrate Eq. 1 with respect to *y*, noting that this is a *partial* integration and we must add an arbitrary function of the other variable, *x*, rather than a simple constant of integration.

$$\psi = \frac{1}{2\mu}\frac{dP}{dx}\left(\frac{y^3}{3} - h\frac{y^2}{2}\right) + g\left(x\right) \qquad (2)$$

Now we choose the other part of the definition of ψ, differentiate Eq. 2, and rearrange as follows:

$$v = -\frac{\partial \psi}{\partial x} = -g'\left(x\right) \qquad (3)$$

where $g'(x)$ denotes dg/dx since g is a function of only one variable, x. We now have two expressions for velocity component v, the given equation and Eq. 3. We equate these and integrate with respect to x to find $g(x)$,

$$v = 0 = -g'\left(x\right) \qquad g'\left(x\right) = 0 \qquad g\left(x\right) = C \qquad (4)$$

Note that here we have added an arbitrary constant of integration C since g is a function of x only. Finally, plugging Eq. 4 into Eq. 2 yields the final expression for ψ,

Stream function:

$$\psi = \frac{1}{2\mu}\frac{dP}{dx}\left(\frac{y^3}{3} - h\frac{y^2}{2}\right) + C \qquad (5)$$

We find constant C by employing the boundary condition on ψ. Here, $\psi = 0$ along $y = 0$ (the bottom wall). Thus C is equal to zero by Eq. 5, and

Stream function:

$$\boxed{\psi = \frac{1}{2\mu}\frac{dP}{dx}\left(\frac{y^3}{3} - h\frac{y^2}{2}\right)} \qquad (6)$$

Along the top wall, $y = h$, and thus

Stream function along top wall:

$$\boxed{\psi_{\text{top}} = -\frac{1}{12\mu}\frac{dP}{dx}h^3} \qquad (7)$$

Discussion The stream function of Eq. 6 is valid not only along the vertical dashed line of the figure provided in the problem statement, but *everywhere* in the flow since the flow is fully developed and there is nothing special about any particular *x* location.

9-50

Solution We are to generate an expression for the volume flow rate per unit width for fully developed channel flow. We are to compare results from two methods of calculation.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** The flow is fully developed.

Analysis We integrate the *x* component of velocity times cross-sectional area to obtain volume flow rate,

$$\dot{V} = \int_A u dA = \int_{y=0}^{y=h} \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)W dy = \left[\frac{1}{2\mu}\frac{dP}{dx}\left(\frac{y^3}{3} - h\frac{y^2}{2}\right)W\right]_{y=0}^{y=h}$$
$$= \frac{1}{2\mu}\frac{dP}{dx}\left(-\frac{h^3}{6}\right)W = -\frac{1}{12\mu}\frac{dP}{dx}h^3 W \qquad (1)$$

where *W* is the width of the channel into the page. On a per unit width basis, we divide Eq. 1 by *W* to get

Volume flow rate per unit width: $$\boxed{\frac{\dot{V}}{W} = -\frac{1}{12\mu}\frac{dP}{dx}h^3} \qquad (2)$$

The volume flow rate per unit width between any two streamlines ψ_2 and ψ_1 is equal to $\psi_2 - \psi_1$. We take the streamlines representing the top wall and the bottom wall of the channel. Using the result from the previous problem,

Volume flow rate per unit width:
$$\boxed{\frac{\dot{V}}{W} = \psi_{\text{top}} - \psi_{\text{bottom}} = -\frac{1}{12\mu}\frac{dP}{dx}h^3 - 0 = -\frac{1}{12\mu}\frac{dP}{dx}h^3} \qquad (3)$$

Equations 2 and 3 agree, as they must.

Discussion The integration of Eq. 1 can be performed at any *x* location in the channel since the flow is fully developed.

9-51

Solution We are to plot several streamlines using evenly spaced values of ψ and discuss the spacing between the streamlines.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** The flow is fully developed.

Properties The viscosity of water at T = 20°C is 1.002 × 10^{-3} kg/(m·s).

Analysis The stream function is obtained from the result of Problem 9-43,

Stream function:
$$\psi = \frac{1}{2\mu}\frac{dP}{dx}\left(\frac{y^3}{3} - h\frac{y^2}{2}\right) \qquad (1)$$

We need to solve Eq. 1 (a cubic equation) for y as a function of ψ so that we can plot streamlines. First we re-write Eq. 1 in standard cubic form,

Standard cubic form:
$$y^3 - \frac{3h}{2}y^2 - \frac{6\mu\psi}{dP/dx} = 0 \qquad (2)$$

We can either look up the solution for cubic equations or use Newton's iteration method to obtain y for a given value of ψ. In general there are three roots – we choose the positive real root with $0 < y < h$, which is the only one that has physical meaning for this problem. Along the top wall, $y = h$, and Eq. 1 yields

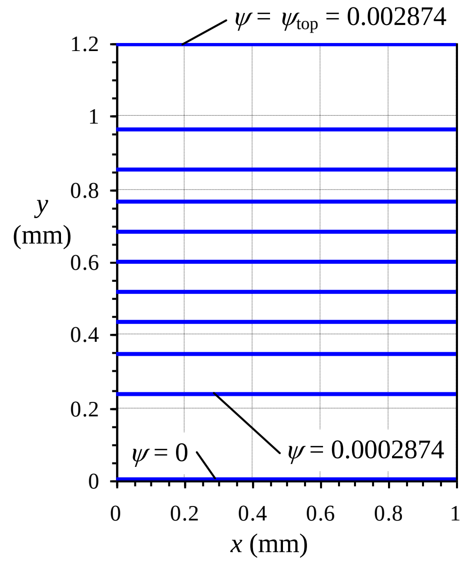


FIGURE 1
Streamlines for 2-D channel flow with evenly spaced values of stream function. Values of ψ are in units of m²/s.

Stream function along top wall:
$$\psi_{\text{top}} = -\frac{1}{12\mu}\frac{dP}{dx}h^3 = \frac{1}{12\left(1.002\times10^{-3}\ \text{kg/m}\cdot\text{s}\right)}\left(20,000\ \text{N/m}^3\right)\left(0.00120\ \text{m}\right)^3\left(\frac{\text{kg}\cdot\text{m}}{\text{s}^2\text{N}}\right)$$
$$= 2.874\times10^{-3}\ \text{m}^2/\text{s}$$

The streamlines themselves are straight, flat horizontal lines as seen by Eq. 1. We divide ψ_{top} by 10 to generate evenly spaced stream functions. We plot 11 streamlines in Fig. 1 (counting the streamlines on both walls) by plugging these values of ψ into Eq. 2 and solving for y.

The streamlines are not evenly spaced. This is because the volume flow rate per unit width between two streamlines ψ_2 and ψ_1 is equal to $\psi_2 - \psi_1$. The flow speeds in the middle of the channel are higher than those near the top or bottom of the channel, so we expect the streamlines to be closer near the middle.

Discussion The extent of the x axis in Fig. 1 is arbitrary since the flow is fully developed. You can immediately see from a streamline plot like Fig. 1 where flow speeds are high and low (relatively speaking).

9-52

Solution We are to calculate the volume flow rate and average speed of air being sucked through a sampling probe.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional.

Analysis For 2-D incompressible flow the difference in the value of the stream function between two streamlines is equal to the volume flow rate per unit width between the two streamlines. Thus,

Volume flow rate through the sampling probe:

$$\dot{V} = \psi_u - \psi_l \times W = (0.150 - 0.093)\ \mathrm{m^2/s} \times 0.0395\ \mathrm{m} = 0.0022515\ \mathrm{m^3/s} \cong \mathbf{0.00225\ m^3/s} \qquad (1)$$

The average speed of air in the probe is obtained by dividing volume flow rate by cross-sectional area,

Average speed through the sampling probe:

$$V_{\text{avg}} = \frac{\dot{V}}{hW} = \frac{0.022515\ \mathrm{m^3/s}}{(0.00458\ \mathrm{m})(0.0395\ \mathrm{m})} = 12.4454\ \mathrm{m/s} \cong \mathbf{12.4\ m/s} \qquad (2)$$

Discussion Notice that the streamlines inside the probe are more closely packed than are those outside the probe because the flow speed is higher inside the probe.

9-53

Solution We are to sketch streamlines for the case of a sampling probe with too little suction, and we are to name this type of sampling and label the lower and upper dividing streamlines.

Analysis If the suction were too weak, the volume flow rate through the probe would be too low and the average air speed through the probe would be lower than that of the air stream. The dividing streamlines would diverge outward rather than inward as sketched in Fig. 1. We would call this type of sampling **subisokinetic sampling**.

Discussion We have drawn the streamlines inside the probe further apart than those in the air stream because the flow speed is lower inside the probe.

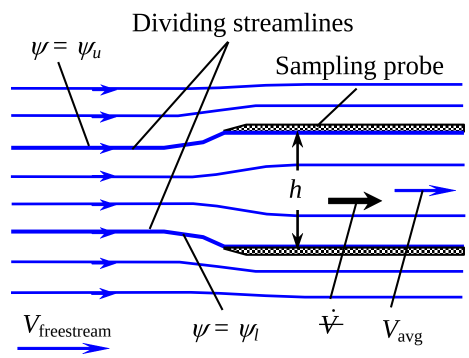


FIGURE 1
Streamlines for *subisokinetic sampling*.

9-54

Solution We are to calculate the speed of the air stream of a previous problem.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional.

Analysis In the air stream far upstream of the probe,

Volume flow rate per unit width:
$$\frac{\dot{V}}{W} = \psi_u - \psi_l = V_{\text{freestream}} y_u - V_\infty y_l = V_{\text{freestream}}\left(y_u - y_l\right) \qquad (1)$$

By definition of streamlines, the volume flow rate between the two dividing streamlines must be the same as that through the probe itself. We know the volume flow rate through the probe from the results of the previous problem. The value of the stream function on the lower and upper dividing streamlines are the same as those of the previous problem, namely ψ_l = 0.093 m^2/s and ψ_u = 0.150 m^2/s respectively. We also know $y_u - y_l$ from the information given here. Thus, Eq. 1 yields

Freestream speed:
$$V_{\text{free stream}} = \frac{\psi_u - \psi_i}{y_u - y_i} = \frac{(0.150 - 0.093)\,\text{m}^2/\text{s}}{0.00624\,\text{m}} = 9.134624\ \text{m/s} \cong \mathbf{9.13\,m/s} \qquad (2)$$

Discussion We verify by these calculations that the sampling is superisokinetic (average speed through the probe is higher than that of the upstream air stream).

9-55

Solution For a given stream function we are to generate expressions for the velocity components.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *r*-θ plane.

Analysis We differentiate ψ to find the velocity components in cylindrical coordinates,

Radial velocity component:
$$\boxed{u_\theta = \frac{1}{r}\frac{\partial \psi}{\partial \theta} = V\cos\theta\left(1 - \frac{a^2}{r^2}\right)}$$

Tangential velocity component:
$$\boxed{u_\theta = -\frac{\partial \psi}{\partial r} = -V\sin\theta\left(1 + \frac{a^2}{r^2}\right)}$$

Discussion The radial velocity component is zero at the cylinder surface (*r* = *a*), but the tangential velocity component is not. In other words, this approximation does not satisfy the no-slip boundary condition along the cylinder surface. See Chap. 10 for a more detailed discussion about such approximations.

9-56

Solution We are to verify that the given ψ satisfies the continuity equation, and we are to discuss any restrictions.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric (ψ is a function of r and z only).

Analysis We plug the given velocity components into the axisymmetric continuity equation,

$$\frac{1}{r}\frac{\partial(ru_r)}{\partial r}+\frac{\partial(u_z)}{\partial z}=\frac{1}{r}\frac{\partial\left(-\frac{\partial\psi}{\partial z}\right)}{\partial r}+\frac{\partial\left(\frac{1}{r}\frac{\partial\psi}{\partial r}\right)}{\partial z}=\frac{1}{r}\left(-\frac{\partial^2\psi}{\partial r\partial z}+\frac{\partial^2\psi}{\partial z\partial r}\right)=0$$

Thus we see that **continuity is satisfied by the given stream function**. The only restriction on ψ is that **ψ must be a smooth function of *r* and *z***.

Discussion For a smooth function of two variables, the order of differentiation does not matter.

9-57

Solution For a given velocity field we are to generate an expression for ψ.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis We start by picking one of the two definitions of the stream function (it doesn't matter which part we choose – the solution will be identical).

$$\frac{\partial\psi}{\partial y}=u=V\cos\alpha \qquad (1)$$

Next we integrate Eq. 1 with respect to y, noting that this is a *partial* integration and we must add an arbitrary function of the other variable, x, rather than a simple constant of integration.

$$\psi=yV\cos\alpha+g(x) \qquad (2)$$

Now we choose the other part of the definition of ψ, differentiate Eq. 2, and rearrange as follows:

$$v=-\frac{\partial\psi}{\partial x}=-g'(x) \qquad (3)$$

where $g'(x)$ denotes dg/dx since g is a function of only one variable, x. We now have two expressions for velocity component v, the given equation and Eq. 3. We equate these and integrate with respect to x to find $g(x)$,

$$v=V\sin\alpha=-g'(x) \qquad g'(x)=-V\sin\alpha \qquad g(x)=-xV\sin\alpha+C \qquad (4)$$

Note that here we have added an arbitrary constant of integration C since g is a function of x only. Finally, plugging Eq. 4 into Eq. 2 yields the final expression for ψ,

Stream function: $$\boxed{\psi=V(y\cos\alpha-x\sin\alpha)+C} \qquad (5)$$

Constant C is arbitrary; it is common to set it to zero, although it can be set to any desired value.

Discussion You can verify by differentiating ψ that Eq. 5 yields the correct values of u and v.

9-58

Solution For a given stream function, we are to calculate the velocity components and verify incompressibility.

Assumptions **1** The flow is steady. **2** The flow is incompressible (this assumption is to be verified). **3** The flow is two-dimensional in the *x*-*y* plane, implying that $w = 0$ and neither u nor v depend on z.

Analysis (*a*) We use the definition of ψ to obtain expressions for u and v.

Velocity components:
$$\boxed{u = \frac{\partial \psi}{\partial y} = bx + 2cy \qquad v = -\frac{\partial \psi}{\partial x} = -2ax - by} \qquad (1)$$

(*b*) We check if the incompressible continuity equation in the *x*-*y* plane is satisfied by the velocity components of Eq. 1,

Incompressible continuity:
$$\underbrace{\frac{\partial u}{\partial x}}_{b} + \underbrace{\frac{\partial v}{\partial y}}_{-b} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0} = 0 \qquad b - b = 0 \qquad (2)$$

We conclude that the flow is indeed incompressible.

Discussion Since ψ is a smooth function of x and y, it automatically satisfies the continuity equation by its definition. Equation 2 confirms this. If it did not, we would go back and look for an algebra mistake somewhere.

9-59

Solution We are to plot several streamlines for a given velocity field.

Analysis We re-write the stream function equation of the previous problem with all the terms on one side,

$$cy^2 + bxy + ax^2 - \psi = 0 \qquad (1)$$

For any constant value of ψ (along a streamline), Eq. 1 is in a form that enables us to use the quadratic rule to solve for y as a function of x,

Equation for a streamline:
$$y = \frac{-bx \pm \sqrt{b^2x^2 - 4c\left(ax^2 - \psi\right)}}{2c} \qquad (2)$$

We plot the streamlines in Fig. 1. For each value of ψ there are two curves – one for the positive root and one for the negative root of Eq. 2. There is symmetry about a diagonal line through the origin. The streamlines appear to be hyperbolae. We determine the flow direction by plugging in a couple values of x and y and calculating the velocity components; e.g., at x = 1 m and y = 3 m, u = 2.7 m/s and v = 4.9 m/s. The flow at this point is in the upper right direction. Similarly, at x = 1 m and y = -2 m, u = -3.3 m/s and v = -1.6 m/s. The flow at this point is in the lower left direction.

Discussion This flow may not represent any particular physical flow field, but it produces an interesting flow pattern.

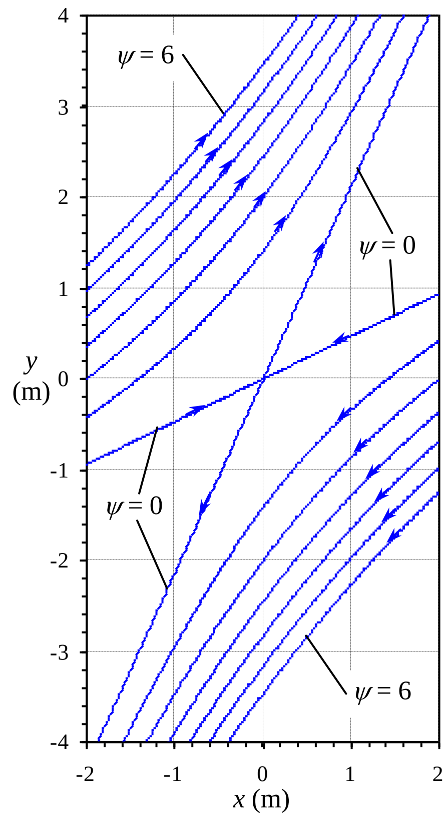


FIGURE 1
Streamlines for a given velocity field. Values of ψ are in units of m^2/s.

9-60

Solution For a given stream function, we are to calculate the velocity components and verify incompressibility.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane, implying that $w = 0$ and neither u nor v depend on z.

Analysis (*a*) We use the definition of ψ to obtain expressions for u and v.

Velocity components:
$$\boxed{u = \frac{\partial \psi}{\partial y} = -2by + dx \qquad v = -\frac{\partial \psi}{\partial x} = -2ax - c - dy} \qquad (1)$$

(*b*) We check if the incompressible continuity equation in the *x*-*y* plane is satisfied by the velocity components of Eq. 1,

Incompressible continuity:
$$\underbrace{\frac{\partial u}{\partial x}}_{d} + \underbrace{\frac{\partial v}{\partial y}}_{-d} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0} = 0 \qquad d - d = 0 \qquad (2)$$

We conclude that the flow is indeed incompressible.

Discussion Since ψ is a smooth function of x and y, it automatically satisfies the continuity equation by its definition. Eq. 2 confirms this. If it did not, we would go back and look for an algebra mistake somewhere.

9-61

Solution We are to make up a stream function $\psi(x,y)$, calculate the velocity components and verify incompressibility.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis Every student should have a different stream function. He or she then takes the derivatives with respect to y and x to find u and v. The student should then plug his/her velocity components into the incompressible continuity equation. **Continuity will be satisfied regardless of $\psi(x,y)$, provided that $\psi(x,y)$ is a smooth function of x and y**.

Discussion As long as ψ is a smooth function of x and y, it automatically satisfies the continuity equation by its definition.

9-62

Solution We are to calculate the percentage of flow going through one branch of a branching duct.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis For 2-D incompressible flow the difference in the value of the stream function between two streamlines is equal to the volume flow rate per unit width between the two streamlines. Thus,

Main branch:
$$\left.\frac{\dot{V}}{W}\right)_{\text{main}} = \psi_{\text{upper wall}} - \psi_{\text{lower wall}} = \left(4.35 - 2.03\right)\frac{\text{m}^2}{\text{s}} = 2.32\frac{\text{m}^2}{\text{s}} \qquad (1)$$

Similary, in the upper branch,

Upper branch:
$$\left.\frac{\dot{V}}{W}\right)_{\text{upper}} = \psi_{\text{upper wall}} - \psi_{\text{branch wall}} = (4.35 - 3.10)\frac{\text{m}^2}{\text{s}} = 1.25\frac{\text{m}^2}{\text{s}} \qquad (2)$$

On a percentage basis, the percentage of volume flow through the upper branch is calculated as

$$\frac{\dot{V}_{\text{upper}}}{\dot{V}_{\text{main}}} = \frac{\left.\frac{\dot{V}}{W}\right)_{\text{upper}}}{\left.\frac{\dot{V}}{W}\right)_{\text{main}}} = \frac{1.25\frac{\text{m}^2}{\text{s}}}{2.32\frac{\text{m}^2}{\text{s}}} = 0.53879 \cong \mathbf{53.9\%} \qquad (3)$$

Discussion No dimensions are given, so it is not possible to calculate velocities.

9-63

Solution We are to calculate duct height *h* for a given average velocity through a duct and values of stream function along the duct walls.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis The volume flow rate through the main branch of the duct is equal to the average velocity times the cross-sectional area of the duct,

Volume flow rate:
$$\dot{V} = V_{\text{avg}} W h \qquad (1)$$

We solve for h in Eq. 1, using the results of the previous problem,

Duct height:
$$h = \frac{1}{V_{\text{avg}}} \frac{\dot{V}}{W}\Bigg)_{\text{main}} = \frac{1}{13.4\ \frac{\text{m}}{\text{s}}} \times 2.32\ \frac{\text{m}^2}{\text{s}} \left(\frac{100\ \text{cm}}{\text{m}}\right) = 17.3134\ \text{cm} \cong \mathbf{17.3\ cm} \qquad (2)$$

An alternative way to solve for height *h* is to assume uniform flow in the main branch, for which $\psi = V_{\text{avg}} y$. We take the difference between ψ at the top of the duct and ψ at the bottom of the duct to find *h*,

$$\psi_{\text{upper wall}} - \psi_{\text{lower wall}} = V_{\text{avg}} y_{\text{upper wall}} - V_{\text{avg}} y_{\text{lower wall}} = V_{\text{avg}} \left(y_{\text{upper wall}} - y_{\text{lower wall}} \right) = V_{\text{avg}} h$$

Thus,

Duct height:
$$h = \frac{\psi_{\text{upper wall}} - \psi_{\text{lower wall}}}{V_{\text{avg}}} = \frac{(4.35 - 2.03)\ \frac{\text{m}^2}{\text{s}}}{13.4\ \frac{\text{m}}{\text{s}}} \left(\frac{100\ \text{cm}}{\text{m}}\right) = 17.3134\ \text{cm} \cong \mathbf{17.3\ cm} \qquad (3)$$

You can see that we get the same result as that of Eq. 2.

Discussion The result is correct even if the velocity profile through the duct is not uniform, since we have used the *average* velocity in our calculations.

9-64E

Solution For a given velocity field we are to generate an expression for ψ and plot several streamlines for given values of constants a and b.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the x-y plane, implying that w = 0 and neither u nor v depend on z.

Analysis We plug the given equation into the steady incompressible continuity equation,

Condition for incompressibility:
$$\frac{\partial v}{\partial y}=-\underbrace{\frac{\partial u}{\partial x}}_{2ax-by}-\underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0} \qquad \frac{\partial v}{\partial y}=-2ax+by$$

Next we integrate with respect to y. Note that since the integration is a *partial* integration, we must add some arbitrary function of x instead of simply a constant of integration.

y component of velocity:
$$v=-2axy+\frac{by^2}{2}+f(x)$$

If the flow were three-dimensional, we would add a function of x and z instead. We are told that v = 0 for all values of x when y = 0. This is only possible if $f(x)$ = 0. Thus,

y component of velocity:
$$v=-2axy+\frac{by^2}{2} \qquad (1)$$

To obtain the stream function, we start by picking one of the two parts of the definition of the stream function,

$$\frac{\partial\psi}{\partial y}=u=ax^2-bxy$$

Next we integrate the above equation with respect to y, noting that this is a *partial* integration and we must add an arbitrary function of the other variable, x, rather than a simple constant of integration.

$$\psi=ax^2y-\frac{bxy^2}{2}+g(x) \qquad (2)$$

Now we choose the other part of the definition of ψ, differentiate Eq. 2, and rearrange as follows:

$$v=-\frac{\partial\psi}{\partial x}=-2axy+\frac{by^2}{2}-g'(x) \qquad (3)$$

where $g'(x)$ denotes dg/dx since g is a function of only one variable, x. We now have two expressions for velocity component v, Eq. 1 and Eq. 3. We equate these and integrate with respect to x to find $g(x)$,

$$g'(x)=0 \qquad g(x)=C \qquad (4)$$

Note that here we have added an arbitrary constant of integration C since g is a function of x only. But C must be zero in order for ψ to be zero for any value of x when y = 0. Finally, Eq. 2 yields the final expression for ψ,

Solution:
$$\boxed{\psi=ax^2y-\frac{bxy^2}{2}} \qquad (5)$$

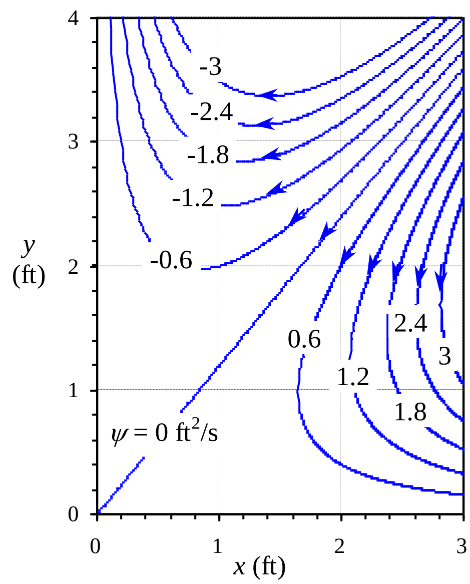


FIGURE 1
Streamlines for a given velocity field; the value of constant ψ is indicated for each streamline in units of ft^2/s.

To plot the streamlines, we note that Eq. 5 represents a *family* of curves, one unique curve for each value of the stream function ψ. We solve Eq. 5 for y as a function of x. A bit of algebra (the quadratic rule) yields

Equation for streamlines:
$$y = \frac{ax^2 \pm \sqrt{a^2x^4 - 2\psi bx}}{bx} \qquad (6)$$

For the given values of constants a and b, we plot Eq. 6 for several values of ψ in Fig. 1; these curves of constant ψ are streamlines of the flow. Note that both the positive and negative roots of Eq. 6 are required to plot each streamline. The direction of the flow is found by calculating u and v at some point in the flow field. We pick x = 2 ft, y = 2 ft, where u = –1.2 ft/s and v = –2.1 ft/s. This indicates flow to the lower left near this location. We fill in the rest of the arrows in Fig. 1 to be consistent. We see that the flow enters from the upper right, and splits into two parts – one to the lower right and one to the upper left.

Discussion It is always a good idea to check your algebra. In this example, you should differentiate Eq. 5 to verify that the velocity components of the given equation are obtained.

9-65

Solution We are to generate an expression for the stream function that describes a given velocity field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric (ψ is a function of r and z only).

Analysis The r and z velocity components from Problem 9-34 are

Velocity field:
$$u_r = -\frac{r}{2}\frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L} \qquad u_z = u_{z,\text{entrance}} + \frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L} z \qquad (1)$$

To generate the stream function we use the definition of ψ for steady, incompressible, axisymmetric flow,

Axisymmetric stream function:
$$u_r = -\frac{1}{r}\frac{\partial \psi}{\partial z} \qquad u_z = \frac{1}{r}\frac{\partial \psi}{\partial r} \qquad (2)$$

We choose one of the definitions of Eq. 2 to integrate. We pick the second one,

Integration:
$$\begin{aligned}\psi &= \int r u_z dr = \int r\left(u_{z,\text{entrance}} + \frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L} z\right) dr \\ &= \frac{r^2}{2}\left(u_{z,\text{entrance}} + \frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L} z\right) + f(z)\end{aligned} \qquad (3)$$

We added a function of z instead of a constant of integration since this is a partial integration. Now we take the z derivative of Eq. 3 and use the other half of Eq. 2,

Differentiation:
$$u_r = -\frac{1}{r}\frac{\partial \psi}{\partial z} = -\frac{r}{2}\frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L} - \frac{1}{r} f'(z) \qquad (4)$$

We equate Eq. 4 to the known value of u_r from Eq. 1,

Comparison:
$$u_r = -\frac{r}{2}\frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L} - \frac{1}{r} f'(z) = -\frac{r}{2}\frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L} \quad \text{or} \quad f'(z) = 0 \qquad (5)$$

Since f is a function of z only, integration of Eq. 5 yields $f(z)$ = constant. The final result is thus

Stream function:
$$\boxed{\psi = \frac{r^2}{2}\left(u_{z,\text{entrance}} + \frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L} z\right) + \text{constant}} \qquad (6)$$

Discussion The constant of integration can be any value since velocity components are determined by taking *derivatives* of the stream function.

9-66E

Solution We are to calculate the axial speed at the entrance and exit of the nozzle, and we are to plot several streamlines for a given axisymmetric flow field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric (ψ is a function of r and z only).

Analysis (a) Since u_z is not a function of radius, the axial velocity profile across a cross section of the nozzle is uniform. (This is consistent with the assumption that frictional effects along the nozzle walls are neglected.) Thus, at any cross section the axial speed is equal to the volume flow rate divided by cross-sectional area,

Entrance axial speed:

$$u_{z,\text{entrance}} = \frac{4\dot{V}}{\pi D_{\text{entrance}}^{\ 2}} = \frac{4\times 2.0\ \dfrac{\text{gal}}{\text{min}}}{\pi\left(0.50\text{ in}\right)^2}\left(\frac{0.1337\text{ ft}^3}{\text{gal}}\right)\left(\frac{12\text{ in}}{\text{ft}}\right)^2\left(\frac{\text{min}}{60\text{ s}}\right) = \mathbf{3.268\ \frac{ft}{s}} \qquad (1)$$

Similarly,

Exit axial speed:

$$u_{z,\text{exit}} = \frac{4\dot{V}}{\pi D_{\text{exit}}^{\ 2}} = \mathbf{41.69\ \frac{ft}{s}} \qquad (2)$$

(b) We use the stream function developed in Problem 9-61. Setting the constant to zero for simplicity, we have

Stream function:

$$\psi = \frac{r^2}{2}\left(u_{z,\text{entrance}} + \frac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L}z\right) \qquad (3)$$

We solve Eq. 3 for r as a function of z and plot several streamlines in Fig. 1,

Streamlines:

$$r = \pm\sqrt{\frac{2\psi}{u_{z,\text{entrance}} + \dfrac{u_{z,\text{exit}} - u_{z,\text{entrance}}}{L}z}} \qquad (4)$$

At the nozzle entrance (z = 0), the wall is at $r = D_{\text{entrance}}/2$ = 0.25 inches. Eq. 3 yields ψ_{wall} = 0.0007073 ft^3/s for the streamline that passes through this point. This streamline thus represents the shape of the nozzle wall, and we have designed the nozzle shape.

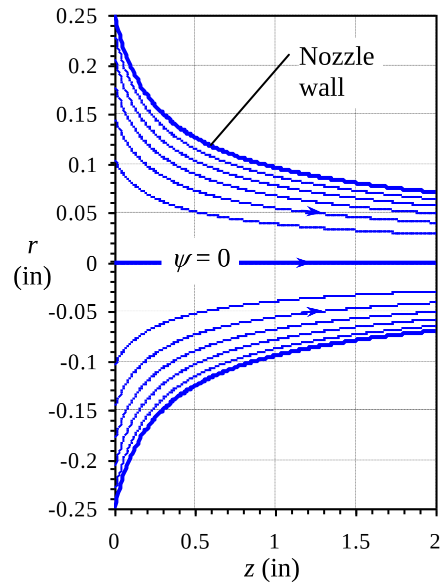


FIGURE 1
Streamlines for flow through an axisymmetric garden hose nozzle. Note that the vertical axis is highly magnified.

Discussion You can verify that the diameter between the outermost streamlines varies from D_{entrance} to D_{exit}.

9-67

Solution We are to discuss the sign of the stream function in a separation bubble, and determine where ψ is a minimum.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis For 2-D incompressible flow the difference in the value of the stream function between two streamlines is equal to the volume flow rate per unit width between the two streamlines. For example, $\psi_{upper} - \psi_0$ is positive and represents the volume flow rate per unit width between the wall and the uppermost streamline, The flow between these two streamlines is to the right. Likewise, the difference between ψ along the dividing streamline and $\psi = \psi_1$ along a streamline in the upper part of the separation bubble must also be positive as sketched in Fig. 1. The arc-shaped dividing streamline divides fluid within the separation bubble from fluid outside of the separation bubble. The stream function along this dividing streamline must be zero since it intersects the wall where $\psi = 0$. The only way we can have flow to the right in the upper part of the separation bubble is if ψ_1 is *negative* (Fig. 1). We conclude that **for this problem, all streamlines within the separation bubble have negative values of stream function**. **The minimum value of ψ occurs in the center of the separation bubble as sketched in Fig. 1**.

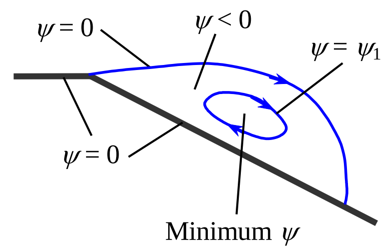


FIGURE 1
Close-up of streamlines near the separation bubble. The minimum value of the stream function occurs in the middle of the separation bubble.

Discussion We cannot conclude that ψ is always negative within a separation bubble, since we can add any arbitrary constant to all the ψ values, and it will not change the flow.

9-68

Solution We are to discuss how someone can interpret the relative speed of a flow based solely on contours of constant stream function.

Assumptions **1** The flow is steady. **2** The flow is incompressible.

Analysis For 2-D incompressible flow the difference in the value of the stream function between two streamlines is equal to the volume flow rate per unit width between the two streamlines. Thus, if the streamlines are very close together, the speed of the fluid between them is large relative to locations where the same two streamlines are far apart. **Professor Flows noticed a region in which the streamlines were very close together, implying high relative speed in that region of the flow**.

Discussion If the values of ψ on the contour plot are labeled, we can actually infer the fluid speed by measuring the distance between streamlines.

9-69

Solution For the given set of streamlines, we are to discuss how we can tell the relative speed of the fluid.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric.

Analysis As with 2-D flow, when streamlines that are initially equally spaced spread away from each other, it indicates that the flow speed has decreased in that region. Likewise, if the streamlines come closer together, the flow speed has increased in that region. From the figure provided in the problem statement, we infer that the flow far upstream of the plate is straight and uniform, since the streamlines are parallel. The fluid decelerates as it approaches the front face of the cylinder, especially near the stagnation point, as indicated by the wide gap between streamlines. The flow accelerates rapidly to very high speeds around the corner of the cylinder as indicated by the tightly spaced streamlines there. The flow is seen to separate on top of the cylinder. Since the streamlines are very sparse in this region, **we infer that the fluid moves relatively slowly inside the separation bubble**.

Discussion Such analyses in axisymmetric flow fields are more difficult than those in 2-D planar flow fields because streamlines of equally spaced stream function are *not* spaced equally apart in a uniform axisymmetric flow field. This is due to the fact that the cross-sectional area between streamlines increases with radius (a factor of $2\pi r$ is introduced). Nevertheless, we can still tell where the flow speeds up and slows down in this example.

9-70E

Solution We are to interpret a streamline plot by determining the direction of flow and by estimating the speed of the flow at a point.

Assumptions **1** The flow is steady. **2** The flow is incompressible **3** The flow is two-dimensional.

Analysis (*a*) We must tilt our heads nearly upside down to see an increase in stream function ψ in the mathematically positive manner. In other words, since ψ increases in the downward direction, the flow is to the lower left, following our left side rule. Arrows are drawn in Fig. 1.

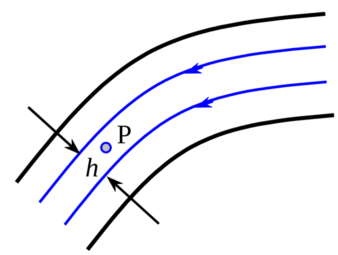


FIGURE 1
Streamlines with direction shown.

(*b*) For 2-D incompressible flow the difference in the value of the stream function between two streamlines is equal to the volume flow rate per unit width between the two streamlines. We approximate the flow as uniform between the two labeled streamlines in the figure provided in the problem statement. The speed at point P is thus

$$V_P \approx \frac{\dot{V}}{Wh} = \frac{1}{h}\frac{\dot{V}}{W} = \frac{1}{h}\left(\psi_1 - \psi_2\right) = \frac{1}{1.58\text{ in}}\left(0.45 - 0.32\right)\frac{\text{ft}^2}{\text{s}}\left(\frac{12\text{ in}}{\text{ft}}\right) = 0.987342\frac{\text{ft}}{\text{s}} \cong \mathbf{0.99}\,\frac{\mathbf{ft}}{\mathbf{s}} \qquad (1)$$

(*c*) Nowhere did we use any property of the fluid, so changing to water does not change our result. **For either air or water (or any incompressible fluid), V_P = 0.99 ft/s** (to two significant digits).

Discussion Streamlines and stream functions are *kinematic* properties, as discussed in Chap. 4. That is why fluid density, viscosity, etc. are irrelevant here.

9-71

Solution We are to find the primary dimensions and primary units of the compressible stream function.

Analysis From the given definition, we see that ψ_ρ is the product of a density, a velocity, and a length,

Primary dimensions of ψ_ρ: $$\boxed{\{\psi_\rho\} = \left\{\frac{\text{mass}}{\text{length}^3}\times\frac{\text{length}}{\text{time}}\times\text{length}\right\} = \left\{\frac{\text{m}}{\text{Lt}}\right\}}$$

The primary units of ψ_ρ are kg/(m·s) (SI) and lbm/(ft s) (English).

Discussion Ironically, although the stream function is often applied to potential flows where viscosity is not a parameter, ψ_ρ has the same units as μ.

9-72

Solution We are to generate an expression for the compressible stream function for a given flow field.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis We start by picking one of the two definitions of the compressible stream function (it doesn't matter which part we choose – the solution will be identical).

$$\frac{\partial\psi_\rho}{\partial y} = \rho u = \left(\rho_1 + C_\rho x\right)\left(u_1 + C_u x\right) = \rho_1 u_1 + \left(\rho_1 C_u + u_1 C_\rho\right)x + C_\rho C_u x^2 \qquad (1)$$

Next we integrate Eq. 1 with respect to *y*, noting that this is a *partial* integration and we must add an arbitrary function of the other variable, *x*, rather than a simple constant of integration.

$$\psi_\rho = \rho_1 u_1 y + \left(\rho_1 C_u + u_1 C_\rho\right)xy + C_\rho C_u x^2 y + g(x) \qquad (2)$$

Now we choose the other part of the definition of ψ, differentiate Eq. 2, and rearrange as follows:

$$-\rho v = \frac{\partial\psi_\rho}{\partial x} = \left(\rho_1 C_u + u_1 C_\rho\right)y + 2C_\rho C_u xy + g'(x) \qquad (3)$$

where $g'(x)$ denotes dg/dx since g is a function of only one variable, x. We now have two expressions for $-\rho v$, Eq. 3 and the value computed from the known density and velocity, i.e.

$$-\rho v = \left(\rho_1 C_u + u_1 C_\rho\right)y - 2C_u C_\rho xy \qquad (4)$$

We equate Eqs. 3 and 4 and integrate with respect to x to find $g(x)$,

$$g'(x) = 0 \qquad g(x) = C \qquad (5)$$

Note that here we have added an arbitrary constant of integration C since g is a function of x only. Plugging Eq. 5 into Eq. 2 yields

$$\psi_\rho = \rho_1 u_1 y + \left(\rho_1 C_u + u_1 C_\rho\right)xy + C_\rho C_u x^2 y + C \qquad (6)$$

We determine constant C by setting $\psi_\rho = 0$ at $y = 0$ in Eq. 6, yielding $C = 0$. Thus the final expression for the compressible stream function is

Compressible stream function: $$\boxed{\psi_\rho = \rho_1 u_1 y + \left(\rho_1 C_u + u_1 C_\rho\right)xy + C_\rho C_u x^2 y} \qquad (7)$$

Discussion You can verify by differentiating ψ_ρ that Eq. 7 yields the correct values of u and v.

9-73

Solution We are to generate an expression for the compressible stream function for a given flow field.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis We start by picking one of the two definitions of the compressible stream function (it doesn't matter which part we choose – the solution will be identical).

$$\frac{\partial \psi_\rho}{\partial y} = \rho u = \left(\rho_1 + C_\rho x\right)\left(u_1 + C_u x\right) = \rho_1 u_1 + \left(\rho_1 C_u + u_1 C_\rho\right)x + C_\rho C_u x^2 \quad (1)$$

Next we integrate Eq. 1 with respect to *y*, noting that this is a *partial* integration and we must add an arbitrary function of the other variable, *x*, rather than a simple constant of integration.

$$\psi_\rho = \rho_1 u_1 y + \left(\rho_1 C_u + u_1 C_\rho\right)xy + C_\rho C_u x^2 y + g\left(x\right) \quad (2)$$

Now we choose the other part of the definition of ψ, differentiate Eq. 2, and rearrange as follows:

$$-\rho v = \frac{\partial \psi_\rho}{\partial x} = \left(\rho_1 C_u + u_1 C_\rho\right)y + 2C_\rho C_u xy + g'\left(x\right) \quad (3)$$

where $g'(x)$ denotes dg/dx since *g* is a function of only one variable, *x*. We now have two expressions for $-\rho v$, Eq. 3 and the value computed from the known density and velocity, i.e.

$$-\rho v = \left(\rho_1 C_u + u_1 C_\rho\right)y - 2C_u C_\rho xy \quad (4)$$

We equate Eqs. 3 and 4 and integrate with respect to *x* to find $g(x)$,

$$g'\left(x\right) = 0 \qquad g\left(x\right) = C \quad (5)$$

Note that here we have added an arbitrary constant of integration *C* since *g* is a function of *x* only. Plugging Eq. 5 into Eq. 2 yields

$$\psi_\rho = \rho_1 u_1 y + \left(\rho_1 C_u + u_1 C_\rho\right)xy + C_\rho C_u x^2 y + C \quad (6)$$

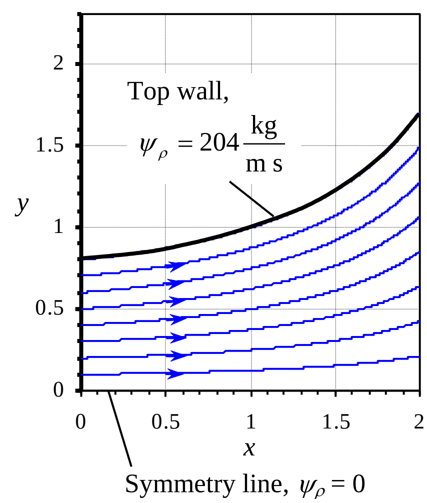


FIGURE 1
Streamlines for a diverging duct.

We determine constant *C* by setting $\psi_\rho = 0$ at $y = 0$ in Eq. 6, yielding $C = 0$. Thus the final expression for the compressible stream function is

Compressible stream function: $$\boxed{\psi_\rho = \rho_1 u_1 y + \left(\rho_1 C_u + u_1 C_\rho\right)xy + C_\rho C_u x^2 y} \quad (7)$$

We solve Eq. 7 for *y* as a function of *x* and ψ_ρ so that we can plot streamlines,

Equation for plotting streamlines: $$y = \frac{\psi_\rho}{\rho_1 u_1 + \left(\rho_1 C_u + u_1 C_\rho\right)x + C_\rho C_u x^2} \quad (8)$$

We plot Eq. 8 in Fig. 1 for several values of ψ_ρ, using the values of constants u_1, ρ_1, C_u, and C_ρ given in Problem 9-21. The agreement with the streamlines of Problem 9-21 is excellent.

The streamline starting at $x = 0$, $y = 0.8$ m is the top wall of the duct. Therefore the value of ψ_ρ at the top wall of the diverging duct is found be setting at $x = 0$ and $y = 0.8$ m,

ψ_ρ at the top wall: $$\psi_{\rho,\text{top}} = \rho_1 u_1 y = \left(0.85\frac{\text{kg}}{\text{m}^3}\right)\left(300\frac{\text{m}}{\text{s}}\right)\left(0.8\text{ m}\right) = \mathbf{204\frac{kg}{m\cdot s}} \quad (9)$$

Discussion You can verify by differentiating ψ_ρ that Eq. 9 yields the correct values of *u* and *v*.

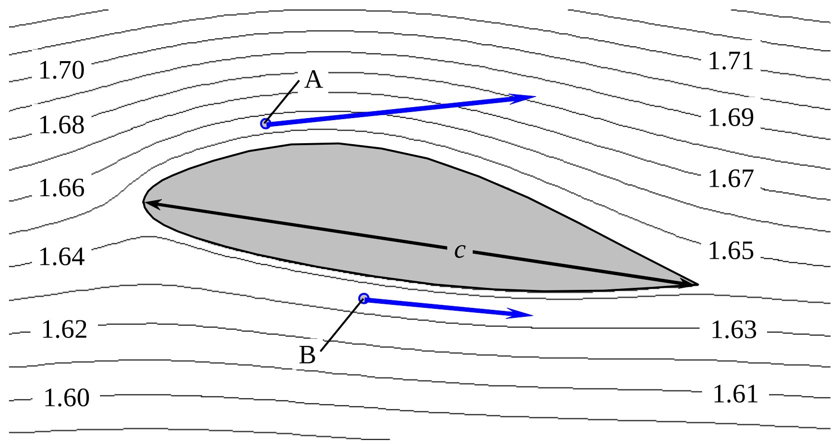
9-74

Solution We are to interpret a streamline plot by determining the direction of flow and by estimating the speed of the flow at a point.

Assumptions **1** The flow is steady. **2** The flow is incompressible **3** The flow is two-dimensional.

Analysis (*a*) We can tell the direction of the flow by whether ψ_ρ increases or decreases in the vertical direction (left side rule). We see that at points A and B the flow is to the right. Furthermore, since the streamlines near point B are somewhat further apart than those near point A (by a factor of about 1.6), the speed at point A is a factor of about 1.6 greater than that at point B. Arrows are drawn in Fig. 1.

FIGURE 1
Relative velocity vectors at points A and B, added to the streamline plot.



In terms of lift, it is obvious that the flow speeds near the upper surface of the hydrofoil are greater than those near the lower surface. From the Bernoulli equation we know that low speeds lead to (relatively) higher pressures; thus the pressure on the lower half of the hydrofoil is greater than that on the upper half, leading to lift.

(*b*) For 2-D incompressible flow the difference in the value of the stream function between two streamlines is equal to the volume flow rate per unit width between the two streamlines. We approximate the flow as uniform between the two streamlines that enclose point A in Fig. P9-70. By measurement with a ruler, we find that the distance δ between streamlines 1.65 and 1.66 is about 0.034*c*, or about (0.034)(9.0 mm) = 0.306 mm. The speed at point A is thus

$$V_{\text{A}} \approx \frac{\dot{V}}{W\delta} = \frac{1}{\delta}\frac{\dot{V}}{W} = \frac{1}{\delta}\left(\psi_{1.66} - \psi_{1.65}\right)$$

$$= \frac{1}{0.306\times10^{-3}\ \text{m}}\left(1.66-1.65\right)\frac{\text{m}^2}{\text{s}} = 32.7\ \frac{\text{m}}{\text{s}} \cong \mathbf{33.}\ \frac{\mathbf{m}}{\mathbf{s}}$$

We give our answer to only two significant digits here because of the difficulty of measuring the distance between the two streamlines.

Discussion Students' answers may vary somewhat depending on how accurately they measure the distance between streamlines. Values between 30 and 40 m/s are reasonable.

9-75

Solution We are to interpret a streamline plot by determining the direction of flow and by estimating the speed of the flow at a point.

Assumptions **1** The flow is steady (time-averaged). **2** The flow is incompressible **3** The flow is two-dimensional.

Properties The density of air at $T = 20^{\circ}\mathrm{C}$ is 1.18 kg/m^3.

Analysis (*a*) We can tell the direction of the flow by whether ψ_ρ increases or decreases in the vertical direction (left side rule). We see that at point A, the flow is to the left, while at point B the flow is to the right. Furthermore, since the streamlines near point B are much closer together than those near point A (by a factor of about five), the speed at point B is a factor of about five greater than that at point A. Arrows are drawn in Fig. 1.

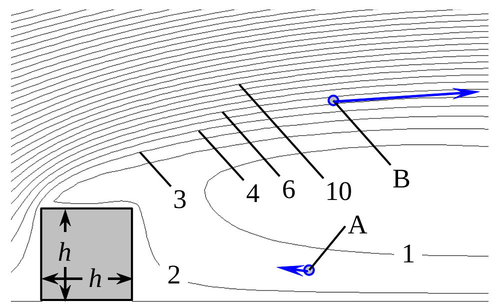


FIGURE 1
Relative velocity vectors at points A and B, added to the streamline plot.

(*b*) For 2-D incompressible flow the difference in the value of the compressible stream function between two streamlines is equal to the mass flow rate per unit width between the two streamlines. We approximate the flow as uniform between the two streamlines that enclose point B in Fig. P9-71. By measurement with a ruler, we find that the distance δ between streamlines 5 and 6 is about h/10, or about 0.10 m. The speed at point B is thus

$$V_{\mathrm{B}} \approx \frac{\dot{m}}{\rho W \delta} = \frac{1}{\rho\delta}\frac{\dot{m}}{W} = \frac{1}{\rho\delta}\left(\psi_6 - \psi_5\right) = \frac{1}{\left(1.18\,\frac{\mathrm{kg}}{\mathrm{m}^3}\right)\left(0.10\ \mathrm{m}\right)}\left(6-5\right)\frac{\mathrm{kg}}{\mathrm{m}\cdot\mathrm{s}} = \mathbf{8.5\,\frac{m}{s}} \qquad (1)$$

We are only accurate to one digit here because of the difficulty of measuring the distance between the two streamlines. We give our final result as $\boldsymbol{V_{\mathrm{B}}}$ **= 8 or 9 m/s**.

Discussion Students' answers may vary considerably depending on how accurately they measure the distance between streamlines.

9-76

Solution We are to determine the value of the stream function along the positive *y* axis and the negative *x* axis for the case of a line source at the origin.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* or *r*-θ plane.

Analysis For 2-D incompressible flow the difference in the value of the stream function between two streamlines is equal to the volume flow rate per unit width between the two streamlines. Let us take the arc of the circle of radius *r* between the positive *x* axis and the positive *y* axis of the figure provided with the problem statement. The volume flow rate per unit width through this arc is one-fourth of $\dot{V}/L$, the total volume flow rate per unit width, since the arc spans exactly one-fourth of the circumference of the circle.

$$\psi_{\text{positive } y \text{ axis}} - \psi_{\text{positive } x \text{ axis}} = \frac{1}{4}\frac{\dot{V}}{L} \quad (1)$$

Since $\psi = 0$ along the positive *x* axis, we conclude that

ψ along positive y axis:

$$\boxed{\psi_{\text{positive } y \text{ axis}} = \frac{1}{4}\frac{\dot{V}}{L}} \quad (2)$$

Similarly, the volume flow rate through the top half of the circle is half of the total volume flow rate and we conclude that

ψ along negative x axis:

$$\boxed{\psi_{\text{negative } y \text{ axis}} = \frac{1}{2}\frac{\dot{V}}{L}} \quad (3)$$

Discussion Some CFD codes use ψ as a variable, and we thus need to specify the value of ψ along boundaries of the computational domain. Simple calculations such as this are useful in these situations.

9-77

Solution We are to determine the value of the stream function along the positive *y* axis and the negative *x* axis for the case of a line sink at the origin.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* or *r*-θ plane.

Analysis Everything is the same as in Problem 9-59 except that the flow direction is reversed everywhere. The volume flow rate per unit width through the arc of radius *r* between the positive *x* axis and the positive *y* axis of Fig. P9-59 is now *negative* one-fourth of $\dot{V}/L$ since the flow is now mathematically negative.

$$\psi_{\text{positive } y \text{ axis}} - \psi_{\text{positive } x \text{ axis}} = -\frac{1}{4}\frac{\dot{V}}{L} \quad (1)$$

Since $\psi = 0$ along the positive *x* axis, we conclude that

ψ along positive y axis:

$$\boxed{\psi_{\text{positive } y \text{ axis}} = -\frac{1}{4}\frac{\dot{V}}{L}} \quad (2)$$

Similarly, the volume flow rate through the top half of the circle is half of the total volume flow rate and is negative. We conclude that

ψ along negative x axis:

$$\boxed{\psi_{\text{negative } x \text{ axis}} = -\frac{1}{2}\frac{\dot{V}}{L}} \quad (3)$$

Discussion We need to be careful of the sign of ψ.

Linear Momentum Equation, Boundary Conditions, and Applications

9-78C

Solution We are to define mechanical pressure and discuss its application.

Analysis **Mechanical pressure is the mean normal stress acting inwardly on a fluid element**. For an incompressible fluid, the density is constant and therefore we have no equation of state available for calculation of the thermodynamic pressure. In fact, thermodynamic pressure cannot even be defined for an incompressible fluid. Fluid elements and surfaces still "feel" a pressure, however, and this pressure is the so-called mechanical pressure.

Discussion When dealing with incompressible fluid flows, pressure variable P is always interpreted as the mechanical pressure P_m.

9-79C

Solution We are to describe the constitutive equations and name the equation to which they are applied.

Analysis **The constitutive equations are relationships between the components of the stress tensor and the primary unknowns of the problem, namely pressure and velocity**. The constitutive equations enable us to write the components of the stress tensor in *Cauchy's equation* in terms of the velocity field and the pressure field.

Discussion Cauchy's equation by itself is useless without the constitutive equations, because we would have too many unknowns for the number of available equations.

9-80C

Solution We are to discuss velocity boundary conditions in a stationary and a moving frame of reference for the case of an airplane flying through the air.

Analysis (*a*) From the stationary frame of reference, $\vec{V} = \vec{V}_{\text{airplane}}$ on all surfaces of the airplane, (no-slip boundary condition). Far from the airplane **the air is still** ($\vec{V}$ **= 0**).

(*b*) From the reference frame moving with the airplane, $\vec{V} = 0$ on all surfaces, (no-slip boundary condition). Far from the airplane the air is moving towards the airplane at **a speed that is opposite the airplane's speed** ($\vec{V} = -\vec{V}_{\text{airplane}}$).

Discussion The no-slip condition requires that the fluid velocity equal the airplane velocity everywhere on the airplane surface, regardless of the geometry of the airplane, and regardless of the frame of reference.

9-81C

Solution We are to discuss the difference between Newtonian fluids and non-Newtonian fluids, and we are to give examples of each.

Analysis The main distinction between a Newtonian fluid and a non-Newtonian fluid is that **for flow of a Newtonian fluid, shear stress is linearly proportional to shear strain rate**, whereas for flow of a non-Newtonian fluid, the relationship between shear stress and shear strain rate is nonlinear.

There are many examples of Newtonian fluids. Most pure, common liquids like water, oil, gasoline, alcohol, etc. are Newtonian. Most gases also behave like Newtonian fluids. Non-newtonian fluids include paint, pastes and creams, polymer solutions, cake batter, slurries and colloidal suspensions like quicksand, blood, etc.

Discussion The Navier-Stokes equations apply only to Newtonian fluids. For non-Newtonian fluids, you would need to insert nonlinear constitutive equations into Cauchy's equations in order to obtain a useful differential equation for conservation of linear momentum.

9-82C

Solution We are to define or describe each type of fluid.

Analysis

(*a*) **A viscoelastic fluid is a fluid that returns (either fully or partially) to its original shape after the applied stress is released**.

(*b*) **A pseudoplastic fluid is a shear thinning fluid – the more the fluid is sheared, the less viscous it becomes**.

(*c*) **A dilatant fluid is a shear thickening fluid – the more the fluid is sheared, the more viscous it becomes**.

(*d*) **A Bingham plastic fluid is an extreme type of pseudoplastic fluid that requires a finite stress called the *yield stress* in order for the fluid to flow at all**.

Discussion All of the above are examples of non-Newtonian fluids.

9-83C

Solution We are to discuss each term, and write the equation as a word equation.

Analysis **Term I is the net body force acting on the control volume. Term II is the net surface force acting on the control volume. Term III is the net rate of change of linear momentum within the control volume. Term IV is the net rate of outflow of linear momentum through the control surface**. In words, the equation can be expressed as: "**The total force acting on the control volume is the sum of body forces and surface forces, and is equal to the rate at which momentum changes within the control volume plus the rate at which momentum flows out of the control volume**."

Discussion The dimensions of each term in the equation are those of momentum per time. Each term has primary dimensions of $\{mLt^{-2}\}$.

9-84

Solution We are to generate and discuss velocity and pressure boundary conditions for the given flow problem.

Assumptions **1** The flow is steady. **2** Surface tension effects are negligibly small.

Analysis We must satisfy the no-slip boundary condition on all tank walls, $\vec{V}_{\text{liquid}} = \vec{V}_{\text{tank}}$. Mathematically, we write $\boldsymbol{u_r = u_z = 0}$ **and** $\boldsymbol{u_\theta = R\omega}$ **at** $\boldsymbol{r = R}$ (the tank side walls). We also write $\boldsymbol{u_r = u_z = 0}$ **and** $\boldsymbol{u_\theta = r\omega}$ **at** $\boldsymbol{z = 0}$ (the bottom wall of the tank). We do not specify the pressure along the tank walls. **At the free surface** $\boldsymbol{P = P_{atm}}$ since the free surface is exposed to atmospheric air. In addition, the vertical and radial components of velocity $\boldsymbol{u_z}$ **and** $\boldsymbol{u_r}$ **must equal zero at the free surface**, but the angular velocity component u_θ is set to $\boldsymbol{u_\theta = r\omega}$ **at the free surface**. We also know that the shear stress at free surface must be zero (negligible shear due to the air). This boundary condition is not needed however since we already know the velocity field. In fact, the velocity field is known right from the start since we are told that the liquid is in solid body rotation: $u_z = u_r = 0$ and $u_\theta = r\omega$ everywhere.

Discussion This is a degenerate case of the Navier-Stokes equation since the fluid is in solid body rotation. Nevertheless, it is useful to think about the required boundary conditions.

9-85

Solution We are to compare Eqs. 1 and 2 to see if they are the same or not.

Analysis We use the product rule to differentiate Eq. 1,

$$\tau_{r\theta} = \tau_{\theta r} = \mu\left(r u_\theta \frac{-1}{r^2} + \frac{r}{r}\frac{\partial u_\theta}{\partial r} + \frac{1}{r}\frac{\partial u_r}{\partial \theta} \right) = \mu\left(\frac{1}{r}\left(\frac{\partial u_r}{\partial \theta} - u_\theta \right) + \frac{\partial u_\theta}{\partial r} \right) \qquad (3)$$

Thus we see that **Eq. 1 and Eq. 2 are equivalent**.

Discussion The viscous stress tensor is defined identically in the other texts; the terms are simply grouped together in a different fashion.

9-86

Solution We are to estimate the volume flow rate of oil between two plates, and we are to calculate the Reynolds number.

Assumptions **1** The flow is steady. **2** The oil is incompressible. **3** Since the gap is so small compared to the plate dimensions, we assume 2-D flow in the *x*-*y* plane. **4** We ignore entrance effects and end effects and assume that the flow can be approximated as fully developed channel flow everywhere in the gap.

Properties The viscosity and density of unused engine oil at T = 60°C are 72.5 × 10^{-3} kg/(m·s) and 864 kg/m^3 respectively.

Analysis The velocity field for fully developed channel flow is

Velocity components, 2-D channel flow:
$$u = \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right) \qquad v = 0 \tag{1}$$

We integrate the *x* component of velocity times cross-sectional area to obtain volume flow rate (see also Problem 9-44),

Volume flow rate:
$$\dot{V} = \int_A u\,dA = \int_{y=0}^{y=h} \frac{1}{2\mu}\frac{dP}{dx}\left(y^2 - hy\right)W\,dy = -\frac{1}{12\mu}\frac{dP}{dx}h^3 W \tag{2}$$

The pressure gradient is approximated as

$$\frac{dP}{dx} \approx \frac{P_{\text{out}} - P_{\text{in}}}{L} = \frac{(0-1)\text{ atm}}{1.25\text{ m}}\left(\frac{101{,}300\text{ N/m}^2}{\text{atm}}\right) = -81{,}040\text{ N/m}^3 \tag{3}$$

We plug Eq. 3 into Eq. 2 and solve for the volume flow rate,

Volume flow rate:
$$\dot{V} = -\frac{1}{12\left(72.5\times10^{-3}\ \frac{\text{kg}}{\text{m}\cdot\text{s}}\right)}\left(-81{,}040\ \frac{\text{N}}{\text{m}^3}\right)(0.00360\text{ m})^3(0.550\text{ m})\left(\frac{\text{kg m}}{\text{s}^2\text{N}}\right) \tag{4}$$
$$= 2.39029\times10^{-3}\text{ m}^3\text{/s} \cong \mathbf{2.39\times10^{-3}\ m^3/s}$$

The average velocity of the oil through the channel is

Average velocity:
$$V = \frac{\dot{V}}{hW} = \frac{2.39029\times10^{-3}\text{ m}^3\text{/s}}{(0.00360\text{ m})(0.550\text{ m})} = 1.20722\text{ m/s} \cong 1.21\text{ m/s} \tag{5}$$

Finally, the characteristic Reynolds number is

$$\text{Re} = \frac{\rho V h}{\mu} = \frac{(864\text{ kg/m}^3)(1.20722\text{ m/s})(0.00360\text{ m})}{72.5\times10^{-3}\text{ kg/m}\cdot\text{s}} = \mathbf{51.8} \tag{6}$$

The flow is definitely laminar.

Discussion We give our final results to three significant digits.

9-87

Solution For a given velocity field, we are to calculate the pressure field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the x-y plane. **4** Gravity does not act in either the x or the y plane.

Analysis The flow field must satisfy the steady, two-dimensional, incompressible continuity *and* momentum equations. We check each equation separately; let's consider continuity first:

Continuity:
$$\underbrace{\frac{\partial u}{\partial x}}_{a}+\underbrace{\frac{\partial v}{\partial y}}_{-a}+\underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0\text{ (2-D)}}=0$$

Continuity is satisfied. Now we look at the x *component* of the Navier-Stokes equation:

x momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial u}{\partial t}}}_{0\text{ (steady)}}+\underbrace{u\frac{\partial u}{\partial x}}_{(ax+b)a}+\underbrace{v\cancel{\frac{\partial u}{\partial y}}}_{(-ay+c)0}+\underbrace{\cancel{w\frac{\partial u}{\partial z}}}_{0\text{ (2-D)}}\right)=-\frac{\partial P}{\partial x}+\underbrace{\cancel{\rho g_x}}_{0}+\mu\left(\underbrace{\cancel{\frac{\partial^2 u}{\partial x^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 u}{\partial y^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 u}{\partial z^2}}}_{0\text{ (2-D)}}\right) \quad (1)$$

The x momentum equation reduces to

x momentum:
$$\frac{\partial P}{\partial x}=\rho\left(-a^2x-ab\right) \quad (2)$$

The x momentum equation is satisfied provided we can generate a smooth pressure field that satisfies Eq. 2. In similar fashion (we don't show the details), the y momentum equation reduces to

y momentum:
$$\frac{\partial P}{\partial y}=\rho\left(-a^2y+ac\right) \quad (3)$$

The y momentum equation is satisfied provided we can generate a smooth pressure field that satisfies Eq. 3. The pressure field $P(x,y)$ must be a smooth function of x and y. Mathematically, this requires that the order of differentiation (x then y verses y then x) should not matter. We therefore check whether this is so by differentiating Eqs. 3 and 2 respectively:

Cross-differentiation:
$$\frac{\partial^2 P}{\partial x\partial y}=\frac{\partial}{\partial x}\left(\frac{\partial P}{\partial y}\right)=0 \qquad \frac{\partial^2 P}{\partial y\partial x}=\frac{\partial}{\partial y}\left(\frac{\partial P}{\partial x}\right)=0 \quad (4)$$

Equation 4 shows that indeed, P is a smooth function of x and y. Thus, *we should be able to calculate the pressure field*. To calculate $P(x,y)$, we start with either Eq. 2 or Eq. 3 and integrate. We pick Eq. 2, which we can partially integrate (with respect to x) to obtain an expression for $P(x,y)$,

Pressure field from x-momentum:
$$P(x,y)=\rho\left(-\frac{a^2x^2}{2}-abx\right)+g(y) \quad (5)$$

Note that we added an arbitrary function of the other variable y rather than a constant of integration since this is a partial integration. We then take the partial derivative of Eq. 5 with respect to y to obtain

$$\frac{\partial P}{\partial y}=g'(y)=\rho\left(-a^2y+ac\right) \quad (6)$$

where we have equated our result to Eq. 3 for consistency. We can now integrate Eq. 6 to obtain the function $g(y)$:

$$g(y)=\rho\left(-\frac{a^2y^2}{2}+acy\right)+C \quad (7)$$

where C is an arbitrary constant of integration. Finally, we plug Eq. 7 into Eq. 5 to obtain our final expression for $P(x,y)$. The result is

Solution:

$$\boxed{P(x,y)=\rho\left(-\frac{a^2x^2}{2}-\frac{a^2y^2}{2}-abx+acy\right)+C} \qquad (8)$$

Discussion For practice, you should differentiate Eq. 8 with respect to both *x* and *y*, and compare to Eqs. 2 and 3. (This also serves as a check of our algebra.) In addition, try to obtain Eq. 8 by starting with Eq. 3 rather than Eq. 2; you should get the same answer. Pressure is found to within some arbitrary constant *C* since the absolute magnitude of pressure is irrelevant; only pressure *gradients* are important.

9-88

Solution For a given velocity field, we are to calculate the pressure field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** Gravity does not act in either the *x* or the *y* direction.

Analysis The flow field must satisfy the steady, two-dimensional, incompressible continuity *and* momentum equations. We check each equation separately; let's consider continuity first:

Continuity:
$$\underbrace{\frac{\partial u}{\partial x}}_{-2ax}+\underbrace{\frac{\partial v}{\partial y}}_{2ax}+\underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0\text{ (2-D)}}=0$$

Continuity is satisfied. Now we look at the *x component* of the Navier-Stokes equation:

x momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial u}{\partial t}}}_{0\text{ (steady)}}+\underbrace{u\frac{\partial u}{\partial x}}_{(-ax^2)(-2ax)}+\underbrace{v\cancel{\frac{\partial u}{\partial y}}}_{(2axy)(0)}+\underbrace{\cancel{w\frac{\partial u}{\partial z}}}_{0\text{ (2-D)}}\right)=-\frac{\partial P}{\partial x}+\underbrace{\cancel{\rho g_x}}_{0}+\mu\left(\underbrace{\frac{\partial^2 u}{\partial x^2}}_{-2a}+\underbrace{\cancel{\frac{\partial^2 u}{\partial y^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 u}{\partial z^2}}}_{0\text{ (2-D)}}\right) \qquad (1)$$

equation 1 reduces to

x momentum:
$$\frac{\partial P}{\partial x}=-2\rho a^2x^3-2\mu a \qquad (2)$$

The *x* momentum equation is satisfied provided we can generate a pressure field that satisfies Eq. 2. In similar fashion we examine the *y* momentum equation,

y momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial v}{\partial t}}}_{0\text{ (steady)}}+\underbrace{u\frac{\partial v}{\partial x}}_{(-ax^2)(2ay)}+\underbrace{v\frac{\partial v}{\partial y}}_{(2axy)(2ax)}+\underbrace{\cancel{w\frac{\partial v}{\partial z}}}_{0\text{ (2-D)}}\right)=-\frac{\partial P}{\partial y}+\underbrace{\cancel{\rho g_y}}_{0}+\mu\left(\underbrace{\cancel{\frac{\partial^2 v}{\partial x^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 v}{\partial y^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 v}{\partial z^2}}}_{0\text{ (2-D)}}\right)$$

The *y* momentum equation reduces to

y momentum:
$$\frac{\partial P}{\partial y}=-2\rho a^2x^2y \qquad (3)$$

The *y* momentum equation is satisfied provided we can generate a pressure field that satisfies Eq. 3.

In the two-dimensional flow under discussion here, the pressure field $P(x,y)$ must be a smooth function of *x* and *y*. Mathematically, this requires that the order of differentiation (*x* then *y* versus *y* then *x*) should not matter. We check whether this is so by differentiating Eqs. 3 and 2 respectively:

Cross-differentiation:
$$\frac{\partial^2 P}{\partial y\partial x}=\frac{\partial}{\partial y}\left(\frac{\partial P}{\partial x}\right)=0 \qquad \frac{\partial^2 P}{\partial x\partial y}=\frac{\partial}{\partial x}\left(\frac{\partial P}{\partial y}\right)=-2\rho a^2x^2 \qquad (4)$$

Since the cross-derivative terms in Eq. 4 do not match, *P* is *not* a smooth function of *x* and *y*. Thus, **we are unable to calculate a steady, incompressible, two-dimensional pressure field with the given velocity field**. We cannot proceed any further.

Discussion This problem shows that even if a velocity field satisfies the continuity equation (conservation of mass), and even if we can plot streamlines for the flow field, this does not necessarily guarantee that the velocity field is physically possible. In the present case, for instance, we are unable to find a pressure field that can satisfy the steady Navier-Stokes equation.

9-89

Solution For a given velocity field, we are to calculate the pressure field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the r-θ plane. **4** Gravity does not act in either the r or the θ direction.

Analysis The flow must satisfy the steady, two-dimensional, incompressible continuity *and* momentum equations. We check each equation separately, starting with continuity,

Continuity:
$$\frac{1}{r}\underbrace{\cancel{\frac{\partial(ru_r)}{\partial r}}}_{0}+\frac{1}{r}\underbrace{\cancel{\frac{\partial(u_\theta)}{\partial \theta}}}_{0}+\underbrace{\cancel{\frac{\partial(u_z)}{\partial z}}}_{0}=0$$

Continuity is satisfied. Now we look at the θ component of the Navier-Stokes equation,

$$\rho\left(\underbrace{\cancel{\frac{\partial u_\theta}{\partial t}}}_{0\text{ (steady)}}+\underbrace{u_r\frac{\partial u_\theta}{\partial r}}_{\frac{C}{r}\left(-\frac{K}{r^2}\right)}+\underbrace{\frac{u_\theta}{r}\cancel{\frac{\partial u_\theta}{\partial \theta}}}_{\frac{K}{r^2}(0)}+\underbrace{\frac{u_r u_\theta}{r}}_{\frac{CK}{r^3}}+\underbrace{\cancel{u_z\frac{\partial u_\theta}{\partial z}}}_{0\text{ (2-D)}}\right)$$
$$=-\frac{1}{r}\frac{\partial P}{\partial \theta}+\underbrace{\cancel{\rho g_\theta}}_{0}+\mu\left(\underbrace{\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_\theta}{\partial r}\right)}_{\frac{K}{r^3}}-\underbrace{\frac{u_\theta}{r^2}}_{\frac{K}{r^3}}+\underbrace{\frac{1}{r^2}\cancel{\frac{\partial^2 u_\theta}{\partial \theta^2}}}_{0}-\underbrace{\frac{2}{r^2}\cancel{\frac{\partial u_r}{\partial \theta}}}_{0}+\underbrace{\cancel{\frac{\partial^2 u_\theta}{\partial z^2}}}_{0\text{ (2-D)}}\right) \quad (1)$$

The θ momentum equation reduces to

θ momentum:
$$\frac{\partial P}{\partial \theta}=0 \quad (2)$$

The θ momentum equation is satisfied provided we can generate a pressure field that satisfies Eq. 2. As a side note, we might have expected Eq. 2 without even working through the algebra, since in this problem the velocity field is independent of angle θ; we expect that pressure does not depend on θ either. In similar fashion the r momentum equation is

$$\rho\left(\underbrace{\cancel{\frac{\partial u_r}{\partial t}}}_{0\text{ (steady)}}+\underbrace{u_r\frac{\partial u_r}{\partial r}}_{\frac{C}{r}\left(\frac{-C}{r^2}\right)}+\underbrace{\frac{u_\theta}{r}\cancel{\frac{\partial u_r}{\partial \theta}}}_{\frac{K}{r^2}(0)}-\underbrace{\frac{{u_\theta}^2}{r}}_{\frac{K^2}{r^3}}+\underbrace{\cancel{u_z\frac{\partial u_r}{\partial z}}}_{0\text{ (2-D)}}\right)$$
$$=-\frac{\partial P}{\partial r}+\underbrace{\cancel{\rho g_r}}_{0}+\mu\left(\underbrace{\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_r}{\partial r}\right)}_{\frac{C}{r^3}}-\underbrace{\frac{u_r}{r^2}}_{\frac{C}{r^3}}+\underbrace{\frac{1}{r^2}\cancel{\frac{\partial^2 u_r}{\partial \theta^2}}}_{0}-\underbrace{\frac{2}{r^2}\cancel{\frac{\partial u_\theta}{\partial \theta}}}_{0}+\underbrace{\cancel{\frac{\partial^2 u_r}{\partial z^2}}}_{0\text{ (2-D)}}\right)$$

which reduces to

r momentum:
$$\frac{\partial P}{\partial r}=\rho\frac{K^2+C^2}{r^3} \quad (3)$$

The r momentum equation is satisfied provided we can generate a pressure field that satisfies Eq. 3.

The pressure field $P(r,\theta)$ must be a smooth function of r and θ. Mathematically, this requires that the order of differentiation (r then θ versus θ then r) should not matter. We therefore check whether this is so by differentiating Eqs. 2 and 3 respectively:

Cross-differentiation: $$\frac{\partial^2 P}{\partial r \partial \theta} = \frac{\partial}{\partial r}\left(\frac{\partial P}{\partial \theta}\right) = 0 \qquad \frac{\partial^2 P}{\partial \theta \partial r} = \frac{\partial}{\partial \theta}\left(\frac{\partial P}{\partial r}\right) = 0 \qquad (4)$$

Equation 4 shows that indeed, P is a smooth function of r and θ. Thus, *we should be able to calculate the pressure field*.

To calculate $P(r,\theta)$, we start with either Eq. 2 or Eq. 3 and integrate. We pick Eq. 2, which we can partially integrate (with respect to θ) to obtain an expression for $P(r,\theta)$,

Pressure field from θ-momentum: $$P(r,\theta) = 0 + g(r) \qquad (5)$$

Note that we added an arbitrary function of the other variable r rather than a constant of integration since this is a partial integration. We then take the partial derivative of Eq. 5 with respect to r to obtain

$$\frac{\partial P}{\partial r} = g'(r) = \rho \frac{K^2 + C^2}{r^3} \qquad (6)$$

where we have equated our result to Eq. 3 for consistency. We can now integrate Eq. 6 to obtain the function $g(r)$:

$$g(r) = -\frac{1}{2}\rho \frac{K^2 + C^2}{r^2} + C_1 \qquad (7)$$

where C_1 is an arbitrary constant of integration. Finally, we plug Eq. 7 into Eq. 5 to obtain our final expression for $P(x,y)$. The result is

Answer: $$\boxed{P(r,\theta) = -\frac{1}{2}\rho \frac{K^2 + C^2}{r^2} + C_1} \qquad (8)$$

Thus the pressure field for this flow decreases like $1/r^2$ as we approach the origin. (The origin itself is a singularity point.) This flow field is a simplistic model of a tornado or hurricane, and the low pressure at the center is the "eye of the storm". We note that this flow field is irrotational, and thus Bernoulli's equation can be used instead to calculate the pressure. If we call the pressure P_∞ far away from the origin ($r \to \infty$), where the local velocity approaches zero, Bernoulli's equation shows that at any distance r from the origin,

Bernoulli equation: $$P + \frac{1}{2}\rho V^2 = P_\infty \qquad P = P_\infty - \frac{1}{2}\rho \frac{K^2 + C^2}{r^2} \qquad (9)$$

Equation 9 agrees with our solution (Eq. 8) from the full Navier-Stokes equation if we set constant C_1 equal to P_∞. A region of rotational flow near the origin would avoid the singularity there, and would yield a more physically realistic model of a real tornado.

Discussion For practice, try to obtain Eq. 8 by starting with Eq. 3 rather than Eq. 2; you should get the same answer.

9-90

Solution For a given velocity field, we are to calculate the pressure field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x-y* plane. **4** Gravity does not act in either the *x* or the *y* direction.

Analysis The flow field must satisfy the steady, two-dimensional, incompressible continuity *and* momentum equations. We check each equation separately; let's consider continuity first:

Continuity:
$$\underbrace{\frac{\partial u}{\partial x}}_{a} + \underbrace{\frac{\partial v}{\partial y}}_{-a} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{0\text{ (2-D)}} = 0$$

Continuity is satisfied. Now we look at the *x component* of the Navier-Stokes equation:

x momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial u}{\partial t}}}_{0\text{ (steady)}} + \underbrace{u\frac{\partial u}{\partial x}}_{(ax+b)a} + \underbrace{v\cancel{\frac{\partial u}{\partial y}}}_{(-ay+cx^2)0} + \underbrace{\cancel{w\frac{\partial u}{\partial z}}}_{0\text{ (2-D)}}\right) = -\frac{\partial P}{\partial x} + \underbrace{\cancel{\rho g_x}}_{0} + \mu\left(\underbrace{\cancel{\frac{\partial^2 u}{\partial x^2}}}_{0} + \underbrace{\cancel{\frac{\partial^2 u}{\partial y^2}}}_{0} + \underbrace{\cancel{\frac{\partial^2 u}{\partial z^2}}}_{0\text{ (2-D)}}\right) \qquad (1)$$

Equation 1 reduces to

x momentum:
$$\frac{\partial P}{\partial x} = \rho\left(-a^2x - ab\right) \qquad (2)$$

The *x* momentum equation is satisfied provided we can generate a pressure field that satisfies Eq. 2. In similar fashion we examine the *y* momentum equation,

$$y\text{ momentum: } \rho\left(\underbrace{\cancel{\frac{\partial v}{\partial t}}}_{0\text{ (steady)}} + \underbrace{u\frac{\partial v}{\partial x}}_{(ax+b)2cx} + \underbrace{v\frac{\partial v}{\partial y}}_{(-ay+cx^2)(-a)} + \underbrace{\cancel{w\frac{\partial v}{\partial z}}}_{0\text{ (2-D)}}\right) = -\frac{\partial P}{\partial y} + \underbrace{\cancel{\rho g_y}}_{0} + \mu\left(\underbrace{\frac{\partial^2 v}{\partial x^2}}_{2c} + \underbrace{\cancel{\frac{\partial^2 v}{\partial y^2}}}_{0} + \underbrace{\cancel{\frac{\partial^2 v}{\partial z^2}}}_{0\text{ (2-D)}}\right)$$

The *y* momentum equation reduces to

y momentum:
$$\frac{\partial P}{\partial y} = \rho\left(-acx^2 - 2bcx - a^2y\right) + 2c\mu \qquad (3)$$

The *y* momentum equation is satisfied provided we can generate a pressure field that satisfies Eq. 3.

In the two-dimensional flow under discussion here, the pressure field $P(x,y)$ must be a smooth function of *x* and *y*. Mathematically, this requires that the order of differentiation (*x* then *y* versus *y* then *x*) should not matter. We check whether this is so by differentiating Eqs. 3 and 2 respectively:

Cross-differentiation:
$$\frac{\partial^2 P}{\partial y \partial x} = \frac{\partial}{\partial y}\left(\frac{\partial P}{\partial x}\right) = 0 \qquad \frac{\partial^2 P}{\partial x \partial y} = \frac{\partial}{\partial x}\left(\frac{\partial P}{\partial y}\right) = \rho\left(-2acx - 2bc\right) \qquad (4)$$

Since the cross-derivative terms in Eq. 4 do not match, *P* is *not* a smooth function of *x* and *y*. Thus, **we are unable to calculate a steady, incompressible, two-dimensional pressure field with the given velocity field**. We cannot proceed any farther – **the pressure cannot be found** with the given velocity field and restrictions.

Discussion This problem shows that if a velocity field satisfies the continuity equation (conservation of mass), this does not necessarily guarantee that the velocity field is physically possible. In the present case, for instance, we are unable to find a pressure field that can satisfy the steady form of the Navier-Stokes equation.

9-91

Solution For a given geometry and set of boundary conditions, we are to calculate the velocity field, and plot the nondimensionalized velocity profile.

Assumptions We number and list the assumptions for clarity:

1. The walls are infinite in the *y*-*z* plane (*y* is into the page).
2. The flow is steady, i.e. time derivatives of any quantity are zero.
3. The flow is parallel (the *x* component of velocity, *u*, is zero everywhere).
4. The fluid is incompressible and Newtonian, and the flow is laminar.
5. Pressure *P* = constant everywhere. In other words, there is no applied pressure gradient pushing the flow; the flow establishes itself due to a balance between gravitational forces and viscous forces.
6. The velocity field is purely two-dimensional, which implies that $v = 0$ and all *y* derivatives are zero.
7. Gravity acts in the negative *z* direction. We can express this mathematically as $\vec{g} = -g\vec{k}$, or $g_x = g_y = 0$ and $g_z = -g$.

Analysis We obtain the velocity and pressure fields by following the step-by-step procedure for differential fluid flow solutions.

Step 1 Set up the problem and the geometry. See problem statement.

Step 2 List assumptions and boundary conditions. We have already listed seven assumptions. The boundary conditions come from the no-slip condition at the walls (1) at $x = -h/2$, $u = v = w = 0$. (2) At $x = h/2$, $u = v = w = 0$.

Step 3 Write out and simplify the differential equations. We start with the continuity equation in Cartesian coordinates,

Continuity:
$$\underbrace{\cancel{\frac{\partial u}{\partial x}}}_{\text{Assumption 3}} + \underbrace{\cancel{\frac{\partial v}{\partial y}}}_{\text{Assumption 6}} + \frac{\partial w}{\partial z} = 0 \qquad \text{or} \qquad \frac{\partial w}{\partial z} = 0 \qquad (1)$$

Equation 1 tells us that *w* is not a function of *z*. In other words, it doesn't matter where we place our origin – the flow is the same at any *z* location. In other words the flow is *fully developed*. Since *w* is not a function of time (Assumption 2), *z* (Eq. 1), or *y* (Assumption 6), we conclude that *w* is at most a function of *x*,

Result of continuity:
$$w = w(x) \text{ only} \qquad (2)$$

We now simplify each component of the Navier-Stokes equation as far as possible. Since $u = v = 0$ everywhere and gravity does not act in the *x* or *y* directions, the *x* and *y* momentum equations are satisfied exactly (in fact all terms are zero in both equations). The *z* momentum equation reduces to

z momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial w}{\partial t}}}_{\text{Assumption 2}} + \underbrace{u\cancel{\frac{\partial w}{\partial x}}}_{\text{Assumption 3}} + \underbrace{v\cancel{\frac{\partial w}{\partial y}}}_{\text{Assumption 6}} + \underbrace{w\cancel{\frac{\partial w}{\partial z}}}_{\text{Continuity}}\right) = \underbrace{-\cancel{\frac{\partial P}{\partial z}}}_{\text{Assumption 5}} + \underbrace{\rho g_z}_{-\rho g}$$
$$+\mu\left(\frac{\partial^2 w}{\partial x^2} + \underbrace{\cancel{\frac{\partial^2 w}{\partial y^2}}}_{\text{Assumption 6}} + \underbrace{\cancel{\frac{\partial^2 w}{\partial z^2}}}_{\text{continuity}}\right) \qquad \text{or} \qquad \frac{d^2 w}{dx^2} = \frac{\rho g}{\mu} \qquad (3)$$

We have changed from a partial derivative (∂/∂*x*) to a total derivative (d/dx) in Eq. 3 as a direct result of Eq. 2, reducing the PDE to an ODE.

Step 4 Solve the differential equations. Continuity and *x* and *y* momentum have already been "solved". Equation 3 (*z* momentum) is integrated twice to get

Integration of z momentum:
$$w = \frac{\rho g}{2\mu}x^2 + C_1 x + C_2 \qquad (4)$$

Step 5 We apply boundary conditions (1) and (2) from Step 2 above to obtain constants C_1 and C_2, *Boundary condition* (1): $$0 = \frac{\rho g}{8\mu} h^2 - C_1 \frac{h}{2} + C_2$$

and

Boundary condition (2): $$0 = \frac{\rho g}{8\mu} h^2 + C_1 \frac{h}{2} + C_2$$

We solve the above two equations simultaneously to obtain expressions for C_1 and C_2,

Constants of integration: $$C_1 = 0 \qquad C_2 = \frac{-\rho g}{8\mu} h^2$$

Finally, Eq. 4 becomes

Final result for velocity field: $$w = \frac{\rho g}{2\mu}\left(x^2 - \left(\frac{h}{2}\right)^2\right) \qquad (5)$$

Since $-h/2 < x < h/2$ everywhere, w is negative everywhere as expected (flow is downward).

Step 6 Verify the results. You can plug in the velocity field to verify that all the differential equations and boundary conditions are satisfied.

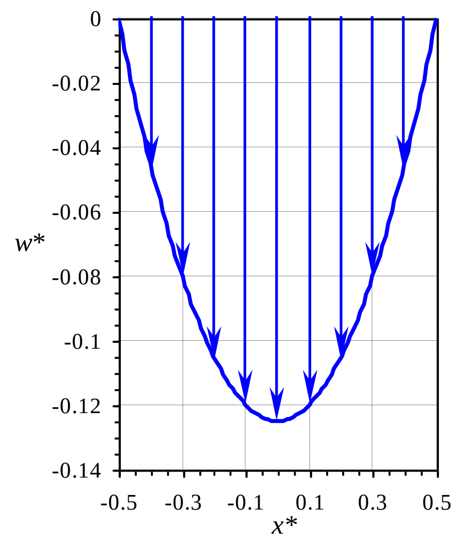


FIGURE 1
The velocity profile for liquid falling between two vertical walls.

We nondimensionalize Eq. 5 by inspection: we let $x^* = x/h$ and $w^* = w\mu/(\rho g h^2)$. Eq. 5 becomes

Nondimensionalized velocity profile: $$\boxed{w^* = \frac{1}{2}\left((x^*)^2 - \frac{1}{4}\right)} \qquad (6)$$

We plot the nondimensional velocity field in Fig. 1. The velocity profile is parabolic.

Discussion Equation 4 for the z component of velocity is identical to that of Example 9-17. In fact, the present problem is identical to Example 9-17 except for the boundary conditions and the location of the origin. Comparing the two results, we see that the maximum nondimensional velocity for the case with two walls is one-fourth than that for the case with only one wall. This is not unexpected – the additional wall leads to more viscous forces that retard the flow.

9-92

Solution We are to calculate and compare the volume flow rate per unit width of fluid falling between two vertical walls and fluid falling along one vertical wall.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The walls are infinitely wide and very long so that all of the parallel flow, fully developed approximations of the previous problem hold.

Analysis We calculate the volume flow rate per unit width by integration of the velocity:

Volume flow rate per unit depth, two vertical walls:

$$\boxed{\frac{\dot{V}}{L} = \int_{-h/2}^{h/2} w\,dx = \int_{-h/2}^{h/2} \left[\frac{\rho g}{2\mu}\left(x^2 - \left(\frac{h}{2}\right)^2\right)\right]dx = \frac{\rho g}{2\mu}\left[\frac{x^3}{3} - \frac{h^2}{4}x\right]_{x=-h/2}^{x=h/2} = \frac{\rho g}{2\mu}\left[\frac{h^3}{24} - \frac{h^3}{8} + \frac{h^3}{24} - \frac{h^3}{8}\right] = \frac{-\rho g h^3}{12\mu}} \qquad (1)$$

The result is negative since we have defined positive volume flow rate upward, since z is upward, but the flow is downward. For the case with only one vertical wall and a free surface, we calculate the vertical component of velocity to be $w = \frac{\rho g x}{2\mu}(x - 2h)$ (see Example 9-17). Thus, we calculate $\dot{V}/L$ for the case of one vertical wall to be

Volume flow rate per unit depth, one vertical wall with a free surface:

$$\boxed{\frac{\dot{V}}{L} = \int_0^h w\,dx = \int_0^h \left[\frac{\rho g x}{2\mu}(x - 2h)\right]dx = \frac{\rho g}{2\mu}\left[\frac{x^3}{3} - x^2 h\right]_{x=0}^{x=h} = \frac{\rho g}{2\mu}\left[\frac{h^3}{3} - h^3 - 0 + 0\right] = \frac{-\rho g h^3}{3\mu}} \qquad (2)$$

Comparing the two cases we see that $\dot{V}/L$ ***for the case of one vertical wall and a free surface is four times greater than the case of two vertical walls with no free surface***. The physical explanation is that with two walls, the fluid is held back by more viscous stresses, leading to a parabolic velocity profile. For the single-wall case the free surface has no shear stress and thus the fluid flows more freely.

Discussion The two flows being compared here are identical except for the boundary conditions. This illustrates the importance of setting proper boundary conditions.

9-93

Solution For a given geometry and set of boundary conditions, we are to calculate the velocity and pressure fields, and plot the nondimensional velocity profile.

Assumptions We number and list the assumptions for clarity:

1 The wall is infinite in the *s*-*y* plane (*y* is out of the page for a right-handed coordinate system).

2 The flow is steady, i.e. $\frac{\partial}{\partial t}(\text{anything}) = 0$.

3 The flow is parallel and fully developed (we assume the normal component of velocity, u_n, is zero, and we assume that the streamwise component of velocity u_s is independent of streamwise coordinate *s*).

4 The fluid is incompressible and Newtonian, and the flow is laminar.

5 Pressure P = constant = P_{atm} at the free surface. In other words, there is no applied pressure gradient pushing the flow; the flow establishes itself due to a balance between gravitational forces and viscous forces along the wall. Atmospheric pressure is constant everywhere since we are neglecting the change of air pressure with elevation.

6 The velocity field is purely two-dimensional, which implies that $v = 0$ and $\frac{\partial}{\partial y}$ (any velocity component) = 0.

7 Gravity acts in the negative *z* direction. We can express this mathematically as $\vec{g} = -g\vec{k}$. In the *s*-*n* plane, $g_s = g\sin\alpha$ and $g_n = -g\cos\alpha$.

Analysis We obtain the velocity and pressure fields by following the step-by-step procedure for differential fluid flow solutions.

Step 1 Set up the problem and the geometry. See Problem statement.

Step 2 List assumptions and boundary conditions. We have already listed seven assumptions. The boundary conditions are (1) No slip at the wall: at $n = 0$, $u_s = v = u_n = 0$. (2) At the free surface ($n = h$) there is no shear, which in this coordinate system at the vertical free surface means $\partial u_s/\partial n = 0$. (3) $P = P_{atm}$ at $n = h$.

Step 3 Write out and simplify the differential equations. We start with the continuity equation in modified Cartesian coordinates, (s,y,n) and (u_s,v,u_n),

Continuity:

$$\frac{\partial u_s}{\partial s} + \underbrace{\cancel{\frac{\partial v}{\partial y}}}_{\text{Assumption 6}} + \underbrace{\cancel{\frac{\partial u_n}{\partial n}}}_{\text{Assumption 3}} = 0 \quad \text{or} \quad \frac{\partial u_s}{\partial s} = 0 \qquad (1)$$

Equation 1 tells us that u_s is not a function of *s*. In other words, it doesn't matter where we place our origin – the flow is the same at any *s* location. This does not tell us anything new; we have already assumed that the flow is fully developed (Assumption 3). Furthermore, since u_s is not a function of time (Assumption 2) or *y* (Assumption 6), we conclude that u_s is at most a function of *n*,

Result of continuity:

$$u_s = u_s(n) \text{ only} \qquad (2)$$

We now simplify each component of the Navier-Stokes equation as far as possible. Since $v = 0$ everywhere and gravity does not act in the *y* direction, the *y* momentum equation is satisfied exactly (in fact all terms are zero). Since $u_n = 0$ everywhere, the only non-zero terms in the *n* momentum equation are the pressure term and the gravity term. The *n* momentum equation reduces to

n momentum:

$$\rho \underbrace{\cancel{\frac{Du_n}{Dt}}}_{\text{Assumption 3}} = -\frac{\partial P}{\partial n} + \underbrace{\rho g_n}_{-\rho g\cos\alpha} + \mu \underbrace{\cancel{\nabla^2 u_n}}_{\text{Assumption 3}} \quad \text{or} \quad \frac{\partial P}{\partial n} = -\rho g\cos\alpha \qquad (3)$$

We integrate Eq. 3 to solve for the pressure,

Pressure:

$$P = -\rho g n\cos\alpha + f(s) \qquad (4)$$

where we have added a function of *s* rather than a simple constant of integration. But from boundary condition (3), at $n = h$, $P = P_{atm}$. Thus Eq. 4 yields $f(s) = P_{atm} + \rho g h\cos\alpha$. In other words, $f(s)$ is really not a function of *s* at all. Equation 4 then becomes

Final expression for pressure:

$$\boxed{P = P_{atm} + \rho g(h-n)\cos\alpha} \qquad (5)$$

The s momentum equation reduces to

$$\rho\left(\underbrace{\cancel{\frac{\partial u_s}{\partial t}}}_{\text{Assumption 2}}+\underbrace{u_s\cancel{\frac{\partial u_s}{\partial s}}}_{\text{Continuity}}+\underbrace{v\cancel{\frac{\partial u_s}{\partial y}}}_{\text{Assumption 6}}+\underbrace{u_n\cancel{\frac{\partial u_s}{\partial n}}}_{\text{Assumption 3}}\right)=-\underbrace{\cancel{\frac{\partial P}{\partial s}}}_{\text{Eq. 5}}+\underbrace{\rho g_s}_{\rho g\sin\alpha} \tag{6}$$

$$+\mu\left(\underbrace{\cancel{\frac{\partial^2 u_s}{\partial s^2}}}_{\text{continuity}}+\underbrace{\cancel{\frac{\partial^2 u_s}{\partial y^2}}}_{\text{Assumption 6}}+\frac{\partial^2 u_s}{\partial n^2}\right) \qquad \text{or} \qquad \frac{d^2u_s}{dn^2}=-\frac{\rho g\sin\alpha}{\mu}$$

We have changed from a partial derivative ($\partial/\partial n$) to a total derivative (d/dn) in Eq. 6 as a direct result of Eq. 2, reducing the PDE to an ODE.

Step 4 Solve the differential equations. Continuity and n and y momentum have already been "solved". Equation 6 (s momentum) is integrated twice to get

$$u_s=-\frac{\rho g\sin\alpha}{2\mu}n^2+C_1n+C_2 \tag{7}$$

Step 5 We apply boundary conditions (1) and (2) from Step 2 above to obtain constants C_1 and C_2,

Boundary condition (1): $u_s=0+0+C_2$ at $n=0$ $\qquad C_2=0$

and

Boundary condition (2):

$$\left.\frac{du_s}{dn}\right)_{n=h}=-\frac{\rho g\sin\alpha}{\mu}h+C_1=0 \qquad C_1=\frac{\rho gh\sin\alpha}{\mu}$$

Finally, Eq. 4 becomes

Final result for velocity field: $$\boxed{u_s=\frac{\rho g\sin\alpha}{2\mu}n\left(2h-n\right)} \tag{8}$$

Since $n < h$ in the film, u_s is positive everywhere as expected (flow is downward).

Step 6 Verify the results. You can plug in the velocity field to verify that all the differential equations and boundary conditions are satisfied.

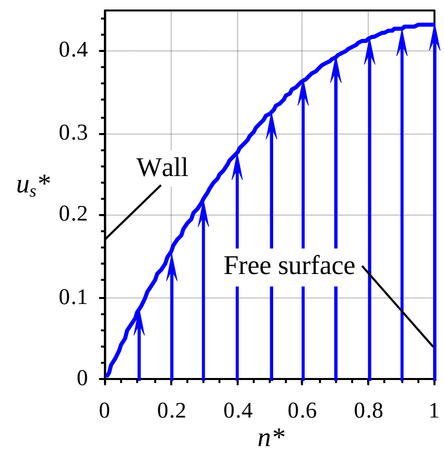


FIGURE 1
The velocity profile for an oil film falling down an inclined wall, $\alpha = 60^{o}$.

When $\alpha = 90^{o}$ $\sin\alpha = 1$ and Eq. 8 is equivalent to Eq. 5 of Example 9-17. (The signs are opposite since s is down while z is up.) Also, Eq. 5 above reduces to $P = P_{atm}$ everywhere when $\alpha = 90^{o}$ since $\cos\alpha = 0$; this also agrees with the results of Example 9-17. We nondimensionalize Eq. 8 by inspection: we let $n^* = n/h$ and $u_s^* = u_s\mu/(\rho gh^2)$. Eq. 8 becomes

Nondimensional velocity profile: $$\boxed{u_s^*=\frac{n^*}{2}\left(2-n^*\right)\sin\alpha} \tag{9}$$

We plot the nondimensional velocity field in Fig. 1 for the case in which $\alpha = 60^{o}$.

Discussion The profile shape is identical to that of Example 9-17, but is scaled by the factor $\sin\alpha$. This problem could also have been solved in standard Cartesian coordinates (x,y,z), but the algebra would be more involved.

9-94

Solution We are to calculate the volume flow rate per unit width of oil falling down a vertical wall.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The wall is infinitely wide and very long so that all of the parallel flow, fully developed approximations of Problem 9-89 hold.

Analysis We calculate the volume flow rate per unit width by integration of the velocity:

Volume flow rate per unit depth:
$$\boxed{\frac{\dot{V}}{L} = \int_0^h u_s dn = \int_0^h \left[\frac{\rho g \sin\alpha}{2\mu} n(2h-n)\right] dn = \frac{\rho g \sin\alpha}{3\mu} h^3} \qquad (1)$$

For an oil film of thickness 5.0 mm with ρ = 888 kg/m^3 and μ = 0.80 kg/(m·s), we calculate $\dot{V}/L$ using Eq. 1,

Result:
$$\frac{\dot{V}}{L} = \frac{\rho g \sin\alpha}{3\mu} h^3 = \frac{\left(888 \text{ kg/m}^3\right)\left(9.81 \text{ m/s}^2\right)\sin\left(60^\circ\right)\left(0.005 \text{ m}\right)^3}{3\left(0.80 \text{ kg/m}\cdot\text{s}\right)} = \mathbf{3.93\times10^{-4}\ m^2/s}$$

Discussion Since viscosity is in the denominator of Eq. 1, a low viscosity liquid (like water) would yield a larger volume flow rate; this agrees with our intuition. Likewise, a larger density liquid and/or a thicker film would yield a larger volume flow rate, again agreeing with our intuition. Finally, if $\alpha = 0^\circ$ there is no flow.

9-95

Solution We are to expand two terms into three terms, and then compress the three terms into one term.

Analysis We use the product rule to differentiate the expression,

$$\boxed{\mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_\theta}{\partial r}\right) - \frac{u_\theta}{r^2}\right) = \mu\left(\frac{\partial^2 u_\theta}{\partial r^2} + \frac{1}{r}\frac{\partial u_\theta}{\partial r} - \frac{u_\theta}{r^2}\right)}$$

The second part of this question involves some trial and error, using the product rule in reverse. After some effort we get

$$\boxed{\mu\left(\frac{\partial^2 u_\theta}{\partial r^2} + \frac{1}{r}\frac{\partial u_\theta}{\partial r} - \frac{u_\theta}{r^2}\right) = \mu\left(\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r u_\theta\right)\right)\right)} \qquad (1)$$

You can apply the product rule to verify Eq. 1.

Discussion The grouping of these terms into one term as in Eq. 1 turns out to be useful for some analytical solutions of the Navier-Stokes equation.

9-96

Solution For a given geometry and set of boundary conditions, we are to calculate the velocity field.

Assumptions We number and list the assumptions for clarity:

1 The cylinders are infinite in the z direction (z is out of the page in the figure of the problem statement for a right-handed coordinate system). The velocity field is purely two-dimensional, which implies that $w = 0$ and derivatives of any velocity component with respect to z are zero.
2 The flow is steady, meaning that all time derivatives are zero.
3 The flow is circular, meaning that the radial velocity component u_r is zero.
4 The flow is rotationally symmetric, meaning that nothing is a function of θ.
5 The fluid is incompressible and Newtonian, and the flow is laminar.
6 Gravitational effects are ignored. (Note that gravity may act in the z direction, leading to an additional hydrostatic pressure distribution in the z direction. This would not affect the present analysis.)

Analysis We obtain the velocity and pressure fields by following the step-by-step procedure for differential fluid flow solutions.

Step 1 Set up the problem and the geometry. See the problem statement.

Step 2 List assumptions and boundary conditions. We have already listed five assumptions. The boundary conditions are (1) No slip at the inner wall: at $r = R_i$, $u_\theta = \omega_i R_i$. (2) No slip at the outer wall: at $r = R_o$, $u_\theta = 0$.

Step 3 Write out and simplify the differential equations. We start with the continuity equation in cylindrical coordinates, (r,θ,z) and (u_r,u_θ,u_z),

Continuity:
$$\underbrace{\cancel{\frac{1}{r}\frac{\partial(ru_r)}{\partial r}}}_{\text{Assumption 3}}+\underbrace{\cancel{\frac{1}{r}\frac{\partial(u_\theta)}{\partial\theta}}}_{\text{Assumption 4}}+\underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{Assumption 1}}=0 \qquad \text{or} \qquad 0=0 \qquad (1)$$

Thus continuity is satisfied exactly by our assumptions.

We now simplify each component of the Navier-Stokes equation as far as possible. Since $w = 0$ everywhere and gravity is ignored, the z momentum equation is satisfied exactly (in fact all terms are zero). Since $u_r = 0$ everywhere, the only non-zero terms in the r momentum equation are the pressure term and the "extra" term that involves u_θ. The r momentum equation reduces to

r momentum:
$$\frac{\partial P}{\partial r}=\rho\frac{u_\theta^{\ 2}}{r} \qquad \text{or} \qquad \frac{dP}{dr}=\rho\frac{u_\theta^{\ 2}}{r} \qquad (2)$$

We have changed the partial derivatives to total derivatives since P is a function only of r. Equation 2 could be used to solve for $P(r)$ once we find u_θ.

The θ momentum equation is written out, using the result of the previous problem,

θ momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial u_\theta}{\partial t}}}_{\text{Assumption 2}}+\underbrace{\cancel{u_r\frac{\partial u_\theta}{\partial r}}}_{\text{Assumption 3}}+\underbrace{\cancel{\frac{u_\theta}{r}\frac{\partial u_\theta}{\partial\theta}}}_{\text{Assumption 4}}+\underbrace{\cancel{\frac{u_r u_\theta}{r}}}_{\text{Assumption 3}}+\underbrace{\cancel{u_z\frac{\partial u_\theta}{\partial z}}}_{\text{Assumption 1}}\right)$$
$$=-\underbrace{\cancel{\frac{1}{r}\frac{\partial P}{\partial\theta}}}_{\text{Assumption 4}}+\underbrace{\cancel{\rho g_\theta}}_{\text{Assumption 6}}+\mu\left(\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(ru_\theta)\right)+\underbrace{\cancel{\frac{1}{r^2}\frac{\partial^2 u_\theta}{\partial\theta^2}}}_{\text{Assumption 4}}-\underbrace{\cancel{\frac{2}{r^2}\frac{\partial u_r}{\partial\theta}}}_{\text{Assumption 3}}+\underbrace{\cancel{\frac{\partial^2 u_\theta}{\partial z^2}}}_{\text{Assumption 6}}\right)$$

Again we change from partial derivatives ($\partial/\partial r$) to a total derivatives (d/dr), reducing the PDE to an ODE. The θ momentum equation reduces to

Reduced θ momentum:
$$\frac{d}{dr}\left(\frac{1}{r}\frac{d}{dr}(ru_\theta)\right)=0 \qquad (3)$$

Step 4 Solve the differential equations. Continuity and z momentum have already been "solved". Equation 3 (θ momentum) is integrated once, *Integration of θ momentum*: $$\frac{1}{r}\frac{d}{dr}\left(ru_\theta\right)=C_1$$

After multiplying by *r* we integrate again. After division by *r* we get

u_θ: $$u_\theta = C_1\frac{r}{2}+\frac{C_2}{r} \qquad (4)$$

Step 5 We apply boundary conditions (1) and (2) from Step 2 above to obtain constants C_1 and C_2,

Boundary condition (2): $$0 = C_1\frac{R_o}{2}+\frac{C_2}{R_o} \quad \text{or} \quad C_2 = -C_1\frac{R_o^{\ 2}}{2}$$

and

Boundary condition (1): $$R_i\omega_i = C_1\frac{R_i}{2}+\frac{C_2}{R_i} = C_1\frac{R_i}{2}-C_1\frac{R_o^{\ 2}}{2R_i}$$

Which can be solved for C_1. The two constants of integration are thus

Constants of integration: $$C_1 = \frac{-2R_i^{\ 2}\omega_i}{R_o^{\ 2}-R_i^{\ 2}} \qquad C_2 = \frac{R_o^{\ 2}R_i^{\ 2}\omega_i}{R_o^{\ 2}-R_i^{\ 2}}$$

Finally, Eq. 4 becomes (after a bit of algebra)

Final result for velocity field: $$\boxed{u_\theta = \frac{R_i^{\ 2}\omega_i}{R_o^{\ 2}-R_i^{\ 2}}\left(\frac{R_o^{\ 2}}{r}-r\right)} \qquad (5)$$

Step 6 Verify the results. You can plug in the velocity field to verify that all the differential equations and boundary conditions are satisfied.

Discussion There are valid alternative forms of Eq. 5. We could integrate Eq. 2 to solve for the pressure since we now know u_θ from Eq. 5. The algebra is laborious, but not difficult.

9-97

Solution For a given geometry and set of boundary conditions, we are to calculate the velocity field.

Assumptions We number and list the assumptions for clarity:

1 The cylinders are infinite in the z direction (z is out of the page in the figure of the problem statement for a right-handed coordinate system). The velocity field is purely two-dimensional, which implies that $w = 0$ and derivatives of any velocity component with respect to z are zero.
2 The flow is steady, meaning that all time derivatives are zero.
3 The flow is circular, meaning that the radial velocity component u_r is zero.
4 The flow is rotationally symmetric, meaning that nothing is a function of θ.
5 The fluid is incompressible and Newtonian, and the flow is laminar.
6 Gravitational effects are ignored. (Note that gravity may act in the z direction, leading to an additional hydrostatic pressure distribution in the z direction. This would not affect the present analysis.)

Analysis We obtain the velocity and pressure fields by following the step-by-step procedure for differential fluid flow solutions.

Step 1 Set up the problem and the geometry. See the problem statement.
Step 2 List assumptions and boundary conditions. We have already listed five assumptions. The boundary conditions are (1) No slip at the inner wall: at $r = R_i$, $u_\theta = 0$. (2) No slip at the outer wall: at $r = R_o$, $u_\theta = \omega_o R_o$.
Step 3 Write out and simplify the differential equations. We start with the continuity equation in cylindrical coordinates, (r,θ,z) and (u_r,u_θ,u_z),

Continuity:
$$\underbrace{\cancel{\frac{1}{r}\frac{\partial(ru_r)}{\partial r}}}_{\text{Assumption 3}} + \underbrace{\cancel{\frac{1}{r}\frac{\partial(u_\theta)}{\partial \theta}}}_{\text{Assumption 4}} + \underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{Assumption 1}} = 0 \qquad \text{or} \qquad 0 = 0 \qquad (1)$$

Thus continuity is satisfied exactly by our assumptions.

We now simplify each component of the Navier-Stokes equation as far as possible. Since $w = 0$ everywhere and gravity is ignored, the z momentum equation is satisfied exactly (in fact all terms are zero). Since $u_r = 0$ everywhere, the only non-zero terms in the r momentum equation are the pressure term and the "extra" term that involves u_θ. The r momentum equation reduces to

r momentum:
$$\frac{\partial P}{\partial r} = \rho\frac{u_\theta^{\,2}}{r} \qquad \text{or} \qquad \frac{dP}{dr} = \rho\frac{u_\theta^{\,2}}{r} \qquad (2)$$

We have changed the partial derivatives to total derivatives since P is a function only of r. Equation 2 could be used to solve for $P(r)$ once we find u_θ.

The θ momentum equation is written out, using the result of the previous problem,

θ momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial u_\theta}{\partial t}}}_{\text{Assumption 2}} + \underbrace{\cancel{u_r\frac{\partial u_\theta}{\partial r}}}_{\text{Assumption 3}} + \underbrace{\cancel{\frac{u_\theta}{r}\frac{\partial u_\theta}{\partial \theta}}}_{\text{Assumption 4}} + \underbrace{\cancel{\frac{u_r u_\theta}{r}}}_{\text{Assumption 3}} + \underbrace{\cancel{u_z\frac{\partial u_\theta}{\partial z}}}_{\text{Assumption 1}}\right)$$
$$= -\underbrace{\cancel{\frac{1}{r}\frac{\partial P}{\partial \theta}}}_{\text{Assumption 4}} + \underbrace{\cancel{\rho g_\theta}}_{\text{Assumption 6}} + \mu\left(\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(ru_\theta)\right) + \underbrace{\cancel{\frac{1}{r^2}\frac{\partial^2 u_\theta}{\partial \theta^2}}}_{\text{Assumption 4}} - \underbrace{\cancel{\frac{2}{r^2}\frac{\partial u_r}{\partial \theta}}}_{\text{Assumption 3}} + \underbrace{\cancel{\frac{\partial^2 u_\theta}{\partial z^2}}}_{\text{Assumption 6}}\right)$$

Again we change from partial derivatives ($\partial/\partial r$) to a total derivatives (d/dr), reducing the PDE to an ODE. The θ momentum equation reduces to

Reduced θ momentum:
$$\frac{d}{dr}\left(\frac{1}{r}\frac{d}{dr}(ru_\theta)\right) = 0 \qquad (3)$$

Step 4 Solve the differential equations. Continuity and z momentum have already been "solved". Equation 3 (θ momentum) is integrated once, *Integration of θ momentum*: $$\frac{1}{r}\frac{d}{dr}\left(ru_\theta\right) = C_1$$

After multiplying by r we integrate again. After division by r we get

u_θ: $$u_\theta = C_1\frac{r}{2} + \frac{C_2}{r} \qquad (4)$$

Step 5 We apply boundary conditions (1) and (2) from Step 2 above to obtain constants C_1 and C_2,

Boundary condition (1): $$0 = C_1\frac{R_i}{2} + \frac{C_2}{R_i} \quad \text{or} \quad C_2 = -C_1\frac{R_i^2}{2}$$

and

Boundary condition (2): $$R_o\omega_o = C_1\frac{R_o}{2} + \frac{C_2}{R_o} = C_1\frac{R_o}{2} - C_1\frac{R_i^2}{2R_o}$$

Which can be solved for C_1. The two constants of integration are thus

Constants of integration: $$C_1 = \frac{2R_o^2\omega_o}{R_o^2 - R_i^2} \qquad C_2 = \frac{-R_o^2R_i^2\omega_o}{R_o^2 - R_i^2}$$

Finally, Eq. 4 becomes (after a bit of algebra)

Final result for velocity field: $$\boxed{u_\theta = \frac{R_o^2\omega_o}{R_o^2 - R_i^2}\left(r - \frac{R_i^2}{r}\right)} \qquad (5)$$

Step 6 Verify the results. You can plug in the velocity field to verify that all the differential equations and boundary conditions are satisfied.

Discussion There are valid alternative forms of Eq. 5. We could integrate Eq. 2 to solve for the pressure since we now know u_θ from Eq. 5. The algebra is laborious, but not difficult.

9-98

Solution We are to simplify the velocity field for two limiting cases of Problem 9-96 and discuss.

Assumptions The same assumptions of Problem 9-96 apply here.

Analysis (*a*) First we re-write the velocity profile from Problem 9-92,

Exact velocity profile:

$$u_\theta = \frac{R_i^2\omega_i}{R_o^2 - R_i^2}\left(\frac{R_o^2}{r} - r\right) = \frac{R_i^2\omega_i}{(R_o - R_i)(R_o + R_i)}\left(\frac{(R_o - r)(R_o + r)}{r}\right) \qquad (1)$$

Note that Eq. 1 is still exact. When the gap is very small, (R_o – R_i) << R_o, and $R_o \approx R_i$. Thus we replace R_o + R_i in the denominator of Eq. 1 by $2R_i$. Similarly, $r \approx R_i$ and we replace $R_o + r$ in the numerator of Eq. 1 by $2R_i$. Likewise we replace r in the denominator of Eq. 1 by R_i. As suggested we define $y = R_o - r$, h = gap thickness = $R_o - R_i$, and V = speed of the "upper plate" = $R_i\omega_i$ (Fig. 1). Plugging all of these approximations and definitions into Eq. 1 we get

Approximate velocity for small gap:

$$\boxed{u_\theta \approx \frac{R_i^2\omega_i}{h\cdot 2R_i}\left(\frac{y\cdot 2R_i}{R_i}\right) = \frac{y\omega_i R_i}{h} = V\frac{y}{h}} \qquad (2)$$

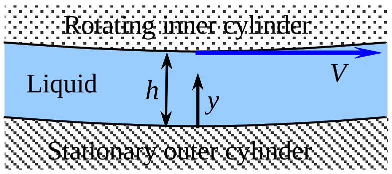


FIGURE 1
A magnified view near the bottom for the case in which the gap between the two cylinders is very small.

We verify that Eq. 2 is linear in the small gap and is the same velocity profile as we generated for 2-D Couette flow between two infinite flat plates.

(*b*) As the outer cylinder radius approaches infinity, R_i << R_o, and R_i can be ignored when added to or subtracted from R_o. Similarly, r << R_o, and r can be ignored when added to or subtracted from R_o. Equation 1 becomes

Approximate velocity for infinite R_o:

$$\boxed{u_\theta \approx \frac{R_i^2\omega_i}{(R_o)(R_o)}\left(\frac{(R_o)(R_o)}{r}\right) = \frac{R_i^2\omega_i}{r}} \qquad (3)$$

We recognize Eq. 3 as of the form u_θ = constant/r which is the velocity field for a **line vortex**.

Discussion Imagine a long, thin cylinder spinning in a vat of liquid. After a long time, the flow field given by Eq. 3 would emerge – basically a line vortex for all radii greater than R_i.

9-99

Solution For a given geometry and set of boundary conditions, we are to calculate the velocity field.

Assumptions The assumptions are identical to those of Problem 9-96.

Analysis We obtain the velocity and pressure fields by following the step-by-step procedure for differential fluid flow solutions. Everything is identical to Problem 9-96 except for the boundary condition at the outer cylinder wall. We re-write boundary condition (2): at $r = R_o$, $u_\theta = \omega_o R_o$. We will not repeat all the algebra associated with the equations of motion. The tangential velocity component is still

u_θ:
$$u_\theta = C_1 \frac{r}{2} + \frac{C_2}{r} \tag{1}$$

Now we apply boundary conditions (1) and (2) to obtain constants C_1 and C_2,

Boundary condition (1):
$$\frac{R_i}{2} C_1 + \frac{1}{R_i} C_2 = R_i \omega_i \tag{2}$$

Boundary condition (2):
$$\frac{R_o}{2} C_1 + \frac{1}{R_o} C_2 = R_o \omega_o \tag{3}$$

We solve Eqs. 2 and 3 simultaneously for C_1 and C_2. The result is

Constants of integration:
$$C_1 = \frac{2\left(R_o^{\,2}\omega_o - R_i^{\,2}\omega_i\right)}{R_o^{\,2} - R_i^{\,2}} \qquad C_2 = \frac{R_o^{\,2} R_i^{\,2}\left(\omega_i - \omega_o\right)}{R_o^{\,2} - R_i^{\,2}} \tag{4}$$

Finally, Eq. 4 becomes (after a bit of algebra)

Final result for velocity field:
$$\boxed{u_\theta = \frac{1}{R_o^{\,2} - R_i^{\,2}}\left[\left(R_o^{\,2}\omega_o - R_i^{\,2}\omega_i\right) r + \frac{R_o^{\,2} R_i^{\,2}\left(\omega_i - \omega_o\right)}{r}\right]} \tag{5}$$

We set $\omega_o = 0$ in Eq. 5 to verify that it simplifies to the result of Problem 9-92,

Simplified velocity field:
$$\boxed{u_\theta = \frac{R_i^{\,2}\omega_i}{R_o^{\,2} - R_i^{\,2}}\left(\frac{R_o^{\,2}}{r} - r\right)} \tag{6}$$

Discussion There are valid alternative forms of Eq. 5.

9-100

Solution We are to discuss a simplified version of the velocity field of the previous problem.

Assumptions The assumptions are identical to those of the previous problem.

Analysis We set $R_i = \omega_i = 0$ in Eq. 5 of the previous problem. The tangential velocity component simplifies to

Simplified u_θ:

$$u_\theta = \frac{1}{R_o^{\,2}}\left[R_o^{\,2}\omega_o r\right] = \omega_o r \qquad (1)$$

We recognize Eq. 1 as the velocity field for *solid body rotation*. To set up this velocity field in a physical experiment, we would place a cylindrical container of liquid on a rotating table. After a long time, the entire tank, including the liquid, would be in solid body rotation.

Discussion If you imagine flow between the inner and outer cylinders, and then imagine that the inner cylinder stops spinning and shrinks to infinitesimal radius, you can convince yourself that solid body rotation would result.

9-101

Solution For flow in a pipe annulus we are to calculate the velocity field.

Assumptions We number and list the assumptions for clarity:

1 The pipe is infinitely long in the x direction.

2 The flow is steady, i.e. $\frac{\partial}{\partial t}(\text{anything}) = 0$.

3 This is a parallel flow (the r component of velocity, u_r, is zero).

4 The fluid is incompressible and Newtonian, and the flow is laminar.

5 A constant pressure gradient is applied in the x direction such that pressure changes linearly with respect to x according to the given expression.

6 The velocity field is axisymmetric with no swirl, implying that $u_\theta = 0$ and $\frac{\partial}{\partial \theta}(\text{anything}) = 0$.

7 We ignore the effects of gravity.

Analysis We obtain the velocity field by following the step-by-step procedure for differential fluid flow solutions.

Step 1 Lay out the problem and the geometry. See the problem statement.

Step 2 List assumptions and boundary conditions. We have already listed seven assumptions. The boundary conditions come from imposing the no slip condition at both pipe walls: (1) at $r = R_i$, $\vec{V} = 0$. (2) at $r = R_o$, $\vec{V} = 0$.

Step 3 Write out and simplify the differential equations. We start with the continuity equation in cylindrical coordinates,

Continuity:
$$\underbrace{\cancel{\frac{1}{r}\frac{\partial(r u_r)}{\partial r}}}_{\text{Assumption 3}} + \underbrace{\cancel{\frac{1}{r}\frac{\partial(u_\theta)}{\partial \theta}}}_{\text{Assumption 6}} + \frac{\partial u}{\partial x} = 0 \qquad \text{or} \qquad \frac{\partial u}{\partial x} = 0 \tag{1}$$

Equation 1 tells us that u is not a function of x. In other words, it doesn't matter where we place our origin – the flow is the same at any x location. This can also be inferred directly from Assumption 1, which tells us that there is nothing special about any x location since the pipe is infinite in length – the flow is fully developed. Furthermore, since u is not a function of time (Assumption 2) or θ (Assumption 6), we conclude that u is at most a function of r,

Result of continuity:
$$u = u(r) \text{ only} \tag{2}$$

Next we simplify the x momentum equation as far as possible:

x momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial u}{\partial t}}}_{\text{Assumption 2}} + \underbrace{\cancel{u_r\frac{\partial u}{\partial r}}}_{\text{Assumption 3}} + \underbrace{\cancel{\frac{u_\theta}{r}\frac{\partial u}{\partial \theta}}}_{\text{Assumption 6}} + \underbrace{\cancel{u\frac{\partial u}{\partial x}}}_{\text{Continuity}}\right) = -\frac{\partial P}{\partial x} + \underbrace{\cancel{\rho g_x}}_{\text{Assumption 7}} + \mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u}{\partial r}\right) + \underbrace{\cancel{\frac{1}{r^2}\frac{\partial^2 u}{\partial \theta^2}}}_{\text{Assumption 6}} + \underbrace{\cancel{\frac{\partial^2 u}{\partial x^2}}}_{\text{Continuity}}\right) \tag{3}$$

or

Result of x momentum:
$$\frac{1}{r}\frac{d}{dr}\left(r\frac{du}{dr}\right) = \frac{1}{\mu}\frac{\partial P}{\partial x} \tag{4}$$

Note that we have replaced the partial derivative operators for the u derivatives with total derivative operators because of Eq. 2. Every term in the r momentum equation is zero except the pressure gradient term, forcing that lone term to also be zero,

r momentum:
$$\frac{\partial P}{\partial r} = 0 \tag{5}$$

In other words, P is not a function of r. Since P is also not a function of time (Assumption 2) or θ (Assumption 6), P can be at most a function of x, *Result of r momentum*: $$P = P(x) \text{ only} \qquad (6)$$

Therefore we can replace the partial derivative operator for the pressure gradient in Eq. 4 by the total derivative operator since P varies only with x. Finally, all terms of the θ component of the Navier-Stokes equation go to zero.

Step 4 Solve the differential equations. Continuity and r momentum have already been "solved", resulting in Eqs. 2 and 6 respectively. The θ momentum equation has vanished, and thus we are left with Eq. 4 (x momentum). After multiplying both sides by r, we integrate once to obtain

Integration of x momentum: $$r\frac{du}{dr} = \frac{r^2}{2\mu}\frac{dP}{dx} + C_1 \qquad (7)$$

where C_1 is a constant of integration. Note that the pressure gradient dP/dx is a constant here. After dividing both sides of Eq. 7 by r, we can integrate a second time to get

Second integration of x momentum: $$u = \frac{r^2}{4\mu}\frac{dP}{dx} + C_1 \ln r + C_2 \qquad (8)$$

where C_2 is a second constant of integration.

Step 5 Apply boundary conditions from Step 2 above to obtain constants C_1 and C_2:

Boundary condition (1): $$0 = \frac{R_i^2}{4\mu}\frac{dP}{dx} + C_1 \ln R_i + C_2$$

Boundary condition (2): $$0 = \frac{R_o^2}{4\mu}\frac{dP}{dx} + C_1 \ln R_o + C_2$$

We solve the above two equations simultaneously to find C_1 and C_2,

Constants: $$C_1 = -\frac{\left(R_o^2 - R_i^2\right)}{4\mu \ln \frac{R_o}{R_i}}\frac{dP}{dx} \qquad C_2 = \frac{\left(R_o^2 \ln R_i - R_i^2 \ln R_o\right)}{4\mu \ln \frac{R_o}{R_i}}\frac{dP}{dx}$$

After some algebra and rearrangement, Eq. 7 becomes

Final result for axial velocity: $$\boxed{u = \frac{1}{4\mu}\frac{dP}{dx}\left(r^2 + \frac{R_i^2 \ln \frac{r}{R_o} - R_o^2 \ln \frac{r}{R_i}}{\ln \frac{R_o}{R_i}}\right)} \qquad (9)$$

Step 6 Verify the results. You can plug in the velocity field to verify that all the differential equations and boundary conditions are satisfied.

Discussion There are other valid forms of Eq. 9. For example, after some rearrangement, Eq. 9 can be written as

Alternative form: $$\boxed{u = \frac{1}{4\mu}\frac{dP}{dx}\left(r^2 - R_o^2 - \frac{R_o^2 - R_i^2}{\ln \frac{R_o}{R_i}} \ln \frac{r}{R_o}\right)} \qquad (10)$$

9-102

Solution We are to generate the velocity field for a given flow setup.

Assumptions All assumptions are the same as those of the previous problem except for the fifth one, which we modify here: **5** Pressure P is constant everywhere.

Analysis Most of the algebra is identical to that of the previous problem except that the pressure gradient is zero, making this problem easier. Also, the first boundary condition changes: at $r = R_i$, $u = V$. The x momentum equation reduces to

Result of x momentum:
$$\frac{1}{r}\frac{d}{dr}\left(r\frac{du}{dr}\right) = 0 \qquad (1)$$

After integration, division by r, and a second integration, Eq. 1 yields

x component of velocity:
$$u = C_1 \ln r + C_2 \qquad (2)$$

We apply boundary conditions:

Boundary condition (1):
$$V = C_1 \ln R_i + C_2$$

and

Boundary condition (2):
$$0 = C_1 \ln R_o + C_2$$

We solve the above two equations simultaneously to yield the constants,

Constants of integration:
$$C_1 = \frac{-V}{\ln\dfrac{R_o}{R_i}} \qquad C_2 = \frac{V \ln R_o}{\ln\dfrac{R_o}{R_i}} \qquad (3)$$

and thus Eq. 2 becomes

Result for u:
$$\boxed{u = \frac{V}{\ln\dfrac{R_o}{R_i}}\left(\ln R_o - \ln r\right) = \frac{V \ln\dfrac{R_o}{r}}{\ln\dfrac{R_o}{R_i}}} \qquad (4)$$

Discussion In this and other parallel flow problems, the nonlinear terms in the Navier-Stokes equation drop out, simplifying the problem and enabling an exact analytical solution to be found.

9-103

Solution We are to generate the velocity field for a given flow setup.

Assumptions All assumptions are the same as those of the previous problem.

Analysis The algebra is identical to that of the previous problem except that the boundary conditions are swapped: at $r = R_i$, $u = 0$ and at $r = R_o$, $u = V$. The x momentum equation reduces to

Result of x momentum:
$$\frac{1}{r}\frac{d}{dr}\left(r\frac{du}{dr}\right)=0 \qquad (1)$$

After integration, division by r, and a second integration, Eq. 1 yields

x component of velocity:
$$u = C_1 \ln r + C_2 \qquad (2)$$

We apply boundary conditions:

Boundary condition (1):
$$V = C_1 \ln R_o + C_2$$

and

Boundary condition (2):
$$0 = C_1 \ln R_i + C_2$$

We solve the above two equations simultaneously to yield the constants,

Constants of integration:
$$C_1 = \frac{-V}{\ln\frac{R_i}{R_o}} \qquad C_2 = \frac{V\ln R_i}{\ln\frac{R_i}{R_o}} \qquad (3)$$

and thus Eq. 2 becomes

Result for u:
$$\boxed{u = \frac{V}{\ln\frac{R_i}{R_o}}\left(\ln R_i - \ln r\right) = \frac{V\ln\frac{R_i}{r}}{\ln\frac{R_i}{R_o}}} \qquad (4)$$

Discussion Since the boundary conditions of the present problem are the same as those of the previous problem except that R_o and R_i are swapped, it turns out that the result is also identical except that the two radii are swapped.

9-104

Solution For modified Couette flow with two immiscible fluids we are to list the boundary conditions and then solve for both the velocity and pressure fields. Finally we are to plot the velocity profile across the channel.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Properties The density and viscosity of water at T = 80°C are 971.8 kg/m^3 and 0.355 × 10^{-3} kg/(m·s) respectively. The density and viscosity of unused engine oil at T = 80°C are 852 kg/m^3 and 32.0 × 10^{-3} kg/(m·s) respectively.

Analysis (*a*) The velocity boundary conditions come from the no-slip condition at the walls:

Boundary condition (1): At $z = 0$, $u_1 = 0$ (1)

and

Boundary condition (2): At $z = h_1 + h_2$, $u_2 = V$ (2)

At the interface we know that both the velocities and the shear stresses must match,

Boundary condition (3): At $z = h_1$, $u_1 = u_2$ (3)

and

Boundary condition (4): At $z = h_1$, $\mu_1 \dfrac{du_1}{dz} = \mu_2 \dfrac{du_2}{dz}$ (4)

The first pressure boundary condition comes from the known pressure on the bottom,

Boundary condition (5): At $z = 0$, $P = P_0$ (5)

The second pressure boundary condition comes from the fact that the pressure cannot have a discontinuity at the interface since we are ignoring surface tension,

Boundary condition (6): At $z = h_1$, $P_1 = P_2$ (6)

(*b*) We solve for the velocity field using the step-by-step procedure outlined in this chapter. However, we leave out the details because the algebra is identical to that of simple Couette flow – the only difference is in the boundary conditions. For parallel, fully developed flow in the *x* direction, *u* is the only non-zero velocity component and it is a function of *z* only. The *x* momentum equations in the two fluids reduce to

x momentum:
$$\frac{d^2u_1}{dz^2} = 0 \qquad \frac{d^2u_2}{dz^2} = 0 \tag{7}$$

We integrate both parts of Eq. 7 twice, introducing four constants of integration,

Expressions for u:
$$u_1 = C_1 z + C_2 \qquad u_2 = C_3 z + C_4 \tag{8}$$

We apply the first four boundary conditions to find these constants,

Boundary conditions (1) and (2):
$$C_2 = 0 \qquad V = C_3\left(h_1 + h_2\right) + C_4$$

and

Boundary conditions (3) and (4):
$$C_1 h_1 = C_3 h_1 + C_4 \qquad \mu_1 C_1 = \mu_2 C_3$$

After some algebra, we solve simultaneously for all the constants,

$$C_1 = \frac{\mu_2 V}{\mu_2 h_1 + \mu_1 h_2} \qquad C_2 = 0 \qquad C_3 = \frac{\mu_1 V}{\mu_2 h_1 + \mu_1 h_2} \qquad C_4 = V\left(\frac{\mu_2 h_1 - \mu_1 h_1}{\mu_2 h_1 + \mu_1 h_2}\right) \tag{9}$$

And the velocity components of Eq. 8 become

$$\boxed{u_1 = \frac{\mu_2 V}{\mu_2 h_1 + \mu_1 h_2} z} \tag{10}$$

and

$$\boxed{u_2 = \frac{\mu_1 V}{\mu_2 h_1 + \mu_1 h_2} z + V\left(\frac{\mu_2 h_1 - \mu_1 h_1}{\mu_2 h_1 + \mu_1 h_2}\right) = \frac{V}{\mu_2 h_1 + \mu_1 h_2}\left(\mu_1 (z - h_1) + \mu_2 h_1\right)} \qquad (11)$$

You should plug in the boundary conditions to verify that Eqs. 10 and 11 are correct.

(*c*) We analyze the *z* momentum equation to find the pressure. Since w = 0 everywhere, the only non-zero terms are the pressure and gravity terms. Thus we have

z momentum: $$\frac{dP_1}{dz} = -\rho_1 g \qquad \frac{dP_2}{dz} = -\rho_2 g \qquad (12)$$

We integrate Eqs. 12 to obtain

Pressure: $$P_1 = -\rho_1 g z + C_5 \qquad P_2 = -\rho_2 g z + C_6 \qquad (13)$$

After applying boundary conditions (5) and (6) we obtain the final expressions for the two pressures,

$$\boxed{P_1 = P_0 - \rho_1 g z} \text{ and } \boxed{P_2 = P_0 + (\rho_1 + \rho_2) g h_1 - \rho_2 g z} \qquad (14)$$

Again you can verify that the boundary conditions are satisfied by Eq. 14.

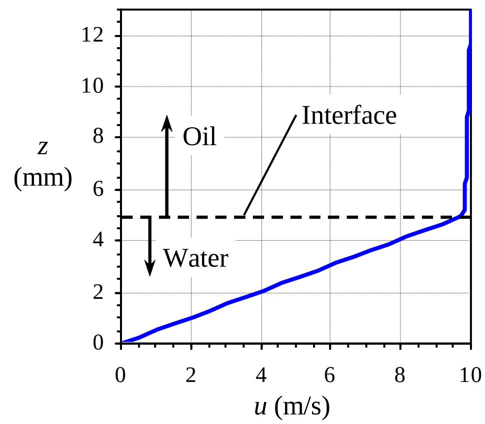


FIGURE 1
The velocity profile for Couette flow with two immiscible liquids.

(*d*) For the given fluid properties we plot the velocity profile in Fig. 1. Since the oil is so much more viscous than the water, the oil velocity is nearly constant (small slope) while the water velocity varies rapidly (large slope). At the interface the viscosity times the slope must match, so this should not be surprising.

Discussion Both velocity profiles are linear. The pressure is simply hydrostatic since *P* is a function of *z* only. The oil must be on top since it is less dense than water.

9-105

Solution We are to calculate $u(r)$ for flow inside an inclined round pipe.

Assumptions We number and list the assumptions for clarity:

1 The pipe is infinitely long in the x direction.
2 The flow is steady, i.e. any time derivative is zero.
3 This is a parallel flow (the r component of velocity, u_r, is zero).
4 The fluid is incompressible and Newtonian, and the flow is laminar.
5 The pressure is constant everywhere except for hydrostatic pressure.
6 The velocity field is axisymmetric with no swirl, implying that $u_\theta = 0$ and all derivatives with respect to θ are zero.

Analysis To obtain the velocity and pressure fields, we follow the step-by-step procedure outlined above.

Step 1 Lay out the problem and the geometry. See the problem statement.

Step 2 List assumptions and boundary conditions. We have already listed six assumptions. The first boundary condition comes from imposing the no slip condition at the pipe wall: (1) at $r = R$, $\vec{V} = 0$. The second boundary condition comes from the fact that the centerline of the pipe is an axis of symmetry: (2) at $r = 0$, $du/dr = 0$.

Step 3 Write out and simplify the differential equations. We start with the continuity equation in cylindrical coordinates, a modified version of Eq. 9-62a,

Continuity:
$$\underbrace{\cancel{\frac{1}{r}\frac{\partial(ru_r)}{\partial r}}}_{\text{Assumption 3}} + \underbrace{\cancel{\frac{1}{r}\frac{\partial(u_\theta)}{\partial \theta}}}_{\text{Assumption 6}} + \frac{\partial u}{\partial x} = 0 \qquad \text{or} \qquad \frac{\partial u}{\partial x} = 0 \qquad (1)$$

Equation 1 tells us that u is not a function of x. In other words, it doesn't matter where we place our origin – the flow is the same at any x location. This can also be inferred directly from Assumption 1, which tells us that there is nothing special about any x location since the pipe is infinite in length – the flow is fully developed. Furthermore, since u is not a function of time (Assumption 2) or θ (Assumption 6), we conclude that u is at most a function of r,

Result of continuity:
$$u = u(r) \text{ only} \qquad (2)$$

We now simplify the x momentum equation as far as possible:

x momentum:
$$\rho\left(\underbrace{\cancel{\frac{\partial u}{\partial t}}}_{\text{Assumption 2}} + \underbrace{\cancel{u_r\frac{\partial u}{\partial r}}}_{\text{Assumption 3}} + \underbrace{\cancel{\frac{u_\theta}{r}\frac{\partial u}{\partial \theta}}}_{\text{Assumption 6}} + \underbrace{\cancel{u\frac{\partial u}{\partial x}}}_{\text{Continuity}}\right)$$
$$= -\underbrace{\cancel{\frac{\partial P}{\partial x}}}_{\text{Assumption 5}} + \underbrace{\rho g_x}_{\rho g \sin\alpha} + \mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u}{\partial r}\right) + \underbrace{\cancel{\frac{1}{r^2}\frac{\partial^2 u}{\partial \theta^2}}}_{\text{Assumption 6}} + \underbrace{\cancel{\frac{\partial^2 u}{\partial x^2}}}_{\text{Continuity}}\right)$$

or

Result of x momentum:
$$\frac{1}{r}\frac{d}{dr}\left(r\frac{du}{dr}\right) = \frac{-\rho g \sin\alpha}{\mu} \qquad (3)$$

As in previous examples the material acceleration (entire left hand side of the x momentum equation) is zero, implying that fluid particles are not accelerating at all in this flow field, and linearizing the Navier-Stokes equation. Also notice that we have replaced the partial derivative operators for the u derivatives with total derivative operators because of Eq. 2.

You can show in similar fashion that every term in the r momentum equation and in the θ momentum equation goes to zero.

Step 4 Solve the differential equations. Continuity, r momentum, and θ momentum have already been solved, and thus we are left with Eq. 3 (x momentum). After multiplying both sides by r, integrating, dividing by r, and integrating again, *Axial velocity component*: $$u = \frac{-\rho g \sin\alpha}{4\mu} r^2 + C_1 \ln r + C_2 \quad (4)$$

where C_1 and C_2 are constants of integration.

Step 5 Apply boundary conditions from Step 2 above to obtain constants C_1 and C_2. We apply boundary condition (2) first:

Boundary condition (2): $$\frac{du}{dr} = 0 + \frac{C_1}{0} = 0$$

Since $C_1/0$ is undefined (∞), the only way for du/dr to equal zero at $r = 0$ is for C_1 to equal 0. An alternative way to think of this boundary condition is to say that u must remain finite at the centerline of the pipe. Again this is possible only if constant C_1 is equal to 0.

$$C_1 = 0$$

Now we apply the first boundary condition,

Boundary condition (1):
$$u = \frac{-\rho g \sin\alpha}{4\mu} R^2 + 0 + C_2 = 0 \qquad \text{or} \qquad C_2 = \frac{\rho g \sin\alpha}{4\mu} R^2$$

Finally, Eq. 4 becomes

Final result for axial velocity: $$\boxed{u = \frac{\rho g \sin\alpha}{4\mu}\left(R^2 - r^2\right)} \quad (5)$$

The axial velocity profile is thus in the shape of a paraboloid, just as in Example 9-18.

Step 6 Verify the results. You can plug in the velocity field to verify that all the differential equations and boundary conditions are satisfied.

The volume flow rate through the pipe is found by integrating Eq. 5 through the whole cross-sectional area of the pipe,

Volume flow rate:
$$\boxed{\dot{V} = \int_{\theta=0}^{2\pi}\int_{r=0}^{R} u\,dr = \frac{2\pi\rho g \sin\alpha}{4\mu}\int_{r=0}^{R}\left(R^2 - r^2\right)r\,dr = \frac{\pi R^4}{8\mu}\rho g \sin\alpha} \quad (6)$$

Since volume flow rate is also equal to the average axial velocity times cross-sectional area, we can easily determine the average axial velocity, V:

Average axial velocity: $$\boxed{V = \frac{\dot{V}}{A} = \frac{\frac{\pi R^4}{8\mu}\rho g \sin\alpha}{\pi R^2} = \frac{R^2}{8\mu}\rho g \sin\alpha} \quad (7)$$

Discussion There is no such thing as an "inviscid" fluid. For example, if μ were zero in this problem, the axial velocity, volume flow rate, and average velocity would all go to infinity since μ appears in the denominator of Eqs. 5 through 7.

9-106

Solution We are to generate and discuss velocity and pressure boundary conditions for the given flow problem.

Assumptions **1** The flow is steady in the mean. **2** Surface tension effects are negligibly small.

Analysis On all tank walls, $\vec{V} = 0$ since the tank walls are stationary (no-slip boundary condition). Mathematically, we write $\boldsymbol{u_r = u_\theta = u_z = 0}$ **at** $\boldsymbol{r = R_{\text{tank}}}$ (the tank side walls) and at $z = 0$ (the bottom wall of the tank). On the blade surfaces, the fluid velocity must equal that of the blades (also the no-slip condition). At any radial location r the velocity of the blade surface is $\vec{V}_{\text{blade}} = r\omega\vec{e}_\theta$. In other words $\boldsymbol{u_\theta = r\omega}$ **at the blade surfaces**. Since the blades do not move at all in the radial or vertical directions, $\boldsymbol{u_r = u_z = 0}$ **along the blade surfaces**. Finally, **at the free surface** $\boldsymbol{P = P_{\text{atm}}}$ since the free surface is exposed to atmospheric air. In addition, the vertical component of velocity $\boldsymbol{u_z}$ **must equal zero at the free surface**. We note that the other two velocity components (u_r and u_θ) may be non-zero at the free surface, but the shear stress in the horizontal plane of the free surface must be zero (negligible shear due to the air). Mathematically, $\boldsymbol{\partial u_r/\partial z = \partial u_\theta/\partial z = 0}$ **at the free surface**.

Discussion The no-slip condition requires that $u_\theta = r\omega$ everywhere on the blade surface, regardless of the geometry of the blades.

9-107

Solution We are to generate and discuss velocity and pressure boundary conditions for the stirrer flow problem from a rotating frame of reference.

Assumptions **1** The flow is steady in the mean. **2** Surface tension effects are negligibly small.

Analysis On all tank walls, $\vec{V}_{\text{tank}} = 0$ from a stationary frame of reference since the tank walls are stationary (no-slip boundary condition). From the *rotating* frame of reference however, the tank walls are rotating in the opposite direction of ω. Mathematically, we write $\boldsymbol{u_r = u_z = 0}$ **and** $\boldsymbol{u_\theta = -R_{\text{tank}}\omega}$ **at** $\boldsymbol{r = R_{\text{tank}}}$ (the tank side walls). **At the bottom wall of the tank we write** $\boldsymbol{u_r = u_z = 0}$ **and** $\boldsymbol{u_\theta = -r\omega}$ **at** $\boldsymbol{z = 0}$. On the blade surfaces, the fluid velocity must equal that of the blades (also the no-slip condition). Since the blades are stationary in this rotating frame of reference, $\boldsymbol{u_r = u_z = u_\theta = 0}$ **at the blade surfaces**. Finally, **at the free surface** $\boldsymbol{P = P_{\text{atm}}}$ since the free surface is exposed to atmospheric air. In addition, the vertical component of velocity u_z must equal zero at the free surface. We note that the other two velocity components (u_r and u_θ) may be non-zero at the free surface, but the shear stress in the horizontal plane of the free surface must be zero (negligible shear due to the air). Mathematically, $\boldsymbol{\partial u_r/\partial z = \partial u_\theta/\partial z = 0}$ **at the free surface**.

Discussion In this problem the free surface boundary conditions are independent of frame of reference.

Review Problems

9-108C

Solution We are to list the six steps used to solve fluid flow problem with the continuity and Navier-Stokes equations, for the case in which the fluid is incompressible and has constant properties.

Analysis The steps are listed below:

Step 1 Lay out the problem and the geometry. Identify all relevant dimensions and parameters.
Step 2 List all appropriate assumptions, approximations, simplifications, and boundary conditions.
Step 3 Write out and simplify the differential equations (continuity and the required components of Navier-Stokes) as much as possible.
Step 4 Solve (integrate) the differential equations. This leads to one or more constants of integration.
Step 5 Apply boundary conditions to obtain values for the constants of integration.
Step 6 Verify the results by checking that the flow field meets all the specifications and boundary conditions.

Discussion These steps are not always followed in the same order. For example, in CFD applications the boundary conditions are applied *before* the equations are integrated.

9-109C

Solution We are to name each equation, and then discuss its restrictions and its physical meaning.

Analysis (*a*) This is the **continuity equation**. The form given here is valid for any fluid. It describes conservation of mass in a fluid flow.

(*b*) This is **Cauchy's equation**. The form given here is valid for any fluid. It describes conservation of linear momentum in a fluid flow.

(*c*) This is the **Navier-Stokes equation**. The form given here is valid for a specific type of fluid, namely an incompressible Newtonian fluid. The equation describes conservation of linear momentum in a fluid flow.

Discussion It is important that you be able to recognize these notable equations of fluid mechanics.

9-110C

Solution We are to discuss the connection(s) between the incompressible flow approximation and the constant temperature approximation.

Analysis For an incompressible flow, the density is assumed to be constant. In addition, the incompressible flow approximation usually implies that *all* fluid properties (viscosity, thermal conductivity, etc.) are constant as well. These assumptions go hand in hand because a flow with constant density implies a flow with little or no temperature changes and no buoyancy effects. Since viscosity is a strong function of temperature but generally a weak function of pressure, the fluid's viscosity is approximately constant whenever temperature is constant. When dealing with incompressible fluid flows, pressure variable P is interpreted as the mechanical pressure P_m, and we don't need an equation of state. In effect, the equation of state is replaced by the assumption of constant density and constant temperature.

Discussion Mechanical pressure P_m is determined by the flow field, not by thermodynamics.

9-111C

Solution

(*a*) **True**: The unknowns for an incompressible flow problem with constant fluid properties are pressure and the three components of velocity. Density and viscosity are constants and are therefore not unknowns.

(*b*) **False**: The unknowns for a compressible flow problem are pressure, the three components of velocity, and the density. However, density is a thermodynamic function of pressure and temperature. Hence, temperature appears as an additional unknown, as does some kind of equation of state. In summary, there are actually at least 6 unknowns (P, u, v, w, ρ, and T). We therefore need 6 equations (continuity, 3 components of Navier-Stokes, equation of state, and energy). In addition, fluid properties such as viscosity may change as well, and we need either more equations or some kind of look-up table for these properties.

(*c*) **False**: Cauchy's equation contains additional unknowns – the components of the stress tensor, which must be written in terms of the velocity and pressure fields through some kind of constitutive equation.

(d) **True**: For an incompressible flow problem involving a Newtonian fluid, there are only 4 unknowns (P, u, v, and w). We therefore need only 4 equations (continuity and 3 components of Navier-Stokes).

9-112C

Solution We are to discuss the relationship between volumetric strain rate and the continuity equation.

Analysis Volumetric strain rate is defined as the rate of increase of volume of a fluid element per unit volume. In a compressible flow field, the volume of a fluid particle may increase or decrease as it moves along in the flow, but its mass must remain constant. (This is a fundamental statement of conservation of mass of a system, since the fluid particle can be thought of as an infinitesimal system.) Mathematically it turns out that **volumetric strain rate is the sum of the three normal strain rates, and is identically zero for incompressible flow** (density cannot change, and hence volume cannot change). The continuity equation is based on the same fundamental principle of mass conservation. It is a differential form of the equation of conservation of mass. Its incompressible form also shows that the sum of the three normal strain rates must be zero. On the other hand, if the density is *not* constant, the sum of the three normal strain rates is not zero, but is still equal to the volumetric strain rate, which is also non-zero.

Discussion Volumetric strain rate is derived and discussed in Chap. 4 as a kinematic property.

9-113

Solution For a given geometry and set of boundary conditions, we are to calculate the velocity and pressure fields, and plot the velocity profile.

Assumptions The assumptions are identical to those of Example 9-17. We do not list them here.

Analysis We obtain the velocity and pressure fields by following the step-by-step procedure for differential fluid flow solutions. Everything is identical to Example 9-17 except for the boundary condition at the wall. Boundary condition (1), the no-slip condition, becomes: at x = 0, u = v = 0. w = V. Steps 1 through 4 are otherwise identical, and the result is

Result of integration of z momentum: $$w = \frac{\rho g}{2\mu}x^2 + C_1 x + C_2 \qquad (1)$$

We continue, beginning with Step 5:

Step 5 We apply boundary conditions (1) and (2) from Step 2 to obtain constants C_1 and C_2,

Boundary condition (1): $$w = 0 + 0 + C_2 = V \qquad C_2 = V$$

and

Boundary condition (2): $$\left.\frac{dw}{dx}\right)_{x=h} = \frac{\rho g}{\mu}h + C_1 = 0 \qquad C_1 = -\frac{\rho g h}{\mu}$$

Finally, Eq. 1 becomes

Result: $$\boxed{w = \frac{\rho g}{2\mu}x^2 - \frac{\rho g}{\mu}hx + V = \frac{\rho g x}{2\mu}\left(x - 2h\right) + V} \qquad (2)$$

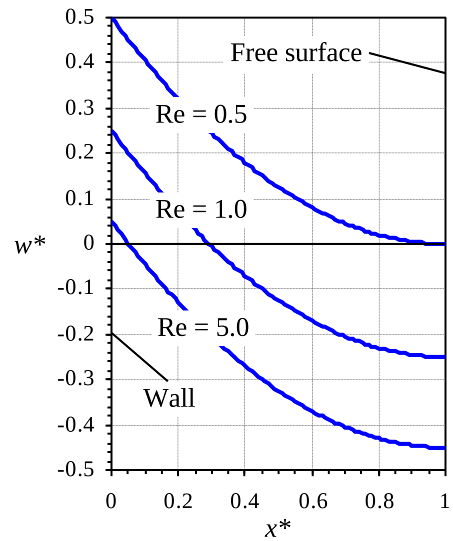


FIGURE 1
Velocity profiles for an oil film falling down a moving vertical wall. For all three Reynolds numbers, Fr = 0.5.

Since $x < h$ in the film, the first term in Eq. 2 is negative, but the second term is positive. Depending on the relative magnitude of the terms, part or all of the vertical velocity may be positive. The pressure field is still $P = P_{atm}$ everywhere.

Step 6 Verify the results. You can plug in the velocity field to verify that all the differential equations and boundary conditions are satisfied.

We nondimensionalize Eq. 2 by inspection: we let $x^* = x/h$ and $w^* = w\mu/(\rho g h^2)$. Eq. 2 becomes

Nondimensional velocity profile: $$w^* = \frac{x^*}{2}\left(x^* - 2\right) + \frac{V\mu}{\rho g h^2} \qquad (3)$$

We verify by inspection that when V = 0, Eq. 3 reduces to the velocity profile of Example 9-17. After some algebra we see that Eq. 3 can be re-written as

Final nondimensional velocity profile: $$w^* = \frac{x^*}{2}\left(x^* - 2\right) + \frac{\mathrm{Fr}^2}{\mathrm{Re}} \qquad (4)$$

where Froude number Fr = $V/\sqrt{gh}$ and Reynolds number Re = $\rho Vh/\mu$. We plot the nondimensional velocity field in Fig. 1 for Fr = 0.5 and Re = 0.5, 1.0, and 5.0.

Discussion Notice that the velocity profile has zero slope at the free surface regardless of the values of Fr and Re. For large enough V, the net mass flow rate is upward rather than downward.

9-114

Solution We are to calculate the volume flow rate per unit width of oil falling down a moving vertical wall, and then calculate the wall speed such that the net volume flow rate of oil is zero.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The wall is infinitely wide and very long so that all of the parallel flow, fully developed approximations of the previous problem hold.

Analysis We calculate the volume flow rate per unit width by integration of the velocity:

Volume flow rate per unit depth:

$$\frac{\dot{V}}{L} = \int_0^h w\,dx = \int_0^h \left[\frac{\rho g x}{2\mu}\left(x-2h\right)+V\right]dx \;= \boxed{Vh - \frac{\rho g h^3}{3\mu}} \qquad (1)$$

The volume flow rate is zero when the two terms in Eq. 1 cancel,

Zero volume flow rate:

$$\frac{\dot{V}}{L} = 0 \;\text{ when }\; Vh = \frac{\rho g h^3}{3\mu} \;\text{ or }\; \boxed{V = \frac{\rho g h^2}{3\mu}} \qquad (2)$$

For an oil film of thickness 5.65 mm with ρ = 888 kg/m^3 and μ = 0.80 kg/(m·s), we calculate V using Eq. 2,

Result for V:

$$V = \frac{\rho g h^2}{3\mu} = \frac{(888\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.00412\text{ m})^2}{3(0.801\text{ kg/m}\cdot\text{s})} = 0.061535\text{ m/s} \cong \mathbf{0.0615\,m/s} \qquad (3)$$

Discussion For any V greater than the value calculated in Eq. 3, the net oil flow is up, while for V less than this value, the net oil flow is down. Since viscosity is in the denominator of Eq. 2, a low viscosity liquid (like water) would require a very large vertical velocity in order to achieve a net upward flow of the liquid.

9-115E

Solution For a given axial velocity component in an axisymmetric flow field, we are to validate the incompressible approximation, generate the radial velocity component, generate an expression for the stream function, and then plot some streamlines and design the shape of the contraction.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is axisymmetric implying that $u_\theta = 0$ and there is no variation in the θ direction.

Properties At room temperature and pressure, the speed of sound in air is about 1130 ft/s.

Analysis (*a*) The maximum speed occurs in the test section, where the Mach number is

Mach number:

$$\text{Ma} = \frac{u_{z,L}}{c} = \frac{120 \frac{\text{ft}}{\text{s}}}{1130 \frac{\text{ft}}{\text{s}}} = 0.106 \qquad (1)$$

Since Ma is much less than 0.3, *the incompressible flow approximation is reasonable*.

(*b*) Between $z = 0$ and $z = L$, the axial velocity component is given by

Axial velocity component:

$$u_z = u_{z,0} + \frac{u_{z,L} - u_{z,0}}{L} z \qquad (2)$$

We use the incompressible continuity equation in cylindrical coordinates, simplified as follows for axisymmetric flow,

Incompressible axisymmetric continuity equation:

$$\frac{1}{r}\frac{\partial (r u_r)}{\partial r} + \frac{\partial (u_z)}{\partial z} = 0$$

After rearranging,

$$\frac{\partial (r u_r)}{\partial r} = -r\frac{\partial (u_z)}{\partial z} = -r\frac{u_{z,L} - u_{z,0}}{L} \qquad (3)$$

We integrate Eq. 3 with respect to r,

$$r u_r = -\frac{r^2}{2}\frac{u_{z,L} - u_{z,0}}{L} + f(z) \qquad (4)$$

Notice that since we performed a *partial* integration with respect to r, we add a function of the other variable z rather than simply a constant of integration. We divide all terms in Eq. 4 by r and recognize that the term with $f(z)$ will go to infinity at the centerline of the contraction ($r = 0$) unless $f(z) = 0$. Our final expression for u_r is thus

Radial velocity component:

$$\boxed{u_r = -\frac{r}{2}\frac{u_{z,L} - u_{z,0}}{L}} \qquad (5)$$

(*c*) The algebra for generating the stream function is identical to that of Problem 9-61 except for a change in notation. The result is thus

Stream function: $$\boxed{\psi = \frac{r^2}{2}\left(u_{z,0} + \frac{u_{z,L} - u_{z,0}}{L}z\right) + \text{constant}} \qquad (6)$$

The constant can be anything. We set it to zero for simplicity.

(*d*) First we calculate the axial speed at the entrance to the contraction. By conservation of mass,

$$u_{z,0}A_0 = u_{z,L}A_L \qquad \text{or} \qquad u_{z,0}\frac{\pi D_0^{\ 2}}{4} = u_{z,L}\frac{\pi D_L^{\ 2}}{4}$$

from which

$$u_{z,0} = u_{z,L}\frac{D_L^{\ 2}}{D_0^{\ 2}} = 120\,\frac{\text{ft}}{\text{s}} \times \frac{(1.5\text{ ft})^2}{(5.0\text{ ft})^2} = 10.8\,\frac{\text{ft}}{\text{s}}$$

We solve Eq. 6 for r as a function of z and plot several streamlines in Fig. 1,

Streamlines: $$r = \pm\sqrt{\frac{2\psi}{u_{z,0} + \dfrac{u_{z,L} - u_{z,0}}{L}z}} \qquad (7)$$

At the entrance of the contraction (z = 0), the wall is at $r = D_0/2$ = 2.5 ft. Eq. 6 yields ψ_{wall} = 33.75 ft^3/s for the streamline that passes through this point. This streamline thus represents the shape of the nozzle wall, and we have designed the nozzle shape.

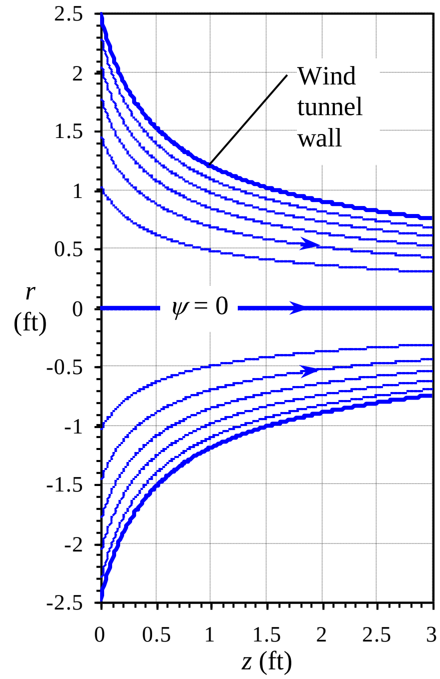


FIGURE 1
Streamlines for flow through an axisymmetric wind tunnel contraction.

Discussion Since the boundary layers along the walls of the contraction are very small, the assumption about negligible friction effects is reasonable. This contraction shape should deliver the desired axial flow speed quite nicely.

9-116

Solution We are to determine a relationship between constants *a*, *b*, *c*, *d*, and *e* that ensures incompressibility, and we are to determine the primary dimensions of each constant.

Assumptions **1** The flow is steady. **2** The flow is incompressible (under certain restraints to be determined).

Analysis We plug the velocity components into the incompressible continuity equation,

Condition for incompressibility:

$$\underbrace{\frac{\partial u}{\partial x}}_{az^2}+\underbrace{\frac{\partial v}{\partial y}}_{cxz}+\underbrace{\frac{\partial w}{\partial z}}_{3dz^2+2exz}=0 \qquad az^2+cxz+3dz^2+2exz=0 \qquad (1)$$

To guarantee incompressibility, the above equation must be satisfied everywhere. We equate similar terms to obtain the following relationships:

Conditions for incompressibility:

$$\boxed{a=-3d \qquad c=-2e} \qquad (2)$$

The units are found by observing that each component of the velocity field must be dimensionally homogeneous – each term must have dimensions of velocity. We examine each term:

$$\left\{axz^2\right\}=\left\{a\times \mathrm{L}^3\right\}=\left\{\frac{\mathrm{L}}{\mathrm{t}}\right\} \qquad \boxed{\{a\}=\left\{\frac{1}{\mathrm{L}^2\mathrm{t}}\right\}}$$

$$\{by\}=\{b\times \mathrm{L}\}=\left\{\frac{\mathrm{L}}{\mathrm{t}}\right\} \qquad \boxed{\{b\}=\left\{\frac{1}{\mathrm{t}}\right\}}$$

$$\{cxyz\}=\left\{c\times \mathrm{L}^3\right\}=\left\{\frac{\mathrm{L}}{\mathrm{t}}\right\} \qquad \boxed{\{c\}=\left\{\frac{1}{\mathrm{L}^2\mathrm{t}}\right\}}$$

$$\left\{dz^3\right\}=\left\{d\times \mathrm{L}^3\right\}=\left\{\frac{\mathrm{L}}{\mathrm{t}}\right\} \qquad \boxed{\{d\}=\left\{\frac{1}{\mathrm{L}^2\mathrm{t}}\right\}}$$

$$\left\{exz^2\right\}=\left\{e\times \mathrm{L}^3\right\}=\left\{\frac{\mathrm{L}}{\mathrm{t}}\right\} \qquad \boxed{\{e\}=\left\{\frac{1}{\mathrm{L}^2\mathrm{t}}\right\}}$$

Discussion If Eq. 2 were not satisfied, the given velocity field might still represent a valid flow field, but density would have to vary with location in the flow field – in other words the flow would be *compressible*.

9-117

Solution We are to simplify the incompressible Navier-Stokes equation for the case of rigid body motion with arbitrary acceleration.

Analysis We begin with the vector form of the incompressible Navier-Stokes equation,

Incompressible Navier-Stokes equation: $$\rho \frac{D\vec{V}}{Dt} = -\vec{\nabla}P + \rho\vec{g} + \mu\nabla^2\vec{V} \qquad (1)$$

In rigid body motion, $\vec{V}$ is not zero, but since the liquid moves as a solid body there is no *relative* motion between fluid particles. Thus the viscous term in Eq. 1 disappears. (Fluid particles do not rub against each other or shear against each other in any way, so the viscous term must vanish.) The material acceleration term $D\vec{V}/Dt$ is the acceleration following a fluid particle; hence it is identical to the imposed acceleration $\vec{a}$. Finally, $\vec{g} = -g\vec{k}$. Thus Eq. 1 reduces to

Equation for rigid body acceleration: $$\boxed{\vec{\nabla}P + \rho g\vec{k} = -\rho\vec{a}} \qquad (2)$$

Discussion You can verify that Eq. 2 agrees with the rigid body acceleration equation of Chap. 3.

9-118

Solution We are to simplify the incompressible Navier-Stokes equation for the case of hydrostatics.

Analysis We begin with the vector form of the incompressible Navier-Stokes equation,

Incompressible Navier-Stokes equation: $$\rho \frac{D\vec{V}}{Dt} = -\vec{\nabla}P + \rho\vec{g} + \mu\nabla^2\vec{V} \qquad (1)$$

In hydrostatics, $\vec{V}$ = 0 everywhere (no flow). Thus the first and last terms in Eq. 1 disappear. In addition, $\vec{g} = -g\vec{k}$. Thus Eq. 1 reduces to

Hydrostatics equation: $$\boxed{\vec{\nabla}P = -\rho g\vec{k}} \qquad (2)$$

Discussion We verify from Eq. 2 that pressure does not change horizontally, but increases downward.

9-119

Solution We are to specify boundary conditions in terms of stream function.

Assumptions **1** The flow is steady. **2** The flow is incompressible **3** The flow is two-dimensional.

Analysis (*a*) For 2-D incompressible flow the difference in the value of the stream function between two streamlines is equal to the volume flow rate per unit width between the two streamlines. Since the entire flow is confined between the lower and upper channel walls, we know that stream function ψ must be constant along the upper wall. We calculate ψ on the upper channel wall as follows:

$$V_1 = \frac{\dot{V}}{H_1 W} = \frac{1}{H_1}\frac{\dot{V}}{W} = \frac{1}{H_1}\left(\psi_{\text{upper}} - \psi_{\text{lower}}\right) \qquad (1)$$

from which

ψ_{upper}:
$$\psi_{\text{upper}} = \psi_{\text{lower}} + H_1 V = 0 + \left(0.12\text{ m}\right)\left(18.5\text{ m/s}\right) = \mathbf{2.22\ m^2/s} \qquad (2)$$

(*b*) Since the inlet flow is uniform, ψ must increase linearly from ψ_{lower} to ψ_{upper} along the left edge of the computational domain. In equation form,

ψ_{left}:
$$\psi_{\text{left}} = \psi_{\text{lower}} + \frac{\left(\psi_{\text{upper}} - \psi_{\text{lower}}\right)}{H_1} y = \frac{2.22\text{ m}^2\text{/s}}{0.12\text{ m}} y = \boxed{\left(18.5\text{ m/s}\right) y} \qquad (3)$$

We notice that Eq. 3 could have been obtained directly from $u = V_1 = \partial\psi/\partial y$.

(*c*) We have some options for the right edge of the computational domain. If that boundary is far enough away that it does not adversely affect the flow near the sudden contraction, we might specify a uniform velocity distribution along the right edge, similar to Eq. 3 above, but with a higher velocity determined by conservation of mass,

Average outlet speed:
$$V_2 = V_1 \frac{H_1}{H_2} = \left(18.5\text{ m/s}\right)\frac{0.12\text{ m}}{0.046\text{ m}} = 48.26\text{ m/s}$$

In other words, we would specify

ψ_{right}:
$$\boxed{\psi_{\text{right}} = \left(48.26\text{ m/s}\right) y} \qquad (4)$$

Eq. 4 is not a very good boundary condition because we know that viscous effects will surely slow down the flow near the walls – the velocity profile at the outlet will *not* be uniform.

A much better boundary condition (if the code permits it) is to specify that ψ not change with x along the right edge of the domain. Mathematically, we would specify

ψ_{right}:
$$\frac{\partial \psi_{\text{right}}}{\partial x} = 0 \qquad (5)$$

You can see from the definition of ψ that Eq. 5 is identical to forcing velocity component v to be zero at the outlet. In other words, we specify that the flow at the outlet is *parallel*.

A third option would be to locate the right edge very far downstream so that the flow there is fully developed channel flow, for which we can specify the stream function as a function of y along the edge. ψ can be obtained from Problem 9-43.

Discussion CFD and boundary conditions are discussed in detail in Chap. 15.

9-120

Solution For each equation we are to tell whether it is linear or nonlinear and explain.

Analysis (*a*) The incompressible continuity equation is

The incompressible continuity equation:
$$\vec{\nabla}\cdot\vec{V}=0 \tag{1}$$

This equation is linear. There are no nonlinear terms.

(*b*) The compressible continuity equation is

The compressible continuity equation:
$$\frac{\partial\rho}{\partial t}+\vec{\nabla}\cdot\left(\rho\vec{V}\right)=0 \tag{2}$$

This equation is nonlinear. The second term has a product of two variables, ρ and $\vec{V}$ – this is what makes the equation nonlinear.

(*c*) The incompressible Navier-Stokes equation is

The incompressible Navier-Stokes equation:
$$\rho\frac{D\vec{V}}{Dt}=-\vec{\nabla}P+\rho\vec{g}+\mu\nabla^2\vec{V} \tag{3}$$

This equation is nonlinear. The material acceleration term on the left can be written as

The incompressible Navier-Stokes equation:
$$\frac{D\vec{V}}{Dt}=\underbrace{\frac{\partial\vec{V}}{\partial t}}_{\text{Unsteady or local part}}+\underbrace{\left(\vec{V}\cdot\vec{\nabla}\right)\vec{V}}_{\text{Advective (or convective) part}} \tag{4}$$

The advective part of Eq. 4 contains products of variable $\vec{V}$ and derivatives of variable $\vec{V}$ – this is what makes the equation nonlinear.

Discussion Density is treated as a constant in Eq. 3, and does not affect the nonlinearity of the equation. For *compressible* flow however, variable density causes the nonlinearity.

9-121

Solution We are to sketch some streamlines for boundary layer flow.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis We can offer only quantitative sketches of the streamlines. Since there is no flow reversal, we can be sure that **δ(*x*) is not a streamline**. In fact, streamlines must *cross* $\delta(x)$. Furthermore, at any given *y* location above the plate the fluid speed decreases as the boundary layer grows downstream. Hence, *the streamlines must diverge*. The bottom line is that the streamlines veer slightly upward away from the wall to compensate for the loss of speed in the boundary layer. Streamlines are sketched in Fig. 1.

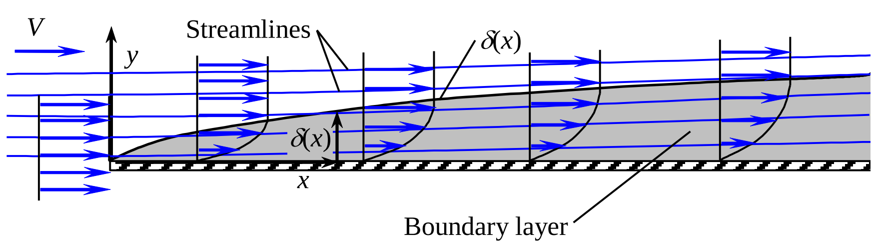


FIGURE 1
Streamlines above and within a flat plate boundary layer; since streamlines cross the curve $\delta(x)$, $\delta(x)$ cannot itself be a streamline of the flow. Furthermore, streamlines within the boundary layer veer up because of decreasing speeds within the boundary layer.

Discussion As the boundary layer grows in thickness, more and more streamlines end up inside the boundary layer.

9-122

Solution We are to define a ψ that satisfies the continuity equation, and increases in the positive z direction when the flow is from right to left in the x-z plane.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the x-z plane.

Analysis We propose the following stream function,

Stream function:
$$\boxed{u = -\frac{\partial \psi}{\partial z} \qquad w = \frac{\partial \psi}{\partial x}} \tag{1}$$

We verify that the continuity equation is satisfied by Eq. 1,

Steady, incompressible, 2-D continuity equation:
$$\underbrace{\frac{\partial u}{\partial x}}_{-\frac{\partial^2 \psi}{\partial x \partial z}} + \underbrace{\frac{\partial w}{\partial z}}_{\frac{\partial^2 \psi}{\partial z \partial x}} = 0 \tag{2}$$

The only restriction is that ψ must be a smooth function of x and z. We check if we picked the proper signs by examining freestream flow from right to left in the x-z plane:

Freestream flow:
$$u = -U \qquad w = 0 \qquad \psi = Uz + C \tag{3}$$

where U is a positive constant and C is an arbitrary constant. Thus we verify that as z increases, ψ increases, and the flow is from right to left as desired.

Discussion If we had defined ψ with the opposite signs of Eq. 1, the flow would be from left to right as ψ increases.

9-123

Solution We are to analyze this problem two ways: with the exact (differential) technique, and with dimensional analysis, and we are to compare the results.

Assumptions **1** The flow is steady. **2** The flow is incompressible, Newtonian, laminar, parallel, and fully developed ($u = u(y)$ only, where x is in the direction of motion and y is normal to the direction of motion). **3** We ignore aerodynamic drag on the block.

Analysis (*a*) We draw a free-body diagram of the block in Fig. 1 and sum all the forces acting on it. There are only two forces in the x direction: the x component of weight $W\sin\alpha$ and the force τA due to viscous shear at the bottom surface of the block. Since the block slides at constant speed, these two forces must balance.

Force balance:
$$W\sin\alpha = \tau A = \frac{\mu VA}{h} \qquad (1)$$

where we have used the exact analytical expression for the shear stress for Couette flow, namely $\tau = \mu(du/dy) = \mu V/h$. Solving for h,

Exact solution for h:
$$\boxed{h = \frac{\mu VA}{W\sin\alpha}} \qquad (2)$$

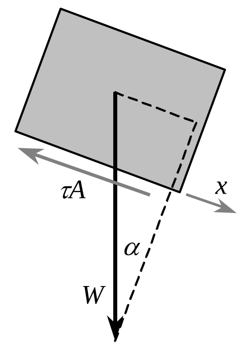


FIGURE 1
Free-body diagram of the block.

(*b*) We perform a dimensional analysis leaving out many of the details. There are 6 parameters in the problem: h as a function of V, A, W, α, and μ. There are three primary dimensions represented in the problem, namely m, L, and t. Thus we expect 6-3 = 3 Πs. We choose three repeating variables, V, A, and W. The Πs are

Dimensionless parameters:
$$\Pi_1 = \frac{h}{\sqrt{A}} \qquad \Pi_2 = \frac{\mu V\sqrt{A}}{W} \qquad \Pi_3 = \alpha$$

The dimensionless relationship is

Result of dimensional analysis:
$$\boxed{\frac{h}{\sqrt{A}} = f\left(\frac{\mu V\sqrt{A}}{W},\alpha\right)} \qquad (3)$$

To put the Πs of Eq. 3 into the form of Eq. 2 we do the following:

Relationship between Πs:
$$\Pi_1 = \frac{\Pi_2}{\sin\Pi_3} \quad\rightarrow\quad \frac{h}{\sqrt{A}} = \frac{\mu V\sqrt{A}}{W\sin\alpha} \quad\rightarrow\quad \boxed{h = \frac{\mu VA}{W\sin\alpha}} \qquad (4)$$

Thus we see that dimensional analysis is indeed consistent with the exact solution. Of course, we could not know the relationship of Eq. 4 by dimensional reasoning alone.

Discussion The agreement between Parts (*a*) and (*b*) is satisfying and emphasizes two different approaches to the same engineering problem.

9-124

Solution We are to write Poisson's equation in standard form and discuss its similarities and differences compared to Laplace's equation.

Analysis Poisson's equation in standard form is

Poisson's equation: $\boxed{\nabla^2\phi = s}$ (1)

where ϕ is a dependent variable that is a function of space, ∇^2 is the Laplacian operator, and s is the right hand side of the equation, which may be a function of space, but cannot be a function of ϕ itself. Poisson's equation is similar to Laplace's equation in that the left hand sides are identical. The difference is that Poisson's equation has a non-zero right hand side whereas the right hand side of Laplace's equation is zero. Note: Poisson's equation reduces to Laplace's equation if $s = 0$.

Discussion We discuss Poisson's equation briefly in this chapter in relation to pressure correction algorithms used by CFD codes.

9-125

Solution We are to analyze this problem three ways: with the control volume technique, with the differential technique, and with dimensional analysis, and we are to compare the results.

Assumptions **1** The flow is steady. **2** The flow is axisymmetric, incompressible, Newtonian, laminar, parallel, and fully developed ($u = u(r)$ only).

Analysis (*a*) We use the head form of the energy equation from point 1 to point 2. Since there are no pumps, turbines, or minor losses the energy equation reduces to

Energy equation:
$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_f \tag{1}$$

The pressure terms cancel since $P_1 = P_2 = P_{atm}$. The velocity terms cancel since the flow is fully developed. Upon substitution of the major head loss equation we have

Reduced energy equation:
$$\Delta z = z_1 - z_2 = h_f = f\frac{L}{D}\frac{V^2}{2g} \tag{2}$$

But for fully developed laminar pipe flow we know from Chap. 6 that the Darcy friction factor f = 64/Re. Thus Eq. 2 becomes

$$\Delta z = \frac{64}{\text{Re}}\frac{L}{D}\frac{V^2}{2g} = \frac{64\mu}{\rho VD}\frac{L}{D}\frac{V^2}{2g} = \frac{32\mu LV}{\rho D^2 g}$$

from which we can solve for average velocity V through the pipe,

V from control volume analysis:
$$\boxed{V = \frac{\rho g D^2 \Delta z}{32\mu L}} \tag{3}$$

(*b*) An exact analysis of this flow was performed in Problem 9-100. We refer to the solution of that problem and do not show the details here. The average velocity through the pipe was found to be

$$V = \frac{R^2}{8\mu}\rho g \sin\alpha$$

But $R = D/2$, and from the figure provided in the problem statement we see that $\sin\alpha = \Delta z/L$. Thus, our result is

V from differential analysis:
$$\boxed{V = \frac{\rho g D^2 \Delta z}{32\mu L}} \tag{4}$$

The agreement with the result of Part (*a*) is exact.

(*c*) Finally we perform a dimensional analysis. We leave out the details, providing only a summary here; this is a good review of the material of Chap. 7. There are 7 parameters in the problem: V as a function of ρ, g, D, Δz, μ, and L. There are three primary dimensions represented in the problem, namely m, L, and t. Thus we expect 7-3 = 4 Πs. We choose three repeating variables, ρ, g, and D. The Πs are

Dimensionless parameters:
$$\Pi_1 = \frac{V}{\sqrt{gD}} \qquad \Pi_2 = \frac{\rho D\sqrt{gD}}{\mu} \qquad \Pi_3 = \frac{\Delta z}{D} \qquad \Pi_4 = \frac{L}{D}$$

The first Π is a Froude number and the second Π is a Reynolds number. The dimensionless relationship is

Result of dimensional analysis:
$$\boxed{\frac{V}{\sqrt{gD}} = f\left(\frac{\rho D\sqrt{gD}}{\mu},\frac{\Delta z}{D},\frac{L}{D}\right)} \tag{5}$$

To put the Πs of Eq. 5 into the form of Eq. 4 we do the following:

Relationship between Πs:

$$\Pi_1 = \frac{\Pi_2 \Pi_3}{32\Pi_4} \quad \rightarrow \quad \frac{V}{\sqrt{gD}} = \frac{\rho D\sqrt{gD}}{32\mu}\frac{\Delta z}{D}\frac{D}{L} \quad \rightarrow \quad \boxed{V = \frac{\rho g D^2 \Delta z}{32\mu L}} \tag{6}$$

Thus we see that dimensional analysis is indeed consistent with the exact solution. Of course, we could not know the relationship of Eq. 6 by dimensional reasoning alone.

Discussion The agreement between Parts (*a*), (*b*), and (*c*) is satisfying and emphasizes three different approaches to the same engineering problem.

9-126

Solution We are to determine the primary dimensions of ψ, nondimensionalize Eq. 1, and then plot several nondimensional streamlines for this flow field.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x-y* or *r-θ* plane.

Analysis

(*a*) There are several ways to calculate the primary dimensions of ψ. First, from Eq. 1 we see that

Dimensions of stream function:
$$\boxed{\{\psi\} = \left\{\frac{\dot{V}}{2\pi L}\right\} = \left\{\frac{\mathrm{L^3 t^{-1}}}{\mathrm{L}}\right\} = \left\{\frac{\mathrm{L^2}}{\mathrm{t}}\right\}} \quad (2)$$

We could also use the definition of ψ. Since velocities are obtained by spatial derivatives of ψ, ψ must have an additional length dimension in the numerator compared to the dimensions of velocity. This reasoning also yields $\{\psi\} = \{\mathrm{L^2/t}\}$.

(*b*) The nondimensional form of the stream function is straightforward. Eq. 1 becomes

Nondimensional stream function:
$$\psi^* = -\arctan\frac{\sin 2\theta}{\cos 2\theta + \dfrac{1}{r^{*2}}} \quad (3)$$

(*c*) We solve Eq. 3 for r^*,

Equation for nondimensional streamlines:
$$\boxed{r^* = \pm\sqrt{\frac{\tan(-\psi^*)}{\sin 2\theta - \cos 2\theta \tan(-\psi^*)}}} \quad (4)$$

We pick the positive root to avoid negative radii. We plot several streamlines in the desired range in Fig. 1. The range of ψ^* is 0 on the positive *x* axis to $-\pi$ on the positive *y* axis to -2π on the negative *x* axis.

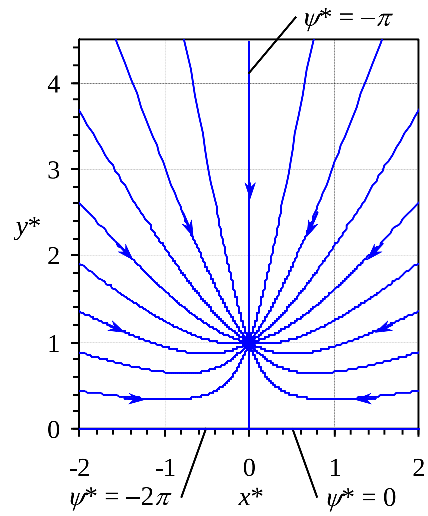


FIGURE 1
Nondimensional streamlines for flow into a vacuum cleaner attachment; ψ^* is incremented uniformly from 2π (negative *x* axis) to 0 (positive *x* axis).

Discussion The point ($x = 0$, $y = b$) is a singularity point with infinite velocity.

9-127

Solution Two examples in Chapter 9 developed a velocity profile and flow rate equation by making assumptions in pipe flow where one of these examples yielded the Poiseuille flow equation. In this problem, we assume the fluid, blood here, is not Newtonian but a Bingham Plastic fluid. The velocity profile and flow rate are to be determined. In addition, the velocity profile is to be plotted along with Newtonian and Pseudoplastic fluids.

Assumptions: **1** The flow is steady and incompressible. **2** The flow occurs within a rigid, circular pipe with uniformity and is axisymmetric.

Analysis Following the Power Law example in Chap. 9, we want to derive the velocity profile and flow rate using the shear stress equation provided for a Bingham Plastic model as noted below,

$\tau = -\mu \frac{du}{dr} + \tau_y$and recall that $\tau = \frac{r}{2}\frac{dP}{dz}$

So, if we substitute for τ and rearrange to get

$$\frac{du}{dr} = -\frac{r}{2\mu}\frac{dP}{dz} + \frac{\tau_y}{\mu}$$

$$du = -\frac{1}{2\mu}\frac{dP}{dz} r\,dr + \frac{\tau_y}{\mu} dr$$

Integrate,

$$u = -\frac{1}{4\mu}\frac{dP}{dz} r^2 + \frac{\tau_y}{\mu} r + C$$

where *C* is a constant of integration.

Apply the boundary condition, $at\ r = R, u = 0$, which yields

$$C = \frac{1}{4\mu}\frac{dP}{dz} R^2 + \frac{\tau_y}{\mu} R$$

We then substitute back in to arrive at a velocity equation

$$u = \frac{1}{4\mu}\frac{dP}{dz}(R^2 - r^2) - \frac{\tau_y}{\mu}(R - r)$$

However, this is not complete. Because there is a yield stress, we will have to treat the solution a little differently and need to break up the flow into 2 sections. For flow to occur, the wall shear stress must exceed the Bingham Plastic yield stress. Therefore, we must define some radius (R_y) where the shear stress will exceed the yield stress so flow can occur.

$$\tau_y = \frac{R_y}{2}\frac{dP}{dz}$$

In terms of our velocity profile, the fluid velocity will be constant from R=0 to R=R_y and written as

$$u_y = \frac{1}{4\mu}\frac{dP}{dz}\left(R^2 - R_y^{\,2}\right) - \frac{\tau_y}{\mu}\left(R - R_y\right)$$

Substitute for τ_y,

$$u_y = \frac{1}{4\mu}\frac{dP}{dz}\left(R^2 - R_y^{\,2}\right) - \frac{R_y}{2\mu}\frac{dP}{dz}\left(R - R_y\right)$$

So, the velocity profile in the outer region (from R_y to R) is

$$u = \frac{1}{\mu}\frac{dP}{dz}\left[\frac{(R^2 - r^2)}{4} - \frac{R_y}{2}(R - r)\right]$$

So, for the flow rate, this must be looked as two separate regions as well where there is the central region of flow and aouter region of flow.

$$Q_{total} = Q_{central} + Q_{peripheral}$$

Let's take these two separately.

$Q_{central}$ is easy since we know the velocity is constant through this region. Thus, the equation becomes

$Q_{central} = u_y \pi R_y^{\,2}$

Therefore,

$$Q_{central} = \frac{\pi}{\mu}\frac{dP}{dz}\left[\frac{\left(R^2 - R_y^{\,2}\right)R_y^{\,2}}{4} - \frac{R_y^{\,3}}{2}\left(R - R_y\right)\right]$$

In the outer region, the flow rate is defined by an integral over that region,

$$Q_{peripheral} = \int_{R_y}^{R} 2\pi r u dr$$

After we substitute for *u*, the equation becomes

$$Q_{peripheral} = \int_{R_y}^{R} 2\pi r \frac{1}{\mu}\frac{dP}{dz}\left[\frac{(R^2 - r^2)}{4} - \frac{R_y}{2}(R - r)\right] dr$$

Integrating yields,

$$Q_{peripheral} = \frac{\pi}{\mu}\frac{dP}{dz}\left[\frac{R^4}{8} - \frac{R^2 {R_y}^2}{4} - \frac{5}{24}{R_y}^4 - \frac{R^3 R_y}{6} + \frac{R {R_y}^3}{2}\right]$$

Combining both flow rate equations,

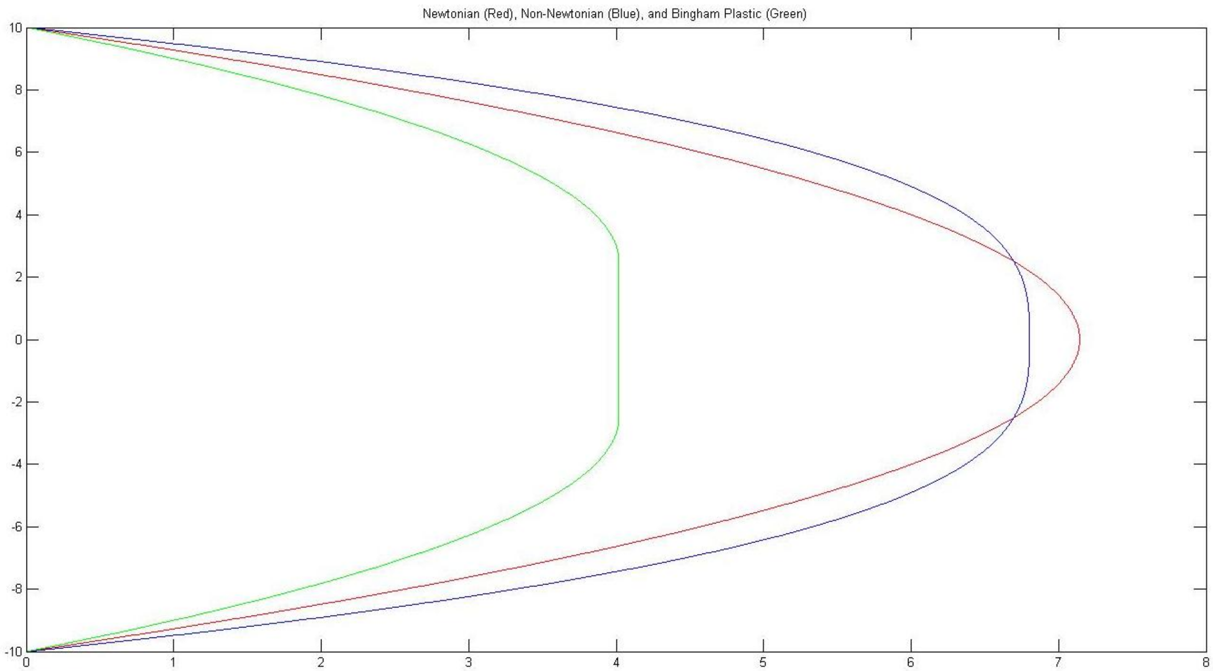
$$Q_{total} = \frac{\pi}{\mu}\frac{dP}{dz}\left[\frac{R^4}{8} + \frac{{R_y}^4}{24} - \frac{R^3 R_y}{6}\right]$$

If we then substitute back in for R_y, the following equation is developed,

$$Q_{total} = \frac{\pi R^4 dP}{8\mu dz}\left[1 - \frac{4}{3}\left(\frac{2\tau_y}{R\left({dP}/{dz}\right)}\right) + \frac{1}{3}\left(\frac{2\tau_y}{R\left({dP}/{dz}\right)}\right)\right]$$

Note: If the yield stress does not exist ($\tau_y = 0$), the equation reduces to that for Poiseuille Flow for a Newtonian Fluid.

The velocity profiles then become for Pseudoplastic, Newtonian, and Bingham Plastic fluids,



Fundamentals of Engineering (FE) Exam Problems

9-128

The continuity equation is also known as

(*a*) Conservation of mass (*b*) Conservation of energy (*c*) Conservation of momentum

(*d*) Newton's second law (*e*) Cauchy's equation

Answer (*a*) Conservation of mass

9-129

The Navier-Stokes equation is also known as

(*a*) Newton's first law (*b*) Newton's second law (*c*) Newton's third law

(*d*) Continuity equation (*e*) Energy equation

Answer (*b*) Newton's second law

9-130

Which one is the general differential equation form of the continuity equation for a control volume?

(*a*) $\int_{\mathrm{CS}} \rho \vec{V} \cdot \vec{n}\, dA = 0$ (*b*) $\int_{\mathrm{CV}} \frac{\partial \rho}{\partial t} dV + \int_{\mathrm{CS}} \rho \vec{V} \cdot \vec{n}\, dA = 0$ (*c*) $\vec{\nabla} \cdot \left(\rho \vec{V}\right) = 0$

(*d*) $\frac{\partial \rho}{\partial t} + \vec{\nabla} \cdot \left(\rho \vec{V}\right) = 0$ (*e*) None of these

Answer (*d*) $\frac{\partial \rho}{\partial t} + \vec{\nabla} \cdot \left(\rho \vec{V}\right) = 0$

9-131

Which one is differential, incompressible, two-dimensional continuity equation in Cartesian coordinates?

(*a*) $\int_{CS} \rho \vec{V} \cdot \vec{n}\, dA = 0$ (*b*) $\frac{1}{r}\frac{\partial (ru_r)}{\partial r} + \frac{1}{r}\frac{\partial (u_\theta)}{\partial \theta} = 0$ (*c*) $\vec{\nabla} \cdot \left(\rho \vec{V}\right) = 0$

(*d*) $\vec{\nabla} \cdot \vec{V} = 0$ (*e*) $\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0$

Answer (*e*) $\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0$

9-132

A steady velocity field is given by $\vec{V} = (u, v, w) = 2ax^2 y\vec{i} + 3bxy^2 \vec{j} + cy\vec{k}$, where *a*, *b*, and *c* are constants. Under what conditions is this flow field incompressible?

(*a*) $a = b$ (*b*) $a = -b$ (*c*) $2a = -3b$ (*d*) $3a = 2b$ (*e*) $a = 2b$

Answer (*c*) $2a = -3b$

Solution

du/dx + dv/dy = 0 → 4axy + 6bxy = 0 → 4a = -6b → 2a = -3b

9-133

A steady, two-dimensional, incompressible flow field in the *xy*-plane has a stream function given by $\psi = ax^2 + by^2 + cy$, where *a*, *b*, and *c* are constants. The expression for the velocity component *u* is

(*a*) $2ax$ (*b*) $2by + c$ (*c*) $-2ax$ (*d*) $-2by-c$ (*e*) $2ax + 2by + c$

Answer (*b*) $2by + c$

Solution

u = d(psi)/dy = 2by + c

9-134

A steady, two-dimensional, incompressible flow field in the *xy*-plane has a stream function given by $\psi = ax^2 + by^2 + cy$, where *a*, *b*, and *c* are constants. The expression for the velocity component *v* is

(*a*) $2ax$ (*b*) $2by + c$ (*c*) $-2ax$ (*d*) $-2by-c$ (*e*) $2ax + 2by + c$

Answer (*c*) $-2ax$

Solution

v = - d(psi)/dx = -2ax

9-135

If a fluid flow is both incompressible and isothermal, which property is not expected to be constant?

(*a*) Temperature (*b*) Density (*c*) Dynamic viscosity

(*d*) Kinematic viscosity (*e*) Specific heat

Answer (*e*) Specific heat

9-136

Which one is incompressible Navier-Stokes equation with constant viscosity?

(*a*) $\rho \dfrac{D\vec{V}}{Dt} + \vec{\nabla}P - \rho\vec{g} = 0$ (*b*) $-\vec{\nabla}P + \rho\vec{g} + \mu\vec{\nabla}^2\vec{V} = 0$ (*c*) $\rho \dfrac{D\vec{V}}{Dt} = -\vec{\nabla}P + \mu\vec{\nabla}^2\vec{V}$

(*d*) $\rho \dfrac{D\vec{V}}{Dt} = -\vec{\nabla}P + \rho\vec{g} + \mu\vec{\nabla}^2\vec{V}$ (*e*) $\rho \dfrac{D\vec{V}}{Dt} = -\vec{\nabla}P + \rho\vec{g} + \mu\vec{\nabla}^2\vec{V} + \vec{\nabla}\cdot\vec{V} = 0$

Answer (*d*) $\rho \dfrac{D\vec{V}}{Dt} = -\vec{\nabla}P + \rho\vec{g} + \mu\vec{\nabla}^2\vec{V}$

9-137

Which one is not correct regarding the Navier-Stokes equation?

(*a*) Nonlinear equation (*b*) Unsteady equation (*c*) Second-order equation

(*d*) Partial differential equation (*e*) None of these

Answer (*e*) None of these

9-138

In fluid flow analyses, which boundary condition can be expressed as $\vec{V}_{\text{fluid}} = \vec{V}_{\text{wall}}$

(*a*) No-slip (*b*) Interface (*c*) Free-surface (*d*) Symmetry (*e*) Inlet

Answer (*a*) No-slip

Solutions Manual for

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Chapter 10

APPROXIMATE SOLUTIONS OF THE NAVIER-STOKES EQUATION

Introductory Problems and Modified Pressure

10-1C

Solution We are to discuss the role of nondimensionalization of the Navier-Stokes equations.

Analysis When we properly nondimensionalize the Navier-Stokes equation, **all the terms are re-written in the form of some nondimensional parameter times a quantity of order unity**. Thus, we can simply compare the orders of magnitude of the nondimensional parameters to see which terms (if any) can be ignored because they are very small compared to other terms. For example, if the Strouhal number is much smaller than the Euler number, we can ignore the term that contains the Strouhal number, but must retain the term that contains the Euler number.

Discussion This method works only if the characteristic scales of the problem (length, speed, frequency, etc.) are chosen properly.

10-2C

Solution We are to label regions in a flow field where certain approximations are likely to be appropriate.

Assumptions **1** The flow is incompressible. **2** The flow is steady in the mean (we ignore the unsteady flow field close to the rotating blades).

Analysis A boundary layer grows along the floor, both upstream and downstream of the fan. The flow upstream of the fan is largely irrotational except very close to the floor. The air is nearly static far upstream and far above the fan. Downstream of the fan, the flow is most likely swirling and turbulent, and none of the approximations are expected to be appropriate there. In other words, the full Navier-Stokes equation must be solved in that region. **We sketch all these regions in Fig. 1**.

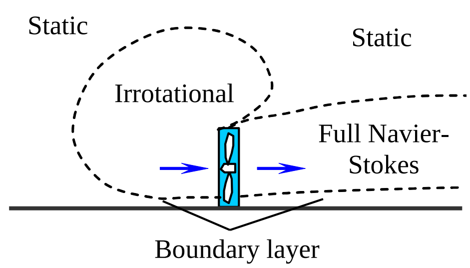


FIGURE 1
Regions of appropriate approximations for the flow produced by a box fan sitting on the floor of a large room.

Discussion The regions sketched in Fig. 1 are not well defined, nor are they necessarily to scale.

10-3C

Solution We are to discuss the difference between an "exact" solution and an approximate solution of the Navier-Stokes equation.

Analysis **In an "exact" solution, we begin with the full Navier-Stokes equation**. As we solve the problem, some terms may drop out due to the specified geometry or other simplifying assumptions in the problem. **In an approximate solution, we eliminate some terms in the Navier-Stokes equation right from the start**. In other words, we *begin* with a reduced or simplified *approximate* form of the equation.

Discussion The approximations are based on the class of flow problem and/or the *region* in which such approximations are appropriate (e.g. irrotational, boundary layer, etc.).

10-4C

Solution We are to discuss which nondimensional parameter is eliminated by use of the modified pressure.

Analysis Modified pressure effectively combines the effects of actual pressure and gravity. In the nondimensionalized Navier-Stokes equation in terms of modified pressure, **the Froude number disappears**. The reason Froude number is eliminated is because **the gravity term is eliminated from the equation.**

Discussion Keep in mind that we can employ modified pressure only for flows without free surface effects.

10-5C

Solution We are to discuss the criteria used to determine whether an approximation of the Navier-Stokes equation is appropriate or not

Analysis We determine if an approximation is appropriate **by comparing the orders of magnitude of the various terms in the equations of motion**. If the neglected terms are negligibly small compared to other terms, then the approximation is appropriate. If not, then it is not appropriate to neglect those terms.

Discussion It is important that the proper scales be used for the nondimensionalization of the equation. Otherwise, the order of magnitude analysis may be incorrect.

10-6C

Solution We are to discuss the physical significance of the four nondimensional parameters in the nondimensionalized incompressible Navier-Stokes equation.

Analysis The four parameters are discussed individually below:

- **Strouhal number**: St is the ratio of some characteristic flow time to some period of oscillation. If St << 1, the oscillation period is very large compared to the characteristic flow time, and the problem is quasi-steady; the unsteady term in the Navier-Stokes equation may be ignored. If St >> 1, the oscillation period is very short compared to the characteristic flow time, and the unsteadiness dominates the problem; the unsteady term must remain.
- **Euler number**: Eu is the ratio of a characteristic pressure difference to a characteristic pressure due to fluid inertia. If Eu << 1, pressure gradients are very small compared to inertial pressure, and the pressure term can be neglected in the Navier-Stokes equation. If Eu >> 1, the pressure term is very large compared to the inertial term, and must remain in the equation.
- **Froude number**: Fr is the ratio of inertial forces to gravitational forces. Note that Fr appears in the *denominator* of the nondimensionalized Navier-Stokes equation. If Fr << 1, gravitational forces are very large compared to inertial forces, and the gravity term must remain in the Navier-Stokes equation. If Fr >> 1, gravitational forces are negligible compared to inertial forces, and the gravity term in the Navier-Stokes equation can be ignored.
- **Reynolds number**: Re is the ratio of inertial forces to viscous forces. Note that Re appears in the *denominator* of the nondimensionalized Navier-Stokes equation. If Re << 1, viscous forces are very large compared to inertial forces, and the viscous term must remain. (In fact, it may dominate the other terms, as in creeping flow). If Re >> 1, viscous forces are negligible compared to inertial forces, and the viscous term in the Navier-Stokes equation can be ignored. Note that this applies only to regions outside of boundary layers, because the characteristic length scale for a boundary layer is generally much smaller than that for the overall flow.

Discussion You must keep in mind that the approximations discussed here are appropriate only in certain *regions* of the flow field. In other regions of the same flow field, different approximations may apply.

10-7C

Solution We are to discuss the criterion for using modified pressure.

Analysis **Modified pressure can be used only when there are *no free surface effects* in the problem**.

Discussion Modified pressure is simply a combination of thermodynamic pressure and hydrostatic pressure. It turns out that if there are **no free surface effects**, the hydrostatic pressure component is independent of the flow pressure component, and these two can be separated.

10-8C

Solution We are to discuss the most significant danger that arises with an approximate solution, and we are to come up with an example.

Analysis The danger of an approximate solution of the Navier-Stokes equation is this: **If the approximation is not appropriate to begin with, our solution will be incorrect** – even if we perform all the mathematics correctly. There are many examples. For instance, we may assume that a boundary layer exists in a region of flow. However, if the Reynolds number is not large enough, the boundary layer is too thick and the boundary layer approximations break down. Another example is that we may assume a fluid statics region, when in reality there are swirling eddies in that region. The unsteady motion of the eddies makes the problem unsteady and dynamic – the approximation of fluid statics would be inappropriate.

Discussion When you make an approximation and solve the problem, it is best to go back and verify that the approximation is appropriate.

10-9

Solution We are to write all three components of the Navier-Stokes equation in terms of modified pressure, and show that they are equivalent to the equations with regular pressure. We are also to discuss the advantage of using modified pressure.

Analysis In terms of modified pressure, the Navier-Stokes equation is written in Cartesian components as

x component:
$$\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial P'}{\partial x}+\mu\nabla^2 u \qquad (1)$$

and

y component:
$$\rho\left(\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial P'}{\partial y}+\mu\nabla^2 v \qquad (2)$$

and

z component:
$$\rho\left(\frac{\partial w}{\partial t}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial P'}{\partial z}+\mu\nabla^2 w \qquad (3)$$

The definition of modified pressure is

Modified pressure:
$$P'=P+\rho g z \qquad (4)$$

When Eq. 4 is plugged into Eqs. 1 and 2, the gravity term disappears since z is independent of x and y. The result is

x component:
$$\boxed{\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial P}{\partial x}+\mu\nabla^2 u} \qquad (5)$$

and

y component:
$$\boxed{\rho\left(\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial P}{\partial y}+\mu\nabla^2 v} \qquad (6)$$

However, when Eq. 4 is plugged into Eq. 3, the result is

z component:
$$\boxed{\rho\left(\frac{\partial w}{\partial t}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial P}{\partial z}-\rho g+\mu\nabla^2 w} \qquad (7)$$

Equations 5 through 7 are the appropriate components of the Navier-Stokes equation in terms of regular pressure, so long as gravity acts downward (in the –z direction).

The advantage of using modified pressure is that the gravity term disappears from the Navier-Stokes equation.

Discussion Modified pressure can be used only when there are no free surfaces.

10-10

Solution We are to sketch the profile of modified pressure and shade in the region representing hydrostatic pressure.

Assumptions **1** The flow is incompressible. **2** The flow is fully developed. **3** Gravity acts vertically downward. **4** There are no free surface effects in this flow field.

Analysis By definition, modified pressure $P' = P + \rho gz$. So we add hydrostatic pressure component ρgz to the given profile for P to obtain the profile for P'. Recall from Example 9-16, that for the case in which gravity does not act in the x-z plane, the pressure would be constant along any slice $x = x_1$. Thus we infer that here (with gravity), the linear increase in P as we move down vertically in the channel is due to hydrostatic pressure. Therefore, when we add ρgz to P to obtain the modified pressure, it turns out that P' is *constant* at this horizontal location.

We show two solutions in Fig. 1: (a) datum plane $z = 0$ located at the bottom wall, and (b) datum plane $z = 0$ located at the top wall. The shaded region in Fig. 1b represents the hydrostatic pressure component. P' is constant along the slice $x = x_1$ for either case, and the datum plane can be drawn at any arbitrary elevation.

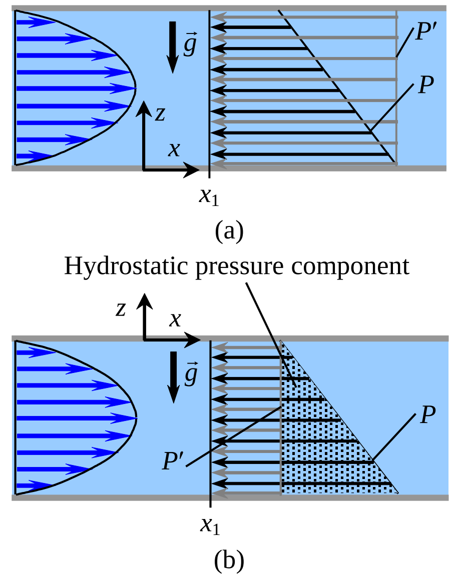


FIGURE 1
Actual pressure P (black arrows) and modified pressure P' (gray arrows) for fully developed planar Poiseuille flow. (a) Datum plane at bottom wall and (b) datum plane at top wall. The hydrostatic pressure component ρgz is the shaded area in (b).

Discussion It should be apparent why it is advantageous to use modified pressure; namely, the gravity term is eliminated from the Navier-Stokes equation, and P' is in general simpler than P.

10-11

Solution We are to discuss how modified pressure varies with downstream distance in planar Poiseuille flow.

Assumptions **1** The flow is incompressible. **2** The flow is fully developed. **3** Gravity acts vertically downward. **4** There are no free surface effects in this flow field.

Analysis For fully developed planar Poiseuille flow between two parallel plates, we know that pressure P decreases linearly with x, the distance down the channel. Modified pressure is defined as $P' = P + \rho gz$. However, since the flow is horizontal, elevation z does not change as we move axially down the channel. Thus we conclude that **modified pressure P' decreases linearly with x**. We sketch both P and P' in Fig. 1 at two axial locations, $x = x_1$ and $x = x_2$. The shaded region in Fig. 1 represents the hydrostatic pressure component ρgz. Since channel height is constant, the hydrostatic component does not change with x. P' is constant along any vertical slice, but its magnitude decreases linearly with x as sketched.

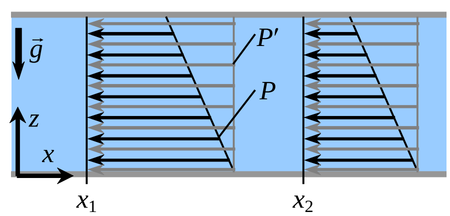


FIGURE 1
Actual pressure P (black arrows) and modified pressure P' (gray arrows) at two axial locations for fully developed planar Poiseuille flow.

Discussion The pressure gradient dP'/dx in terms of modified pressure is the same as the pressure gradient $\partial P/\partial x$ in terms of actual pressure.

10-12

Solution We are to generate an "exact" solution of the Navier-Stokes equation for fully developed Couette flow, using modified pressure. We are to compare to the solution of Chap. 9 that does *not* use modified pressure.

Assumptions We number and list the assumptions for clarity:

1 The plates are infinite in x and z (z is out of the page in the figure associated with this problem).

2 The flow is steady.

3 This is a parallel flow (we assume the y component of velocity, v, is zero).

4 The fluid is incompressible and Newtonian, and the flow is laminar.

5 Pressure P = constant with respect to x. In other words, there is no applied pressure gradient pushing the flow in the x direction; the flow establishes itself due to viscous stresses caused by the moving upper wall. In terms of modified pressure, P' is also constant with respect to x.

6 The velocity field is purely two-dimensional, which implies that $w = 0$ and

$$\frac{\partial}{\partial z}\left(\text{any velocity component}\right)=0 .$$

7 Gravity acts in the negative z direction.

Analysis To obtain the velocity and pressure fields, we follow the step-by-step procedure outlined in Chap. 9.

Step 1 Set up the problem and the geometry. See the figure associated with this problem.

Step 2 List assumptions and boundary conditions. We have already listed seven assumptions. The boundary conditions come from imposing the no slip condition: (1) At the bottom plate (y = 0), $u = v = w = 0$. (2) At the top plate ($y = h$), $u = V$, v = 0, and w = 0. (3) At z = 0, $P = P_0$, and thus $P' = P + \rho gz = P_0$.

Step 3 Write out and simplify the differential equations. We start with the continuity equation in Cartesian coordinates,

Continuity:
$$\frac{\partial u}{\partial x}+\underbrace{\cancel{\frac{\partial v}{\partial y}}}_{\text{Assumption 3}}+\underbrace{\cancel{\frac{\partial w}{\partial z}}}_{\text{Assumption 6}}=0 \qquad \text{or} \qquad \frac{\partial u}{\partial x}=0 \tag{1}$$

Equation 1 tells us that u is not a function of x. In other words, it doesn't matter where we place our origin – the flow is the same at any x location. I.e., the flow is fully developed. Furthermore, since u is not a function of time (Assumption 2) or z (Assumption 6), we conclude that u is at most a function of y,

Result of continuity:
$$u=u\left(y\right)\ \text{only} \tag{2}$$

We now simplify the x momentum equation as far as possible:

$$\rho\left(\underbrace{\cancel{\frac{\partial u}{\partial t}}}_{\text{Assumption 2}}+\underbrace{u\cancel{\frac{\partial u}{\partial x}}}_{\text{Continuity}}+\underbrace{\cancel{v\frac{\partial u}{\partial y}}}_{\text{Assumption 3}}+\underbrace{\cancel{w\frac{\partial u}{\partial z}}}_{\text{Assumption 6}}\right)=\underbrace{-\cancel{\frac{\partial P'}{\partial x}}}_{\text{Assumption 5}}+\mu\left(\underbrace{\cancel{\frac{\partial^2 u}{\partial x^2}}}_{\text{Continuity}}+\frac{\partial^2 u}{\partial y^2}+\underbrace{\cancel{\frac{\partial^2 u}{\partial z^2}}}_{\text{Assumption 6}}\right) \rightarrow \frac{d^2u}{dy^2}=0 \tag{3}$$

All other terms in Eq. 3 have disappeared except for a lone viscous term, which must then itself equal zero. Notice that we have changed from a partial derivative ($\partial/\partial y$) to a total derivative (d/dy) in Eq. 3 as a direct result of Eq. 2. We do not show the details here, but you can show in similar fashion that every term except the pressure term in the y momentum equation goes to zero, forcing that lone term to also be zero,

y momentum:
$$\frac{\partial P'}{\partial y}=0 \tag{4}$$

The same thing happens to the z momentum equation; the result is

z momentum:
$$\frac{\partial P'}{\partial z}=0 \tag{5}$$

In other words, P' is not a function of y or z. Since P' is also not a function of time (Assumption 2) or x (Assumption 5), P' is a constant,

Result of y and z momentum:
$$P'=\ \text{constant}=C_3 \tag{6}$$

Step 4 Solve the differential equations. Continuity, y momentum, and z momentum have already been "solved", resulting in Eqs. 2 and 6. Equation 3 (x momentum) is integrated twice to get

Integration of x momentum: $$u = C_1 y + C_2 \qquad (7)$$

where C_1 and C_2 are constants of integration.

Step 5 We apply boundary condition (3), $P' = P_0$ at $z = 0$. Eq. 6 yields $C_3 = P_0$, and

Final solution for pressure field: $$\boxed{P' = P_0 \quad \rightarrow \quad P = P_0 - \rho g z} \qquad (9)$$

We next apply boundary conditions (1) and (2) to obtain constants C_1 and C_2.

Boundary condition (1): $$u = C_1(0) + C_2 = 0 \qquad \text{or} \qquad C_2 = 0$$

and

Boundary condition (2): $$u = C_1(h) + 0 = V \qquad \text{or} \qquad C_1 = \frac{V}{h}$$

Finally, Eq. 7 becomes

Final result for velocity field: $$\boxed{u = V\frac{y}{h}} \qquad (10)$$

The velocity field reveals a simple linear velocity profile from $u = 0$ at the bottom plate to $u = V$ at the top plate.

Step 6 Verify the results. You can plug in the velocity and pressure fields to verify that all the differential equations and boundary conditions are satisfied.

We verify that **the results are identical to those of Example 9-15**. Thus, we get the same result using modified pressure throughout the calculation as we do using the regular (thermodynamic) pressure throughout the calculation.

Discussion Since there are no free surfaces in this problem, the gravity term in the Navier-Stokes equation is absorbed into the modified pressure, and the pressure and gravity terms are combined into one term. This is possible since flow pressure and hydrostatic pressure are uncoupled.

10-13

Solution We are to plug the given scales for this flow problem into the nondimensionalized Navier-Stokes equation to show that only two terms remain in the region consisting of most of the tank.

Assumptions **1** The flow is incompressible. **2** $d << D$. **3** D is of the same order of magnitude as H.

Analysis The characteristic frequency is taken as the inverse of the characteristic time, $f = 1/t_{\text{drain}}$. The Strouhal number is thus

Strouhal number:
$$\text{St} = \frac{fL}{V} = \frac{H}{t_{\text{drain}}V} \sim 1 \qquad (1)$$

St is of order of magnitude 1 since the order of magnitude of t_{drain} is H/V. The Euler number is

Euler number:
$$\text{Eu} = \frac{P_0 - P_\infty}{\rho V^2} = \frac{\rho g H}{\rho V^2} \sim \frac{V_{\text{jet}}^{\ 2}}{V^2} \sim \frac{D^4}{d^4} \qquad (2)$$

where we have used the order of magnitude estimate that $V_{\text{jet}} \sim \sqrt{gH}$. We have also used conservation of mass, namely $V_{\text{jet}}d^2 = V_{\text{tank}}D^2$. Similarly, the Froude number is

Froude number:
$$\text{Fr} = \frac{V}{\sqrt{gH}} \sim \frac{V}{V_{\text{jet}}} \sim \frac{d^2}{D^2} \qquad (3)$$

Finally, the Reynolds number is

Reynolds number:
$$\text{Re} = \frac{\rho V H}{\mu} \sim \frac{\rho V D}{\mu} = \frac{\rho V_{\text{jet}} D}{\mu}\frac{V}{V_{\text{jet}}} \sim \text{Re}_{\text{jet}}\frac{d^2}{D^2} \qquad (4)$$

We plug Eqs. 1 through 4 into the nondimensionalized incompressible Navier-Stokes equation and compare orders of magnitude of each term,

Nondimensionalized incompressible Navier-Stokes equation:
$$\underbrace{[\text{St}]\frac{\partial \vec{V}^*}{\partial t^*}}_{\sim 1} + \underbrace{\left(\vec{V}^*\cdot\vec{\nabla}^*\right)\vec{V}^*}_{\sim 1} = \underbrace{-[\text{Eu}]\vec{\nabla}^*P^*}_{\sim \frac{D^2}{d^2}} + \underbrace{\left[\frac{1}{\text{Fr}^2}\right]\vec{g}^*}_{\sim \frac{D^2}{d^2}} + \underbrace{\left[\frac{1}{\text{Re}}\right]\vec{\nabla}^{*2}\vec{V}^*}_{\sim \text{Re}_{\text{jet}}\frac{d^2}{D^2}} \qquad (5)$$

Clearly, the first two terms (the unsteady and inertial terms) in Eq. 5 are negligible compared to the second two terms (the pressure and gravity terms) since $D >> d$. The last term (the viscous term) is a little trickier. We know that if the flow remains laminar, the order of magnitude of Re_{jet} is at most 10^3. Thus, in order for the viscous term to be of the same order of magnitude as the inertial term, d^2/D^2 must be of order of magnitude 10^{-3}. Thus, provided that these criteria are met, the only two remaining terms in the Navier-Stokes equation are the pressure and gravity terms. The final dimensional form of the equation is the same as that of fluid statics,

Incompressible Navier-Stokes equation for fluid statics:
$$\vec{\nabla}P = \rho\vec{g} \qquad (6)$$

The criteria for Carrie's approximation to be appropriate depends on the desired precision. For 1% error, D must be at least 10 times greater than d to ignore the unsteady term and the inertial term. The viscous term, however, depends on the value of Re_{jet}. To be safe, Carrie should assume the highest possible value of Re_{jet}, for which we know from the above order of magnitude estimates that D must be at least $10^{3/2}$ times greater than d.

Discussion We cannot use the modified pressure in this problem since there is a free surface.

10-14

Solution We are to sketch the profile of actual pressure and shade in the region representing hydrostatic pressure.

Assumptions **1** The flow is incompressible. **2** Gravity acts vertically downward. **3** There are no free surface effects in this flow field.

Analysis By definition, modified pressure $P' = P + \rho gz$. Thus, to obtain actual pressure P, we subtract the hydrostatic component ρgz from the given profile of P'. Using the given value of P at the mid-way point as a guide, **we sketch the actual pressure in Fig. 1** such that the difference between P' and P increases linearly. In other words, we subtract the hydrostatic pressure component ρgz from the modified pressure P' to obtain the profile for actual pressure P.

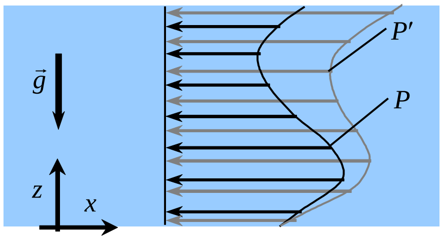


FIGURE 1
Actual pressure P (black arrows) and modified pressure P' (gray arrows) for the given pressure field.

Discussion We assume that there are no free surface effects in the problem; otherwise modified pressure should not be used. The datum plane is set in the problem statement, but any arbitrary elevation could be used instead. If the datum plane were set at the top of the domain, P' would be less than P everywhere because of the negative values of z in the transformation from P to P'.

10-15

Solution We are to solve the Navier-Stokes equation in terms of modified pressure for the case of steady, fully developed, laminar flow in a round pipe. We are to obtain expressions for the pressure and velocity fields, and compare the actual pressure at the top of the pipe to that at the bottom of the pipe.

Assumptions We make the same assumptions as in Example 9-18, except we use modified pressure P' in place of actual pressure P.

Analysis The Navier-Stokes equation with gravity, written in terms of modified pressure P', is identical to the Navier-Stokes equation with no gravity, written in terms of actual pressure P. In other words, all of the algebra of Example 9-18 remains the same, except we use modified pressure P' in place of actual pressure P. The velocity field does not change, and the result is

Axial velocity field:
$$\boxed{u = \frac{1}{4\mu}\frac{dP}{dx}\left(r^2 - R^2\right)} \qquad (1)$$

The modified pressure field is

Modified pressure field:
$$P' = P'(x) = P_1' + \frac{dP'}{dx}dx \qquad (2)$$

where P_1' is the modified pressure at location $x = x_1$. In Example 9-18, the actual pressure varies only with x. In fact it decreases linearly with x (note that the pressure gradient is negative for flow from left to right). Here, Eq. 2 shows that modified pressure behaves in the same way, namely P' varies only with x, and in fact decreases linearly with x.

We simply subtract the hydrostatic pressure component ρgz from modified pressure P' (Eq. 2) to obtain the final expression for actual pressure P,

Actual pressure field:
$$\boxed{P = P' - \rho gz \quad \rightarrow \quad P = P_1' + \frac{dP'}{dx}dx - \rho gz} \qquad (3)$$

Since the pipe is horizontal, the bottom of the pipe is lower than the top of the pipe. Thus, z_{top} is greater than z_{bottom}, and therefore by Eq. 3 P_{top} *is less than* P_{bottom}. This agrees with our experience that pressure increases downward.

Discussion Since there are no free surfaces in this flow, the gravity term does not directly influence the velocity field, and a hydrostatic component is added to the pressure field. You can see the advantage of using modified pressure.

Creeping Flow

10-16C

Solution We are to discuss why density is not a factor in aerodynamic drag on a particle in creeping flow.

Analysis It turns out that fluid density drops out of the creeping flow equations, since **the terms that contain ρ in the Navier-Stokes equation are negligibly small compared to the pressure and viscous terms (which do not contain ρ)**. Another way to think about this is: In creeping flow, there is no fluid inertia, and since inertia is associated with fluid mass (density), density cannot contribute to the aerodynamic drag on a particle moving in creeping flow. In creeping flow, there is a balance between pressure forces and viscous forces, neither of which depend on fluid density.

Discussion Density does have an *indirect* influence on creeping flow drag. Namely, ρ is needed in the Reynolds number calculation, and Re determines whether the flow is in the creeping flow regime or not.

10-17C

Solution We are to name each term in the Navier-Stokes equation, and then discuss which terms remain when the creeping flow approximation is made.

Analysis The terms in the equation are identified as follows:

- **I** ***Unsteady*** **term**
- **II** ***Inertial*** **term**
- **III** ***Pressure*** **term**
- **IV** ***Gravity*** **term**
- **V** ***Viscous*** **term**

When the creeping flow approximation is made, **only terms III (pressure) and V (viscous) remain**. The other three terms are very small compared to these two and can be ignored. The significance is that all unsteady and inertial effects (terms I and II) have disappeared, as has gravity. We are left with a flow in which pressure forces and viscous forces must balance. Another significant result is that density has disappeared from the creeping flow equation, as discussed in the text.

Discussion There are other acceptable one-word descriptions of some of the terms in the equation. For example, the inertial term can also be called the convective term, the advective term, or the acceleration term.

10-18

Solution Aluminum balls of different diameters are dropped into a tank filled with glycerin. Experimental ball velocities are to be compared with theoretical ones.

Analysis The free body diagram is shown in the figure. Let's apply Newton's second law in the vertical direction:

$$F_{net} = ma$$

$$m_s g - F_D - F_B = m\frac{dV}{dt} \quad (1)$$

where m_s is the mass of ball, F_D the drag force, F_B the buoyancy force, and D the ball velocity at any time.

$$\rho_s \frac{\pi D^3}{6} g - \rho_f g \frac{\pi D^3}{6} - 3\pi\mu DV = \rho_s \frac{\pi D^3}{6}\frac{dV}{dt} \qquad (2)$$

After some manipulations we get

$$g\left(1 - \frac{\rho_f}{\rho s}\right) - \frac{18\mu}{\rho_s D^2} V = \frac{dV}{dt} \qquad (3)$$

Let's introduce two constants such A, B. Eq. 3 is then

$$C_1 - C_2 V = \frac{dV}{dt} \quad \text{or} \quad \frac{dV}{C_1 - C_2 V} = dt \qquad (4)$$

By integrating Eq. 4 we obtain

$$t = \frac{-\ln(C_1 - C_2 V)}{C_2} \qquad (5)$$

or

$$V(t) = \frac{1}{C_2}\left(C_1 - e^{-C_2 t}\right) = \frac{\rho_s D^2}{18\mu}\left[\left(1 - \frac{\rho_f}{\rho_s}\right)g - EXP\left(-\frac{18\mu}{\rho_s D^2}t\right)\right] \qquad (6)$$

Comparisons:

For D=2 mm, using Eq. 6 we obtain: V=3.13 mm/s: Error= $\frac{3.2 - 3.13}{3.2} * 100 = 2.2\ \%$.

For D=4 mm, using Eq. 6 we obtain: V=12.54 mm/s: Error= $\frac{12.8 - 12.54}{12.8} * 100 = 2.03\ \%$

For D=10 mm, using Eq. 6 we obtain: V=78.4 mm/s: Error= $\frac{60.4 - 78.4}{60.4} * 100 = -28.8\ \%$

There is a good agreement for the first two diameters. However the error for third one is not acceptable. Let's check Re number:

$$\text{Re} = \frac{1260 \times 78.4 \times 10^{-3} * 10 \times 10^{-3}}{1.0} = 0.988$$

Reynold's number seems to be beyond the range of validity of the Stokes' equation. This probably causes too large error in the prediction. If we used the general form of the equation (see Prob. 10–2) we would find V=60.1 mm/s, which is pretty close to the experimental result (60.4 mm/s).

10-19

Solution Aluminum balls of different diameters are dropped into a tank filled with glycerin. Experimental ball velocities are to be compared with theoretical ones.

Analysis For this case the differential equation will be in the following form:

$$m_s g - F_D - F_B = m\frac{dV}{dt}$$

$$\rho_s \frac{\pi D^3}{6} g - \rho_f g \frac{\pi D^3}{6} - 3\pi\mu DV - \frac{9\pi}{16}\rho D^2 V^2 = \rho_s \frac{\pi D^3}{6}\frac{dV}{dt} \text{ or}$$

$$g\left(1 - \frac{\rho_f}{\rho s}\right) - \frac{18\mu}{\rho_s D^2} V - \frac{3.375}{D} V^2 = \frac{dV}{dt}$$

Introducing some constants yields

$$C_1 - C_2 V - C_3 V^2 = \frac{dV}{dt},$$

or by separating variables we obtain

$$\frac{dV}{C_1 - C_2 V - C_3 V^2} = dt$$

By integrating we get

$$t = \frac{2\tanh^{-1}\left(\dfrac{2C_3 V + C_2}{\sqrt{4C_1 C_3 + C_2^2}}\right)}{\sqrt{4C_1 C_3 + C_2^2}}$$

This equation can also be solved for V as below:

$$V = V(t) = \frac{-C_2 + \beta \tanh\left(\dfrac{\beta t}{2}\right)}{2C_3},$$

where

$$\beta = \sqrt{4C_1 C_3 + C_2^2}$$

10-20

Solution We are to estimate the maximum speed of honey through a hole such that the Reynolds number remains below 0.1, at two different temperatures.

Analysis The density of honey is equal to its specific gravity times the density of water,

Density of honey: $$\rho_{\text{honey}} = SG_{\text{honey}}\rho_{\text{water}} = 1.42\left(998.0\ \text{kg/m}^3\right) = 1420\ \text{kg/m}^3 \qquad (1)$$

We convert the viscosity of honey from poise to standard SI units,

Viscosity of honey at 20°:

$$\mu_{\text{honey}} = 190\ \text{poise}\left(\frac{\text{g}}{\text{cm}\cdot\text{s}\cdot\text{poise}}\right)\left(\frac{\text{kg}}{1000\ \text{g}}\right)\left(\frac{100\ \text{cm}}{\text{m}}\right) = 19.0\frac{\text{kg}}{\text{m}\cdot\text{s}} \qquad (2)$$

Finally, we plug Eqs. 1 and 2 into the definition of Reynolds number, and set Re = 0.1 to solve for the maximum speed to ensure creeping flow,

Maximum speed for creeping flow at 20°

$$V_{\max} = \frac{\text{Re}_{\max}\,\mu_{\text{honey}}}{\rho_{\text{honey}}D} = \frac{(0.1)(19.0\ \text{kg/m}\cdot\text{s})}{(1420\ \text{kg/m}^3)(0.0060\ \text{m})} = \mathbf{0.22\ m/s} \qquad (3)$$

At the higher temperature of 50°C, the calculations yield $V_{\max}$ = 0.01174 ≈ **0.012 m/s**. Thus, it is much easier to achieve creeping flow with honey at lower temperatures since the viscosity of honey increases rapidly as the temperature drops.

Discussion We used Re < 0.1 as the maximum Reynolds number for creeping flow, but experiments reveal that in many flows, the creeping flow approximation is acceptable at Reynolds numbers as high as nearly 1.0.

10-21

Solution We are to compare the number of body lengths per second of a swimming human and a swimming sperm.

Analysis We let *BLPS* denote "body lengths per second". For the human swimmer,

Human: $$BLPS_{\text{human}} = \frac{100\ \text{m/min}}{1.85\ \text{m/body length}}\left(\frac{\text{min}}{60\ \text{s}}\right) = \mathbf{0.90\ body\ length/s}$$

For the sperm, we use the speed calculated in Problem 10-19. The total body length of the sperm (head and tail) is about 40 μm, as measured from the figure.

Sperm: $$BLPS_{\text{sperm}} = \frac{1.5\times10^{-4}\ \text{m/s}}{40\ \mu\text{m/body length}}\left(\frac{10^6\ \mu\text{m}}{\text{m}}\right) = \mathbf{3.8\ body\ length/s}$$

So, on an equal basis of comparison, the sperm swims faster than the human! This result is perhaps surprising since the human benefits from inertia, while the sperm feels no inertial effects. However, we must keep in mind that the sperm's body is designed to swim, while the human body is designed for multiple uses – it is not optimized for swimming.

Note: Students' answers may differ widely since the measurements from the photograph are not very accurate.

Discussion Perhaps a more fair comparison would be between a *fish* and a sperm.

10-22

Solution We are to calculate how fast air must move vertically to keep a water drop suspended in the air.

Assumptions **1** The drop is spherical. **2** The creeping flow approximation is appropriate.

Properties For air at T = 25°C, ρ = 1.184 kg/m^3 and μ = 1.849 × 10^{-5} kg/m·s. The density of the water at T = 25°C is 997.0 kg/m^3.

Analysis Since the drop is sitting still, its downward force must exactly balance its upward force when the vertical air speed V is "just right". The downward force is the weight of the particle:

Downward force on the particle:
$$F_{\text{down}} = \pi \frac{D^3}{6} \rho_{\text{particle}} g \tag{1}$$

The upward force is the aerodynamic drag force acting on the particle plus the buoyancy force on the particle. The aerodynamic drag force is obtained from the creeping flow drag on a sphere,

Upward force on the particle:
$$F_{\text{up}} = 3\pi\mu VD + \pi \frac{D^3}{6} \rho_{\text{air}} g \tag{2}$$

We equate Eqs. 1 and 2, i.e., $F_{\text{down}} = F_{\text{up}}$,

Balance:
$$\pi \frac{D^3}{6}\left(\rho_{\text{particle}} - \rho_{\text{air}}\right) g = 3\pi\mu VD$$

and solve for the required air speed V,

$$V = \frac{D^2}{18\mu}\left(\rho_{\text{particle}} - \rho_{\text{air}}\right) g = \frac{\left(42.5\times10^{-6}\ \text{m}\right)^2}{18\left(1.849\times10^{-5}\ \text{kg/m}\cdot\text{s}\right)}\left[\left(998.0-1.184\right)\ \text{kg/m}^3\right]\left(9.81\ \text{m/s}^2\right) = \mathbf{0.0531\ m/s}$$

Finally, we must verify that the Reynolds number is small enough that the creeping flow approximation is appropriate.

Check of Reynolds number:
$$\text{Re} = \frac{\rho_{\text{air}} VD}{\mu} = \frac{\left(1.184\ \text{kg/m}^3\right)\left(0.0531\ \text{m/s}\right)\left(42.5\times10^{-6}\ \text{m}\right)}{1.849\times10^{-5}\ \text{kg/m}\cdot\text{s}} = 0.144$$

Since Re << 1, The creeping flow approximation is appropriate, although it is not as small as we'd like to be real confident in the creeping flow approximation.

Discussion Notice that although air density does appear in the calculation of V, it is very small compared to the density of water. (If we ignore ρ_{air} in that calculation, we get the same answer to 3 significant digits. However, ρ_{air} *is* required in the calculation of Reynolds number – to verify that the creeping flow approximation is appropriate.

10-23

Solution We are to generate a characteristic pressure scale for flow through a slipper-pad bearing.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional in the *x*-*y* plane. **3** The creeping flow approximation is appropriate.

Analysis The *x* component of the creeping flow momentum equation is

x momentum:
$$\frac{\partial P}{\partial x} \approx \mu \nabla^2 u \quad \rightarrow \quad \frac{\partial P}{\partial x} \approx \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \underbrace{\cancel{\frac{\partial^2 u}{\partial z^2}}}_{2-D} \right)$$

We plug in the characteristic scales to get

Orders of magnitude:
$$\underbrace{\frac{\partial P}{\partial x}}_{\frac{\Delta P}{L}} \approx \mu \underbrace{\frac{\partial^2 u}{\partial x^2}}_{\frac{V}{L^2}} + \mu \underbrace{\frac{\partial^2 u}{\partial y^2}}_{\frac{V}{h_0^2}} \qquad (1)$$

The first term on the right of Eq. 1 is clearly much smaller than the second term on the right since $h_0 << L$. Equating the orders of magnitude of the two remaining terms,

Characteristic pressure scale:
$$\frac{\Delta P}{L} \sim \mu \frac{V}{h_0^2} \rightarrow \boxed{\Delta P \sim \frac{\mu V L}{h_0^2}} \qquad (2)$$

Discussion The characteristic pressure scale differs from that in the text because there are two length scales in this problem rather than just one.

10-24

Solution We are to find a characteristic velocity scale for v, compare the inertial terms of the x momentum equation to the pressure and viscous terms, and discuss how the creeping flow equations can still be used even if Re is not small.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional in the x-y plane. **3** Gravity forces are negligible.

Analysis (a) We use the continuity equation to obtain the characteristic velocity scale for v,

Continuity:

$$\underbrace{\frac{\partial u}{\partial x}}_{\frac{V}{L}}+\underbrace{\frac{\partial v}{\partial y}}_{\frac{v}{h_0}}=0 \rightarrow \boxed{v \sim \frac{Vh_0}{L}} \tag{1}$$

(b) We analyze the orders of magnitude of each term in the steady, 2-D, incompressible x momentum equation without gravity,

x momentum:

$$\underbrace{\rho u\frac{\partial u}{\partial x}}_{\frac{\rho V^2}{L}}+\underbrace{\rho v\frac{\partial u}{\partial y}}_{\rho\frac{Vh_0}{L}\frac{V}{h_0}=\frac{\rho V^2}{L}}=-\underbrace{\frac{\partial P}{\partial x}}_{\frac{\mu V}{h_0^2}}+\underbrace{\mu\frac{\partial^2 u}{\partial x^2}}_{\frac{\mu V}{L^2}}+\underbrace{\mu\frac{\partial^2 u}{\partial y^2}}_{\frac{\mu V}{h_0^2}} \tag{2}$$

where we have also used the result of Problem 10-23. The first viscous term of Eq. 2 is clearly much smaller than the second viscous term since $h_0 << L$. We multiply the order of magnitude of all the remaining terms by $L/(\rho V^2)$ to compare terms,

Comparison of orders of magnitude:

$$\underbrace{\rho u\frac{\partial u}{\partial x}}_{1}+\underbrace{\rho v\frac{\partial u}{\partial y}}_{1}=-\underbrace{\frac{\partial P}{\partial x}}_{\frac{\mu}{\rho V h_0}\frac{L}{h_0}}+\underbrace{\mu\frac{\partial^2 u}{\partial y^2}}_{\frac{\mu}{\rho V h_0}\frac{L}{h_0}} \tag{3}$$

We recognize the Reynolds number based on gap height, Re = $\rho V h_0/\mu$. Since the pressure and viscous terms contain the product of 1/Re, which is large for creeping flow, and L/h_0, which is also large, it is clear that **the inertial terms (left side of Eq. 3) are negligibly small compared to the pressure and viscous terms**.

(c) Since the pressure and viscous terms contain the product of 1/Re and L/h_0, when $h_0 << L$, **the creeping flow equations can still be appropriate even if Reynolds number is not less than one**. For example, if $L/h_0 \sim 10{,}000$ and Re ~ 10, the pressure and viscous terms are still three orders of magnitude larger than the inertial terms.

Discussion In the limit as $L/h_0 \rightarrow \infty$, the inertial terms disappear regardless of the Reynolds number. This limiting case is the Couette flow problem of Chap. 9.

10-25

Solution We are to analyze the *y* momentum equation by order of magnitude analysis, and we are to comment about the pressure gradient $\partial P/\partial y$.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional in the *x*-*y* plane. **3** Gravity forces are negligible.

Analysis We analyze the orders of magnitude of each term in the steady, 2-D, incompressible *y* momentum equation without gravity,

y momentum:

$$\underbrace{\rho u\frac{\partial v}{\partial x}}_{\rho V\frac{Vh_0}{L}\frac{1}{L}=\frac{\rho V^2 h_0}{L^2}} + \underbrace{\rho v\frac{\partial v}{\partial y}}_{\rho\frac{V^2h_0^2}{L^2}\frac{1}{h_0}=\frac{\rho V^2 h_0}{L^2}} = -\underbrace{\frac{\partial P}{\partial y}}_{\frac{\mu VL}{h_0^3}} + \underbrace{\mu\frac{\partial^2 v}{\partial x^2}}_{\mu\frac{Vh_0}{L}\frac{1}{L^2}=\frac{\mu Vh_0}{L^3}} + \underbrace{\mu\frac{\partial^2 v}{\partial y^2}}_{\mu\frac{Vh_0}{L}\frac{1}{h_0^2}=\frac{\mu V}{Lh_0}} \tag{1}$$

The first viscous term of Eq. 1 is clearly much smaller than the second viscous term, since $h_0 << L$. We multiply the order of magnitude of all the remaining terms by $L^2/(\rho V^2 h_0)$ to compare terms,

Comparison of orders of magnitude:

$$\underbrace{\rho u\frac{\partial v}{\partial x}}_{1} + \underbrace{\rho v\frac{\partial v}{\partial y}}_{1} = -\underbrace{\frac{\partial P}{\partial y}}_{\frac{\mu}{\rho Vh_0}\left(\frac{L}{h_0}\right)^3} + \underbrace{\mu\frac{\partial^2 v}{\partial y^2}}_{\frac{\mu}{\rho Vh_0}\left(\frac{L}{h_0}\right)} \tag{2}$$

We recognize the Reynolds number based on gap height, $\mathrm{Re} = \rho Vh_0/\mu$. Since the pressure and viscous terms contain the product of 1/Re, which is large for creeping flow, and L/h_0, which is also large, it is clear that the inertial terms (left side of Eq. 2) are negligibly small compared to the pressure and viscous terms. This is expected, of course, for creeping flow. Now we compare the pressure and viscous terms. Both contain 1/Re, but the pressure term has and additional factor of $(L/h_0)^2$, which is very large. Thus the pressure term is the only remaining term in Eq. 2. How can this be? Since there are no terms that can balance the pressure term, the pressure term itself must be very small. In other words, the *y* momentum equation reduces to

Final form of y momentum:

$$\boxed{\frac{\partial P}{\partial y} \approx 0} \tag{3}$$

In other words, **pressure is a function of *x*, but a very weak, negligible function of *y***.

Discussion The result here is very similar to that for boundary layers, where we also find that $\partial P/\partial y \approx 0$ through the boundary layer.

10-26

Solution We are to list boundary conditions and solve the *x* momentum equation for *u*. Then we are to nondimensionalize our result.

Assumptions **1** The flow is steady and incompressible. **2** Gravity forces are negligible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** *P* is not a function of *y*.

Analysis (*a*) From the figure associated with this problem, we write two boundary conditions on *u*,

Boundary condition (1):
$$\boxed{u = V \text{ at } y = 0 \text{ for all } x} \qquad (1)$$

and

Boundary condition (2):
$$\boxed{u = 0 \text{ at } y = h \text{ for all } x} \qquad (2)$$

We note that *h* is not a constant, but rather a function of *x*.

(*b*) We write the creeping flow *x* momentum equation, and integrate once with respect to *y*, noting that *P* is not a function of *y*. This is a *partial* integration.

Integration of x momentum:
$$\frac{\partial^2 u}{\partial y^2} = \frac{1}{\mu}\frac{dP}{dx} \quad \rightarrow \quad \frac{\partial u}{\partial y} = \frac{1}{\mu}\frac{dP}{dx}y + f_1(x)$$

We integrate again to obtain

Second integration:
$$u = \frac{1}{2\mu}\frac{dP}{dx}y^2 + yf_1(x) + f_2(x) \qquad (3)$$

We apply boundary conditions to find the two unknown functions of *x*. From Eq. 1,

Result of boundary condition (1):
$$f_2(x) = V$$

and from Eq. 2,

Result of boundary condition (2):
$$f_1(x) = \frac{-V - \dfrac{1}{2\mu}\dfrac{dP}{dx}h^2}{h}$$

From these, the final expression for *u* is obtained,

Final expression for u, dimensional:
$$\boxed{u(x,y) = V\left(1 - \frac{y}{h}\right) + \frac{h^2}{2\mu}\frac{dP}{dx}\frac{y}{h}\left(\frac{y}{h} - 1\right)} \qquad (4)$$

We recognize two distinct components of the velocity profile in Eq. 4, namely *a Couette flow component and a Poiseuille flow component*. Thus, the axial velocity is a superposition of Couette flow due to the moving bottom wall and Poiseuille flow due to the pressure gradient.

(*c*) We nondimensionalize Eq. 4 by applying the nondimensional variables given in the problem statement. After some algebra,

Nondimensional expression for u:
$$\boxed{u^* = (1 - y^*) + \frac{h^{*2}}{2}\frac{dP^*}{dx^*}y^*(y^* - 1)} \qquad (5)$$

Discussion Although we have a final expression for *u*, it is in terms of the pressure gradient *dP*/*dx*, which is not known. Pressure boundary conditions and further algebra are required to solve for the pressure field.

10-27

Solution We are to generate an expression for axial velocity for a slipper-pad bearing with arbitrary gap shape.

Assumptions **1** The flow is steady and incompressible. **2** Gravity forces are negligible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** *P* is not a function of *y*.

Analysis In the solution of Problem 10-26, we never used the fact that *h*(*x*) was linear. In fact, our solution is in terms of *h*(*x*), the specific form of which was never specified. Thus, the solution of Problem 10-26 is still appropriate, and no further work needs to be done here. The result is

Expression for u for arbitrary h(x):
$$\boxed{u(x,y)=V\left(1-\frac{y}{h}\right)+\frac{h^2}{2\mu}\frac{dP}{dx}\frac{y}{h}\left(\frac{y}{h}-1\right)} \qquad (1)$$

Discussion As gap height *h*(*x*) changes, so does the pressure distribution.

10-28

Solution We are to prove the given equation for the slipper-pad bearing.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional in the *x*-*y* plane.

Analysis We solve the 2-D continuity equation for *v* by integration,

Continuity:
$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \quad \rightarrow \quad \int_0^h \frac{\partial v}{\partial y}dy=-\int_0^h \frac{\partial u}{\partial x}dy \quad \rightarrow \quad v(h)-v(0)=-\int_0^h \frac{\partial u}{\partial x}dy \qquad (1)$$

But the no-slip condition tells us that $v = 0$ at both the bottom ($y = 0$) and top ($y = h$) plates. Thus Eq. 1 reduces to

Result of continuity:
$$\int_0^h \frac{\partial u}{\partial x}dy=0 \qquad (2)$$

The 1-D Leibnitz theorem is discussed in Chap. 4 and is repeated here:

1-D Leibnitz theorem:
$$\frac{d}{dx}\int_{a(x)}^{b(x)} G(x,y)\,dy=\int_a^b \frac{\partial G}{\partial x}dy+\frac{db}{dx}G(x,b)-\frac{da}{dx}G(x,a) \qquad (3)$$

In our case (comparing Eqs. 2 and 3), $a = 0$, $b = h(x)$, and $G = u$. Thus,

$$\frac{d}{dx}\int_0^h u\,dy=\int_0^h \frac{\partial u}{\partial x}dy+\frac{dh}{dx}u(x,h) \qquad (4)$$

But $u(h) = 0$ for all values of *x* (no-slip condition). Finally then, we combine Eqs. 2 and 4 to yield the desired result,

Final result:
$$\boxed{\frac{d}{dx}\int_0^h u\,dy=0}$$

Discussion This result could also be obtained by control volume conservation of mass. Now we finally have the means of calculating the pressure distribution in the slipper-pad bearing.

10-29

Solution We are to prove the given equation for flow through a 2-D slipper-pad bearing.

Assumptions **1** The flow is steady and incompressible. **2** Gravity forces are negligible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** *P* is not a function of *y*.

Analysis We substitute the expression for *u* from Problem 10-26 into the equation of Problem 10-28,

$$\frac{d}{dx}\int_0^h u\,dy = 0 \quad \rightarrow \quad \frac{d}{dx}\int_0^h \left[V\left(1-\frac{y}{h}\right)+\frac{h^2}{2\mu}\frac{dP}{dx}\frac{y}{h}\left(\frac{y}{h}-1\right)\right]dy = 0 \qquad (1)$$

The integral in Eq. 1 is easily evaluated since both *h* and *dP*/*dx* are functions of *x* only. After some algebra,

$$\frac{d}{dx}\left[V\frac{h}{2}-\frac{h^3}{12\mu}\frac{dP}{dx}\right]=0 \qquad (2)$$

Finally, we take the *x* derivative, recognizing that *h* and *dP*/*dx* are functions of *x*,

Steady, 2-D Reynolds equation for lubrication:

$$\boxed{\frac{d}{dx}\left(h^3\frac{dP}{dx}\right)=6\mu V\frac{dh}{dx}} \qquad (3)$$

Discussion For a given geometry (*h* as a known function of *x*), we can integrate Eq. 3 to obtain the pressure distribution along the slipper-pad bearing.

10-30

Solution We are to find the pressure distribution for flow through a 2-D slipper-pad bearing with linearly decreasing gap height and atmospheric pressure at both ends of the slipper-pad.

Assumptions **1** The flow is steady and incompressible. **2** Gravity forces are negligible. **3** The flow is two-dimensional in the *x*-*y* plane. **4** *P* is not a function of *y*.

Analysis We integrate the Reynolds equation of Problem 10-29, and rearrange:

First integration:

$$h^3\frac{dP}{dx}=6\mu Vh+C_1 \quad \rightarrow \quad \frac{dP}{dx}=6\mu Vh^{-2}+C_1h^{-3} \qquad (1)$$

where C_1 is a constant of integration. Next we substitute the given equation for *h*,

$$\frac{dP}{dx}=6\mu V\left(h_0+\alpha x\right)^{-2}+C_1\left(h_0+\alpha x\right)^{-3} \qquad (2)$$

Equation 2 is in the desired form, i.e., *dP*/*dx* as a function of *x*. We integrate Eq. 2,

Second integration:

$$P=-\frac{6\mu V}{\alpha}\left(h_0+\alpha x\right)^{-1}-\frac{C_1}{2\alpha}\left(h_0+\alpha x\right)^{-2}+C_2 \qquad (3)$$

where C_2 is a second constant of integration. We plug in the two boundary conditions on *P* to find constants C_1 and C_2, namely $P = P_{atm}$ at $x = 0$ and $P = P_{atm}$ at $x = L$. After some algebra, the results are

Constants:

$$C_1=-\frac{12\mu Vh_0h_L}{h_0+h_L} \quad \text{and} \quad C_2=P_{atm}+\frac{6\mu V}{\alpha\left(h_0+h_L\right)} \qquad (4)$$

with which we generate our final expression for *P* from Eq. 3. After some algebra,

Pressure distribution:

$$\boxed{P=P_{atm}+6\mu Vx\left[\frac{h_0-h_L+\alpha x}{\left(h_0+h_L\right)\left(h_0+\alpha x\right)^2}\right]} \qquad (5)$$

Discussion There are other equivalent ways to write the expression for *P*, but Eq. 5 is about as compact as we can get.

10-31E

Solution We are to calculate α, we are to calculate P_{gage} at a given x location, and we are to plot nondimensional gage pressure as a function of nondimensional axial distance for the case of a slipper-pad bearing with linearly decreasing gap height. Finally, we are to estimate the total force that this slipper-pad bearing can support.

Assumptions **1** The flow is steady and incompressible. **2** Gravity forces are negligible in the oil flow. **3** The flow is two-dimensional in the x-y plane. **4** P is not a function of y.

Properties Unused engine oil at T = 40°C: ρ = 876.0 kg/m^3, μ = 0.2177 kg/m·s.

Analysis (a) The convergence is calculated by its definition (see Problem 10-30), and its tangent is also calculated,

$$\alpha = \frac{h_L - h_0}{L} = \frac{(0.0005 - 0.001)\text{ inch}}{1.0\text{ inch}} = \mathbf{-0.0005}$$

$$\rightarrow \qquad \tan\alpha = \mathbf{-0.0005}$$

Note that we must set α to radians when taking the tangent.

(b) At x = 0.5 inches (0.0127 m), we calculate $P_{gage} = P - P_{atm}$ using the result of Problem 10-30; the gage pressure at the mid-way point is

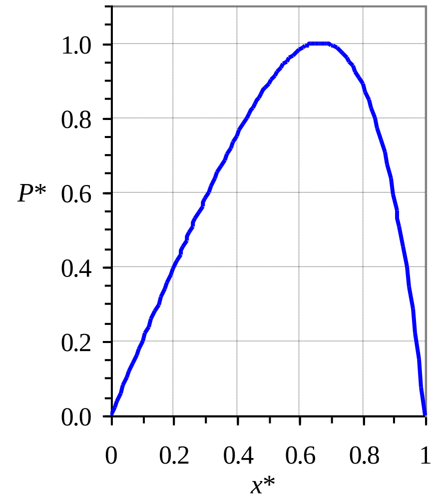


FIGURE 1
Nondimensional gage pressure in a slipper-pad bearing as a function of nondimensional axial distance along the slipper-pad.

$$P_{gage} = P - P_{atm} = 6\mu V x\left[\frac{h_0 - h_L + \alpha x}{(h_0 + h_L)(h_0 + \alpha x)^2}\right]$$

$$= 6\left(0.2177\frac{\text{kg}}{\text{m}\cdot\text{s}}\right)\left(3.048\frac{\text{m}}{\text{s}}\right)(0.0127\text{ m})\left(\frac{\text{N s}^2}{\text{kg m}}\right)\left(\frac{\text{Pa m}^2}{\text{N}}\right)\left[\frac{\left[(2.54-1.27)\times10^{-5}\text{ m}\right]-0.0005(0.0127\text{ m})}{\left[(2.54+1.27)\times10^{-5}\text{ m}\right]\left[2.54\times10^{-5}\text{ m}-0.0005(0.0127\text{ m})\right]^2}\right]$$

$$= 2.32\times10^{7}\text{ Pa} = 3370\text{ psig} = \mathbf{229\text{ atm}}$$

The gage pressure in the middle of the slipper-pad is more than 200 atmospheres. This is quite large, and illustrates how a small slipper-pad bearing can support a large amount of force.

(c) We repeat the calculations of Part (b) for values of x between 0 and L. We nondimensionalize both x and P_{gage} using x^* = x/L and $P^* = (P - P_{atm})h_0^2/\mu VL$. A plot of P^* versus x^* is shown in Fig. 1. The gage pressure is constrained to be zero at both ends of the pad, but reaches a peak near the middle, but more towards the end. For these conditions the maximum value of P^* is 1.0.

(d) To calculate the total weight that the slipper-pad bearing can support, we integrate pressure over the surface area of the plate. We used the trapezoidal rule to integrate numerically in a spreadsheet. The result is

Total vertical force (load): $$F_{load} = \int_{x=0}^{x=L} P_{gage} b\,dx = 62{,}600\text{ N} = \mathbf{14{,}100\text{ lbf}}$$

You can also obtain a reasonable estimate by simply taking the average pressure in the gap times the area – this yields F_{load} = 62,000 N = 13,900 lbf.

Discussion This slipper-pad bearing can hold an enormous amount of weight (7 tons!) due to the extremely high pressures encountered in the oil passage.

10-32

Solution We are to discuss what happens to the load when the oil temperature increases.

Analysis Oil viscosity appears only once in the equation for gap pressure. Thus, pressure and load increase linearly as oil viscosity increases. However, as the oil heats up, its viscosity goes down rapidly. For example, at $T = 40^{\circ}\mathrm{C}$, $\mu = 0.2177$ kg/m·s, but at $T = 80^{\circ}\mathrm{C}$, μ drops to 0.03232 kg/m·s. This is more than a factor of six decrease in viscosity for only a $20^{\circ}\mathrm{C}$ increase in temperature. So, **the load would decrease rapidly as oil temperature rises**.

Discussion This problem illustrates why engineers need to look at extreme operating conditions when designing products – just in case.

10-33

Solution We are to see if the Reynolds number is low enough that the flow can be approximated as creeping flow.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the *x*-*y* plane.

Properties Unused engine oil at $T = 40^{\circ}\mathrm{C}$: $\rho = 876.0\ \mathrm{kg/m^3}$, $\mu = 0.2177$ kg/m·s.

Analysis We base Re on the largest gap height, h_0,

Reynolds number:
$$\mathrm{Re} = \frac{\rho h_0 V}{\mu} = \frac{\left(876.0\ \mathrm{kg/m^3}\right)\left(2.54\times10^{-5}\ \mathrm{m}\right)\left(3.048\ \mathrm{m/s}\right)}{0.2177\ \mathrm{kg/m\cdot s}} = 0.312$$

We see that the Reynolds number is less than one, but we cannot say that Re << 1. So, the flow is not really in the creeping flow regime. However, the creeping flow approximation is generally reasonable up to Reynolds numbers near one. Also, as discussed in Problem 10-24, **the creeping flow approximation is still reasonable** in this case since L/h_0 is so large.

Discussion The error introduced by making the creeping flow approximation is probably less than the error associated with measurement of gap height.

10-34

Solution We are to calculate how much the gap compresses when the load on the bearing is doubled.

Analysis There are several ways to approach this problem: You can try to integrate the pressure distribution analytically to calculate the total load, or you can integrate numerically on a spreadsheet or math program. This is an "inverse" problem in that we can calculate the load for a given value of h_0, but we cannot do the reverse calculation directly – we must do it implicitly. One way to do this is graphically – plot load as a function of h_0, and pick off the value of h_0 where the load has doubled. Another way is by trial and error, or by a convergence technique like Newton's method. It turns out that **the load is doubled when h_0 = 0.0008535 inches (2.168×10^{-5} m). This represents a decrease in initial gap height of about 14.7%.**

Discussion The relationship between gap height and load is clearly nonlinear. When the load doubles, the gap height decreases by less than 15%.

10-35

Solution We are to estimate the speed at which a human being swimming in water would be in the creeping flow regime.

Properties For water at T = 20°C, ρ = 998.0 kg/m³ and μ = 1.002 × 10⁻³ kg/m·s.

Analysis The characteristic length scale of a human body is of order 1 m. To be in the creeping flow regime, the Reynolds number of the body should be below 1. Thus,

$$\mathrm{Re} = \frac{\rho L V}{\mu} \quad \rightarrow \quad V = \frac{\mu \mathrm{Re}}{\rho \mathrm{L}} = \frac{\left(1.002\times10^{-3}\ \mathrm{kg/m\cdot s}\right)(1)}{\left(998\ \mathrm{kg/m^3}\right)(1\ \mathrm{m})} \sim \mathbf{1\times10^{-6}\ m/s}$$

So, we would have to move at about one-millionth of a meter per second, or less. This speed is so slow that it is not measurable. Natural currents in the water, even in a "stagnant" pool of water, would be much greater than this. Hence, **we could never experience creeping flow in water**.

Discussion If we were to use a Reynolds number of 0.1 instead of 1, the result would be even slower.

10-36

Solution For each case we are to calculate the Reynolds number and determine if the creeping flow approximation is appropriate.

Assumptions **1** The values given are characteristic scales of the motion.

Properties For water at T = 20°C, ρ = 998.0 kg/m³ and μ = 1.002 × 10⁻³ kg/m·s. For unused engine oil at T = 140°C, ρ = 816.8 kg/m³ and μ = 6.558 × 10⁻³ kg/m·s. For air at T = 30°C, ρ = 1.164 kg/m³ and μ = 1.872 × 10⁻⁵ kg/m·s.

Analysis (a) The Reynolds number of the microorganism is

$$\mathrm{Re} = \frac{\rho D V}{\mu} = \frac{\left(998.0\ \mathrm{kg/m^3}\right)\left(5.0\times10^{-6}\ \mathrm{m}\right)\left(0.25\ \mathrm{mm/s}\right)}{1.002\times10^{-3}\ \mathrm{kg/m\cdot s}}\left(\frac{\mathrm{m}}{1000\ \mathrm{mm}}\right) = \mathbf{1.25\times10^{-3}}$$

Since Re << 1, **the creeping flow approximation is certainly appropriate**.

(b) The Reynolds number of the oil in the gap is

$$\mathrm{Re} = \frac{\rho D V}{\mu} = \frac{\left(816.8\ \mathrm{kg/m^3}\right)\left(0.0012\ \mathrm{mm}\right)\left(15.0\ \mathrm{m/s}\right)}{6.558\times10^{-3}\ \mathrm{kg/m\cdot s}}\left(\frac{\mathrm{m}}{1000\ \mathrm{mm}}\right) = \mathbf{2.24}$$

Since Re > 1, **the creeping flow approximation is *not* appropriate**.

(c) The Reynolds number of the fog droplet is

$$\mathrm{Re} = \frac{\rho D V}{\mu} = \frac{\left(1.164\ \mathrm{kg/m^3}\right)\left(10\times10^{-6}\ \mathrm{m}\right)\left(2.5\ \mathrm{mm/s}\right)}{1.872\times10^{-5}\ \mathrm{kg/m\cdot s}}\left(\frac{\mathrm{m}}{1000\ \mathrm{mm}}\right) = \mathbf{1.55\times10^{-3}}$$

Since Re << 1, **the creeping flow approximation is certainly appropriate**.

Discussion At room temperature, the oil viscosity increases by a factor of more than a hundred, and the Reynolds number of the bearing of Part (b) would be of order 10^{-2}, which *is* in the creeping flow range.

10-37

Solution We are to estimate the speed and Reynolds number from a multiple-image photograph.

Assumptions **1** The characteristic speed is taken as the average over 10 images.

Properties For water at $T = 20^{o}$C, ρ = 998.0 kg/m^3 and μ = 1.002 × 10^{-3} kg/m·s.

Analysis By measurement with a ruler, we estimate the sperm's diameter as 2.4 μm, and it moves about 7.7 μm in 10 frames. This represents a time of

Time for 10 frames:
$$T = \frac{10 \text{ frames}}{200 \text{ frames/s}} = 0.050 \text{ s}$$

Thus the sperm's speed is

Approximate speed:
$$V = \frac{x}{T} = \frac{7.7\ \mu\text{m}}{0.050\text{ s}}\left(\frac{\text{m}}{10^{6}\ \mu\text{m}}\right) = \mathbf{1.5\times10^{-4}\ m/s}$$

and its Reynolds number is

Reynolds number:
$$\text{Re} = \frac{\rho DV}{\mu} = \frac{\left(998.0\text{ kg/m}^3\right)\left(2.4\times10^{-6}\text{ m}\right)\left(1.5\times10^{-4}\text{ m/s}\right)}{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}} = 3.59\times10^{-4} \cong \mathbf{3.6\times10^{-4}}$$

Since Re << 1, the creeping flow approximation is certainly appropriate.

Note: Students' answers may differ widely since the measurements from the photograph are not very accurate. We report the final answer to only two significant digits because of the inherent error in measuring distances from the photograph.

Discussion If you use the cell's length rather than its diameter as the characteristic length scale, Re increases by a factor of about two, but the flow is still well within the creeping flow regime.

Inviscid Flow

10-38C

Solution We are to discuss the main difference between the steady, incompressible Bernoulli equation when applied to irrotational regions of flow vs. rotational but inviscid regions of flow.

Analysis The Bernoulli equation itself is identical in these two cases, but the "constant" for the case of rotational but inviscid regions of flow is constant only along streamlines of the flow, not everywhere. **For irrotational regions of flow, the same Bernoulli constant holds everywhere**.

Discussion A simple example is that of solid body rotation, which is rotational but inviscid. In this flow, as discussed in the text, the Bernoulli "constant" changes from one streamline to another.

10-39C

Solution We are to discuss the approximation associated with the Euler equation.

Analysis The Euler equation is simply the Navier-Stokes equation with *the viscous term neglected*; **it is therefore an inviscid approximation of the Navier-Stokes equation**. **The Euler equation is appropriate in high Reynolds number regions of the flow where net viscous forces are negligible, far away from walls and wakes.**

Discussion The Euler equation is not appropriate very close to solid walls, since frictional forces are always present there. Note that the same Euler equation is appropriate in an *irrotational* region of flow as well.

10-40

Solution We are to show that the region of flow given by this velocity field is inviscid.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis We consider the viscous terms of the x and y momentum equations:

x momentum viscous terms:

$$\mu\left(\underbrace{\cancel{\frac{\partial^2 u}{\partial x^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 u}{\partial y^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 u}{\partial z^2}}}_{0\ (2\text{-D})}\right)=0 \qquad (1)$$

y momentum viscous terms:

$$\mu\left(\underbrace{\cancel{\frac{\partial^2 v}{\partial x^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 v}{\partial y^2}}}_{0}+\underbrace{\cancel{\frac{\partial^2 v}{\partial z^2}}}_{0\ (2\text{-D})}\right)=0 \qquad (2)$$

Since the viscous terms are identically zero in both components of the Navier-Stokes equation, **this region of flow can indeed be considered inviscid**.

Discussion With the viscous terms removed, the Navier-Stokes equation is reduced to the Euler equation.

10-41

Solution We are to use an alternative method to show that the Euler equation given in the problem statement reduces to the Bernoulli equation for regions of inviscid flow.

Analysis We take the dot product of both sides of the equation with $\vec{V}$. The Euler equation dotted with velocity becomes

$$\vec{\nabla}\left(\frac{P}{\rho}+\frac{V^2}{2}+gz\right)\cdot\vec{V}=\left(\vec{V}\times\vec{\zeta}\right)\cdot\vec{V} \qquad (1)$$

The cross product on the right side of Eq. 1 is a vector that is perpendicular to $\vec{V}$. However, the dot product of two perpendicular vectors is zero by definition of the dot product. Thus, the right hand side of Eq. 1 is identically zero,

$$\vec{\nabla}\left(\frac{P}{\rho}+\frac{V^2}{2}+gz\right)\cdot\vec{V}=0 \qquad (2)$$

Now we use the same argument on the left hand side of Eq. 2, but in reverse. Namely, there are three ways for the dot product of the two vectors in Eq. 2 to be identically zero: (a) the first vector is zero,

Option (a): $$\vec{\nabla}\left(\frac{P}{\rho}+\frac{V^2}{2}+gz\right)=0 \qquad (3)$$

(b) the second vector is zero,

Option (b): $$\vec{V}=0 \qquad (4)$$

or (c) the two vectors are everywhere perpendicular to each other,

Option (c): $$\vec{\nabla}\left(\frac{P}{\rho}+\frac{V^2}{2}+gz\right)\perp\vec{V} \qquad (5)$$

Option (a) represents the *restricted* case in which the quantity in parentheses in Eq. 3 is constant everywhere. Option (b) is the *trivial* case in which there is no flow (fluid statics). Option (c) is the most *general* option, and we work with Eq. 5. Since $\vec{V}$ is everywhere parallel to streamlines of the flow, $\vec{\nabla}\left(\frac{P}{\rho}+\frac{V^2}{2}+gz\right)$ must therefore be everywhere *perpendicular* to streamlines (Fig. 1). Finally, we argue that the gradient of a scalar is a vector that points perpendicular to an imaginary surface on which the scalar is constant. Thus, we argue that the scalar $\frac{P}{\rho}+\frac{V^2}{2}+gz$ must be *constant along a streamline*. Our final result is the steady incompressible Bernoulli equation for inviscid regions of flow,

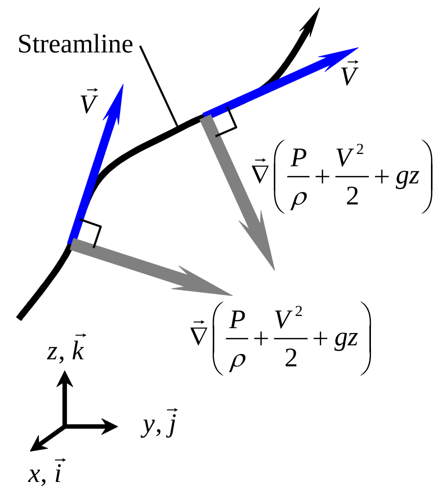


FIGURE 1

Along a streamline, $\vec{\nabla}\left(\frac{P}{\rho}+\frac{V^2}{2}+gz\right)$ is a vector everywhere perpendicular to the streamline; hence $\frac{P}{\rho}+\frac{V^2}{2}+gz$ is constant along the streamline.

$$\boxed{\frac{P}{\rho}+\frac{V^2}{2}+gz=\text{ constant along streamlines}} \qquad (6)$$

Discussion Since we have a vector identity, it must be true regardless of our choice of coordinate system.

10-42

Solution We are to expand the Euler equation into Cartesian coordinates.

Analysis We begin with the vector form of the Euler equation,

Euler equation:
$$\rho\left(\frac{\partial\vec{V}}{\partial t}+\left(\vec{V}\cdot\vec{\nabla}\right)\vec{V}\right)=-\vec{\nabla}P+\rho\vec{g} \qquad (1)$$

The *x* component of Eq. 1 is

x component:
$$\boxed{\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial P}{\partial x}+\rho g_x} \qquad (2)$$

The *y* component of Eq. 1 is

y component:
$$\boxed{\rho\left(\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial P}{\partial y}+\rho g_y} \qquad (3)$$

The *z* component of Eq. 1 is

z component:
$$\boxed{\rho\left(\frac{\partial w}{\partial t}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial P}{\partial z}+\rho g_z} \qquad (4)$$

Discussion The expansion of the Euler equation into components is identical to that of the Navier-Stokes equation, except that the viscous terms are gone.

10-43

Solution We are to expand the Euler equation into cylindrical coordinates.

Analysis We begin with the vector form of the Euler equation,

Euler equation:
$$\rho\left(\frac{\partial\vec{V}}{\partial t}+\left(\vec{V}\cdot\vec{\nabla}\right)\vec{V}\right)=-\vec{\nabla}P+\rho\vec{g} \qquad (1)$$

We must be careful to include the "extra" terms in the convective acceleration. The *r* component of Eq. 1 is

r component:
$$\boxed{\rho\left(\frac{\partial u_r}{\partial t}+u_r\frac{\partial u_r}{\partial r}+\frac{u_\theta}{r}\frac{\partial u_r}{\partial\theta}-\frac{u_\theta{}^2}{r}+u_z\frac{\partial u_r}{\partial z}\right)=-\frac{\partial P}{\partial r}+\rho g_r} \qquad (2)$$

The *y* component of Eq. 1 is

θ component:
$$\boxed{\rho\left(\frac{\partial u_\theta}{\partial t}+u_r\frac{\partial u_\theta}{\partial r}+\frac{u_\theta}{r}\frac{\partial u_\theta}{\partial\theta}+\frac{u_r u_\theta}{r}+u_z\frac{\partial u_\theta}{\partial z}\right)=-\frac{1}{r}\frac{\partial P}{\partial\theta}+\rho g_\theta} \qquad (3)$$

The *z* component of Eq. 1 is

z component:
$$\boxed{\rho\left(\frac{\partial u_z}{\partial t}+u_r\frac{\partial u_z}{\partial r}+\frac{u_\theta}{r}\frac{\partial u_z}{\partial\theta}+u_z\frac{\partial u_z}{\partial z}\right)=-\frac{\partial P}{\partial z}+\rho g_z} \qquad (4)$$

Discussion The expansion of the Euler equation into components is identical to that of the Navier-Stokes equation, except that the viscous terms are gone.

10-44

Solution We are to calculate the pressure field and the shape of the free surface for solid body rotation of water in a container.

Assumptions **1** The flow is steady and incompressible. **2** The flow is rotationally symmetric, meaning that all derivatives with respect to θ are zero. **3** Gravity acts in the negative z direction.

Properties For water at T = 20°C, ρ = 998.0 kg/m³ and μ = 1.002 × 10⁻³ kg/m·s.

Analysis We reduce the components of the Euler equation in cylindrical coordinates (Problem 10-41) as far as possible, noting that $u_r = u_z = 0$ and $u_\theta = \omega r$. The θ component disappears. The r component reduces to

r component of Euler equation:
$$-\rho\frac{u_\theta^2}{r} = -\frac{\partial P}{\partial r} \quad \rightarrow \quad \frac{\partial P}{\partial r} = \rho\omega^2 r \qquad (1)$$

and the z component reduces to

z component of Euler equation:
$$0 = -\frac{\partial P}{\partial z} - \rho g \quad \rightarrow \quad \frac{\partial P}{\partial z} = -\rho g \qquad (2)$$

We find $P(r,z)$ by cross integration. First we integrate Eq. 1 with respect to r,

$$P = \frac{\rho\omega^2 r^2}{2} + f(z) \qquad (3)$$

Note that we add a function of z instead of a constant of integration since this is a *partial* integration. We take the z derivative of Eq. 3, equate to Eq. 2, and integrate,

$$\frac{\partial P}{\partial z} = f'(z) = -\rho g \quad \rightarrow \quad f(z) = -\rho g z + C_1 \qquad (4)$$

Plugging Eq. 4 into Eq. 3 yields our expression for $P(r,z)$,

$$P = \frac{\rho\omega^2 r^2}{2} - \rho g z + C_1 \qquad (5)$$

Now we apply the boundary condition at the origin to find the value of constant C_1,

Boundary condition: At $r = 0$ and $z = 0$, $P = P_{\text{atm}} = C_1 \quad \rightarrow \quad C_1 = P_{\text{atm}}$

Finally, Eq. 5 becomes

Pressure field:
$$\boxed{P = \frac{\rho\omega^2 r^2}{2} - \rho g z + P_{\text{atm}}} \qquad (6)$$

At the free surface, we know that $P = P_{\text{atm}}$, and Eq. 6 yields the equation for the shape of the free surface,

Free surface shape:
$$\boxed{z_{\text{surface}} = \frac{\omega^2 r^2}{2g}} \qquad (7)$$

Discussion Since we know the velocity field from the start, the Euler equation is not needed for obtaining the velocity field. Instead, it is used only to calculate the pressure field. Similarly, the continuity equation is identically satisfied and is not needed here.

10-45

Solution We are to calculate the pressure field and the shape of the free surface for solid body rotation of engine oil in a container.

Assumptions **1** The flow is steady and incompressible. **2** The flow is rotationally symmetric, meaning that all derivatives with respect to θ are zero. **3** Gravity acts in the negative z direction.

Analysis In Problem 10-44, water density appears only as a constant in the pressure equation. Thus, nothing is different here except the value of density, and **the results are identical to those of Problem 10-44**.

Discussion In solid body rotation, the density of the fluid does not affect the shape of the free surface. For oil (less dense than water), pressure increases with depth at a slower rate compared to water.

10-46

Solution We are to calculate the Bernoulli constant for solid body rotation of water in a container.

Assumptions **1** The flow is steady and incompressible. **2** The flow is rotationally symmetric, meaning that all derivatives with respect to θ are zero. **3** Gravity acts in the negative z direction.

Analysis From Problem 10-42, we have the pressure field,

Pressure field:
$$P = \frac{\rho\omega^2 r^2}{2} - \rho g z + P_{\text{atm}} \tag{1}$$

The Bernoulli equation for steady, incompressible, inviscid regions of flow is

$$\frac{P}{\rho} + \frac{V^2}{2} + gz = C_r = \text{constant along streamlines} \tag{2}$$

The velocity field is $u_r = u_z = 0$ and $u_\theta = \omega r$, $V^2 = \omega^2 r^2$, and Eq. 2 becomes

$$C_r = \frac{P}{\rho} + \frac{\omega^2 r^2}{2} + gz \tag{3}$$

Substitution of Eq. 1 into Eq. 3 yields the final expression for C_r,

Bernoulli "constant":
$$C_r = \frac{\omega^2 r^2}{2} - gz + \frac{P_{\text{atm}}}{\rho} + \frac{\omega^2 r^2}{2} + gz \rightarrow \boxed{C_r = \frac{P_{\text{atm}}}{\rho} + \omega^2 r^2} \tag{4}$$

Discussion Streamlines in this flow field are circles about the z axis (lines of constant r). The Bernoulli "constant" C_r is constant along any given streamline, but changes from streamline to streamline. This is typical of rotating flow fields.

10-47

Solution For a given volume flow rate, we are to generate an expression for u_r assuming inviscid flow, and then discuss the velocity profile shape for a real (viscous) flow.

Assumptions **1** The flow remains radial at all times (no u_θ component). **2** The flow is steady, two-dimensional, and incompressible.

Analysis If the flow were inviscid, we could not enforce the no-slip condition at the walls of the duct. At any r location, the volume flow rate must be the same,

Volume flow rate at any r location: $$\dot{V} = u_r rb\Delta\theta \qquad (1)$$

where $\Delta\theta$ is the angle over which the contraction is bound (see Fig. 1). Thus,

$$u_r = \frac{\dot{V}}{rb\Delta\theta} \qquad (2)$$

At radius $r = R$, Eq. 2 becomes

Radial velocity at r = R: $$u_r(R) = \frac{\dot{V}}{Rb\Delta\theta} \qquad (3)$$

Upon substitution of Eq. 3 into Eq. 2, we get

Radial velocity at any r location: $$\boxed{u_r = \frac{R}{r}u_r(R)} \qquad (4)$$

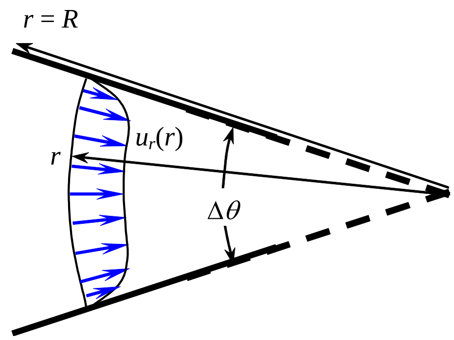


FIGURE 1
Possible shape of the velocity profile for a real (viscous) flow.

In other words, the radial velocity component increases as the reciprocal of r as r approaches zero (the origin).

In a real flow (with viscous effects), we would expect that the velocity near the center of the duct is somewhat larger, while that near the walls is somewhat smaller. Right at the walls, of course, the velocity is zero by the no-slip condition. In Fig. 1 is a sketch of what the velocity profile might look like in a real flow.

Discussion In either case, the radial velocity is infinite at the origin. This is actually a portion of a line sink, as discussed in this chapter.

10-48

Solution We are to show that the given vector identity is satisfied in Cartesian coordinates.

Analysis We expand each term in the vector identity carefully. The first term is

$$\left(\vec{V}\cdot\vec{\nabla}\right)\vec{V}=\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)\vec{i}+\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)\vec{j}+\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)\vec{k} \tag{1}$$

The second term is

$$\vec{\nabla}\left(\frac{V^2}{2}\right)=\frac{1}{2}\left[\left(\frac{\partial u^2}{\partial x}+\frac{\partial v^2}{\partial x}+\frac{\partial w^2}{\partial x}\right)\vec{i}+\left(\frac{\partial u^2}{\partial y}+\frac{\partial v^2}{\partial y}+\frac{\partial w^2}{\partial y}\right)\vec{j}+\left(\frac{\partial u^2}{\partial z}+\frac{\partial v^2}{\partial z}+\frac{\partial w^2}{\partial z}\right)\vec{k}\right]$$

which reduces to

$$\vec{\nabla}\left(\frac{V^2}{2}\right)=\left(u\frac{\partial u}{\partial x}+v\frac{\partial v}{\partial x}+w\frac{\partial w}{\partial x}\right)\vec{i}+\left(u\frac{\partial u}{\partial y}+v\frac{\partial v}{\partial y}+w\frac{\partial w}{\partial y}\right)\vec{j}+\left(u\frac{\partial u}{\partial z}+v\frac{\partial v}{\partial z}+w\frac{\partial w}{\partial z}\right)\vec{k} \tag{2}$$

The third term is

$$\vec{V}\times\left(\vec{\nabla}\times\vec{V}\right)=\left[v\left(\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}\right)-w\left(\frac{\partial u}{\partial z}-\frac{\partial w}{\partial x}\right)\right]\vec{i}+\left[w\left(\frac{\partial w}{\partial y}-\frac{\partial v}{\partial z}\right)-u\left(\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}\right)\right]\vec{j}+\left[u\left(\frac{\partial u}{\partial z}-\frac{\partial w}{\partial x}\right)-v\left(\frac{\partial w}{\partial y}-\frac{\partial v}{\partial z}\right)\right]\vec{k} \tag{3}$$

When we substitute Eqs. 1 through 3 into the given equation, we see that all the terms disappear, and **the equation is satisfied**. We show this for the x direction only (all terms with unit vector $\vec{i}$):

$$\cancel{u\frac{\partial u}{\partial x}}+\cancel{v\frac{\partial u}{\partial y}}+\cancel{w\frac{\partial u}{\partial z}}=\cancel{u\frac{\partial u}{\partial x}}+\cancel{v\frac{\partial v}{\partial x}}+\cancel{w\frac{\partial w}{\partial x}}-\cancel{v\frac{\partial v}{\partial x}}+\cancel{v\frac{\partial u}{\partial y}}+\cancel{w\frac{\partial u}{\partial z}}-\cancel{w\frac{\partial w}{\partial x}} \tag{4}$$

The algebra is similar for the $\vec{j}$ and $\vec{k}$ terms, and the vector identity is shown to be true for Cartesian coordinates.

Discussion Since we have a vector identity, it must be true regardless of our choice of coordinate system.

Irrotational (Potential) Flow

10-49C

Solution We are to discuss D'Alembert's paradox.

Analysis D'Alembert's paradox states that **with the irrotational flow approximation, the aerodynamic drag force on any non-lifting body of any shape immersed in a uniform stream is zero**. It is a paradox because we know from experience that bodies in a flow field have non-zero aerodynamic drag.

Discussion Irrotational flow over a non-lifting immersed body has neither pressure drag nor viscous drag. In a real flow, both of these drag components are present.

10-50C

Solution We are to identify regions in the flow field that are irrotational, and regions that are rotational.

Assumptions **1** The air in the room would be calm if not for the presence of the hair dryer.

Analysis **Flow in the air far away from the hair dryer and its jet is certainly irrotational. As the air approaches the inlet, it is irrotational except very close to the surface of the hair dryer. Flow in the jet is rotational, but flow outside of the jet can be approximated as irrotational**.

Discussion Flow near solid walls is nearly always rotational because of the viscous rotational boundary layer that grows there. There are sharp velocity gradients in a jet, so the vorticity cannot be zero in that region, and the flow must be rotational in the jet as well.

10-51C

Solution We are to discuss the role of the momentum equation in an irrotational region of flow.

Assumptions **1** The flow is steady and incompressible. **2** The region of interest in the flow field is irrotational.

Analysis Although it is true that the momentum equation is not required in order to solve for the velocity field, **it *is* required in order to solve for the *pressure* field**. In particular, the Navier-Stokes equation reduces to the Bernoulli equation in an irrotational region of flow.

Discussion Mathematically, it turns out that in an irrotational flow field the continuity equation is *uncoupled* from the momentum equation, meaning that we can solve continuity for ϕ by itself, without need of the momentum equation. However, the momentum equation cannot be solved by itself.

10-52C

Solution We are to discuss similarities and differences between two approximations: inviscid regions of flow and irrotational regions of flow.

Assumptions **1** The flow is incompressible and steady.

Analysis The two approximations are similar in that in both cases, the viscous terms in the Navier-Stokes equation drop out, leaving the Euler equation. Also, in both cases the Bernoulli equation results form integration of the Euler equation. However, these two approximations differ significantly from each other. **When making the inviscid flow approximation, we assume that the viscous terms are negligibly small**. A good example, as discussed in this chapter, is solid body rotation. In this case, although the fluid itself is viscous, all effects of viscosity are gone, and the flow field can be considered "inviscid" (although it is rotational). On the other hand, **the irrotational approximation is made when the vorticity (a measure of rotationality of fluid particles) is negligibly small**. In this case, viscosity still acts on fluid particles – it shears them and distorts them, yet the net rate of rotation of fluid particles is zero. In other words, in an irrotational region of flow, the net viscous force on a fluid particle is zero, but viscous stresses on the fluid particle are certainly *not* zero. Examples of irrotational, but viscous flows include any irrotational flow field with curved streamlines, such as a line vortex, a doublet, irrotational flow over a circular cylinder, etc. Freestream flow is both inviscid and irrotational since fluid particles do not shear or distort or rotate, and viscosity does not enter into the picture.

Discussion In either case, the viscous terms in the Navier-Stokes equation disappear, but for different reasons. In the inviscid flow approximation, the viscous terms disappear because we neglect viscosity. In the irrotational flow approximation, the viscous terms disappear because they cancel each other out due to the fact that the vorticity (hence the rate of rotation) of fluid particles is negligibly small.

10-53C

Solution We are to discuss the flow property that determines whether a region of flow is rotational or irrotational.

Analysis The **vorticity** determines whether a region of flow is rotational or irrotational. Specifically, if the vorticity is zero (or negligibly small), the flow is approximated as irrotational, but if the vorticity is not negligibly small, the flow is rotational.

Discussion Another acceptable answer is the rate of rotation vector or the angular velocity vector of a fluid particle.

10-54

Solution We are to compare the Bernoulli equation and its restrictions for inviscid, rotational regions of flow and viscous, irrotational regions of flow.

Assumptions **1** The flow is incompressible and steady.

Analysis The Bernoulli equation is the same in both cases, namely

Steady incompressible Bernoulli equation: $$\boxed{\frac{P}{\rho}+\frac{V^2}{2}+gz=C} \quad (1)$$

However, in an inviscid, rotational region of flow, Eq. 1 is applicable only along a streamline. The Bernoulli "constant" C is constant along any particular streamline, but may change from streamline to streamline. In a viscous, irrotational region of flow, however, the Bernoulli constant is constant everywhere, even across streamlines. Thus, **the inviscid, rotational region of flow has more restrictions on the use of the Bernoulli equation**.

Discussion In either case, the viscous terms in the Navier-Stokes equation disappear, but for different reasons.

10-55

Solution For a given set of streamlines, we are to sketch the corresponding set of equipotential curves and explain how we obtain them.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional in the plane of the figure associated with this problem. **3** The flow in the region shown in the figure is irrotational.

Analysis **Some possible equipotential lines are sketched in Fig. 1**. We draw these based on the fact that the streamlines and equipotential curves must intersect at 90^o angles. To find the "correct" shape, it helps to sketch a few extra streamlines in between the given ones to guide in construction of the equipotential curves. These "interpolated" streamlines are shown in Fig. 1 as thin, dotted blue lines.

Discussion The exact shape of the equipotential curves is not known, and individuals may sketch curves of other shapes that are equally valid. The important thing to emphasize is that the curves of constant ϕ are everywhere *perpendicular* to the streamlines.

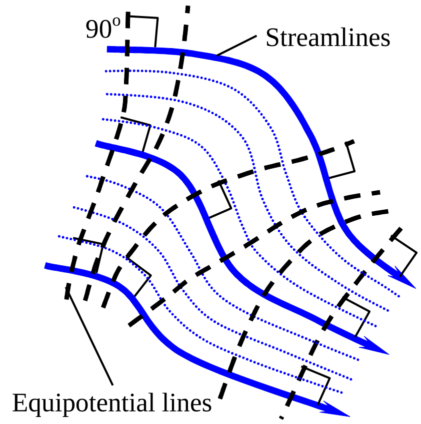


FIGURE 1
Possible equipotential curves (dashed black lines) and intermediate streamlines (dotted blue lines).

10-56

Solution For a given velocity field, we are to assess whether the flow field is irrotational. If so, we are to generate an expression for the velocity potential function.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x-y* plane.

Analysis For the flow to be irrotational, the vorticity must be zero. Since the flow is planar in the *x-y* plane, the only non-zero component of vorticity is in the *z* direction,

z-component of vorticity:
$$\zeta_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} = 0 - 0 = 0 \qquad (1)$$

Since the vorticity is zero, **this flow field can be considered irrotational**, and we should be able to generate a velocity potential function that describes the flow. In two dimensions we have

Velocity components in terms of potential function:
$$u = \frac{\partial \phi}{\partial x} \qquad v = \frac{\partial \phi}{\partial y} \qquad (2)$$

We pick one of these (the first one) and integrate to obtain an expression for ϕ,

Velocity potential function:
$$\frac{\partial \phi}{\partial x} = u = ax + b \qquad \phi = a\frac{x^2}{2} + bx + f(y) \qquad (3)$$

Note that we have added a function of *y* rather than a constant of integration since we have performed a partial integration with respect to *x*. Using Eq. 2, we differentiate Eq. 3 with respect to *y* and equate the result to the *v* component of velocity,

$$\frac{\partial \phi}{\partial y} = f'(y) = v = -ay + c \qquad (4)$$

Equation 4 is integrated with respect to *y* to find function *f*(*y*),

$$f(y) = -a\frac{y^2}{2} + cy + \text{constant} \qquad (5)$$

This time, a constant of integration is added since this is a total integration. Finally, we plug Eq. 5 into Eq. 3 to obtain our final expression for the velocity potential function,

Result, velocity potential function:
$$\boxed{\phi = a\frac{\left(x^2 - y^2\right)}{2} + bx + cy + \text{constant}} \qquad (6)$$

Discussion You should plug Eq. 6 into Eq. 2 to verify that it is correct.

10-57

Solution For a given velocity field, we are to assess whether the flow field is irrotational. If so, we are to generate an expression for the velocity potential function.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis For the flow to be irrotational, the vorticity must be zero. Since the flow is planar in the *x*-*y* plane, the only non-zero component of vorticity is in the *z* direction,

z-component of vorticity:
$$\zeta_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} = ay - ay = 0 \tag{1}$$

Since the vorticity is zero, **this flow field can be considered irrotational**, and we should be able to generate a velocity potential function that describes the flow. In two dimensions we have

Velocity components in terms of potential function:
$$u = \frac{\partial \phi}{\partial x} \qquad v = \frac{\partial \phi}{\partial y} \tag{2}$$

We pick one of these (the first one) and integrate to obtain an expression for ϕ,

Velocity potential function:
$$\frac{\partial \phi}{\partial x} = u = \tfrac{1}{2}ay^2 + b \qquad \phi = \tfrac{1}{2}ay^2x + bx + f(y) \tag{3}$$

Note that we have added a function of *y* rather than a constant of integration since we have performed a partial integration with respect to *x*. Using Eq. 2, we differentiate Eq. 3 with respect to *y* and equate the result to the *v* component of velocity,

$$\frac{\partial \phi}{\partial y} = axy + f'(y) = v = axy + c \quad \rightarrow \quad f'(y) = c \tag{4}$$

Equation 4 is integrated with respect to *y* to find function *f*(*y*),

$$f(y) = cy + C_1 \tag{5}$$

This time, a constant of integration (C_1) is added since this is a total integration. Finally, we plug Eq. 5 into Eq. 3 to obtain our final expression for the velocity potential function,

Result, velocity potential function:
$$\boxed{\phi = \tfrac{1}{2}ay^2x + bx + cy + C_1} \tag{6}$$

Discussion You should plug Eq. 6 into Eq. 2 to verify that it is correct.

10-58

Solution We are to generate expressions for the stream function and the velocity potential function for a line source, beginning with the first equation above.

Assumptions **1** The flow is steady and incompressible. **2** The flow is irrotational in the region of interest. **3** The flow is two-dimensional in the *x*-*y* or *r*-θ plane.

Analysis To find the stream function, we integrate the first equation with respect to θ, and then differentiate with respect to the other variable *r*,

$$\frac{\partial\psi}{\partial\theta}=\frac{\dot{V}/L}{2\pi} \quad \rightarrow \quad \psi=\frac{\dot{V}/L}{2\pi}\theta+f(r) \quad \rightarrow \quad \frac{\partial\psi}{\partial r}=f'(r)=-u_\theta=0 \qquad (1)$$

We integrate Eq. 1 to obtain

$$f(r)=\text{constant} \qquad (2)$$

We set the arbitrary constant of integration to zero since we can add back a constant as desired at any time without changing the flow. Thus,

Line source at the origin:
$$\boxed{\psi=\frac{\dot{V}/L}{2\pi}\theta} \qquad (3)$$

We perform a similar analysis for ϕ by beginning with the first equation:

$$\frac{\partial\phi}{\partial r}=\frac{\dot{V}/L}{2\pi r} \quad \rightarrow \quad \phi=\frac{\dot{V}/L}{2\pi}\ln r+f(\theta) \quad \rightarrow \quad \frac{\partial\phi}{\partial\theta}=f'(\theta)=ru_\theta=0 \qquad (4)$$

We integrate Eq. 4 to obtain

$$f(\theta)=\text{constant}$$

We set the arbitrary constant of integration to zero since we can add back a constant as desired at any time without changing the flow. Thus,

Line source at the origin:
$$\boxed{\phi=\frac{\dot{V}/L}{2\pi}\ln r} \qquad (5)$$

Discussion You can easily verify by differentiation that Eqs. 3 and 5 yield the correct velocity components. Also note that if $\dot{V}/L$ is negative, the flow field is that of a line sink rather than a line source.

10-59

Solution We are to calculate the velocity components from a given potential function, verify that the velocity field is irrotational, and generate an expression for ψ.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional in the *x*-*y* plane. **3** The flow is irrotational in the region in which Eq. 1 applies.

Analysis (*a*) The velocity components are found by taking the *x* and *y* partial derivatives of ϕ,

Velocity components:
$$\boxed{u = \frac{\partial \phi}{\partial x} = 6x + 4y - 2 \qquad v = \frac{\partial \phi}{\partial y} = -6y + 4x - 5} \tag{1}$$

(*b*) We plug in *u* and *v* from Eq. 1 into the *z* component of vorticity to get

z-component of vorticity:
$$\zeta_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} = 4 - 4 = 0 \tag{2}$$

Since $\zeta_z = 0$, and the only component of vorticity in a 2-D flow in the *x*-*y* plane is in the *z* direction, the vorticity is zero, and **the flow is irrotational in the region of interest**.

(*c*) The stream function is found by integration of the velocity components. We begin by integrating the *x* component, $\partial\psi/\partial y = u$, and then taking the *x* derivative to compare with the known value of *v*,

$$\psi = 6xy + 2y^2 - 2y + f(x) \quad \rightarrow \quad v = -\frac{\partial \psi}{\partial x} = -6y - f'(x) = -6y + 4x - 5 \tag{3}$$

From which we see that $f'(x) = 5 - 4x$. Integrating with respect to *x*,

$$f(x) = 5x - 2x^2 + \text{constant} \tag{4}$$

The constant is arbitrary since velocity components are always derivatives of ψ. Thus,

Stream function:
$$\boxed{\psi = 6xy + 2y^2 - 2y + 5x - 2x^2 + \text{constant}} \tag{5}$$

Discussion You can verify that the partial derivatives of Eq. 5 yield the same velocity components as those of Eq. 1.

10-60

Solution We are to calculate the velocity components from a given potential function, verify that the velocity field is irrotational, and generate an expression for ψ.

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional in the *x*-*y* plane. **3** The flow is irrotational in the region in which Eq. 1 applies.

Analysis (*a*) The velocity components are found by taking the *x* and *y* partial derivatives of ϕ,

Velocity components:
$$\boxed{u = \frac{\partial \phi}{\partial x} = 8x + 6 \qquad v = \frac{\partial \phi}{\partial y} = -8y - 4} \tag{1}$$

(*b*) We plug in *u* and *v* from Eq. 1 into the *z* component of vorticity to get

z-component of vorticity:
$$\zeta_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} = 0 - 0 = 0 \tag{2}$$

Since $\zeta_z = 0$, and the only component of vorticity in a 2-D flow in the *x*-*y* plane is in the *z* direction, the vorticity is zero, and **the flow is irrotational in the region of interest**.

(*c*) The stream function is found by integration of the velocity components. We begin by integrating the *x* component, $\partial\psi/\partial y = u$, and then taking the *x* derivative to compare with the known value of *v*,

$$\psi = 8xy + 6y + f(x) \quad \rightarrow \quad v = -\frac{\partial \psi}{\partial x} = -8y - f'(x) = -8y - 4 \tag{3}$$

From which we see that $f'(x) = 4$. Integrating with respect to *x*,

$$f(x) = 4x + \text{constant} \tag{4}$$

The constant is arbitrary since velocity components are always derivatives of ψ. Thus,

Stream function:
$$\boxed{\psi = 8xy + 6y + 4x + \text{constant}} \tag{5}$$

Discussion You can verify that the partial derivatives of Eq. 5 yield the same velocity components as those of Eq. 1.

10-61

Solution We are to show that the stream function for a planar irrotational region of flow satisfies the Laplace equation in cylindrical coordinates.

Assumptions **1** This region of flow is planar in the r-θ plane. **2** The flow is incompressible. **3** This region of flow is irrotational.

Analysis We defined stream function ψ as

Planar flow stream function in cylindrical coordinates: $$u_r = \frac{1}{r}\frac{\partial \psi}{\partial \theta} \qquad u_\theta = -\frac{\partial \psi}{\partial r} \tag{1}$$

We also know that for irrotational flow the vorticity must be zero. Since the only non-zero component of vorticity is in the z direction,

z-component of vorticity: $$\zeta_z = \frac{1}{r}\left(\frac{\partial(ru_\theta)}{\partial r} - \frac{\partial u_r}{\partial \theta}\right) = \frac{1}{r}\left(\frac{\partial}{\partial r}\left(-r\frac{\partial \psi}{\partial r}\right) - \frac{\partial}{\partial \theta}\left(\frac{1}{r}\frac{\partial \psi}{\partial \theta}\right)\right) = 0 \tag{2}$$

Since r is not a function of θ, it can come outside the derivative operator in the last term. Also, the negative sign in both terms can be disposed of. Thus,

Result of irrotationality condition: $$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial \psi}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 \psi}{\partial \theta^2} = 0 \tag{3}$$

Since Eq. 3 is the Laplace equation for 2-D planar flow in the r-θ plane, we have shown that **the stream function indeed satisfies the Laplace equation**.

Discussion Since the Laplace equation for stream function is satisfied in Cartesian coordinates for the case of 2-D planar flow in the x-y plane, it must also be satisfied for the same flow in cylindrical coordinates. All we have done is use a different coordinate system to describe the *same* flow.

10-62

Solution We are to write the Laplace equation in two dimensions (r and θ) in spherical polar coordinates.

Assumptions **1** The flow is independent of angle ϕ (about the x axis). **2** The flow is irrotational.

Analysis We look up the Laplace equation in spherical polar coordinates in any vector analysis book. Ignoring derivatives with respect to ϕ, we get

Laplace equation, axisymmetric flow, (r,θ): $$\boxed{\frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2\frac{\partial \phi}{\partial r}\right) + \frac{1}{r^2\sin\theta}\frac{\partial}{\partial \theta}\left(\sin\theta\frac{\partial \phi}{\partial \theta}\right) = 0}$$

Discussion Even though ϕ satisfies the Laplace equation in an irrotational region of flow, ψ does *not* for the present case of axisymmetric flow.

10-63

Solution We are to prove that the given stream function exactly satisfies the continuity equation for the case of axisymmetric flow in spherical polar coordinates.

Assumptions **1** The flow is axisymmetric, implying that there is no variation rotationally around the axis of symmetry. **2** The flow is incompressible.

Analysis We plug the stream function into the continuity equation, and perform the algebra,

$$\frac{1}{r}\frac{\partial}{\partial r}\left(-\frac{1}{\sin\theta}\frac{\partial\psi}{\partial\theta}\right)+\frac{1}{\sin\theta}\frac{\partial}{\partial\theta}\left(\frac{1}{r}\frac{\partial\psi}{\partial r}\right)=0 \qquad (1)$$

since θ is not a function of r and vice-versa, Eq. 1 can be rearranged as

$$-\frac{1}{r\sin\theta}\frac{\partial^2\psi}{\partial r\partial\theta}+\frac{1}{r\sin\theta}\frac{\partial^2\psi}{\partial\theta\partial r}=0 \qquad -\frac{\partial^2\psi}{\partial r\partial\theta}+\frac{\partial^2\psi}{\partial\theta\partial r}=0 \qquad (2)$$

Equation 2 is identically satisfied as long as ψ is a smooth function of r and θ.

Discussion If ψ were *not* smooth, the order of differentiation (r then θ versus θ then r) would be important and Eq. 2 would not necessarily be zero. In the definition of stream function, it is somewhat arbitrary whether we put the negative sign on u_r or u_θ, and you may find the opposite sign convention in other textbooks.

10-64

Solution We are to generate expressions for velocity potential function and stream function for the case of a uniform stream of magnitude V inclined at angle α.

Assumptions **1** The flow is planar, incompressible, and irrotational. **2** The flow is uniform everywhere in the flow field, with magnitude V and inclination angle α.

Analysis For planar flow in Cartesian coordinates, we write

$$u = \frac{\partial\phi}{\partial x} = \frac{\partial\psi}{\partial y} = V\cos\alpha \qquad v = \frac{\partial\phi}{\partial y} = -\frac{\partial\psi}{\partial x} = V\sin\alpha \qquad (1)$$

By integrating the first of these with respect to x, and then differentiating the result with respect to y, we generate an expression for the velocity potential function for a uniform stream,

$$\phi = Vx\cos\alpha + f(y) \qquad v = \frac{\partial\phi}{\partial y} = f'(y) = V\sin\alpha \qquad (2)$$

Integrating with respect to y,

$$f(y) = Vy\sin\alpha + \text{constant} \qquad (3)$$

The constant is arbitrary since velocity components are always derivatives of ϕ. We set the constant to zero, knowing that we can always add an arbitrary constant later on if desired. Thus,

Velocity potential function:
$$\boxed{\phi = Vx\cos\alpha + Vy\sin\alpha} \qquad (4)$$

We do a similar analysis for the stream function, beginning again with Eq. 1.

$$\psi = Vy\cos\alpha + g(x) \qquad v = -\frac{\partial\psi}{\partial x} = -g'(x) = V\sin\alpha \qquad (5)$$

Integrating with respect to x,

$$g(x) = -Vx\sin\alpha + \text{constant} \qquad (6)$$

The constant is arbitrary since velocity components are always derivatives of ψ. We set the constant to zero, knowing that we can always add an arbitrary constant later on if desired. Thus,

Stream function:
$$\boxed{\psi = Vy\cos\alpha - Vx\sin\alpha} \qquad (7)$$

Discussion You should be able to obtain the same answers by starting with the *opposite* equations in Eq. 1 (i.e., integrate first with respect to y to obtain ϕ and with respect to x to obtain ψ).

10-65

Solution For a given velocity field, we are to assess whether the flow field is irrotational. If so, we are to generate an expression for the velocity potential function.

Assumptions **1** The flow is steady. **2** The flow is incompressible. **3** The flow is two-dimensional in the *x*-*y* plane.

Analysis For the flow to be irrotational, the vorticity must be zero. Since the flow is planar in the *x*-*y* plane, the only non-zero component of vorticity is in the *z* direction,

z-component of vorticity:
$$\zeta_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} = ay^2 - ay \neq 0 \qquad (1)$$

Since the vorticity is not zero, **this flow field cannot be considered irrotational**, and we cannot generate a velocity potential function that describes the flow. We cannot continue – the solution stops here.

Discussion Fluid particles in this region of flow apparently have significant rotation.

10-66

Solution We are to show that the vorticity components are zero in an irrotational region of flow.

Analysis The first component of vorticity becomes

r-component of vorticity vector:
$$\zeta_r = \frac{1}{r}\frac{\partial u_z}{\partial \theta} - \frac{\partial u_\theta}{\partial z} = \frac{1}{r}\frac{\partial^2 \phi}{\partial \theta \partial z} - \frac{1}{r}\frac{\partial^2 \phi}{\partial z \partial \theta} = 0$$

which is valid as long as ϕ is a smooth function of θ and *z*. Similarly, the second component of vorticity becomes

θ-component of vorticity vector:
$$\zeta_\theta = \frac{\partial u_r}{\partial z} - \frac{\partial u_z}{\partial r} = \frac{\partial^2 \phi}{\partial z \partial r} - \frac{\partial^2 \phi}{\partial r \partial z} = 0$$

which is valid as long as ϕ is a smooth function of *r* and *z*. Finally, the third component of vorticity becomes

z-component of vorticity vector:
$$\zeta_z = \frac{1}{r}\frac{\partial}{\partial r}\left(r u_\theta\right) - \frac{1}{r}\frac{\partial u_r}{\partial \theta} = \frac{1}{r}\frac{\partial^2 \phi}{\partial r \partial \theta} - \frac{1}{r}\frac{\partial^2 \phi}{\partial \theta \partial r} = 0$$

which is valid as long as ϕ is a smooth function of *r* and *z*. Thus **all three components of vorticity are zero**.

Discussion By mathematical identity, the velocity potential function is definable only when the vorticity vector is zero; therefore the results are not surprising. Note that in a three-dimensional flow, ϕ must be a smooth function of *r*, θ, and *z*.

10-67

Solution We are to verify that the Laplace equation holds in an irrotational flow field in cylindrical coordinates.

Analysis We plug in the components of velocity from Problem 10-48 into the left hand side of the Laplace equation in cylindrical coordinates,

Laplace equation in cylindrical coordinates:
$$\nabla^2\phi = \frac{1}{r}\frac{\partial}{\partial r}(ru_r) + \frac{1}{r^2}\frac{\partial}{\partial \theta}(ru_\theta) + \frac{\partial u_z}{\partial z} \qquad (1)$$

But since r is not a function of θ, we simplify Eq. 1 to

$$\nabla^2\phi = \frac{1}{r}\frac{\partial}{\partial r}(ru_r) + \frac{1}{r}\frac{\partial u_\theta}{\partial \theta} + \frac{\partial u_z}{\partial z} \qquad (2)$$

We recognize the terms on the right side of Eq. 2 as those of the incompressible form of the continuity equation in cylindrical coordinates; Eq. 2 is thus equal to zero, and the Laplace equation holds,

$$\boxed{\nabla^2\phi = 0} \qquad (3)$$

Discussion The Laplace equation is valid for any incompressible irrotational region of flow, regardless of whether the flow is two- or three-dimensional.

10-68

Solution We are to generate expressions for the stream function and the velocity potential function for a line vortex.

Assumptions **1** The flow is steady and incompressible. **2** The flow is irrotational in the region of interest. **3** The flow is two-dimensional in the *x*-*y* or r-θ plane.

Analysis To find the stream function we integrate the first equation with respect to θ, and then differentiate with respect to the other variable *r*,

$$\frac{\partial \psi}{\partial \theta} = 0 \quad \rightarrow \quad \psi = f(r) \quad \rightarrow \quad \frac{\partial \psi}{\partial r} = f'(r) = -u_\theta = -\frac{\Gamma}{2\pi r} \qquad (1)$$

We integrate Eq. 1 to obtain

$$f(r) = -\frac{\Gamma}{2\pi}\ln r + \text{constant} \qquad (2)$$

We set the arbitrary constant of integration to zero since we can add back a constant as desired at any time without changing the flow. Thus,

Line vortex at the origin:
$$\boxed{\psi = -\frac{\Gamma}{2\pi}\ln r} \qquad (3)$$

We perform a similar analysis for ϕ. Beginning with the first equation:

$$\frac{\partial \phi}{\partial r} = 0 \quad \rightarrow \quad \phi = f(\theta) \quad \rightarrow \quad \frac{\partial \phi}{\partial \theta} = f'(\theta) = r u_\theta = \frac{\Gamma}{2\pi} \qquad (4)$$

We integrate Eq. 4 to obtain

$$f(\theta) = \frac{\Gamma}{2\pi}\theta + \text{constant}$$

We set the arbitrary constant of integration to zero since we can add back a constant as desired at any time without changing the flow. Thus,

Line vortex at the origin:
$$\boxed{\phi = \frac{\Gamma}{2\pi}\theta} \qquad (5)$$

Discussion You can easily verify by differentiation that Eqs. 3 and 5 yield the correct velocity components. Also note that if Γ is positive, the vortex is counterclockwise, and if Γ is negative, the vortex is clockwise.

10-69

Solution We are to calculate Reynolds number and estimate maximum and minimum pressure and speed for potential flow over a circular cylinder.

Assumptions **1** The flow is two-dimensional, and thus the end effects (front and back of the cylinder) are negligible. **2** The flow is far enough in extent so that there are negligible boundary effects.

Analysis (*a*) The Reynolds number is calculated as follows:

$$\mathrm{Re} = \frac{\rho V_\infty d}{\mu} = \frac{\left(998.2\ \mathrm{kg/m^3}\right)\left(0.100481\ \mathrm{m/s}\right)\left(1.00\ \mathrm{m}\right)}{1.003\times 10^{-3}\ \mathrm{kg/m\cdot s}} = 100{,}000$$

Or, **Re = 1.00 × 10⁵** to three significant digits. This **Re is sufficiently high that potential flow should be a reasonable approximation, but only along the front portion of the cylinder**, as discussed in the text.

(*b*) The **minimum speed occurs at the stagnation point, where $|V|_{min} = 0$**. This is also the location of the maximum pressure. We use the Bernoulli equation to estimate the pressure at the stagnation point,

$$P + \frac{1}{2}\rho V^2 = P_\infty + \frac{1}{2}\rho V_\infty^{\ 2} \quad \rightarrow \quad P - P_\infty = \frac{1}{2}\rho V_\infty^{\ 2}$$

where we have set $V = 0$ at the stagnation point. This yields

$$\begin{aligned} P - P_\infty &= \frac{1}{2}\rho V_\infty^{\ 2} = \frac{1}{2}\left(998.2\ \mathrm{kg/m^3}\right)\left(0.100481\ \mathrm{m/s}\right)^2\left(\frac{\mathrm{N}}{\mathrm{kg\cdot m/s^2}}\right) \\ &= 5.0391\ \mathrm{N/m^2} \end{aligned}$$

Thus, the **maximum pressure occurs at the stagnation point, where $(P - P_\infty)_{max} = 5.04\ \mathrm{N/m^2}$** (to three significant digits).

The **maximum speed occurs at the shoulder of the cylinder, where $|V|_{max} = 2V_\infty = 0.201$ m/s**. This is also the location of the minimum pressure. At the shoulder,

$$P + \frac{1}{2}\rho V^2 = P_\infty + \frac{1}{2}\rho V_\infty^{\ 2} \quad \rightarrow \quad P - P_\infty = \frac{1}{2}\rho\left(V_\infty^{\ 2} - V^2\right)$$

Setting $V = 2V_\infty$ yields

$$\begin{aligned} P - P_\infty &= \frac{-3}{2}\rho V_\infty^{\ 2} = \frac{-3}{2}\left(998.2\ \mathrm{kg/m^3}\right)\left(0.100481\ \mathrm{m/s}\right)^2\left(\frac{\mathrm{N}}{\mathrm{kg\cdot m/s^2}}\right) \\ &= -15.1174\ \mathrm{N/m^2} \end{aligned}$$

Thus, the **minimum pressure occurs at the shoulder, where $(P - P_\infty)_{min} = -15.1\ \mathrm{N/m^2}$** (to three significant digits).

Discussion Keep in mind that the potential flow analysis is an approximation – it assumes irrotational flow everywhere, whereas we know that the flow is rotational in the boundary layer very close to the cylinder wall.

10-70

Solution For a given stream function, we are to calculate the velocity potential function

Assumptions **1** The flow is steady and incompressible. **2** The flow is two-dimensional in the r-θ plane. **3** The flow is approximated as irrotational.

Analysis There are two ways to approach this problem: (1) Calculate the velocity components from the stream function, and then integrate to obtain ϕ. (2) Superpose a freestream and a doublet to generate ϕ directly. We show both methods here.

Method (1): We calculate the velocity components everywhere in the flow field by differentiating the stream function,

Velocity components:
$$u_r = \frac{1}{r}\frac{\partial\psi}{\partial\theta} = V_\infty \cos\theta\left(1-\frac{a^2}{r^2}\right) \qquad u_\theta = -\frac{\partial\psi}{\partial r} = -V_\infty \sin\theta\left(1+\frac{a^2}{r^2}\right) \tag{1}$$

Now we integrate to obtain the velocity potential function. We begin by integrating the expression for u_r in Eq. 1,

$$u_r = \frac{\partial\phi}{\partial r} = V_\infty \cos\theta\left(1-\frac{a^2}{r^2}\right) \quad \rightarrow \quad \phi = V_\infty \cos\theta\left(r+\frac{a^2}{r}\right) + f(\theta) \tag{2}$$

We differentiate Eq. 2 with respect to θ and divide by r to get

$$u_\theta = \frac{1}{r}\frac{\partial\phi}{\partial\theta} = -V_\infty \sin\theta\left(1+\frac{a^2}{r^2}\right) + \frac{f'(\theta)}{r} = -V_\infty \sin\theta\left(1+\frac{a^2}{r^2}\right) \tag{3}$$

Equation 3 reduces to $f'(\theta) = 0$, or $f(\theta)$ = constant. The constant is arbitrary, and we set it to zero for convenience. Hence, Eq. 2 reduces to

Velocity potential, flow over a cylinder:
$$\boxed{\phi = V_\infty \cos\theta\left(r+\frac{a^2}{r}\right)} \tag{4}$$

Method (2): The velocity potential functions for a freestream and a doublet are superposed (added) to yield

Superposition:
$$\phi = V_\infty r\cos\theta + K\frac{\cos\theta}{r} \tag{5}$$

To find the doublet strength (K), we set the radial velocity component u_r to zero at the cylinder surface ($r = a$),

$$u_r = \frac{\partial\phi}{\partial r} = V_\infty \cos\theta - K\frac{\cos\theta}{r^2} \quad \rightarrow \quad 0 = V_\infty \cos\theta - K\frac{\cos\theta}{a^2} \tag{6}$$

Equation 6 reduces to $K = a^2V_\infty$. Hence, Eq. 5 becomes

Velocity potential, flow over a cylinder:
$$\boxed{\phi = V_\infty \cos\theta\left(r+\frac{a^2}{r}\right)} \tag{7}$$

Discussion Both methods yield the same answer, as they must.

10-71

Solution We are to analyze potential flow over a 2-D half-body (the Rankine half-body).

Assumptions **1** The flow is 2-D, incompressible, and steady. **2** The flow is irrotational everywhere.

Analysis (*a*) We superpose a uniform stream and a line source. Working in cylindrical coordinates,

Superposition:
$$\psi = V_\infty r\sin\theta + \frac{\dot{V}/L}{2\pi}\theta \tag{1}$$

To calculate ψ on the dividing streamline, we set $r = a$ and $\theta = \pi$, which yields

Dividing streamline:
$$\psi_{\text{dividing}} = V_\infty a \cancel{\sin\pi} + \frac{\dot{V}/L}{2\pi}\pi \quad \rightarrow \quad \boxed{\psi_{\text{dividing}} = \frac{\dot{V}/L}{2}} \tag{2}$$

(*b*) To calculate the half-height b we look very far downstream where the flow returns to a uniform stream with $\psi = V_\infty y$. Since we know the value of ψ on the dividing streamline, and we set $y = b$ there, we calculate

Half-height:
$$\psi_{\text{dividing}} = \frac{\dot{V}/L}{2} = V_\infty b \quad \rightarrow \quad \boxed{b = \frac{\dot{V}/L}{2V_\infty}}$$

(*c*) To generate an equation for the dividing streamline, we plug Eq. 2 into Eq. 1,

Dividing streamline:
$$\psi_{\text{dividing}} = \frac{\dot{V}/L}{2} = V_\infty r\sin\theta + \frac{\dot{V}/L}{2\pi}\theta \quad \rightarrow \quad \boxed{r = \frac{\dot{V}/L}{2\pi V_\infty \sin\theta}\left(\pi - \theta\right)} \tag{3}$$

(*d*) To calculate the location of the stagnation point, we set $r = a$ and $\theta = \pi$ in Eq. 3. Since both numerator and denominator go to zero, we use L'Hopital's rule to get

Stagnation point:
$$a = \frac{\dot{V}/L}{2\pi V_\infty}\frac{(\pi-\theta)}{\sin\theta} = \frac{\dot{V}/L}{2\pi V_\infty}\lim_{\theta\to\pi}\left[\frac{(\pi-\theta)}{\sin\theta}\right] = \frac{\dot{V}/L}{2\pi V_\infty}\frac{-1}{\cos\pi} \quad \rightarrow \quad \boxed{a = \frac{\dot{V}/L}{2\pi V_\infty}} \tag{4}$$

(*e*) To generate an expression for $(V/V_\infty)^2$, we differentiate Eq. 1 to find u_r and u_θ, and then add the squares of these two velocity components,

Velocity components:
$$u_r = \frac{1}{r}\frac{\partial\psi}{\partial\theta} = V_\infty\cos\theta + \frac{\dot{V}/L}{2\pi r} \quad \text{and} \quad u_\theta = -\frac{\partial\psi}{\partial r} = -V_\infty\sin\theta$$

and

Velocity magnitude squared:
$$V^2 = u_r^{\ 2} + u_\theta^{\ 2} = V_\infty^{\ 2}\cos^2\theta + 2V_\infty\cos\theta\frac{\dot{V}/L}{2\pi r} + \left(\frac{\dot{V}/L}{2\pi r}\right)^2$$

which, after some algebra yields

$$\boxed{\left(\frac{V}{V_\infty}\right)^2 = 1 + \left(\frac{\dot{V}/L}{2\pi r V_\infty}\right)^2 + \frac{\dot{V}/L}{\pi r V_\infty}\cos\theta} \tag{5}$$

Or, substituting Eq. 4 into Eq. 5,

$$\boxed{\left(\frac{V}{V_\infty}\right)^2 = 1 + \left(\frac{a}{r}\right)^2 + \frac{2a}{r}\cos\theta} \tag{6}$$

Discussion You can verify Eq. 4 by setting $V = 0$, $r = a$, and $\theta = \pi$ in Eq. 5, since this is the stagnation point at the nose of the body.

Boundary Layers

10-72C

Solution We are to name three flows (other than flow along a wall) where the boundary layer approximation is appropriate, and we are to explain why.

Analysis The boundary layer approximation is appropriate for the three basic types of shear layers: **wakes**, **jets**, and **mixing layers**. These flows have a predominant flow direction, and for high Reynolds numbers, the shear layer is very *thin*, causing the viscous terms to be much smaller than the inertial terms, just as in the case of a boundary layer along a wall.

Discussion For the wake and the mixing layer, there is an irrotational outer flow in the streamwise direction, but for the jet, the flow outside the jet is nearly stagnant.

10-73C

Solution

(*a*) ***False***: If the Reynolds number at a given x location were to increase, all else being equal, viscous forces would decrease in magnitude relative to inertial forces, rendering the boundary layer thinner.

(*b*) ***False***: Actually, as V increases, so does Re, and the boundary layer thickness decreases with increasing Reynolds number.

(*c*) ***True***: Since μ appears in the denominator of the Reynolds number, Re decreases as μ increases, causing the boundary layer thickness to increase.

(d) ***False***: Since ρ appears in the numerator of the Reynolds number, Re increases as ρ increases, causing the boundary layer thickness to decrease.

10-74C

Solution We are to explain why the boundary layer approximation bridges the gap between the Euler equation and the Navier-Stokes equation.

Analysis The Euler equation neglects the viscous terms compared to the inertial terms. For external flow around a body, this is a reasonable approximation over the majority of the flow field, except very close to the body, where viscous effects dominate. The Navier-Stokes equation, on the other hand, includes both viscous and inertial terms, but is much more difficult to solve. The boundary layer equations bridge the gap between these two: **we solve the simpler Euler equation away from walls, and then fit in a thin boundary layer to account for the no-slip condition at walls**.

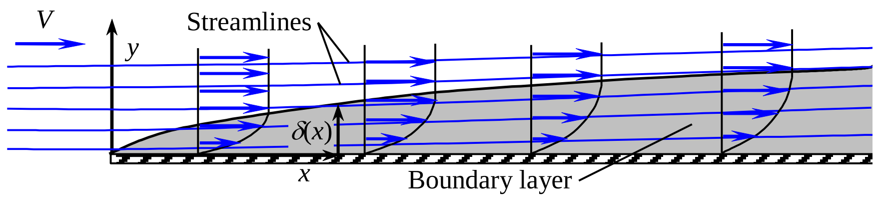
Discussion Students' discussions should be in their own words.

10-75C

Solution We are to sketch several streamlines and discuss whether the curve representing $\delta(x)$ is a streamline or not.

Analysis **Five streamlines are sketched in Fig. 1**. In order to satisfy conservation of mass, the streamlines must cross the curve $\delta(x)$. Thus, $\delta(x)$ cannot itself be a streamline of the flow.

FIGURE 1
Several streamlines and the curve representing δ as a function of x for a flat plate boundary layer. Since streamlines cross the curve $\delta(x)$, $\delta(x)$ cannot itself be a streamline of the flow.



Discussion As the boundary layer grows in thickness, streamlines diverge slowly away from the wall (and become farther apart from each other) in order to conserve mass. However, the upward displacement of the streamlines is not as fast as the growth of $\delta(x)$.

10-76C

Solution We are to define trip wire and explain its purpose.

Analysis A trip wire is **a rod or wire stretched normal to the streamwise direction along the wall**. Its purpose is to create a large disturbance in the laminar boundary layer that causes the boundary layer to "trip" to turbulence much more quickly than it would otherwise.

Discussion Dimples on a golf ball serve the same purpose.

10-77C

Solution We are to discuss the role of an inflection point in a boundary layer profile.

Analysis As sketched in Fig. 10-124, **the existence of an inflection point in the boundary layer profile indicates an adverse or unfavorable pressure gradient**. The reason for this is due to the fact that the second derivative of the velocity profile $u(y)$ at the wall is directly proportional to the pressure gradient (Eq. 10-86). In an adverse pressure gradient field, dP/dx is *positive*, and thus, $\partial^2 u/\partial y^2)_{y=0}$ is also *positive*. However, since $\partial^2 u/\partial y^2$ must be *negative* as u approaches $U(x)$ at the edge of the boundary layer, there has to be an *inflection point* ($\partial^2 u/\partial y^2 = 0$) somewhere in the boundary layer.

Discussion If the adverse pressure gradient is large enough, the boundary layer separates off the wall, leading to reverse flow near the wall.

10-78C

Solution We are to compare laminar and turbulent boundary layer separation, and explain why golf balls have dimples.

Analysis **Turbulent boundary layers are more "full" than are laminar boundary layers**. Because of this, **a turbulent boundary layer is much less likely to separate compared to a laminar boundary layer under the same adverse pressure gradient**. A smooth golf ball, for example, would maintain a laminar boundary layer on its surface, and the boundary layer would separate fairly easily, leading to large aerodynamic drag. **Golf balls have dimples (a type of surface roughness) in order to create an early transition to a turbulent boundary layer**. Flow still separates from the golf ball surface, but much farther downstream in the boundary layer, resulting in significantly reduced aerodynamic drag.

Discussion Turbulent boundary layers have more skin friction drag than do laminar boundary layers, but this effect is less significant than the pressure drag caused by flow separation. Thus, a rough golf ball (at appropriate Reynolds numbers) ends up with less overall drag, compared to a smooth golf ball at the same conditions.

10-79C

Solution We are to list the five steps of the boundary layer procedure.

Analysis **We list the five steps below**, with a description of each:

Step 1 Solve for the outer flow, ignoring the boundary layer (assuming that the region of flow outside the boundary layer is approximately inviscid and/or irrotational). Transform coordinates as necessary to obtain $U(x)$.

Step 2 Assume a thin boundary layer – so thin in fact that it does not affect the outer flow solution of Step 1.

Step 3 Solve the boundary layer equations. For this step we use the no-slip boundary condition at the wall, $u = v = 0$ at $y = 0$, the known outer flow condition at the edge of the boundary layer, $u \to U(x)$ as $y \to \infty$, and some known starting profile, $u = u_{\text{starting}}(y)$ at $x = x_{\text{starting}}$.

Step 4 Calculate quantities of interest in the flow field. For example, once the boundary layer equations have been solved (Step 3), we can calculate $\delta(x)$, shear stress along the wall, total skin friction drag, etc.

Step 5 Verify that the boundary layer approximations are appropriate. In other words, verify that the boundary layer is indeed *thin* – otherwise the approximation is not justified.

Discussion Students' discussions should be in their own words.

10-80C

Solution We are to list at least three "red flags" to look for when performing boundary layer calculations.

Analysis **We list four below**. (Students are asked to list at least three.)

- The boundary layer approximation breaks down if Reynolds number is not large enough. For example, $\delta/L \sim 0.01$ (1%) for $\text{Re}_L = 10{,}000$.
- The assumption of zero pressure gradient in the y direction breaks down if wall curvature is of similar magnitude as δ. In such cases, centripetal acceleration effects due to streamline curvature cannot be ignored. Physically, the boundary layer is not "thin" enough for the approximation to be appropriate when δ is not $<< R$.
- When Reynolds number is too *high*, the boundary layer does not remain laminar. The boundary layer approximation itself may still be appropriate, but the laminar boundary layer equations are *not* valid if the flow is transitional or fully turbulent. The laminar boundary layer on a smooth flat plate under clean flow conditions begins to transition towards turbulence at $\text{Re}_x \approx 1 \times 10^5$. In practical engineering applications, walls may not be smooth and there may be vibrations, noise, and fluctuations in the freestream flow above the wall, all of which contribute to an even earlier start of the transition process.
- If flow separation occurs, the boundary layer approximation is no longer appropriate in the separated flow region. The main reason for this is that a separated flow region contains *reverse flow*, and the parabolic nature of the boundary layer equations is lost.

Discussion Students' discussions should be in their own words.

10-81C

Solution We are to define displacement thickness and discuss whether it is larger or smaller than boundary layer thickness.

Assumptions **1** The flow is steady and incompressible. **2** The boundary layer growing on the flat plate is laminar.

Analysis The two definitions of displacement thickness are:

- Displacement thickness is the distance that a streamline just outside of the boundary layer is deflected away from the wall due to the effect of the boundary layer.
- Displacement thickness is the imaginary increase in thickness of the wall, as seen by the outer flow, due to the effect of the growing boundary layer.

For a laminar boundary layer, **δ is larger than δ^***. δ is defined by the overall thickness of the boundary layer, whereas δ^* is an integrated thickness across the boundary layer that averages the mass deficit across the boundary layer. Therefore, it is not surprising that δ^* is less than δ.

Discussion The definitions given by students should be in their own words.

10-82C

Solution We are to discuss the difference between a favorable and an adverse pressure gradient.

Analysis **When the pressure decreases downstream, the boundary layer is said to experience to a favorable pressure gradient. When the pressure increases downstream, the boundary layer is subjected to an adverse pressure gradient.** The term “favorable” is used because the boundary layer is unlikely to separate off the wall. On the other hand, “adverse” or “unfavorable” indicates that the boundary layer is more likely to separate off the wall.

Discussion A favorable pressure gradient occurs typically at the front of a body, whereas an adverse pressure gradient occurs typically at the back portion of a body.

10-83

Solution We are to calculate the location of transition and turbulence along a flat plate boundary layer.

Assumptions **1** The flow is incompressible and steady in the mean. **2** Free stream disturbances are small. **3** The surface of the plate is very smooth.

Properties The density and viscosity of air at $T = 30^{\circ}$C are 1.164 kg/m^3 and 1.872×10^{-5} kg/m·s respectively.

Analysis Transition begins at the critical Reynolds number, which is approximately 100,000 for "clean" flow along a smooth flat plate. Thus,

Beginning of transition:
$$\mathrm{Re}_{x,\mathrm{critical}} = \frac{\rho V x_{\mathrm{critical}}}{\mu} = 100{,}000 \qquad (1)$$

Solving for x yields

$$x_{\mathrm{critical}} = \frac{100{,}000\mu}{\rho V} = \frac{100{,}000\left(1.872\times10^{-5}\ \mathrm{kg/m\cdot s}\right)}{\left(1.164\ \mathrm{kg/m^3}\right)\left(29.1\ \mathrm{m/s}\right)} = 5.53\times10^{-2}\ \mathrm{m} \qquad (2)$$

That is, transition begins at ***x*** **≈ 5 or 6 cm**. Fully turbulent flow in the boundary layer occurs at approximately 30 times x_{critical}, at $\mathrm{Re}_{x,\mathrm{transition}} \approx 3 \times 10^6$. So, the boundary layer is expected to be fully turbulent by $x \approx 1.5$ to 1.8 m, or to one digit, ***x*** **≈ 2 m**.

Discussion Final results are given to only one significant digit, since the locations of transition and turbulence are only approximations. The actual locations are influenced by many things, such as noise, roughness, vibrations, free stream disturbances, etc.

10-84E

Solution We are to assess whether the boundary layer on the hull of a boat is laminar or turbulent or transitional.

Assumptions **1** The flow is steady and incompressible. **2** The hull surface is smooth.

Properties The density and viscosity of water at $T = 40^{\circ}$F are 62.42 lbm/ft^3 and 1.038×10^{-3} lbm/ft·s, respectively. The kinematic viscosity is thus $\nu = 1.663 \times 10^{-5}$ ft^2/s.

Analysis Although the hull is not a flat plate, the flat plate boundary layer values are useful as a reasonable approximation to determine whether the boundary layer is laminar or turbulent. We calculate the Reynolds number at the trailing edge of the hull, using L as the approximate streamwise distance along the flat plate,

$$\mathrm{Re}_x = \frac{VL}{\nu} = \frac{\left(26.0\ \mathrm{mi/hr}\right)\left(2.4\ \mathrm{ft}\right)}{1.663\times10^{-5}\ \mathrm{ft^2/s}}\left(\frac{5280\ \mathrm{ft}}{\mathrm{mi}}\right)\left(\frac{\mathrm{hr}}{3600\ \mathrm{s}}\right) = 5.50\times10^{6}$$

The critical Reynolds number for transition to turbulence is 1×10^5 for the case of a smooth flat plate with very clean, low-noise free stream conditions. Our Reynolds number is much higher than this. The engineering value of critical Reynolds number for real engineering flows is $\mathrm{Re}_{x,cr} = 5 \times 10^5$. Our Re_x is much higher than this. The typical value where a smooth flat plate BL becomes fully turbulent is $\mathrm{Re}_{x,\mathrm{transition}}$ (3×10^6), and our Re_x is much higher than this. Thus, **the boundary layer is fully turbulent by the trailing edge of the hull**.

Discussion In a real-life situation, the free stream flow is not very "clean" – there are eddies and other disturbances, the hull surface is not perfectly smooth, and the vehicle may be vibrating. Thus, transition and turbulence are likely to occur much earlier than predicted for a smooth flat plate, and the boundary layer is probably fully turbulent for most of the hull.

10-85

Solution We are to assess whether the boundary layer on the surface of a sign is laminar or turbulent or transitional.

Assumptions **1** The flow is steady and incompressible. **2** The sign surface is smooth.

Properties The density and viscosity of air at T = 25°C are 1.184 kg/m^3 and 1.849 × 10^{-5} kg/m·s respectively. The kinematic viscosity is thus ν = 1.562 × 10^{-5} m^2/s.

Analysis Since the air flow is parallel to the sign, this flow is that of a flat plate boundary layer. We calculate the Reynolds number at the downstream edge of the sign, using W as the streamwise distance along the flat plate,

$$\mathrm{Re}_x = \frac{VW}{\nu} = \frac{(8.5\ \mathrm{m/s})(0.45\ \mathrm{m})}{1.562\times10^{-5}\ \mathrm{m^2/s}} = 2.45\times10^5 \qquad (1)$$

The critical Reynolds number for transition to turbulence is 1 × 10^5 for the case of a smooth flat plate with very clean, low-noise freestream conditions. Our Reynolds number is higher than this, but just barely so. The engineering value of critical Reynolds number for transition to a turbulent boundary layer in real engineering flows is $\mathrm{Re}_{x,cr}$ = 5 × 10^5; our value of Re_x is less than $\mathrm{Re}_{x,cr}$. Since Re_x is a bit greater than $\mathrm{Re}_{x,critical}$, but less than $\mathrm{Re}_{x,cr}$ (5 × 10^5), and much less than $\mathrm{Re}_{x,transition}$ (30 × 10^5), **the boundary layer is laminar for a while, and then becomes transitional by the trailing edge of the fin**.

Discussion The flow over the sign is not very "clean" – there are eddies from the passing vehicles, and other atmospheric disturbances. In addition, the sign surface is not perfectly smooth, and most signs tend to oscillate somewhat in the wind. Thus, transition and turbulence are likely to occur much earlier than predicted for a smooth flat plate. The boundary layer on this sign is definitely transitional, but probably not turbulent, by the downstream edge of the sign.

10-86E

Solution We are to assess whether the boundary layer on the wall of a wind tunnel is laminar or turbulent or transitional.

Assumptions **1** The flow is steady and incompressible. **2** The surface of the wind tunnel is smooth. **3** There are minimal disturbances in the freestream flow.

Properties The density and viscosity of air at T = 80°F are 0.07350 lbm/ft^3 and 1.247 × 10^{-5} lbm/ft·s respectively. The kinematic viscosity is thus ν = 1.697 × 10^{-4} ft^2/s.

Analysis We calculate the Reynolds number at the downstream end of the wall, using L = 1.5 ft as the streamwise distance along the flat plate,

$$\mathrm{Re}_x = \frac{VL}{\nu} = \frac{(7.5\ \mathrm{ft/s})(1.5\ \mathrm{ft})}{1.697\times10^{-4}\ \mathrm{ft^2/s}} = 6.63\times10^4 \qquad (1)$$

The engineering value of critical Reynolds number for transition to a turbulent boundary layer in real engineering flows is $\mathrm{Re}_{x,cr}$ = 5 × 10^5; our value of Re_x is much less than $\mathrm{Re}_{x,cr}$. In fact, our Reynolds number is even lower than the critical Reynolds number for transition to turbulence (1 × 10^5) for the case of a smooth flat plate with very clean, low-noise freestream conditions. Since the flow is clean and Re_x is less than $\mathrm{Re}_{x,critical}$, **the boundary layer is definitely laminar**.

Discussion There is typically a contraction just upstream of the test section of a wind tunnel. Upstream of that are typically some screens and/or honeycombs to make the flow clean and uniform. Thus, the disturbances are likely to be quite small, and the boundary layer is most likely laminar.

10-87

Solution We are to generate an expression for the outer flow velocity at point 2 in the boundary layer.

Assumptions **1** The flow is steady, incompressible, and laminar. **2** The boundary layer approximation is appropriate.

Analysis The boundary layer approximation tells us that P is constant *normal* to the boundary layer, but not necessarily *along* the boundary layer. Therefore, at any streamwise location along the boundary layer, the pressure in the outer flow region just above the boundary layer is the same as that at the wall. In the outer flow region, the Bernoulli equation reduces to

Outer flow:
$$U\frac{dU}{dx} = -\frac{1}{\rho}\frac{dP}{dx} \quad \rightarrow \quad \frac{dU}{dx} = -\frac{1}{\rho U}\frac{dP}{dx} \qquad (1)$$

For small values of Δx, we can approximate $U_2 \approx U_1 + (dU/dx)\Delta x$, and $P_2 \approx P_1 + (dP/dx)\Delta x$. Substitution of these approximations into Eq. 1 yields

$$U_2 \approx U_1 - \frac{1}{\rho U_1}\frac{dP}{dx}\Delta x = U_1 - \frac{1}{\rho U_1}\frac{P_2 - P_1}{\Delta x}\Delta x \quad \rightarrow \quad \boxed{U_2 \approx U_1 - \frac{P_2 - P_1}{\rho U_1}} \qquad (2)$$

Discussion It turns out that U_2 does not depend on Δx or μ, but only on P_1, P_2, U_1, and ρ.

10-88

Solution We are to estimate U_2, and explain whether it is less than, equal to, or greater than U_1.

Assumptions **1** The flow is steady, incompressible, and laminar. **2** The boundary layer approximation is appropriate.

Properties The density and viscosity of air at $T = 25^{\circ}$C are 1.184 kg/m^3 and 1.849×10^{-5} kg/m·s respectively.

Analysis The Bernoulli equation is valid in the outer flow region. Thus, we know that as P increases, U decreases, and vice-versa. In this case, P increases, and thus **we expect U_2 to be less than U_1**. From the results of Problem 10-74,

$$U_2 \approx U_1 - \frac{P_2 - P_1}{\rho U_1} = 10.3\frac{\text{m}}{\text{s}} - \frac{2.44\ \text{N/m}^2}{\left(1.186\ \text{kg/m}^3\right)\left(10.3\ \text{m/s}\right)}\left(\frac{\text{kg m}}{\text{N s}^2}\right) = \mathbf{10.1\ m/s}$$

Thus, the outer flow velocity indeed decreases by a small amount.

Discussion The approximation is first order, and thus appropriate only if the distance between x_1 and x_2 is small.

10-89

Solution We are to prove that $\tau_w = 0.332\dfrac{\rho U^2}{\sqrt{\text{Re}_x}}$ for a flat plate boundary layer.

Assumptions **1** The flow is steady and incompressible. **2** The Reynolds number is in the range where the Blasius solution is appropriate.

Analysis Equation 4 of Example 10-10 gives the definition of similarity variable η, which we re-write in terms of y as a function of η,

y as a function of η:

$$\eta = y\sqrt{\frac{U}{\nu x}} \quad \rightarrow \quad y = \eta\sqrt{\frac{\nu x}{U}} \tag{1}$$

From the chain rule and Eq. 1, we obtain an expression for $d/d\eta$,

Derivative with respect to similarity variable η:

$$\frac{d}{d\eta} = \frac{d}{dy}\frac{dy}{d\eta} = \frac{d}{dy}\sqrt{\frac{\nu x}{U}} \tag{2}$$

We apply Eq. 2 above to Eq. 8 of Example 10-10,

$$\left.\frac{d\left(u/U\right)}{d\eta}\right)_{\eta=0} = \left.\frac{du}{dy}\right)_{y=0}\sqrt{\frac{\nu x}{U^3}} = 0.332 \tag{3}$$

But by definition, $\tau_w = \mu\left.\dfrac{\partial u}{\partial y}\right)_{y=0}$, and Eq. 3 yields

Shear stress at the wall:

$$\tau_w = \mu\left.\frac{du}{dy}\right)_{y=0}\sqrt{\frac{\nu x}{U^3}} = 0.332\rho\nu\sqrt{\frac{U^3}{\nu x}} = 0.332\rho U^2\sqrt{\frac{\nu}{Ux}} = 0.332\frac{\rho U^2}{\sqrt{\text{Re}_x}} \tag{4}$$

which is the desired expression for the shear stress at the wall in physical variables.

Discussion The chain rule algebra is valid here since *U* and *x* are functions of *x* only – they are not functions of *y*.

10-90E

Solution We are to calculate δ, δ^*, and θ at the end of the wind tunnel test section.

Assumptions **1** The flow is steady and incompressible. **2** The surface of the wind tunnel is smooth. **3** The boundary layer remains laminar all the way to the end of the test section.

Properties The density and viscosity of air at $T = 80^{\circ}\text{F}$ are 0.07350 lbm/ft^3 and 1.247×10^{-5} lbm/ft·s respectively.

Analysis In Problem 10-73E, we calculated the Reynolds number at the downstream end of the wall, $\text{Re}_x = 6.63 \times 10^4$ (keeping an extra digit for the calculations). All of the desired quantities are functions of Re_x:

Boundary layer thickness:

$$\delta = \frac{4.91x}{\sqrt{\text{Re}_x}} = \frac{4.91(1.5\text{ ft})}{\sqrt{6.63\times10^4}} = 0.0286\text{ ft} \approx 0.34\text{ in} \tag{1}$$

and

Displacement thickness:

$$\delta^* = \frac{1.72x}{\sqrt{\text{Re}_x}} = \frac{1.72(1.5\text{ ft})}{\sqrt{6.63\times10^4}} = 0.0100\text{ ft} \approx 0.12\text{ in} \tag{2}$$

and

Momentum thickness:

$$\theta = \frac{0.664x}{\sqrt{\text{Re}_x}} = \frac{0.664(1.5\text{ ft})}{\sqrt{6.63\times10^4}} = 0.00387\text{ ft} \approx 0.046\text{ in} \tag{3}$$

Thus, $\boldsymbol{\delta}$ **= 0.34 inches**, $\boldsymbol{\delta^*}$ **= 0.12 inches**, and $\boldsymbol{\theta}$ **= 0.046 inches** at the end of the wind tunnel test section. As expected, $\delta > \delta^* > \theta$.

Discussion All answers are given to two significant digits.

10-91

Solution We are to calculate H for an infinitesimally thin boundary layer.

Analysis By definition,

Shape factor:

$$H = \frac{\delta^*}{\theta} = \frac{\int_0^\infty \left(1-\frac{u}{U}\right)dy}{\int_0^\infty \frac{u}{U}\left(1-\frac{u}{U}\right)dy} \tag{1}$$

But for the limiting case under consideration, $u/U = 1$ through the entire boundary layer, yielding $\delta^* = 0$ and $\theta = 0$. To calculate the ratio in Eq. 1, we use l'Hopital's rule, where the variable u approaches U in the limit,

$$H = \lim_{u\to U}\frac{\frac{d}{du}\int_0^\infty \left(1-\frac{u}{U}\right)dy}{\frac{d}{du}\int_0^\infty \frac{u}{U}\left(1-\frac{u}{U}\right)dy} = \lim_{u\to U}\frac{-\int_0^\infty \frac{1}{U}dy}{\int_0^\infty \left(\frac{1}{U}-2\frac{u}{U}\frac{1}{U}\right)dy} = \frac{\int_0^\infty \frac{1}{U}dy}{\int_0^\infty \frac{1}{U}dy} = \mathbf{1}$$

In other words, H is always greater than unity for any real boundary layer.

Discussion Since turbulent boundary layers are fuller than laminar boundary layers, it is no surprise that $H_{\text{turbulent}}$ is closer to unity than is H_{laminar}.

10-92

Solution The acceleration of air through the round test section of a wind tunnel is to be calculated.

Assumptions **1** The flow is steady and incompressible. **2** The walls are smooth, and disturbances and vibrations are kept to a minimum. **3** The boundary layers is laminar.

Properties The kinematic viscosity of air at 20°C is ν = 1.516 × 10^{-5} m^2/s.

Analysis (*a*) As the boundary layer grows along the wall of the wind tunnel test section, air in the region of irrotational flow in the central portion of the test section accelerates in order to satisfy conservation of mass. The Reynolds number at the end of the test section is

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(2.0\ \mathrm{m/s})(0.80\ \mathrm{m})}{1.516\times10^{-5}\ \mathrm{m^2/s}} = 1.055\times10^5$$

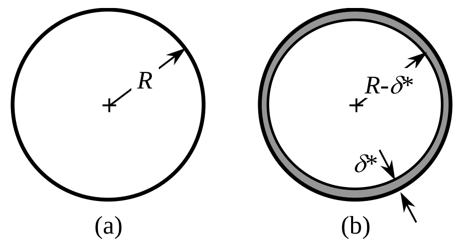


FIGURE 1
Cross-sectional views of the test section of the wind tunnel: (a) beginning of test section, and (b) end of test section.

The engineering value of critical Reynolds number for transition to a turbulent boundary layer in real engineering flows is $\mathrm{Re}_{x,\mathrm{cr}} = 5 \times 10^5$; our value of Re_x is less than $\mathrm{Re}_{x,\mathrm{cr}}$. In fact, Re_x is lower than the critical Reynolds number, $\mathrm{Re}_{x,\mathrm{critical}} \approx 1 \times 10^5$, for a smooth flat plate with a clean free stream. Since the walls are smooth and the flow is clean, we may assume that the boundary layer on the wall remains laminar throughout the length of the test section. We estimate the displacement thickness at the end of the test section,

$$\delta^* \approx \frac{1.72x}{\sqrt{\mathrm{Re}_x}} = \frac{1.72(0.80\ \mathrm{m})}{\sqrt{1.055\times10^5}} = 0.00424\ \mathrm{m} = 4.24\ \mathrm{mm} \qquad (1)$$

Two cross-sectional views of the test section are sketched in Fig. 1, one at the beginning and one at the end of the test section. The effective radius at the end of the test section is reduced by δ^* as calculated by Eq. 1. We apply conservation of mass to calculate the air speed at the end of the test section,

$$V_{\mathrm{end}}A_{\mathrm{end}} = V_{\mathrm{beginning}}A_{\mathrm{beginning}} \quad \rightarrow \quad V_{\mathrm{end}} = V_{\mathrm{beginning}}\frac{\pi R^2}{\pi\left(R-\delta^*\right)^2} \qquad (2)$$

We plug in the numerical values to obtain

Result:

$$V_{\mathrm{end}} = (2.0\ \mathrm{m/s})\frac{(0.15\ \mathrm{m})^2}{(0.15-0.00423\ \mathrm{m})^2} = \mathbf{2.12\ m/s} \qquad (3)$$

Thus **the air speed increases by approximately 6%** through the test section, due to the effect of displacement thickness.

Discussion The same displacement thickness technique may be applied to turbulent boundary layers; however, a different equation for $\delta^*(x)$ is required.

10-93

Solution The acceleration of air through the square test section of a wind tunnel is to be calculated and compared to that through a round wind tunnel test section.

Assumptions **1** The flow is steady and incompressible. **2** The walls are smooth, and disturbances and vibrations are kept to a minimum. **3** The boundary layers is laminar.

Properties The kinematic viscosity of air at 20°C is $\nu = 1.516 \times 10^{-5}$ m²/s.

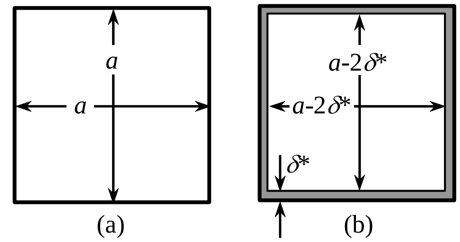


FIGURE 1
Cross-sectional views of the test section of the wind tunnel: (a) beginning of test section, and (b) end of test section.

Analysis (*a*) As the boundary layer grows along the walls of the wind tunnel test section, air in the region of irrotational flow in the central portion of the test section accelerates in order to satisfy conservation of mass. The Reynolds number at the end of the test section is

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(2.0\ \mathrm{m/s})(0.80\ \mathrm{m})}{1.516\times10^{-5}\ \mathrm{m^2/s}} = 1.055\times10^5$$

The engineering value of critical Reynolds number for transition to a turbulent boundary layer in real engineering flows is $\mathrm{Re}_{x,cr} = 5 \times 10^5$; our value of Re_x is less than $\mathrm{Re}_{x,cr}$. In fact, Re_x is lower than the critical Reynolds number, $\mathrm{Re}_{x,\mathrm{critical}} \approx 1 \times 10^5$, for a smooth flat plate with a clean free stream. Since the walls are smooth and the flow is clean, we may assume that the boundary layer on the wall remains laminar throughout the length of the test section. We estimate the displacement thickness at the end of the test section,

$$\delta^* \approx \frac{1.72x}{\sqrt{\mathrm{Re}_x}} = \frac{1.72(0.80\ \mathrm{m})}{\sqrt{1.055\times10^5}} = 0.00424\ \mathrm{m} = 4.24\ \mathrm{mm} \qquad (1)$$

Two cross-sectional views of the test section are sketched in Fig. 1, one at the beginning and one at the end of the test section. The effective dimensions at the end of the test section are reduced by $2\delta^*$. We apply conservation of mass to calculate the air speed at the end of the test section,

$$V_{\mathrm{end}}A_{\mathrm{end}} = V_{\mathrm{beginning}}A_{\mathrm{beginning}} \quad \rightarrow \quad V_{\mathrm{end}} = V_{\mathrm{beginning}}\frac{a^2}{\left(a-2\delta^*\right)^2} \qquad (2)$$

We plug in the numerical values to obtain

Result:

$$V_{\mathrm{end}} = (2.0\ \mathrm{m/s})\frac{(0.30\ \mathrm{m})^2}{(0.30-2\times0.00423\ \mathrm{m})^2} = \mathbf{2.12\ m/s} \qquad (3)$$

Thus **the air speed increases by approximately 6% through the test section**, due to the effect of displacement thickness.

The result for the square test section is identical to that of the round test section. We might have expected the square test section to do better since its cross-sectional area is larger than that of the round test section. However, the square test section also has more wall surface area than does the round test section, and thus, the acceleration due to displacement thickness on the walls is the same in both cases.

Discussion The same displacement thickness technique may be applied to turbulent boundary layers; however, a different equation for $\delta^*(x)$ is required.

10-94

Solution The apparent thickness of a flat plate is to be calculated.

Assumptions **1** The flow is steady and incompressible. **2** The walls are smooth. **3** The boundary layers starts growing at x = 0.

Properties The kinematic viscosity of air at 20°C is $\nu = 1.516 \times 10^{-5}$ m²/s.

Analysis (*a*) As the boundary layer grows along the plate, the Reynolds number increases. The Reynolds number at location x is

Re$_x$:
$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(8.5\ \mathrm{m/s})(0.10\ \mathrm{m})}{1.516\times10^{-5}\ \mathrm{m^2/s}} = 5.607\times10^4$$

The engineering value of critical Reynolds number for transition to a turbulent boundary layer in real engineering flows is $\mathrm{Re}_{x,cr} = 5 \times 10^5$; our value of Re_x is less than $\mathrm{Re}_{x,cr}$. In fact, Re_x is lower than the critical Reynolds number, $\mathrm{Re}_{x,\mathrm{critical}} \approx 1 \times 10^5$, for a smooth flat plate with a clean free stream. Since Re_x is lower than the critical Reynolds number, and since the walls are smooth and the flow is clean, we may assume that the boundary layer on the wall remains laminar, at least to location x. We estimate the displacement thickness at x = 10 cm,

$$\delta^* \approx \frac{1.72x}{\sqrt{\mathrm{Re}_x}} = \frac{1.72(0.10\ \mathrm{m})}{\sqrt{5.607\times10^4}} = 7.264\times10^{-4}\ \mathrm{m} = 0.07264\ \mathrm{cm} \qquad (1)$$

This "extra" thickness is seen by the outer flow. Since the plate is 0.75 cm thick, and since a similar boundary layer forms on the bottom as on the top, the total apparent thickness of the plate is

Apparent thickness:
$$h_{\mathrm{apparent}} = h + 2\delta^* = 0.75\ \mathrm{cm} + 2(0.07264\ \mathrm{cm}) = \mathbf{0.895\ cm}$$

Discussion We have kept 5 digits of precision in intermediate steps, but report our final answer to three significant digits. The Reynolds number is pretty close to critical. If the freestream air flow were noisy and/or the plate were rough or vibrating, we might expect the boundary layer to be transitional, and then the apparent thickness would be greater.

10-95E

Solution The acceleration of air through the round test section of a wind tunnel is to be calculated.

Assumptions **1** The flow is steady and incompressible. **2** The walls are smooth, and disturbances and vibrations are kept to a minimum. **3** The boundary layers is laminar.

Properties The kinematic viscosity of air at 70°F is $\nu = 1.643 \times 10^{-4}$ ft^2/s.

Analysis (*a*) As the boundary layer grows along the wall of the wind tunnel test section, air in the region of irrotational flow in the central portion of the test section accelerates in order to satisfy conservation of mass. The Reynolds number at the end of the test section is

$$\text{Re}_x = \frac{Vx}{\nu} = \frac{(5.0 \text{ ft/s})(10.0 \text{ in})}{1.643\times10^{-4}\text{ ft}^2\text{/s}}\left(\frac{1\text{ft}}{12 \text{ in}}\right) = 2.54\times10^4$$

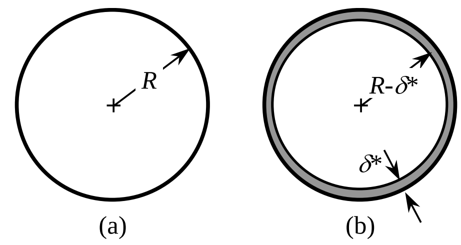


FIGURE 1
Cross-sectional views of the test section of the wind tunnel: (a) beginning of test section, and (b) end of test section.

The engineering value of critical Reynolds number for transition to a turbulent boundary layer in real engineering flows is $\text{Re}_{x,\text{cr}} = 5 \times 10^5$; our value of Re_x is less than $\text{Re}_{x,\text{cr}}$. In fact, Re_x is lower than the critical Reynolds number, $\text{Re}_{x,\text{critical}} \approx 1 \times 10^5$, for a smooth flat plate with a clean free stream. Since the walls are smooth and the flow is clean, we may assume that the boundary layer on the wall remains laminar throughout the length of the test section. We estimate the displacement thickness at the end of the test section,

$$\delta^* \approx \frac{1.72x}{\sqrt{\text{Re}_x}} = \frac{1.72(10.0 \text{ in})}{\sqrt{2.54\times10^4}} = 0.108 \text{ in} \tag{1}$$

Two cross-sectional views of the test section are sketched in Fig. 1, one at the beginning and one at the end of the test section. The effective radius at the end of the test section is reduced by δ^* as calculated by Eq. 1. We apply conservation of mass to calculate the air speed at the end of the test section,

$$V_{\text{end}}A_{\text{end}} = V_{\text{beginning}}A_{\text{beginning}} \quad \rightarrow \quad V_{\text{end}} = V_{\text{beginning}}\frac{\pi R^2}{\pi(R-\delta^*)^2} \tag{2}$$

We plug in the numerical values to obtain

Result:

$$V_{\text{end}} = (5.0 \text{ ft/s})\frac{(6.68 \text{ in})^2}{(6.68 \text{ in} - 0.108 \text{ in})^2} = 5.16568 \text{ ft/s} \cong \mathbf{5.17\ ft/s} \tag{3}$$

Thus the air speed increases by approximately 3.3% through the test section, due to the effect of displacement thickness.

To eliminate this acceleration, the engineers can either diverge the test section walls, or add some suction along the sides to remove some air.

Discussion The same displacement thickness technique may be applied to turbulent boundary layers; however, a different equation for $\delta^*(x)$ is required.

10-96E

Solution We are to determine if a boundary layer is laminar, turbulent, or transitional, and then compare the laminar and turbulent boundary layer thicknesses.

Properties The kinematic viscosity of air at 70°F is $\nu = 1.643 \times 10^{-4}$ ft²/s.

Analysis First, we calculate the Reynolds number at the end of the plate,

Re$_x$, end of plate:
$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(15.5 \text{ ft/s})(10.6 \text{ ft})}{1.643\times 10^{-4} \text{ ft}^2\text{/s}} = 1.00\times 10^6$$

The engineering value of critical Reynolds number for transition to a turbulent boundary layer in real engineering flows is $\mathrm{Re}_{x,\mathrm{cr}} = 5 \times 10^5$; our value of Re_x is greater than $\mathrm{Re}_{x,\mathrm{cr}}$, leading us to suspect that the boundary layer is turbulent. However, Re_x is lower than the transition Reynolds number, $\mathrm{Re}_{x,\mathrm{transition}} \approx 3 \times 10^6$, for a smooth flat plate with a clean free stream. Thus, we suspect that **this boundary layer is laminar at the front of the plate, and then transitional farther downstream**. If the plate is vibrating and/or the freestream is noisy, the boundary layer may possibly be fully turbulent by the end of the plate.

If the boundary layer were to remain laminar to the end of the plate, its thickness would be

Laminar:
$$\delta = \frac{4.91x}{\sqrt{\mathrm{Re}_x}} = \frac{4.91(10.6 \text{ ft})}{\sqrt{1.00\times 10^6}} = 0.0520 \text{ ft} = \mathbf{0.625 \text{ in}} \qquad (1)$$

If the boundary layer at the end of the plate were fully turbulent (and turbulent from the beginning of the plate), its thickness would be

Turbulent, Table 10-4a:
$$\delta \approx \frac{0.16x}{(\mathrm{Re}_x)^{1/7}} = \frac{0.16(10.6 \text{ ft})}{(1.00\times 10^6)^{1/7}} = 0.236 \text{ ft} = \mathbf{2.83 \text{ in}} \qquad (1)$$

or,

Turbulent, Table 10-4b:
$$\delta \approx \frac{0.38x}{(\mathrm{Re}_x)^{1/5}} = \frac{0.38(10.6 \text{ ft})}{(1.00\times 10^6)^{1/5}} = 0.254 \text{ ft} = \mathbf{3.05 \text{ in}} \qquad (1)$$

Thus, the turbulent boundary layer thickness is about 4.5 to 4.9 times thicker than the corresponding laminar boundary layer thickness at the same Reynolds number. We expect the actual boundary layer thickness to lie somewhere between these two extremes.

Discussion The difference between the two turbulent boundary layer equations for δ is about 7 or 8 percent. This is larger than we might hope, but keep in mind that at $\mathrm{Re}_x = 1.00 \times 10^6$, the boundary layer is not yet fully turbulent, and the equations for δ are not accurate at such low Reynolds numbers.

10-97

Solution The height of a boundary layer scoop in a wind tunnel test section is to be calculated.

Assumptions **1** The flow is steady and incompressible. **2** The walls are smooth. **3** The boundary layers starts growing at x = 0.

Properties The kinematic viscosity of air at 20°C is $\nu = 1.516 \times 10^{-5}$ m²/s.

Analysis (*a*) As the boundary layer grows along the wall of the wind tunnel test section, the Reynolds number increases. The Reynolds number at location x is

Re

$$\text{Re}_x:\qquad \text{Re}_x = \frac{Vx}{\nu} = \frac{(45.0\text{ m/s})(1.45\text{ m})}{1.516\times10^{-5}\,\text{m}^2/\text{s}} = 4.3041\times10^6$$

Since Re_x is greater than the transition Reynolds number, $\text{Re}_{x,\text{transition}} \approx 3 \times 10^6$, we assume that the boundary layer is turbulent throughout the length of the test section. We estimate the boundary layer thickness at the location of the scoop,

Table 10-4a:

$$\delta \approx \frac{0.16x}{(\text{Re}_x)^{1/7}} = \frac{0.16(1.45\text{ m})}{(4.3041\times10^6)^{1/7}} = 2.6169\times10^{-2}\text{ m} \cong 26.2\text{ mm} \qquad (1)$$

or,

Table 10-4b:

$$\delta \approx \frac{0.38x}{(\text{Re}_x)^{1/5}} = \frac{0.38(1.45\text{ m})}{(4.3041\times10^6)^{1/5}} = 2.5964\times10^{-2}\text{ m} \cong 26.0\text{ mm} \qquad (1)$$

We design the scoop height to be greater than or equal to the boundary layer thickness at the location of the scoop. Thus, we set $\boldsymbol{h \approx \delta \approx}$ **26. mm, or about an inch**.

Discussion The suction pressure of the scoop must be adjusted carefully so as not to suck too much or too little – otherwise it would disturb the flow. Since the early portion of the boundary layer is laminar, the actual boundary layer thickness will be somewhat lower than that calculated here. Thus, our calculation represents an upper limit. However, some of the large turbulent eddies in the boundary layer may actually exceed height δ, so our calculated h may actually not be sufficient to remove the complete boundary layer.

10-98

Solution We are to plot the mean boundary layer profile $u(y)$ at the end of a flat plate using three different approximations.

Assumptions **1** The plate is smooth. **2** The boundary layer is turbulent from the beginning of the plate. **3** The flow is steady in the mean. **4** The plate is infinitesimally thin and is aligned parallel to the freestream.

Properties The kinematic viscosity of air at 20°C is $\nu = 1.516 \times 10^{-5}$ m^2/s.

Analysis First we calculate the Reynolds number at $x = L$,

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(80.0\text{ m/s})(17.5\text{ m})}{1.516\times 10^{-5}\text{ m}^2\text{/s}} = 9.23\times 10^7$$

This value of Re_x is well above the transitional Reynolds number for a flat plate boundary layer, so the assumption of turbulent flow from the beginning of the plate is reasonable.

Using the column (a) values of Table 10-4, we estimate the boundary layer thickness and the local skin friction coefficient at the end of the plate,

$$\delta \approx \frac{0.16x}{\left(\mathrm{Re}_x\right)^{1/7}} = 0.204\text{ m} \qquad C_{f,x} \approx \frac{0.027}{\left(\mathrm{Re}_x\right)^{1/7}} = 1.97\times 10^{-3} \tag{1}$$

We calculate the friction velocity by using the definition of $C_{f,x}$,

$$u_* = \sqrt{\tau_w/\rho} = U\sqrt{C_{f,x}/2} = (80.0\text{ m/s})\sqrt{\left(1.97\times 10^{-3}\right)/2} = 2.51\text{ m/s} \tag{2}$$

where $U(x) = V$ = constant for a flat plate. It is trivial to generate a plot of the one-seventh-power law. We follow Example 10-13 to plot the log law, namely,

$$y = \frac{\nu}{u_*}e^{\kappa\left(\frac{u}{u_*}-B\right)} \tag{3}$$

Since we know that u varies from 0 at the wall to U at the boundary layer edge, we are also able to plot the log law velocity profile in physical variables. Finally, Spalding's law of the wall is also written in terms of y as a function of u. **We plot all three profiles on the same plot for comparison (Fig. 1)**. All three are close, and we cannot distinguish the log law from Spalding's law on this scale.

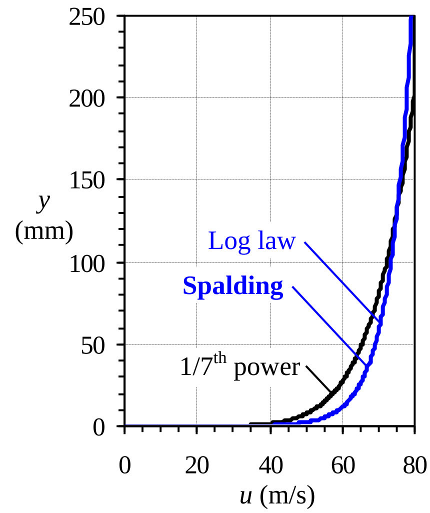


FIGURE 1
Comparison of turbulent flat plate boundary layer profile expressions in physical variables at $\mathrm{Re}_x = 9.23 \times 10^7$: one-seventh-power approximation, log law, and Spalding's law of the wall.

Discussion Neither the one-seventh-power law nor the log law are valid real close to the wall, but Spalding's law is valid all the way to the wall. However, on the scale shown in Fig. 1, we cannot see the differences between the log law and the Spalding law very close to the wall.

10-99

Solution We are to generate expressions for δ^* and θ, and compare to Blasius.

Analysis First, we set $U(x) = V$ = constant for a flat plate. We integrate using the definition of δ^*,

$$\delta^* = \int_0^\infty \left(1-\frac{u}{U}\right)dy = \int_0^\delta \left[1-\frac{y}{\delta}\right]dy = \left[y-\frac{y^2}{2\delta}\right]_{y=0}^{y=\delta}$$

We integrate only to $y = \delta$, since beyond that, the integrand is identically zero. After substituting the limits of integration, we obtain δ^* as a function of δ,

$$\delta^* = [\delta - \delta/2] - [0-0] = \delta/2 \quad \rightarrow \quad \boldsymbol{\delta^* = \delta/2}$$

Similarly,

$$\theta = \int_0^\infty \frac{u}{U}\left(1-\frac{u}{U}\right)dy = \int_0^\delta \left[\frac{y}{\delta}-\frac{y^2}{\delta^2}\right]dy = \left[\frac{y^2}{2\delta}-\frac{y^3}{3\delta^2}\right]_{y=0}^{y=\delta}$$

After substituting the limits of integration, we obtain θ as a function of δ,

$$\theta = [\delta/2 - \delta/3] - [0-0] = \delta/6 \quad \rightarrow \quad \boldsymbol{\theta = \delta/6}$$

The ratios are **δ^*/δ = 1/2 = 0.500**, and **θ/δ = 1/6 = 0.167**, to three significant digits. We compare these approximate results to those obtained from the Blasius solution, i.e., δ^*/δ = 1.72/4.91 = 0.350, and θ/δ = 0.664/4.91 = 0.135. Thus, **our approximate velocity profile yields δ^*/δ to about 43% error, and θ/δ to about 23% error**.

Discussion The linear approximation is not very accurate.

10-100

Solution We are to generate an expression for δ/x, and compare to Blasius.

Analysis By definition of local skin friction coefficient $C_{f,x}$,

$$C_{f,x} = \frac{2\tau_w}{\rho U^2} = \frac{2}{\rho U^2}\left(\mu \frac{du}{dy}\right)_{y=0} = \frac{2}{\rho U^2}\left[\mu \frac{U}{\delta}\right]_{y=0} = \frac{2\mu}{\rho U \delta} \qquad (1)$$

For a flat plate, the Kármán integral equation reduces to

$$C_{f,x} = 2\frac{d\theta}{dx} = 2\left(\frac{1}{6}\right)\frac{d\delta}{dx} \qquad (2)$$

where we have used the expression for θ as a function of δ from Problem 10-99. Substitution of Eq. 1 into Eq. 2 gives

$$\frac{d\delta}{dx} = 3C_{f,x} = \frac{6\mu}{\rho U \delta}$$

We separate variables and integrate,

$$\delta d\delta = \frac{6\mu}{\rho U}dx \quad \rightarrow \quad \frac{\delta^2}{2} = \frac{6\mu}{\rho U}x \quad \rightarrow \quad \frac{\delta}{x} = \sqrt{12\frac{\mu}{\rho U x}}$$

or, collecting terms, rounding to three digits, and setting $\rho Ux/\mu = \mathrm{Re}_x$,

$$\boxed{\frac{\delta}{x} = \frac{3.46}{\sqrt{\mathrm{Re}_x}}}$$

Compared to the Blasius result, $\delta/x = 4.91/\sqrt{\mathrm{Re}_x}$, our approximation based on the linear function velocity profile yields less than 30% error.

Discussion The Kármán integral equation is useful for obtaining approximate relations, and is “forgiving” because of the integration. Even so, the linear approximation is not very good. Nevertheless, a 30% error is sometimes reasonable for “back of the envelope” calculations. The sine wave approximation does much better, as in Prob. 10-114.

10-101

Solution We are to compare *H* for laminar vs. turbulent boundary layers, and discuss its significance.

Analysis Shape factor *H* is defined as the ratio of displacement thickness to momentum thickness. Thus,

Shape factor:
$$H = \frac{\delta^*}{\theta} = \frac{\delta^*/x}{\theta/x} \qquad (1)$$

For the laminar boundary layer on a flat plate, Eq. 1 becomes

Laminar:
$$H = \frac{\delta^*/x}{\theta/x} = \frac{1.72/\sqrt{\mathrm{Re}_x}}{0.664/\sqrt{\mathrm{Re}_x}} = \mathbf{2.59}$$

For the turbulent boundary layer, using both columns for comparison, Eq. 1 yields

Table 10-4a:
$$H = \frac{\delta^*/x}{\theta/x} = \frac{0.020/\left(\mathrm{Re}_x\right)^{1/7}}{0.016/\left(\mathrm{Re}_x\right)^{1/7}} = \mathbf{1.25}$$

Table 10-4b:
$$H = \frac{\delta^*/x}{\theta/x} = \frac{0.048/\left(\mathrm{Re}_x\right)^{1/5}}{0.037/\left(\mathrm{Re}_x\right)^{1/5}} = \mathbf{1.30}$$

Thus, **the shape factor for a laminar boundary layer is about twice that of turbulent boundary layer**. This implies that the smaller the value of *H*, the more full is the boundary layer. We may also infer that the smaller the value of *H*, the less likely is the boundary layer to separate. ***H*** **depends on the shape of the velocity profile – hence its name, shape factor**.

Discussion In fact, at the separation point of a laminar boundary layer, $H \approx 3.5$.

10-102

Solution We are to determine which orientation of a rectangular flat plate produces the higher drag.

Assumptions **1** The flow is steady and incompressible. **2** The Reynolds number is high enough for a laminar boundary layer to form on the plate, but not high enough for the boundary layer to become turbulent.

Analysis Reynolds number appears in the denominator of the equation for shear stress along the wall of a laminar boundary layer. Thus, wall shear stress decreases with increasing *x*, the distance down the plate. Hence, the average wall shear stress is higher for the case with the plate oriented with its short dimension aligned with the wind (case (b) of Fig. P10-80). Since the surface area of the plate is the same regardless of orientation, the plate with the higher average value of τ_w has the higher overall drag. **Case (b) has the higher drag**.

Discussion Another way to think about this problem is that since the boundary layer is thinner near the leading edge, the shear stress is higher there, and the front portion of the plate contributes to more of the total drag than does the rear portion of the plate.

10-103

Solution We are to integrate an expression for δ.

Analysis We start with Eq. 5 of Example 10-14,

$$\frac{d\delta}{dx} = \frac{72}{14}0.027\left(\mathrm{Re}_x\right)^{-1/7} = 0.139\left(\frac{Ux}{\nu}\right)^{-1/7}$$

Integration with respect to *x* yields

$$\delta = \frac{7}{6}(0.139)\left(\frac{Ux}{\nu}\right)^{6/7}\frac{\nu}{U} \quad \rightarrow \quad \frac{\delta}{x} = 0.162\left(\frac{Ux}{\nu}\right)^{6/7}\frac{\nu}{Ux}$$

or, collecting terms, rounding to two digits, and setting $Ux/\nu = \mathrm{Re}_x$,

$$\boxed{\frac{\delta}{x} \approx \frac{0.16}{\left(\mathrm{Re}_x\right)^{1/7}}}$$

Discussion This approximate result is based on the $1/7^{\text{th}}$ power law.

10-104

Solution We are to generate an expression for δ/x.

Analysis For a flat plate, the Kármán integral equation reduces to

$$C_{f,x} = 2\frac{d\theta}{dx} = 2(0.097)\frac{d\delta}{dx}$$

where we have used the given expression for θ as a function of δ. Substitution of the given expression for $C_{f,x}$ gives

$$\frac{d\delta}{dx} = \frac{C_{f,x}}{2(0.097)} = \frac{0.059(\mathrm{Re}_x)^{-1/5}}{0.194} = 0.304\left(\frac{Ux}{\nu}\right)^{-1/5}$$

Integration with respect to x yields

$$\delta = \frac{5}{4}(0.304)\left(\frac{Ux}{\nu}\right)^{4/5}\frac{\nu}{U} \quad \rightarrow \quad \frac{\delta}{x} = 0.380\left(\frac{Ux}{\nu}\right)^{4/5}\frac{\nu}{Ux}$$

or, collecting terms, rounding to two digits, and setting $Ux/\nu = \mathrm{Re}_x$,

$$\boxed{\frac{\delta}{x} \approx \frac{0.38}{(\mathrm{Re}_x)^{1/5}}}$$

The result is identical to that of Table 10-4, column (b).

Discussion The Kármán integral equation is useful for obtaining approximate relations like those of Table 10-4.

10-105

Solution We are to calculate the location of transition and turbulence along a flat plate boundary layer.

Assumptions **1** The flow is incompressible and steady in the mean. **2** Free stream disturbances are small. **3** The surface of the plate is very smooth.

Properties The density and viscosity of air at $T = 30^{\circ}$C are 1.164 kg/m^3 and 1.872×10^{-5} kg/m·s respectively.

Analysis Transition begins at the critical Reynolds number, which is approximately 100,000 for "clean" flow along a smooth flat plate. Thus,

Beginning of transition:

$$\mathrm{Re}_{x,\mathrm{critical}} = \frac{\rho V x_{\mathrm{critical}}}{\mu} = 100{,}000 \qquad (1)$$

Solving for x yields

$$x_{\mathrm{critical}} = \frac{100{,}000\mu}{\rho V} = \frac{100{,}000\left(1.872\times10^{-5}\ \mathrm{kg/m\cdot s}\right)}{\left(1.164\ \mathrm{kg/m^3}\right)\left(35.0\ \mathrm{m/s}\right)} = 4.59\times10^{-2}\ \mathrm{m} \qquad (2)$$

That is, transition begins at $\boldsymbol{x \approx}$ **4 to 5 cm**. Fully turbulent flow in the boundary layer occurs at approximately 30 times x_{critical}, at $\mathrm{Re}_{x,\mathrm{transition}} \approx 3 \times 10^6$. So, the boundary layer is expected to be fully turbulent by $x \approx 1.2$ to 1.5 m, or to one digit, $\boldsymbol{x \approx}$ **1 to 2 m**.

Discussion Final results are given to only one significant digit, since the locations of transition and turbulence are only approximations. The actual locations are influenced by many things, such as noise, roughness, vibrations, free stream disturbances, etc.

10-106E

Solution We are to assess whether the boundary layer on the bottom of a canoe is laminar or turbulent.

Assumptions **1** The flow is steady and incompressible. **2** Ridges, dings, and other non-uniformities in the bottom of the canoe are ignored – the bottom is assumed to be a smooth flat plate aligned exactly with the direction of flow. **3** From the frame of reference of the canoe, the water below the boundary layer under the canoe moves at uniform speed V = 3.5 mph.

Properties The kinematic viscosity of water at $T = 50^{o}$F is $\nu = 1.407 \times 10^{-5}$ ft^2/s.

Analysis First, we calculate the Reynolds number at the stern of the canoe,

$$\text{Re}_x = \frac{Vx}{\nu} = \frac{(3.5\text{ mi/h})(20\text{ ft})}{1.407\times10^{-5}\text{ ft}^2/\text{s}}\left(\frac{5280\text{ ft}}{1\text{ mi}}\right)\left(\frac{1\text{ h}}{3600\text{ s}}\right) = 7.30\times10^6 \tag{1}$$

Since Re_x is much greater than $\text{Re}_{x,cr}$ (5×10^5), and is even greater than $\text{Re}_{x,\text{transition}}$ (50×10^5), **the boundary layer is definitely turbulent by the back of the canoe**.

Discussion Since the canoe bottom is not perfectly smooth nor perfectly flat, and since we expect some disturbances in the lake water due to waves, the paddles, swimming fish, etc., transition to turbulence is expected to occur much earlier and more rapidly than illustrated for the ideal case in Fig. 10-81. Hence we are even more confident that this boundary layer is turbulent.

Review Problems

10-107C

Solution

(*a*) ***True***: We do not have to make the 2-D approximation in order to define the velocity potential function – ϕ can be defined for any flow if the vorticity is zero.
(*b*) ***False***: The stream function is definable for any two-dimensional flow field, regardless of the value of vorticity.
(*c*) ***True***: The velocity potential function is valid only for irrotational flow regions where the vorticity is zero.
(d) ***True***: The stream function is defined from the continuity equation, and is valid only for two-dimensional flows. Note that some researchers have defined three-dimensional forms of the stream function, but these are beyond the scope of the present introductory text book.

10-108

Solution We are to compute the viscous term for the given velocity field, and show that the flow is rotational.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis We consider the viscous terms of the θ component of the Navier-Stokes equation,

Viscous terms:

$$\mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_\theta}{\partial r}\right)-\frac{u_\theta}{r^2}+\frac{1}{r^2}\frac{\partial^2 u_\theta}{\partial \theta^2}-\frac{2}{r^2}\frac{\partial u_r}{\partial \theta}+\frac{\partial^2 u_\theta}{\partial z^2}\right)$$

$$=\mu\left(\frac{\omega}{r}-\frac{\omega}{r}+0-0+0\right)=0$$

The viscous terms are zero, implying that **there are no net viscous forces acting on fluid elements**. This does not necessarily mean that the flow is inviscid – it could also mean that the flow is irrotational, since the viscous terms disappear in both inviscid and irrotational flows. We obtain the z component of vorticity in cylindrical coordinates from Chap. 4,

z component of vorticity:

$$\zeta_z=\frac{1}{r}\left(\frac{\partial(ru_\theta)}{\partial r}-\frac{\partial u_r}{\partial \theta}\right)=\frac{1}{r}\left(\frac{\partial(\omega r^2)}{\partial r}-0\right)=\boldsymbol{2\omega}$$

Thus, since the vorticity is non-zero, **this flow field is rotational**. Finally, since the flow is not irrotational, the only other way that the net viscous force can be zero is if the flow is inviscid. We conclude, then, that **this flow field is also inviscid**.

Discussion The vorticity is twice the angular velocity, as discussed in Chap. 4.

10-109

Solution We are to calculate the viscous stress tensor for a given velocity field.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis The viscous stress tensor is given in Chap. 9 as

Viscous stress tensor in cylindrical coordinates:

$$\tau_{ij} = \begin{pmatrix} \tau_{rr} & \tau_{r\theta} & \tau_{rz} \\ \tau_{\theta r} & \tau_{\theta\theta} & \tau_{\theta z} \\ \tau_{zr} & \tau_{z\theta} & \tau_{zz} \end{pmatrix} = \begin{pmatrix} 2\mu\dfrac{\partial u_r}{\partial r} & \mu\left(r\dfrac{\partial}{\partial r}\left(\dfrac{u_\theta}{r}\right)+\dfrac{1}{r}\dfrac{\partial u_r}{\partial\theta}\right) & \mu\left(\dfrac{\partial u_r}{\partial z}+\dfrac{\partial u_z}{\partial r}\right) \\ \mu\left(r\dfrac{\partial}{\partial r}\left(\dfrac{u_\theta}{r}\right)+\dfrac{1}{r}\dfrac{\partial u_r}{\partial\theta}\right) & 2\mu\left(\dfrac{1}{r}\dfrac{\partial u_\theta}{\partial\theta}+\dfrac{u_r}{r}\right) & \mu\left(\dfrac{\partial u_\theta}{\partial z}+\dfrac{1}{r}\dfrac{\partial u_z}{\partial\theta}\right) \\ \mu\left(\dfrac{\partial u_r}{\partial z}+\dfrac{\partial u_z}{\partial r}\right) & \mu\left(\dfrac{\partial u_\theta}{\partial z}+\dfrac{1}{r}\dfrac{\partial u_z}{\partial\theta}\right) & 2\mu\dfrac{\partial u_z}{\partial z} \end{pmatrix} \quad (1)$$

We plug in the velocity field from Problem 10-102 into Eq. 1, and we get

$$\tau_{ij} = \begin{pmatrix} \tau_{rr} & \tau_{r\theta} & \tau_{rz} \\ \tau_{\theta r} & \tau_{\theta\theta} & \tau_{\theta z} \\ \tau_{zr} & \tau_{z\theta} & \tau_{zz} \end{pmatrix} = \begin{pmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{pmatrix} \quad (2)$$

Thus, we conclude that there are no viscous stresses in this flow field (solid body rotation). Thus, **this flow can be considered inviscid**.

Discussion Since the fluid moves as a solid body, no fluid particles move relative to any other fluid particles; hence we expect no viscous stresses.

10-110

Solution We are to compute the viscous term for the velocity field and show that the flow is irrotational.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis First, we write out and simplify the viscous terms of the θ component of the Navier-Stokes equation,

$$\mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_\theta}{\partial r}\right)-\frac{u_\theta}{r^2}+\frac{1}{r^2}\frac{\partial^2 u_\theta}{\partial\theta^2}-\frac{2}{r^2}\frac{\partial u_r}{\partial\theta}+\frac{\partial^2 u_\theta}{\partial z^2}\right)=\mu\left(\frac{\Gamma}{2\pi r^3}-\frac{\Gamma}{2\pi r^3}\ +0\ -0\ +0\right)=0$$

The viscous terms are zero, implying that **there are no net viscous forces acting on fluid elements**. This does not necessarily mean that the flow is inviscid – it could also mean that the flow is irrotational, since the viscous terms disappear in both inviscid and irrotational flows.

We obtain the z component of vorticity in cylindrical coordinates from Chap. 4,

z component of vorticity:
$$\zeta_z = \frac{1}{r}\left(\frac{\partial(ru_\theta)}{\partial r}-\frac{\partial u_r}{\partial\theta}\right)=\frac{1}{r}(0-0)=0$$

Thus, since the vorticity is zero, **this flow field is irrotational**.

Discussion We cannot say for sure whether the flow is inviscid unless we calculate the viscous shear stresses, as in the following problem.

10-111

Solution We are to calculate the viscous stress tensor for a given velocity field.

Assumptions **1** The flow is steady. **2** The flow is two-dimensional in the r-θ plane.

Analysis The viscous stress tensor in cylindrical coordinates is given in Chap. 9 as

$$\tau_{ij} = \begin{pmatrix} \tau_{rr} & \tau_{r\theta} & \tau_{rz} \\ \tau_{\theta r} & \tau_{\theta\theta} & \tau_{\theta z} \\ \tau_{zr} & \tau_{z\theta} & \tau_{zz} \end{pmatrix} = \begin{pmatrix} 2\mu \dfrac{\partial u_r}{\partial r} & \mu\left(r\dfrac{\partial}{\partial r}\left(\dfrac{u_\theta}{r}\right) + \dfrac{1}{r}\dfrac{\partial u_r}{\partial \theta}\right) & \mu\left(\dfrac{\partial u_r}{\partial z} + \dfrac{\partial u_z}{\partial r}\right) \\ \mu\left(r\dfrac{\partial}{\partial r}\left(\dfrac{u_\theta}{r}\right) + \dfrac{1}{r}\dfrac{\partial u_r}{\partial \theta}\right) & 2\mu\left(\dfrac{1}{r}\dfrac{\partial u_\theta}{\partial \theta} + \dfrac{u_r}{r}\right) & \mu\left(\dfrac{\partial u_\theta}{\partial z} + \dfrac{1}{r}\dfrac{\partial u_z}{\partial \theta}\right) \\ \mu\left(\dfrac{\partial u_r}{\partial z} + \dfrac{\partial u_z}{\partial r}\right) & \mu\left(\dfrac{\partial u_\theta}{\partial z} + \dfrac{1}{r}\dfrac{\partial u_z}{\partial \theta}\right) & 2\mu \dfrac{\partial u_z}{\partial z} \end{pmatrix} \qquad (1)$$

We plug in the velocity field from Problem 10-104 into Eq. 1, and we get

$$\tau_{ij} = \begin{pmatrix} \tau_{rr} & \tau_{r\theta} & \tau_{rz} \\ \tau_{\theta r} & \tau_{\theta\theta} & \tau_{\theta z} \\ \tau_{zr} & \tau_{z\theta} & \tau_{zz} \end{pmatrix} = \begin{pmatrix} 0 & \mu\dfrac{\Gamma}{\pi r^2} & 0 \\ \mu\dfrac{\Gamma}{\pi r^2} & 0 & 0 \\ 0 & 0 & 0 \end{pmatrix} \qquad (2)$$

Thus, we conclude that there are indeed some non-zero viscous stresses in this flow field. Hence, **this flow is not inviscid, even though it is irrotational**.

Discussion The fluid particles move relative to each other, generating viscous shear stresses. However, the *net* viscous force on a fluid particle is zero since the flow is irrotational.

10-112

Solution We are to calculate modified pressure P' and sketch profiles of P' at two vertical locations in the pipe.

Assumptions **1** The flow is incompressible. **2** Gravity acts vertically downward. **3** There are no free surface effects in this flow field.

Analysis By definition, modified pressure $P' = P + \rho gz$. So we add hydrostatic pressure component ρgz to the given profile $P = P_{atm}$ to obtain the profile for P'. At any z location in the pipe,

Modified pressure: $P' = P + \rho gz \quad \rightarrow \quad P' = P_{atm} + \rho gz$

We see that P' is uniform at any vertical location (P' does not vary radially), but P' varies with elevation z. At $z = z_1$,

Modified pressure at z_1: $\boxed{P_1' = P_{atm} + \rho g z_1}$ (1)

and at $z = z_2$,

Modified pressure at z_2: $\boxed{P_2' = P_{atm} + \rho g z_2}$ (2)

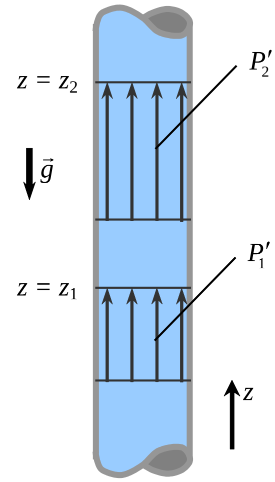


FIGURE 1
Modified pressure profiles at two vertical locations in the pipe.

Comparing Eqs. 1 and 2, the modified pressure is higher at location z_2 since z_2 is higher in elevation than z_1. In this problem there is no forced pressure gradient in terms of actual pressure. However, in terms of *modified* pressure, there is a linearly decreasing modified pressure along the axis of the pipe. In other words, there *is* a forced pressure gradient in terms of modified pressure.

Discussion Since modified pressure eliminates the gravity term from the Navier-Stokes equation, we have replaced the effect of gravity by a gradient of modified pressure P'.

10-113

Solution We are to calculate the required pressure drop between two axial locations of a horizontal pipe that would yield the same volume flow rate as that of the vertical pipe of Problem 10-106.

Assumptions **1** The flow is incompressible. **2** Gravity acts vertically downward. **3** There are no free surface effects in this flow field.

Analysis For the vertical case of Problem 10-106, we know that at $z = z_1$,

Modified pressure at z_1: $P_1' = P_{atm} + \rho g z_1$ (1)

and at $z = z_2$,

Modified pressure at z_2: $P_2' = P_{atm} + \rho g z_2$ (2)

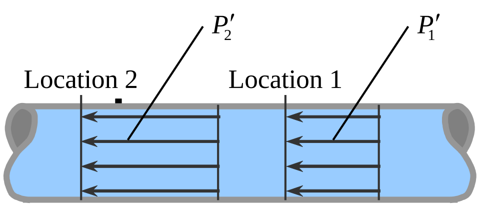


FIGURE 1
Modified pressure profiles at two horizontal locations in the pipe.

Since modified pressure effectively eliminates gravity from the problem, we expect that at the same volume flow rate, *the difference in modified pressure from z_2 to z_1 does not change with changes in the orientation of the pipe*. From Eqs. 1 and 2,

Change in modified pressure from z_2 to z_1: $P_2' - P_1' = \rho g\left(z_2 - z_1\right)$ (3)

The modified pressure profiles at two axial locations in the horizontal pipe are sketched in Fig. 1. We convert the modified pressures in Eq. 3 to actual pressures using the definition of modified pressure, $P = P' - \rho gz$. We note however, that for the horizontal pipe, z does not change along the pipe. Thus we conclude that the required difference in actual pressure is

Change in pressure from location 2 to location 1: $\boxed{P_2 - P_1 = \rho g\left(z_2 - z_1\right)}$ (4)

where z_2 and z_1 are the elevations of the vertical pipe case.

Discussion In order to achieve the same flow rate, the forced pressure gradient in the horizontal case must be the same as the hydrostatic pressure difference supplied by gravity in the vertical case.

10-114

Solution We are to compare the sine wave approximation to the Blasius velocity profile.

Analysis We plot both profiles in Fig. 1. There is not much difference between the two, and thus, **the sine wave profile is a very good approximation of the Blasius profile**.

Discussion The slope of the two profiles at the wall is indistinguishable on the plot (Fig. 1); thus, the sine wave approximation should yield reasonable results for skin friction (shear stress) along the wall as well.

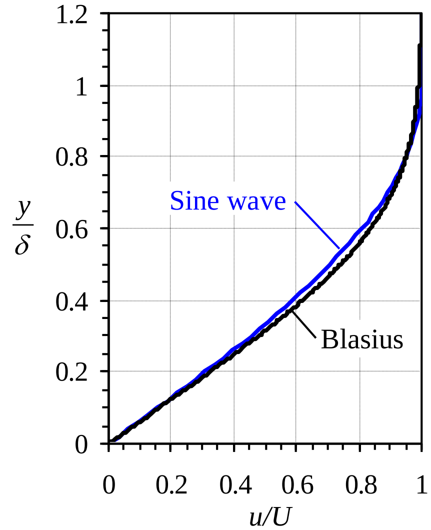


FIGURE 1
Comparison of Blasius and sine wave velocity profiles.

10-115

Solution We are to generate expressions for δ^* and θ, and compare to Blasius.

Analysis First, we set $U(x) = V$ = constant for a flat plate. We integrate using the definition of δ^*,

$$\delta^* = \int_0^\infty \left(1-\frac{u}{U}\right)dy = \int_0^\delta \left[1-\sin\left(\frac{\pi}{2}\frac{y}{\delta}\right)\right]dy = \left[y+\cos\left(\frac{\pi}{2}\frac{y}{\delta}\right)\frac{2\delta}{\pi}\right]_{y=0}^{y=\delta}$$

We integrate only to $y = \delta$, since beyond that, the integrand is identically zero. After substituting the limits of integration, we obtain δ^* as a function of δ,

$$\delta^* = \left[\delta+0\right]-\left[0+\frac{2\delta}{\pi}\right] = \delta-\frac{2}{\pi}\delta \quad \rightarrow \quad \boldsymbol{\delta^* = 0.3634\delta}$$

Similarly,

$$\theta = \int_0^\infty \frac{u}{U}\left(1-\frac{u}{U}\right)dy = \int_0^\delta \left[\sin\left(\frac{\pi}{2}\frac{y}{\delta}\right)-\sin^2\left(\frac{\pi}{2}\frac{y}{\delta}\right)\right]dy$$
$$= \left\{-\cos\left(\frac{\pi}{2}\frac{y}{\delta}\right)\frac{2\delta}{\pi}-\left[\frac{y}{2}-\frac{\delta}{2\pi}\sin\left(\pi\frac{y}{\delta}\right)\right]\right\}_{y=0}^{y=\delta}$$

where we obtained the integral for $\sin^2$ from integration tables. After substituting the limits of integration, we obtain θ as a function of δ,

$$\theta = \left\{0-\left[\frac{\delta}{2}-0\right]\right\}-\left\{-\frac{2\delta}{\pi}-\left[0-0\right]\right\} = -\frac{\delta}{2}+\frac{2\delta}{\pi} \quad \rightarrow \quad \boldsymbol{\theta = 0.1366\delta}$$

The ratios are $\boldsymbol{\delta^*/\delta = 0.363}$, and $\boldsymbol{\theta/\delta = 0.137}$, to three significant digits. We compare these approximate results to those obtained from the Blasius solution, i.e., δ^*/δ = 1.72/4.91 = 0.350, and θ/δ = 0.664/4.91 = 0.135. Thus, **our approximate velocity profile yields $\boldsymbol{\delta^*/\delta}$ to less than 4% error, and $\boldsymbol{\theta/\delta}$ to about 1% error**.

Discussion Integration is "forgiving", and reasonable results can be obtained from integration, even when the velocity profile shape is not exact.

10-116

Solution We are to generate an expression for δ/x, and compare to Blasius.

Analysis By definition of local skin friction coefficient $C_{f,x}$,

$$C_{f,x} = \frac{2\tau_w}{\rho U^2} = \frac{2}{\rho U^2}\left(\mu \frac{du}{dy}\right)_{y=0} = \frac{2}{\rho U^2}\left[\mu U \cos\left(\frac{\pi}{2}\frac{y}{\delta}\right)\frac{\pi}{2\delta}\right]_{y=0} = \frac{2\mu}{\rho U}\left[\frac{\pi}{2\delta}\right] = \frac{\mu\pi}{\rho U\delta} \qquad (1)$$

For a flat plate, the Kármán integral equation reduces to

$$C_{f,x} = 2\frac{d\theta}{dx} = 2(0.1366)\frac{d\delta}{dx} \qquad (2)$$

where we have used the expression for θ as a function of δ from Problem 10-115. Substitution of Eq. 1 into Eq. 2 gives

$$\frac{d\delta}{dx} = \frac{C_{f,x}}{2(0.1366)} = \frac{\mu\pi}{0.2732\rho U\delta}$$

We separate variables and integrate,

$$\delta d\delta = \frac{\mu\pi}{0.2732\rho U}dx \quad \rightarrow \quad \frac{\delta^2}{2} = \frac{\mu\pi}{0.2732\rho U}x \quad \rightarrow \quad \frac{\delta}{x} = \sqrt{\frac{2\pi}{0.2732}\frac{\mu}{\rho Ux}}$$

Collecting terms, rounding to three digits, and setting $\rho Ux/\mu = \mathrm{Re}_x$,

$$\boxed{\frac{\delta}{x} \approx \frac{4.80}{\sqrt{\mathrm{Re}_x}}}$$

Compared to the Blasius result, $\delta/x = 4.91/\sqrt{\mathrm{Re}_x}$, **the approximation yields less than 3% error**.

Discussion The Kármán integral equation is useful for approximations, and is "forgiving" because of the integration.

Fundamentals of Engineering (FE) Exam Problems

10-117

If the fluid velocity is zero in a flow field, the Navier-Stokes equation becomes

(*a*) $\vec{\nabla}P - \rho\vec{g} = 0$ (*b*) $-\vec{\nabla}P + \rho\vec{g} + \mu\vec{\nabla}^2\vec{V} = 0$ (*c*) $\rho\frac{D\vec{V}}{Dt} = -\vec{\nabla}P + \mu\vec{\nabla}^2\vec{V}$

(*d*) $\rho\frac{D\vec{V}}{Dt} = -\vec{\nabla}P + \rho\vec{g} + \mu\vec{\nabla}^2\vec{V}$ (*e*) $\rho\frac{D\vec{V}}{Dt} + \vec{\nabla}P - \rho\vec{g} = 0$

Answer (*a*) $\vec{\nabla}P - \rho\vec{g} = 0$

10-118

Which one is not a scaling parameter used to nondimensionalize the equations of motion?

(*a*) Characteristic length, L (*b*) Characteristic speed, V (*c*) Characteristic viscosity, μ

(*d*) Characteristic frequency, f (*e*) Gravitational acceleration, g

Answer (*c*) Characteristic viscosity, μ

10-119

Which one is not a nondimensional variable defined to nondimensionalize the equations of motion?

(*a*) $t^* = ft$ (*b*) $\vec{x}^* = \frac{\vec{x}}{L}$ (*c*) $\vec{V}^* = \frac{\vec{V}}{V}$ (*d*) $\vec{g}^* = \frac{\vec{g}}{g}$ (*e*) $P^* = \frac{P}{P_0}$

Answer (*e*) $P^* = \frac{P}{P_0}$

10-120

Which dimensionless parameter does not appear in nondimensionalized Navier-Stokes equation?

(*a*) Reynolds number (*b*) Prandtl number (*c*) Strouhal number

(*d*) Euler number (*e*) Froude number

Answer (*b*) Prandtl number

10-121

Which dimensionless parameter is zero in nondimensionalized Navier-Stokes equation when the flow is quasi-steady?

(*a*) Euler number (*b*) Prandtl number (*c*) Froude number
(*d*) Strouhal number (*e*) Reynolds number

Answer (*d*) Strouhal number

10-122

If pressure P is replaced by modified pressure $P' = P + \rho gz$ in nondimensionalized Navier-Stokes equation, which dimensionless parameter drops out?

(*a*) Froude number (*b*) Reynolds number (*c*) Strouhal number
(*d*) Euler number (*e*) Prandtl number

Answer (*a*) Froude number

10-123

In creeping flow, the value of Reynolds number is typically

(*a*) Re < 1 (*b*) Re << 1 (*c*) Re > 1 (*d*) Re >> 1 (*e*) Re = 0

Answer (*b*) Re << 1

10-124

Which one is approximate Navier-Stokes equation in dimensional form for creeping flow?

(*a*) $\vec{\nabla}P - \rho\vec{g} = 0$ (*b*) $-\vec{\nabla}P + \mu\vec{\nabla}^2\vec{V} = 0$ (*c*) $-\vec{\nabla}P + \rho\vec{g} + \mu\vec{\nabla}^2\vec{V} = 0$

(*d*) $\rho\dfrac{D\vec{V}}{Dt} = -\vec{\nabla}P + \rho\vec{g} + \mu\vec{\nabla}^2\vec{V}$ (*e*) $\rho\dfrac{D\vec{V}}{Dt} + \vec{\nabla}P - \rho\vec{g} = 0$

Answer (*b*) $-\vec{\nabla}P + \mu\vec{\nabla}^2\vec{V} = 0$

10-125

For creeping flow over a three-dimensional object, the aerodynamic drag on the object does not depend on

(*a*) Velocity, V (*b*) Fluid viscosity, μ (*c*) Characteristic length, L

(*d*) Fluid density, ρ (*e*) None of these

Answer (*d*) Fluid density, ρ

10-126

Consider a spherical ash particle of diameter 65 μm, falling from a volcano at a high elevation in air whose temperature is –50°C and whose pressure is 55 kPa. The density of air is 0.8588 kg/m^3 and its viscosity is 1.474×10^{-5} kg/m·s. The density of the particle is 1240 kg/m^3. The drag force on a sphere in creeping flow is given by $F_D = 3\pi\mu VD$. The terminal velocity of this particle at this altitude is

(*a*) 0.096 m/s (*b*) 0.123 m/s (*c*) 0.194 m/s (*d*) 0.225 m/s (*e*) 0.276 m/s

Answer (*c*) 0.194 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=65E-6 [m]
rho_particle=1240 [kg/m^3]
rho_air=0.8588 [kg/m^3]
mu=1.474E-5 [kg/m-s]
g=9.81 [m/s^2]
W=pi*D^3/6*rho_particle*g
F_down=W
F_D=3*pi*mu*V*D
F_buoyancy=pi*D^3/6*rho_air*g
F_up=F_D+F_buoyancy
F_down=F_up
```

10-127

Which one is not correct regarding inviscid regions of flow?

(*a*) Inertial forces are not negligible.

(*b*) Pressure forces are not negligible.

(*c*) Reynolds number is large.

(*d*) Not valid in boundary layers and wakes.

(*e*) Solid body rotation of a fluid is an example.

Answer (*e*) Solid body rotation of a fluid is an example.

10-128

For which regions of flow is the Laplace equation $\vec{\nabla}^2\phi = 0$ applicable?

(*a*) Irrotational (*b*) Inviscid (*c*) Boundary layer (*d*) Wake (*e*) Creeping

Answer (*a*) Irrotational

10-129

A very thin region of flow near a solid wall where viscous forces and rotationality cannot be ignored is called

(*a*) Inviscid regions of flow (*b*) Irrotational flow (*c*) Boundary layer

(*d*) Outer flow region (*e*) Creeping flow

Answer (*c*) Boundary layer

10-130

Which one is not a flow region where the boundary layer approximation may be appropriate?

(*a*) Jet (*b*) Inviscid (*c*) Wake (*d*) Mixing layer

(*e*) Thin region near a solid wall

Answer (*b*) Inviscid

10-131

Which one is not correct regarding the boundary layer approximation?

(*a*) The higher the Reynolds number, the thinner the boundary layer.

(*b*) The boundary layer approximation may be appropriate for free shear layers.

(*c*) The boundary layer equations are approximations of the Navier-Stokes equation.

(*d*) The curve representing boundary layer thickness δ as a function of *x* is a streamline.

(*e*) The boundary layer approximation bridges the gap between the Euler equation and the Navier-Stokes equation.

Answer (*d*) The curve representing boundary layer thickness δ as a function of *x* is a streamline.

10-132

For a laminar boundary layer growing on a flat plate, the boundary layer thickness δ is not a function of

(*a*) Velocity, V (*b*) Distance from the leading edge, x (*c*) Fluid density, ρ

(*d*) Fluid viscosity, μ (*e*) Gravitational acceleration, g

Answer (*e*) Gravitational acceleration, g

10-133

For flow along a flat plate with x being the distance from the leading edge, the boundary layer thickness grows like

(*a*) x (*b*) $\sqrt{x}$ (*c*) x^2 (*d*) $1/x$ (*e*) $1/x^2$

Answer (*b*) $\sqrt{x}$

10-134

Air flows at 25°C with a velocity of 3 m/s in a wind tunnel whose test section is 25-cm long. The displacement thickness at the end of the test section is (The kinematic viscosity of air is 1.562×10^{-5} m^2/s.)

(*a*) 0.955 mm (*b*) 1.18 mm (*c*) 1.33 mm (*d*) 1.70 mm (*e*) 1.96 mm

Answer (*e*) 1.96 mm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=25 [C]
V=3 [m/s]
x=0.25 [m]
nu=1.562E-5 [m^2/s]
Re_x=V*x/nu
delta_star=1.72*x/sqrt(Re_x)
```

10-135

Air flows at 25°C with a velocity of 6 m/s over a flat plate whose length is 40 cm. The momentum thickness at the center of the plate is (The kinematic viscosity of air is 1.562×10^{-5} m^2/s.)

(*a*) 0.479 mm (*b*) 0.678 mm (*c*) 0.832 mm (*d*) 1.08 mm (*e*) 1.34 mm

Answer (*a*) 0.479 mm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=25 [C]
V=6 [m/s]
x=0.2 [m]
nu=1.562E-5 [m^2/s]
Re_x=V*x/nu
theta=0.664*x/sqrt(Re_x)
```

10-136

Water flows at 20°C with a velocity of 1.1 m/s over a flat plate whose length is 15 cm. The boundary layer thickness at the end of the plate is (The density and viscosity of water are 998 kg/m^3 and 1.002×10^{-3} kg/m·s, respectively.)

(*a*) 1.14 mm (*b*) 1.35 mm (*c*) 1.56 mm (*d*) 1.82 mm (*e*) 2.09 mm

Answer (*d*) 1.82 mm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=20 [C]
V=1.1[m/s]
x=0.15 [m]
rho=998 [kg/m^3]
mu=1.002E-3 [kg/m-s]
nu=mu/rho
Re_x=V*x/nu
delta=4.91*x/sqrt(Re_x)
```

10-137

Air flows at 15°C with a velocity of 12 m/s over a flat plate whose length is 80 cm. Using one-seventh power law of the turbulent flow, what is the boundary layer thickness at the end of the plate? (The kinematic viscosity of air is 1.470×10^{-5} m^2/s.)

(*a*) 1.54 cm (*b*) 1.89 cm (*c*) 2.16 cm (*d*) 2.45 cm (*e*) 2.82 cm

Answer (*b*) 1.89 cm

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=15 [C]
V=12 [m/s]
x=0.8 [m]
nu=1.470E-5 [m^2/s]
Re_x=V*x/nu
delta=0.16*x/Re_x^(1/7)
```

10-138

Air at 15°C flows at 10 m/s over a flat plate of length 2 m. Using one-seventh power law of the turbulent flow, what is the ratio of local skin friction coefficient for the turbulent and laminar flow cases? (The kinematic viscosity of air is 1.470×10^{-5} m^2/s.)

(*a*) 1.25 (*b*) 3.72 (*c*) 6.31 (*d*) 8.64 (*e*) 12.0

Answer (*c*) 6.31

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=15 [C]
V=10 [m/s]
x=2 [m]
nu=1.470E-5 [m^2/s]
Re_x=V*x/nu
C_f_x_laminar=0.664/sqrt(Re_x)
C_f_x_turbulent=0.027/Re_x^(1/7)
Ratio=C_f_x_turbulent/C_f_x_laminar
```

Design and Essay Problem

10-139

Solution We are to discuss the velocity overshoot in Fig. 10-136.

Analysis **The velocity overshoot is a direct result of the displacement effect and the effect of inertia**. At very *low* values of Re_L (less than about 10^1), where the displacement effect is most prominent, the velocity overshoot is almost non-existent. This can be explained by the lack of inertia at these low Reynolds numbers. Without inertia, there is no mechanism to accelerate the flow around the plate; rather, viscosity *retards* the flow everywhere in the vicinity of the plate, and the influence of the plate extends tens of plate lengths beyond the plate in all directions. At *moderate* values of Reynolds number (Re_L between about 10^1 and 10^4), the displacement effect is significant, and inertial terms are no longer negligible. Hence, fluid is able to accelerate around the plate and the velocity overshoot is significant. At very *high* values of Reynolds number ($\mathrm{Re}_L > 10^4$), inertial terms dominate viscous terms, and the boundary layer is so thin that the displacement effect is almost negligible – the small displacement effect leads to very small velocity overshoot at high Reynolds numbers.

Discussion We can imagine that the flat plate appears thicker from the point of view of the outer flow, and therefore, the flow must accelerate around this "fat" plate.

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

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Yunus A. Çengel & John M. Cimbala

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Chapter 11
EXTERNAL FLOW: DRAG AND LIFT

Drag, Lift, and Drag Coefficients

11-1C

Solution We are to compare the speed of two bicyclists.

Analysis **The bicyclist who leans down and brings his body closer to his knees goes faster** since the frontal area and thus the drag force is less in that position. The drag coefficient also goes down somewhat, but this is a secondary effect.

Discussion This is easily experienced when riding a bicycle down a long hill.

11-2C

Solution We are to discuss how the local skin friction coefficient changes with position along a flat plate in laminar flow.

Analysis The local friction coefficient **decreases with downstream distance** in laminar flow over a flat plate.

Discussion At the front of the plate, the boundary layer is very thin, and thus the shear stress at the wall is large. As the boundary layer grows downstream, however, the boundary layer grows in size, decreasing the wall shear stress.

11-3C

Solution We are to define the frontal area of a body and discuss its applications.

Analysis The *frontal area* of a body is the **area seen by a person when looking from upstream** (the area projected on a plane normal to the direction of flow of the body). The frontal area is appropriate to use in drag and lift calculations for blunt bodies such as cars, cylinders, and spheres.

Discussion The drag force on a body is proportional to both the drag coefficient *and* the frontal area. Thus, one is able to reduce drag by reducing the drag coefficient or the frontal area (or both).

11-4C

Solution We are to define the planform area of a body and discuss its applications.

Analysis The *planform area* of a body is the **area that would be seen by a person looking at the body from above in a direction normal to flow**. The planform area is the area projected on a plane parallel to the direction of flow and normal to the lift force. The planform area is appropriate to use in drag and lift calculations for **slender bodies such as flat plate and airfoils when the frontal area is very small**.

Discussion Consider for example an extremely thin flat plate aligned with the flow. The frontal area is nearly zero, and is therefore not appropriate to use for calculation of drag or lift coefficient.

11-5C

Solution We are to explain when a flow is 2-D, 3-D, and axisymmetric.

Analysis The flow over a body is said to be *two-dimensional* when **the body is very long and of constant cross-section, and the flow is normal to the body** (such as the wind blowing over a long pipe perpendicular to its axis). There is no significant flow along the axis of the body. The flow along a body that **possesses symmetry along an axis in the flow direction** is said to be *axisymmetric* (such as a bullet piercing through air). Flow over a body that **cannot be modeled as two-dimensional or axisymmetric** is *three-dimensional*. The flow over a car is three-dimensional.

Discussion As you might expect, 3-D flows are much more difficult to analyze than 2-D or axisymmetric flows.

11-6C

Solution We are to discuss the difference between upstream and free-stream velocity.

Analysis The **velocity of the fluid relative to the immersed solid body sufficiently far away from a body** is called the *free-stream velocity*, *V*. The *upstream* (or *approach*) *velocity V* is the **velocity of the approaching fluid far ahead of the body**. These two velocities are equal if the flow is uniform and the body is small relative to the scale of the free-stream flow.

Discussion This is a subtle difference, and the two terms are often used interchangeably.

11-7C

Solution We are to discuss the difference between streamlined and blunt bodies.

Analysis A body is said to be *streamlined* if a conscious effort is made to **align its shape with the anticipated streamlines in the flow**. Otherwise, a body tends to **block the flow**, and is said to be *blunt*. A tennis ball is a blunt body (unless the velocity is very low and we have "creeping flow").

Discussion In creeping flow, the streamlines align themselves with the shape of any body – this is a much different regime than our normal experiences with flows in air and water. A low-drag body shape in creeping flow looks much different than a low-drag shape in high Reynolds number flow.

11-8C

Solution We are to discuss applications in which a large drag is desired.

Analysis Some applications in which a large drag is desirable: **parachuting**, **sailing**, and the **transport of pollens**.

Discussion When sailing efficiently, however, the *lift* force on the sail is more important than the drag force in propelling the boat.

11-9C

Solution We are to define drag and discuss why we usually try to minimize it.

Analysis The **force a flowing fluid exerts on a body in the flow direction** is called *drag*. **Drag is caused by friction between the fluid and the solid surface, and the pressure difference between the front and back of the body**. We try to minimize drag in order to **reduce fuel consumption** in vehicles, **improve safety and durability** of structures subjected to high winds, and to **reduce noise and vibration**.

Discussion In some applications, such as parachuting, high drag rather than low drag is desired.

11-10C

Solution We are to define lift, and discuss its cause and the contribution of wall shear to lift.

Analysis The **force a flowing fluid exerts on a body in the normal direction to flow that tends to move the body in that direction** is called *lift*. It is caused by the **components of the pressure and wall shear forces in the direction normal to the flow**. The wall shear contributes to lift (unless the body is very slim), but **its contribution is usually small**.

Discussion Typically the nonsymmetrical *shape* of the body is what causes the lift force to be produced.

11-11C

Solution We are to explain how to calculate the drag coefficient, and discuss the appropriate area.

Analysis When the drag force F_D, the upstream velocity V, and the fluid density ρ are measured during flow over a body, the drag coefficient is determined from

$$\boxed{C_D = \frac{F_D}{\frac{1}{2}\rho V^2 A}}$$

where ***A*** **is ordinarily the** ***frontal area*** (the area projected on a plane normal to the direction of flow) of the body.

Discussion In some cases, however, such as flat plates aligned with the flow or airplane wings, the planform area is used instead of the frontal area. Planform area is the area projected on a plane parallel to the direction of flow and normal to the lift force.

11-12C

Solution We are to explain how to calculate the lift coefficient, and discuss the appropriate area.

Analysis When the lift force F_L, the upstream velocity V, and the fluid density ρ are measured during flow over a body, the lift coefficient can be determined from

$$\boxed{C_L = \frac{F_L}{\frac{1}{2}\rho V^2 A}}$$

where ***A*** **is ordinarily the** ***planform area***, which is the area that would be seen by a person looking at the body from above in a direction normal to the body.

Discussion In some cases, however, such as flat plates aligned with the flow or airplane wings, the planform area is used instead of the frontal area. Planform area is the area projected on a plane parallel to the direction of flow and normal to the lift force.

11-13C

Solution We are to define and discuss terminal velocity.

Analysis The **maximum velocity a free falling body can attain** is called the *terminal velocity*. **It is determined by setting the weight of the body equal to the drag and buoyancy forces**, $W = F_D + F_B$.

Discussion When discussing the settling of small dust particles, terminal velocity is also called *terminal settling speed* or *settling velocity*.

11-14C

Solution We are to discuss the difference between skin friction drag and pressure drag, and which is more significant for slender bodies.

Analysis The **part of drag that is due directly to wall shear stress** τ_w is called the *skin friction drag* $F_{D,\text{ friction}}$ since it is caused by frictional effects, and **the part that is due directly to pressure** ***P*** **and depends strongly on the shape of the body** is called the *pressure drag* $F_{D,\text{ pressure}}$. For slender bodies such as airfoils, **the friction drag is usually more significant**.

Discussion For blunt bodies, on the other hand, pressure drag is usually more significant than skin friction drag.

11-15C

Solution We are to discuss the effect of surface roughness on drag coefficient.

Analysis The friction drag coefficient is **independent of surface roughness** in *laminar flow*, but is a **strong function of surface roughness** in *turbulent flow* due to surface roughness elements protruding farther into the viscous sublayer.

Discussion If the roughness is very large, however, the drag on bodies is increased even for laminar flow, due to pressure effects on the roughness elements.

11-16C

Solution We are to discuss the effect of streamlining, and its effect on friction drag and pressure drag.

Analysis As a result of streamlining, (*a*) **friction drag increases**, (*b*) **pressure drag decreases**, and (*c*) **total drag decreases** at high Reynolds numbers (the general case), but increases at very low Reynolds numbers (creeping flow) since the friction drag dominates at low Reynolds numbers.

Discussion Streamlining can significantly reduce the overall drag on a body at high Reynolds number.

11-17C

Solution We are to define and discuss flow separation.

Analysis At sufficiently high velocities, **the fluid stream detaches itself from the surface of the body**. This is called *separation*. It is caused by a fluid flowing over a curved surface at a high velocity (or technically, by adverse pressure gradient). Separation increases the drag coefficient drastically.

Discussion A boundary layer has a hard time resisting an adverse pressure gradient, and is likely to separate. A turbulent boundary layer is in general more resilient to flow separation than a laminar flow.

11-18C

Solution We are to define and discuss drafting.

Analysis *Drafting* is when **a moving body follows another moving body by staying close behind in order to reduce drag**. It reduces the pressure drag and thus the drag coefficient for the drafted body by **taking advantage of the low pressure wake region of the moving body in front**.

Discussion We often see drafting in automobile and bicycle racing.

11-19C

Solution We are to discuss how drag coefficient varies with Reynolds number.

Analysis (*a*) In general, the drag coefficient **decreases with Reynolds number at low and moderate Reynolds numbers**. (*b*) The drag coefficient is **nearly independent of Reynolds number at high Reynolds numbers** (Re > 10^4).

Discussion When the drag coefficient is independent of Re at high values of Re, we call this Reynolds number independence.

11-20C

Solution We are to discuss the effect of adding a fairing to a circular cylinder.

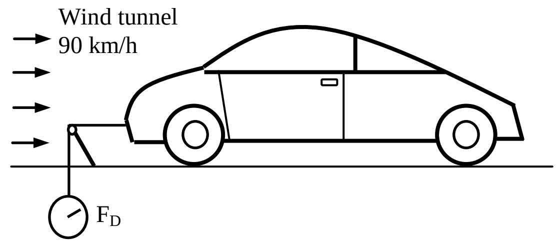
Analysis As a result of attaching fairings to the front and back of a cylindrical body at high Reynolds numbers, (*a*) **friction drag increases**, (*b*) **pressure drag decreases**, and (*c*) **total drag decreases**.

Discussion In creeping flow (very low Reynolds numbers), however, adding a fairing like this would actually *increase* the overall drag, since the surface area and therefore the skin friction drag would increase significantly.

11-21

Solution The drag force acting on a car is measured in a wind tunnel. The drag coefficient of the car at the test conditions is to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** The cross-section of the tunnel is large enough to simulate free flow over the car. **3** The bottom of the tunnel is also moving at the speed of air to approximate actual driving conditions or this effect is negligible. **4** Air is an ideal gas.



Properties The density of air at 1 atm and 25°C is ρ = 1.164 kg/m^3.

Analysis The drag force acting on a body and the drag coefficient are given by

$$F_D = C_D A \frac{\rho V^2}{2} \quad \text{and} \quad C_D = \frac{2F_D}{\rho A V^2}$$

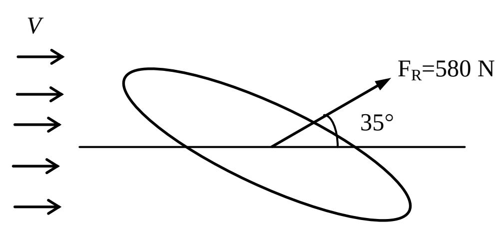
where A is the frontal area. Substituting and noting that 1 m/s = 3.6 km/h, the drag coefficient of the car is determined to be

$$C_D = \frac{2\times(220\text{ N})}{(1.164\text{ kg/m}^3)(1.25\times1.65\text{ m}^2)(90/3.6\text{ m/s})^2}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{0.29}$$

Discussion Note that the drag coefficient depends on the design conditions, and its value will be different at different conditions. Therefore, the published drag coefficients of different vehicles can be compared meaningfully only if they are determined under identical conditions. This shows the importance of developing standard testing procedures in industry.

11-22

Solution The resultant of the pressure and wall shear forces acting on a body is given. The drag and the lift forces acting on the body are to be determined.



Analysis The drag and lift forces are determined by decomposing the resultant force into its components in the flow direction and the normal direction to flow,

Drag force: $F_D = F_R \cos\theta = (580\text{ N})\cos 35° = \mathbf{475 N}$

Lift force: $F_L = F_R \sin\theta = (580\text{ N})\sin 35° = \mathbf{333 N}$

Discussion Note that the greater the angle between the resultant force and the flow direction, the greater the lift.

11-23

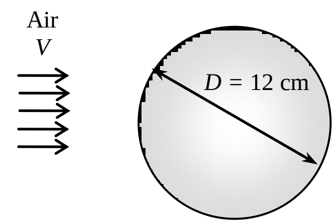
Solution The total drag force acting on a spherical body is measured, and the pressure drag acting on the body is calculated by integrating the pressure distribution. The friction drag coefficient is to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** The surface of the sphere is smooth. **3** The flow over the sphere is turbulent (to be verified).

Properties The density and kinematic viscosity of air at 1 atm and 5°C are ρ = 1.269 kg/m^3 and ν = 1.382×10^{-5} m^2/s. The drag coefficient of sphere in turbulent flow is C_D = 0.2, and its frontal area is $A = \pi D^2/4$ (Table 11-2).

Analysis The total drag force is the sum of the friction and pressure drag forces. Therefore,

$$F_{D,\text{friction}} = F_D - F_{D,\text{pressure}} = 5.2 - 4.9 = 0.3\ \text{N}$$



where $F_D = C_D A \dfrac{\rho V^2}{2}$ and $F_{D,\text{friction}} = C_{D,\text{friction}} A \dfrac{\rho V^2}{2}$

Taking the ratio of the two relations above gives

$$C_{D,\text{friction}} = \frac{F_{D,\text{friction}}}{F_D} C_D = \frac{0.3\ \text{N}}{5.2\ \text{N}}(0.2) = \mathbf{0.0115}$$

Now we need to verify that the flow is turbulent. This is done by calculating the flow velocity from the drag force relation, and then the Reynolds number:

$$F_D = C_D A \frac{\rho V^2}{2} \rightarrow V = \sqrt{\frac{2F_D}{\rho C_D A}} = \sqrt{\frac{2(5.2\ \text{N})}{(1.269\ \text{kg/m}^3)(0.2)[\pi(0.12\ \text{m})^2/4]}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)} = 60.2\ \text{m/s}$$

$$\text{Re} = \frac{VD}{\nu} = \frac{(60.2\ \text{m/s})(0.12\ \text{m})}{1.382\times10^{-5}\ \text{m}^2/s} = 5.23\times10^5$$

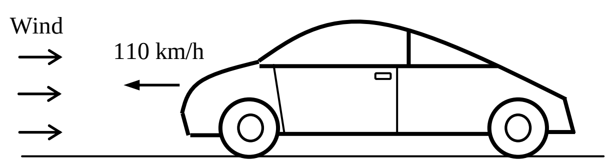
which is greater than 2×10^5. Therefore, the flow is turbulent as assumed.

Discussion Note that knowing the flow regime is important in the solution of this problem since the total drag coefficient for a sphere is 0.5 in laminar flow and 0.2 in turbulent flow.

11-24

Solution A car is moving at a constant velocity. The upstream velocity to be used in fluid flow analysis is to be determined for the cases of calm air, wind blowing against the direction of motion of the car, and wind blowing in the same direction of motion of the car.

Analysis In fluid flow analysis, the velocity used is the relative velocity between the fluid and the solid body. Therefore:



(*a*) Calm air: $V = V_{car}$ = **110 km/h**

(*b*) Wind blowing against the direction of motion:
$V = V_{car} + V_{wind}$ = 110 + 30 = **140 km/h**

(*c*) Wind blowing in the same direction of motion:
$V = V_{car} - V_{wind}$ = 110 - 30 = **80 km/h**

Discussion Note that the wind and car velocities are added when they are in opposite directions, and subtracted when they are in the same direction.

11-25E

Solution The frontal area of a car is reduced by redesigning. The amount of fuel and money saved per year as a result are to be determined.

Assumptions **1** The car is driven 12,000 miles a year at an average speed of 55 km/h. **2** The effect of reduction of the frontal area on the drag coefficient is negligible.

Properties The densities of air and gasoline are given to be 0.075 lbm/ft^3 and 50 lbm/ft^3, respectively. The heating value of gasoline is given to be 20,000 Btu/lbm. The drag coefficient is C_D = 0.3 for a passenger car (Table 11-2).

Analysis The drag force acting on a body is determined from

$$F_D = C_D A \frac{\rho V^2}{2}$$

where A is the frontal area of the body. The drag force acting on the car before redesigning is

$$F_D = 0.3(18\ \text{ft}^2)\frac{(0.075\ \text{lbm/ft}^3)(55\ \text{mph})^2}{2}\left(\frac{1.4667\ \text{ft/s}}{1\ \text{mph}}\right)^2\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = 40.9\ \text{lbf}$$

Noting that work is force times distance, the amount of work done to overcome this drag force and the required energy input for a distance of 12,000 miles are

$$W_{\text{drag}} = F_D L = (40.9\ \text{lbf})(12{,}000\ \text{miles/year})\left(\frac{5280\ \text{ft}}{1\ \text{mile}}\right)\left(\frac{1\ \text{Btu}}{778.169\ \text{lbf}\cdot\text{ft}}\right) = 3.330\times10^6\ \text{Btu/year}$$

$$E_{in} = \frac{W_{\text{drag}}}{\eta_{\text{car}}} = \frac{3.330\times10^6\ \text{Btu/year}}{0.30} = 1.110\times10^7\ \text{Btu/year}$$

Then the amount and costs of the fuel that supplies this much energy are

$$\text{Amont of fuel} = \frac{m_{\text{fuel}}}{\rho_{\text{fuel}}} = \frac{E_{\text{in}}/\text{HV}}{\rho_{\text{fuel}}} = \frac{(1.110\times10^7\ \text{Btu/year})/(20{,}000\ \text{Btu/lbm})}{50\ \text{lbm/ft}^3} = 11.10\ \text{ft}^3/\text{year}$$

$$\text{Cost} = (\text{Amount of fuel})(\text{Unit cost}) = (11.10\ \text{ft}^3/\text{year})(\$3.10/\text{gal})\left(\frac{7.4804\ \text{gal}}{1\ \text{ft}^3}\right) = \$257.4/\text{year}$$

That is, the car uses 11.10 ft^3 = 83.03 gallons of gasoline at a cost of \$257.4 per year to overcome the drag.

The drag force and the work done to overcome it are directly proportional to the frontal area. Then the percent reduction in the fuel consumption due to reducing frontal area is equal to the percent reduction in the frontal area:

$$\text{Reduction ratio} = \frac{A - A_{\text{new}}}{A} = \frac{18-15}{18} = 0.1667$$

$$\text{Amount reduction} = (\text{Reduction ratio})(\text{Amount})$$

$$= 0.1667(83.03\ \text{gal/year}) = \mathbf{13.8\ gal/year}$$

$$\text{Cost reduction} = (\text{Reduction ratio})(\text{Cost}) = 0.1667(\$257.4/\text{year}) = \mathbf{\$42.9/year}$$

Therefore, reducing the frontal area reduces the fuel consumption due to drag by 16.7%.

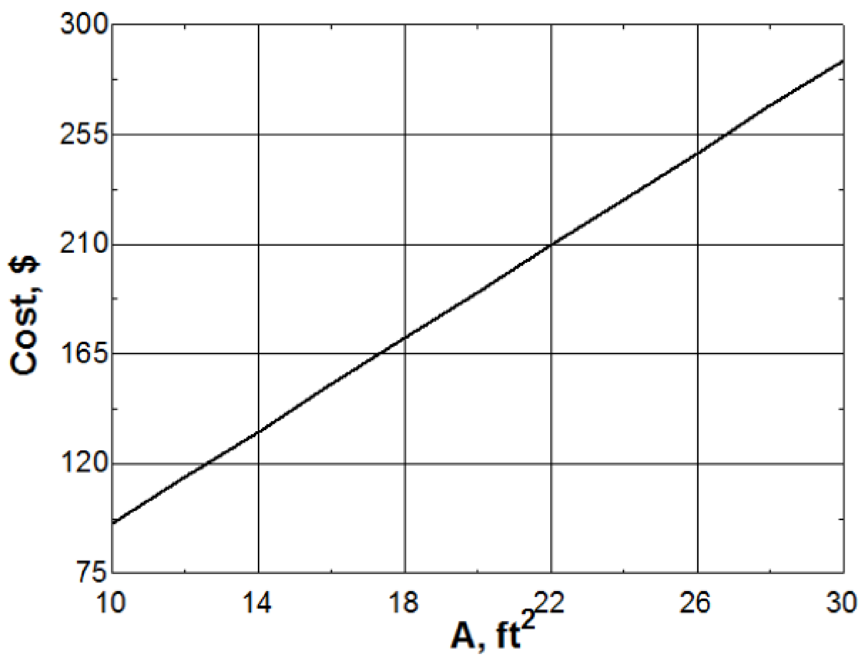
Discussion Note from this example that significant reductions in drag and fuel consumption can be achieved by reducing the frontal area of a vehicle.

11-26E

Solution The previous problem is reconsidered. The effect of frontal area on the annual fuel consumption of the car as the frontal area varied from 10 to 30 ft^2 in increments of 2 ft^2 are to be investigated.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
CD=0.3
rho=0.075 "lbm/ft3"
V=55*1.4667 "ft/s"
Eff=0.32
Price=2.20 "$/gal"
efuel=20000 "Btu/lbm"
rho_gas=50 "lbm/ft3"
L=12000*5280 "ft"
FD=CD*A*(rho*V^2)/2/32.2 "lbf"
Wdrag=FD*L/778.169 "Btu"
Ein=Wdrag/Eff
m=Ein/efuel "lbm"
Vol=(m/rho_gas)*7.4804 "gal"
Cost=Vol*Price
```



A, ft^2	F_{drag} lbf	Amount, gal	Cost, $
10	22.74	43.27	95.2
12	27.28	51.93	114.2
14	31.83	60.58	133.3
16	36.38	69.24	152.3
18	40.92	77.89	171.4
20	45.47	86.55	190.4
22	50.02	95.2	209.4
24	54.57	103.9	228.5
26	59.11	112.5	247.5
28	63.66	121.2	266.6
30	68.21	129.8	285.6

Discussion As you might expect, the cost goes up linearly with area, since drag force goes up linearly with area.

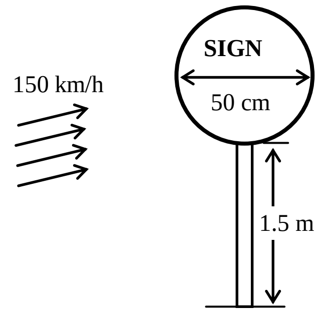
11-27

Solution A circular sign is subjected to high winds. The drag force acting on the sign and the bending moment at the bottom of its pole are to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** The drag force on the pole is negligible. **3** The flow is turbulent so that the tabulated value of the drag coefficient can be used.

Properties The drag coefficient for a thin circular disk is C_D = 1.1 (Table 11-2). The density of air at 100 kPa and 10°C = 283 K is

$$\rho = \frac{P}{RT} = \frac{100\ \text{kPa}}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg.K})(283\ \text{K})} = 1.231\ \text{kg/m}^3$$



Analysis The frontal area of a circular plate subjected to normal flow is $A = \pi D^2/4$. Then the drag force acting on the sign is

$$F_D = C_D A \frac{\rho V^2}{2}$$

$$= (1.1)[\pi(0.5\ \text{m})^2/4]\frac{(1.231\ \text{kg/m}^3)(150/3.6\ \text{m/s})^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{231\ N}$$

Noting that the resultant force passes through the center of the stop sign, the bending moment at the bottom of the pole becomes

$$M_{bottom} = F_D \times L = (231\ \text{N})(1.5 + 0.25)\ \text{m} = \mathbf{404\ Nm}$$

Discussion Note that the drag force is equivalent to the weight of over 23 kg of mass. Therefore, the pole must be strong enough to withstand the weight of 23 kg hanged at one of its end when it is held from the other end horizontally.

11-28E

Solution We are to estimate how much money is wasted by driving with a pizza sign on a car roof.

Properties ρ_{fuel} = 50.2 lbm/ft^3, HV_{fuel} = 1.53 × 10^7 ft·lbf/lbm, ρ_{air} = 0.07518 lbm/ft^3, μ_{air} = 1.227 × 10^{-5} lbm/ft·s

Analysis First some conversions: V = 45 mph = 66.0 ft/s and the total distance traveled in one year = L = 10,000 miles = 5.280 × 10^7 ft. The additional drag force due to the sign is

$$F_D = \tfrac{1}{2}\rho_{air}V^2 C_D A$$

where A is the frontal area. The work required to overcome this additional drag is force times distance. So, letting L be the total distance driven in a year,

$$\text{Work}_{drag} = F_D L = \tfrac{1}{2}\rho_{air}V^2 C_D A L$$

The energy required to perform this work is much greater than this due to overall efficiency of the car engine, transmission, etc. Thus,

$$E_{required} = \frac{\text{Work}_{drag}}{\eta_{overall}} = \frac{\tfrac{1}{2}\rho_{air}V^2 C_D A L}{\eta_{overall}}$$

But the required energy is also equal to the heating value of the fuel HV times the mass of fuel required. In terms of required fuel *volume*, volume = mass/density. Thus,

$$\boxed{V_{\text{fuel required}} = \frac{m_{\text{fuel required}}}{\rho_{fuel}} = \frac{E_{required}/\text{HV}}{\rho_{fuel}} = \frac{\tfrac{1}{2}\rho_{air}V^2 C_D A L}{\rho_{fuel}\eta_{overall}\text{HV}}}$$

The above is our answer in variable form. Finally, we plug in the given values and properties to obtain the numerical answer,

$$\boxed{V_{\text{fuel required}} = \frac{\tfrac{1}{2}\left(0.07518\text{ lbm/ft}^3\right)\left(66.0\text{ ft/s}\right)^2\left(0.94\right)\left(0.612\text{ ft}^2\right)\left(5.280\times10^7\text{ ft}\right)}{\left(50.2\text{ lbm/ft}^3\right)\left(0.332\right)\left(1.53\times10^7\text{ ft}\cdot\text{lbf/lbm}\right)}\left(\frac{\text{lbf}}{32.174\text{ lbm}\cdot\text{ft/s}^2}\right)}$$

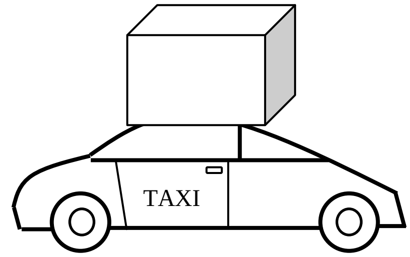
which yields $V_{\text{fuel required}}$ = 0.6062 ft^3, which is equivalent to 4.535 gallons per year. At $3.50 per gallon, the total cost is about $15.87 per year, or rounding to two significant digits, **the total cost is about $16 per year**.

Discussion Any more than 2 significant digits in the final answer cannot be justified. Bill would be wise to lobby for a more aerodynamic pizza sign.

11-29

Solution An advertisement sign in the form of a rectangular block that has the same frontal area from all four sides is mounted on top of a taxicab. The increase in the annual fuel cost due to this sign is to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** The car is driven 60,000 km a year at an average speed of 50 km/h. **3** The overall efficiency of the engine is 28%. **4** The effect of the sign and the taxicab on the drag coefficient of each other is negligible (no interference), and the edge effects of the sign are negligible (a crude approximation). **5** The flow is turbulent so that the tabulated value of the drag coefficient can be used.



Properties The densities of air and gasoline are given to be 1.25 kg/m^3 and 0.72 kg/L, respectively. The heating value of gasoline is given to be 42,000 kJ/kg. The drag coefficient for a square rod for normal flow is C_D = 2.2 (Table 11-1).

Analysis Noting that 1 m/s = 3.6 km/h, the drag force acting on the sign is

$$F_D = C_D A \frac{\rho V^2}{2} = (2.2)(0.9 \times 0.3\,\text{m}^2)\frac{(1.25\,\text{kg/m}^3)(50/3.6\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = 71.61\,\text{N}$$

Noting that work is force times distance, the amount of work done to overcome this drag force and the required energy input for a distance of 60,000 km are

$$W_{\text{drag}} = F_D L = (71.61\,\text{N})(60{,}000\,\text{km/year}) = 4.30\times10^6\ \text{kJ/year}$$

$$E_{in} = \frac{W_{\text{drag}}}{\eta_{\text{car}}} = \frac{4.30\times10^6\ \text{kJ/year}}{0.28} = 1.54\times10^7\ \text{kJ/year}$$

Then the amount and cost of the fuel that supplies this much energy are

$$\text{Amont of fuel} = \frac{m_{\text{fuel}}}{\rho_{\text{fuel}}} = \frac{E_{\text{in}}/\text{HV}}{\rho_{\text{fuel}}} = \frac{(1.54\times10^7\ \text{kJ/year})/(42{,}000\,\text{kJ/kg})}{0.72\,\text{kg/L}} = \mathbf{509\,L/year}$$

$$\text{Cost} = (\text{Amount of fuel})(\text{Unit cost}) = (509\ \text{L/year})(\$1.10/\text{L}) = \mathbf{\$560/year}$$

That is, the taxicab will use 509 L of gasoline at a cost of \$560 per year to overcome the drag generated by the advertisement sign.

Discussion Note that the advertisement sign increases the fuel cost of the taxicab significantly. The taxicab operator may end up losing money by installing the sign if he/she is not aware of the major increase in the fuel cost, and negotiate accordingly.

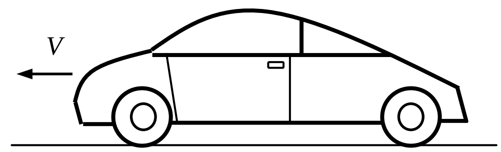
11-30E

Solution A person who normally drives at 55 mph now starts driving at 75 mph. The percentage increase in fuel consumption of the car is to be determined.

Assumptions **1** The fuel consumption is approximately proportional to the drag force on a level road (as stated). **2** The drag coefficient remains the same.

Analysis The drag force is proportional to the square of the velocity, and power is force times velocity. Therefore,

$$F_D = C_D A \frac{\rho V^2}{2} \quad \text{and} \quad \dot{W}_{\text{drag}} = F_D V = C_D A \frac{\rho V^3}{2}$$



Then the ratio of the power used to overcome drag force at V_2= 75 mph to that at V_1 = 55 mph becomes

$$\frac{\dot{W}_{\text{drag2}}}{\dot{W}_{\text{drag1}}} = \frac{V_2^3}{V_1^3} = \frac{75^3}{55^3} = \mathbf{2.54}$$

Therefore, the power to overcome the drag force and thus fuel consumption per unit time more than doubles as a result of increasing the velocity from 55 to 75 mph.

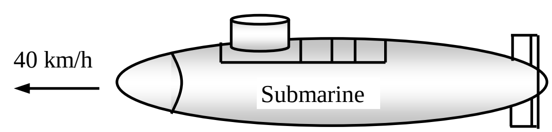
Discussion This increase appears to be large. This is because all the engine power is assumed to be used entirely to overcome drag. Still, the simple analysis above shows the strong dependence of the fuel consumption on the cruising speed of a vehicle. A better measure of fuel consumption is the amount of fuel used per unit distance (rather than per unit time). A car cruising at 55 mph will travel a distance of 55 miles in 1 hour. But a car cruising at 75 mph will travel the same distance at 55/75 = 0.733 h or 73.3% of the time. Therefore, for a given distance, the increase in fuel consumption is 2.54×0.733 = 1.86 – an increase of **86%.** This is large, especially with the high cost of gasoline these days.

11-31

Solution A submarine is treated as an ellipsoid at a specified length and diameter. The powers required for this submarine to cruise horizontally in seawater and to tow it in air are to be determined.

Assumptions **1** The submarine can be treated as an ellipsoid. **2** The flow is turbulent. **3** The drag of the towing rope is negligible. **4** The motion of submarine is steady and horizontal.

Properties The drag coefficient for an ellipsoid with *L/D* = 25/5 = 5 is C_D = 0.1 in turbulent flow (Table 11-2). The density of sea water is given to be 1025 kg/m^3. The density of air is given to be 1.30 kg/m^3.



Analysis Noting that 1 m/s = 3.6 km/h, the velocity of the submarine is equivalent to V = 40/3.6 = 11.11 m/s. The frontal area of an ellipsoid is $A = \pi D^2/4$. Then the drag force acting on the submarine becomes

In water: $$F_D = C_D A \frac{\rho V^2}{2} = (0.1)[\pi(5\text{ m})^2/4]\frac{(1025\text{ kg/m}^3)(11.11\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 124.2\text{ kN}$$

In air: $$F_D = C_D A \frac{\rho V^2}{2} = (0.1)[\pi(5\text{ m})^2/4]\frac{(1.30\text{ kg/m}^3)(11.11\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 0.1575\text{ kN}$$

Noting that power is force times velocity, the power needed to overcome this drag force is

In water: $$\dot{W}_{\text{drag}} = F_D V = (124.2\text{ kN})(11.11\text{ m/s})\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right) = \mathbf{1380\,kW}$$

In air: $$\dot{W}_{\text{drag}} = F_D V = (0.1575\text{ kN})(11.11\text{ m/s})\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right) = \mathbf{1.75\,kW}$$

Therefore, the power required for this submarine to cruise horizontally in seawater is 1380 kW and the power required to tow this submarine in air at the same velocity is 1.75 kW.

Discussion Note that the power required to move the submarine in water is about 800 times the power required to move it in air. This is due to the higher density of water compared to air (sea water is about 800 times denser than air).

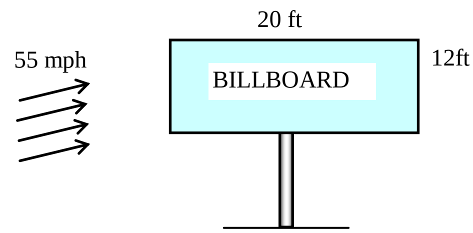
11-32E

Solution A billboard is subjected to high winds. The drag force acting on the billboard is to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** The drag force on the supporting poles are negligible. **3** The flow is turbulent so that the tabulated value of the drag coefficient can be used. **4** The 3-D effects are negligible, and thus the billboard is approximated as a very long flat plate (two-dimensioal).

Properties If the plate were two dimensional, the drag coefficient would be C_D = 1.9 (Table 11-1). However, for this 3-D case, Table 11-2 yields C_D = 1.10 + 0.02(20/8 +8/20) = 1.158, which is a better approximation of the drag coefficient. We calculate the gas constant for air (Table A-1E) as

$$R_{\text{air}} = \frac{R_u}{M} = \frac{10.732\ \text{psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R}}{28.97\ \text{lbm/lbmol}} = 0.37045\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$$



The density of air at 14.3 psia and 40°F = 40 + 459.67 = 499.67 R is thus

$$\rho = \frac{P}{RT} = \frac{14.3\ \text{psia}}{(0.37045\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(499.67\ \text{R})} = 0.077254\ \text{lbm/ft}^3$$

Analysis The drag force acting on the billboard is determined from

$$F_D = C_D A\frac{\rho V^2}{2} = (1.158)(12\times 20\ \text{ft}^2)\frac{(0.077254\ \text{lbm/ft}^3)(55\ \text{mi/h})^2}{2}\left(\frac{1\ \text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1.4667\ \text{ft/s}}{1\ \text{mi/h}}\right)^2$$

$$= 2171\ \text{lbf} \cong \mathbf{2170\ lbf}$$

Discussion Note that the drag force is equivalent to the weight of 2170 lbm of mass. Therefore, the support bars must be strong enough to withstand the weight of 2170 lbm hanging on one of their ends when they are held from the other end horizontally (which causes a fairly large moment or torque on the support bars).

11-33

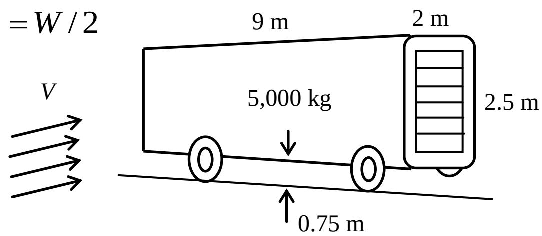
Solution A semi truck is exposed to winds from its side surface. The wind velocity that will tip the truck over to its side is to be determined.

Assumptions **1** The flow of air in the wind is steady and incompressible. **2** The edge effects on the semi truck are negligible (a crude approximation), and the resultant drag force acts through the center of the side surface. **3** The flow is turbulent so that the tabulated value of the drag coefficient can be used. **4** The distance between the wheels on the same axle is also 2 m. **5** The semi truck is loaded uniformly so that its weight acts through its center.

Properties The density of air is given to be $\rho = 1.10\ \text{kg/m}^3$. The drag coefficient for a body of rectangular cross-section corresponding to $L/D = 2/2 = 1$ is $C_D = 2.2$ when the wind is normal to the side surface (Table 11-2).

Analysis When the truck is first tipped, the wheels on the wind-loaded side of the truck will be off the ground, and thus all the reaction forces from the ground will act on wheels on the other side. Taking the moment about an axis passing through these wheels and setting it equal to zero gives the required drag force to be

$$\sum M_{\text{wheels}} = 0 \quad\rightarrow\quad F_D \times (2\ \text{m}) - W \times (1\ \text{m}) = 0 \quad\rightarrow\quad F_D = W/2$$



Substituting, the required drag force is determined to be

$$F_D = mg/2 = (5000\ \text{kg})(9.81\ \text{m/s}^2)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)/2 = 24{,}525\ \text{N}$$

The wind velocity that will cause this drag force is determined to be

$$F_D = C_D A \frac{\rho V^2}{2} \quad\rightarrow\quad 24{,}525\ \text{N} = (2.2)(2.5\times 9\ \text{m}^2)\frac{(1.10\ \text{kg/m}^3)V^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) \quad\rightarrow\quad V = 31.48\ \text{m/s}$$

which is equivalent to a wind velocity of $V = 31.48\times 3.6 =$ **113 km/h.**

Discussion This is very high velocity, and it can be verified easily by calculating the Reynolds number that the flow is turbulent as assumed.

11-34

Solution A bicyclist is riding his bicycle downhill on a road with a specified slope without pedaling or breaking. The terminal velocity of the bicyclist is to be determined for the upright and racing positions.

Assumptions **1** The rolling resistance and bearing friction are negligible. **2** The drag coefficient remains constant. **3** The buoyancy of air is negligible.

Properties The density of air is given to be ρ = 1.25 kg/m^3. The frontal area and the drag coefficient are given to be 0.45 m^2 and 1.1 in the upright position, and 0.4 m^2 and 0.9 on the racing position.

Analysis The terminal velocity of a free falling object is reached when the drag force equals the component of the total weight (bicyclist + bicycle) in the flow direction,

$$F_D = W_{\text{total}} \sin\theta \quad \rightarrow \quad C_D A \frac{\rho V^2}{2} = m_{\text{total}} g \sin\theta$$

Solving for V gives $$V = \sqrt{\frac{2 g m_{\text{total}} \sin\theta}{C_D A \rho}}$$

The terminal velocities for both positions are obtained by substituting the given values:

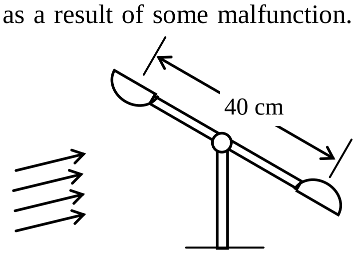
Upright position: $$V = \sqrt{\frac{2(9.81\text{ m/s}^2)(70+15\text{ kg})\sin 8°}{1.1(0.45\text{ m}^2)(1.25\text{ kg/m}^3)}} = 19.4\text{ m/s} = 69.7\text{ km/h} \cong \mathbf{70\ km/h}$$

Racing position: $$V = \sqrt{\frac{2(9.81\text{ m/s}^2)(70+15\text{ kg})\sin 8°}{0.9(0.4\text{ m}^2)(1.25\text{ kg/m}^3)}} = 22.7\text{ m/s} = 81.8\text{ km/h} \cong \mathbf{82\ km/h}$$

Discussion Note that the position of the bicyclist has a significant effect on the drag force, and thus the terminal velocity. So it is no surprise that the bicyclists maintain the racing position during a race.

11-35

Solution The pivot of a wind turbine with two hollow hemispherical cups is stuck as a result of some malfunction. For a given wind speed, the maximum torque applied on the pivot is to be determined.



Assumptions **1** The flow of air in the wind is steady and incompressible. **2** The air flow is turbulent so that the tabulated values of the drag coefficients can be used.

Properties The density of air is given to be ρ = 1.25 kg/m^3. The drag coefficient for a hemispherical cup is 0.4 and 1.2 when the hemispherical and plain surfaces are exposed to wind flow, respectively.

Analysis The maximum torque occurs when the cups are normal to the wind since the length of the moment arm is maximum in this case. Noting that the frontal area is $\pi D^2/4$ for both cups, the drag force acting on each cup is determined to be

Convex side: $$F_{D1} = C_{D1} A \frac{\rho V^2}{2} = (0.4)[\pi(0.08\text{ m})^2/4]\frac{(1.25\text{ kg/m}^3)(15\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 0.283\text{ N}$$

Concave side: $$F_{D2} = C_{D2} A \frac{\rho V^2}{2} = (1.2)[\pi(0.08\text{ m})^2/4]\frac{(1.25\text{ kg/m}^3)(15\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 0.848\text{ N}$$

The moment arm for both forces is 20 cm since the distance between the centers of the two cups is given to be 25 cm. Taking the moment about the pivot, the net torque applied on the pivot is determined to be

$$M_{\max} = F_{D2}L - F_{D1}L = (F_{D2} - F_{D1})L = (0.848 - 0.283\text{ N})(0.20\text{ m}) = \mathbf{0.113\ N\cdot m}$$

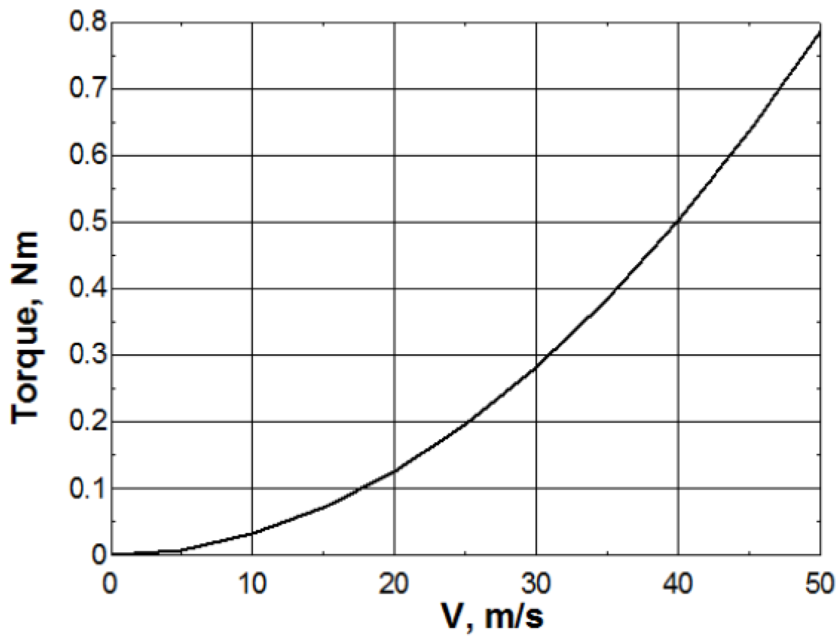
Discussion Note that the torque varies between zero when both cups are aligned with the wind to the maximum value calculated above.

11-36

Solution The previous problem is reconsidered. The effect of wind speed on the torque applied on the pivot as the wind speed varies from 0 to 50 m/s in increments of 5 m/s is to be investigated.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
CD1=0.40 "Curved bottom"
CD2=1.2 "Plain frontal area"
"rho=density(Air, T=T, P=P)" "kg/m^3"
rho=1.25 "kg/m3"
D=0.08 "m"
L=0.25 "m"
A=pi*D^2/4 "m^2"
FD1=CD1*A*(rho*V^2)/2 "N"
FD2=CD2*A*(rho*V^2)/2 "N"
FD_net=FD2-FD1
Torque=(FD2-FD1)*L/2
```



V, m/s	$F_{\text{drag, net}}$, N	Torque, Nm
0	0.00	0.000
5	0.06	0.008
10	0.25	0.031
15	0.57	0.071
20	1.01	0.126
25	1.57	0.196
30	2.26	0.283
35	3.08	0.385
40	4.02	0.503
45	5.09	0.636
50	6.28	0.785

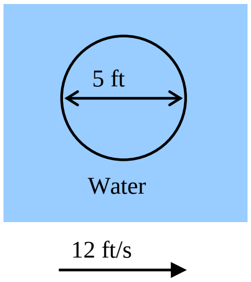
Discussion Since drag force grows as velocity squared, the torque also grows as velocity squared.

11-37E

Solution A spherical tank completely submerged in fresh water is being towed by a ship at a specified velocity. The required towing power is to be determined.

Assumptions **1** The flow is turbulent so that the tabulated value of the drag coefficient can be used. **2** The drag of the towing bar is negligible.

Properties The drag coefficient for a sphere is C_D = 0.2 in turbulent flow (it is 0.5 for laminar flow). We take the density of water to be 62.4 lbm/ft^3.



Analysis The frontal area of a sphere is $A = \pi D^2/4$. Then the drag force acting on the spherical tank is

$$F_D = C_D A \frac{\rho V^2}{2} = (0.2)[\pi(5\text{ ft})^2/4]\frac{(62.4\text{ lbm/ft}^3)(12\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 548\text{ lbf}$$

Since power is force times velocity, the power needed to overcome this drag force during towing is

$$\dot{W}_{\text{Towing}} = \dot{W}_{\text{drag}} = F_D V = (548\text{ lbf})(12\text{ ft/s})\left(\frac{1\text{ kW}}{737.56\text{ lbf}\cdot\text{ft/s}}\right) = 8.92\text{ kW} = 12.0\text{ hp}$$

Therefore, the additional power needed to tow the tank is **12.0 hp**.

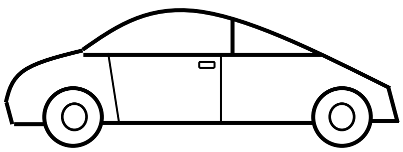
Discussion Note that the towing power is proportional the cube of the velocity. Therefore, the towing power can be reduced to one-eight (which is 1.5 hp) by reducing the towing velocity by half to 6 ft/s. But the towing time will double this time for a given distance.

11-38

Solution The power delivered to the wheels of a car is used to overcome aerodynamic drag and rolling resistance. For a given power, the speed at which the rolling resistance is equal to the aerodynamic drag and the maximum speed of the car are to be determined.

Assumptions **1** The air flow is steady and incompressible. **2** The bearing friction is negligible. **3** The drag and rolling resistance coefficients of the car are constant. **4** The car moves horizontally on a level road.

Properties The density of air is given to be ρ = 1.20 kg/m^3. The drag and rolling resistance coefficients are given to be C_D = 0.32 and C_{RR} = 0.04, respectively.



Analysis (*a*) The rolling resistance of the car is

$$F_{RR} = C_{RR}W = 0.04(950\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 372.8\text{ N}$$

The velocity at which the rolling resistance equals the aerodynamic drag force is determined by setting these two forces equal to each other,

$$F_D = C_D A\frac{\rho V^2}{2} \quad\rightarrow\quad 372.8\text{ N} = (0.32)(1.8\text{ m}^2)\frac{(1.20\text{ kg/m}^3)V^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) \quad\rightarrow\quad V = 32.8\text{ m/s (or 118 km/h)}$$

(*b*) Power is force times speed, and thus the power needed to overcome drag and rolling resistance is the product of the sum of the drag force and the rolling resistance and the velocity of the car,

$$\dot{W}_{\text{total}} = \dot{W}_{\text{drag}} + \dot{W}_{\text{RR}} = (F_D + F_{RR})V = C_D A\frac{\rho V^3}{2} + F_{RR}V$$

Substituting the known quantities, the maximum speed corresponding to a wheel power of 80 kW is determined to be

$$(0.32)(1.8\text{ m}^2)\frac{(1.20\text{ kg/m}^3)V^3}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) + 372.8V = 80{,}000\text{ W}\left(\frac{1\text{ N}\cdot\text{m/s}}{1\text{ W}}\right) \quad\text{or,}\quad 0.3456V^3 + 372.8V = 80{,}000$$

whose solution is V = 55.56 m/s **= 200 km/h**.

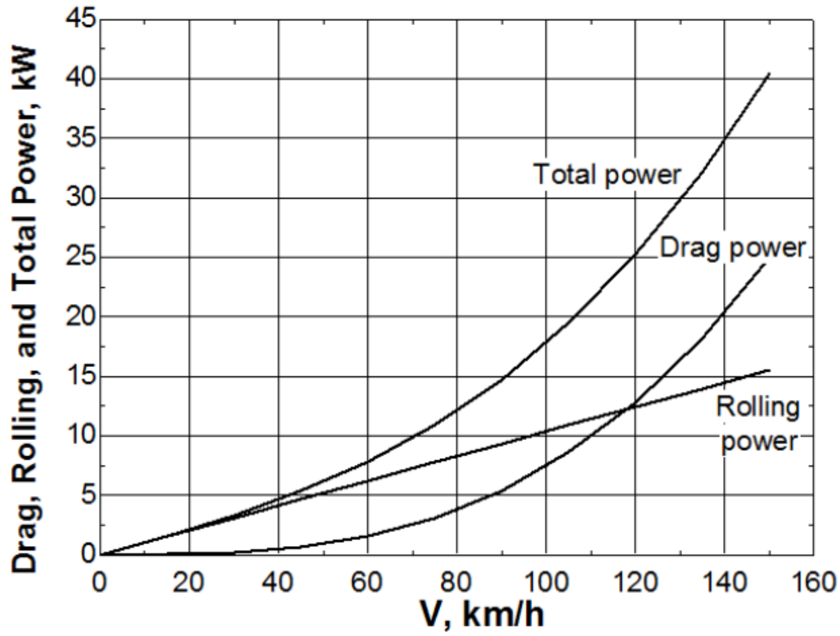
Discussion A net power input of 80 kW is needed to overcome rolling resistance and aerodynamic drag at a velocity of 200 km/h. About 75% of this power is used to overcome drag and the remaining 25% to overcome the rolling resistance. At much higher velocities, the fraction of drag becomes even higher as it is proportional to the cube of car velocity.

11-39

Solution The previous problem is reconsidered. The effect of car speed on the required power to overcome (*a*) rolling resistance, (*b*) the aerodynamic drag, and (*c*) their combined effect as the car speed varies from 0 to 150 km/h in increments of 15 km/h is to be investigated.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
rho=1.20 "kg/m3"
C_roll=0.04
m=950 "kg"
g=9.81 "m/s2"
V=Vel/3.6 "m/s"
W=m*g
F_roll=C_roll*W
A=1.8 "m2"
C_D=0.32
F_D=C_D*A*(rho*V^2)/2 "N"
Power_RR=F_roll*V/1000 "W"
Power_Drag=F_D*V/1000 "W"
Power_Total=Power_RR+Power_Drag
```



V, m/s	W_{drag} kW	$W_{rolling}$, kW	W_{total}, *kW*
0	0.00	0.00	0.00
15	0.03	1.55	1.58
30	0.20	3.11	3.31
45	0.68	4.66	5.33
60	1.60	6.21	7.81
75	3.13	7.77	10.89
90	5.40	9.32	14.72
105	8.58	10.87	19.45
120	12.80	12.43	25.23
135	18.23	13.98	32.20
150	25.00	15.53	40.53

Discussion Notice that the rolling power curve and drag power curve intersect at about 118 km/h (73.3 mph). So, near highway speeds, the overall power is split nearly 50% between rolling drag and aerodynamic drag.

11-40

Solution We are to estimate how many additional liters of fuel are wasted per year by driving with a ball on a car antenna.

Properties ρ_{fuel} = 0.802 kg/L, HV_{fuel} = 44,020 kJ/kg, ρ_{air} = 1.204 kg/m^3, μ_{air} = 1.825 × 10^{-5} kg/m·s

Analysis The additional drag force due to the sun ball is

$$F_D = \tfrac{1}{2}\rho_{\text{air}}V^2C_DA$$

where A is the frontal area of the sphere. The work required to overcome this additional drag is force times distance. So, letting L be the total distance driven in a year,

$$\text{Work}_{\text{drag}} = F_DL = \tfrac{1}{2}\rho_{\text{air}}V^2C_DAL$$

The energy required to perform this work is much greater than this due to overall efficiency of the car engine, transmission, etc. Thus,

$$E_{\text{required}} = \frac{\text{Work}_{\text{drag}}}{\eta_{\text{overall}}} = \frac{\tfrac{1}{2}\rho_{\text{air}}V^2C_DAL}{\eta_{\text{overall}}}$$

But the required energy is also equal to the heating value of the fuel HV times the mass of fuel required. In terms of required fuel *volume*, volume = mass/density. Thus,

$$\boxed{V_{\text{fuel required}} = \frac{m_{\text{fuel required}}}{\rho_{\text{fuel}}} = \frac{E_{\text{required}}/\text{HV}}{\rho_{\text{fuel}}} = \frac{\tfrac{1}{2}\rho_{\text{air}}V^2C_DAL}{\rho_{\text{fuel}}\eta_{\text{overall}}\text{HV}}}$$

The above is our answer in variable form. Finally, we plug in the given values and properties to obtain the numerical answer,

$$V_{\text{fuel required}} = \frac{0.5(1.204\,\text{kg/m}^3)(20.8\,\text{m/s})^2(0.87)(2.08\times10^{-3}\,\text{m}^2)(15{,}000{,}000\,\text{m})}{(0.802\,\text{kg/L})(0.312)(44{,}020\,\text{kJ/kg})}\left(\frac{\text{N}}{\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{\text{kJ}}{1000\,\text{N}\cdot\text{m}}\right)$$

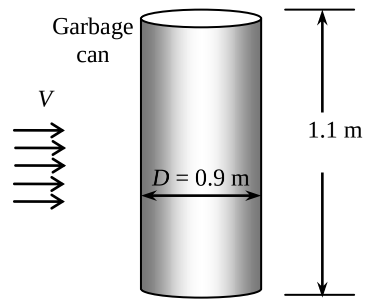
$$= 0.6418\,\text{L} \approx \mathbf{0.64\,L}$$

Discussion If fuel costs around $1 per liter, the total cost is around 64 cents per year to drive with the ball compared to without the ball – Suzy should not worry about wasting this negligible amount of money if the sun ball on her antenna gives her some pleasure when she drives. We give the answer to two significant digits – more than that is unwarranted.

11-41

Solution A garbage can is found in the morning tipped over due to high winds the night before. The wind velocity during the night when the can was tipped over is to be determined.

Assumptions **1** The flow of air in the wind is steady and incompressible. **2** The ground effect on the wind and the drag coefficient is negligible (a crude approximation) so that the resultant drag force acts through the center of the side surface. **3** The garbage can is loaded uniformly so that its weight acts through its center.



Properties The density of air is given to be $\rho = 1.25$ kg/m^3, and the average density of the garbage inside the can is given to be 150 kg/m^3. The drag coefficient of the garbage can is given to be 0.7.

Analysis The volume of the garbage can and the weight of the garbage are

$$\boldsymbol{V} = [\pi D^2/4]H = [\pi(0.90\text{ m})^2/4](1.1\text{ m}) = 0.6998\text{ m}^2$$

$$W = mg = \rho g\boldsymbol{V} = (150\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.6998\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 1029.7\text{ N}$$

When the garbage can is first tipped, the edge on the wind-loaded side of the can will be off the ground, and thus all the reaction forces from the ground will act on the other side. Taking the moment about an axis passing through the contact point and setting it equal to zero gives the required drag force to be

$$\sum M_{\text{contact}} = 0 \;\rightarrow\; F_D \times (H/2) - W \times (D/2) = 0 \quad\rightarrow\quad F_D = \frac{WD}{H} = \frac{(1029.7\text{ N})(0.90\text{ m})}{1.1\text{ m}} = 842.5\text{ N}$$

Noting that the frontal area is *DH*, the wind velocity that will cause this drag force is determined to be

$$F_D = C_D A\frac{\rho V^2}{2} \;\rightarrow\; 842.5\text{ N} = (0.7)[1.1\times 0.90\text{ m}^2]\frac{(1.25\text{ kg/m}^3)V^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) \;\rightarrow\; V = 44.10\text{ m/s}$$

which is equivalent to a wind velocity of $V = 44.10 \times 3.6$ **= 159 km/h.**

Discussion The analysis above shows that under the stated assumptions, the wind velocity at some moment exceeded 159 km/h. But we cannot tell how high the wind velocity has been. Such analysis and predictions are commonly used in forensic engineering.

11-42

Solution A plastic sphere is dropped into water. The terminal velocity of the sphere in water is to be determined.

Assumptions **1** The fluid flow over the sphere is laminar (to be verified). **2** The drag coefficient remains constant.

Properties The density of plastic sphere is 1150 kg/m^3. The density and dynamic viscosity of water at 20°C are ρ = 998 kg/m^3 and μ = 1.002×10^{-3} kg/m·s, respectively. The drag coefficient for a sphere in laminar flow is C_D = 0.5.

Analysis The terminal velocity of a free falling object is reached when the drag force equals the weight of the solid object less the buoyancy force applied by the fluid,

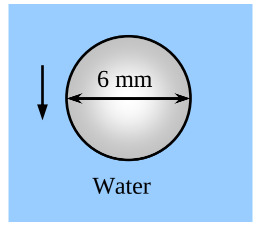
$$F_D = W - F_B \quad \text{where} \quad F_D = C_D A \frac{\rho_f V^2}{2}, \quad W = \rho_s g \boldsymbol{V}, \quad \text{and} \quad F_B = \rho_f g \boldsymbol{V}$$

Here $A = \pi D^2/4$ is the frontal area and $\boldsymbol{V} = \pi D^3/6$ is the volume of the sphere. Substituting and simplifying,

$$C_D A \frac{\rho_f V^2}{2} = \rho_s g \boldsymbol{V} - \rho_f g \boldsymbol{V} \quad \rightarrow \quad C_D \frac{\pi D^2}{4} \frac{\rho_f V^2}{2} = (\rho_s - \rho_f) g \frac{\pi D^3}{6} \quad \rightarrow \quad C_D \frac{V^2}{8} = \left(\frac{\rho_s}{\rho_f} - 1 \right) \frac{gD}{6}$$

Solving for V and substituting,

$$V = \sqrt{\frac{4gD(\rho_s/\rho_f - 1)}{3C_D}} = \sqrt{\frac{4(9.81\ \text{m/s}^2)(0.006\ \text{m})(1150/998 - 1)}{3 \times 0.5}} = \mathbf{0.155\ m/s}$$



The Reynolds number is

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(998\ \text{kg/m}^3)(0.155\ \text{m/s})(6 \times 10^{-3}\ \text{m})}{1.002 \times 10^{-3}\ \text{kg} \cdot \text{m/s}} = 926$$

which is less than 2×10^5. Therefore, the flow is laminar as assumed.

Discussion This problem can also be solved "usually more accurately" using a trial-and-error approach by using C_D data from Fig. 11-34. The C_D value corresponding to Re = 926 is about 0.5, and thus the terminal velocity is the same.

11-43

Solution A spherical hot air balloon that stands still in the air is subjected to winds. The initial acceleration of the balloon is to be determined.

Assumptions **1** The entire hot air balloon can be approximated as a sphere. **2** The wind is steady and turbulent, and blows parallel to the ground.

Properties The drag coefficient for turbulent flow over a sphere is C_D = 0.2 (Table 11-2). We take the density of air to be 1.20 kg/m^3.

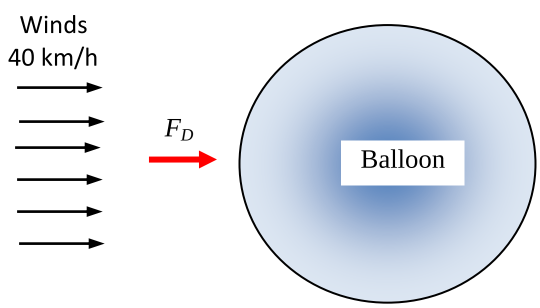
Analysis The frontal area of a sphere is $A = \pi D^2/4$. Noting that 1 m/s = 3.6 km/h, the drag force acting on the balloon is

$$F_D = C_D A \frac{\rho V^2}{2} = (0.2)[\pi(7\text{ m})^2/4]\frac{(1.20\text{ kg/m}^3)(40/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 570.1\text{ N}$$

Then from Newton's 2nd law of motion, the initial acceleration in the direction of the winds becomes

$$a = \frac{F_D}{m} = \frac{570.1\text{ N}}{350\text{ kg}}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{1.63\ m/s^2}$$

Discussion Note that wind-induced drag force is the only mechanism to move a balloon horizontally. The balloon can be moved vertically by dropping weights from the cage of the balloon or by the change of conditions of the light gas in the balloon.

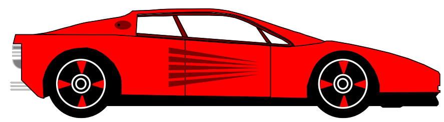


11-44E

Solution The drag coefficient of a sports car increases when the sunroof is open, and it requires more power to overcome aerodynamic drag. The additional power consumption of the car when the sunroof is opened is to be determined at two different velocities.

Assumptions **1** The car moves steadily at a constant velocity on a straight path. **2** The effect of velocity on the drag coefficient is negligible.

Properties The density of air is given to be 0.075 lbm/ft^3. The drag coefficient of the car is given to be C_D = 0.32 when the sunroof is closed, and C_D = 0.41 when it is open.



Analysis (*a*) Noting that 1 mph = 1.4667 ft/s and that power is force times velocity, the drag force acting on the car and the power needed to overcome it at 35 mph are:

Open sunroof: $$F_{D1} = 0.32(18\text{ ft}^2)\frac{(0.075\text{ lbm/ft}^3)(35\times1.4667\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 17.7\text{ lbf}$$

$$\dot{W}_{\text{drag}1} = F_{D1}V = (17.7\text{ lbf})(35\times1.4667\text{ ft/s})\left(\frac{1\text{ kW}}{737.56\text{ lbf}\cdot\text{ft/s}}\right) = 1.23\text{ kW}$$

Closed sunroof: $$F_{D2} = 0.41(18\text{ ft}^2)\frac{(0.075\text{ lbm/ft}^3)(35\times1.4667\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 22.7\text{ lbf}$$

$$\dot{W}_{\text{drag}2} = F_{D2}V = (22.7\text{ lbf})(35\times1.4667\text{ ft/s})\left(\frac{1\text{ kW}}{737.56\text{ lbf}\cdot\text{ft/s}}\right) = 1.58\text{ kW}$$

Therefore, the additional power required for this car when the sunroof is open is

$$\dot{W}_{\text{extra}} = \dot{W}_{\text{drag}2} - \dot{W}_{\text{drag}2} = 1.58 - 1.23 = \mathbf{0.350\ kW} \quad \text{(at 35 mph)}$$

(*b*) We now repeat the calculations for 70 mph:

Open sunroof: $$F_{D1} = 0.32(18\text{ ft}^2)\frac{(0.075\text{ lbm/ft}^3)(70\times1.4667\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 70.7\text{ lbf}$$

$$\dot{W}_{\text{drag}1} = F_{D1}V = (70.7\text{ lbf})(70\times1.4667\text{ ft/s})\left(\frac{1\text{ kW}}{737.56\text{ lbf}\cdot\text{ft/s}}\right) = 9.82\text{ kW}$$

Closed sunroof: $$F_{D2} = 0.41(18\text{ ft}^2)\frac{(0.075\text{ lbm/ft}^3)(70\times1.4667\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = 90.6\text{ lbf}$$

$$\dot{W}_{\text{drag}2} = F_{D2}V = (90.6\text{ lbf})(70\times1.4667\text{ ft/s})\left(\frac{1\text{ kW}}{737.56\text{ lbf}\cdot\text{ft/s}}\right) = 12.6\text{ kW}$$

Therefore, the additional power required for this car when the sunroof is open is

$$\dot{W}_{\text{extra}} = \dot{W}_{\text{drag}2} - \dot{W}_{\text{drag}2} = 12.6 - 9.82 = \mathbf{2.78 kW} \quad \text{(at 70 mph)}$$

Discussion Note that the additional drag caused by open sunroof is 0.35 kW at 35 mph, and 2.78 kW at 70 mph, which is an increase of 8 folds when the velocity is doubled. This is expected since the power consumption to overcome drag is proportional to the cube of velocity.

11-45

Solution The flat mirror of a car is replaced by one with hemispherical back. The amount of fuel and money saved per year as a result are to be determined.

Assumptions **1** The car is driven 24,000 km a year at an average speed of 95 km/h. **2** The effect of the car body on the flow around the mirror is negligible (no interference).

Properties The densities of air and gasoline are taken to be 1.20 kg/m^3 and 750 kg/m^3, respectively. The heating value of gasoline is given to be 44,000 kJ/kg. The drag coefficients C_D are 1.1 for a circular disk, and 0.40 for a hemispherical body.

Analysis The drag force acting on a body is determined from

$$F_D = C_D A \frac{\rho V^2}{2}$$

where A is the frontal area of the body, which is $A = \pi D^2/4$ for both the flat and rounded mirrors. The drag force acting on the flat mirror is

$$F_D = 1.1 \frac{\pi (0.13\,\text{m})^2}{4} \frac{(1.20\,\text{kg/m}^3)(95\,\text{km/h})^2}{2} \left(\frac{1\,\text{m/s}}{3.6\,\text{km/h}}\right)^2 \left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = 6.10\,\text{N}$$

Noting that work is force times distance, the amount of work done to overcome this drag force and the required energy input for a distance of 24,000 km are

$$W_{\text{drag}} = F \times L = (6.10\,\text{N})(24{,}000\,\text{km/year}) = 146{,}400\,\text{kJ/year}$$

$$E_{in} = \frac{W_{\text{drag}}}{\eta_{\text{car}}} = \frac{146{,}400\,\text{kJ}}{0.3} = 488{,}000\,\text{kJ/year}$$

Then the amount and costs of the fuel that supplies this much energy are

$$\text{Amount of fuel} = \frac{m_{\text{fuel}}}{\rho_{\text{fuel}}} = \frac{E_{\text{in}}/\text{HV}}{\rho_{\text{fuel}}} = \frac{(488{,}000\,\text{kJ/year})/(44{,}000\,\text{kJ/kg})}{0.75\,\text{kg/L}} = 14.79\,\text{L/year}$$

$$\text{Cost} = (\text{Amount of fuel})(\text{Unit cost}) = (14.79\,\text{L/year})(\$0.90/\text{L}) = \$13.31/\text{year}$$

That is, the car uses 14.79 L of gasoline at a cost of $13.31 per year to overcome the drag generated by a flat mirror extending out from the side of a car.

The drag force and the work done to overcome it are directly proportional to the drag coefficient. Then the percent reduction in the fuel consumption due to replacing the mirror is equal to the percent reduction in the drag coefficient:

$$\text{Reduction ratio} = \frac{C_{D,\,\text{flat}} - C_{D,\,\text{hemisp}}}{C_{D,\,\text{flat}}} = \frac{1.1 - 0.4}{1.1} = 0.636$$

$$\text{Amount reduction} = (\text{Reduction ratio})(\text{Amount})$$

$$= 0.636(\$14.79\,\text{L/year}) = \mathbf{9.41\,L/year}$$

$$\text{Cost reduction} = (\text{Reduction ratio})(\text{Cost}) = 0.636(\$13.32/\text{year}) = \mathbf{\$8.47/ye}$$

Since a typical car has two rearview mirrors, the driver saves about $17 per year in gasoline by replacing the flat mirrors by the hemispherical ones.

Discussion Note from this example that significant reductions in drag and fuel consumption can be achieved by streamlining the shape of various components and the entire car. So it is no surprise that the sharp corners are replaced in late model cars by rounded contours. This also explains why large airplanes retract their wheels after takeoff, and small airplanes use contoured fairings around their wheels

Flow over Flat Plates

11-46C

Solution We are to define and discuss the average skin friction coefficient over a flat plate.

Analysis The *average friction coefficient* in flow over a flat plate is determined by **integrating the local friction coefficient over the entire length of the plate**, and then dividing it by the length of the plate. Or, it can be determined experimentally by measuring the drag force, and dividing it by the dynamic pressure.

Discussion For the case of a flat plate aligned with the flow, there is no pressure drag, only skin friction drag. Thus, the average friction coefficient is the same as the drag coefficient. This is not true for other body shapes.

11-47C

Solution We are to discuss the fluid property responsible for the development of a boundary layer.

Analysis The **fluid viscosity** is responsible for the development of the velocity boundary layer. Velocity forces the boundary layer closer to the wall. Therefore, the higher the velocity (and thus Reynolds number), the lower the thickness of the boundary layer.

Discussion All fluids have viscosity – a measure of frictional forces in a fluid. There is no such thing as an *inviscid fluid*, although there are *regions*, called *inviscid flow regions*, in which viscous effects are negligible.

11-48C

Solution We are to define and discuss the friction coefficient for flow over a flat plate.

Analysis The *friction coefficient* represents **the resistance to fluid flow over a flat plate**. It is proportional to the drag force acting on the plate. The drag coefficient for a flat surface is equivalent to the mean friction coefficient.

Discussion In flow over a flat plate aligned with the flow, there is no pressure (form) drag – only friction drag.

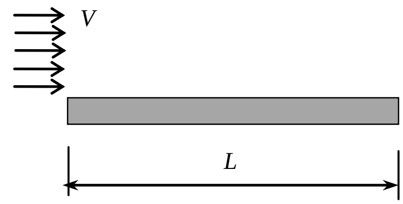
11-49

Solution Laminar flow of a fluid over a flat plate is considered. The change in the drag force is to be determined when the free-stream velocity of the fluid is tripled.

Analysis For the laminar flow of a fluid over a flat plate the drag force is given by

$$F_{D1} = C_f A \frac{\rho V^2}{2} \quad \text{where} \quad C_f = \frac{1.328}{\text{Re}^{0.5}}$$

Therefore



$$F_{D1} = \frac{1.328}{\text{Re}^{0.5}} A \frac{\rho V^2}{2}$$

Substituting Reynolds number relation, we get

$$F_{D1} = \frac{1.328}{\left(\frac{VL}{\upsilon}\right)^{0.5}} A \frac{\rho V^2}{2} = 0.664 V^{3/2} A \frac{\upsilon^{0.5}}{L^{0.5}}$$

When the free-stream velocity of the fluid is tripled, the new value of the drag force on the plate becomes

$$F_{D2} = \frac{1.328}{\left(\frac{(3V)L}{\upsilon}\right)^{0.5}} A \frac{\rho (3V)^2}{2} = 0.664 (3V)^{3/2} A \frac{\upsilon^{0.5}}{L^{0.5}}$$

The ratio of drag forces corresponding to V and $2V$ is

$$\frac{F_{D2}}{F_{D1}} = \frac{(3V)^{3/2}}{V^{3/2}} = \mathbf{3^{3/2}} = \mathbf{5.20}$$

In other words, there is a **5.20-fold increase** in the drag force on the plate.

Discussion Note that the drag force increases almost three times in laminar flow when the fluid velocity is doubled.

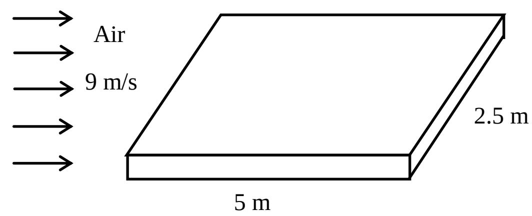
11-50

Solution Air flows over a plane surface at high elevation. The drag force acting on the top surface of the plate is to be determined for flow along the two sides of the plate.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The surface of the plate is smooth.

Properties The dynamic viscosity is independent of pressure, and for air at 25°C it is $\mu = 1.849\times10^{-5}$ kg/m·s. The air density at 25°C = 298 K and 83.4 kPa is

$$\rho = \frac{P}{RT} = \frac{83.4\ \text{kPa}}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\ \text{K})} = 0.9751\ \text{kg/m}^3$$



Analysis (a) If the air flows parallel to the 8 m side, the Reynolds number becomes

$$\mathrm{Re}_L = \frac{\rho VL}{\mu} = \frac{(0.9751\ \text{kg/m}^3)(9\ \text{m/s})(5\ \text{m})}{1.849\times10^{-5}\ \text{kg/m}\cdot\text{s}} = 2.373\times10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow, and the friction coefficient is determined to be

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} - \frac{1742}{\mathrm{Re}_L} = \frac{0.074}{(2.373\times10^6)^{1/5}} - \frac{1742}{2.373\times10^6} = 0.003194$$

Noting that the pressure drag is zero and thus $C_D = C_f$ for a flat plate, the drag force acting on the top surface of the plate becomes

$$F_D = C_f A\frac{\rho V^2}{2} = 0.003194\times(5\times2.5\ \text{m}^2)\frac{(0.9751\ \text{kg/m}^3)(9\ \text{m/s})^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{1.58\ N}$$

(b) If the air flows parallel to the 2.5 m side, the Reynolds number is

$$\mathrm{Re}_L = \frac{\rho VL}{\mu} = \frac{(0.9751\ \text{kg/m}^3)(9\ \text{m/s})(2.5\ \text{m})}{1.849\times10^{-5}\ \text{kg/m}\cdot\text{s}} = 1.187\times10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow, and the friction coefficient is determined to be

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} - \frac{1742}{\mathrm{Re}_L} = \frac{0.074}{(1.187\times10^6)^{1/5}} - \frac{1742}{1.187\times10^6} = 0.003044$$

Then the drag force acting on the top surface of the plate becomes

$$F_D = C_f A\frac{\rho V^2}{2} = 0.003044\times(5\times2.5\ \text{m}^2)\frac{(0.9751\ \text{kg/m}^3)(9\ \text{m/s})^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{1.50\ N}$$

Discussion Note that the drag force is proportional to density, which is proportional to the pressure. Therefore, the altitude has a major influence on the drag force acting on a surface. Commercial airplanes take advantage of this phenomenon and cruise at high altitudes where the air density is much lower to save fuel.

11-51

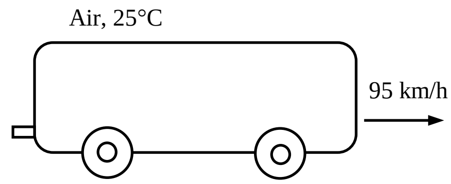
Solution A train is cruising at a specified velocity. The drag force acting on the top surface of a passenger car of the train is to be determined.

Assumptions **1** The air flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The top surface of the train is smooth (in reality it can be rough). **5** The air is calm (no significant winds).

Properties The density and kinematic viscosity of air at 1 atm and 25°C are $\rho = 1.184$ kg/m^3 and $\nu = 1.562\times10^{-5}$ m^2/s .

Analysis The Reynolds number is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{[95/3.6 \text{ m/s}](8 \text{ m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 1.352\times10^7$$



which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow, and the friction coefficient is determined to be

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} - \frac{1742}{\mathrm{Re}_L} = \frac{0.074}{(1.352\times10^7)^{1/5}} - \frac{1742}{1.352\times10^7} = 0.002645$$

Noting that the pressure drag is zero and thus $C_D = C_f$ for a flat plate, the drag force acting on the surface becomes

$$F_D = C_f A\frac{\rho V^2}{2} = 0.002645\times(8\times2.1 \text{ m}^2)\frac{(1.184 \text{ kg/m}^3)(95/3.6 \text{ m/s})^2}{2}\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{18.3 N}$$

Discussion Note that we can solve this problem using the turbulent flow relation (instead of the combined laminar-turbulent flow relation) without much loss in accuracy since the Reynolds number is much greater than the critical value. Also, the actual drag force will probably be greater because of the surface roughness effects.

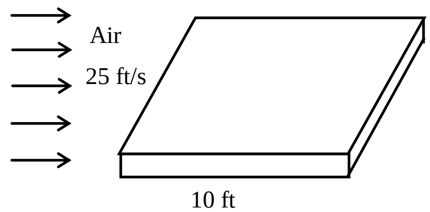
11-52E

Solution Air flows over a flat plate. The local friction coefficients at intervals of 1 ft is to be determined and plotted against the distance from the leading edge.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The surface of the plate is smooth.

Properties The density and kinematic viscosity of air at 1 atm and 70°F are $\rho = 0.07489$ lbm/ft^3 and $\nu = 0.5913$ ft^2/h = 1.643×10^{-4} ft^2/s .

Analysis For the first 1 ft interval, the Reynolds number is



$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(25\text{ ft/s})(1\text{ ft})}{1.643\times10^{-4}\text{ ft}^2\text{/s}} = 1.522\times10^5$$

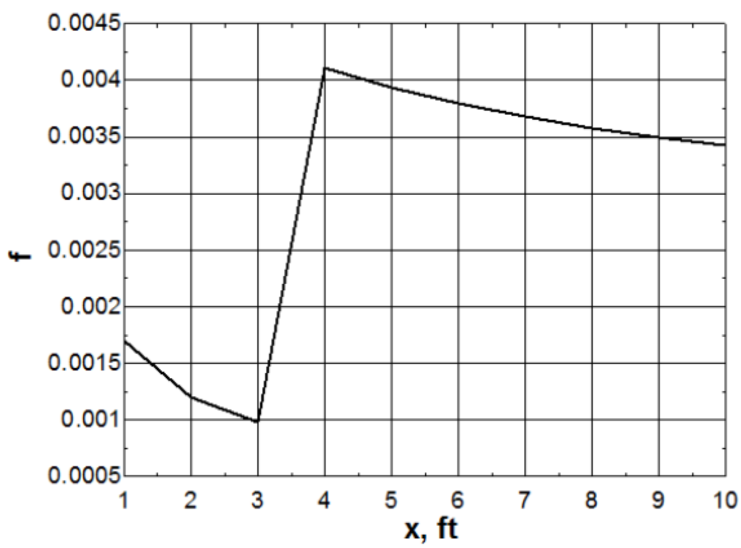
which is less than the critical value of 5×10^5. Therefore, the flow is laminar. The local friction coefficient is

$$C_{f,x} = \frac{0.664}{\mathrm{Re}^{0.5}} = \frac{0.664}{(1.522\times10^5)^{0.5}} = 0.001702$$

We repeat calculations for all 1-ft intervals. The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
rho=0.07489 "lbm/ft3"
nu=0.5913/3600 "ft2/s"
V=25
"Local Re and C_f"
Re=x*V/nu
"f=0.664/Re^0.5"
f=0.059/Re^0.2
```

x, ft	Re	C_f
1	1.522E+05	0.001702
2	3.044E+05	0.001203
3	4.566E+05	0.000983
4	6.088E+05	0.004111
5	7.610E+05	0.003932
6	9.132E+05	0.003791
7	1.065E+06	0.003676
8	1.218E+06	0.003579
9	1.370E+06	0.003496
10	1.522E+06	0.003423



Discussion Note that the Reynolds number exceeds the critical value for $x > 3$ ft, and thus the flow is turbulent over most of the plate. For $x > 3$ ft, we used $C_f = 0.074/\mathrm{Re}_L^{1/5}$-$1742/\mathrm{Re}_L$ for friction coefficient. Note that C_f decreases with Re in both laminar and turbulent flows.

11-53

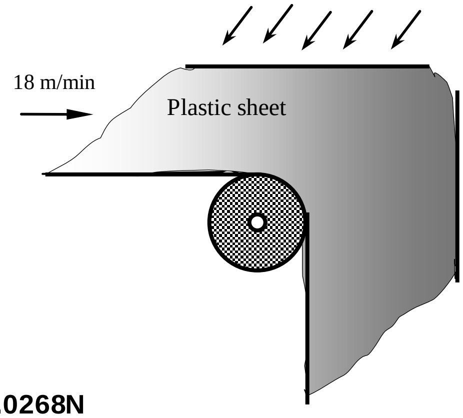
Solution Air flows on both sides of a continuous sheet of plastic. The drag force air exerts on the plastic sheet in the direction of flow is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** Both surfaces of the plastic sheet are smooth. **5** The plastic sheet does not vibrate and thus it does not induce turbulence in air flow.

Properties The density and kinematic viscosity of air at 1 atm and 60°C are $\rho = 1.059$ kg/m^3 and $\nu = 1.896\times10^{-5}$ m^2/s .

Analysis The length of the cooling section is

$$L = V_{\text{sheet}}\Delta t = [(18/60)\ \text{m/s}](2\ \text{s}) = 0.6\ \text{m}$$



The Reynolds number is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(4\ \text{m/s})(1.2\ \text{m})}{1.896\times10^{-5}\ \text{m}^2/\text{s}} = 2.532\times10^5$$

which is less than the critical Reynolds number. Thus the flow is laminar. The area on both sides of the sheet exposed to air flow is

$$A = 2wL = 2(1.2\ \text{m})(0.6\ \text{m}) = 1.44\ \text{m}^2$$

Then the friction coefficient and the drag force become

$$C_f = \frac{1.328}{\mathrm{Re}_L^{\ 0.5}} = \frac{1.328}{(2.532\times10^5)^{0.5}} = 0.002639$$

$$F_D = C_f A\frac{\rho V^2}{2} = (0.002639)(1.2\ \text{m}^2)\frac{(1.059\ \text{kg/m}^3)(4\ \text{m/s})^2}{2} = \mathbf{0.0268\ N}$$

Discussion Note that the Reynolds number remains under the critical value, and thus the flow remains laminar over the entire plate. In reality, the flow may be turbulent because of the motion of the plastic sheet.

11-54E

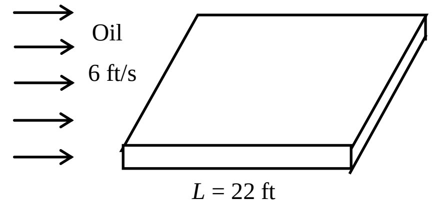
Solution Light oil flows over a flat plate. The total drag force per unit width of the plate is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$. **3** The surface of the plate is smooth.

Properties The density and kinematic viscosity of light oil at 75°F are $\rho = 55.3$ lbm/ft^3 and $\nu = 7.751\times10^{-3}$ ft^2/s.

Analysis Noting that L = 22 ft, the Reynolds number at the end of the plate is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(6\ \mathrm{ft/s})(22\ \mathrm{ft})}{7.751\times10^{-3}\ \mathrm{ft^2/s}} = 1.703\times10^4$$



which is less than the critical Reynolds number. Thus we have *laminar flow* over the entire plate, and the average friction coefficient is determined from

$$C_f = 1.328\,\mathrm{Re}_L^{-0.5} = 1.328\times(1.703\times10^4)^{-0.5} = 0.01018$$

Noting that the pressure drag is zero and thus $C_D = C_f$ for a flat plate, the drag force acting on the top surface of the plate per unit width becomes

$$F_D = C_f A\frac{\rho V^2}{2} = 0.01018\times(22\times1\ \mathrm{ft^2})\frac{(55.3\ \mathrm{lbm/ft^3})(6\ \mathrm{ft/s})^2}{2}\left(\frac{1\ \mathrm{lbf}}{32.2\ \mathrm{lbm\cdot ft/s^2}}\right) = \mathbf{6.92\ lbf}$$

The total drag force acting on the entire plate can be determined by multiplying the value obtained above by the width of the plate.

Discussion The force per unit width corresponds to the weight of a mass of 6.92 lbm. Therefore, a person who applies an equal and opposite force to the plate to keep it from moving will feel like he or she is using as much force as is necessary to hold a 6.92 lbm mass from dropping.

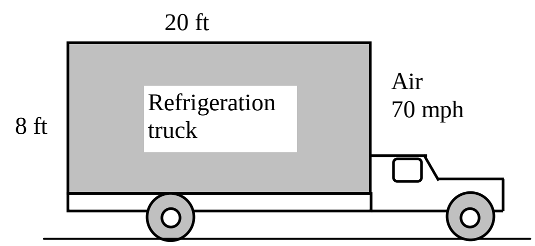
11-55E

Solution A refrigeration truck is traveling at a specified velocity. The drag force acting on the top and side surfaces of the truck and the power needed to overcome it are to be determined.

Assumptions **1** The process is steady and incompressible. **2** The airflow over the entire outer surface is turbulent because of constant agitation. **3** Air is an ideal gas. **4** The top and side surfaces of the truck are smooth (in reality they can be rough). **5** The air is calm (no significant winds).

Properties The density and kinematic viscosity of air at 1 atm and 80°F are ρ = 0.07350 lbm/ft^3 and ν = 0.6110 ft^2/s = 1.697×10^{-4} ft^2/s .

Analysis The Reynolds number is



$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{[70 \times 1.4667 \text{ ft/s}](20 \text{ ft})}{1.697 \times 10^{-4} \text{ ft}^2/\text{s}} = 1.210 \times 10^7$$

The air flow over the entire outer surface is assumed to be turbulent. Then the friction coefficient becomes

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} = \frac{0.074}{(1.210 \times 10^7)^{1/5}} = 0.002836$$

The area of the top and side surfaces of the truck is

$$A = A_{\text{top}} + 2A_{\text{side}} = 9{\times}20{+}2{\times}8{\times}20 = 500 \text{ ft}^2$$

Noting that the pressure drag is zero and thus $C_D = C_f$ for a plane surface, the drag force acting on these surfaces becomes

$$F_D = C_f A \frac{\rho V^2}{2} = 0.002836 \times (500 \text{ ft}^2) \frac{(0.07350 \text{ lbm/ft}^3)(70 \times 1.4667 \text{ ft/s})^2}{2} \left(\frac{1 \text{ lbf}}{32.2 \text{ lbm} \cdot \text{ft/s}^2} \right) = \mathbf{17.1 \text{ lbf}}$$

Noting that power is force times velocity, the power needed to overcome this drag force is

$$\dot{W}_{\text{drag}} = F_D V = (17.1 \text{ lbf})(70 \times 1.4667 \text{ ft/s}) \left(\frac{1 \text{ kW}}{737.56 \text{ lbf} \cdot \text{ft/s}} \right) = \mathbf{2.37 \text{ kW}}$$

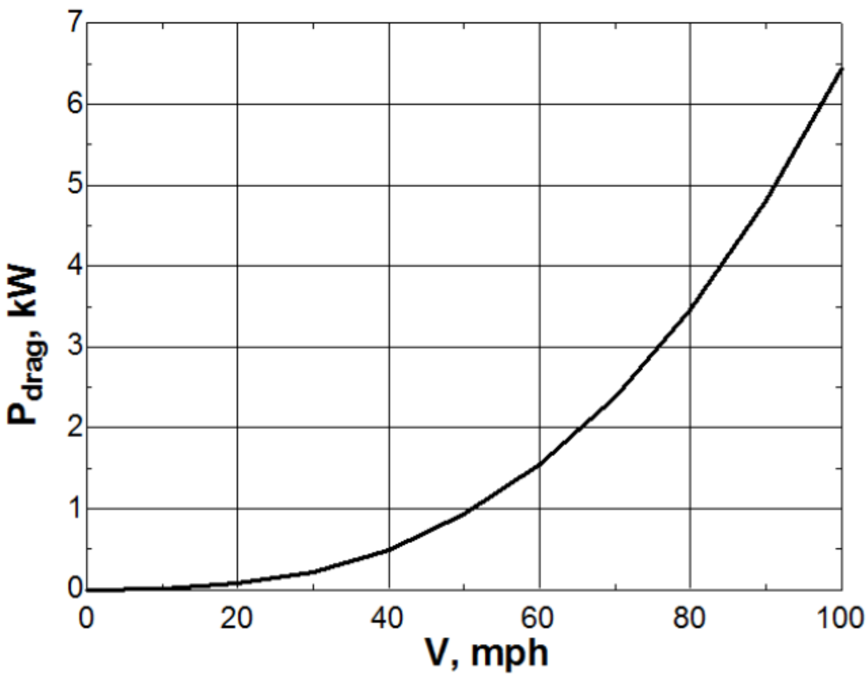
Discussion Note that the calculated drag force (and the power required to overcome it) is very small. This is not surprising since the drag force for blunt bodies is almost entirely due to pressure drag, and the friction drag is practically negligible compared to the pressure drag.

11-56E

Solution The previous problem is reconsidered. The effect of truck speed on the total drag force acting on the top and side surfaces, and the power required to overcome as the truck speed varies from 0 to 100 mph in increments of 10 mph is to be investigated.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
rho=0.07350 "lbm/ft3"
nu=0.6110/3600 "ft2/s"
V=Vel*1.4667 "ft/s"
L=20 "ft"
W=2*8+9
A=L*W
Re=L*V/nu
Cf=0.074/Re^0.2
g=32.2 "ft/s2"
F=Cf*A*(rho*V^2)/2/32.2 "lbf"
Pdrag=F*V/737.56 "kW"
```



V, mph	Re	F_{drag}, lbf	P_{drag}, kW
0	0	0.00	0.000
10	1.728E+06	0.51	0.010
20	3.457E+06	1.79	0.071
30	5.185E+06	3.71	0.221
40	6.913E+06	6.23	0.496
50	8.642E+06	9.31	0.926
60	1.037E+07	12.93	1.542
70	1.209E+07	17.06	2.375
80	1.382E+07	21.69	3.451
90	1.555E+07	26.82	4.799
100	1.728E+07	32.42	6.446

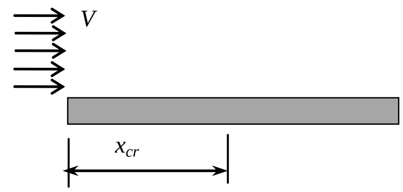
Discussion The required power increases rapidly with velocity – in fact, as velocity *cubed*.

11-57

Solution Air is flowing over a long flat plate with a specified velocity. The distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The surface of the plate is smooth.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are $\rho = 1.184$ kg/m^3 and $\nu = 1.562\times10^{-5}$ m^2/s .



Analysis The critical Reynolds number is given to be $\text{Re}_{cr} = 5\times10^5$. The distance from the leading edge of the plate where the flow becomes turbulent is the distance x_{cr} where the Reynolds number becomes equal to the critical Reynolds number,

$$\text{Re}_{cr} = \frac{Vx_{cr}}{\nu} \quad\rightarrow\quad x_{cr} = \frac{\nu \text{Re}_{cr}}{V} = \frac{\left(1.562\times10^{-5}\ \text{m}^2/\text{s}\right)\left(5\times10^5\right)}{8\ \text{m/s}} = \mathbf{0.976\ m}$$

The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness relation,

$$\delta_{v,x} = \frac{4.91x}{\text{Re}_x^{1/2}} \quad\rightarrow\quad \delta_{v,cr} = \frac{4.91x_{cr}}{\text{Re}_{cr}^{1/2}} = \frac{4.91(0.976\ \text{m})}{(5\times10^5)^{1/2}} = 0.00678\ \text{m} = \mathbf{0.678 cm}$$

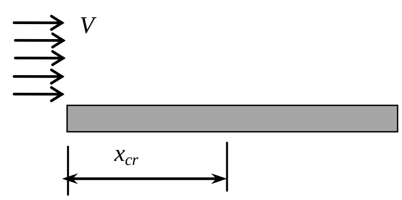
Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

11-58

Solution Water is flowing over a long flat plate with a specified velocity. The distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** The surface of the plate is smooth.

Properties The density and dynamic viscosity of water at 1 atm and 25°C are $\rho = 997$ kg/m^3 and $\mu = 0.891\times10^{-3}$ kg/m·s.



Analysis The critical Reynolds number is given to be $\text{Re}_{cr} = 5\times10^5$. The distance from the leading edge of the plate where the flow becomes turbulent is the distance x_{cr} where the Reynolds number becomes equal to the critical Reynolds number,

$$\text{Re}_{cr} = \frac{\rho V x_{cr}}{\mu} \quad\rightarrow\quad x_{cr} = \frac{\mu \text{Re}_{cr}}{\rho V} = \frac{(0.891\times10^{-3}\ \text{kg/m}\cdot\text{s})(5\times10^5)}{(997\ \text{kg/m}^3)(8\ \text{m/s})} = \mathbf{0.056m}$$

The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness relation,

$$\delta_{v,x} = \frac{5x}{\text{Re}_x^{1/2}} \quad\rightarrow\quad \delta_{v,cr} = \frac{4.91x_{cr}}{\text{Re}_{cr}^{1/2}} = \frac{4.91(0.056\ \text{m})}{(5\times10^5)^{1/2}} = 0.00039\ \text{m} = \mathbf{0.39mm}$$

Therefore, the flow becomes turbulent after about 5.6 cm from the leading edge of the plate, and the thickness of the boundary layer at that location is 0.39 mm.

Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

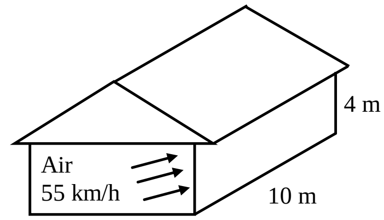
11-59

Solution Wind is blowing parallel to the side wall of a house. The drag force acting on the wall is to be determined for two different wind velocities.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The wall surface is smooth (the actual wall surface is usually very rough). **5** The wind blows parallel to the wall.

Properties The density and kinematic viscosity of air at 1 atm and 5°C are $\rho = 1.269$ kg/m^3 and $\nu = 1.382\times10^{-5}$ m^2/s .

Analysis The Reynolds number is



$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{\left[(55/3.6)\text{ m/s}\right](10\text{ m})}{1.382\times10^{-5}\text{ m}^2\text{/s}} = 1.105\times10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow, and the friction coefficient is

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} - \frac{1742}{\mathrm{Re}_L} = \frac{0.074}{(1.105\times10^7)^{1/5}} - \frac{1742}{1.105\times10^7} = 0.002730$$

Noting that the pressure drag is zero and thus $C_D = C_f$ for a flat plate, the drag force acting on the wall surface is

$$F_D = C_f A\frac{\rho V^2}{2} = 0.00273\times(10\times4\text{ m}^2)\frac{(1.269\text{ kg/m}^3)(55/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 16.17\text{ N} \cong \mathbf{16\ N}$$

(b) When the wind velocity is doubled to 110 km/h, the Reynolds number becomes

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{\left[(110/3.6)\text{ m/s}\right](10\text{ m})}{1.382\times10^{-5}\text{ m}^2\text{/s}} = 2.211\times10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow, and the friction coefficient and the drag force become

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} - \frac{1742}{\mathrm{Re}_L} = \frac{0.074}{(2.211\times10^7)^{1/5}} - \frac{1742}{2.211\times10^7} = 0.002435$$

$$F_D = C_f A\frac{\rho V^2}{2} = 0.002435\times(10\times4\text{ m}^2)\frac{(1.269\text{ kg/m}^3)(110/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 57.70\text{ N} \cong \mathbf{58\ N}$$

Treating flow over the side wall of a house as flow over a flat plate is not quite realistic. When flow hits a bluff body like a house, it separates at the sharp corner and a separation bubble exists over most of the side panels of the house. Therefore, flat plat boundary layer equations are not appropriate for this problem, and the entire house should considered in the solution instead.

Discussion Note that the actual drag will probably be much higher since the wall surfaces are typically very rough. Also, we can solve this problem using the turbulent flow relation (instead of the combined laminar-turbulent flow relation) without much loss in accuracy. Finally, the drag force nearly quadruples when the velocity is doubled. This is expected since the drag force is proportional to the square of the velocity, and the effect of velocity on the friction coefficient is small.

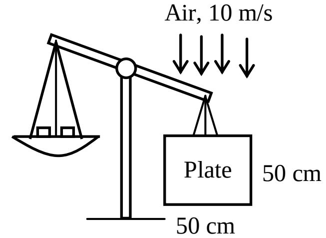
11-60

Solution The weight of a thin flat plate exposed to air flow on both sides is balanced by a counterweight. The mass of the counterweight that needs to be added in order to balance the plate is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The surfaces of the plate are smooth.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are $\rho = 1.184$ kg/m^3 and $\nu = 1.562\times10^{-5}$ m^2/s .

Analysis The Reynolds number is



$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(10 \text{ m/s})(0.5 \text{ m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 3.201\times10^5$$

which is less than the critical Reynolds number of 5×10^5 . Therefore the flow is laminar. The average friction coefficient, drag force and the corresponding mass are

$$C_f = \frac{1.328}{\mathrm{Re}_L^{\,0.5}} = \frac{1.328}{(3.201\times10^5)^{0.5}} = 0.002347$$

$$F_D = C_f A \frac{\rho V^2}{2} = (0.002347)[(2\times0.5\times0.5)\text{ m}^2]\frac{(1.184 \text{ kg/m}^3)(10 \text{ m/s})^2}{2} = 0.0695 \text{ kg}\cdot\text{m/s}^2 = 0.0695 \text{ N}$$

The mass whose weight is 0.0695 N is

$$m = \frac{F_D}{g} = \frac{0.0695 \text{ kg.m/s}^2}{9.81 \text{ m/s}^2} = \mathbf{0.0071 kg} = \mathbf{7.1 g}$$

Therefore, the mass of the counterweight must be 7.1 g to counteract the drag force acting on the plate.

Discussion Note that the apparatus described in this problem provides a convenient mechanism to measure drag force and thus drag coefficient.

Flow across Cylinders and Spheres

11-61C

Solution We are to discuss why flow separation is delayed in turbulent flow over circular cylinders.

Analysis Flow separation in flow over a cylinder is delayed in turbulent flow **because of the extra mixing due to random fluctuations and the transverse motion**.

Discussion As a result of the turbulent mixing, a turbulent boundary layer can resist flow separation better than a laminar boundary layer can, under otherwise similar conditions.

11-62C

Solution We are to discuss how pressure drag and friction drag differ in flow over blunt bodies.

Analysis *Friction drag* is due to the **shear stress at the surface** whereas *pressure drag* is due to **the pressure differential between the front and back sides of the body** because of the wake that is formed in the rear.

Discussion For a blunt or bluff body, pressure drag is usually greater than friction drag, while for a well-streamlined body, the opposite is true. For the case of a flat plate aligned with the flow, *all* of the drag is friction drag.

11-63C

Solution We are to discuss why the drag coefficient suddenly drops when the flow becomes turbulent.

Analysis Turbulence **moves the fluid separation point further back on the rear of the body, reducing the size of the wake, and thus the magnitude of the pressure drag** (which is the dominant mode of drag). As a result, the drag coefficient suddenly drops. In general, turbulence increases the drag coefficient for flat surfaces, but the drag coefficient usually remains constant at high Reynolds numbers when the flow is turbulent.

Discussion The sudden drop in drag is sometimes referred to as the *drag crisis*.

11-64

Solution A spherical dust particle is suspended in the air at a fixed point as a result of an updraft air motion. The magnitude of the updraft velocity is to be determined using Stokes law.

Assumptions **1** The Reynolds number is low (at the order of 1) so that Stokes law is applicable (to be verified). **2** The updraft is steady and incompressible. **3** The buoyancy force applied by air to the dust particle is negligible since $\rho_{air} << \rho_{dust}$ (besides, the uncertainty in the density of dust is greater than the density of air). (We will solve the problem without utilizing this assumption for generality).

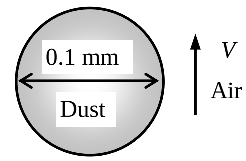
Properties The density of dust is given to be ρ_s = 2.1 g/cm^3 = 2100 kg/m^3. The density and dynamic viscosity of air at 1 atm and 25°C are ρ_f = 1.184 kg/m^3 and μ = 1.849×10^{-5} kg/m·s.

Analysis The terminal velocity of a free falling object is reached (or the suspension of an object in a flow stream is established) when the drag force equals the weight of the solid object less the buoyancy force applied by the surrounding fluid,

$$F_D = W - F_B \quad \text{where} \quad F_D = 3\pi\mu VD \text{ (Stokes law)}, \quad W = \rho_s g \boldsymbol{V}, \quad \text{and } F_B = \rho_f g \boldsymbol{V}$$

Here $\boldsymbol{V} = \pi D^3/6$ is the volume of the sphere. Substituting,

$$3\pi\mu VD = \rho_s g \boldsymbol{V} - \rho_f g \boldsymbol{V} \quad \rightarrow \quad 3\pi\mu VD = (\rho_s - \rho_f) g \frac{\pi D^3}{6}$$



Solving for the velocity V and substituting the numerical values, the updraft velocity is determined to be

$$V = \frac{gD^2(\rho_s - \rho_f)}{18\mu} = \frac{(9.81 \text{ m/s}^2)(0.0001 \text{ m})^2(2100\text{-}1.184) \text{ kg/m}^3}{18(1.849\times10^{-5} \text{ kg/m}\cdot\text{s})} = 0.6186 \text{ m/s} \cong \mathbf{0.62 \ m/s}$$

The Reynolds number in this case is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(1.184 \text{ kg/m}^3)(0.619 \text{ m/s})(0.0001 \text{ m})}{1.849\times10^{-5} \text{ kg}\cdot\text{m/s}} = 4.0$$

which is in the order of 1. Therefore, the creeping flow idealization and thus Stokes law is applicable, and the value calculated is valid.

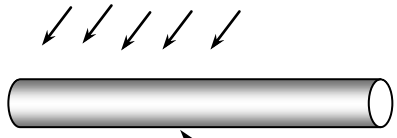
Discussion Flow separation starts at about Re = 10. Therefore, Stokes law can be used as an approximation for Reynolds numbers up to this value, but this should be done with care.

11-65

Solution A pipe is exposed to high winds. The drag force exerted on the pipe by the winds is to be determined.

Assumptions **1** The outer surface of the pipe is smooth so that Fig. 11-34 can be used to determine the drag coefficient. **2** Air flow in the wind is steady and incompressible. **3** The turbulence in the wind is not considered. **4**The direction of wind is normal to the pipe.

Properties The density and kinematic viscosity of air at 1 atm and 10°C are ρ = 1.246 kg/m^3 and ν = 1.426×10^{-5} m^2/s.



Analysis Noting that D = 0.05 m and 1 m/s = 3.6 km/h, the Reynolds number for flow over the pipe is

$$\text{Re} = \frac{VD}{\nu} = \frac{(50/3.6\text{ m/s})(0.05\text{ m})}{1.426\times10^{-5}\text{ m}^2/\text{s}} = 0.4870\times10^5$$

The drag coefficient corresponding to this value is, from Fig. 11-34, C_D = 1.0. Also, the frontal area for flow past a cylinder is $A = LD$. Then the drag force becomes

$$F_D = C_D A\frac{\rho V^2}{2} = 1.0(1\times0.05\text{ m}^2)\frac{(1.246\text{ kg/m}^3)(50/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{6.01N}\ \text{(per m length)}$$

Discussion Note that the drag force acting on a unit length of the pipe is equivalent to the weight of 0.6 kg mass. The total drag force acting on the entire pipe can be obtained by multiplying the value obtained by the pipe length. It should be kept in mind that wind turbulence may reduce the drag coefficients by inducing turbulence and delaying flow separation.

11-66

Solution Spherical hail is falling freely in the atmosphere. The terminal velocity of the hail in air is to be determined.

Assumptions **1** The surface of the hail is smooth so that Fig. 11-34 can be used to determine the drag coefficient. **2** The variation of the air properties with altitude is negligible. **3** The buoyancy force applied by air to hail is negligible since $\rho_{air} << \rho_{hail}$ (besides, the uncertainty in the density of hail is greater than the density of air). **4** Air flow over the hail is steady and incompressible when terminal velocity is established. **5** The atmosphere is calm (no winds or drafts).

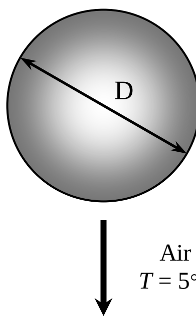
Properties The density and kinematic viscosity of air at 1 atm and 5°C are ρ = 1.269 kg/m^3 and ν = 1.382×10^{-5} m^2/s. The density of hail is given to be 910 kg/m^3.

Analysis The terminal velocity of a free falling object is reached when the drag force equals the weight of the solid object less the buoyancy force applied by the fluid, which is negligible in this case,

$$F_D = W - F_B \quad \text{where} \quad F_D = C_D A \frac{\rho_f V^2}{2}, \quad W = mg = \rho_s g \boldsymbol{V} = \rho_s g(\pi D^3/6), \quad \text{and} \quad F_B \cong 0$$

and $A = \pi D^2/4$ is the frontal area. Substituting and simplifying,

$$C_D A \frac{\rho_f V^2}{2} = W \quad \rightarrow \quad C_D \frac{\pi D^2}{4} \frac{\rho_f V^2}{2} = \rho_s g \frac{\pi D^3}{6} \quad \rightarrow \quad C_D \rho_f V^2 = \rho_s g \frac{4D}{3}$$



Solving for V and substituting,

$$V = \sqrt{\frac{4g\rho_s D}{3C_D \rho_f}} = \sqrt{\frac{4(9.81\,\text{m/s}^2)(910\text{ kg/m}^3)(0.008\text{ m})}{3C_D(1.269\text{ kg/m}^3)}} \quad \rightarrow \quad V = \frac{8.662}{\sqrt{C_D}} \qquad (1)$$

The drag coefficient C_D is to be determined from Fig. 11-34, but it requires the Reynolds number which cannot be calculated since we do not know velocity. Therefore, the solution requires a trial-error approach. First we express the Reynolds number as

$$\text{Re} = \frac{VD}{\nu} = \frac{V(0.008\text{ m})}{1.382\times10^{-5}\text{ m}^2/\text{s}} \quad \rightarrow \quad \text{Re} = 578.9V \qquad (2)$$

Now we choose a velocity in m/s, calculate the Re from Eq. 2, read the corresponding C_D from Fig. 11-34, and calculate V from Eq. 1. Repeat calculations until the assumed velocity matches the calculated velocity. With this approach the terminal velocity is determined to be

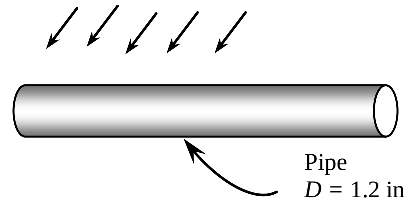
V = **13.7 m/s**

The corresponding Re and C_D values are Re = 7930 and C_D = 0.40. Therefore, the velocity of hail will remain constant when it reaches the terminal velocity of 13.7 m/s = 49 km/h.

Discussion The simple analysis above gives us a reasonable value for the terminal velocity. A more accurate answer can be obtained by a more detailed (and complex) analysis by considering the variation of air properties with altitude, and by considering the uncertainty in the drag coefficient (a hail is not necessarily spherical and smooth).

11-67E

Solution A pipe is crossing a river while remaining completely immersed in water. The drag force exerted on the pipe by the river is to be determined.



Assumptions **1** The outer surface of the pipe is smooth so that Fig. 11-34 can be used to determine the drag coefficient. **2** Water flow in the river is steady. **3** The turbulence in water flow in the river is not considered. **4** The direction of water flow is normal to the pipe.

Properties The density and dynamic viscosity of water at 70°F are ρ = 62.30 lbm/ft^3 and μ = 2.36 lbm/ft·h = 6.556×10^{-4} lbm/ft·s.

Analysis Noting that D = 1.2 in = 0.1 ft, the Reynolds number for flow over the pipe is

$$\text{Re} = \frac{VD}{\nu} = \frac{\rho VD}{\mu} = \frac{(62.30\ \text{lbm/ft}^3)(10\ \text{ft/s})(0.1\ \text{ft})}{6.556\times10^{-4}\ \text{lbm/ft}\cdot\text{s}} = 9.50\times10^4$$

The drag coefficient corresponding to this value is, from Fig. 11-34, C_D = 1.1. Also, the frontal area for flow past a cylinder is $A = LD$. Then the drag force acting on the cylinder becomes

$$F_D = C_D A\frac{\rho V^2}{2} = 1.1\times(140\times0.1\ \text{ft}^2)\frac{(62.30\ \text{lbm/ft}^3)(10\ \text{ft/s})^2}{2}\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = \mathbf{1490\ lbf}$$

Discussion Note that this force is equivalent to the weight of a 1490 lbm mass. Therefore, the drag force the river exerts on the pipe is equivalent to hanging a mass of 1490 lbm on the pipe supported at its ends 70 ft apart. The necessary precautions should be taken if the pipe cannot support this force. Also, the fluctuations in water flow may reduce the drag coefficients by inducing turbulence and delaying flow separation.

11-68

Solution Dust particles that are unsettled during high winds rise to a specified height, and start falling back when things calm down. The time it takes for the dust particles to fall back to the ground and their velocity are to be determined using Stokes law.

Assumptions **1** The Reynolds number is low (at the order of 1) so that Stokes law is applicable (to be verified). **2** The atmosphere is calm during fall back (no winds or drafts). **3** The initial transient period during which the dust particle accelerates to its terminal velocity is negligible. **4** The buoyancy force applied by air to the dust particle is negligible since $\rho_{air} << \rho_{dust}$ (besides, the uncertainty in the density of dust is greater than the density of air). (We will solve this problem without utilizing this assumption for generality).

Properties The density of dust is given to be ρ_s = 1.6 g/cm^3 = 1600 kg/m^3. The density and dynamic viscosity of air at 1 atm and 30°C are ρ_f = 1.164 kg/m^3 and μ = 1.872×10^{-5} kg/m·s.

Analysis The terminal velocity of a free falling object is reached when the drag force equals the weight of the solid object less the buoyancy force applied by the surrounding fluid,

$$F_D = W - F_B \quad \text{where} \quad F_D = 3\pi\mu VD \text{ (Stokes law)}, \quad W = \rho_s g \boldsymbol{V}, \quad \text{and} \quad F_B = \rho_f g \boldsymbol{V}$$

Here $\boldsymbol{V} = \pi D^3/6$ is the volume of the sphere. Substituting,

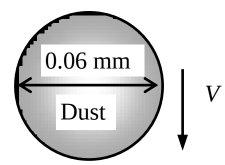
$$3\pi\mu VD = \rho_s g \boldsymbol{V} - \rho_f g \boldsymbol{V} \quad \rightarrow \quad 3\pi\mu VD = (\rho_s - \rho_f) g \frac{\pi D^3}{6}$$

Solving for the velocity V and substituting the numerical values, the terminal velocity is determined to be

$$V = \frac{gD^2(\rho_s - \rho_f)}{18\mu} = \frac{(9.81\text{ m/s}^2)(6\times10^{-5}\text{ m})^2(1600 - 1.164)\text{ kg/m}^3}{18(1.872\times10^{-5}\text{ kg/m}\cdot\text{s})} = 0.1676\text{ m/s} \cong \mathbf{0.168\ m/s}$$

Then the time it takes for the dust particle to travel 200 m at this velocity becomes

$$\Delta t = \frac{L}{V} = \frac{200\text{ m}}{0.1676\text{ m/s}} = 1194\text{ s} = \mathbf{19.9\ min}$$



The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(1.164\text{ kg/m}^3)(0.1676\text{ m/s})(6\times10^{-5}\text{ m})}{1.872\times10^{-5}\text{ kg}\cdot\text{m/s}} = 0.625$$

which is in the order of 1. Therefore, the creeping flow idealization and thus Stokes law is applicable.

Discussion Note that the dust particle reaches a terminal velocity of 0.168 m/s, and it takes about 20 min to fall back to the ground. The presence of drafts in air may significantly increase the settling time.

11-69

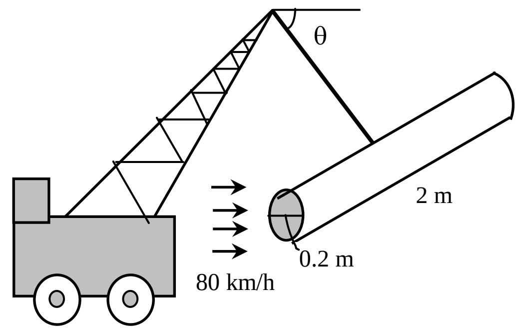
Solution A cylindrical log suspended by a crane is subjected to normal winds. The angular displacement of the log and the tension on the cable are to be determined.

Assumptions **1** The surfaces of the log are smooth so that Fig. 11-34 can be used to determine the drag coefficient (not a realistic assumption). **2** Air flow in the wind is steady and incompressible. **3** The turbulence in the wind is not considered. **4**The direction of wind is normal to the log, which always remains horizontal. **5** The end effects of the log are negligible. **6** The weight of the cable and the drag acting on it are negligible. **7** Air is an ideal gas.

Properties The dynamic viscosity of air at 5°C (independent of pressure) is μ = 1.754×10^{-5} kg/m·s. Then the density and kinematic viscosity of air are calculated to be

$$\rho = \frac{P}{RT} = \frac{88\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(278\text{ K})} = 1.103\text{ kg/m}^3$$

$$\nu = \frac{\mu}{\rho} = \frac{1.754\times10^{-5}\text{ kg/m}\cdot\text{s}}{1.103\text{ kg/m}^3} = 1.590\times10^{-5}\text{ m}^2\text{/s}$$



Analysis Noting that D = 0.2 m and 1 m/s = 3.6 km/h, the Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(40/3.6\text{ m/s})(0.2\text{ m})}{1.590\times10^{-5}\text{ m}^2\text{/s}} = 1.398\times10^5$$

The drag coefficient corresponding to this value is, from Fig. 11-34, C_D = 1.2. Also, the frontal area for flow past a cylinder is $A = LD$. Then the total drag force acting on the log becomes

$$F_D = C_D A\frac{\rho V^2}{2} = 1.2(2\times0.2\text{ m}^2)\frac{(1.103\text{ kg/m}^3)(40/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 32.7\text{ N}$$

The weight of the log is

$$W = mg = \rho g \mathbf{V} = \rho g\frac{\pi D^2 L}{4} = (513\text{ kg/m}^3)(9.81\text{ m/s}^2)\frac{\pi(0.2\text{ m})^2(2\text{ m})}{4}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 316\text{ N}$$

Then the resultant force acting on the log and the angle it makes with the horizontal become

$$F_{\log} = R = \sqrt{W^2 + F_D^2} = \sqrt{32.7^2 + 316^2} = \mathbf{318\,N}$$

$$\tan\theta = \frac{W}{F_D} = \frac{316}{32.7} = 9.66 \quad\rightarrow\quad \theta = \mathbf{84}°$$

Drawing a free body diagram of the log and doing a force balance will show that the magnitude of the tension on the cable must be equal to the resultant force acting on the log. Therefore, the tension on the cable is 318 N and the cable makes 84° with the horizontal.

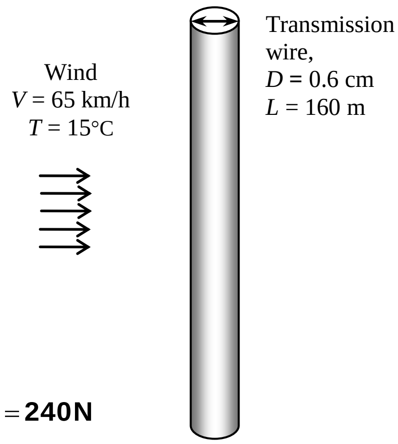
Discussion Note that the wind in this case has rotated the cable by 6° from its vertical position, and increased the tension action on it somewhat. At very high wind speeds, the increase in the cable tension can be very significant, and wind loading must always be considered in bodies exposed to high winds.

11-70

Solution Wind is blowing across the wire of a transmission line. The drag force exerted on the wire by the wind is to be determined.

Assumptions **1** The wire surfaces are smooth so that Fig. 11-34 can be used to determine the drag coefficient. **2** Air flow in the wind is steady and incompressible. **3** The turbulence in the wind is not considered. **4**The direction of wind is normal to the wire.

Properties The density and kinematic viscosity of air at 1 atm and 15°C are $\rho = 1.225$ kg/m^3 and $\nu = 1.470\times10^{-5}$ m^2/s.



Analysis Noting that D = 0.006 m and 1 m/s = 3.6 km/h, the Reynolds number for the flow is

$$\text{Re} = \frac{VD}{\nu} = \frac{(65/3.6\text{ m/s})(0.006\text{ m})}{1.470\times10^{-5}\text{ m}^2\text{/s}} = 7.370\times10^3$$

The drag coefficient corresponding to this value is, from Fig. 11-34, C_D = 1.25. Also, the frontal area for flow past a cylinder is $A = LD$. Then the drag force becomes

$$F_D = C_D A\frac{\rho V^2}{2} = 1.25(160\times0.006\text{ m}^2)\frac{(1.225\text{ kg/m}^3)(65/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{240 N}$$

Therefore, the drag force acting on the wire is 240 N, which is equivalent to the weight of about 24 kg mass hanging on the wire.

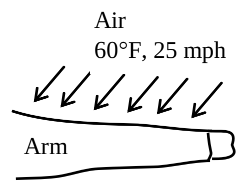
Discussion It should be kept in mind that wind turbulence may reduce the drag coefficients by inducing turbulence and delaying flow separation.

11-71E

Solution A person extends his uncovered arms into the windy air outside. The drag force exerted on both arms by the wind is to be determined.

Assumptions **1** The surfaces of the arms are smooth so that Fig. 11-34 can be used to determine the drag coefficient. **2** Air flow in the wind is steady and incompressible. **3** The turbulence in the wind is not considered. **4**The direction of wind is normal to the arms. **5** The arms can be treated as 2-ft-long and 4-in.-diameter cylinders with negligible end effects.

Properties The density and kinematic viscosity of air at 1 atm and 60°F are $\rho = 0.07633$ lbm/ft^3 and $\nu = 0.5718$ ft^2/h = 1.588×10^{-4} ft^2/s.



Analysis Noting that D = 4 in = 0.3333 ft and 1 mph = 1.4667 ft/s, the Reynolds number for flow over the arm is

$$\text{Re} = \frac{VD}{\nu} = \frac{(25\times1.4667\text{ ft/s})(0.3333\text{ ft})}{1.588\times10^{-4}\text{ ft}^2\text{/s}} = 7.697\times10^4$$

The drag coefficient corresponding to this value is, from Fig. 11-34, C_D = 1.0. Also, the frontal area for flow past a cylinder is $A = LD$. Then the total drag force acting on *both* arms becomes

$$F_D = C_D A\frac{\rho V^2}{2} = 1.0\times(2\times2\times0.3333\text{ ft}^2)\frac{(0.07633\text{ lbm/ft}^3)(25\times1.4667\text{ ft/s})^2}{2}\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{2.12 lbf}$$

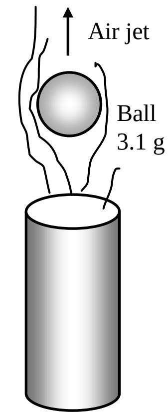
Discussion Note that this force is equivalent to the weight of 1 lbm mass. Therefore, the drag force the wind exerts on the arms of this person is equivalent to hanging 0.5 lbm of mass on each arm. Also, it should be kept in mind that the wind turbulence and the surface roughness may affect the calculated result significantly.

11-72

Solution A ping-pong ball is suspended in air by an upward air jet. The velocity of the air jet is to be determined, and the phenomenon that the ball returns to the center of the air jet after a disturbance is to be explained.

Assumptions **1** The surface of the ping-pong ball is smooth so that Fig. 11-34 can be used to determine the drag coefficient. **2** Air flow over the ball is steady and incompressible.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are ρ = 1.184 kg/m^3 and $\nu = 1.562\times10^{-5}$ m^2/s.



Analysis The terminal velocity of a free falling object is reached when the drag force equals the weight of the solid object less the buoyancy force applied by the fluid,

$$F_D = W - F_B \quad \text{where} \quad F_D = C_D A \frac{\rho_f V^2}{2}, \quad W = mg, \quad \text{and } F_B = \rho_f g \boldsymbol{V}$$

Here $A = \pi D^2/4$ is the frontal area and $V = \pi D^3/6$ is the volume of the sphere. Also,

$$W = mg = (0.0031\,\text{kg})(9.81\,\text{m/s}^2) = 0.03041\,\text{kg}\cdot\text{m/s}^2 = 0.03041\,\text{N}$$

$$F_B = \rho_f g \frac{\pi D^3}{6} = (1.184\,\text{kg/m}^3)(9.81\,\text{m/s}^2)\frac{\pi(0.042\,\text{m})^3}{6} = 0.000451\,\text{kg}\cdot\text{m/s}^2 = 0.000451\,\text{N}$$

Substituting and solving for V,

$$C_D \frac{\pi D^2}{4}\frac{\rho_f V^2}{2} = W - F_B \;\rightarrow V = \sqrt{\frac{8(W-F_B)}{\pi D^2 C_D \rho_f}} = \sqrt{\frac{8(0.03041 - 0.000451)\,\text{kg}\cdot\text{m/s}^2)}{\pi(0.042\,\text{m})^2 C_D (1.184\,\text{kg/m}^3)}} \rightarrow V = \frac{6.044}{\sqrt{C_D}} \quad (1)$$

The drag coefficient C_D is to be determined from Fig. 11-34, but it requires the Reynolds number which cannot be calculated since we do not know velocity. Therefore, the solution requires a trial-error approach. First we express the Reynolds number as

$$\text{Re} = \frac{VD}{\nu} = \frac{V(0.042\,\text{m})}{1.562\times10^{-5}\,\text{m}^2/\text{s}} \quad \rightarrow \quad \text{Re} = 2689V \qquad (2)$$

Now we choose a velocity in m/s, calculate the Re from Eq. 2, read the corresponding C_D from Fig. 11-34, and calculate V from Eq. 1. Repeat calculations until the assumed velocity matches the calculated velocity. With this approach the velocity of the fluid jet is determined to be

V = **9.2 m/s**

The corresponding Re and C_D values are Re = 24,700 and C_D = 0.43. Therefore, the ping-pong ball will remain suspended in the air jet when the air velocity reaches 9.2 m/s = 33.1 km/h.

Discussion

1 If the ball is pushed to the side by a finger, the ball will come back to the center of the jet (instead of falling off) due to the Bernoulli effect. In the core of the jet the velocity is higher, and thus the pressure is lower relative to a location away from the jet.

2 Note that this simple apparatus can be used to determine the drag coefficients of certain object by simply measuring the air velocity, which is easy to do.

3 This problem can also be solved roughly by taking C_D = 0.5 from Table 11-2 for a sphere in laminar flow, and then verifying that the flow is laminar.

Lift

11-73C

Solution We are to discuss why the contribution of viscous effects to lift of airfoils is usually negligible.

Analysis The contribution of viscous effects to lift is usually negligible for airfoils since **the wall shear is nearly parallel to the surfaces of such devices and thus nearly normal to the direction of lift**.

Discussion However, viscous effects *are* extremely important for airfoils at high angles of attack, since the viscous effects near the wall (in the boundary layer) cause the flow to separate and the airfoil to stall, losing significant lift.

11-74C

Solution We are to discuss the lift and drag on a symmetrical airfoil at 5° angle of attack.

Analysis When air flows past a symmetrical airfoil at an angle of attack of 5°, **both the (*a*) lift and (*b*) drag acting on the airfoil are nonzero**.

Discussion Because of the lack of symmetry with respect to the free-stream flow, the flow is different on the top and bottom surfaces of the airfoil, leading to lift. There is drag too, just as there is drag even at zero angle of attack.

11-75C

Solution We are to define and discuss stall.

Analysis The **decrease of lift with an increase in the angle of attack** is called *stall*. When the flow separates over nearly the entire upper half of the airfoil, the lift is reduced dramatically (the separation point is near the leading edge). Stall is caused by **flow separation and the formation of a wide wake region over the top surface of the airfoil**. Commercial aircraft are not allowed to fly at velocities near the stall velocity **for safety reasons**. Airfoils stall at high angles of attack (flow cannot negotiate the curve around the leading edge). **If a plane stalls, it loses mush of its lift, and it can crash**.

Discussion At angles of attack above the stall angle, the drag also increases significantly.

11-76C

Solution We are to discuss the lift and drag on a nonsymmetrical airfoil at zero angle of attack.

Analysis When air flows past a *nonsymmetrical* airfoil at zero angle of attack, **both the (*a*) lift and (*b*) drag acting on the airfoil are nonzero**.

Discussion Because of the lack of symmetry, the flow is different on the top and bottom surfaces of the airfoil, leading to lift. There is drag too, just as there is drag even on a symmetrical airfoil.

11-77C

Solution We are to discuss the lift and drag on a symmetrical airfoil at zero angle of attack.

Analysis When air flows past a *symmetrical* airfoil at zero angle of attack, (*a*) **the lift is zero**, but (*b*) **the drag acting on the airfoil is nonzero**.

Discussion In this case, because of symmetry, there is no lift, but there is still skin friction drag, along with a small amount of pressure drag.

11-78C

Solution We are to discuss which increases at a greater rate – lift or drag – with increasing angle of attack.

Analysis Both the lift and the drag of an airfoil increase with an increase in the angle of attack, but in general, **the lift increases at a much higher rate than does the drag**.

Discussion In other words, the lift-to-drag ratio increases with increasing angle of attack – at least up to the stall angle.

11-79C

Solution We are to why flaps are used on aircraft during takeoff and landing.

Analysis Flaps are used at the leading and trailing edges of the wings of large aircraft during takeoff and landing to alter the shape of the wings to **maximize lift and to enable the aircraft to land or takeoff at low speeds**. **An aircraft can take off or land without flaps, but it can do so at very high velocities**, which is undesirable during takeoff and landing.

Discussion In simple terms, the planform area of the wing increases as the flaps are deployed. Thus, even if the lift coefficient were to remain constant, the actual lift would still increase. In fact, however, flaps *increase the lift coefficient* as well, leading to even further increases in lift. Lower takeoff and landing speeds lead to shorter runway length requirements.

11-80C

Solution We are to discuss the lift on a spinning and non-spinning ball.

Analysis When air is flowing past a spherical ball, **the lift exerted on the ball is zero if the ball is not spinning**, and **it is nonzero if the ball is spinning about an axis normal to the free stream velocity** (no lift is generated if the ball is spinning about an axis parallel to the free stream velocity).

Discussion In the parallel spinning case, however, a *side force* would be generated (e.g., a curve ball).

11-81C

Solution We are to discuss the effect of wing tip vortices on drag and lift.

Analysis The effect of wing tip vortices is to **increase drag (induced drag) and to decrease lift**. This effect is also due to the *downwash*, which **causes an effectively smaller angle of attack**.

Discussion Induced drag is a three-dimensional effect; there is no induced drag on a 2-D airfoil since there are no tips.

11-82C

Solution We are to discuss induced drag and how to minimize it.

Analysis *Induced drag* is **the additional drag caused by the tip vortices**. The tip vortices have a lot of kinetic energy, all of which is wasted and is ultimately dissipated as heat in the air downstream. **Induced drag can be reduced by using long and narrow wings, and by modifying the geometry of the wing tips**.

Discussion Birds are designed with feathers that fan out at the tips of their wings in order to reduce induced drag.

11-83C

Solution We are to explain why some airplane wings have endplates or winglets.

Analysis *Endplates* or *winglets* are added at the tips of airplane wings **to reduce induced drag**. In short, the winglets disrupt flow from the high pressure lower part of the wing to the low pressure upper part of the wing, thereby reducing the strength of the tip vortices.

Discussion Comparing a wing with and without winglets, the one with winglets can produce the same lift with less drag, thereby saving fuel or increasing range. Winglets are especially useful in glider planes because high lift with small drag is critical to their operation (and safety!).

11-84C

Solution We are to discuss how flaps affect the lift and drag of airplane wings.

Analysis Flaps **increase both the lift and the drag of the wings**. But the increase in drag during takeoff and landing is not much of a concern because of the relatively short time periods involved. This is the penalty we pay willingly to take off and land at safe speeds.

Discussion Note, however, that the engine must operate at nearly full power during takeoff to overcome the large drag.

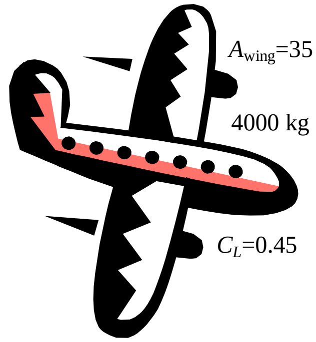
11-85

Solution The wing area, lift coefficient at takeoff settings, the cruising drag coefficient, and total mass of a small aircraft are given. The takeoff speed, the wing loading, and the required power to maintain a constant cruising speed are to be determined.

Assumptions **1** Standard atmospheric conditions exist. **2** The drag and lift produced by parts of the plane other than the wings are not considered.

Properties The density of standard air at sea level is ρ= 1.225 kg/m^3.

Analysis (*a*) An aircraft will takeoff when lift equals the total weight. Therefore,



$$W = F_L \quad \rightarrow \quad W = \tfrac{1}{2} C_L \rho V^2 A \quad \rightarrow \quad V = \sqrt{\frac{2W}{\rho C_L A}}$$

Substituting, the takeoff speed is determined to be

$$V_{\text{takeoff}} = \sqrt{\frac{2mg}{\rho C_{L,\text{takeoff}} A}} = \sqrt{\frac{2(4000\text{ kg})(9.81\text{ m/s}^2)}{(1.225\text{ kg/m}^3)(0.45)(35\text{ m}^2)}}$$
$$= 63.8\text{ m/s} = \mathbf{230\ km/h}$$

(*b*) Wing loading is the average lift per unit planform area, which is equivalent to the ratio of the lift to the planform area of the wings since the lift generated during steady cruising is equal to the weight of the aircraft. Therefore,

$$F_{\text{loading}} = \frac{F_L}{A} = \frac{W}{A} = \frac{(4000\text{ kg})(9.81\text{ m/s}^2)}{35\text{ m}^2} = \mathbf{1121\ N/m^2}$$

(*c*) When the aircraft is cruising steadily at a constant altitude, the net force acting on the aircraft is zero, and thus thrust provided by the engines must be equal to the drag force, which is

$$F_D = C_D A \frac{\rho V^2}{2} = (0.035)(35\text{ m}^2)\frac{(1.225\text{ kg/m}^3)(300/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 5.211\text{ kN}$$

Noting that power is force times velocity, the propulsive power required to overcome this drag is equal to the thrust times the cruising velocity,

$$\text{Power} = \text{Thrust} \times \text{Velocity} = F_D V = (5.211\text{ kN})(300/3.6\text{ m/s})\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right) = \mathbf{434\ kW}$$

Therefore, the engines must supply 434 kW of propulsive power to overcome the drag during cruising.

Discussion The power determined above is the power to overcome the drag that acts on the wings only, and does not include the drag that acts on the remaining parts of the aircraft (the fuselage, the tail, etc). Therefore, the total power required during cruising will be greater. The required rate of energy input can be determined by dividing the propulsive power by the propulsive efficiency.

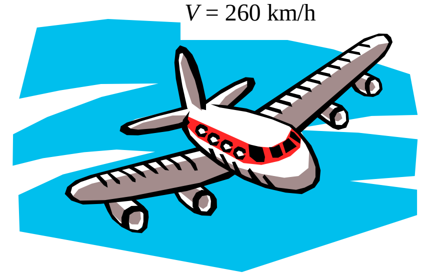
11-86

Solution The takeoff speed of an aircraft when it is fully loaded is given. The required takeoff speed when the weight of the aircraft is increased by 10% as a result of overloading is to be determined.

Assumptions **1** The atmospheric conditions (and thus the properties of air) remain the same. **2** The settings of the plane during takeoff are maintained the same so that the lift coefficient of the plane remains the same.

Analysis An aircraft will takeoff when lift equals the total weight. Therefore,

$$W = F_L \quad \rightarrow \quad W = \tfrac{1}{2} C_L \rho V^2 A \quad \rightarrow \quad V = \sqrt{\frac{2W}{\rho C_L A}}$$



We note that the takeoff velocity is proportional to the square root of the weight of the aircraft. When the density, lift coefficient, and area remain constant, the ratio of the velocities of the overloaded and fully loaded aircraft becomes

$$\frac{V_2}{V_1} = \frac{\sqrt{2W_2 / \rho C_L A}}{\sqrt{2W_1 / \rho C_L A}} = \frac{\sqrt{W_2}}{\sqrt{W_1}} \qquad \rightarrow \qquad V_2 = V_1 \sqrt{\frac{W_2}{W_1}}$$

Substituting, the takeoff velocity of the overloaded aircraft is determined to be

$$V_2 = V_1 \sqrt{\frac{1.2W_1}{W_1}} = (260 \text{ km/h})\sqrt{1.1} = \mathbf{273\,km/h}$$

Discussion A similar analysis can be performed for the effect of the variations in density, lift coefficient, and planform area on the takeoff velocity.

11-87

Solution The takeoff speed and takeoff time of an aircraft at sea level are given. The required takeoff speed, takeoff time, and the additional runway length required at a higher elevation are to be determined.

Assumptions **1** Standard atmospheric conditions exist. **2** The settings of the plane during takeoff are maintained the same so that the lift coefficient of the plane and the planform area remain constant. **3** The acceleration of the aircraft during takeoff remains constant.

Properties The density of standard air is ρ_1 = 1.225 kg/m^3 at sea level, and ρ_2 = 1.048 kg/m^3 at 1600 m altitude.

Analysis (*a*) An aircraft will takeoff when lift equals the total weight. Therefore,

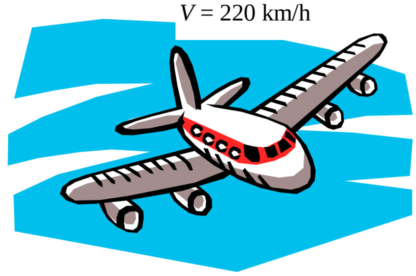
$$W = F_L \quad \rightarrow \quad W = \tfrac{1}{2} C_L \rho V^2 A \quad \rightarrow \quad V = \sqrt{\frac{2W}{\rho C_L A}}$$

We note that the takeoff speed is inversely proportional to the square root of air density. When the weight, lift coefficient, and area remain constant, the ratio of the speeds of the aircraft at high altitude and at sea level becomes

$$\frac{V_2}{V_1} = \frac{\sqrt{2W / \rho_2 C_L A}}{\sqrt{2W / \rho_1 C_L A}} = \frac{\sqrt{\rho_1}}{\sqrt{\rho_2}} \quad \rightarrow \quad V_2 = V_1 \sqrt{\frac{\rho_1}{\rho_2}} = (220\text{ km/h})\sqrt{\frac{1.225}{1.048}} = \mathbf{238\ km/h}$$

Therefore, the takeoff velocity of the aircraft at higher altitude is 238 km/h.

(*b*) The acceleration of the aircraft at sea level is



$$a = \frac{\Delta V}{\Delta t} = \frac{220\text{ km/h} - 0}{15\text{ s}}\left(\frac{1\text{ m/s}}{3.6\text{ km/h}}\right) = 4.074\text{ m/s}^2$$

which is assumed to be constant both at sea level and the higher altitude. Then the takeoff time at the higher altitude becomes

$$a = \frac{\Delta V}{\Delta t} \quad \rightarrow \quad \Delta t = \frac{\Delta V}{a} = \frac{238\text{ km/h} - 0}{4.074\text{ m/s}^2}\left(\frac{1\text{ m/s}}{3.6\text{ km/h}}\right) = \mathbf{16.2\ s}$$

(*c*) The additional runway length is determined by calculating the distance traveled during takeoff for both cases, and taking their difference:

$$L_1 = \tfrac{1}{2} a t_1^2 = \tfrac{1}{2}(4.074\text{ m/s}^2)(15\text{ s})^2 = 458\text{ m}$$

$$L_2 = \tfrac{1}{2} a t_2^2 = \tfrac{1}{2}(4.074\text{ m/s}^2)(16.2\text{ s})^2 = 535\text{ m}$$

$$\Delta L = L_2 - L_1 = 535 - 458 = \mathbf{77\ m}$$

Discussion Note that altitude has a significant effect on the length of the runways, and it should be a major consideration on the design of airports. It is interesting that a 1.2 second increase in takeoff time increases the required runway length by about 100 m.

11-88E

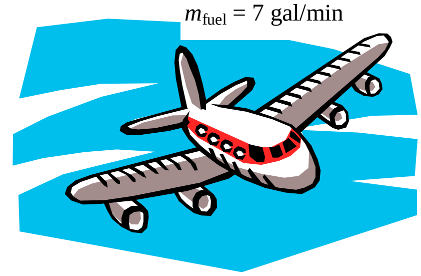
Solution The rate of fuel consumption of an aircraft while flying at a low altitude is given. The rate of fuel consumption at a higher altitude is to be determined for the same flight velocity.

Assumptions **1** Standard atmospheric conditions exist. **2** The settings of the plane during takeoff are maintained the same so that the drag coefficient of the plane and the planform area remain constant. **3** The velocity of the aircraft and the propulsive efficiency remain constant. **4** The fuel is used primarily to provide propulsive power to overcome drag, and thus the energy consumed by auxiliary equipment (lights, etc) is negligible.

Properties The density of standard air is ρ_1 = 0.05648 lbm/ft^3 at 10,000 ft, and ρ_2 = 0.02866 lbm/ft^3 at 30,000 ft altitude.

Analysis When an aircraft cruises steadily (zero acceleration) at a constant altitude, the net force acting on the aircraft is zero, and thus the thrust provided by the engines must be equal to the drag force. Also, power is force times velocity (distance per unit time), and thus the propulsive power required to overcome drag is equal to the thrust times the cruising velocity. Therefore,

$$\dot{W}_{\text{propulsive}} = \text{Thrust} \times V = F_D V = C_D A \frac{\rho V^2}{2} V = C_D A \frac{\rho V^3}{2}$$



The propulsive power is also equal to the product of the rate of fuel energy supplied (which is the rate of fuel consumption times the heating value of the fuel, $\dot{m}_{\text{fuel}}\text{HV}$) and the propulsive efficiency. Then,

$$\dot{W}_{\text{prop}} = \eta_{\text{prop}} \dot{m}_{\text{fuel}} \text{HV} \quad \rightarrow \quad C_D A \frac{\rho V^3}{2} = \eta_{\text{prop}} \dot{m}_{\text{fuel}} \text{HV}$$

We note that the rate of fuel consumption is proportional to the density of air. When the drag coefficient, the wing area, the velocity, and the propulsive efficiency remain constant, the ratio of the rates of fuel consumptions of the aircraft at high and low altitudes becomes

$$\frac{\dot{m}_{\text{fuel},2}}{\dot{m}_{\text{fuel},1}} = \frac{C_D A \rho_2 V^3 / 2\eta_{\text{prop}} \text{HV}}{C_D A \rho_1 V^3 / 2\eta_{\text{prop}} \text{HV}} = \frac{\rho_2}{\rho_1} \quad \rightarrow \quad \dot{m}_{\text{fuel},2} = \dot{m}_{\text{fuel},1} \frac{\rho_2}{\rho_1} = (7 \text{ gal/min}) \frac{0.02866}{0.05648} = \mathbf{3.55\ gal/min}$$

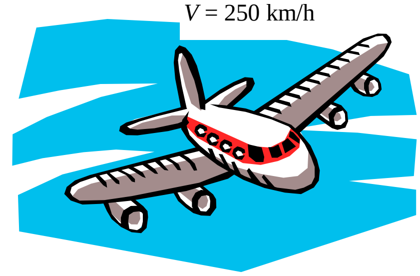
Discussion Note the fuel consumption drops by half when the aircraft flies at 30,000 ft instead of 10,000 ft altitude. Therefore, large passenger planes routinely fly at high altitudes (usually between 30,000 and 40,000 ft) to save fuel. This is especially the case for long flights.

11-89

Solution The takeoff speed of an aircraft when it is fully loaded is given. The required takeoff speed when the aircraft has 100 empty seats is to be determined.

Assumptions **1** The atmospheric conditions (and thus the properties of air) remain the same. **2** The settings of the plane during takeoff are maintained the same so that the lift coefficient of the plane remains the same. **3** A passenger with luggage has an average mass of 140 kg.

Analysis An aircraft will takeoff when lift equals the total weight. Therefore,



$$W = F_L \quad \rightarrow \quad W = \tfrac{1}{2} C_L \rho V^2 A \quad \rightarrow \quad V = \sqrt{\frac{2W}{\rho C_L A}}$$

We note that the takeoff velocity is proportional to the square root of the weight of the aircraft. When the density, lift coefficient, and wing area remain constant, the ratio of the velocities of the under-loaded and fully loaded aircraft becomes

$$\frac{V_2}{V_1} = \frac{\sqrt{2W_2 / \rho C_L A}}{\sqrt{2W_1 / \rho C_L A}} = \frac{\sqrt{W_2}}{\sqrt{W_1}} = \frac{\sqrt{m_2 g}}{\sqrt{m_1 g}} = \frac{\sqrt{m_2}}{\sqrt{m_1}} \qquad \rightarrow \qquad V_2 = V_1 \sqrt{\frac{m_2}{m_1}}$$

where $m_2 = m_1 - m_{\text{unusedcapacity}} = 400{,}000 \text{ kg} - (140 \text{ kg/passanger}) \times (100 \text{ passengers}) = 386{,}000 \text{ kg}$

Substituting, the takeoff velocity of the overloaded aircraft is determined to be

$$V_2 = V_1 \sqrt{\frac{m_2}{m_1}} = (250 \text{ km/h}) \sqrt{\frac{386{,}000}{400{,}000}} = \mathbf{246\,km/h}$$

Discussion Note that the effect of empty seats on the takeoff velocity of the aircraft is small. This is because the most weight of the aircraft is due to its empty weight (the aircraft itself rather than the passengers and their luggage.)

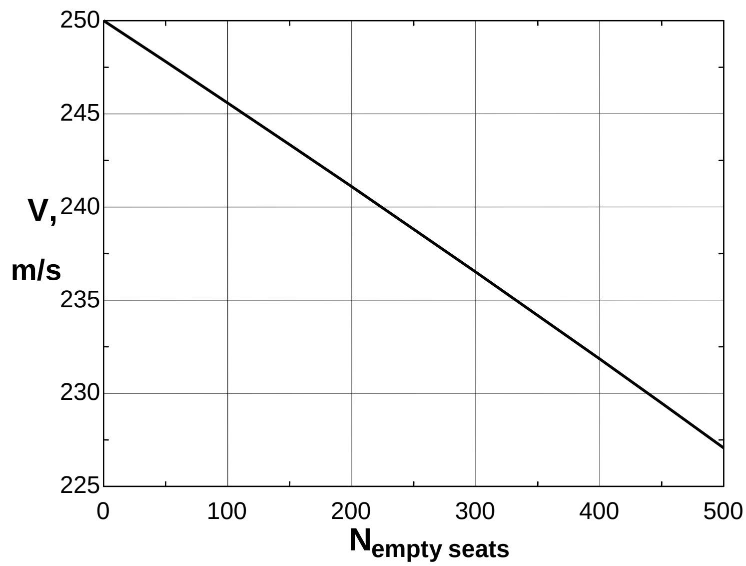
11-90

Solution The previous problem is reconsidered. The effect of empty passenger count on the takeoff speed of the aircraft as the number of empty seats varies from 0 to 500 in increments of 50 is to be investigated.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
m_passenger=140 "kg"
m1=400000 "kg"
m2=m1-N_empty*m_passenger
V1=250 "km/h"
V2=V1*SQRT(m2/m1)
```

Empty seats	$m_{airplane,1}$, kg	$m_{airplane,1}$, kg	$V_{takeoff,}$ m/s
0	400000	400000	250.0
50	400000	393000	247.8
100	400000	386000	245.6
150	400000	379000	243.3
200	400000	372000	241.1
250	400000	365000	238.8
300	400000	358000	236.5
350	400000	351000	234.2
400	400000	344000	231.8
450	400000	337000	229.5
500	400000	330000	227.1

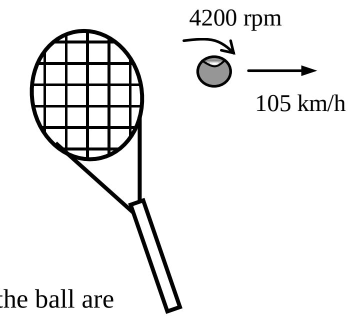


Discussion As expected, the takeoff speed decreases as the number of empty seats increases. On the scale plotted, the curve appears nearly linear, but it is not; the curve is actually a small portion of a square-root curve.

11-91

Solution A tennis ball is hit with a backspin. It is to be determined if the ball will fall or rise after being hit.

Assumptions **1** The outer surface of the ball is smooth enough for Fig. 11-53 to be applicable. **2** The ball is hit horizontally so that it starts its motion horizontally.



Properties The density and kinematic viscosity of air at 1 atm and 25°C are ρ = 1.184 kg/m^3 and ν = 1.562×10^{-5} m^2/s.

Analysis The ball is hit horizontally, and thus it would normally fall under the effect of gravity without the spin. The backspin will generate a lift, and the ball will rise if the lift is greater than the weight of the ball. The lift can be determined from

$$F_L = C_L A \frac{\rho V^2}{2}$$

where A is the frontal area of the ball, $A = \pi D^2/4$. The regular and angular velocities of the ball are

$$V = (105\text{ km/h})\left(\frac{1\text{ m/s}}{3.6\text{ km/h}}\right) = 29.17\text{ m/s} \qquad \text{and} \qquad \omega = (4200\text{ rev/min})\left(\frac{2\pi\text{ rad}}{1\text{ rev}}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right) = 440\text{ rad/s}$$

Then,

$$\frac{\omega D}{2V} = \frac{(440\text{ rad/s})(0.064\text{ m})}{2(29.17\text{ m/s})} = 0.483\text{ rad}$$

From Fig. 11-53, the lift coefficient corresponding to this value is C_L = 0.095. Then the lift acting on the ball is

$$F_L = (0.095)\frac{\pi(0.064\text{ m})^2}{4}\frac{(1.184\text{ kg/m}^3)(29.17\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 0.15\text{ N}$$

The weight of the ball is $W = mg = (0.057\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 0.56\text{ N}$

which is more than the lift. Therefore, the ball will **drop** under the combined effect of gravity and lift due to spinning after hitting, with a net force of 0.56 - 0.15 = 0.41 N.

Discussion The Reynolds number for this problem is $\text{Re}_L = \frac{VD}{\nu} = \frac{(29.17\text{ m/s})(0.064\text{ m})}{1.562\times10^{-5}\text{ m}^2\text{/s}} = 1.20\times10^5$, which is close enough to 6×10^4 for which Fig. 11-53 is prepared. Therefore, the result should be close enough to the actual answer.

11-92E

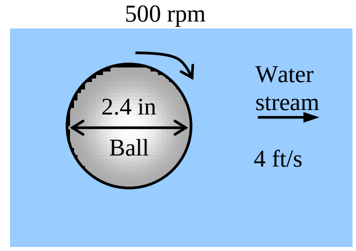
Solution A spinning ball is dropped into a water stream. The lift and drag forces acting on the ball are to be determined.

Assumptions **1** The outer surface of the ball is smooth enough for Fig. 11-53 to be applicable. **2** The ball is completely immersed in water.

Properties The density and dynamic viscosity of water at 60°F are ρ = 62.36 lbm/ft^3 and μ = 2.713 lbm/ft·h = 7.536×10^{-4} lbm/ft·s.

Analysis The drag and lift forces can be determined from

$$F_D = C_D A \frac{\rho V^2}{2} \quad \text{and} \quad F_L = C_L A \frac{\rho V^2}{2}$$



where A is the frontal area of the ball, which is $A = \pi D^2/4$, and D = 2.4/12 = 0.2 ft. The Reynolds number and the angular velocity of the ball are

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(62.36\ \text{lbm/ft}^3)(4\ \text{ft/s})(0.2\ \text{ft})}{7.536\times10^{-4}\ \text{lbm}\cdot\text{ft/s}} = 6.62\times10^4$$

$$\omega = (500\ \text{rev/min})\left(\frac{2\pi\ \text{rad}}{1\ \text{rev}}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = 52.4\ \text{rad/s}$$

and

$$\frac{\omega D}{2V} = \frac{(52.4\ \text{rad/s})(0.2\ \text{ft})}{2(4\ \text{ft/s})} = 1.31\ \text{rad}$$

From Fig. 11-53, the drag and lift coefficients corresponding to this value are C_D = 0.56 and C_L = 0.35. Then the drag and the lift acting on the ball are

$$F_D = (0.56)\frac{\pi(0.2\ \text{ft})^2}{4}\frac{(62.36\ \text{lbm/ft}^3)(4\ \text{ft/s})^2}{2}\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = \mathbf{0.27\ lbf}$$

$$F_L = (0.35)\frac{\pi(0.2\ \text{ft})^2}{4}\frac{(62.36\ \text{lbm/ft}^3)(4\ \text{ft/s})^2}{2}\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = \mathbf{0.17\ lbf}$$

Discussion The Reynolds number for this problem is 6.62×10^4 which is close enough to 6×10^4 for which Fig. 11-53 is prepared. Therefore, the result should be close enough to the actual answer.

11-93

Solution An airfoil has a given lift-to drag ratio at 0° angle of attack. The angle of attack that will raise this ratio to 80 is to be determined.

Analysis The ratio C_L/C_D for the given airfoil is plotted against the angle of attack in Fig. 11-43. The angle of attack corresponding to C_L/C_D = 80 is θ = **3°**.

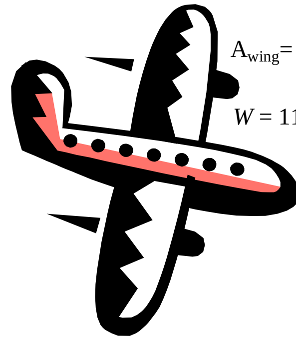
Discussion Note that different airfoils have different C_L/C_D vs. θ charts.

11-94

Solution The wings of a light plane resemble the NACA 23012 airfoil with no flaps. Using data for that airfoil, the takeoff speed at a specified angle of attack and the stall speed are to be determined.

Assumptions **1** Standard atmospheric conditions exist. **2** The drag and lift produced by parts of the plane other than the wings are not considered.

Properties The density of standard air at sea level is ρ = 1.225 kg/m^3. At an angle of attack of 5°, the lift and drag coefficients are read from Fig. 11-45 to be C_L = 0.6 and C_D = 0.015 [*Note*: Student values may differ significantly because these values are very hard to read from the plots]. The maximum lift coefficient is $C_{L,\max}$ = 1.52 and it occurs at an angle of attack of 15°.



Analysis An aircraft will takeoff when lift equals the total weight. Therefore,

$$W = F_L \quad \rightarrow \quad W = \tfrac{1}{2} C_L \rho V^2 A \quad \rightarrow \quad V = \sqrt{\frac{2W}{\rho C_L A}}$$

Substituting, the takeoff speed is determined to be

$$V_{\text{takeoff}} = \sqrt{\frac{2(11{,}000\text{ N})}{(1.225\text{ kg/m}^3)(0.6)(39\text{ m}^2)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)} = 27.70\text{ m/s} \cong \mathbf{99.7\ km/h}$$

since 1 m/s = 3.6 km/h. The stall velocity (the minimum takeoff velocity corresponding the stall conditions) is determined by using the maximum lift coefficient in the above equation,

$$V_{\min} = \sqrt{\frac{2W}{\rho C_{L,\text{takeoff}} A}} = \sqrt{\frac{2(11{,}000\text{ N})}{(1.225\text{ kg/m}^3)(1.52)(39\text{ m}^2)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)} = 17.41\text{ m/s} \cong \mathbf{62.7\ km/h}$$

Discussion The "safe" minimum velocity to avoid the stall region is obtained by multiplying the stall velocity by 1.2:

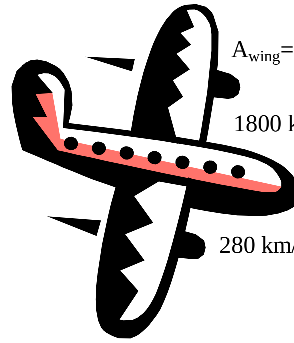
$$V_{\text{min,safe}} = 1.2V_{\min} = 1.2 \times 17.41\text{ m/s} = 20.9\text{ m/s} = 75.2\text{ km/h}$$

Note that the takeoff velocity decreased from 107 km/h at an angle of attack of 5° to 80.8 km/s under stall conditions with a safety margin.

11-95

Solution The total mass, wing area, cruising speed, and propulsive power of a small aircraft are given. The lift and drag coefficients of this airplane while cruising are to be determined.

Assumptions **1** Standard atmospheric conditions exist. **2** The drag and lift produced by parts of the plane other than the wings are not considered. **3** The fuel is used primarily to provide propulsive power to overcome drag, and thus the energy consumed by auxiliary equipment (lights, etc) is negligible.



Properties The density of standard air at an altitude of 4000 m is ρ= 0.819 kg/m^3.

Analysis Noting that power is force times velocity, the propulsive power required to overcome this drag is equal to the thrust times the cruising velocity. Also, when the aircraft is cruising steadily at a constant altitude, the net force acting on the aircraft is zero, and thus thrust provided by the engines must be equal to the drag force. Then,

$$\dot{W}_{\text{prop}} = \text{Thrust} \times \text{Velocity} = F_D V \quad \rightarrow \quad F_D = \frac{\dot{W}_{\text{prop}}}{V} = \frac{190 \text{ kW}}{280/3.6 \text{ m/s}} \left(\frac{1000 \text{ N} \cdot \text{m/s}}{1 \text{ kW}} \right) = 2443 \text{ N}$$

Then the drag coefficient becomes

$$F_D = C_D A \frac{\rho V^2}{2} \quad \rightarrow \quad C_D = \frac{2F_D}{\rho A V^2} = \frac{2(2443 \text{ N})}{(0.819 \text{ kg/m}^3)(42 \text{ m}^2)(280/3.6 \text{ m/s})^2} \left(\frac{1 \text{ kg} \cdot \text{m/s}^2}{1 \text{ N}} \right) = \mathbf{0.0235}$$

An aircraft cruises at constant altitude when lift equals the total weight. Therefore,

$$W = F_L = \tfrac{1}{2} C_L \rho V^2 A \quad \rightarrow \quad C_L = \frac{2W}{\rho V^2 A} = \frac{2(1800 \text{ kg})(9.81 \text{ m/s}^2)}{(0.819 \text{ kg/m}^3)(42 \text{ m}^2)(280/3.6 \text{ m/s})^2} = \mathbf{0.17}$$

Therefore, the drag and lift coefficients of this aircraft during cruising are 0.0235 and 0.17, respectively, with a C_L/C_D ratio of 7.2.

Discussion The drag and lift coefficient determined are for cruising conditions. The values of these coefficient can be very different during takeoff because of the angle of attack and the wing geometry.

11-96

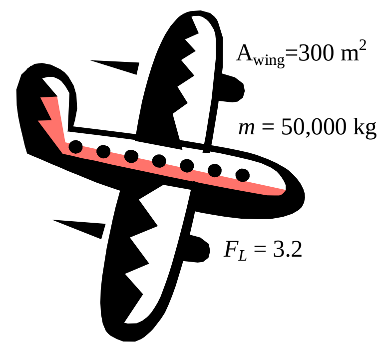
Solution The mass, wing area, the maximum (stall) lift coefficient, the cruising speed and the cruising drag coefficient of an airplane are given. The safe takeoff speed at sea level and the thrust that the engines must deliver during cruising are to be determined.

Assumptions **1** Standard atmospheric conditions exist **2** The drag and lift produced by parts of the plane other than the wings are not considered. **3** The takeoff speed is 20% over the stall speed. **4** The fuel is used primarily to provide propulsive power to overcome drag, and thus the energy consumed by auxiliary equipment (lights, etc) is negligible.

Properties The density of standard air is ρ_1 = 1.225 kg/m^3 at sea level, and ρ_2 = 0.312 kg/m^3 at 12,000 m altitude. The cruising drag coefficient is given to be C_D = 0.03. The maximum lift coefficient is given to be $C_{L,\max}$ = 3.2.

Analysis (*a*) An aircraft will takeoff when lift equals the total weight. Therefore,

$$W = F_L \quad \rightarrow \quad W = \tfrac{1}{2} C_L \rho V^2 A \quad \rightarrow \quad V = \sqrt{\frac{2W}{\rho C_L A}} = \sqrt{\frac{2mg}{\rho C_L A}}$$



The stall velocity (the minimum takeoff velocity corresponding the stall conditions) is determined by using the maximum lift coefficient in the above equation,

$$V_{\min} = \sqrt{\frac{2mg}{\rho_1 C_{L,\max} A}} = \sqrt{\frac{2(50{,}000 \text{ kg})(9.81 \text{ m/s}^2)}{(1.225 \text{ kg/m}^3)(3.2)(300 \text{ m}^2)}} = 28.9 \text{ m/s} = 104 \text{ km/h}$$

since 1 m/s = 3.6 km/h. Then the "safe" minimum velocity to avoid the stall region becomes

$$V_{\min,\text{safe}} = 1.2 V_{\min} = 1.2 \times (28.9 \text{ m/s}) = 34.7 \text{ m/s} = \mathbf{125\ km/h}$$

(*b*) When the aircraft cruises steadily at a constant altitude, the net force acting on the aircraft is zero, and thus the thrust provided by the engines must be equal to the drag force, which is

$$F_D = C_D A \frac{\rho_2 V^2}{2} = (0.03)(300 \text{ m}^2) \frac{(0.312 \text{ kg/m}^3)(700/3.6 \text{ m/s})^2}{2} \left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2} \right) = 53.08 \text{ kN}$$

Noting that power is force times velocity, the propulsive power required to overcome this drag is equal to the thrust times the cruising velocity,

$$\text{Power} = \text{Ttrust} \times \text{Velocity} = F_D V = (53.08 \text{ kN})(700/3.6 \text{ m/s}) \left(\frac{1 \text{ kW}}{1 \text{ kN}\cdot\text{m/s}} \right) = \mathbf{10{,}300 kW}$$

Therefore, the engines must supply 10,300 kW of propulsive power to overcome drag during cruising.

Discussion The power determined above is the power to overcome the drag that acts on the wings only, and does not include the drag that act on the remaining parts of the aircraft (the fuselage, the tail, etc). Therefore, the total power required during cruising will be greater. The required rate of energy input can be determined by dividing the propulsive power by the propulsive efficiency.

Review Problems

11-97

Solution A blimp connected to the ground by a rope is subjected to parallel winds. The rope tension when the wind is at a specified value is submarine is to be determined.

Assumptions **1** The blimp can be treated as an ellipsoid. **2** The wind is steady and turbulent, and blows parallel to the ground.

Properties The drag coefficient for an ellipsoid with L/D = 8/3 = 2.67 is C_D = 0.1 in turbulent flow (Table 11-2). We take the density of air to be 1.20 kg/m^3.

Analysis The frontal area of an ellipsoid is $A = \pi D^2/4$. Noting that 1 m/s = 3.6 km/h, the drag force acting on the blimp becomes

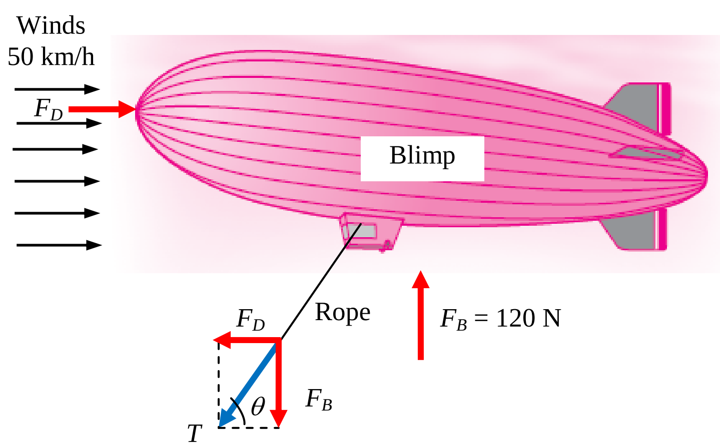
$$F_D = C_D A \frac{\rho V^2}{2} = (0.1)[\pi(3\,\text{m})^2/4]\frac{(1.20\,\text{kg/m}^3)(50/3.6\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = 81.8\,\text{N}$$

This is the horizontal component of the rope tension. The vertical component is equal to the rope tension when there is no wind, and is given to be F_B = 120 N. Knowing the horizontal and vertical components, the magnitude of rope tension becomes

$$T = \sqrt{F_D^2 + F_B^2} = \sqrt{(81.8\,\text{N}) + (120\,\text{N})} = \mathbf{145\,N} \quad \text{with} \quad \theta = \arctan\frac{F_B}{F_D} = \arctan\frac{120\,\text{N}}{81.8\,\text{N}} = \mathbf{55.7°}$$

where θ is the angle rope makes with the horizontal. Therefore

Discussion Note that the drag force acting on the blimp is proportional to the square of the blimp diameter. Therefore, the blimp diameter should be minimized to minimize the drag force.

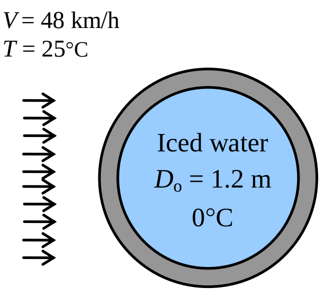


11-98

Solution A large spherical tank located outdoors is subjected to winds. The drag force exerted on the tank by the winds is to be determined.

Assumptions **1** The outer surfaces of the tank are smooth. **2** Air flow in the wind is steady and incompressible, and flow around the tank is uniform. **3** Turbulence in the wind is not considered. **4** The effect of any support bars on flow and drag is negligible.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are ρ = 1.184 kg/m^3 and ν = 1.562×10^{-5} m^2/s.



Analysis Noting that D = 1.2 m and 1 m/s = 3.6 km/h, the Reynolds number for the flow is

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{(48/3.6\ \mathrm{m/s})(1.2\ \mathrm{m})}{1.562\times 10^{-5}\ \mathrm{m^2/s}} = 1.024\times 10^{6}$$

The drag coefficient for a smooth sphere corresponding to this Reynolds number is, from Fig. 11-36, C_D = 0.14. Also, the frontal area for flow past a sphere is $A = \pi D^2/4$. Then the drag force becomes

$$F_D = C_D A \frac{\rho V^2}{2} = 0.14[\pi(1.2\ \mathrm{m})^2/4]\frac{(1.184\ \mathrm{kg/m^3})(48/3.6\ \mathrm{m/s})^2}{2}\left(\frac{1\ \mathrm{N}}{1\ \mathrm{kg\cdot m/s^2}}\right) = \mathbf{16.7\ N}$$

Discussion Note that the drag coefficient is very low in this case since the flow is turbulent (Re > 2×10^5). Also, it should be kept in mind that wind turbulence may affect the drag coefficient.

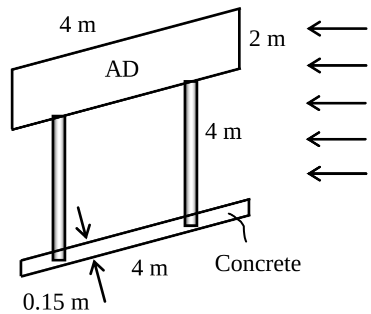
11-99

Solution A rectangular advertisement panel attached to a rectangular concrete block by two poles is to withstand high winds. For a given maximum wind speed, the maximum drag force on the panel and the poles, and the minimum length L of the concrete block for the panel to resist the winds are to be determined.

Assumptions **1** The flow of air is steady and incompressible. **2** The wind is normal to the panel (to check for the worst case). **3** The flow is turbulent so that the tabulated value of the drag coefficients can be used.

Properties In turbulent flow, the drag coefficient is $C_D = 0.3$ for a circular rod, and $C_D = 2.0$ for a thin rectangular plate (Table 11-2). The densities of air and concrete block are given to be $\rho = 1.30$ kg/m^3 and $\rho_c = 2300$ kg/m^3.

Analysis (*a*) The drag force acting on the panel is



$$F_{D,\text{panel}} = C_D A \frac{\rho V^2}{2}$$

$$= (2.0)(2\times 4\text{ m}^2)\frac{(1.30\text{ kg/m}^3)(150/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= \mathbf{18{,}000\,N}$$

(*b*) The drag force acting on each pole is

$$F_{D,\text{pole}} = C_D A \frac{\rho V^2}{2}$$

$$= (0.3)(0.05\times 4\text{ m}^2)\frac{(1.30\text{ kg/m}^3)(150/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= \mathbf{68\,N}$$

Therefore, the drag force acting on both poles is 68 × 2 = **136 N**. Note that the drag force acting on poles is negligible compared to the drag force acting on the panel.

(*c*) The weight of the concrete block is

$$W = mg = \rho g \boldsymbol{V} = (2300\text{ kg/m}^3)(9.81\text{ m/s}^2)(\text{L}\times 4\text{ m}\times 0.15\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 13{,}540L\ \text{ N}$$

Note that the resultant drag force on the panel passes through its center, the drag force on the pole passes through the center of the pole, and the weight of the panel passes through the center of the block. When the concrete block is first tipped, the wind-loaded side of the block will be lifted off the ground, and thus the entire reaction force from the ground will act on the other side. Taking the moment about this side and setting it equal to zero gives

$$\sum M = 0 \ \rightarrow\ F_{D,\text{panel}}\times(1+4+0.15) + F_{D,\text{pole}}\times(2+0.15) - W\times(L/2) = 0$$

Substituting and solving for L gives

$$18{,}000\times 5.15 + 136\times 2.15 - 13{,}540L\times L/2 = 0 \quad\rightarrow\quad \text{L} = 3.70\text{ m}$$

Therefore, the minimum length of the concrete block must be L = **3.70**.

Discussion This length appears to be large and impractical. It can be reduced to a more reasonable value by (*a*) increasing the height of the concrete block, (*b*) reducing the length of the poles (and thus the tipping moment), or (*c*) by attaching the concrete block to the ground (through long nails, for example).

11-100

Solution The bottom surface of a plastic boat is approximated as a flat surface. The friction drag exerted on the bottom surface of the boat by water and the power needed to overcome it are to be determined.

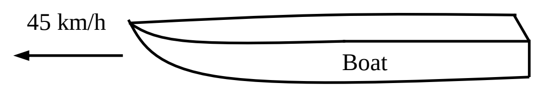
Assumptions **1** The flow is steady and incompressible. **2** The water is calm (no significant currents or waves). **3** The water flow is turbulent over the entire surface because of the constant agitation of the boat. **4** The bottom surface of the boat is a flat surface, and it is smooth.

Properties The density and dynamic viscosity of water at 15°C are ρ = 999.1 kg/m^3 and μ = 1.138×10^{-3} kg/m·s.

Analysis The Reynolds number at the end of the bottom surface of the boat is

$$\text{Re}_L = \frac{\rho VL}{\mu} = \frac{(999.1\,\text{kg/m}^3)(45/3.6\,\text{m/s})(2\,\text{m})}{1.138\times10^{-3}\,\text{kg/m}\cdot\text{s}} = 2.195\times10^7$$

The flow is assumed to be turbulent over the entire surface. Then the average friction coefficient and the drag force acting on the surface becomes



$$C_f = \frac{0.074}{\text{Re}_L^{1/5}} = \frac{0.074}{(2.195\times10^7)^{1/5}} = 0.002517$$

$$F_D = C_f A\frac{\rho V^2}{2} = (0.002517)[1.5\times 2\,\text{m}^2]\frac{(999.1\,\text{kg/m}^3)(45/3.6\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg.m/s}^2}\right) = 589.4\,\text{N} \cong \mathbf{589\,N}$$

Noting that power is force times velocity, the power needed to overcome this drag force is

$$\dot{W}_{\text{drag}} = F_D V = (589.4\,\text{N})(45/3.6\,\text{m/s})\left(\frac{1\,\text{kW}}{1000\,\text{N}\cdot\text{m/s}}\right) = \mathbf{7.37\,kW}$$

Discussion Note that the calculated drag force (and the power required to overcome it) is relatively small. This is not surprising since the drag force for blunt bodies (including those partially immersed in a liquid) is almost entirely due to pressure drag, and the friction drag is practically negligible compared to the pressure drag.

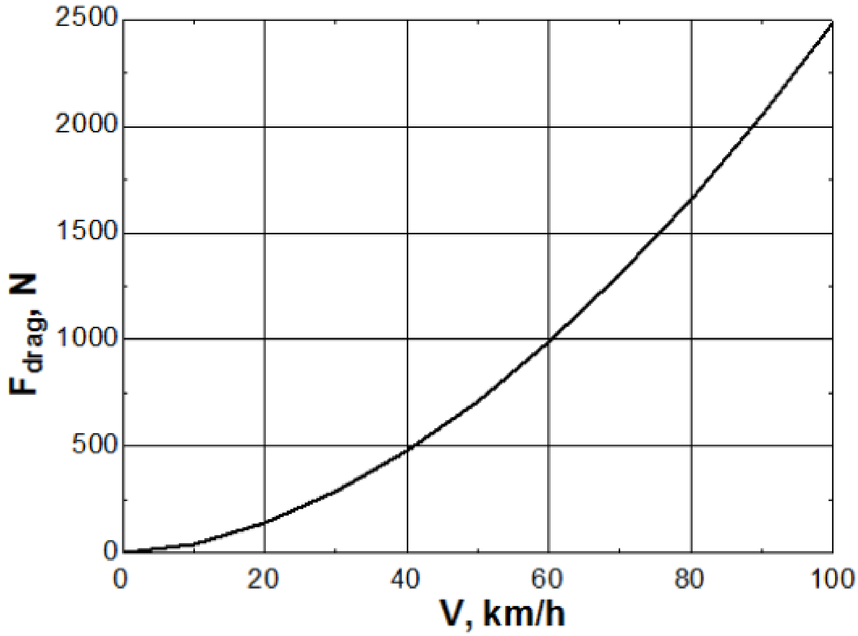
11-101

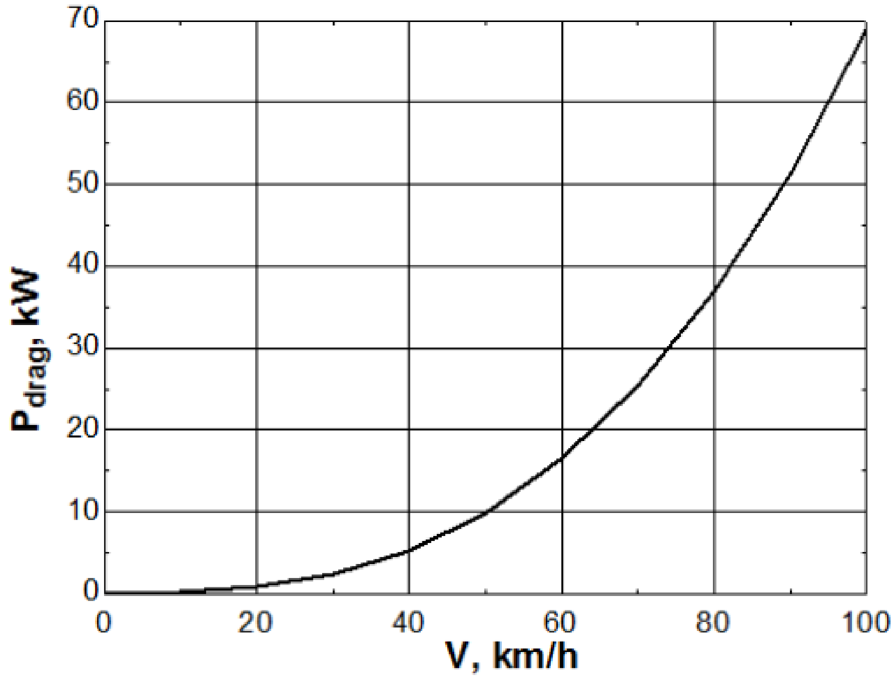
Solution The previous problem is reconsidered. The effect of boat speed on the drag force acting on the bottom surface of the boat and the power needed to overcome as the boat speed varies from 0 to 100 km/h in increments of 10 km/h is to be investigated.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
rho=999.1 "kg/m3"
mu=1.138E-3 "m2/s"
V=Vel/3.6 "m/s"
L=2 "m"
W=1.5 "m"
A=L*W

Re=rho*L*V/mu
Cf=0.074/Re^0.2
g=9.81 "m/s2"
F=Cf*A*(rho*V^2)/2 "N"
P_drag=F*V/1000 "kW"
```

V, km/h	Re	C_f	F_{drag}, N	P_{drag}, kW
0	0	0	0	0.0
10	4.877E+06	0.00340	39	0.1
20	9.755E+06	0.00296	137	0.8
30	1.463E+07	0.00273	284	2.4
40	1.951E+07	0.00258	477	5.3
50	2.439E+07	0.00246	713	9.9
60	2.926E+07	0.00238	989	16.5
70	3.414E+07	0.00230	1306	25.4
80	3.902E+07	0.00224	1661	36.9
90	4.390E+07	0.00219	2053	51.3
100	4.877E+07	0.00215	2481	68.9





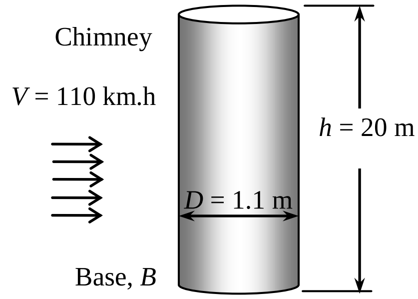
Discussion The curves look similar at first glance, but in fact F_{drag} increases like V^2, while F_{drag} increases like V^3.

11-102

Solution The chimney of a factory is subjected to high winds. The bending moment at the base of the chimney is to be determined.

Assumptions **1** The flow of air in the wind is steady, turbulent, and incompressible. **2** The ground effect on the wind and the drag coefficient is negligible (a crude approximation) so that the resultant drag force acts through the center of the side surface. **3** The edge effects are negligible and thus the chimney can be treated as a 2-D long cylinder.

Properties The drag coefficient for a long cylindrical bar in turbulent flow is 0.3 (Table 11-2). The density of air at 20°C and 1 atm is $\rho = 1.204$ kg/m^3.



Analysis Noting that the frontal area is DH and 1 m/s = 3.6 km/h, the drag force becomes

$$F_D = C_D A \frac{\rho V^2}{2}$$

$$= (0.3)[1.1 \times 20\text{ m}^2]\frac{(1.204\text{ kg/m}^3)(110/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$= 3710\text{ N}$$

The drag force acts through the mid height of the chimney. Noting that moment is force times moment arm, the wind-induced bending moment at the base of the chimney becomes

$$M_{\text{Bending}} = M_B = F_D \times h/2 = (3710\text{ N})(20/2\text{ m}) = \mathbf{3.71 \times 10^4\ N \cdot m}$$

Discussion Forces and moments caused by winds can be very high. Therefore, wind loading is an important consideration in the design of structures.

11-103E

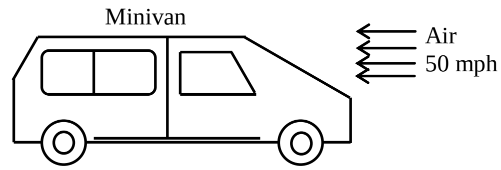
Solution The passenger compartment of a minivan is modeled as a rectangular box. The drag force acting on the top and the two side surfaces and the power needed to overcome it are to be determined.

Assumptions **1** The air flow is steady and incompressible. **2** The air flow over the exterior surfaces is turbulent because of constant agitation. **3** Air is an ideal gas. **4** The top and side surfaces of the minivan are flat and smooth (in reality they can be rough). **5** The atmospheric air is calm (no significant winds).

Properties The density and kinematic viscosity of air at 1 atm and 80°F are ρ = 0.07350 lbm/ft^3 and ν = 0.6110 ft^2/h = 1.697×10^{-4} ft^2/s.

Analysis The Reynolds number at the end of the top and side surfaces is

$$\mathrm{Re} = \frac{VL}{\nu} = \frac{[(50\times 1.4667)\ \mathrm{ft/s}](11\ \mathrm{ft})}{1.697\times 10^{-4}\ \mathrm{ft^2/s}} = 4.754\times 10^6$$



The air flow over the entire outer surface is assumed to be turbulent. Then the friction coefficient becomes

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} = \frac{0.074}{(4.754\times 10^6)^{1/5}} = 0.003418$$

The area of the top and side surfaces of the minivan is

$$A = A_{\mathrm{top}} + 2A_{\mathrm{side}} = 6\times 11 + 2\times 4.5\times 11 = 165\ \mathrm{ft^2}$$

Noting that the pressure drag is zero and thus $C_D = C_f$ for a plane surface, the drag force acting on these surfaces becomes

$$F_D = C_f A\frac{\rho V^2}{2} = 0.003418\times(165\ \mathrm{ft^2})\frac{(0.074\ \mathrm{lbm/ft^3})(50\times 1.4667\ \mathrm{ft/s})^2}{2}\left(\frac{1\ \mathrm{lbf}}{32.2\ \mathrm{lbm\cdot ft/s^2}}\right) = \mathbf{3.49\ lbf}$$

Noting that power is force times velocity, the power needed to overcome this drag force is

$$\dot{W}_{\mathrm{drag}} = F_D V = (3.49\ \mathrm{lbf})(50\times 1.4667\ \mathrm{ft/s})\left(\frac{1\ \mathrm{kW}}{737.56\ \mathrm{lbf\cdot ft/s}}\right) = \mathbf{0.347\ kW}$$

Discussion Note that the calculated drag force (and the power required to overcome it) is very small. This is not surprising since the drag force for blunt bodies is almost entirely due to pressure drag, and the friction drag is practically negligible compared to the pressure drag.

11-104E

Solution Cruising conditions of a passenger plane are given. The minimum safe landing and takeoff speeds with and without flaps, the angle of attack during cruising, and the power required are to be determined

Assumptions **1** The drag and lift produced by parts of the plane other than the wings are not considered. **2** The wings are assumed to be two-dimensional airfoil sections, and the tip effects are neglected. **4** The lift and drag characteristics of the wings can be approximated by NACA 23012 so that Fig. 11-45 is applicable.

Properties The densities of air are 0.075 lbm/ft^3 on the ground and 0.0208 lbm/ft^3 at cruising altitude. The maximum lift coefficients of the wings are 3.48 and 1.52 with and without flaps, respectively (Fig. 11-45).

V = 550 mph
m = 150,000 lbm
A_{wing} = 1800 m^2

Analysis (*a*) The weight and cruising speed of the airplane are

$$W = mg = (150{,}000\ \text{lbm})(32.2\ \text{ft/s}^2)\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = 150{,}000\ \text{lbf}$$

$$V = (550\ \text{mph})\left(\frac{1.4667\ \text{ft/s}}{1\ \text{mph}}\right) = 806.7\ \text{ft/s}$$

Minimum velocity corresponding the stall conditions with and without flaps are

$$V_{\min 1} = \sqrt{\frac{2W}{\rho C_{L,\max 1}A}} = \sqrt{\frac{2(150{,}000\ \text{lbf})}{(0.075\ \text{lbm/ft}^3)(1.52)(1800\ \text{ft}^2)}\left(\frac{32.2\ \text{lbm}\cdot\text{ft/s}^2}{1\ \text{lbf}}\right)} = 217\ \text{ft/s}$$

$$V_{\min 2} = \sqrt{\frac{2W}{\rho C_{L,\max 2}A}} = \sqrt{\frac{2(150{,}000\ \text{lbf})}{(0.075\ \text{lbm/ft}^3)(3.48)(1800\ \text{ft}^2)}\left(\frac{32.2\ \text{lbm}\cdot\text{ft/s}^2}{1\ \text{lbf}}\right)} = 143\ \text{ft/s}$$

The "safe" minimum velocities to avoid the stall region are obtained by multiplying these values by 1.2:

Without flaps: $V_{\min 1,\text{safe}} = 1.2V_{\min 1} = 1.2\times(217\ \text{ft/s}) = 260\ \text{ft/s} = \mathbf{178\ mph}$

With flaps: $V_{\min 2,\text{safe}} = 1.2V_{\min 2} = 1.2\times(143\ \text{ft/s}) = 172\ \text{ft/s} = \mathbf{117\ mph}$

since 1 mph = 1.4667 ft/s. Note that the use of flaps allows the plane to takeoff and land at considerably lower velocities, and thus at a shorter runway.

(*b*) When an aircraft is cruising steadily at a constant altitude, the lift must be equal to the weight of the aircraft, $F_L = W$. Then the lift coefficient is determined to be

$$C_L = \frac{F_L}{\frac{1}{2}\rho V^2 A} = \frac{150{,}000\ \text{lbf}}{\frac{1}{2}(0.0208\ \text{lbm/ft}^3)(806.7\ \text{ft/s})^2(1800\ \text{ft}^2)}\left(\frac{32.2\ \text{lbm}\cdot\text{ft/s}^2}{1\ \text{lbf}}\right) = 0.40$$

For the case of no flaps, the angle of attack corresponding to this value of C_L is determined from Fig. 11-45 to be about α = **3.5°**.

(*c*) When aircraft cruises steadily, the net force acting on the aircraft is zero, and thus thrust provided by the engines must be equal to the drag force. The drag coefficient corresponding to the cruising lift coefficient of 0.40 is C_D = 0.015 (Fig. 11-45). Then the drag force acting on the wings becomes

$$F_D = C_D A\frac{\rho V^2}{2} = (0.015)(1800\ \text{ft}^2)\frac{(0.0208\ \text{lbm/ft}^3)(806.7\ \text{ft/s})^2}{2}\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right) = 5675\ \text{lbf}$$

Noting that power is force times velocity (distance per unit time), the power required to overcome this drag is equal to the thrust times the cruising velocity,

$$\text{Power} = \text{Thrust}\times\text{Velocity} = F_D V = (5675\ \text{lbf})(806.7\ \text{ft/s})\left(\frac{1\ \text{kW}}{737.56\ \text{lbf}\cdot\text{ft/s}}\right) = \mathbf{6200\ kW}$$

Discussion Note that the engines must supply 6200 kW of power to overcome the drag during cruising. This is the power required to overcome the drag that acts on the wings only, and does not include the drag that acts on the remaining parts of the aircraft (the fuselage, the tail, etc).

11-105

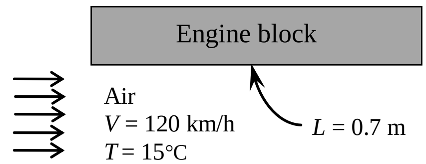
Solution An automotive engine is approximated as a rectangular block. The drag force acting on the bottom surface of the engine is to be determined.

Assumptions **1** The air flow is steady and incompressible. **2** Air is an ideal gas. **3** The atmospheric air is calm (no significant winds). **3** The air flow is turbulent over the entire surface because of the constant agitation of the engine block. **4** The bottom surface of the engine is a flat surface, and it is smooth (in reality it is quite rough because of the dirt collected on it).

Properties The density and kinematic viscosity of air at 1 atm and 15°C are $\rho = 1.225$ kg/m^3 and $\nu = 1.470\times10^{-5}$ m^2/s.

Analysis The Reynolds number at the end of the engine block is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(120/3.6\text{ m/s})(0.7\text{ m})}{1.470\times10^{-5}\text{ m}^2\text{/s}} = 1.587\times10^6$$



The flow is assumed to be turbulent over the entire surface. Then the average friction coefficient and the drag force acting on the surface becomes

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} = \frac{0.074}{(1.587\times10^6)^{1/5}} = 0.004257$$

$$F_D = C_f A\frac{\rho V^2}{2} = (0.004257)[0.6\times0.7\text{ m}^2]\frac{(1.225\text{ kg/m}^3)(120/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg.m/s}^2}\right) = \mathbf{1.22 N}$$

Discussion Note that the calculated drag force (and the power required to overcome it) is very small. This is not surprising since the drag force for blunt bodies is almost entirely due to pressure drag, and the friction drag is practically negligible compared to the pressure drag.

11-106

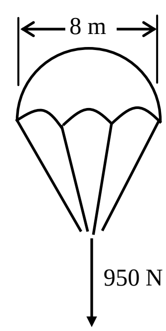
Solution The total weight of a paratrooper and its parachute is given. The terminal velocity of the paratrooper in air is to be determined.

Assumptions **1** The air flow over the parachute is turbulent so that the tabulated value of the drag coefficient can be used. **2** The variation of the air properties with altitude is negligible. **3** The buoyancy force applied by air to the person (and the parachute) is negligible because of the small volume occupied and the low air density.

Properties The density of air is given to be 1.20 kg/m^3. The drag coefficient of a parachute is $C_D = 1.3$.

Analysis The terminal velocity of a free falling object is reached when the drag force equals the weight of the solid object, less the buoyancy force applied by the fluid, which is negligible in this case,

$$F_D = W - F_B \quad \text{where} \quad F_D = C_D A\frac{\rho_f V^2}{2}, \quad W = mg = 950\text{ N}, \quad \text{and} \quad F_B \cong 0$$



where $A = \pi D^2/4$ is the frontal area. Substituting and simplifying,

$$C_D A\frac{\rho_f V^2}{2} = W \quad \rightarrow \quad C_D\frac{\pi D^2}{4}\frac{\rho_f V^2}{2} = W$$

Solving for V and substituting,

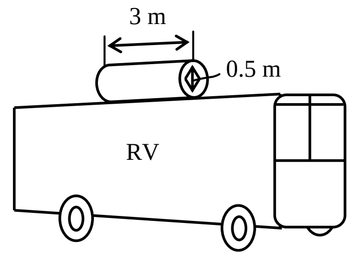
$$V = \sqrt{\frac{8W}{C_D\pi D^2\rho_f}} = \sqrt{\frac{8(950\text{ N})}{1.3\pi(8\text{ m})^2(1.20\text{ kg/m}^3)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)} = \mathbf{4.9 m/s}$$

Therefore, the velocity of the paratrooper will remain constant when it reaches the terminal velocity of 4.9 m/s = 18 km/h.

Discussion The simple analysis above gives us a rough value for the terminal velocity. A more accurate answer can be obtained by a more detailed (and complex) analysis by considering the variation of air density with altitude, and by considering the uncertainty in the drag coefficient.

11-107

Solution The water needs of a recreational vehicle (RV) are to be met by installing a cylindrical tank on top of the vehicle. The additional power requirements of the RV at a specified speed for two orientations of the tank are to be determined.



Assumptions **1** The flow of air is steady and incompressible. **2** The effect of the tank and the RV on the drag coefficient of each other is negligible (no interference). **3** The flow is turbulent so that the tabulated value of the drag coefficient can be used.

Properties The drag coefficient for a cylinder corresponding to L/D = 3/0.5 = 6 is C_D = 0.95 when the circular surfaces of the tank face the front and back, and C_D = 0.8 when the circular surfaces of the tank face the sides of the RV (Table 11-2). The density of air at the specified conditions is

$$\rho = \frac{P}{RT} = \frac{87\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg.K})(295\text{ K})} = 1.028\text{ kg/m}^3$$

Analysis (*a*) The drag force acting on the tank when the circular surfaces face the front and back is

$$F_D = C_D A\frac{\rho V^2}{2} = (0.95)[\pi(0.5\text{ m})^2/4]\frac{(1.028\text{ kg/m}^3)(80\text{ km/h})^2}{2}\left(\frac{1\text{ m/s}}{3.6\text{ km/h}}\right)^2\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 47.35\text{ N}$$

Noting that power is force times velocity, the amount of additional power needed to overcome this drag force is

$$\dot{W}_{\text{drag}} = F_D V = (47.35\text{ N})(80/3.6\text{ m/s})\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right) = \mathbf{1.05\ kW}$$

(*b*) The drag force acting on the tank when the circular surfaces face the sides of the RV is

$$F_D = C_D A\frac{\rho V^2}{2} = (0.8)[0.5\times 3\text{ m}^2]\frac{(1.028\text{ kg/m}^3)(80\text{ km/h})^2}{2}\left(\frac{1\text{ m/s}}{3.6\text{ km/h}}\right)^2\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 304.6\text{ N}$$

Then the additional power needed to overcome this drag force is

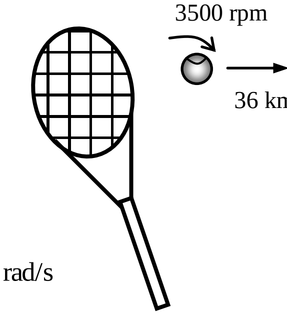
$$\dot{W}_{\text{drag}} = F_D V = (304.6\text{ N})(80/3.6\text{ m/s})\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right) = \mathbf{6.77\ kW}$$

Therefore, the additional power needed to overcome the drag caused by the tank is 1.05 kW and 6.77 W for the two orientations indicated.

Discussion Note that the additional power requirement is the lowest when the tank is installed such that its circular surfaces face the front and back of the RV. This is because the frontal area of the tank is minimum in this orientation.

11-108

Solution A smooth ball is moving at a specified velocity. The increase in the drag coefficient when the ball spins is to be determined.



Assumptions **1** The outer surface of the ball is smooth. **2** The air is calm (no winds or drafts).

Properties The density and kinematic viscosity of air at 1 atm and 25°C are ρ = 1.184 kg/m^3 and ν = 1.562×10^{-5} m^2/s.

Analysis Noting that *D* = 0.09 m and 1 m/s = 3.6 km/h, the regular and angular velocities of the ball are

$$V = (36\text{ km/h})\left(\frac{1\text{ m/s}}{3.6\text{ km/h}}\right) = 10\text{ m/s} \quad \text{and} \quad \omega = (3500\text{ rev/min})\left(\frac{2\pi\text{ rad}}{1\text{ rev}}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right) = 367\text{ rad/s}$$

From these values, we calculate the nondimensional rate of rotation and the Reynolds number:

$$\frac{\omega D}{2V} = \frac{(367\text{ rad/s})(0.09\text{ m})}{2(10\text{ m/s})} = 1.652\text{ rad} \quad \text{and} \quad \text{Re} = \frac{VD}{\nu} = \frac{(10\text{ m/s})(0.09\text{ m})}{1.562\times10^{-5}\text{ m}^2\text{/s}} = 5.762\times10^4$$

Then the drag coefficients for the ball with and without spin are determined from Figs. 11-36 and 11-53 to be:

Without spin: C_D = 0.50 (Fig. 11-36, smooth ball)
With spin: C_D = 0.58 (Fig. 11-53)

Then the increase in the drag coefficient due to spinning becomes

$$\text{Increase in } C_D = \frac{C_{D,\text{spin}} - C_{D,\text{nospin}}}{C_{D,\text{nospin}}} = \frac{0.58 - 0.50}{0.50} = 0.160$$

Therefore, **the drag coefficient in this case increases by about 16% because of spinning**.

Discussion Note that the Reynolds number for this problem is 5.762×10^4 which is close enough to 6×10^4 for which Fig. 11-53 is prepared. Therefore, the result obtained should be fairly accurate.

11-109

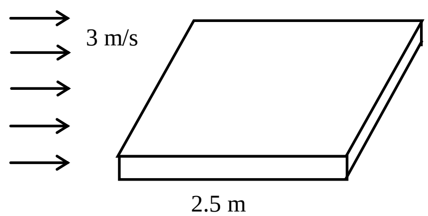
Solution A fluid flows over a 2.5-m long flat plate. The thickness of the boundary layer at intervals of 0.25 m is to be determined and plotted against the distance from the leading edge for air, water, and oil.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The surface of the plate is smooth.

Properties The kinematic viscosity of the three fluids at 1 atm and 20°C are: $\nu = 1.516\times10^{-5}$ m²/s for air , $\nu = \mu/\rho = (1.002\times10^{-3}\ \text{kg/m·s})/(998\ \text{kg/m}^3) = 1.004\times10^{-6}$ m²/s for water, and $\nu = 9.429\times10^{-4}$ m²/s for oil.

Analysis The thickness of the boundary layer along the flow for laminar and turbulent flows is given by

Laminar flow: $\delta_x = \dfrac{4.91x}{\mathrm{Re}_x^{1/2}}$, Turbulent flow: $\delta_x = \dfrac{0.38x}{\mathrm{Re}_x^{1/5}}$



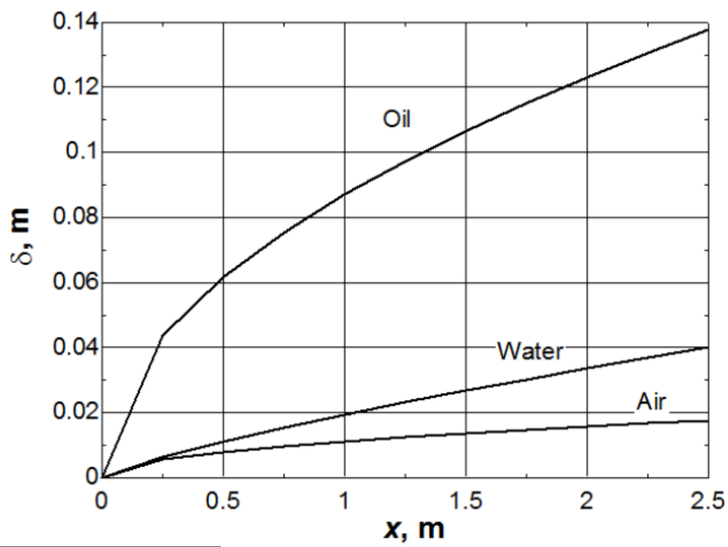
(*a*) AIR: The Reynolds number and the boundary layer thickness at the end of the first 0.25 m interval are

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(3\ \text{m/s})(0.25\ \text{m})}{1.516\times10^{-5}\ \text{m}^2/\text{s}} = 0.495\times10^5,$$

$$\delta_x = \frac{5x}{\mathrm{Re}_x^{1/2}} = \frac{4.91\times(0.25\ \text{m})}{(0.495\times10^5)^{0.5}} = 5.52\times10^{-3}\ \text{m}$$

We repeat calculations for all 0.25 m intervals. The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
V=3 "m/s"
nu1=1.516E-5 "m2/s, Air"
Re1=x*V/nu1
delta1=4.91*x*Re1^(-0.5) "m, laminar flow"
nu2=1.004E-6 "m2/s, water"
Re2=x*V/nu2
delta2=0.38*x*Re2^(-0.2) "m, turbulent flow"
nu3=9.429E-4 "m2/s, oil"
Re3=x*V/nu3
delta3=4.91*x*Re3^(-0.5) "m, laminar flow"
```



x, cm	Air		Water		Oil	
	Re	δ_x	Re	δ_x	Re	δ_x
0.00	0.000E+00	0.0000	0.000E+00	0.0000	0.000E+00	0.0000
0.25	4.947E+04	0.0055	7.470E+05	0.0064	7.954E+02	0.0435
0.50	9.894E+04	0.0078	1.494E+06	0.0111	1.591E+03	0.0616
0.75	1.484E+05	0.0096	2.241E+06	0.0153	2.386E+03	0.0754
1.00	1.979E+05	0.0110	2.988E+06	0.0193	3.182E+03	0.0870
1.25	2.474E+05	0.0123	3.735E+06	0.0230	3.977E+03	0.0973
1.50	2.968E+05	0.0135	4.482E+06	0.0266	4.773E+03	0.1066
1.75	3.463E+05	0.0146	5.229E+06	0.0301	5.568E+03	0.1152
2.00	3.958E+05	0.0156	5.976E+06	0.0335	6.363E+03	0.1231
2.25	4.453E+05	0.0166	6.723E+06	0.0369	7.159E+03	0.1306
2.50	4.947E+05	0.0175	7.470E+06	0.0401	7.954E+03	0.1376

Discussion Note that the flow is laminar for (a) and (c), and turbulent for (b). Also note that the thickness of the boundary layer is very small for air and water, but it is very large for oil. This is due to the high viscosity of oil.

11-110

Solution A fairing is installed to the front of a rig to reduce the drag coefficient. The maximum speed of the rig after the fairing is installed is to be determined.

Assumptions **1** The rig moves steadily at a constant velocity on a straight path in calm weather. **2** The bearing friction resistance is constant. **3** The effect of velocity on the drag and rolling resistance coefficients is negligible. **4** The buoyancy of air is negligible. **5** The power produced by the engine is used to overcome rolling resistance, bearing friction, and aerodynamic drag.

Properties The density of air is given to be 1.25 kg/m^3. The drag coefficient of the rig is given to be C_D = 0.96, and decreases to C_D = 0.76 when a fairing is installed. The rolling resistance coefficient is C_{RR} = 0.05.

Analysis The bearing friction resistance is given to be F_{bearing} = 350 N. The rolling resistance is

$$F_{RR} = C_{RR}W = 0.05(17{,}000\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 8339\text{ N}$$

The maximum drag occurs at maximum velocity, and its value before the fairing is installed is

$$F_{D1} = C_D A\frac{\rho V_1^2}{2} = (0.96)(9.2\text{ m}^2)\frac{(1.25\text{ kg/m}^3)(110/3.6\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 5154\text{ N}$$

Power is force times velocity, and thus the power needed to overcome bearing friction, drag, and rolling resistance is the product of the sum of these forces and the velocity of the rig,

$$\begin{aligned}\dot W_{\text{total}} &= \dot W_{\text{bearing}} + \dot W_{\text{drag}} + \dot W_{\text{RR}} = (F_{\text{bearing}} + F_D + F_{RR})\mathbf{V}\\ &= (350+8339+5154)(110/3.6\text{ m/s})\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right)\\ &= 423\text{ kW}\end{aligned}$$

The maximum velocity the rig can attain at the same power of 423 kW after the fairing is installed is determined by setting the sum of the bearing friction, rolling resistance, and the drag force equal to 423 kW,

$$\dot W_{\text{total}} = \dot W_{\text{bearing}} + \dot W_{\text{drag2}} + \dot W_{\text{RR}} = (F_{\text{bearing}} + F_{D2} + F_{RR})V_2 = \left(350 + C_{D2}A\frac{\rho V_2^2}{2} + 5154\right)V_2$$

Substituting the known quantities,

$$(423\text{ kW})\left(\frac{1000\text{ N}\cdot\text{m/s}}{1\text{ kW}}\right) = \left(350\text{ N} + (0.76)(9.2\text{ m}^2)\frac{(1.25\text{ kg/m}^3)V_2^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) + 5154\text{ N}\right)V_2$$

or,

$$423{,}000 = 5504V_2 + 4.37V_2^3$$

Solving it with an equation solver gives V_2 = **36.9 m/s = 133 km/h**.

Discussion Note that the maximum velocity of the rig increases from 110 km/h to 133 km/h as a result of reducing its drag coefficient from 0.96 to 0.76 while holding the bearing friction and the rolling resistance constant.

11-111E

Solution We are to estimate how much money is wasted by driving with a tennis ball on a car antenna.

Properties ρ_{fuel} = 50.2 lbm/ft^3, HV$_{fuel}$ = 1.47 × 10^7 ft·lbf/lbm, ρ_{air} = 0.07518 lbm/ft^3, μ_{air} = 1.227 × 10^{-5} lbm/ft·s

Analysis First some conversions: V = 55 mph = 80.667 ft/s, D = 2.62 in = 0.21833 ft, and the total distance traveled in one year = L = 16,000 miles = 8.448 × 10^7 ft. We calculate the Reynolds number,

$$\text{Re} = \frac{\rho_{air} V D}{\mu} = \frac{\left(0.07518 \text{ lbm/ft}^3\right)\left(80.667 \text{ ft/s}\right)\left(0.21833 \text{ ft}\right)}{1.227\times10^{-5} \text{ lbm/}\left(\text{ft}\cdot\text{s}\right)} = 107900$$

At this Reynolds number and for the given roughness ratio, we look up C_D for a sphere: $C_D \approx 0.505$. [Note: student answers may vary because this value is not easily read from the graph, but C_D should be around 0.5.]

The additional drag force due to the sphere is

$$F_D = \tfrac{1}{2}\rho_{air} V^2 C_D A$$

where A is the frontal area of the sphere, $A = \pi D^2/4$. The work required to overcome this additional drag is force times distance. So, letting L be the total distance driven in a year,

$$\text{Work}_{drag} = F_D L = \tfrac{1}{2}\rho_{air} V^2 C_D \left(\pi D^2/4\right) L$$

The energy required to perform this work is much greater than this due to overall efficiency of the car engine, transmission, etc. Thus,

$$E_{required} = \frac{\text{Work}_{drag}}{\eta_{overall}} = \frac{\tfrac{1}{2}\rho_{air} V^2 C_D \left(\pi D^2/4\right) L}{\eta_{overall}}$$

But the required energy is also equal to the heating value of the fuel HV times the mass of fuel required. In terms of required fuel *volume*, volume = mass/density. Thus,

$$\boxed{V_{\text{fuel required}} = \frac{m_{\text{fuel required}}}{\rho_{fuel}} = \frac{E_{required}/\text{HV}}{\rho_{fuel}} = \frac{\tfrac{1}{2}\rho_{air} V^2 C_D \left(\pi D^2/4\right) L}{\rho_{fuel}\eta_{overall}\text{HV}}}$$

The above is our answer in variable form. Finally, we plug in the given values and properties to obtain the numerical answer,

$$V_{\text{fuel required}} = \frac{0.5(0.07518\,\text{lbm/ft}^3)(80.667\,\text{ft/s})^2(0.505)[\pi(0.21833\,\text{ft})^2/4](8.448\times10^7\,\text{ft})}{(50.2\,\text{lbm/ft}^3)(0.308)(1.47\times10^7\,\text{ft}\cdot\text{lbf/lbm})}\left(\frac{1\,\text{lbf}}{32.174\,\text{lbm}\cdot\text{ft/s}^2}\right)$$

$$= 0.6418\,\text{L} \approx \mathbf{0.64L}$$

$$\boxed{V_{\text{fuel required}} = \frac{\tfrac{1}{2}\left(0.07518 \text{ lbm/ft}^3\right)\left(80.667 \text{ ft/s}\right)^2\left(0.505\right)\left(\pi\left(0.21833 \text{ ft}\right)^2/4\right)\left(6.336\times10^7 \text{ ft}\right)}{\left(50.2 \text{ lbm/ft}^3\right)\left(0.308\right)\left(1.47\times10^7 \text{ ft}\cdot\text{lbf/lbm}\right)}\left(\frac{\text{lbf}}{32.174 \text{ lbm}\cdot\text{ft/s}^2}\right)}$$

which yields $V_{\text{fuel required}}$ = 0.05343 ft^3, which is equivalent to 0.3997 gallons per year. At $4 per gallon, **the total cost is about $1.60 per year**.

Since the cost is minimal (only about a dollar per year), Janie's friends should get off her back about this.

Discussion I we use $C_D \approx 0.50$, the cost turns out to be $1.58, but we should report our result to only 2 significant digits anyway, so our answer is still $1.6 per year.

11-112

Solution Spherical aluminum balls are dropped into glycerin, and their terminal velocities are measured. The velocities are to be compared to those predicted by Stokes law, and the error involved is to be determined.

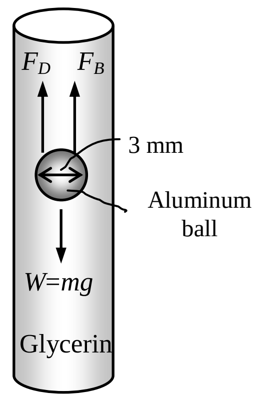
Assumptions **1** The Reynolds number is low (at the order of 1) so that Stokes law is applicable (to be verified). **2** The diameter of the tube that contains the fluid is large enough to simulate free fall in an unbounded fluid body. **3** The tube is long enough to assure that the velocity measured is terminal velocity.

Properties The density of aluminum balls is given to be ρ_s = 2700 kg/m^3. The density and viscosity of glycerin are given to be ρ_f = 1274 kg/m^3 and μ = 1 kg/m·s.

Analysis The terminal velocity of a free falling object is reached when the drag force equals the weight of the solid object, less the buoyancy force applied by the fluid,

$$F_D = W - F_B \quad \text{where} \quad F_D = 3\pi\mu VD \text{ (Stokes law)}, \quad W = \rho_s g \mathbf{V}, \quad \text{and } F_B = \rho_f g \mathbf{V}$$

Here $V = \pi D^3/6$ is the volume of the sphere. Substituting and simplifying,



$$3\pi\mu VD = \rho_s g \mathbf{V} - \rho_f g \mathbf{V} \quad \rightarrow \quad 3\pi\mu VD = (\rho_s - \rho_f) g \frac{\pi D^3}{6}$$

Solving for the terminal velocity V of the ball gives

$$V = \frac{gD^2(\rho_s - \rho_f)}{18\mu}$$

(*a*) D = 2 mm and V = 3.2 mm/s

$$V = \frac{(9.81\,\text{m/s}^2)(0.002\,\text{m})^2(2700\text{-}1274)\ \text{kg/m}^3}{18(1\ \text{kg/m}\cdot\text{s})} = \mathbf{0.00311\,m/s} = \mathbf{3.11\,mm/s}$$

$$\text{Error} = \frac{V_{\text{experimental}} - V_{\text{Stokes}}}{V_{\text{experimental}}} = \frac{3.2 - 3.11}{3.2} = \mathbf{0.029 \quad or\ 2.9\%}$$

(*b*) D = 4 mm and V = 12.8 mm/s

$$V = \frac{(9.81\,\text{m/s}^2)(0.004\,\text{m})^2(2700\text{-}1274)\ \text{kg/m}^3}{18(1\ \text{kg/m}\cdot\text{s})} = \mathbf{0.0124\,m/s} = \mathbf{12.4\,mm/s}$$

$$\text{Error} = \frac{V_{\text{experimental}} - V_{\text{Stokes}}}{V_{\text{experimental}}} = \frac{12.8 - 12.4}{12.8} = \mathbf{0.029 \quad or\ 2.9\%}$$

(*c*) D = 10 mm and V = 60.4 mm/s

$$V = \frac{(9.81\,\text{m/s}^2)(0.010\,\text{m})^2(2700\text{-}1274)\ \text{kg/m}^3}{18(1\ \text{kg/m}\cdot\text{s})} = \mathbf{0.0777\,m/s} = \mathbf{77.7\,mm/s}$$

$$\text{Error} = \frac{V_{\text{experimental}} - V_{\text{Stokes}}}{V_{\text{experimental}}} = \frac{60.4 - 77.7}{60.4} = -\mathbf{0.287 \quad or\ \text{-}28.7\%}$$

There is a good agreement for the first two diameters. However the error for third one is large. The Reynolds number for each case is

$$(a)\ \ \text{Re} = \frac{\rho VD}{\mu} = \frac{(1274\ \text{kg/m}^3)(0.0032\ \text{m/s})(0.002\,\text{m})}{1\ \text{kg}\cdot\text{m/s}} = 0.008, \qquad (b)\ \text{Re} = 0.065, \quad \text{and} \quad (c)\ \text{Re} = 0.770.$$

We observe that Re << 1 for the first two cases, and thus the creeping flow idealization is applicable. But this is not the case for the third case.

Discussion If we used the general form of the equation (see next problem) we would obtain V = 59.7 mm/s for part (*c*), which is very close to the experimental result (60.4 mm/s).

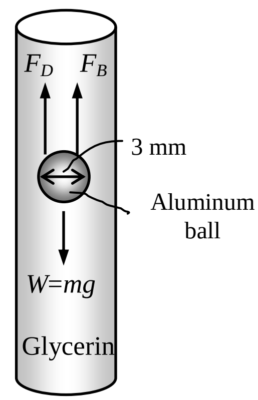
11-113

Solution Spherical aluminum balls are dropped into glycerin, and their terminal velocities are measured. The velocities predicted by a more general form of Stokes law and the error involved are to be determined.

Assumptions **1** The Reynolds number is low (of order 1) so that Stokes law is applicable (to be verified). **2** The diameter of the tube that contains the fluid is large enough to simulate free fall in an unbounded fluid body. **3** The tube is long enough to assure that the velocity measured is terminal velocity.

Properties The density of aluminum balls is given to be ρ_s = 2700 kg/m^3. The density and viscosity of glycerin are given to be ρ_f = 1274 kg/m^3 and μ = 1 kg/m·s.

Analysis The terminal velocity of a free falling object is reached when the drag force equals the weight of the solid object, less the buoyancy force applied by the fluid,



$$F_D = W - F_B$$

where $F_D = 3\pi\mu DV + (9\pi/16)\rho_s V^2 D^2$, $W = \rho_s g\boldsymbol{V}$, and $F_B = \rho_f g\boldsymbol{V}$

Here $\boldsymbol{V} = \pi D^3/6$ is the volume of the sphere. Substituting and simplifying,

$$3\pi\mu VD + (9\pi/16)\rho_s V^2 D^2 = (\rho_s - \rho_f)g\frac{\pi D^3}{6}$$

Solving for the terminal velocity V of the ball gives

$$V = \frac{-b+\sqrt{b^2-4ac}}{2a} \quad \text{where } a = \frac{9\pi}{16}\rho_s D^2,\ b = 3\pi\mu D, \text{ and } c = -(\rho_s - \rho_f)g\frac{\pi D^3}{6}$$

(a) D = 2 mm and V = 3.2 mm/s: a = 0.01909, b = 0.01885, c = -0.0000586

$$V = \frac{-0.01885+\sqrt{(0.01885)^2 - 4\times 0.01909\times(-0.0000586)}}{2\times 0.01909} = \mathbf{0.00310\ m/s} = \mathbf{3.10\ mm/s}$$

$$\text{Error} = \frac{V_{\text{experimental}} - V_{\text{Stokes}}}{V_{\text{experimental}}} = \frac{3.2-3.10}{3.2} = \mathbf{0.032\ \ or\ 3.2\%}$$

(b) D = 4 mm and V = 12.8 mm/s: a = 0.07634, b = 0.0377, c = -0.0004688

$$V = \frac{-0.0377+\sqrt{(0.0377)^2 - 4\times 0.07634\times(-0.0004688)}}{2\times 0.07634} = \mathbf{0.0121\ m/s} = \mathbf{12.1\ mm/s}$$

$$\text{Error} = \frac{V_{\text{experimental}} - V_{\text{Stokes}}}{V_{\text{experimental}}} = \frac{12.8-12.1}{12.8} = \mathbf{0.052\ \ or\ 5.2\%}$$

(c) D = 10 mm and V = 60.4 mm/s: a = 0.4771, b = 0.09425, c = -0.007325

$$V = \frac{-0.09425+\sqrt{(0.09425)^2 - 4\times 0.3771\times(-0.007325)}}{2\times 0.4771} = \mathbf{0.0597\ m/s} = \mathbf{59.7\ mm/s}$$

$$\text{Error} = \frac{V_{\text{experimental}} - V_{\text{Stokes}}}{V_{\text{experimental}}} = \frac{60.4-59.7}{60.4} = \mathbf{0.012\ \ or\ 1.2\%}$$

The Reynolds number for the three cases are

(a) $\text{Re} = \dfrac{\rho VD}{\mu} = \dfrac{(1274\ \text{kg/m}^3)(0.0032\ \text{m/s})(0.002\ \text{m})}{1\ \text{kg}\cdot\text{m/s}} = 0.008$, ($b$) Re = 0.065, and ($c$) Re = 0.770.

Discussion There is a good agreement for the third case (case c), but the general Stokes law increased the error for the first two cases (cases a and b) from 2.9% and 2.9% to 3.2% and 5.2%, respectively. Therefore, the basic form of Stokes law should be preferred when the Reynolds number is much lower than 1.

11-114

Solution A spherical aluminum ball is dropped into oil. A relation is to be obtained for the variation of velocity with time and the terminal velocity of the ball. The variation of velocity with time is to be plotted, and the time it takes to reach 99% of terminal velocity is to be determined.

Assumptions **1** The Reynolds number is low (Re << 1) so that Stokes law is applicable. **2** The diameter of the tube that contains the fluid is large enough to simulate free fall in an unbounded fluid body.

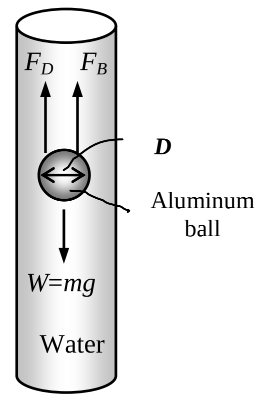
Properties The density of aluminum balls is given to be ρ_s = 2700 kg/m^3. The density and viscosity of oil are given to be ρ_f = 876 kg/m^3 and μ = 0.2177 kg/m·s.

Analysis The free body diagram is shown in the figure. The net force acting downward on the ball is the weight of the ball less the weight of the ball and the buoyancy force applied by the fluid,

$$F_{net} = W - F_D - F_B \quad \text{where} \quad F_D = 3\pi\mu DV, \quad W = m_s g = \rho_s g \boldsymbol{V}, \quad \text{and} \quad F_B = \rho_f g \boldsymbol{V}$$

where F_D is the drag force, F_B is the buoyancy force, and W is the weight. Also, $\boldsymbol{V} = \pi D^3/6$ is the volume, m_s is the mass, D is the diameter, and V the velocity of the ball. Applying Newton's second law in the vertical direction,

$$F_{net} = ma \quad \rightarrow \quad m_s g - F_D - F_B = m\frac{dV}{dt}$$



Substituting the drag and buoyancy force relations,

$$\rho_s \frac{\pi D^3}{6} g - 3\pi\mu DV - \rho_f g \frac{\pi D^3}{6} = \rho_s \frac{\pi D^3}{6}\frac{dV}{dt}$$

or, $$g\left(1 - \frac{\rho_f}{\rho s}\right) - \frac{18\mu}{\rho_s D^2} V = \frac{dV}{dt} \quad \rightarrow \quad a - bV = \frac{dV}{dt}$$

where $a = g(1-\rho_f/\rho_s)$ and $b = 18\mu/(\rho_s D^2)$. It can be rearranged as $\dfrac{dV}{a-bV} = dt$

Integrating from t = 0 where V = 0 to $t = t$ where $V = V$ gives

$$\int_0^V \frac{dV}{a-bV} = \int_0^t dt \quad \rightarrow \quad -\frac{\ln(a-bV)}{b}\Bigg|_0^V = t\,\Big|_0^t \quad \rightarrow \quad \ln\left(\frac{a-bV}{a}\right) = -bt$$

Solving for V gives the desired relation for the variation of velocity of the ball with time,

$$V = \frac{a}{b}\left(1 - e^{-bt}\right) \qquad \text{or} \qquad V = \frac{(\rho_s - \rho_f)gD^2}{18\mu}\left(1 - e^{-\frac{18\mu}{\rho_s D^2}t}\right) \tag{1}$$

Note that as $t \rightarrow \infty$, it gives the terminal velocity as $V_{\text{terminal}} = \dfrac{a}{b} = \dfrac{(\rho_s - \rho_f)gD^2}{18\mu}$ (2)

The time it takes to reach 99% of terminal velocity can to be determined by replacing V in Eq. 1 by $0.99V_{\text{terminal}} = 0.99a/b$. This gives e^{-bt} = 0.01 or

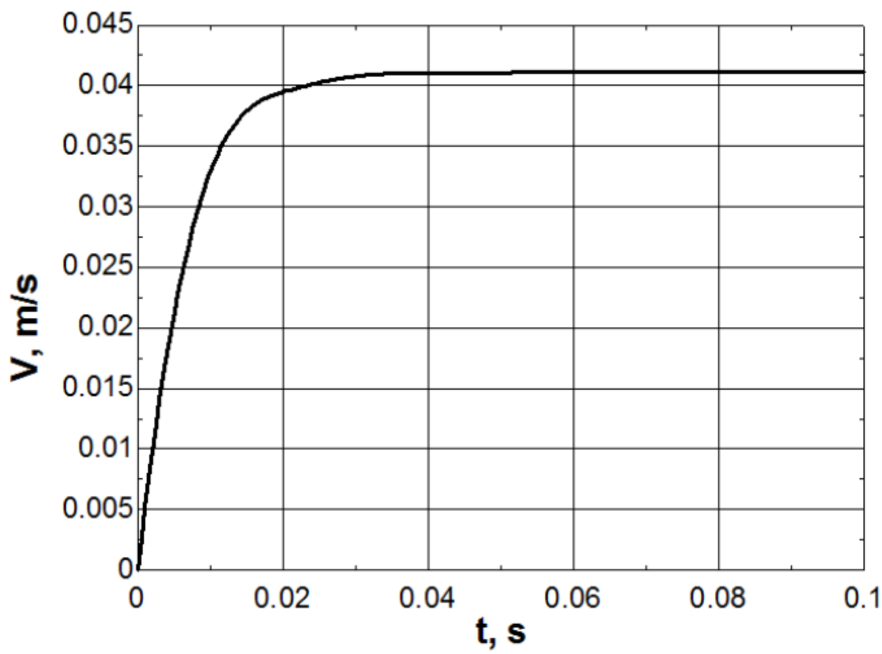
$$t_{99\%} = -\frac{\ln(0.01)}{b} = -\frac{\ln(0.01)\rho_s D^2}{18\mu} \tag{3}$$

Given values: D = 0.003 m, ρ_f = 876 kg/m^3, μ = 0.2177 kg/m·s, g = 9.81 m/s^2.

Calculation results: Re = 0.50, a = 6.627, b = 161.3, $t_{99\%}$ = **0.029 s**, and $V_{\text{terminal}} = a/b$ = **0.04 m/s**.

Continued on next page →

t, s	*V*, m/s
0.00	0.000
0.01	0.033
0.02	0.039
0.03	0.041
0.04	0.041
0.05	0.041
0.06	0.041
0.07	0.041
0.08	0.041
0.09	0.041
0.10	0.041



Discussion The velocity increases rapidly at first, but quickly levels off by around 0.04 s.

11-115

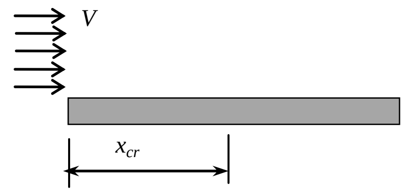
Solution Engine oil flows over a long flat plate. The distance from the leading edge x_{cr} where the flow becomes turbulent is to be determined, and thickness of the boundary layer over a distance of $2x_{cr}$ is to be plotted.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** The surface of the plate is smooth.

Properties The kinematic viscosity of engine oil at 40°C is $\nu = 2.485\times10^{-4}$ m^2/s.

Analysis The thickness of the boundary layer along the flow for laminar and turbulent flows is given by

Laminar flow: $\delta_x = \dfrac{4.91x}{\mathrm{Re}_x^{1/2}}$, Turbulent flow: $\delta_x = \dfrac{0.38x}{\mathrm{Re}_x^{1/5}}$



The distance from the leading edge x_{cr} where the flow turns turbulent is determined by setting Reynolds number equal to the critical Reynolds number,

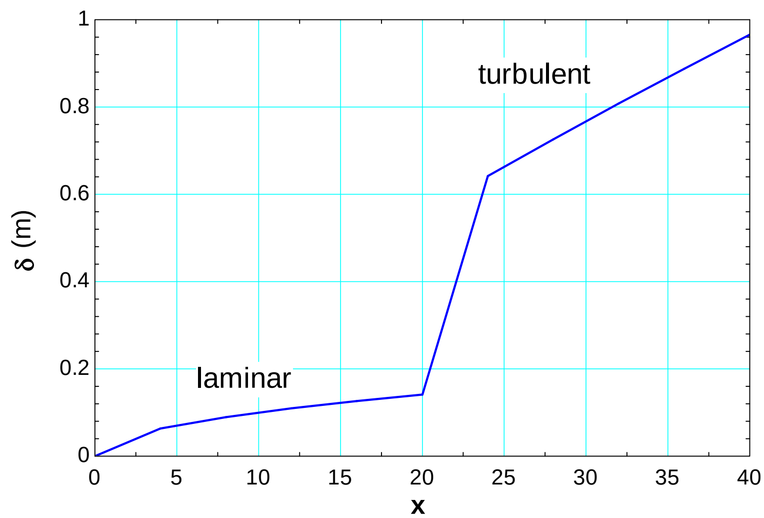
$$\mathrm{Re}_{cr} = \frac{Vx_{cr}}{\nu} \rightarrow x_{cr} = \frac{\mathrm{Re}_{cr}\,\nu}{V} = \frac{(5\times10^5)(2.485\times10^{-4}\ \mathrm{m^2/s})}{6\ \mathrm{m/s}} = \mathbf{20.7\,m}$$

Therefore, we should consider flow over 2×20.7 = 41.4 m long section of the plate, and use the laminar relation for the first half, and the turbulent relation for the second part to determine the boundary layer thickness. For example, the Reynolds number and the boundary layer thickness at a distance 2 m from the leading edge of the plate are

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(6\ \mathrm{m/s})(2\ \mathrm{m})}{2.485\times10^{-4}\ \mathrm{m^2/s}} = 48{,}290, \quad \delta_x = \frac{4.91x}{\mathrm{Re}_x^{1/2}} = \frac{4.91\times(2\ \mathrm{m})}{(48{,}290)^{0.5}} = 0.0447\ \mathrm{m}$$

Calculating the boundary layer thickness and plotting give

x, m	Re	$\delta_{x,\ \mathrm{laminar}}$	$\delta_{x,\ \mathrm{turbulent}}$
0	0	0	
4	96579	0.0632	
8	193159	0.08937	
12	289738	0.1095	
16	386318	0.1264	
20	482897	0.1413	
24	579477		0.6418
28	676056		0.726
32	772636		0.8079
36	869215		0.8877
40	965795		0.9658



Discussion Notice the sudden, rapid rise in boundary layer thickness when the boundary layer becomes turbulent.

11-116

Solution A spherical object is dropped into a fluid, and its terminal velocity is measured. The viscosity of the fluid is to be determined.

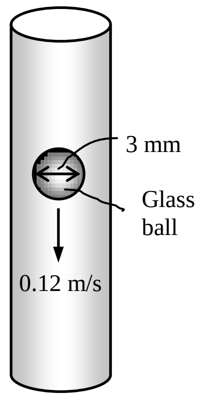
Assumptions **1** The Reynolds number is low (at the order of 1) so that Stokes law is applicable (to be verified). **2** The diameter of the tube that contains the fluid is large enough to simulate free fall in an unbounded fluid body. **3** The tube is long enough to assure that the velocity measured is the terminal velocity.

Properties The density of glass ball is given to be ρ_s = 2500 kg/m^3. The density of the fluid is given to be ρ_f = 875 kg/m^3.

Analysis The terminal velocity of a free falling object is reached when the drag force equals the weight of the solid object, less the buoyancy force applied by the fluid,

$$F_D = W - F_B \quad \text{where} \quad F_D = 3\pi\mu VD \text{ (Stokes law)}, \quad W = \rho_s g \boldsymbol{V}, \quad \text{and} \quad F_B = \rho_f g \boldsymbol{V}$$

Here $\boldsymbol{V} = \pi D^3/6$ is the volume of the sphere. Substituting and simplifying,



$$3\pi\mu VD = \rho_s g \boldsymbol{V} - \rho_f g \boldsymbol{V} \quad \rightarrow \quad 3\pi\mu VD = (\rho_s - \rho_f) g \frac{\pi D^3}{6}$$

Solving for μ and substituting, the dynamic viscosity of the fluid is determined to be

$$\mu = \frac{gD^2(\rho_s - \rho_f)}{18V} = \frac{(9.81\,\text{m/s}^2)(0.003\,\text{m})^2(2500 - 875)\,\text{kg/m}^3}{18(0.12\,\text{m/s})} = \mathbf{0.0664\,kg/m \cdot s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(875\,\text{kg/m}^3)(0.12\,\text{m/s})(0.003\,\text{m})}{0.0664\,\text{kg}\cdot\text{m/s}} = 4.74$$

which is at the order of 1. Therefore, the creeping flow idealization is valid.

Discussion Flow separation starts at about Re = 10. Therefore, Stokes law can be used for Reynolds numbers up to this value, but this should be done with care.

Fundamentals of Engineering (FE) Exam Problems

11-117

Which ones are physical phenomena associated with fluid flow over bodies?

I. Drag force acting on automobiles II. The lift developed by airplane wings

III. Upward draft of rain and snow IV. Power generated by wind turbines

(*a*) I and II (*b*) I and III (*c*) II and III (*d*) I, II, and III (*e*) I, II, III, and IV

Answer (*e*) I, II, III, and IV

11-118

The sum of the components of the pressure and wall shear forces in the direction normal to the flow is called

(*a*) Drag (*b*) Friction (*c*) Lift (*d*) Bluff (*e*) Blunt

Answer (*c*) Lift

11-119

A car is moving at a speed of 70 km/h in air at 20°C. The frontal area of the car is 2.4 m^2. If the drag force acting on the car in the flow direction is 205 N, the drag coefficient of the car is

(*a*) 0.312 (*b*) 0.337 (*c*) 0.354 (*d*) 0.375 (*e*) 0.391

Answer (*d*) 0.375

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=70 [km/h]*Convert(km/h, m/s)
T=20 [C]
A=2.4 [m^2]
F_D=205 [N]
rho=1.204 [kg/m^3] "at 20 C"
F_D=C_D*A*rho*V^2/2
```

11-120

A person is driving his motorcycle at a speed of 110 km/h in air at 20°C. The frontal area of the motorcycle and driver is 0.75 m^2. If the drag coefficient under these conditions is estimated to be 0.9, the drag force acting on the car in the flow direction is

(*a*) 379 N (*b*) 220 N (*c*) 283 N (*d*) 308 N (*e*) 450 N

Answer (*a*) 379 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=110 [km/h]*Convert(km/h, m/s)
T=20 [C]
A=0.75 [m^2]
C_D=0.9
rho=1.204 [kg/m^3] "at 20 C"
F_D=C_D*A*rho*V^2/2
```

11-121

The manufacturer of a car reduces the drag coefficient of the car from 0.38 to 0.33 as a result of some modifications in its shape and design. If, on average, the aerodynamic drag accounts for 20 percent of the fuel consumption, the percent reduction in the fuel consumption of the car due to reducing the drag coefficient is

(*a*) 15% (*b*) 13% (*c*) 6.6% (*d*) 2.6% (*e*) 1.3%

Answer (*d*) 2.6%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_D1=0.38
C_D2=0.33
f_drag=0.20
PercentReduction=f_drag*(C_D1-C_D2)/C_D1*Convert(,%)
```

11-122

The region of flow trailing the body where the effects of the body on velocity are felt is called

(*a*) Wake (*b*) Separated region (*c*) Stall (*d*) Vortice (*e*) Boundary

Answer (*a*) Wake

11-123

The turbulent boundary layer can be considered to consist of four regions. Which one is not one of them?

(*a*) Buffer layer (*b*) Overlap layer (*c*) Transition layer (*d*) Viscous layer

(*e*) Turbulent layer

Answer (*c*) Transition layer

11-124

Water at 10°C flows over a 1.1-m-long flat plate with a velocity of 0.55 m/s. If the width of the plate is 2.5 m, the drag force acting on the top side of the plate is (Water properties at 10°C are: ρ = 999.7 kg/m^3, μ = 1.307 × 10^{-3} kg/m·s.)

(*a*) 0.46 N (*b*) 0.81 N (*c*) 2.75 N (*d*) 4.16 N (*e*) 6.32 N

Answer (*b*) 0.81 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=10 [C]
L=1.1 [m]
V=0.55 [m/s]
W=2.5 [m]
rho=999.7 [kg/m^3]
mu=1.307E-3 [kg/m-s]
nu=mu/rho
Re_L=V*L/nu
C_f=1.328*Re_L^(-0.5)
A=L*W
F_D=C_f*A*rho*V^2/2
```

11-125

Water at 10°C flows over a 3.75-m-long flat plate with a velocity of 1.15 m/s. If the width of the plate is 6.5 m, the average friction coefficient over the entire plate is (Water properties at 10°C are: ρ = 999.7 kg/m^3, μ = 1.307 × 10^{-3} kg/m·s.)

(*a*) 0.00508 (*b*) 0.00447 (*c*) 0.00302 (*d*) 0.00367 (*e*) 0.00315

Answer (*e*) 0.00315

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=10 [C]
L=3.75 [m]
V=1.15 [m/s]
W=6.5 [m]
rho=999.7 [kg/m^3]
mu=1.307E-3 [kg/m-s]
nu=mu/rho
Re_L=V*L/nu
C_f=0.074/Re_L^(1/5)-1742/Re_L
```

11-126

Air at 30°C flows over a 3.0-cm-outer-diameter, 45-m-long pipe with a velocity of 6 m/s. The drag force exerted on the pipe by the air is (Air properties at 30°C are: $\rho = 1.164$ kg/m^3, $\nu = 1.608 \times 10^{-5}$ m^2/s.)

(*a*) 19.3 N (*b*) 36.8 N (*c*) 49.3 N (*d*) 53.9 N (*e*) 60.1 N

Answer (*b*) 36.8 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=30 [C]
D=0.03 [m]
L=45 [m]
V=6 [m/s]
rho=1.164 [kg/m^3]
nu=1.608E-5 [m^2/s]
Re=V*D/nu
C_D=1.3 "From Fig. 11-34 at the calculated Reynolds number"
A=L*D
F_D=C_D*A*rho*V^2/2
```

11-127

A 0.8-m-outer-diameter spherical tank is completely submerged in a flowing water stream at a velocity of 2.5 m/s. The drag force acting on the tank is (Water properties are: $\rho = 998.0$ kg/m^3, $\mu = 1.002 \times 10^{-3}$ kg/m·s.)

(*a*) 878 N (*b*) 627 N (*c*) 545 N (*d*) 356 N (*e*) 220 N

Answer (*e*) 220 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.8 [m]
V=2.5 [m/s]
rho=998 [kg/m^3]
mu=1.002E-3 [kg/m-s]
Re=rho*V*D/mu
C_D=0.14 "From Fig. 11-36 at the calculated Reynolds number"
A=pi*D^2/4
F_D=C_D*A*rho*V^2/2
```

11-128

An airplane has a total mass of 18,000 kg and a wing planform area of 35 m^2. The density of air at the ground is 1.2 kg/m^3. The maximum lift coefficient is 3.48. The minimum safe speed for takeoff and landing with extending the flaps is

(*a*) 305 km/h (*b*) 173 km/h (*c*) 194 km/h (*d*) 212 km/h (*e*) 246 km/h

Answer (*d*) 212 km/h

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=18000 [kg]
A=35 [m^2]
rho=1.2 [kg/m^3]
C_L_max=3.48
g=9.81 [m/s^2]
W=m*g
V_min=sqrt((2*W)/(rho*C_L_max*A))
f_safety=1.2
V_min_safe=f_safety*V_min*convert(m/s, km/h)
```

11-129

An airplane has a total mass of 35,000 kg and a wing planform area of 65 m^2. The airplane is cruising at 10,000 m altitude with a velocity of 1100 km/h. The density of air on cruising altitude is 0.414 kg/m^3. The lift coefficient of this airplane at the cruising altitude is

(*a*) 0.273 (*b*) 0.290 (*c*) 0.456 (*d*) 0.874 (*e*) 1.22

Answer (*a*) 0.273

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=35000 [kg]
A=65 [m^2]
V=1100 [km/h]*Convert(km/h, m/s)
rho=0.414 [kg/m^3]

g=9.81 [m/s^2]
W=m*g
F_L=W
C_L=F_L/(1/2*rho*V^2*A)
```

11-130

An airplane is cruising at a velocity of 800 km/h in air whose density is 0.526 kg/m^3. The airplane has a wing planform area of 90 m^2. The lift and drag coefficients on cruising conditions are estimated to be 2.0 and 0.06, respectively. The power that needs to be supplied to provide enough trust to overcome wing drag is

(*a*) 9760 kW (*b*) 11,300 kW (*c*) 15,600 kW (*d*) 18,200 kW (*e*) 22,600 kW

Answer (*c*) 15,600 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=800 [km/h]*Convert(km/h, m/s)
rho=0.526 [kg/m^3]
A=90 [m^2]
CL=2.0
C_D=0.06
F_D=C_D*A*rho*V^2/2
Power=F_D*V*Convert(W, kW)
```

Design and Essay Problems

11-131 to 11-134

Solution Students' essays and designs should be unique and will differ from each other.

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

Third Edition

Yunus A. Çengel & John M. Cimbala

McGraw-Hill, 2013

Chapter 12
COMPRESSIBLE FLOW

Stagnation Properties

12-1C

Solution We are to discuss the temperature change from an airplane's nose to far away from the aircraft.

Analysis The temperature of the air **rises as it approaches the nose because of the stagnation process**.

Discussion In the frame of reference moving with the aircraft, the air decelerates from high speed to zero at the nose (stagnation point), and this causes the air temperature to rise.

12-2C

Solution We are to define dynamic temperature.

Analysis *Dynamic temperature* is **the temperature rise of a fluid during a stagnation process**.

Discussion When a gas decelerates from high speed to zero speed at a stagnation point, the temperature of the gas rises.

12-3C

Solution We are to discuss the measurement of flowing air temperature with a probe – is there significant error?

Analysis **No, there is not significant error**, because the velocities encountered in air-conditioning applications are very low, and thus the static and the stagnation temperatures are practically identical.

Discussion If the air stream were supersonic, however, the error would indeed be significant.

12-4

Solution Air flows through a device. The stagnation temperature and pressure of air and its velocity are specified. The static pressure and temperature of air are to be determined.

Assumptions **1** The stagnation process is isentropic. **2** Air is an ideal gas.

Properties The properties of air at an anticipated average temperature of 600 K are c_p = 1.051 kJ/kg·K and k = 1.376.

Analysis The static temperature and pressure of air are determined from

$$T = T_0 - \frac{V^2}{2c_p} = 673.2 - \frac{(570\text{ m/s})^2}{2\times 1.051\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 518.6\text{ K} \cong \mathbf{519K}$$

and

$$P_2 = P_{02}\left(\frac{T_2}{T_{02}}\right)^{k/(k-1)} = (0.6\text{ MPa})\left(\frac{518.6\text{ K}}{673.2\text{ K}}\right)^{1.376/(1.376-1)} = \mathbf{0.231MPa}$$

Discussion Note that the stagnation properties can be significantly different than thermodynamic properties.

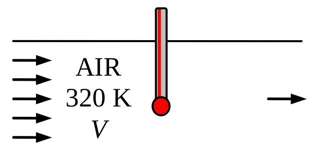
12-5

Solution Air at 320 K is flowing in a duct. The temperature that a stationary probe inserted into the duct will read is to be determined for different air velocities.

Assumptions The stagnation process is isentropic.

Properties The specific heat of air at room temperature is c_p = 1.005 kJ/kg·K.

Analysis The air which strikes the probe will be brought to a complete stop, and thus it will undergo a stagnation process. The thermometer will sense the temperature of this stagnated air, which is the stagnation temperature, T_0. It is determined from $T_0 = T + \dfrac{V^2}{2c_p}$. The results for each case are calculated below:



(*a*) $$T_0 = 320\text{ K} + \frac{(1\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{320.0K}$$

(*b*) $$T_0 = 320\text{ K} + \frac{(10\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{320.1K}$$

(*c*) $$T_0 = 320\text{ K} + \frac{(100\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{325.0K}$$

(*d*) $$T_0 = 320\text{ K} + \frac{(1000\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{817.5K}$$

Discussion Note that the stagnation temperature is nearly identical to the thermodynamic temperature at low velocities, but the difference between the two is significant at high velocities.

12-6

Solution The states of different substances and their velocities are specified. The stagnation temperature and stagnation pressures are to be determined.

Assumptions **1** The stagnation process is isentropic. **2** Helium and nitrogen are ideal gases.

Analysis (*a*) Helium can be treated as an ideal gas with c_p = 5.1926 kJ/kg·K and k = 1.667. Then the stagnation temperature and pressure of helium are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 50°\text{C} + \frac{(240\text{ m/s})^2}{2\times 5.1926\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{55.5°C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.25\text{ MPa})\left(\frac{328.7\text{ K}}{323.2\text{ K}}\right)^{1.667/(1.667-1)} = \mathbf{0.261\ MPa}$$

(*b*) Nitrogen can be treated as an ideal gas with c_p = 1.039 kJ/kg·K and k =1.400. Then the stagnation temperature and pressure of nitrogen are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 50°\text{C} + \frac{(300\text{ m/s})^2}{2\times 1.039\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{93.3°C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.15\text{ MPa})\left(\frac{366.5\text{ K}}{323.2\text{ K}}\right)^{1.4/(1.4-1)} = \mathbf{0.233\ MPa}$$

(*c*) Steam can be treated as an ideal gas with c_p = 1.865 kJ/kg·K and k =1.329. Then the stagnation temperature and pressure of steam are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 350°\text{C} + \frac{(480\text{ m/s})^2}{2\times 1.865\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{411.8°C} = \mathbf{685\ K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.1\text{ MPa})\left(\frac{685\text{ K}}{623.2\text{ K}}\right)^{1.329/(1.329-1)} = \mathbf{0.147\ MPa}$$

Discussion Note that the stagnation properties can be significantly different than thermodynamic properties.

12-7

Solution The state of air and its velocity are specified. The stagnation temperature and stagnation pressure of air are to be determined.

Assumptions **1** The stagnation process is isentropic. **2** Air is an ideal gas.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4.

Analysis The stagnation temperature of air is determined from

$$T_0 = T + \frac{V^2}{2c_p} = 238\text{ K} + \frac{(325\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 290.5 \cong \mathbf{291\ K}$$

Other stagnation properties at the specified state are determined by considering an isentropic process between the specified state and the stagnation state,

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (36\text{ kPa})\left(\frac{290.5\text{ K}}{238\text{ K}}\right)^{1.4/(1.4-1)} = 72.37\text{ kPa} \cong \mathbf{72.4\ kPa}$$

Discussion Note that the stagnation properties can be significantly different than thermodynamic properties.

12-8E

Solution Steam flows through a device. The stagnation temperature and pressure of steam and its velocity are specified. The static pressure and temperature of the steam are to be determined.

Assumptions **1** The stagnation process is isentropic. **2** Steam is an ideal gas.

Properties Steam can be treated as an ideal gas with c_p = 0.4455 Btu/lbm·R and k =1.329.

Analysis The static temperature and pressure of steam are determined from

$$T = T_0 - \frac{V^2}{2c_p} = 700°\text{F} - \frac{(900\text{ ft/s})^2}{2\times 0.4455\text{ Btu/lbm}\cdot°\text{F}}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = \mathbf{663.7°F}$$

$$P = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (120\text{ psia})\left(\frac{1123.7\text{ R}}{1160\text{ R}}\right)^{1.329/(1.329-1)} = \mathbf{105.5 psia}$$

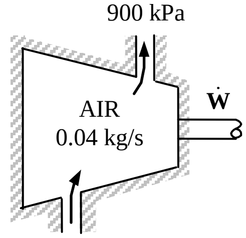
Discussion Note that the stagnation properties can be significantly different than thermodynamic properties.

12-9

Solution The inlet stagnation temperature and pressure and the exit stagnation pressure of air flowing through a compressor are specified. The power input to the compressor is to be determined.

Assumptions **1** The compressor is isentropic. **2** Air is an ideal gas.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4.



Analysis The exit stagnation temperature of air T_{02} is determined from

$$T_{02} = T_{01}\left(\frac{P_{02}}{P_{01}}\right)^{(k-1)/k} = (308.2\text{ K})\left(\frac{900}{100}\right)^{(1.4-1)/1.4} = 577.4\text{ K}$$

From the energy balance on the compressor,

$$\dot{W}_{\text{in}} = \dot{m}(h_{20} - h_{01})$$

or,

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_{02} - T_{01}) = (0.04\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(577.4 - 308.2)\text{K} = \mathbf{10.8 kW}$$

Discussion Note that the stagnation properties can be used conveniently in the energy equation.

12-10

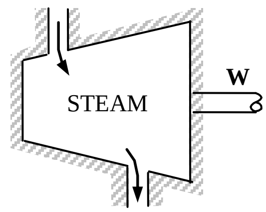
Solution The inlet stagnation temperature and pressure and the exit stagnation pressure of products of combustion flowing through a gas turbine are specified. The power output of the turbine is to be determined.

Assumptions **1** The expansion process is isentropic. **2** Products of combustion are ideal gases.

Properties The properties of products of combustion are c_p = 1.157 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.33.

Analysis The exit stagnation temperature T_{02} is determined to be

$$T_{02} = T_{01}\left(\frac{P_{02}}{P_{01}}\right)^{(k-1)/k} = (963.2\text{ K})\left(\frac{0.1}{0.75}\right)^{(1.33-1)/1.33} = 584.2\text{ K}$$



Also,

$$c_p = kc_v = k\left(c_p - R\right) \longrightarrow c_p = \frac{kR}{k-1}$$
$$= \frac{1.33\left(0.287\text{ kJ/kg}\cdot\text{K}\right)}{1.33-1}$$
$$= 1.157\text{ kJ/kg}\cdot\text{K}$$

From the energy balance on the turbine,

$$-w_{\text{out}} = (h_{20} - h_{01})$$

or, $$w_{\text{out}} = c_p(T_{01} - T_{02}) = (1.157\text{ kJ/kg}\cdot\text{K})(963.2 - 584.2)\text{K} = 438.5\text{ kJ/kg} \cong \mathbf{439\text{ kJ/kg}}$$

Discussion Note that the stagnation properties can be used conveniently in the energy equation.

One Dimensional Isentropic Flow

12-11C

Solution We are to determine if it is possible to accelerate a gas to supersonic velocity in a converging nozzle.

Analysis **No**, it is not possible.

Discussion The only way to do it is to have first a converging nozzle, and then a diverging nozzle.

12-12C

Solution We are to discuss what happens to several variables when a subsonic gas enters a diverging duct.

Analysis (*a*) The **velocity decreases**. (*b*), (*c*), (*d*) The **temperature, pressure, and density of the fluid increase**.

Discussion The velocity decrease is opposite to what happens in supersonic flow.

12-13C

Solution We are to discuss the pressure at the throats of two different converging-diverging nozzles.

Analysis **The pressures at the two throats are identical**.

Discussion Since the gas has the same stagnation conditions, it also has the same sonic conditions at the throat.

12-14C

Solution We are to discuss what happens to several variables when a supersonic gas enters a converging duct.

Analysis (*a*) The **velocity decreases**. (*b*), (*c*), (*d*) The **temperature, pressure, and density of the fluid increase**.

Discussion The velocity decrease is opposite to what happens in subsonic flow.

12-15C

Solution We are to discuss what happens to several variables when a supersonic gas enters a diverging duct.

Analysis (*a*) The **velocity increases**. (*b*), (*c*), (*d*) The **temperature, pressure, and density of the fluid decrease**.

Discussion The velocity increase is opposite to what happens in subsonic flow.

12-16C

Solution We are to discuss what happens to the exit velocity and mass flow rate through a converging nozzle at sonic exit conditions when the nozzle exit area is reduced.

Analysis (*a*) The **exit velocity remains constant at sonic speed**, (*b*) the **mass flow rate through the nozzle decreases because of the reduced flow area**.

Discussion Without a diverging portion of the nozzle, a converging nozzle is limited to sonic velocity at the exit.

12-17C

Solution We are to discuss what happens to several variables when a subsonic gas enters a converging duct.

Analysis (*a*) The **velocity increases**. (*b*), (*c*), (*d*) The **temperature, pressure, and density of the fluid decrease**.

Discussion The velocity increase is opposite to what happens in supersonic flow.

12-18

Solution Helium enters a converging-diverging nozzle at specified conditions. The lowest temperature and pressure that can be obtained at the throat of the nozzle are to be determined.

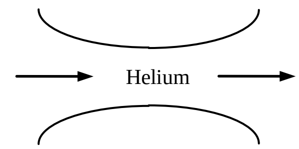
Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of helium are k = 1.667 and c_p = 5.1926 kJ/kg·K.

Analysis The lowest temperature and pressure that can be obtained at the throat are the critical temperature T^* and critical pressure P^*. First we determine the stagnation temperature T_0 and stagnation pressure P_0,

$$T_0 = T + \frac{V^2}{2c_p} = 800\text{ K} + \frac{(100\text{ m/s})^2}{2\times 5.1926\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 801\text{ K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.7\text{ MPa})\left(\frac{801\text{ K}}{800\text{ K}}\right)^{1.667/(1.667-1)} = 0.702\text{ MPa}$$



Thus,

$$T^* = T_0\left(\frac{2}{k+1}\right) = (801\text{ K})\left(\frac{2}{1.667+1}\right) = \mathbf{601\ K}$$

and

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.702\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{0.342 MPa}$$

Discussion These are the temperature and pressure that will occur at the throat when the flow past the throat is supersonic.

12-19

Solution The speed of an airplane and the air temperature are give. It is to be determined if the speed of this airplane is subsonic or supersonic.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is R = 0.287 kJ/kg·K. Its specific heat ratio at room temperature is k = 1.4.

Analysis The temperature is -50 + 273.15 = 223.15 K. The speed of sound is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(223.15\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)}\left(\frac{3.6\text{ km/h}}{1\text{ m/s}}\right) = 1077.97\text{ km/h}$$

and

$$\text{Ma} = \frac{V}{c} = \frac{1050\text{ km/h}}{1077.97\text{ km/h}} = 0.9741\text{ km/h} \cong \mathbf{0.974}$$

The speed of the airplane is **subsonic** since the Mach number is less than 1.

Discussion Subsonic airplanes stay sufficiently far from the Mach number of 1 to avoid the instabilities associated with transonic flights.

12-20

Solution The critical temperature, pressure, and density of air and helium are to be determined at specified conditions.

Assumptions Air and Helium are ideal gases with constant specific heats at room temperature.

Properties The properties of air at room temperature are R = 0.287 kJ/kg·K, k = 1.4, and c_p = 1.005 kJ/kg·K. The properties of helium at room temperature are R = 2.0769 kJ/kg·K, k = 1.667, and c_p = 5.1926 kJ/kg·K.

Analysis (*a*) Before we calculate the critical temperature T*, pressure P*, and density ρ*, we need to determine the stagnation temperature T_0, pressure P_0, and density ρ_0.

$$T_0 = 100°\text{C} + \frac{V^2}{2c_p} = 100 + \frac{(250\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 131.1°\text{C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\text{ kPa})\left(\frac{404.3\text{ K}}{373.2\text{ K}}\right)^{1.4/(1.4-1)} = 264.7\text{ kPa}$$

$$\rho_0 = \frac{P_0}{RT_0} = \frac{264.7\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(404.3\text{ K})} = 2.281\text{ kg/m}^3$$

Thus,

$$T^* = T_0\left(\frac{2}{k+1}\right) = (404.3\text{ K})\left(\frac{2}{1.4+1}\right) = \mathbf{337\,K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (264.7\text{ kPa})\left(\frac{2}{1.4+1}\right)^{1.4/(1.4-1)} = \mathbf{140\,kPa}$$

$$\rho^* = \rho_0\left(\frac{2}{k+1}\right)^{1/(k-1)} = (2.281\text{ kg/m}^3)\left(\frac{2}{1.4+1}\right)^{1/(1.4-1)} = \mathbf{1.45\ kg/m^3}$$

(*b*) For helium, $T_0 = T + \dfrac{V^2}{2c_p} = 40 + \dfrac{(300\text{ m/s})^2}{2\times 5.1926\text{ kJ/kg}\cdot°\text{C}}\left(\dfrac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 48.7°\text{C}$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\text{ kPa})\left(\frac{321.9\text{ K}}{313.2\text{ K}}\right)^{1.667/(1.667-1)} = 214.2\text{ kPa}$$

$$\rho_0 = \frac{P_0}{RT_0} = \frac{214.2\text{ kPa}}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(321.9\text{ K})} = 0.320\text{ kg/m}^3$$

Thus,

$$T^* = T_0\left(\frac{2}{k+1}\right) = (321.9\text{ K})\left(\frac{2}{1.667+1}\right) = \mathbf{241\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (200\text{ kPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{97.4\ kPa}$$

$$\rho^* = \rho_0\left(\frac{2}{k+1}\right)^{1/(k-1)} = (0.320\text{ kg/m}^3)\left(\frac{2}{1.667+1}\right)^{1/(1.667-1)} = \mathbf{0.208\ kg/m^3}$$

Discussion These are the temperature, pressure, and density values that will occur at the throat when the flow past the throat is supersonic.

12-21E

Solution Air flows through a duct at a specified state and Mach number. The velocity and the stagnation pressure, temperature, and density of the air are to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air are R = 0.06855 Btu/lbm·R = 0.3704 psia·ft^3/lbm·R and k = 1.4.

Analysis First, T = 320 + 459.67 = 779.67 K. The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.06855\text{ Btu/1bm}\cdot\text{R})(779.67\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = 1368.72\text{ ft/s}$$

Thus,

$$V = \text{Ma}\times c = (0.7)(1368.72\text{ ft/s}) = 958.10 \cong \mathbf{958\ \ ft/s}$$

Also,

$$\rho = \frac{P}{RT} = \frac{25\text{ psia}}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(779.67\text{ R})} = 0.086568\text{ 1bm/ft}^3$$

Then the stagnation properties are determined from

$$T_0 = T\left(1+\frac{(k-1)\text{Ma}^2}{2}\right) = (779.67\text{ R})\left(1+\frac{(1.4\text{-}1)(0.7)^2}{2}\right) = 856.08\text{ R} \cong \mathbf{856\ \ R}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (25\text{ psia})\left(\frac{856.08\text{ R}}{779.67\text{ R}}\right)^{1.4/(1.4-1)} = 34.678\text{ psia} \cong \mathbf{34.7\ \ psia}$$

$$\rho_0 = \rho\left(\frac{T_0}{T}\right)^{1/(k-1)} = (0.08656\ \ \text{1bm/ft}^3)\left(\frac{856.08\text{ R}}{779.67\text{ R}}\right)^{1/(1.4-1)} = 0.10936\text{ lbm/ft}^3 \cong \mathbf{0.109\ \ lbm/ft^3}$$

Discussion Note that the temperature, pressure, and density of a gas increases during a stagnation process.

12-22

Solution Air enters a converging-diverging nozzle at specified conditions. The lowest pressure that can be obtained at the throat of the nozzle is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The specific heat ratio of air at room temperature is k = 1.4.

Analysis The lowest pressure that can be obtained at the throat is the critical pressure P*, which is determined from

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (1200\text{ kPa})\left(\frac{2}{1.4+1}\right)^{1.4/(1.4-1)} = \mathbf{634\,kPa}$$

Discussion This is the pressure that occurs at the throat when the flow past the throat is supersonic.

12-23

Solution The Mach number of scramjet and the air temperature are given. The speed of the engine is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is R = 0.287 kJ/kg·K. Its specific heat ratio at room temperature is k = 1.4.

Analysis The temperature is -20 + 273.15 = 253.15 K. The speed of sound is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(253.15\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 318.93\text{ m/s}$$

and

$$V = c\text{Ma} = (318.93\text{ m/s})(7)\left(\frac{3.6\text{ km/h}}{1\text{ m/s}}\right) = 8037\text{ km/h} \cong \mathbf{8040\text{ km/h}}$$

Discussion Note that extremely high speed can be achieved with scramjet engines. We cannot justify more than three significant digits in a problem like this.

12-24E

Solution The Mach number of scramjet and the air temperature are given. The speed of the engine is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is R = 0.06855 Btu/lbm·R. Its specific heat ratio at room temperature is k = 1.4 .

Analysis The temperature is 0 + 459.67 = 459.67 R. The speed of sound is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.06855\text{ Btu/lbm}\cdot\text{R})(459.67\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)} = 1050.95\text{ ft/s}$$

and

$$V = c\text{Ma} = (1050.95\text{ ft/s})(7)\left(\frac{1\text{ mi/h}}{1.46667\text{ ft/s}}\right) = 5015.9\text{ mi/h} \cong \mathbf{5020\text{ mi/h}}$$

Discussion Note that extremely high speed can be achieved with scramjet engines. We cannot justify more than three significant digits in a problem like this.

12-25

Solution Air flows through a duct. The state of the air and its Mach number are specified. The velocity and the stagnation pressure, temperature, and density of the air are to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air at room temperature are R = 0.287 kPa.m^3/kg.K and k = 1.4.

Analysis The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(373.2\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 387.2\text{ m/s}$$

Thus,

$$V = \text{Ma}\times c = (0.8)(387.2\text{ m/s}) = \mathbf{310\ m/s}$$

Also,

$$\rho = \frac{P}{RT} = \frac{200\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(373.2\text{ K})} = 1.867\text{ kg/m}^3$$

Then the stagnation properties are determined from

$$T_0 = T\left(1+\frac{(k-1)\text{Ma}^2}{2}\right) = (373.2\text{ K})\left(1+\frac{(1.4\text{-}1)(0.8)^2}{2}\right) = \mathbf{421\ K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\text{ kPa})\left(\frac{421.0\text{ K}}{373.2\text{ K}}\right)^{1.4/(1.4-1)} = \mathbf{305\ kPa}$$

$$\rho_0 = \rho\left(\frac{T_0}{T}\right)^{1/(k-1)} = (1.867\text{ kg/m}^3)\left(\frac{421.0\text{ K}}{373.2\text{ K}}\right)^{1/(1.4-1)} = \mathbf{2.52\ kg/m^3}$$

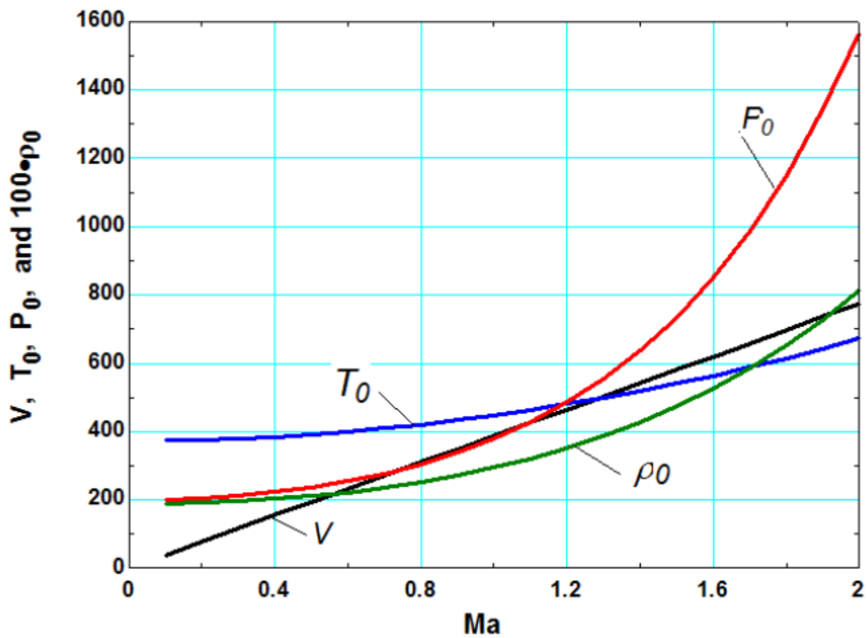
Discussion Note that both the pressure and temperature drop as the gas is accelerated as part of the internal energy of the gas is converted to kinetic energy.

12-26

Solution Problem 12-41 is reconsidered. The effect of Mach number on the velocity and stagnation properties as the Ma is varied from 0.1 to 2 are to be investigated, and the results are to be plotted.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
P=200
T=100+273.15
R=0.287
k=1.4
c=SQRT(k*R*T*1000)
Ma=V/c
rho=P/(R*T)

"Stagnation properties"
T0=T*(1+(k-1)*Ma^2/2)
P0=P*(T0/T)^(k/(k-1))
rho0=rho*(T0/T)^(1/(k-1))
```



Mach num. Ma	Velocity, V, m/s	Stag. Temp, T_0, K	Stag. Press, P_0, kPa	Stag. Density, ρ_0, kg/m^3
0.1	38.7	373.9	201.4	1.877
0.2	77.4	376.1	205.7	1.905
0.3	116.2	379.9	212.9	1.953
0.4	154.9	385.1	223.3	2.021
0.5	193.6	391.8	237.2	2.110
0.6	232.3	400.0	255.1	2.222
0.7	271.0	409.7	277.4	2.359
0.8	309.8	420.9	304.9	2.524
0.9	348.5	433.6	338.3	2.718
1.0	387.2	447.8	378.6	2.946
1.1	425.9	463.5	427.0	3.210
1.2	464.7	480.6	485.0	3.516
1.3	503.4	499.3	554.1	3.867
1.4	542.1	519.4	636.5	4.269
1.5	580.8	541.1	734.2	4.728
1.6	619.5	564.2	850.1	5.250
1.7	658.3	588.8	987.2	5.842
1.8	697.0	615.0	1149.2	6.511
1.9	735.7	642.6	1340.1	7.267
2.0	774.4	671.7	1564.9	8.118

Discussion Note that as Mach number increases, so does the flow velocity and stagnation temperature, pressure, and density.

12-27

Solution An aircraft is designed to cruise at a given Mach number, elevation, and the atmospheric temperature. The stagnation temperature on the leading edge of the wing is to be determined.

Assumptions Air is an ideal gas with constant specific heats at room temperature.

Properties The properties of air are R = 0.287 kPa.m^3/kg.K, c_p = 1.005 kJ/kg·K, and $k = 1.4$.

Analysis The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(236.15\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 308.0\text{ m/s}$$

Thus,

$$V = \text{Ma} \times c = (1.1)(308.0\text{ m/s}) = 338.8\text{ m/s}$$

Then,

$$T_0 = T + \frac{V^2}{2c_p} = 236.15 + \frac{(338.8\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{293K}$$

Discussion Note that the temperature of a gas increases during a stagnation process as the kinetic energy is converted to enthalpy.

12-28

Solution Quiescent carbon dioxide at a given state is accelerated isentropically to a specified Mach number. The temperature and pressure of the carbon dioxide after acceleration are to be determined.

Assumptions Carbon dioxide is an ideal gas with constant specific heats at room temperature.

Properties The specific heat ratio of the carbon dioxide at room temperature is $k = 1.288$.

Analysis The inlet temperature and pressure in this case is equivalent to the stagnation temperature and pressure since the inlet velocity of the carbon dioxide is said to be negligible. That is, $T_0 = T_\text{i}$ = 400 K and $P_0 = P_\text{i}$ = 1200 kPa. Then,

$$T = T_0\left(\frac{2}{2+(k-1)\text{Ma}^2}\right) = (600\text{ K})\left(\frac{2}{2+(1.288\text{-}1)(0.6)^2}\right) = 570.43\text{ K} \cong \mathbf{570\text{ K}}$$

and

$$P = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (1200\text{ kPa})\left(\frac{570.43\text{ K}}{600\text{ K}}\right)^{1.288/(1.288-1)} = 957.23\text{ K} \cong \mathbf{957\ \ kPa}$$

Discussion Note that both the pressure and temperature drop as the gas is accelerated as part of the internal energy of the gas is converted to kinetic energy.

Isentropic Flow Through Nozzles

12-29C

Solution We are to analyze if it is possible to accelerate a fluid to supersonic speeds with a velocity that is not sonic at the throat.

Analysis **No, if the flow in the throat is subsonic**. If the velocity at the throat is subsonic, the diverging section would act like a diffuser and decelerate the flow. **Yes, if the flow in the throat is already supersonic**, the diverging section would accelerate the flow to even higher Mach number.

Discussion In duct flow, the latter situation is not possible unless a second converging-diverging portion of the duct is located upstream, and there is sufficient pressure difference to choke the flow in the upstream throat.

12-30C

Solution We are to discuss what would happen if we add a diverging section to supersonic flow in a duct.

Analysis The fluid would **accelerate even further**, as desired.

Discussion This is the opposite of what would happen in subsonic flow.

12-31C

Solution We are to discuss the difference between Ma* and Ma.

Analysis Ma* is the **local velocity non-dimensionalized with respect to the sonic speed at the throat**, whereas Ma is the **local velocity non-dimensionalized with respect to the local sonic speed**.

Discussion The two are identical at the throat when the flow is choked.

12-32C

Solution We are to consider subsonic flow through a converging nozzle with critical pressure at the exit, and analyze the effect of lowering back pressure below the critical pressure.

Analysis (*a*) **No effect on velocity**. (*b*) **No effect on pressure**. (*c*) **No effect on mass flow rate**.

Discussion In this situation, the flow is already choked initially, so further lowering of the back pressure does not change anything upstream of the nozzle exit plane.

12-33C

Solution We are to compare the mass flow rates through two identical converging nozzles, but with one having a diverging section.

Analysis **If the back pressure is low enough so that sonic conditions exist at the throats, the mass flow rates in the two nozzles would be identical**. However, **if the flow is not sonic at the throat, the mass flow rate through the nozzle with the diverging section would be greater**, because it acts like a subsonic diffuser.

Discussion Once the flow is choked at the throat, whatever happens downstream is irrelevant to the flow upstream of the throat.

12-34C

Solution We are to discuss the hypothetical situation of hypersonic flow at the outlet of a converging nozzle.

Analysis Maximum flow rate through a converging nozzle is achieved when Ma = 1 at the exit of a nozzle. For all other Ma values the mass flow rate decreases. Therefore, **the mass flow rate would decrease if hypersonic velocities were achieved at the throat of a converging nozzle**.

Discussion Note that this is not possible unless the flow upstream of the converging nozzle is already hypersonic.

12-35C

Solution We are to consider subsonic flow through a converging nozzle, and analyze the effect of setting back pressure to critical pressure for a converging nozzle.

Analysis (*a*) The **exit velocity reaches the sonic speed**, (*b*) the **exit pressure equals the critical pressure**, and (*c*) the **mass flow rate reaches the maximum value**.

Discussion In such a case, we say that the flow is *choked*.

12-36C

Solution We are to discuss what happens to several variables in the diverging section of a subsonic converging-diverging nozzle.

Analysis (*a*) The **velocity decreases**, (*b*) the **pressure increases**, and (*c*) the **mass flow rate remains the same**.

Discussion Qualitatively, this is the same as what we are used to (in previous chapters) for incompressible flow.

12-37C

Solution We are to discuss what would happen if we add a diverging section to supersonic flow in a duct.

Analysis The fluid would **accelerate even further** instead of decelerating.

Discussion This is the opposite of what would happen in subsonic flow.

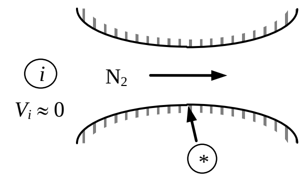
12-38

Solution Nitrogen enters a converging-diverging nozzle at a given pressure. The critical velocity, pressure, temperature, and density in the nozzle are to be determined.

Assumptions **1** Nitrogen is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of nitrogen are k = 1.4 and R = 0.2968 kJ/kg·K.

Analysis The stagnation pressure in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle,



$P_0 = P_i$ = 700 kPa

$T_0 = T_i$ = 400 K

$$\rho_0 = \frac{P_0}{RT_0} = \frac{700\text{ kPa}}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400\text{ K})} = 5.896\text{ kg/m}^3$$

Critical properties are those at a location where the Mach number is Ma = 1. From Table A-13 at Ma =1, we read T/T_0 =0.8333, P/P_0 = 0.5283, and ρ/ρ_0 = 0.6339. Then the critical properties become

$T^* = 0.8333T_0 = 0.8333(400\text{ K}) = \mathbf{333\ K}$

$P^* = 0.5283P_0 = 0.5283(700\text{ kPa}) = \mathbf{370\ MPa}$

$\rho^* = 0.6339\rho_0 = 0.6339(5.896\text{ kg/m}^3) = \mathbf{3.74\ kg/m^3}$

Also,

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.4)(0.2968\text{ kJ/kg}\cdot\text{K})(333\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{372\ m/s}$$

Discussion We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

12-39

Solution For an ideal gas, an expression is to be obtained for the ratio of the speed of sound where Ma = 1 to the speed of sound based on the stagnation temperature, c^*/c_0.

Analysis For an ideal gas the speed of sound is expressed as $c = \sqrt{kRT}$. Thus,

$$\frac{c^*}{c_0} = \frac{\sqrt{kRT^*}}{\sqrt{kRT_0}} = \left(\frac{T^*}{T_0}\right)^{1/2} = \left(\frac{\mathbf{2}}{\boldsymbol{k}\mathbf{+1}}\right)^{\mathbf{1/2}}$$

Discussion Note that a speed of sound changes the flow as the temperature changes.

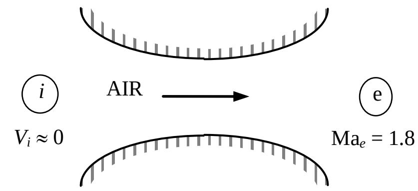
12-40

Solution Air enters a converging-diverging nozzle at a specified pressure. The back pressure that will result in a specified exit Mach number is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The specific heat ratio of air is k = 1.4.

Analysis The stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. It remains constant throughout the nozzle since the flow is isentropic,



$$P_0 = P_i = 1.2 \text{ MPa}$$

From Table A-13 at $\mathrm{Ma_e}$ =1.8, we read P_e/P_0 = 0.1740.

Thus, $P = 0.1740P_0 = 0.1740(1.2 \text{ MPa}) =$ **0.209 MPa** = **209 kPa**

Discussion If we solve this problem using the relations for compressible isentropic flow, the results would be identical.

12-41E

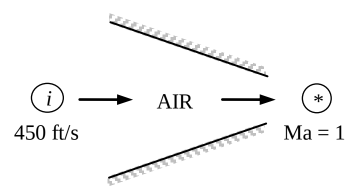
Solution Air enters a nozzle at specified temperature, pressure, and velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of air are k = 1.4 and c_p = 0.240 Btu/lbm·R (Table A-2Ea).

Analysis The properties of the fluid at the location where Ma =1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T + \frac{V_i^2}{2c_p} = 630 \text{ R} + \frac{(450 \text{ ft/s})^2}{2 \times 0.240 \text{ Btu/lbm}\cdot\text{R}}\left(\frac{1 \text{ Btu/1bm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 646.9 \text{ R}$$



$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (30 \text{ psia})\left(\frac{646.9\text{K}}{630\text{K}}\right)^{1.4/(1.4-1)} = 32.9 \text{ psia}$$

From Table A-13 (or from Eqs. 12-18 and 12-19) at Ma =1, we read T/T_0 =0.8333, P/P_0 = 0.5283. Thus,

$T = 0.8333T_0 = 0.8333(646.9 \text{ R}) =$ **539 R** and $P = 0.5283P_0 = 0.5283(32.9 \text{ psia}) =$ **17.4 psia**

Also,

$$c_i = \sqrt{kRT}_i = \sqrt{(1.4)(0.06855 \text{ Btu/1bm}\cdot\text{R})(630 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/1bm}}\right)} = 1230 \text{ ft/s} \quad \text{and}$$

$$\mathrm{Ma}_i = \frac{V_i}{c_i} = \frac{450 \text{ ft/s}}{1230 \text{ ft/s}} = 0.3657$$

From Table A-13 at this Mach number we read A_i/A^* = 1.7426. Thus the ratio of the throat area to the nozzle inlet area is

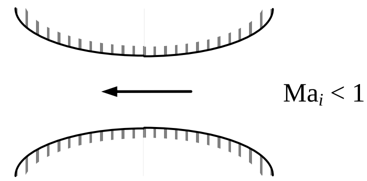
$$\frac{A^*}{A_i} = \frac{1}{1.7426} = \mathbf{0.574}$$

Discussion If we solve this problem using the relations for compressible isentropic flow, the results would be identical.

12-42

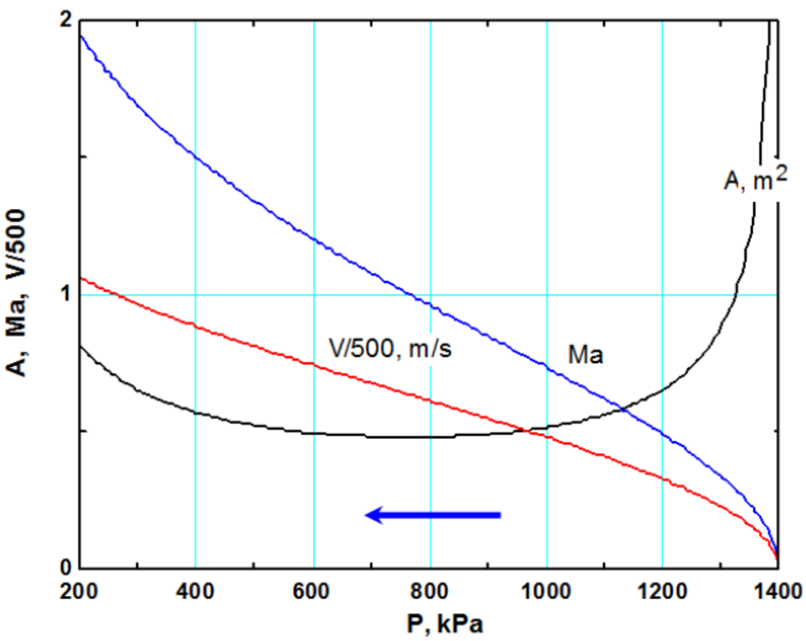
Solution For subsonic flow at the inlet, the variation of pressure, velocity, and Mach number along the length of the nozzle are to be sketched for an ideal gas under specified conditions.

Assumptions **1** The gas is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The flow is choked at the throat.



Analysis Using EES and CO_2 as the gas, we calculate and plot flow area *A*, velocity *V*, and Mach number Ma as the pressure drops from a stagnation value of 1400 kPa to 200 kPa. Note that the curve for *A* is related to the shape of the nozzle, with horizontal axis serving as the centerline. The EES equation window and the plot are shown below.

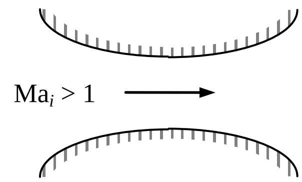
```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
P0=1400 "kPa"

T0=473 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
rho_norm=rho/rho_0 "Normalized density"
T=T0*(P/P0)^((k-1)/k)
Tnorm=T/T0  "Normalized temperature"
V=SQRT(2*Cp*(T0-T)*1000)
V_norm=V/500
A=m/(rho*V)*500
C=SQRT(k*R*T*1000)
Ma=V/C
```



Discussion We are assuming that the back pressure is sufficiently low that the flow is choked at the throat, and the flow downstream of the throat is supersonic without any shock waves. Mach number and velocity continue to rise right through the throat into the diverging portion of the nozzle, since the flow becomes supersonic.

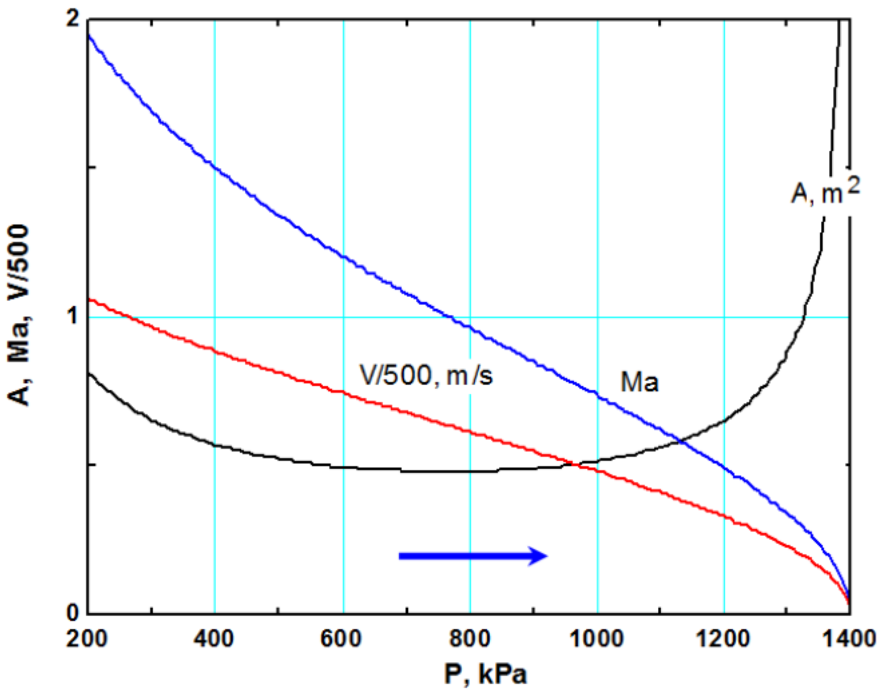
12-43

Solution We repeat the previous problem, but for supersonic flow at the inlet. The variation of pressure, velocity, and Mach number along the length of the nozzle are to be sketched for an ideal gas under specified conditions.



Analysis Using EES and CO_2 as the gas, we calculate and plot flow area A, velocity V, and Mach number Ma as the pressure rises from 200 kPa at a very high velocity to the stagnation value of 1400 kPa. Note that the curve for A is related to the shape of the nozzle, with horizontal axis serving as the centerline.

```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
P0=1400 "kPa"

T0=473 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
rho_norm=rho/rho_0 "Normalized density"
T=T0*(P/P0)^((k-1)/k)
Tnorm=T/T0  "Normalized temperature"
V=SQRT(2*Cp*(T0-T)*1000)
V_norm=V/500
A=m/(rho*V)*500
C=SQRT(k*R*T*1000)
Ma=V/C
```



Discussion Note that this problem is identical to the proceeding one, except the flow direction is reversed. In fact, when plotted like this, the plots are identical.

12-44

Solution It is to be explained why the maximum flow rate per unit area for a given ideal gas depends only on $P_0/\sqrt{T_0}$. Also for an ideal gas, a relation is to be obtained for the constant a in $\dot{m}_{\max}/A^* = a\left(P_0/\sqrt{T_0}\right)$.

Properties The properties of the ideal gas considered are R = 0.287 kPa.m^3/kg·K and k = 1.4.

Analysis The maximum flow rate is given by

$$\dot{m}_{\max} = A^* P_0 \sqrt{k/RT_0}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)} \quad \text{or} \quad \dot{m}_{\max}/A^* = \left(P_0/\sqrt{T_0}\right)\sqrt{k/R}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)}$$

For a given gas, k and R are fixed, and thus the mass flow rate depends on the parameter $P_0/\sqrt{T_0}$. Thus, $\dot{m}_{\max}/A^*$ can be expressed as $\dot{m}_{\max}/A^* = a\left(P_0/\sqrt{T_0}\right)$ where

$$a = \sqrt{k/R}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)} = \sqrt{\frac{1.4}{(0.287 \text{ kJ/kg.K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)}}\left(\frac{2}{1.4+1}\right)^{2.4/0.8} = \mathbf{0.0404\ (m/s)\sqrt{K}}$$

Discussion Note that when sonic conditions exist at a throat of known cross-sectional area, the mass flow rate is fixed by the stagnation conditions.

12-45

Solution An ideal gas is flowing through a nozzle. The flow area at a location where Ma = 1.8 is specified. The flow area where Ma = 0.9 is to be determined.

Assumptions Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The specific heat ratio is given to be k = 1.4.

Analysis The flow is assumed to be isentropic, and thus the stagnation and critical properties remain constant throughout the nozzle. The flow area at a location where Ma_2 = 0.9 is determined using A/A^* data from Table A-13 to be

$$\text{Ma}_1 = 1.8: \quad \frac{A_1}{A^*} = 1.4390 \longrightarrow A^* = \frac{A_1}{1.4390} = \frac{36 \text{ cm}^2}{1.4390} = 25.02 \text{ cm}^2$$

$$\text{Ma}_2 = 0.9: \quad \frac{A_2}{A^*} = 1.0089 \longrightarrow A_2 = (1.0089)A^* = (1.0089)(25.02 \text{ cm}^2) = \mathbf{25.2\ cm^2}$$

Discussion We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

12-46

Solution An ideal gas is flowing through a nozzle. The flow area at a location where Ma = 1.8 is specified. The flow area where Ma = 0.9 is to be determined.

Assumptions Flow through the nozzle is steady, one-dimensional, and isentropic.

Analysis The flow is assumed to be isentropic, and thus the stagnation and critical properties remain constant throughout the nozzle. The flow area at a location where $\mathrm{Ma}_2 = 0.9$ is determined using the A/A^* relation,

$$\frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left\{\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right\}^{(k+1)/2(k-1)}$$

For $k = 1.33$ and $\mathrm{Ma}_1 = 1.8$:

$$\frac{A_1}{A^*} = \frac{1}{1.8}\left\{\left(\frac{2}{1.33+1}\right)\left(1+\frac{1.33-1}{2}1.8^2\right)\right\}^{2.33/2\times 0.33} = 1.4696$$

and, $$A^* = \frac{A_1}{2.570} = \frac{36\ \mathrm{cm}^2}{1.4696} = 24.50\ \mathrm{cm}^2$$

For $k = 1.33$ and $\mathrm{Ma}_2 = 0.9$:

$$\frac{A_2}{A^*} = \frac{1}{0.9}\left\{\left(\frac{2}{1.33+1}\right)\left(1+\frac{1.33-1}{2}0.9^2\right)\right\}^{2.33/2\times 0.33} = 1.0091$$

and $$A_2 = (1.0091)A^* = (1.0091)(24.50\ \mathrm{cm}^2) = \mathbf{24.7\ cm^2}$$

Discussion Note that the compressible flow functions in Table A-13 are prepared for $k = 1.4$, and thus they cannot be used to solve this problem.

12-47E

Solution Air enters a converging-diverging nozzle at a specified temperature and pressure with low velocity. The pressure, temperature, velocity, and mass flow rate are to be calculated in the specified test section.

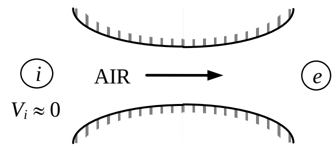
Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of air are k = 1.4 and R = 0.06855 Btu/lbm·R = 0.3704 psia·ft^3/lbm·R.

Analysis The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.

$P_0 = P_i = 150$ psia and $T_0 = T_i = 100°\text{F} \approx 560$ R

Then,



$$T_e = T_0\left(\frac{2}{2+(k-1)\text{Ma}^2}\right) = (560\text{ R})\left(\frac{2}{2+(1.4\text{-}1)2^2}\right) = \mathbf{311R}$$

$$P_e = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (150\text{ psia})\left(\frac{311}{560}\right)^{1.4/0.4} = \mathbf{19.1\ psia}$$

$$\rho_e = \frac{P_e}{RT_e} = \frac{19.1\text{ psia}}{(0.3704\text{ psia.ft}^3/\text{1bm}\cdot\text{R})(311\text{ R})} = 0.166\text{ 1bm/ft}^3$$

The nozzle exit velocity can be determined from $V_e = \text{Ma}_e c_e$, where c_e is the speed of sound at the exit conditions,

$$V_e = \text{Ma}_e c_e = \text{Ma}_e\sqrt{kRT_e} = (2)\sqrt{(1.4)(0.06855\text{ Btu/1bm}\cdot\text{R})(311\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = 1729\text{ ft/s} \cong \mathbf{1730\ \ ft/s}$$

Finally,

$$\dot{m} = \rho_e A_e V_e = (0.\ 166\text{ 1bm/ft}^3)(5\text{ ft}^2)(1729\text{ ft/s}) = 1435\text{ lbm/s} \cong \mathbf{1440\ lbm/s}$$

Discussion Air must be very dry in this application because the exit temperature of air is extremely low, and any moisture in the air will turn to ice particles.

12-48

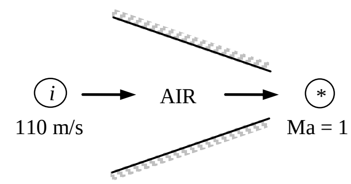
Solution Air enters a nozzle at specified temperature, pressure, and velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of air are k = 1.4 and c_p = 1.005 kJ/kg·K.

Analysis The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T_i + \frac{V_i^2}{2c_p} = 420\text{ K} + \frac{(110\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 426.02$$



and

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.5\text{ MPa})\left(\frac{426.02\text{ K}}{420\text{ K}}\right)^{1.4/(1.4-1)} = 0.52554\text{ MPa}$$

From Table A-13 (or from Eqs. 12-18 and 12-19) at Ma = 1, we read T/T_0 = 0.8333, P/P_0 = 0.5283. Thus,

$T = 0.8333T_0$ = 0.8333(426.02 K) = 355.00 K ≈ **355 K**

and

$P = 0.5283P_0$ = 0.5283(0.52554 MPa) = 0.27764 MPa ≈ **0.278 MPa** = **278 kPa**

Also,

$$c_i = \sqrt{kRT}_i = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(420\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 410.799\text{ m/s}$$

and

$$\text{Ma}_i = \frac{V_i}{c_i} = \frac{110\text{ m/s}}{410.799\text{ m/s}} = 0.2678$$

$$\text{Ma}_i = \frac{V_i}{c_i} = \frac{150\text{ m/s}}{410.799\text{ m/s}} = 0.3651$$

From Table A-13 at this Mach number we read A_i/A^* = 2.3343. Thus the ratio of the throat area to the nozzle inlet area is

$$\frac{A^*}{A} = \frac{1}{2.3343} = 0.42839 \cong \mathbf{0.428}$$

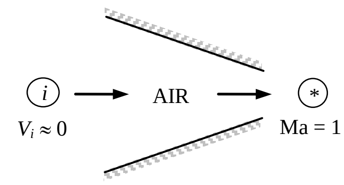
Discussion We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

12-49

Solution Air enters a nozzle at specified temperature and pressure with low velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The specific heat ratio of air is k = 1.4.



Analysis The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. The stagnation temperature and pressure in this case are identical to the inlet temperature and pressure since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.

$T_0 = T_i$ = 420 K and $P_0 = P_i$ = 0.5 MPa

From Table A-13 (or from Eqs. 12-18 and 12-19) at Ma =1, we read T/T_0 =0.8333, P/P_0 = 0.5283. Thus,

$T = 0.8333T_0$ = 0.8333(420 K) = **350 K** and $P = 0.5283P_0$ = 0.5283(0.5 MPa) = **0.264 MPa**

The Mach number at the nozzle inlet is Ma = 0 since $V_i \cong 0$. From Table A-13 at this Mach number we read $A_i/A^* = \infty$. Thus the ratio of the throat area to the nozzle inlet area is $\dfrac{A^*}{A_i} = \dfrac{1}{\infty} = \mathbf{0}$.

Discussion If we solve this problem using the relations for compressible isentropic flow, the results would be identical.

12-50

Solution Air enters a converging nozzle at a specified temperature and pressure with low velocity. The exit pressure, the exit velocity, and the mass flow rate versus the back pressure are to be calculated and plotted.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

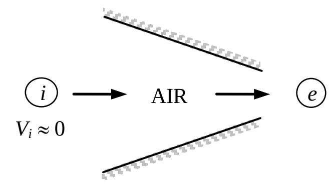
Properties The properties of air are $k = 1.4$, $R = 0.287$ kJ/kg·K, and $c_p = 1.005$ kJ/kg·K.

Analysis The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.,

$$P_0 = P_\mathrm{i} = 900 \text{ kPa}$$

$$T_0 = T_\mathrm{i} = 400 \text{ K}$$

The critical pressure is determined to be



$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (900 \text{ kPa})\left(\frac{2}{1.4+1}\right)^{1.4/0.4} = 475.5 \text{ kPa}$$

Then the pressure at the exit plane (throat) will be

$$P_\mathrm{e} = P_\mathrm{b} \quad \text{for} \quad P_\mathrm{b} \geq 475.5 \text{ kPa}$$

$$P_\mathrm{e} = P^* = 475.5 \text{ kPa} \quad \text{for} \quad P_\mathrm{b} < 475.5 \text{ kPa (choked flow)}$$

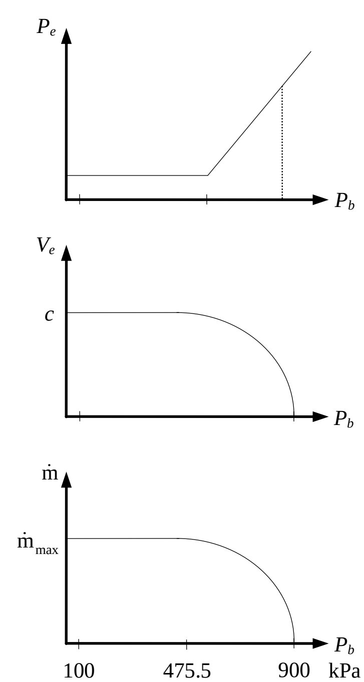
Thus the back pressure will not affect the flow when $100 < P_\mathrm{b} < 475.5$ kPa. For a specified exit pressure P_e, the temperature, the velocity and the mass flow rate can be determined from

Temperature $$T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (400 \text{ K})\left(\frac{P_\mathrm{e}}{900}\right)^{0.4/1.4}$$

Velocity $$V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.005 \text{ kJ/kg}\cdot\text{K})(400 - \mathrm{T_e})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)}$$

Density $$\rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.287 \text{ kPa}\cdot\text{m}^3/kg\cdot K)T_e}$$

Mass flow rate $$\dot{m} = \rho_e V_e A_e = \rho_e V_e (0.001 \text{ m}^2)$$



The results of the calculations are tabulated as

P_b, kPa	P_e, kPa	T_e, K	V_e, m/s	ρ_e, kg/m^3	$\dot{m}$, kg/s
900	900	400	0	7.840	0
800	800	386.8	162.9	7.206	1.174
700	700	372.3	236.0	6.551	1.546
600	600	356.2	296.7	5.869	1.741
500	500	338.2	352.4	5.151	1.815
475.5	475.5	333.3	366.2	4.971	1.820
400	475.5	333.3	366.2	4.971	1.820
300	475.5	333.3	366.2	4.971	1.820
200	475.5	333.3	366.2	4.971	1,820
100	475.5	333.3	366.2	4.971	1.820

Discussion We see from the plots that once the flow is choked at a back pressure of 475.5 kPa, the mass flow rate remains constant regardless of how low the back pressure gets.

12-51

Solution We are to reconsider the previous problem. Using EES (or other) software, we are to solve the problem for the inlet conditions of 0.8 MPa and 1200 K.

Analysis Air at 800 kPa, 1200 K enters a converging nozzle with a negligible velocity. The throat area of the nozzle is 10 cm2. Assuming isentropic flow, calculate and plot the exit pressure, the exit velocity, and the mass flow rate versus the back pressure P_b for 0.8>= P_b >=0.1 MPa.

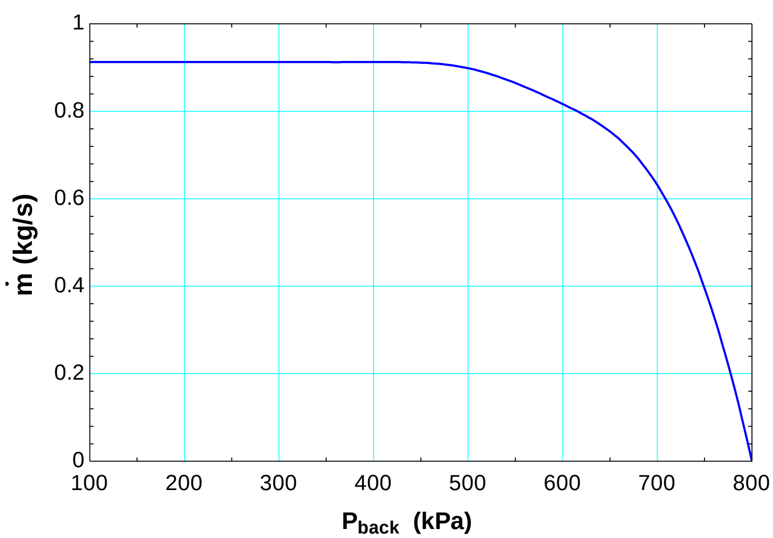
```
Procedure ExitPress(P_back,P_crit : P_exit, Condition$)
If  (P_back>=P_crit)  then
    P_exit:=P_back                                   "Unchoked Flow Condition"
    Condition$:='unchoked'
else
    P_exit:=P_crit                                   "Choked Flow Condition"
    Condition$:='choked'
Endif
End

Gas$='Air'
A_cm2=10  "Throat area, cm2"
P_inlet =800"kPa"
T_inlet= 1200"K"
"P_back =422.7"  "kPa"

A_exit = A_cm2*Convert(cm^2,m^2)
C_p=specheat(Gas$,T=T_inlet)
C_p-C_v=R
k=C_p/C_v
M=MOLARMASS(Gas$)                                    "Molar mass of  Gas$"
R= 8.314/M                                           "Gas constant for Gas$"

"Since the inlet velocity is negligible, the stagnation temperature = T_inlet;
and, since the nozzle is isentropic, the stagnation pressure = P_inlet."

P_o=P_inlet                                          "Stagnation pressure"
T_o=T_inlet                                          "Stagnation temperature"
P_crit /P_o=(2/(k+1))^(k/(k-1))                      "Critical pressure from Eq. 16-22"
Call ExitPress(P_back,P_crit : P_exit, Condition$)

T_exit /T_o=(P_exit/P_o)^((k-1)/k)                   "Exit temperature for isentopic flow, K"

V_exit ^2/2=C_p*(T_o-T_exit)*1000                    "Exit velocity, m/s"

Rho_exit=P_exit/(R*T_exit)                           "Exit density, kg/m3"

m_dot=Rho_exit*V_exit*A_exit                         "Nozzle mass flow rate, kg/s"

"If you wish to redo the plots, hide the diagram window and remove the { } from
the first 4 variables just under the procedure.  Next set the desired range of
back pressure in the parametric table.  Finally, solve the table (F3).  "
```

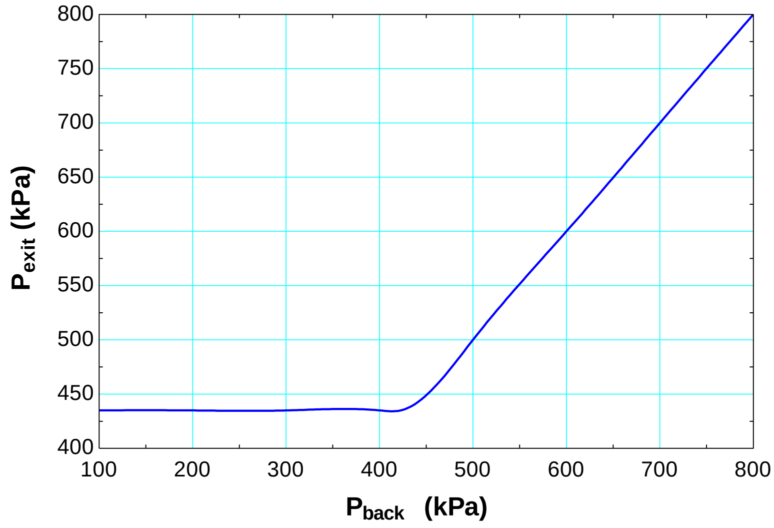
The table of results and the corresponding plot are provided below.

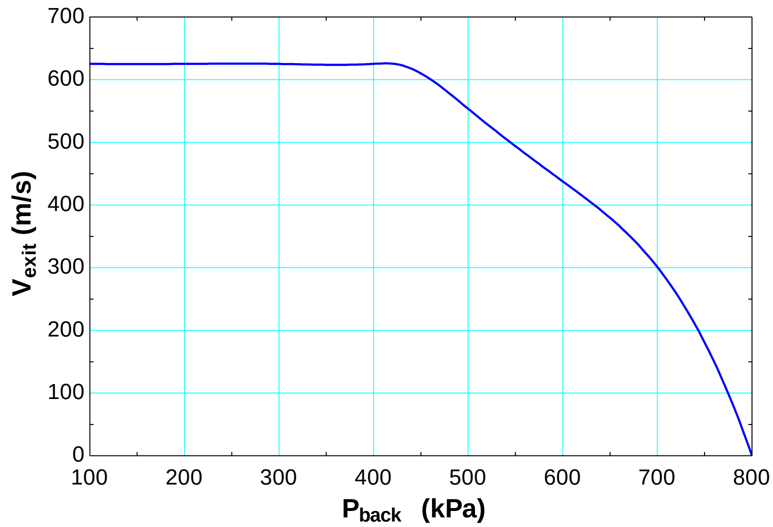
EES SOLUTION

A_cm2=10
A_exit=0.001
Condition$='choked'
C_p=1.208
C_v=0.9211
Gas$='Air'
k=1.312
M=28.97
m_dot=0.9124
P_back=422.7
P_crit=434.9
P_exit=434.9
P_inlet=800
P_o=800
R=0.287
Rho_exit=1.459
T_exit=1038
T_inlet=1200
T_o=1200
V_exit=625.2

P_{back}[kPa]	P_{exit} [kPa]	V_{exit} [m/s]	m [kg/s]	T_{exit} [K]	ρ_{exit} [kg/m^3]
100	434.9	625.2	0.9124	1038	1.459
200	434.9	625.2	0.9124	1038	1.459
300	434.9	625.2	0.9124	1038	1.459
400	434.9	625.2	0.9124	1038	1.459
422.7	434.9	625.2	0.9124	1038	1.459
500	500	553.5	0.8984	1073	1.623
600	600	437.7	0.8164	1121	1.865
700	700	300.9	0.6313	1163	2.098
800	800	0.001523	0.000003538	1200	2.323







Discussion We see from the plot that once the flow is choked at a back pressure of 422.7 kPa, the mass flow rate remains constant regardless of how low the back pressure gets.

Shock Waves and Expansion Waves

12-52C

Solution We are to discuss the applicability of the isentropic flow relations across shocks and expansion waves.

Analysis The isentropic relations of ideal gases are ***not*** **applicable for flows across (*a*) normal shock waves and (*b*) oblique shock waves**, but they ***are*** **applicable for flows across (*c*) Prandtl-Meyer expansion waves**.

Discussion Flow across any kind of shock wave involves irreversible losses – hence, it cannot be isentropic.

12-53C

Solution We are to discuss the states on the Fanno and Rayleigh lines.

Analysis The *Fanno line* represents the **states that satisfy the conservation of mass and energy equations**. The *Rayleigh line* represents the **states that satisfy the conservation of mass and momentum equations**. The *intersections* points of these lines represent the **states that satisfy the conservation of mass, energy, and momentum equations**.

Discussion *T-s* diagrams are quite helpful in understanding these kinds of flows.

12-54C

Solution We are to analyze a claim about oblique shock analysis.

Analysis **Yes, the claim is correct**. Conversely, normal shocks can be thought of as special oblique shocks in which the shock angle is $\beta = \pi/2$, or 90°.

Discussion The component of flow in the direction normal to the oblique shock acts exactly like a normal shock. We can think of the flow parallel to the oblique shock as "going along for the ride" – it does not affect anything.

12-55C

Solution We are to discuss the effect of a normal shock wave on several properties.

Analysis (*a*) **velocity decreases**, (*b*) **static temperature increases**, (*c*) **stagnation temperature remains the same**, (*d*) **static pressure increases**, and (*e*) **stagnation pressure decreases**.

Discussion In addition, the Mach number goes from supersonic (Ma > 1) to subsonic (Ma < 1).

12-56C

Solution We are to discuss the formation of oblique shocks and how they differ from normal shocks.

Analysis *Oblique shocks* occur when **a gas flowing at supersonic speeds strikes a flat or inclined surface**. Normal shock waves are perpendicular to flow whereas inclined shock waves, as the name implies, are typically **inclined relative to the flow direction**. Also, normal shocks form a straight line whereas **oblique shocks can be straight or curved**, depending on the surface geometry.

Discussion In addition, while a normal shock must go from supersonic (Ma > 1) to subsonic (Ma < 1), the Mach number downstream of an oblique shock can be either supersonic or subsonic.

12-57C

Solution We are to discuss whether the flow upstream and downstream of an oblique shock needs to be supersonic.

Analysis **Yes**, the *upstream flow* has to be supersonic for an oblique shock to occur. **No**, the flow *downstream* of an oblique shock can be subsonic, sonic, and even supersonic.

Discussion The latter is not true for normal shocks. For a normal shock, the flow must always go from supersonic (Ma > 1) to subsonic (Ma < 1).

12-58C

Solution We are to determine if Ma downstream of a normal shock can be supersonic.

Analysis **No**, the second law of thermodynamics requires the flow after the shock to be subsonic.

Discussion A normal shock wave always goes from supersonic to subsonic in the flow direction.

12-59C

Solution We are to discuss shock detachment at the nose of a 2-D wedge-shaped body.

Analysis **When the wedge half-angle δ is greater than the maximum deflection angle θ_{max}**, the shock becomes curved and detaches from the nose of the wedge, forming what is called a *detached oblique shock* or a *bow wave*. The numerical value of the shock angle at the nose is $\beta = 90^o$.

Discussion When δ is less than θ_{max}, the oblique shock is attached to the nose.

12-60C

Solution We are to discuss the shock at the nose of a rounded body in supersonic flow.

Analysis When supersonic flow impinges on a blunt body like the rounded nose of an aircraft, the wedge half-angle δ at the nose is 90^o, and an attached oblique shock cannot exist, regardless of Mach number. Therefore, **a detached oblique shock must occur in front of *all* such blunt-nosed bodies**, whether two-dimensional, axisymmetric, or fully three-dimensional.

Discussion Since $\delta = 90^o$ at the nose, δ is always greater than θ_{max}, regardless of Ma or the shape of the rest of the body.

12-61C

Solution We are to discuss if a shock wave can develop in the converging section of a C-V nozzle.

Analysis **No**, because the flow must be supersonic before a shock wave can occur. The flow in the converging section of a nozzle is always subsonic.

Discussion A normal shock (if it is to occur) would occur in the supersonic (diverging) section of the nozzle.

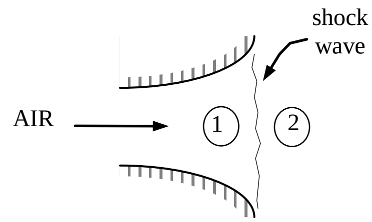
12-62

Solution Air flowing through a nozzle experiences a normal shock. Various properties are to be calculated before and after the shock.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air at room temperature are k= 1.4, R = 0.287 kJ/kg·K, and c_p = 1.005 kJ/kg·K.

Analysis The stagnation temperature and pressure before the shock are



$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 230 + \frac{(815\text{ m/s})^2}{2(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 560.5\text{ K}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (26\text{ kPa})\left(\frac{560.5\text{ K}}{230\text{ K}}\right)^{1.4/(1.4-1)} = 587.3\text{ kPa}$$

The velocity and the Mach number before the shock are determined from

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(230\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{304.0 m/s}$$

and

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{815\text{ m/s}}{304.0\text{ m/s}} = \mathbf{2.681}$$

The fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-14. For Ma_1 = 2.681 we read

$$\text{Ma}_2 = \mathbf{0.4972}, \quad \frac{P_{02}}{P_1} = 9.7330, \quad \frac{P_2}{P_1} = 8.2208, \quad \text{and} \quad \frac{T_2}{T_1} = 2.3230$$

Then the stagnation pressure P_{02} , static pressure P_2 , and static temperature T_2 , are determined to be

$P_{02} = 9.7330P_1$ = (9.7330)(26 kPa) = **253.1 kPa**

$P_2 = 8.2208P_1$ = (8.2208)(26 kPa) = **213.7 kPa**

$T_2 = 2.3230T_1$ = (2.3230)(230 K) = **534.3 K**

The air velocity after the shock can be determined from $V_2 = \text{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.4972)\sqrt{(1.4)(0.287\text{ kJ/kg.K})(534.3\text{K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{230.4 m/s}$$

Discussion This problem could also be solved using the relations for compressible flow and normal shock functions. The results would be identical.

12-63

Solution Air flowing through a nozzle experiences a normal shock. The entropy change of air across the normal shock wave is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air at room temperature are R = 0.287 kJ/kg·K and c_p = 1.005 kJ/kg·K.

Analysis The entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})\ln(2.3230) - (0.287\text{ kJ/kg}\cdot\text{K})\ln(8.2208)$$
$$= \mathbf{0.242\ kJ/kg\cdot K}$$

Discussion A shock wave is a highly dissipative process, and the entropy generation is large during shock waves.

12-64

Solution For an ideal gas flowing through a normal shock, a relation for V_2/V_1 in terms of k, Ma_1, and Ma_2 is to be developed.

Analysis The conservation of mass relation across the shock is $\rho_1 V_1 = \rho_2 V_2$ and it can be expressed as

$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{P_1/RT_1}{P_2/RT_2} = \left(\frac{P_1}{P_2}\right)\left(\frac{T_2}{T_1}\right)$$

From Eqs. 12-35 and 12-38,

$$\frac{V_2}{V_1} = \left(\frac{1+k\text{Ma}_2^2}{1+k\text{Ma}_1^2}\right)\left(\frac{1+\text{Ma}_1^2(k-1)/2}{1+\text{Ma}_2^2(k-1)/2}\right)$$

Discussion This is an important relation as it enables us to determine the velocity ratio across a normal shock when the Mach numbers before and after the shock are known.

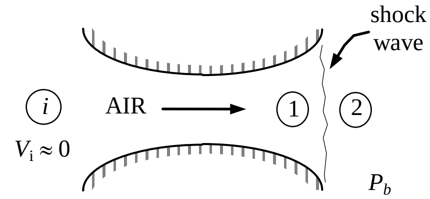
12-65

Solution Air enters a converging-diverging nozzle at a specified state. The required back pressure that produces a normal shock at the exit plane is to be determined for the specified nozzle geometry.

Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs. **3** The shock wave occurs at the exit plane.

Analysis The inlet stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. Since the flow before the shock to be isentropic,

$$P_{01} = P_i = 2 \text{ MPa}$$



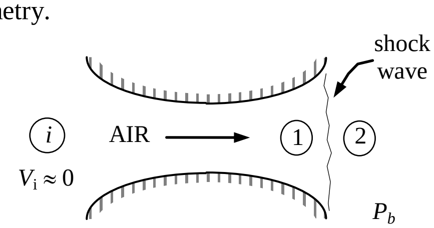
It is specified that A/A^* =3.5. From Table A-13, Mach number and the pressure ratio which corresponds to this area ratio are the Ma_1 =2.80 and P_1/P_{01} = 0.0368. The pressure ratio across the shock for this Ma_1 value is, from Table A-14, P_2/P_1 = 8.98. Thus the back pressure, which is equal to the static pressure at the nozzle exit, must be

$$P_2 = 8.98P_1 = 8.98\times0.0368P_{01} = 8.98\times0.0368\times(2 \text{ MPa}) = \mathbf{0.661\ MPa}$$

Discussion We can also solve this problem using the relations for compressible flow and normal shock functions. The results would be identical.

12-66

Solution Air enters a converging-diverging nozzle at a specified state. The required back pressure that produces a normal shock at the exit plane is to be determined for the specified nozzle geometry.



Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Analysis The inlet stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. Since the flow before the shock to be isentropic,

$$P_{0x} = P_i = 2 \text{ MPa}$$

It is specified that A/A^* = 2. From Table A-13, the Mach number and the pressure ratio which corresponds to this area ratio are the Ma_1 =2.20 and P_1/P_{01} = 0.0935. The pressure ratio across the shock for this M_1 value is, from Table A-14, P_2/P_1 = 5.48. Thus the back pressure, which is equal to the static pressure at the nozzle exit, must be

$$P_2 = 5.48P_1 = 5.48\times0.0935P_{01} = 5.48\times0.0935\times(2 \text{ MPa}) = \mathbf{1.02\ MPa}$$

Discussion We can also solve this problem using the relations for compressible flow and normal shock functions. The results would be identical.

12-67E

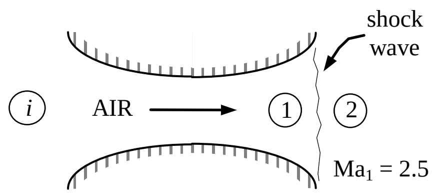
Solution Air flowing through a nozzle experiences a normal shock. Effect of the shock wave on various properties is to be determined. Analysis is to be repeated for helium

Assumptions **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air are k = 1.4 and R = 0.06855 Btu/lbm·R, and the properties of helium are k = 1.667 and R = 0.4961 Btu/lbm·R.

Analysis The air properties upstream the shock are

$\text{Ma}_1 = 2.5$, $P_1 = 10$ psia, and $T_1 = 440.5$ R



Fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-14. For $\text{Ma}_1 = 2.5$,

$$\text{Ma}_2 = \mathbf{0.513},\ \frac{P_{02}}{P_1} = 8.5262,\ \frac{P_2}{P_1} = 7.125,\ \text{and}\ \frac{T_2}{T_1} = 2.1375$$

Then the stagnation pressure P_{02}, static pressure P_2, and static temperature T_2, are determined to be

$P_{02} = 8.5262P_1 = (8.5262)(10\text{ psia}) =$ **85.3 psia**

$P_2 = 7.125P_1 = (7.125)(10\text{ psia}) =$ **71.3 psia**

$T_2 = 2.1375T_1 = (2.1375)(440.5\text{ R}) =$ **942 R**

The air velocity after the shock can be determined from $V_2 = \text{Ma}_2c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.513)\sqrt{(1.4)(0.06855\text{ Btu/1bm}\cdot\text{R})(941.6\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = \mathbf{772\ ft/s}$$

We now repeat the analysis for helium. This time we cannot use the tabulated values in Table A-14 since k is not 1.4. Therefore, we have to calculate the desired quantities using the analytical relations,

$$\text{Ma}_2 = \left(\frac{\text{Ma}_1^2 + 2/(k-1)}{2\text{Ma}_1^2 k/(k-1) - 1}\right)^{1/2} = \left(\frac{2.5^2 + 2/(1.667-1)}{2\times 2.5^2 \times 1.667/(1.667-1) - 1}\right)^{1/2} = \mathbf{0.553}$$

$$\frac{P_2}{P_1} = \frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2} = \frac{1 + 1.667\times 2.5^2}{1 + 1.667\times 0.553^2} = 7.5632$$

$$\frac{T_2}{T_1} = \frac{1 + \text{Ma}_1^2(k-1)/2}{1 + \text{Ma}_2^2(k-1)/2} = \frac{1 + 2.5^2(1.667-1)/2}{1 + 0.553^2(1.667-1)/2} = 2.7989$$

$$\frac{P_{02}}{P_1} = \left(\frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2}\right)\left(1 + (k-1)\text{Ma}_2^2/2\right)^{k/(k-1)}$$

$$= \left(\frac{1 + 1.667\times 2.5^2}{1 + 1.667\times 0.553^2}\right)\left(1 + (1.667-1)\times 0.553^2/2\right)^{1.667/0.667} = 9.641$$

Thus, $P_{02} = 11.546P_1 = (11.546)(10\text{ psia}) =$ **115 psia**

$P_2 = 7.5632P_1 = (7.5632)(10\text{ psia}) =$ **75.6 psia**

$T_2 = 2.7989T_1 = (2.7989)(440.5\text{ R}) =$ **1233 R**

$$V_2 = \text{Ma}_2c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.553)\sqrt{(1.667)(0.4961\text{ Btu/1bm.R})(1232.9\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = \mathbf{2794\ ft/s}$$

Discussion This problem could also be solved using the relations for compressible flow and normal shock functions. The results would be identical.

12-68E

Solution We are to reconsider Prob. 12-67E. Using EES (or other) software, we are to study the effects of both air and helium flowing steadily in a nozzle when there is a normal shock at a Mach number in the range 2 < Mx < 3.5. In addition to the required information, we are to calculate the entropy change of the air and helium across the normal shock, and tabulate the results in a parametric table.

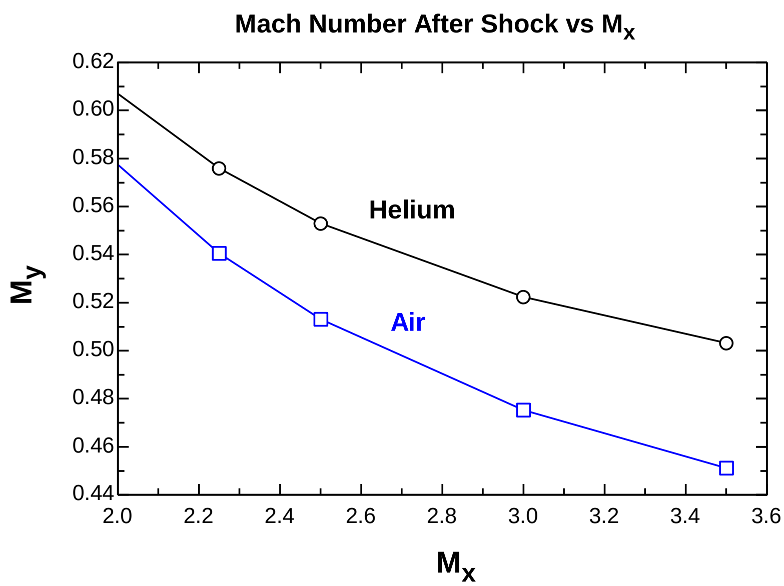
Analysis We use EES to calculate the entropy change of the air and helium across the normal shock. The results are given in the Parametric Table for 2 < M_x < 3.5.

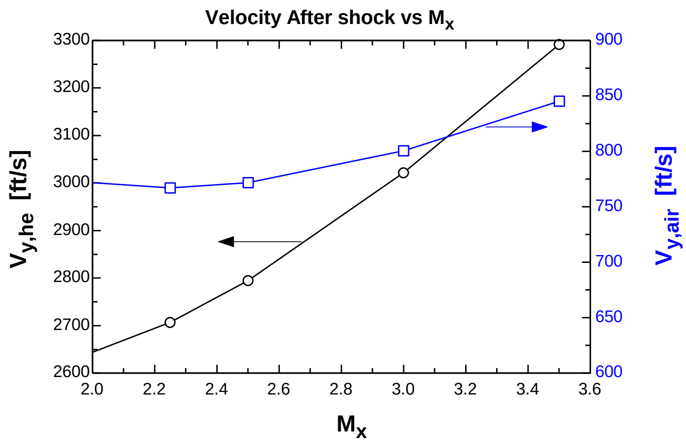
```
Procedure NormalShock(M_x,k:M_y,PyOPx, TyOTx,RhoyORhox, PoyOPox, PoyOPx)
    If M_x < 1 Then
            M_y = -1000;PyOPx=-1000;TyOTx=-1000;RhoyORhox=-1000
            PoyOPox=-1000;PoyOPx=-1000
    else
            M_y=sqrt( (M_x^2+2/(k-1)) / (2*M_x^2*k/(k-1)-1) )
            PyOPx=(1+k*M_x^2)/(1+k*M_y^2)
            TyOTx=( 1+M_x^2*(k-1)/2  )/(1+M_y^2*(k-1)/2  )
            RhoyORhox=PyOPx/TyOTx
            PoyOPox=M_x/M_y*(  (1+M_y^2*(k-1)/2)/ (1+M_x^2*(k-1)/2)   )^((k+1)/(2*(k-1)))
            PoyOPx=(1+k*M_x^2)*(1+M_y^2*(k-1)/2)^(k/(k-1))/(1+k*M_y^2)
    Endif
End

Function ExitPress(P_back,P_crit)
If P_back>=P_crit then ExitPress:=P_back     "Unchoked Flow Condition"
If P_back<P_crit then ExitPress:=P_crit      "Choked Flow Condition"
End

Procedure GetProp(Gas$:Cp,k,R) "Cp and k data are from Text Table A.2E"
   M=MOLARMASS(Gas$)           "Molar mass of  Gas$"
   R= 1545/M                   "Particular gas constant for Gas$, ft-lbf/lbm-R"
                               "k = Ratio of Cp to Cv"
                               "Cp = Specific heat at constant pressure"
   if Gas$='Air' then
                               Cp=0.24"Btu/lbm-R"; k=1.4
   endif
   if Gas$='CO2' then
                               Cp=0.203"Btu/lbm_R"; k=1.289
   endif
   if Gas$='Helium' then
                               Cp=1.25"Btu/lbm-R"; k=1.667
   endif
End

"Variable Definitions:"
"M = flow Mach Number"
"P_ratio = P/P_o  for compressible, isentropic flow"
"T_ratio = T/T_o for compressible, isentropic flow"
"Rho_ratio= Rho/Rho_o for compressible, isentropic flow"
"A_ratio=A/A* for compressible, isentropic flow"
"Fluid properties before the shock are denoted with a subscript x"
"Fluid properties after the shock are denoted with a subscript y"
"M_y = Mach Number down stream of normal shock"
"PyOverPx= P_y/P_x  Pressue ratio across normal shock"
"TyOverTx =T_y/T_x  Temperature ratio across normal shock"
"RhoyOverRhox=Rho_y/Rho_x Density ratio across normal shock"
"PoyOverPox = P_oy/P_ox  Stagantion pressure ratio across normal shock"
"PoyOverPx = P_oy/P_x  Stagnation pressure after normal shock ratioed to pressure before shock"

"Input Data"
{P_x = 10 "psia"}              "Values of P_x, T_x, and M_x are set in the Parametric Table"
{T_x = 440.5 "R"}
{M_x = 2.5}
```

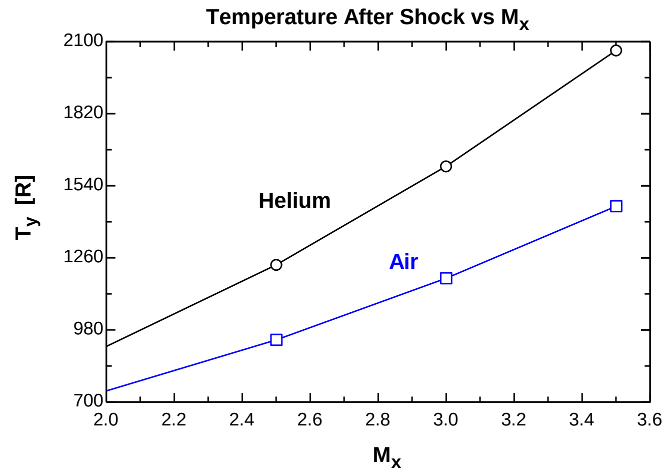
```
Gas$='Air'  "This program has been written for the gases Air, CO2, and Helium"
Call GetProp(Gas$:Cp,k,R)
Call NormalShock(M_x,k:M_y,PyOverPx, TyOverTx,RhoyOverRhox, PoyOverPox, PoyOverPx)
P_oy_air=P_x*PoyOverPx              "Stagnation pressure after the shock"
P_y_air=P_x*PyOverPx                "Pressure after the shock"
T_y_air=T_x*TyOverTx                "Temperature after the shock"
M_y_air=M_y                         "Mach number after the shock"

"The velocity after the shock can be found from the product of the Mach number and
speed of sound after the shock."
C_y_air = sqrt(k*R"ft-lbf/lbm_R"*T_y_air"R"*32.2 "lbm-ft/lbf-s^2")
V_y_air=M_y_air*C_y_air
DELTAs_air=entropy(air,T=T_y_air, P=P_y_air) -entropy(air,T=T_x,P=P_x)

Gas2$='Helium'  "Gas2$ can be either Helium or CO2"
Call GetProp(Gas2$:Cp_2,k_2,R_2)
Call NormalShock(M_x,k_2:M_y2,PyOverPx2, TyOverTx2,RhoyOverRhox2, PoyOverPox2, PoyOverPx2)
P_oy_he=P_x*PoyOverPx2              "Stagnation pressure after the shock"
P_y_he=P_x*PyOverPx2                "Pressure after the shock"
T_y_he=T_x*TyOverTx2                "Temperature after the shock"
M_y_he=M_y2                         "Mach number after the shock"

"The velocity after the shock can be found from the product of the Mach number and
speed of sound after the shock."
C_y_he = sqrt(k_2*R_2"ft-lbf/lbm_R"*T_y_he"R"*32.2 "lbm-ft/lbf-s^2")
V_y_he=M_y_he*C_y_he
DELTAs_he=entropy(helium,T=T_y_he, P=P_y_he) -entropy(helium,T=T_x,P=P_x)
```

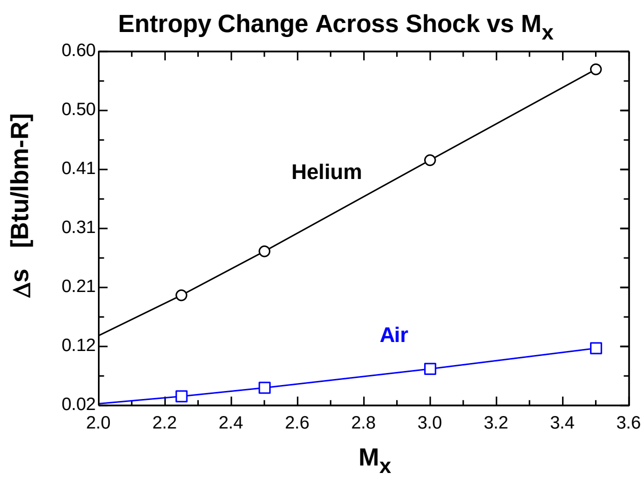
The parametric table and the corresponding plots are shown below.

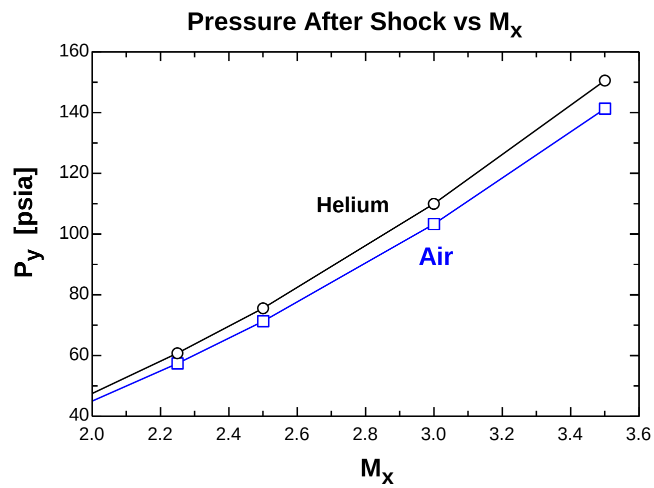
$V_{y,he}$ [ft/s]	$V_{y,air}$ [ft/s]	$T_{y,he}$ [R]	$T_{y,air}$ [R]	T_x [R]	$P_{y,he}$ [psia]	$P_{y,air}$ [psia]	P_x [psia]	$P_{oy,he}$ [psia]	$P_{oy,air}$ [psia]	$M_{y,he}$	$M_{y,air}$	M_x	Δs_{he} [Btu/lbm-R]	Δs_{air} [Btu/lbm-R]
2644	771.9	915.6	743.3	440.5	47.5	45	10	63.46	56.4	0.607	0.5774	2	0.1345	0.0228
2707	767.1	1066	837.6	440.5	60.79	57.4	10	79.01	70.02	0.5759	0.5406	2.25	0.2011	0.0351
2795	771.9	1233	941.6	440.5	75.63	71.25	10	96.41	85.26	0.553	0.513	2.5	0.2728	0.04899
3022	800.4	1616	1180	440.5	110	103.3	10	136.7	120.6	0.5223	0.4752	3	0.4223	0.08
3292	845.4	2066	1460	440.5	150.6	141.3	10	184.5	162.4	0.5032	0.4512	3.5	0.5711	0.1136











Discussion In all cases, regardless of the fluid or the Mach number, entropy increases across a shock wave. This is because a shock wave involves irreversibilities.

12-69

Solution Air flowing through a converging-diverging nozzle experiences a normal shock at the exit. The effect of the shock wave on various properties is to be determined.

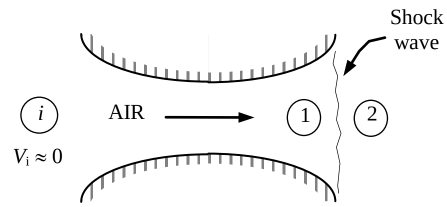
Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs. **3** The shock wave occurs at the exit plane.

Properties The properties of air are $k = 1.4$ and $R = 0.287$ kJ/kg·K.

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Then,

$$P_{01} = P_i = 1 \text{ MPa}$$

$$T_{01} = T_i = 300 \text{ K}$$



Then,

$$T_1 = T_{01}\left(\frac{2}{2+(k-1)\mathrm{Ma}_1^2}\right) = (300 \text{ K})\left(\frac{2}{2+(1.4-1)2.4^2}\right) = 139.4 \text{ K}$$

and

$$P_1 = P_{01}\left(\frac{T_1}{T_0}\right)^{k/(k-1)} = (1 \text{ MPa})\left(\frac{139.4}{300}\right)^{1.4/0.4} = 0.06840 \text{ MPa}$$

The fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-14. For $\mathrm{Ma}_1 = 2.4$ we read

$$\mathrm{Ma}_2 = 0.5231 \cong \mathbf{0.523}, \quad \frac{P_{02}}{P_{01}} = 0.5401, \quad \frac{P_2}{P_1} = 6.5533, \text{ and } \frac{T_2}{T_1} = 2.0403$$

Then the stagnation pressure P_{02}, static pressure P_2, and static temperature T_2, are determined to be

$$P_{02} = 0.5401 P_{01} = (0.5401)(1.0 \text{ MPa}) = \mathbf{0.540 \text{ MPa} = 540 \text{ kPa}}$$

$$P_2 = 6.5533 P_1 = (6.5533)(0.06840 \text{ MPa}) = \mathbf{0.448 \text{ MPa} = 448 \text{ kPa}}$$

$$T_2 = 2.0403 T_1 = (2.0403)(139.4 \text{ K}) = \mathbf{284 \text{ K}}$$

The air velocity after the shock can be determined from $V_2 = \mathrm{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \mathrm{Ma}_2 c_2 = \mathrm{Ma}_2\sqrt{kRT_2} = (0.5231)\sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(284 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{177 \text{ m/s}}$$

Discussion We can also solve this problem using the relations for normal shock functions. The results would be identical.

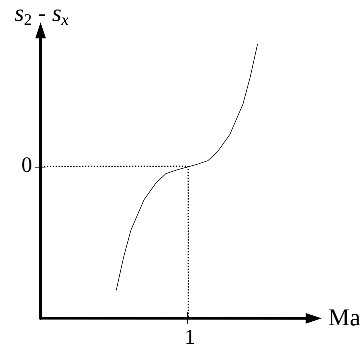
12-70

Solution The entropy change of air across the shock for upstream Mach numbers between 0.5 and 1.5 is to be determined and plotted.

Assumptions **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air are $k = 1.4$, $R = 0.287$ kJ/kg·K, and $c_p = 1.005$ kJ/kg·K.

Analysis The entropy change across the shock is determined to be



$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

where

$$\text{Ma}_2 = \left(\frac{\text{Ma}_1^2 + 2/(k-1)}{2\text{Ma}_1^2 k/(k-1) - 1}\right)^{1/2}, \quad \frac{P_2}{P_1} = \frac{1+k\text{Ma}_1^2}{1+k\text{Ma}_2^2}, \text{ and } \frac{T_2}{T_1} = \frac{1+\text{Ma}_1^2(k-1)/2}{1+\text{Ma}_2^2(k-1)/2}$$

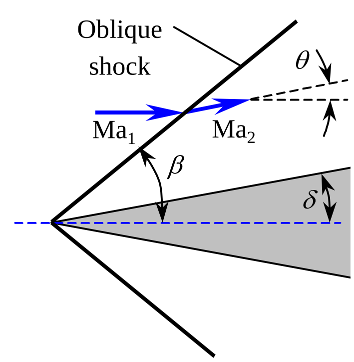
The results of the calculations can be tabulated as

Ma_1	Ma_2	T_2/T_1	P_2/P_1	$s_2 - s_1$
0.5	2.6458	0.1250	0.4375	-1.853
0.6	1.8778	0.2533	0.6287	-1.247
0.7	1.5031	0.4050	0.7563	-0.828
0.8	1.2731	0.5800	0.8519	-0.501
0.9	1.1154	0.7783	0.9305	-0.231
1.0	1.0000	1.0000	1.0000	0.0
1.1	0.9118	1.0649	1.2450	0.0003
1.2	0.8422	1.1280	1.5133	0.0021
1.3	0.7860	1.1909	1.8050	0.0061
1.4	0.7397	1.2547	2.1200	0.0124
1.5	0.7011	1.3202	2.4583	0.0210

Discussion The total entropy change is negative for upstream Mach numbers Ma_1 less than unity. Therefore, normal shocks cannot occur when $\text{Ma}_1 < 1$.

12-71

Solution Supersonic airflow approaches the nose of a two-dimensional wedge and undergoes a straight oblique shock. For a specified Mach number, the minimum shock angle and the maximum deflection angle are to be determined.



Assumptions Air is an ideal gas with a constant specific heat ratio of $k = 1.4$ (so that Fig. 12-41 is applicable).

Analysis For Ma = 5, we read from Fig. 12-41

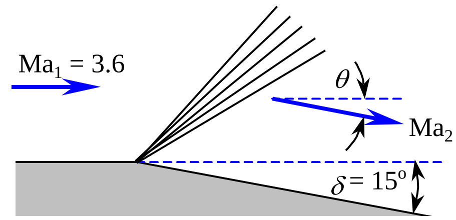
Minimum shock (or wave) angle: $\beta_{\min} = \mathbf{12^\circ}$

Maximum deflection (or turning) angle: $\theta_{\max} = \mathbf{41.5^\circ}$

Discussion Note that the minimum shock angle decreases and the maximum deflection angle increases with increasing Mach number Ma_1.

12-72

Solution Air flowing at a specified supersonic Mach number undergoes an expansion turn. The Mach number, pressure, and temperature downstream of the sudden expansion along a wall are to be determined.



Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is k = 1.4.

Analysis On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 15^{o}$. Then the upstream and downstream Prandtl-Meyer functions are determined to be

$$\nu(\mathrm{Ma}) = \sqrt{\frac{k+1}{k-1}}\tan^{-1}\left(\sqrt{\frac{k-1}{k+1}(\mathrm{Ma}^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}^2-1}\right)$$

Upstream:

$$\nu(\mathrm{Ma}_1) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}(3.6^2-1)}\right) - \tan^{-1}\left(\sqrt{3.6^2-1}\right) = 60.09°$$

Then the downstream Prandtl-Meyer function becomes

$$\nu(\mathrm{Ma}_2) = \theta + \nu(\mathrm{Ma}_1) = 15° + 60.09° = 75.09°$$

Ma_2 is found from the Prandtl-Meyer relation, which is now implicit:

Downstream:
$$\nu(\mathrm{Ma}_2) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}\mathrm{Ma}_2^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}_2^2-1}\right) = 75.09°$$

Solution of this implicit equation gives Ma_2 = **4.81**. Then the downstream pressure and temperature are determined from the isentropic flow relations:

$$P_2 = \frac{P_2/P_0}{P_1/P_0}P_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-k/(k-1)}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-k/(k-1)}}P_1 = \frac{[1+4.81^2(1.4-1)/2]^{-1.4/0.4}}{[1+3.6^2(1.4-1)/2]^{-1.4/0.4}}(32\text{ kPa}) = \mathbf{6.65kPa}$$

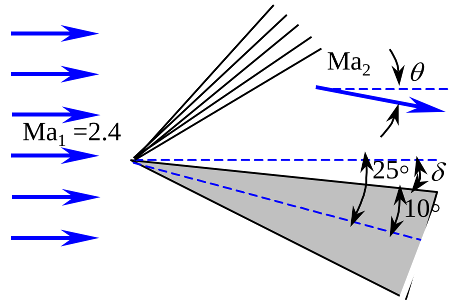
$$T_2 = \frac{T_2/T_0}{T_1/T_0}T_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-1}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-1}}T_1 = \frac{[1+4.81^2(1.4-1)/2]^{-1}}{[1+3.6^2(1.4-1)/2]^{-1}}(240\text{ K}) = \mathbf{153K}$$

Note that this is an expansion, and Mach number increases while pressure and temperature decrease, as expected.

Discussion There are compressible flow calculators on the Internet that solve these implicit equations that arise in the analysis of compressible flow, along with both normal and oblique shock equations; e.g., see www.aoe.vt.edu/~devenpor/aoe3114/calc.html .

12-73

Solution Air flowing at a specified supersonic Mach number undergoes an expansion turn over a tilted wedge. The Mach number, pressure, and temperature downstream of the sudden expansion above the wedge are to be determined.



Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is k = 1.4.

Analysis On the basis of Assumption #2, the deflection angle is determined to be $\theta \approx \delta$ = 25° - 10° = 15°. Then the upstream and downstream Prandtl-Meyer functions are determined to be

$$\nu(\text{Ma}) = \sqrt{\frac{k+1}{k-1}}\tan^{-1}\left(\sqrt{\frac{k-1}{k+1}(\text{Ma}^2-1)}\right) - \tan^{-1}\left(\sqrt{\text{Ma}^2-1}\right)$$

Upstream:

$$\nu(\text{Ma}_1) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}(2.4^2-1)}\right) - \tan^{-1}\left(\sqrt{2.4^2-1}\right) = 36.75°$$

Then the downstream Prandtl-Meyer function becomes

$$\nu(\text{Ma}_2) = \theta + \nu(\text{Ma}_1) = 15° + 36.75° = 51.75°$$

Now Ma$_2$ is found from the Prandtl-Meyer relation, which is now implicit:

Downstream: $\nu(\text{Ma}_2) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}(\text{Ma}_2^2-1)}\right) - \tan^{-1}\left(\sqrt{\text{Ma}_2^2-1}\right) = 51.75°$

It gives Ma$_2$ = **3.105**. Then the downstream pressure and temperature are determined from the isentropic flow relations

$$P_2 = \frac{P_2/P_0}{P_1/P_0}P_1 = \frac{[1+\text{Ma}_2^2(k-1)/2]^{-k/(k-1)}}{[1+\text{Ma}_1^2(k-1)/2]^{-k/(k-1)}}P_1 = \frac{[1+3.105^2(1.4-1)/2]^{-1.4/0.4}}{[1+2.4^2(1.4-1)/2]^{-1.4/0.4}}(70\text{ kPa}) = \mathbf{23.8kPa}$$

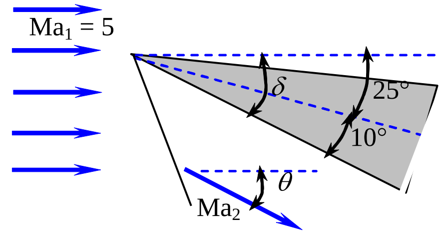
$$T_2 = \frac{T_2/T_0}{T_1/T_0}T_1 = \frac{[1+\text{Ma}_2^2(k-1)/2]^{-1}}{[1+\text{Ma}_1^2(k-1)/2]^{-1}}T_1 = \frac{[1+3.105^2(1.4-1)/2]^{-1}}{[1+2.4^2(1.4-1)/2]^{-1}}(260\text{ K}) = \mathbf{191\ K}$$

Note that this is an expansion, and Mach number increases while pressure and temperature decrease, as expected.

Discussion There are compressible flow calculators on the Internet that solve these implicit equations that arise in the analysis of compressible flow, along with both normal and oblique shock equations; e.g., see www.aoe.vt.edu/~devenpor/aoe3114/calc.html .

12-74

Solution Air flowing at a specified supersonic Mach number undergoes a compression turn (an oblique shock) over a tilted wedge. The Mach number, pressure, and temperature downstream of the shock below the wedge are to be determined.



Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is $k = 1.4$.

Analysis On the basis of Assumption #2, the deflection angle is determined to be $\theta \approx \delta = 25° + 10° = 35°$. Then the two values of oblique shock angle β are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k + \cos 2\beta) + 2} \qquad \rightarrow \qquad \tan 12° = \frac{2(3.4^2 \sin^2\beta - 1)/\tan\beta}{3.4^2(1.4 + \cos 2\beta) + 2}$$

which is implicit in β. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives β_{weak} = 49.86° and β_{strong} = 77.66°. Then for the case of strong oblique shock, the upstream "normal" Mach number $\mathrm{Ma}_{1,n}$ becomes

$$\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 5\sin 77.66° = 4.884$$

Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,n}$ become

$$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(4.884)^2 + 2}{2(1.4)(4.884)^2 - 1.4 + 1}} = 0.4169$$

The downstream pressure and temperature are determined to be

$$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (70\ \text{kPa})\frac{2(1.4)(4.884)^2 - 1.4 + 1}{1.4 + 1} = \mathbf{1940\ kPa}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1}\frac{2 + (k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (260\ \text{K})\frac{1940\ \text{kPia}}{70\ \text{kPa}}\frac{2 + (1.4-1)(4.884)^2}{(1.4+1)(4.884)^2} = \mathbf{1450\ K}$$

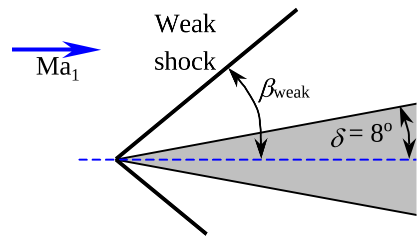
The downstream Mach number is determined to be

$$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.4169}{\sin(77.66° - 35°)} = \mathbf{0.615}$$

Discussion Note that $\mathrm{Ma}_{1,n}$ is supersonic and $\mathrm{Ma}_{2,n}$ and Ma_2 are subsonic. Also note the huge rise in temperature and pressure across the strong oblique shock, and the challenges they present for spacecraft during reentering the earth's atmosphere.

12-75E

Solution Air flowing at a specified supersonic Mach number is forced to turn upward by a ramp, and weak oblique shock forms. The wave angle, Mach number, pressure, and temperature after the shock are to be determined.



Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is $k = 1.4$.

Analysis On the basis of Assumption #2, we take the deflection angle as equal to the ramp, i.e., $\theta \approx \delta = 8^o$. Then the two values of oblique shock angle β are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k + \cos 2\beta) + 2} \qquad \rightarrow \qquad \tan 8° = \frac{2(2^2 \sin^2\beta - 1)/\tan\beta}{2^2(1.4 + \cos 2\beta) + 2}$$

which is implicit in β. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\boldsymbol{\beta_{weak}}$ **= 37.21°** and $\beta_{strong} = 85.05^o$. Then for the case of weak oblique shock, the upstream "normal" Mach number $Ma_{1,n}$ becomes

$$\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 2\sin 37.21° = 1.209$$

Also, the downstream normal Mach numbers $Ma_{2,n}$ become

$$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.209)^2 + 2}{2(1.4)(1.209)^2 - 1.4 + 1}} = 0.8363$$

The downstream pressure and temperature are determined to be

$$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (12\text{ psia})\frac{2(1.4)(1.209)^2 - 1.4 + 1}{1.4 + 1} = \mathbf{18.5\ psia}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1}\frac{2 + (k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (490\text{ R})\frac{18.5\text{ psia}}{12\text{ psia}}\frac{2 + (1.4-1)(1.209)^2}{(1.4+1)(1.209)^2} = \mathbf{556\ R}$$

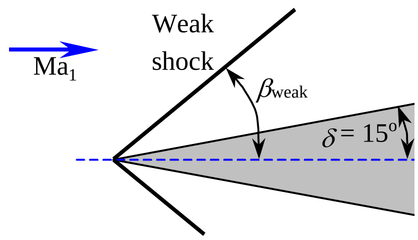
The downstream Mach number is determined to be

$$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.8363}{\sin(37.21° - 8°)} = \mathbf{1.71}$$

Discussion Note that $Ma_{1,n}$ is supersonic and $Ma_{2,n}$ is subsonic. However, Ma_2 is *supersonic* across the weak oblique shock (it is *subsonic* across the strong oblique shock).

12-76E

Solution Air flowing at a specified supersonic Mach number is forced to undergo a compression turn (an oblique shock)., The Mach number, pressure, and temperature downstream of the oblique shock are to be determined.

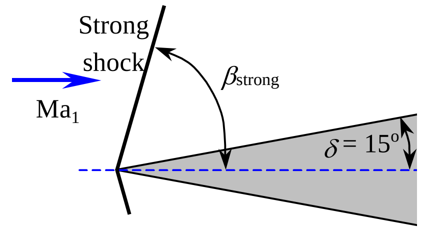


Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is $k = 1.4$.

Analysis On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 15^o$. Then the two values of oblique shock angle β are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k+\cos 2\beta)+2} \quad \rightarrow \quad \tan 15° = \frac{2(2^2\sin^2\beta - 1)/\tan\beta}{2^2(1.4+\cos 2\beta)+2}$$



which is implicit in β. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\beta_{weak} = 45.34^o$ and $\beta_{strong} = 79.83^o$. Then the upstream "normal" Mach number $\mathrm{Ma}_{1,n}$ becomes

Weak shock: $$\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 2\sin 45.34° = 1.423$$

Strong shock: $$\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 2\sin 79.83° = 1.969$$

Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,n}$ become

Weak shock: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.423)^2 + 2}{2(1.4)(1.423)^2 - 1.4 + 1}} = 0.7304$$

Strong shock: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.969)^2 + 2}{2(1.4)(1.969)^2 - 1.4 + 1}} = 0.5828$$

The downstream pressure and temperature for each case are determined to be

Weak shock: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (6\text{ psia})\frac{2(1.4)(1.423)^2 - 1.4 + 1}{1.4+1} = 17.57 \cong \mathbf{17.6 psia}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1}\frac{2+(k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (480\text{ R})\frac{17.57\text{ psia}}{8\text{ psia}}\frac{2+(1.4-1)(1.423)^2}{(1.4+1)(1.423)^2} = 609.5\text{ R} \cong \mathbf{610 R}$$

Strong shock: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (8\text{ psia})\frac{2(1.4)(1.969)^2 - 1.4 + 1}{1.4+1} = 34.85 \cong \mathbf{34.9\ psia}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1}\frac{2+(k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (480\text{ R})\frac{34.85\text{ psia}}{8\text{ psia}}\frac{2+(1.4-1)(1.969)^2}{(1.4+1)(1.969)^2} = 797.9\text{ R} \cong \mathbf{798 R}$$

The downstream Mach number is determined to be

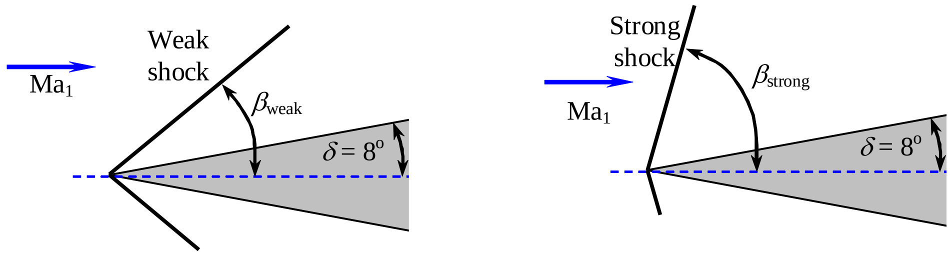
Weak shock: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta-\theta)} = \frac{0.7304}{\sin(45.34° - 15°)} = \mathbf{1.45}$$

Strong shock: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta-\theta)} = \frac{0.5828}{\sin(79.83° - 15°)} = \mathbf{0.644}$$

Discussion Note that the change in Mach number, pressure, temperature across the *strong shock* are much greater than the changes across the *weak shock*, as expected. For both the weak and strong oblique shock cases, $\mathrm{Ma}_{1,n}$ is supersonic and $\mathrm{Ma}_{2,n}$ is subsonic. However, Ma_2 is *supersonic* across the weak oblique shock, but *subsonic* across the strong oblique shock.

12-77

Solution Air flowing at a specified supersonic Mach number impinges on a two-dimensional wedge, The shock angle, Mach number, and pressure downstream of the weak and strong oblique shock formed by a wedge are to be determined.



Assumptions **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is $k = 1.4$.

Analysis On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 8^o$. Then the two values of oblique shock angle β are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k + \cos 2\beta) + 2} \quad \rightarrow \quad \tan 8° = \frac{2(3.4^2 \sin^2\beta - 1)/\tan\beta}{3.4^2(1.4 + \cos 2\beta) + 2}$$

which is implicit in β. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\boldsymbol{\beta_{weak} = 23.15^o}$ and $\boldsymbol{\beta_{strong} = 87.45^o}$. Then the upstream "normal" Mach number $\mathrm{Ma}_{1,n}$ becomes

Weak shock: $\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 3.4 \sin 23.15° = 1.336$

Strong shock: $\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 3.4 \sin 87.45° = 3.397$

Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,n}$ become

Weak shock: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.336)^2 + 2}{2(1.4)(1.336)^2 - 1.4 + 1}} = 0.7681$$

Strong shock: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(3.397)^2 + 2}{2(1.4)(3.397)^2 - 1.4 + 1}} = 0.4553$$

The downstream pressure for each case is determined to be

Weak shock: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (60\ \mathrm{kPa})\frac{2(1.4)(1.336)^2 - 1.4 + 1}{1.4 + 1} = \mathbf{115.0 kPa}$$

Strong shock: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (60\ \mathrm{kPa})\frac{2(1.4)(3.397)^2 - 1.4 + 1}{1.4 + 1} = \mathbf{797.6 kPa}$$

The downstream Mach number is determined to be

Weak shock: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.7681}{\sin(23.15° - 8°)} = \mathbf{2.94}$$

Strong shock: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.4553}{\sin(87.45° - 8°)} = \mathbf{0.463}$$

Discussion Note that the change in Mach number and pressure across the *strong shock* are much greater than the changes across the *weak shock*, as expected. For both the weak and strong oblique shock cases, $\mathrm{Ma}_{1,n}$ is supersonic and $\mathrm{Ma}_{2,n}$ is subsonic. However, Ma_2 is *supersonic* across the weak oblique shock, but *subsonic* across the strong oblique shock.

12-78

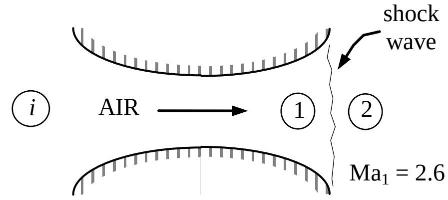
Solution Air flowing through a nozzle experiences a normal shock. The effect of the shock wave on various properties is to be determined. Analysis is to be repeated for helium under the same conditions.

Assumptions **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air are k = 1.4 and R = 0.287 kJ/kg·K, and the properties of helium are k = 1.667 and R = 2.0769 kJ/kg·K.

Analysis The air properties upstream the shock are

Ma_1 = 2.6, P_1 = 58 kPa, and T_1 = 270 K



Fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions in Table A-14. For Ma_1 = 2.6,

$$\mathrm{Ma}_2 = \mathbf{0.5039},\ \frac{P_{02}}{P_1} = 9.1813,\ \frac{P_2}{P_1} = 7.7200,\ \text{and}\ \frac{T_2}{T_1} = 2.2383$$

We obtained these values using analytical relations in Table 14. Then the stagnation pressure P_{02}, static pressure P_2, and static temperature T_2, are determined to be

P_{02} = 9.1813P_1 = (9.1813)(58 kPa) = **532.5 kPa**

P_2 = 7.7200P_1 = (7.7200)(58 kPa) = **447.8 kPa**

T_2 = 2.9220T_1 = (2.2383)(270 K) = **604.3 K**

The air velocity after the shock can be determined from V_2 = $\mathrm{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \mathrm{Ma}_2 c_2 = \mathrm{Ma}_2\sqrt{kRT_2} = (0.5039)\sqrt{(1.4)(0.287\ \mathrm{kJ/kg\cdot K})(604.3\ \mathrm{K})\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = \mathbf{248.3\ m/s}$$

We now repeat the analysis for helium. This time we cannot use the tabulated values in Table A-14 since k is not 1.4. Therefore, we have to calculate the desired quantities using the analytical relations,

$$\mathrm{Ma}_2 = \left(\frac{\mathrm{Ma}_1^2 + 2/(k-1)}{2\mathrm{Ma}_1^2 k/(k-1) - 1}\right)^{1/2} = \left(\frac{2.6^2 + 2/(1.667-1)}{2\times 2.6^2 \times 1.667/(1.667-1) - 1}\right)^{1/2} = \mathbf{0.5455}$$

$$\frac{P_2}{P_1} = \frac{1+k\mathrm{Ma}_1^2}{1+k\mathrm{Ma}_2^2} = \frac{1+1.667\times 2.6^2}{1+1.667\times 0.5455^2} = 8.2009$$

$$\frac{T_2}{T_1} = \frac{1+\mathrm{Ma}_1^2(k-1)/2}{1+\mathrm{Ma}_2^2(k-1)/2} = \frac{1+2.6^2(1.667-1)/2}{1+0.5455^2(1.667-1)/2} = 2.9606$$

$$\frac{P_{02}}{P_1} = \left(\frac{1+k\mathrm{Ma}_1^2}{1+k\mathrm{Ma}_2^2}\right)\left(1+(k-1)\mathrm{Ma}_2^2/2\right)^{k/(k-1)}$$

$$= \left(\frac{1+1.667\times 2.6^2}{1+1.667\times 0.5455^2}\right)\left(1+(1.667-1)\times 0.5455^2/2\right)^{1.667/0.667} = 10.389$$

Thus, P_{02} = 10.389P_1 = (10.389)(58 kPa) = **602.5 kPa**

P_2 = 8.2009P_1 = (8.2009)(58 kPa) = **475.7 kPa**

T_2 = 2.9606T_1 = (2.9606)(270 K) = **799.4 K**

$$V_2 = \mathrm{Ma}_2 c_2 = \mathrm{Ma}_2\sqrt{kRT_2} = (0.5455)\sqrt{(1.667)(2.0769\ \mathrm{kJ/kg\cdot K})(799.4\ \mathrm{K})\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = \mathbf{907.5\ m/s}$$

Discussion The velocity and Mach number are higher for helium than for air due to the different values of k and R.

12-79

Solution Air flowing through a nozzle experiences a normal shock. The entropy change of air across the normal shock wave is to be determined.

Assumptions **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

Properties The properties of air are R = 0.287 kJ/kg·K and c_p = 1.005 kJ/kg·K, and the properties of helium are R = 2.0769 kJ/kg·K and c_p = 5.1926 kJ/kg·K.

Analysis For air, the entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (1.005\text{ kJ/kg}\cdot\text{K})\ln(2.2383) - (0.287\text{ kJ/kg}\cdot\text{K})\ln(7.7200) = \mathbf{0.223\ kJ/kg\cdot K}$$

For helium, the entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (5.1926\text{ kJ/kg}\cdot\text{K})\ln(2.9606) - (2.0769\text{ kJ/kg}\cdot\text{K})\ln(8.2009) = \mathbf{1.27\ kJ/kg\cdot K}$$

Discussion Note that shock wave is a highly dissipative process, and the entropy generation is large during shock waves.

Duct Flow with Heat Transfer and Negligible Friction (Rayleigh Flow)

12-80C

Solution We are to discuss the effect of heating on the flow velocity in subsonic Rayleigh flow.

Analysis Heating the fluid **increases the flow velocity in subsonic Rayleigh flow**, but **decreases the flow velocity in supersonic Rayleigh flow**.

Discussion These results are not necessarily intuitive, but must be true in order to satisfy the conservation laws.

12-81C

Solution We are to discuss what the points on a *T*-*s* diagram of Rayleigh flow represent.

Analysis The points on the Rayleigh line represent the **states that satisfy the conservation of mass, momentum, and energy equations as well as the property relations for a given state**. Therefore, for a given inlet state, the fluid cannot exist at any downstream state outside the Rayleigh line on a *T*-*s* diagram.

Discussion The *T*-*s* diagram is quite useful, since any downstream state must lie on the Rayleigh line.

12-82C

Solution We are to discuss the effect of heat gain and heat loss on entropy during Rayleigh flow.

Analysis In Rayleigh flow, **the effect of heat gain is to increase the entropy** of the fluid, and **the effect of heat loss is to decrease the entropy**.

Discussion You should recall from thermodynamics that the entropy of a system can be lowered by removing heat.

12-83C

Solution We are to discuss how temperature and stagnation temperature change in subsonic Rayleigh flow.

Analysis In Rayleigh flow, the **stagnation temperature T_0 always increases with heat transfer to the fluid**, but the temperature T decreases with heat transfer in the Mach number range of 0.845 < Ma < 1 for air. Therefore, the **temperature in this case will decrease**.

Discussion This at first seems counterintuitive, but if heat were *not* added, the temperature would drop even *more* if the air were accelerated isentropically from Ma = 0.92 to 0.95.

12-84C

Solution We are to discuss the characteristic aspect of Rayleigh flow, and its main assumptions.

Analysis The characteristic aspect of Rayleigh flow is **its involvement of heat transfer**. The main assumptions associated with Rayleigh flow are: the flow is **steady**, **one-dimensional**, and **frictionless** through a constant-area duct, and the fluid is an **ideal gas with constant specific heats**.

Discussion Of course, there is no such thing as frictionless flow. It is better to say that frictional effects are negligible compared to the heating effects.

12-85C

Solution We are to examine the Mach number at the end of a choked duct in Rayleigh flow when more heat is added.

Analysis The flow is choked, and thus the flow at the duct exit **remains sonic**.

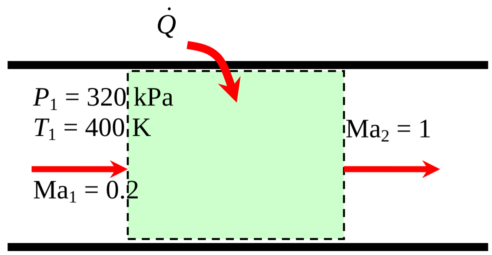
Discussion There is no mechanism for the flow to become supersonic in this case.

12-86

Solution Argon flowing at subsonic velocity in a constant-diameter duct is accelerated by heating. The highest rate of heat transfer without reducing the mass flow rate is to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Mass flow rate remains constant.

Properties We take the properties of argon to be k = 1.667, c_p = 0.5203 kJ/kg·K, and R = 0.2081 kJ/kg·K.



Analysis Heat transfer stops when the flow is choked, and thus $\mathrm{Ma}_2 = V_2/c_2 = 1$. The inlet stagnation temperature is

$$T_{01} = T_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right) = (400\ \mathrm{K})\left(1+\frac{1.667\text{-}1}{2}0.2^2\right) = 405.3\ \mathrm{K}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are

$T_{02}/T_0^* = 1$ (since $\mathrm{Ma}_2 = 1$)

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\mathrm{Ma}_1^2[2+(k-1)\mathrm{Ma}_1^2]}{(1+k\mathrm{Ma}_1^2)^2} = \frac{(1.667+1)0.2^2[2+(1.667-1)0.2^2]}{(1+1.667\times 0.2^2)^2} = 0.1900$$ Therefore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.1900} \qquad \rightarrow \quad T_{02} = T_{01}/0.1900 = (405.3\ \mathrm{K})/0.1900 = 2133\ \mathrm{K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\mathrm{air}} c_p (T_{02} - T_{01}) = (1.2\ \mathrm{kg/s})(0.5203\ \mathrm{kJ/kg\cdot K})(2133-400)\ \mathrm{K} = \mathbf{1080\ kW}$$

Discussion It can also be shown that T_2 = 1600 K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. Also, in the solution of this problem, we cannot use the values of Table A-15 since they are based on k = 1.4.

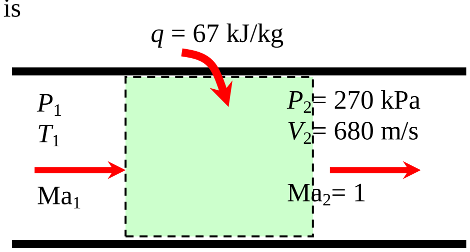
12-87

Solution Air is heated in a duct during subsonic flow until it is choked. For specified pressure and velocity at the exit, the temperature, pressure, and velocity at the inlet are to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis Noting that sonic conditions exist at the exit, the exit temperature is



$$c_2 = V_2/\mathrm{Ma}_2 = (680\ \mathrm{m/s})/1 = 680\ \mathrm{m/s}$$

$$c_2 = \sqrt{kRT_2} \quad\rightarrow\quad \sqrt{(1.4)(0.287\ \mathrm{kJ/kg\cdot K})T_2\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = 680\ \mathrm{m/s}$$

It gives $T_2 = 1151$ K. Then the exit stagnation temperature becomes

$$T_{02} = T_2 + \frac{V_2^2}{2c_p} = 1151\ \mathrm{K} + \frac{(680\ \mathrm{m/s})^2}{2\times 1.005\ \mathrm{kJ/kg\cdot K}}\left(\frac{1\ \mathrm{kJ/kg}}{1000\ \mathrm{m^2/s^2}}\right) = 1381\ \mathrm{K}$$

The inlet stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{01} = T_{02} - \frac{q}{c_p} = 1381\ \mathrm{K} - \frac{67\ \mathrm{kJ/kg}}{1.005\ \mathrm{kJ/kg\cdot K}} = 1314\ \mathrm{K}$$

The maximum value of stagnation temperature T_0^* occurs at Ma = 1, and its value in this case is T_{02} since the flow is choked. Therefore, $T_0^* = T_{02} = 1381$ K. Then the stagnation temperature ratio at the inlet, and the Mach number corresponding to it are, from Table A-15,

$$\frac{T_{01}}{T_0^*} = \frac{1314\ \mathrm{K}}{1381\ \mathrm{K}} = 0.9516 \quad\rightarrow\quad \mathrm{Ma}_1 = 0.7779 \cong \mathbf{0.778}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-15):

$\mathrm{Ma}_1 = 0.7779$: $T_1/T^* = 1.018$, $P_1/P^* = 1.301$, $V_1/V^* = 0.7852$

$\mathrm{Ma}_2 = 1$: $T_2/T^* = 1$, $P_2/P^* = 1$, $V_2/V^* = 1$

Then the inlet temperature, pressure, and velocity are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{1.018} \quad\rightarrow\quad T_1 = 1.018T_2 = 1.018(1151\ \mathrm{K}) = \mathbf{1172\ K}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{1.301} \quad\rightarrow\quad P_1 = 1.301P_2 = 1.301(270\ \mathrm{kPa}) = \mathbf{351.3\ kPa}$$

$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{1}{0.7852} \quad\rightarrow\quad V_1 = 0.7852V_2 = 0.7852(680\ \mathrm{m/s}) = \mathbf{533.9\ m/s}$$

Discussion Note that the temperature and pressure decreases with heating during this subsonic Rayleigh flow while velocity increases. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

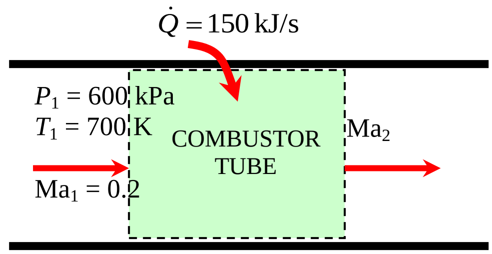
12-88

Solution Air enters the combustion chamber of a gas turbine at a subsonic velocity. For a specified rate of heat transfer, the Mach number at the exit and the loss in stagnation pressure to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The cross-sectional area of the combustion chamber is constant. **3** The increase in mass flow rate due to fuel injection is disregarded.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis The inlet stagnation temperature and pressure are



$$T_{01} = T_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right) = (700\text{ K})\left(1+\frac{1.4-1}{2}0.2^2\right) = 705.6\text{ K}$$

$$P_{01} = P_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{k/(k-1)} = (600\text{ kPa})\left(1+\frac{1.4\text{-}1}{2}0.2^2\right)^{1.4/0.4}$$
$$= 617.0\text{ kPa}$$

The exit stagnation temperature is determined from

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) \quad\rightarrow\quad 150\text{ kJ/s} = (0.3\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(T_{02} - 705.6)\text{ K}$$

It gives

$$T_{02} = 1203\text{ K}$$

At $Ma_1 = 0.2$ we read from $T_{01}/T_0^* = 0.1736$ (Table A-15). Therefore,

$$T_0^* = \frac{T_{01}}{0.1736} = \frac{705.6\text{ K}}{0.1736} = 4064.5\text{ K}$$

Then the stagnation temperature ratio at the exit and the Mach number corresponding to it are (Table A-15)

$$\frac{T_{02}}{T_0^*} = \frac{1203\text{ K}}{4064.5\text{ K}} = 0.2960 \quad\rightarrow\quad \mathrm{Ma}_2 = 0.2706 \cong \mathbf{0.271}$$

Also,

$Ma_1 = 0.2 \quad \rightarrow P_{01}/P_0^* = 1.2346$

$Ma_2 = 0.2706 \quad \rightarrow P_{02}/P_0^* = 1.2091$

Then the stagnation pressure at the exit and the pressure drop become

$$\frac{P_{02}}{P_{01}} = \frac{P_{02}/P_0^*}{P_{01}/P_0^*} = \frac{1.2091}{1.2346} = 0.9794 \quad\rightarrow\quad P_{02} = 0.9794P_{01} = 0.9794(617\text{ kPa}) = 604.3\text{ kPa}$$

and

$$\Delta P_0 = P_{01} - P_{02} = 617.0 - 604.3 = \mathbf{12.7\,kPa}$$

Discussion This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

12-89

Solution Air enters the combustion chamber of a gas turbine at a subsonic velocity. For a specified rate of heat transfer, the Mach number at the exit and the loss in stagnation pressure to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The cross-sectional area of the combustion chamber is constant. **3** The increase in mass flow rate due to fuel injection is disregarded.

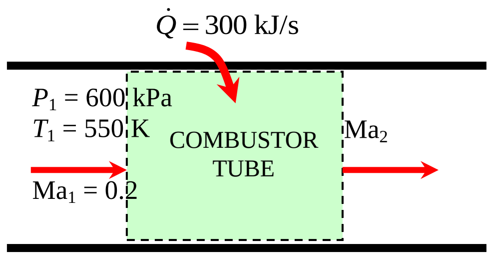
Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K.

Analysis The inlet stagnation temperature and pressure are

$$T_{01} = T_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right) = (700\text{ K})\left(1+\frac{1.4\text{-}1}{2}0.2^2\right) = 705.6\text{ K}$$

$$P_{01} = P_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{k/(k-1)} = (600\text{ kPa})\left(1+\frac{1.4\text{-}1}{2}0.2^2\right)^{1.4/0.4}$$

$$= 617.0\text{ kPa}$$



The exit stagnation temperature is determined from

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) \quad \rightarrow \quad 300\text{ kJ/s} = (0.3\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(T_{02} - 705.6)\text{ K}$$

It gives

$$T_{02} = 1701\text{ K}$$

At Ma_1 = 0.2 we read from T_{01}/T_0^* = 0.1736 (Table A-15). Therefore,

$$T_0^* = \frac{T_{01}}{0.1736} = \frac{705.6\text{ K}}{0.1736} = 4064.5\text{ K}$$

Then the stagnation temperature ratio at the exit and the Mach number corresponding to it are (Table A-15)

$$\frac{T_{02}}{T_0^*} = \frac{1701\text{ K}}{4064.5\text{ K}} = 0.4185 \quad \rightarrow \quad \mathrm{Ma}_2 = 0.3393 \cong \mathbf{0.339}$$

Also,

$\mathrm{Ma}_1 = 0.2 \quad \rightarrow \quad P_{01}/P_0^* = 1.2346$

$\mathrm{Ma}_2 = 0.3393 \quad \rightarrow \quad P_{02}/P_0^* = 1.1820$

Then the stagnation pressure at the exit and the pressure drop become

$$\frac{P_{02}}{P_{01}} = \frac{P_{02}/P_0^*}{P_{01}/P_0^*} = \frac{1.1820}{1.2346} = 0.9574 \quad \rightarrow \quad P_{02} = 0.9574 P_{01} = 0.9574(617\text{ kPa}) = 590.7\text{ kPa}$$

and

$$\Delta P_0 = P_{01} - P_{02} = 617.0 - 590.7 = \mathbf{26.3\,kPa}$$

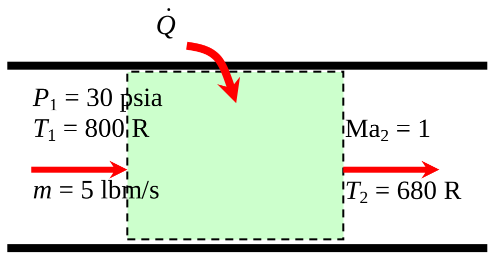
Discussion This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

12-90E

Solution Air flowing with a subsonic velocity in a round duct is accelerated by heating until the flow is choked at the exit. The rate of heat transfer and the pressure drop are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The flow is choked at the duct exit. **3** Mass flow rate remains constant.

Properties We take the properties of air to be k = 1.4, c_p = 0.2400 Btu/lbm·R, and R = 0.06855 Btu/lbm·R = 0.3704 psia·ft^3/lbm·R.



Analysis The inlet density and velocity of air are

$$\rho_1 = \frac{P_1}{RT_1} = \frac{30 \text{ psia}}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(800 \text{ R})} = 0.1012 \text{ lbm/ft}^3$$

$$V_1 = \frac{\dot{m}_{air}}{\rho_1 A_{c1}} = \frac{5 \text{ lbm/s}}{(0.1012 \text{ lbm/ft}^3)[\pi(4/12 \text{ ft})^2/4]} = 565.9 \text{ ft/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 800 \text{ R} + \frac{(565.9 \text{ ft/s})^2}{2\times 0.2400 \text{ Btu/lbm}\cdot\text{R}}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 826.7 \text{ R}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.06855 \text{ Btu/lbm}\cdot\text{R})(800 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = 1386 \text{ ft/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{565.9 \text{ ft/s}}{1386 \text{ ft/s}} = 0.4082$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-15):

$\text{Ma}_1 = 0.4082$: $T_1/T^* = 0.6310$, $P_1/P^* = 1.946$, $T_{01}/T_0^* = 0.5434$

$\text{Ma}_2 = 1$: $T_2/T^* = 1$, $P_2/P^* = 1$, $T_{02}/T_0^* = 1$

Then the exit temperature, pressure, and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.6310} \quad \rightarrow \quad T_2 = T_1/0.6310 = (800 \text{ R})/0.6310 = 1268 \text{ R}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{1.946} \quad \rightarrow \quad P_2 = P_1/2.272 = (30 \text{ psia})/1.946 = 15.4 \text{ psia}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.5434} \quad \rightarrow \quad T_{02} = T_{01}/0.1743 = (826.7 \text{ R})/0.5434 = 1521 \text{ R}$$

Then the rate of heat transfer and the pressure drop become

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (5 \text{ lbm/s})(0.2400 \text{ Btu/lbm}\cdot\text{R})(1521 - 826.7) \text{ R} = \mathbf{834 \text{ Btu/s}}$$

$$\Delta P = P_1 - P_2 = 30 - 15.4 = \mathbf{14.6 \text{ psia}}$$

Discussion Note that the entropy of air increases during this heating process, as expected.

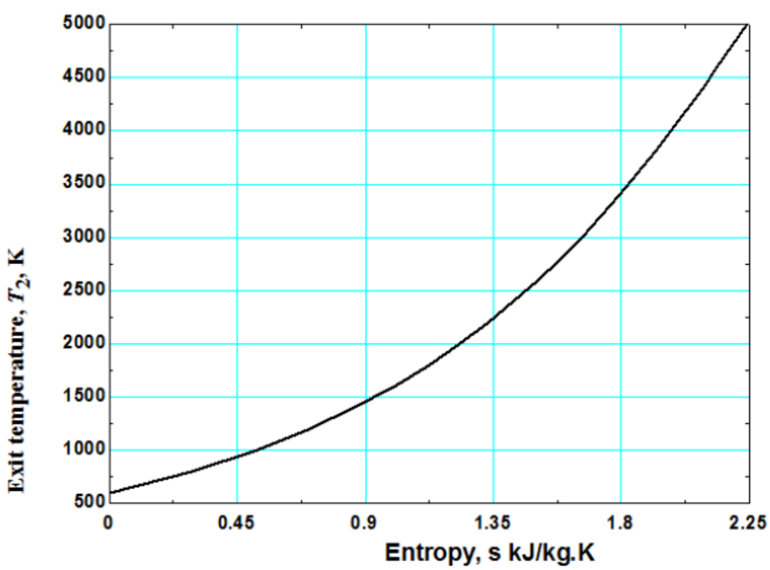
12-91

Solution Air flowing with a subsonic velocity in a duct. The variation of entropy with temperature is to be investigated as the exit temperature varies from 600 K to 5000 K in increments of 200 K. The results are to be tabulated and plotted.

Analysis We solve this problem using EES making use of Rayleigh functions. The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
k=1.4
cp=1.005
R=0.287
P1=350
T1=600
V1=70
C1=sqrt(k*R*T1*1000)
Ma1=V1/C1
T01=T1*(1+0.5*(k-1)*Ma1^2)
P01=P1*(1+0.5*(k-1)*Ma1^2)^(k/(k-1))
F1=1+0.5*(k-1)*Ma1^2
T01Ts=2*(k+1)*Ma1^2*F1/(1+k*Ma1^2)^2
P01Ps=((1+k)/(1+k*Ma1^2))*(2*F1/(k+1))^(k/(k-1))
T1Ts=(Ma1*((1+k)/(1+k*Ma1^2)))^2
P1Ps=(1+k)/(1+k*Ma1^2)
V1Vs=Ma1^2*(1+k)/(1+k*Ma1^2)
F2=1+0.5*(k-1)*Ma2^2
T02Ts=2*(k+1)*Ma2^2*F2/(1+k*Ma2^2)^2
P02Ps=((1+k)/(1+k*Ma2^2))*(2*F2/(k+1))^(k/(k-1))
T2Ts=(Ma2*((1+k)/(1+k*Ma2^2)))^2
P2Ps=(1+k)/(1+k*Ma2^2)
V2Vs=Ma2^2*(1+k)/(1+k*Ma2^2)
T02=T02Ts/T01Ts*T01
P02=P02Ps/P01Ps*P01
T2=T2Ts/T1Ts*T1
P2=P2Ps/P1Ps*P1
V2=V2Vs/V1Vs*V1
Delta_s=cp*ln(T2/T1)-R*ln(P2/P1)
```

Exit temperature T_2, K	Exit Mach number, Ma_2	Exit entropy relative to inlet, s_2, kJ/kg·K
600	0.143	0.000
800	0.166	0.292
1000	0.188	0.519
1200	0.208	0.705
1400	0.227	0.863
1600	0.245	1.001
1800	0.263	1.123
2000	0.281	1.232
2200	0.299	1.331
2400	0.316	1.423
2600	0.333	1.507
2800	0.351	1.586
3000	0.369	1.660
3200	0.387	1.729
3400	0.406	1.795
3600	0.426	1.858
3800	0.446	1.918
4000	0.467	1.975
4200	0.490	2.031
4400	0.515	2.085
4600	0.541	2.138
4800	0.571	2.190
5000	0.606	2.242



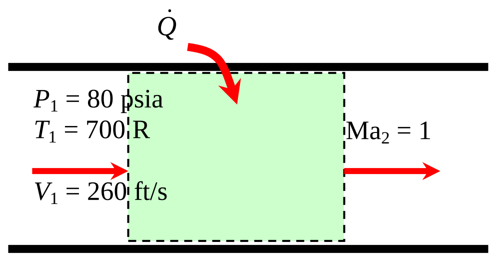
Discussion Note that the entropy of air increases during this heating process, as expected.

12-92E

Solution Air flowing with a subsonic velocity in a square duct is accelerated by heating until the flow is choked at the exit. The rate of heat transfer and the entropy change are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The flow is choked at the duct exit. **3** Mass flow rate remains constant.

Properties We take the properties of air to be k = 1.4, c_p = 0.2400 Btu/lbm·R, and R = 0.06855 Btu/lbm·R = 0.3704 psia·ft^3/lbm·R.



Analysis The inlet density and mass flow rate of air are

$$\rho_1 = \frac{P_1}{RT_1} = \frac{80 \text{ psia}}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(700 \text{ R})} = 0.3085 \text{ lbm/ft}^3$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (0.3085 \text{ lbm/ft}^3)(6\times 6/144 \text{ ft}^2)(260 \text{ ft/s}) = 20.06 \text{ lbm/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 700 \text{ R} + \frac{(260 \text{ ft/s})^2}{2\times 0.2400 \text{ Btu/lbm}\cdot\text{R}}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 705.6 \text{ R}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.06855 \text{ Btu/lbm}\cdot\text{R})(700 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = 1297 \text{ ft/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{260 \text{ ft/s}}{1297 \text{ ft/s}} = 0.2005$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-15):

$\text{Ma}_1 = 0.2005$: $T_1/T^* = 0.2075$, $P_1/P^* = 2.272$, $T_{01}/T_0^* = 0.1743$
$\text{Ma}_2 = 1$: $T_2/T^* = 1$, $P_2/P^* = 1$, $T_{02}/T_0^* = 1$

Then the exit temperature, pressure, and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.2075} \quad \rightarrow \quad T_2 = T_1/0.2075 = (700 \text{ R})/0.2075 = 3374 \text{ R}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{2.272} \quad \rightarrow \quad P_2 = P_1/2.272 = (80 \text{ psia})/2.272 = 35.2 \text{ psia}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.1743} \quad \rightarrow \quad T_{02} = T_{01}/0.1743 = (705.6 \text{ R})/0.1743 = 4048 \text{ R}$$

Then the rate of heat transfer and entropy change become

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (20.06 \text{ lbm/s})(0.2400 \text{ Btu/lbm}\cdot\text{R})(4048 - 705.6) \text{ R} = \mathbf{16{,}090 \text{ Btu/s}}$$

$$\Delta s = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (0.2400 \text{ Btu/lbm}\cdot\text{R})\ln\frac{3374 \text{ R}}{700 \text{ R}} - (0.06855 \text{ Btu/lbm}\cdot\text{R})\ln\frac{35.2 \text{ psia}}{80 \text{ psia}} = \mathbf{0.434 \text{ Btu/lbm}\cdot\text{R}}$$

Discussion Note that the entropy of air increases during this heating process, as expected.

12-93

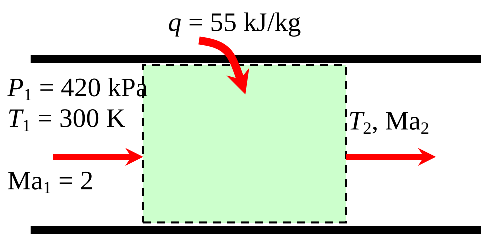
Solution Fuel is burned in a rectangular duct with compressed air. For specified heat transfer, the exit temperature and Mach number are to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K.

Analysis The stagnation temperature and Mach number at the inlet are

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 347.2\text{ m/s}$$



$$V_1 = \text{Ma}_1 c_1 = 2(347.2\text{ m/s}) = 694.4\text{ m/s}$$

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 300\text{ K} + \frac{(694.4\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 539.9\text{ K}$$

The exit stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{02} = T_{01} + \frac{q}{c_p} = 539.9\text{ K} + \frac{55\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 594.6\text{ K}$$

The maximum value of stagnation temperature T_0^* occurs at Ma = 1, and its value can be determined from Table A-15 or from the appropriate relation. At Ma$_1$ = 2 we read T_{01}/T_0^* = 0.7934. Therefore,

$$T_0^* = \frac{T_{01}}{0.7934} = \frac{539.9\text{ K}}{0.7934} = 680.5\text{ K}$$

The stagnation temperature ratio at the exit and the Mach number corresponding to it are, from Table A-15,

$$\frac{T_{02}}{T_0^*} = \frac{594.6\text{ K}}{680.5\text{ K}} = 0.8738 \quad\rightarrow\quad \text{Ma}_2 = 1.642 \cong \mathbf{1.64}$$

Also,

Ma$_1$ = 2 $\rightarrow$ T_1/T^* = 0.5289
Ma$_2$ = 1.642 $\rightarrow$ T_2/T^* = 0.6812

Then the exit temperature becomes

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.6812}{0.5289} = 1.288 \quad\rightarrow\quad T_2 = 1.288T_1 = 1.288(300\text{ K}) = \mathbf{386K}$$

Discussion Note that the temperature increases during this supersonic Rayleigh flow with heating. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

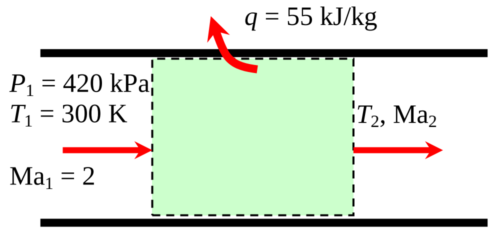
12-94

Solution Compressed air is cooled as it flows in a rectangular duct. For specified heat rejection, the exit temperature and Mach number are to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K.

Analysis The stagnation temperature and Mach number at the inlet are



$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 347.2\text{ m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 2(347.2\text{ m/s}) = 694.4\text{ m/s}$$

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 300\text{ K} + \frac{(694.4\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 539.9\text{ K}$$

The exit stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{02} = T_{01} + \frac{q}{c_p} = 539.9\text{ K} + \frac{-55\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 485.2\text{ K}$$

The maximum value of stagnation temperature T_0^* occurs at Ma = 1, and its value can be determined from Table A-15 or from the appropriate relation. At Ma_1 = 2 we read T_{01}/T_0^* = 0.7934. Therefore,

$$T_0^* = \frac{T_{01}}{0.7934} = \frac{539.9\text{ K}}{0.7934} = 680.5\text{ K}$$

The stagnation temperature ratio at the exit and the Mach number corresponding to it are, from Table A-15,

$$\frac{T_{02}}{T_0^*} = \frac{485.2\text{ K}}{680.5\text{ K}} = 0.7130 \quad\rightarrow\quad \text{Ma}_2 = 2.479 \cong \mathbf{2.48}$$

Also,

$\text{Ma}_1 = 2 \quad\rightarrow\quad T_1/T^* = 0.5289$

$\text{Ma}_2 = 2.479 \quad\rightarrow\quad T_2/T^* = 0.3838$

Then the exit temperature becomes

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.3838}{0.5289} = 0.7257 \quad\rightarrow\quad T_2 = 0.7257T_1 = 0.7257(300\text{ K}) = \mathbf{218K}$$

Discussion Note that the temperature decreases and Mach number increases during this supersonic Rayleigh flow with cooling. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

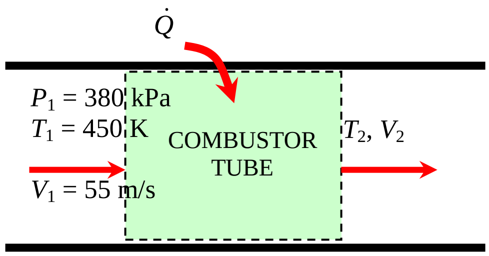
12-95

Solution Fuel is burned in a tubular combustion chamber with compressed air. For a specified exit Mach number, the exit temperature and the rate of fuel consumption are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Combustion is complete, and it is treated as a heat addition process, with no change in the chemical composition of flow. **3** The increase in mass flow rate due to fuel injection is disregarded.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis The inlet density and mass flow rate of air are



$$\rho_1 = \frac{P_1}{RT_1} = \frac{380\text{ kPa}}{(0.287\text{ kJ/kgK})(450\text{ K})} = 2.942\text{ kg/m}^3$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (2.942\text{ kg/m}^3)[\pi(0.16\text{ m})^2/4](55\text{ m/s}) = 3.254\text{ kg/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 450\text{ K} + \frac{(55\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 451.5\text{ K}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(450\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 425.2\text{ m/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{55\text{ m/s}}{425.2\text{ m/s}} = 0.1293$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-15) (We used analytical functions):

$\text{Ma}_1 = 0.1293$: $T_1/T^* = 0.09201$, $T_{01}/T^* = 0.07693$, $V_1/V^* = 0.03923$

$\text{Ma}_2 = 0.8$: $T_2/T^* = 1.0255$, $T_{02}/T^* = 0.9639$, $V_2/V^* = 0.8101$

The exit temperature, stagnation temperature, and velocity are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1.0255}{0.09201} = 11.146 \quad\rightarrow\quad T_2 = 11.146T_1 = 11.146(450\text{ K}) = \mathbf{5016K}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{0.9639}{0.07693} = 12.530 \quad\rightarrow\quad T_{02} = 12.530T_{01} = 12.530(451.5\text{ K}) = 5658\text{ K}$$

$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{0.8101}{0.03923} = 20.650 \quad\rightarrow\quad V_2 = 20.650V_1 = 20.650(55\text{ m/s}) = 1136\text{ m/s}$$

Then the mass flow rate of the fuel is determined to be

$$q = c_p(T_{02} - T_{01}) = (1.005\text{ kJ/kg}\cdot\text{K})(5658 - 451.5)\text{ K} = 5232\text{ kJ/kg}$$

$$\dot{Q} = \dot{m}_{\text{air}}q = (3.254\text{ kg/s})(5232\text{ kJ/kg}) = 17{,}024\text{ kW}$$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}}{HV} = \frac{17{,}024\text{ kJ/s}}{39{,}000\text{ kJ/kg}} = \mathbf{0.4365kg/s}$$

Discussion Note that both the temperature and velocity increase during this subsonic Rayleigh flow with heating, as expected. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

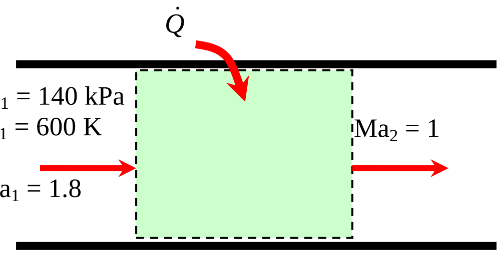
12-96

Solution Air flowing at a supersonic velocity in a duct is decelerated by heating. The highest temperature air can be heated by heat addition and the rate of heat transfer are to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Mass flow rate remains constant.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis Heat transfer will stop when the flow is choked, and thus $\mathrm{Ma}_2 = V_2/c_2 = 1$. Knowing stagnation properties, the static properties are determined to be



$$T_1 = T_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-1} = (600\text{ K})\left(1+\frac{1.4\text{-}1}{2}1.8^2\right)^{-1} = 364.1\text{ K}$$

$$P_1 = P_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-k/(k-1)} = (140\text{ K})\left(1+\frac{1.4\text{-}1}{2}1.8^2\right)^{-1.4/0.4}$$
$$= 24.37\text{ kPa}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{24.37\text{ kPa}}{(0.287\text{ kJ/kgK})(364.1\text{K})} = 0.2332\text{ kg/m}^3$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(364.1\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 382.5\text{ m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 1.8(382.5\text{ m/s}) = 688.5\text{ m/s}$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (0.2332\text{ kg/m}^3)[\pi(0.07\text{ m})^2/4](688.5\text{ m/s}) = 0.6179\text{ kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-15):

$\mathrm{Ma}_1 = 1.8$: $T_1/T^* = 0.6089$, $T_{01}/T_0^* = 0.8363$
$\mathrm{Ma}_2 = 1$: $T_2/T^* = 1$, $T_{02}/T_0^* = 1$

Then the exit temperature and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.6089} \quad\rightarrow\quad T_2 = T_1/0.6089 = (364.1\text{ K})/0.6089 = \mathbf{598K}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.8363} \quad\rightarrow\quad T_{02} = T_{01}/0.8363 = (600\text{ K})/0.8363 = 717.4\text{ K} \cong \mathbf{717\ K}$$

Finally, the rate of heat transfer is

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (0.6179\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(717.4 - 600)\text{ K} = \mathbf{72.9kW}$$

Discussion Note that this is the highest temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature will cause the mass flow rate to decrease. Also, once the sonic conditions are reached, the thermodynamic temperature can be increased further by cooling the fluid and reducing the velocity (see the *T*-*s* diagram for Rayleigh flow).

Adiabatic Duct Flow with Friction (Fanno Flow)

12-97C

Solution We are to discuss the effect of friction on velocity in Fanno flow.

Analysis Friction **increases the flow velocity in subsonic Fanno flow**, but **decreases the flow velocity in supersonic flow**.

Discussion These results may not be intuitive, but they come from following the Fanno line, which satisfies the conservation equations.

12-98C

Solution We are to discuss the T-s diagram for Fanno flow.

Analysis The points on the Fanno line on a *T-s* diagram represent the **states that satisfy the conservation of mass, momentum, and energy equations as well as the property relations for a given inlet state**. Therefore, for a given initial state, the fluid cannot exist at any downstream state outside the Fanno line on a *T-s* diagram.

Discussion The *T-s* diagram is quite useful, since any downstream state must lie on the Fanno line.

12-99C

Solution We are to discuss the effect of friction on the entropy during Fanno flow.

Analysis In Fanno flow, the effect of friction is **always to increase the entropy of the fluid**. Therefore Fanno flow always proceeds in the direction of increasing entropy.

Discussion To do otherwise would violate the second law of thermodynamics.

12-100C

Solution We are to discuss what happens to supersonic Fanno flow, initially sonic at the exit, when the duct is extended.

Analysis **The flow at the duct exit remains sonic**. The **mass flow rate must remain constant** since upstream conditions are not affected by the added duct length.

Discussion The mass flow rate is fixed by the upstream stagnation conditions and the size of the throat – therefore, the mass flow rate does not change by extending the duct. However, a *shock wave* appears in the duct when it is extended.

12-101C

Solution We are to examine what happens when the Mach number of air decreases in supersonic Fanno flow.

Analysis During supersonic Fanno flow, the **stagnation temperature T_0 remains constant**, **stagnation pressure P_0 decreases**, and **entropy s increases**.

Discussion Friction leads to irreversible losses, which are felt as a loss of stagnation pressure and an increase of entropy. However, since the flow is adiabatic, the stagnation temperature does not change downstream.

12-102C

Solution We are to discuss the characteristic aspect of Fanno flow and its main assumptions.

Analysis The characteristic aspect of Fanno flow is its **consideration of friction**. The main assumptions associated with Fanno flow are: the flow is **steady**, **one-dimensional**, and **adiabatic** through a **constant-area duct**, and the fluid is an **ideal gas** with **constant specific heats**.

Discussion Compared to Rayleigh flow, Fanno flow accounts for friction but neglects heat transfer effects, whereas Rayleigh flow accounts for heat transfer but neglects frictional effects.

12-103C

Solution We are to discuss what happens to choked subsonic Fanno flow when the duct is extended.

Analysis The flow is choked, and thus **the flow at the duct exit must remain sonic**. The **mass flow rate has to decrease** as a result of extending the duct length in order to compensate.

Discussion Since there is no way for the flow to become supersonic (e.g., there is no throat), the upstream flow must adjust itself such that the flow at the exit plan remains sonic.

12-104C

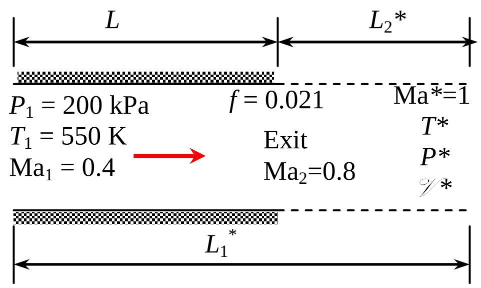
Solution We are to examine what happens when the Mach number of air increases in subsonic Fanno flow.

Analysis During subsonic Fanno flow, the **stagnation temperature T_0 remains constant**, **stagnation pressure P_0 decreases**, and **entropy s increases**.

Discussion Friction leads to irreversible losses, which are felt as a loss of stagnation pressure and an increase of entropy. However, since the flow is adiabatic, the stagnation temperature does not change downstream.

12-105

Solution Subsonic airflow in a constant cross-sectional area adiabatic duct is considered. For a specified exit Mach number, the duct length, temperature, pressure, and velocity at the duct exit are to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor remains constant along the duct.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K. The average friction factor is given to be f = 0.021.

Analysis The inlet velocity is

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(550\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 470.1\text{ m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 0.4(470.1\text{ m/s}) = 188.0\text{ m/s}$$

The Fanno flow functions corresponding to the inlet and exit Mach numbers are, from Table A-16,

Ma_1 = 0.4: $(fL^*/D_h)_1$ = 2.3085 T_1/T^* = 1.1628, P_1/P^* = 2.6958, V_1/V^* = 0.4313
Ma_2 = 0.8: $(fL^*/D_h)_2$ = 0.0723 T_2/T^* = 1.0638, P_2/P^* = 1.2893, V_2/V^* = 0.8251

Then the temperature, pressure, and velocity at the duct exit are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1.0638}{1.1628} = 0.9149 \quad \rightarrow \quad T_2 = 0.9149T_1 = 0.9149(550\text{ K}) = \mathbf{503.2K}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1.2893}{2.6958} = 0.4783 \quad \rightarrow \quad P_2 = 0.4783P_1 = 0.4783(200\text{ kPa}) = \mathbf{95.65kPa}$$

$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{0.8251}{0.4313} = 1.9131 \quad \rightarrow \quad V_2 = 1.9131V_1 = 1.9131(188.0\text{ m/s}) = \mathbf{359.7m/s}$$

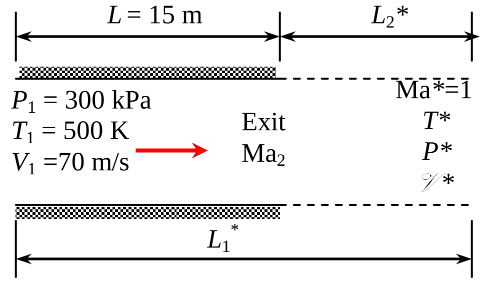
Finally, the actual duct length is determined to be

$$L = L_1^* - L_2^* = \left(\frac{fL_1^*}{D_h} - \frac{fL_2^*}{D_h}\right)\frac{D_h}{f} = (2.3085 - 0.0723)\frac{0.12\text{ m}}{0.021} = \mathbf{12.8m}$$

Discussion Note that it takes a duct length of 12.8 m for the Mach number to increase from 0.4 to 0.8. The Mach number rises at a much higher rate as sonic conditions are approached. The maximum (or sonic) duct lengths at the inlet and exit states in this case are L_1^* = 13.2 m and L_2^* = 0.413 m. Therefore, the flow would reach sonic conditions if a 0.413-m long section were added to the existing duct.

12-106

Solution Air enters a constant-area adiabatic duct of given length at a specified state. The exit Mach number, exit velocity, and the mass flow rate are to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor is constant along the duct.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K. The friction factor is given to be f = 0.023.

Analysis The first thing we need to know is whether the flow is choked at the exit or not. Therefore, we first determine the inlet Mach number and the corresponding value of the function fL^*/D_h ,

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(500\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 448.2\text{ m/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{70\text{ m/s}}{448.2\text{ m/s}} = 0.1562$$

Corresponding to this Mach number we calculate (or read) from Table A-16), $(fL^*/D_h)_1$ = 25.540. Also, using the actual duct length L, we have

$$\frac{fL}{D_h} = \frac{(0.023)(15\text{ m})}{0.04\text{ m}} = 8.625 \;\; < \;\; 25.540$$

Therefore, flow is *not* choked and exit Mach number is less than 1. Noting that $L = L_1^* - L_2^*$, the function fL^*/D_h at the exit state is calculated from

$$\left(\frac{fL^*}{D_h}\right)_2 = \left(\frac{fL^*}{D_h}\right)_1 - \frac{fL}{D_h} = 25.540 - 8.625 = 16.915$$

The Mach number corresponding to this value of fL^*/D is obtained from Table A-16 to be

Ma$_2$ = **0.187**

which is the Mach number at the duct exit. The mass flow rate of air is determined from the inlet conditions to be

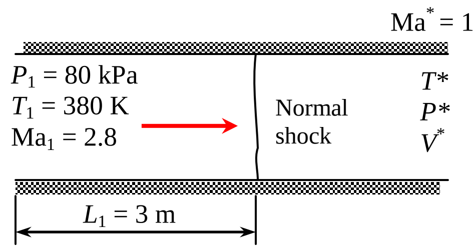
$$\rho_1 = \frac{P_1}{RT_1} = \frac{300\text{ kPa}}{(0.287\text{ kJ/kgK})(500\text{ K})}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 2.091\text{ kg/m}^3$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (2.091\text{ kg/m}^3)[\pi(0.04\text{ m})^2/4](70\text{ m/s}) = \mathbf{0.184\ kg/s}$$

Discussion It can be shown that L_2^* = 29.4 m, indicating that it takes a duct length of 15 m for the Mach number to increase from 0.156 to 0.187, but only 29.4 m to increase from 0.187 to 1. Therefore, the Mach number rises at a much higher rate as sonic conditions are approached.

12-107

Solution Air enters a constant-area adiabatic duct at a specified state, and undergoes a normal shock at a specified location. The exit velocity, temperature, and pressure are to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor is constant along the duct.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K. The friction factor is given to be f = 0.007.

Analysis The Fanno flow functions corresponding to the inlet Mach number of 2.8 are, from Table A-16,

$\text{Ma}_1 = 2.8$: $(fL^*/D_h)_1 = 0.4898$ $\quad T_1/T^* = 0.4673,\quad P_1/P^* = 0.2441$

First we check to make sure that the flow everywhere upstream the shock is supersonic. The required duct length from the inlet L_1^* for the flow to reach sonic conditions is

$$L_1^* = 0.4898\frac{D}{f} = 0.4898\frac{0.05\text{ m}}{0.007} = 3.50\text{ m}$$

which is greater than the actual length 3 m. Therefore, the flow is indeed supersonic when the normal shock occurs at the indicated location. Also, using the actual duct length L_1, we have $\frac{fL_1}{D_h} = \frac{(0.007)(3\text{ m})}{0.05\text{ m}} = 0.4200$. Noting that $L_1 = L_1^* - L_2^*$, the function fL^*/D_h at the exit state and the corresponding Mach number are

$$\left(\frac{fL^*}{D_h}\right)_2 = \left(\frac{fL^*}{D_h}\right)_1 - \frac{fL_1}{D_h} = 0.4898 - 0.4200 = 0.0698 \quad\to\quad \text{Ma}_2 = 1.315$$

From Table A-16, at Ma_2 =1.315: T_2/T^* = 0.8918 and P_2/P^* = 0.7183. Then the temperature, pressure, and velocity before the shock are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.8918}{0.4673} = 1.9084 \qquad\to\qquad T_2 = 1.9084T_1 = 1.9084(380\text{ K}) = 725.2\text{ K}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{0.7183}{0.2441} = 2.9426 \qquad\to\qquad P_2 = 2.9426P_1 = 2.9426(80\text{ kPa}) = 235.4\text{ kPa}$$

The normal shock functions corresponding to a Mach number of 1.315 are, from Table A-14,

Ma_2 =1.315: $\text{Ma}_3 = 0.7786,\quad T_3/T_2 = 1.2001,\quad P_3/P_2 = 1.8495$

Then the temperature and pressure after the shock become

$$T_3 = 1.2001T_2 = 1.2001(725.2\text{ K}) = 870.3\text{ K} \quad\text{and}\quad P_3 = 1.8495P_2 = 1.8495(235.4\text{ kPa}) = 435.4\text{ kPa}$$

Sonic conditions exist at the duct exit, and the flow downstream the shock is still Fanno flow. From Table A-16,

$\text{Ma}_3 = 0.7786$: $\quad T_3/T^* = 1.0702,\quad P_3/P^* = 1.3286$
$\text{Ma}_4 = 1$: $\quad T_4/T^* = 1,\quad P_4/P^* = 1$

Then the temperature, pressure, and velocity at the duct exit are determined to be

$$\frac{T_4}{T_3} = \frac{T_4/T^*}{T_3/T^*} = \frac{1}{1.0702} \qquad\to\qquad T_4 = T_3/1.0702 = (870.3\text{ K})/1.0702 = \mathbf{813\,K}$$

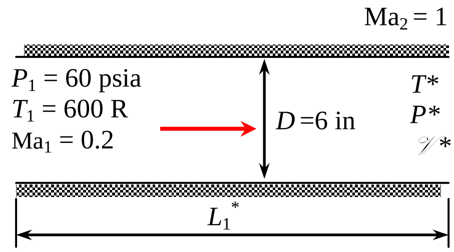
$$\frac{P_4}{P_3} = \frac{P_4/P^*}{P_3/P^*} = \frac{1}{1.3286} \qquad\to\qquad P_4 = P_3/1.3286 = (435.4\text{ kPa})/1.3286 = \mathbf{328\,kPa}$$

$$V_4 = \text{Ma}_4 c_4 = (1)\sqrt{kRT_4} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(813\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{572\,m/s}$$

Discussion It can be shown that L_3^* = 0.67 m, and thus the total length of this duct is 3.67 m. If the duct is extended, the normal shock will move further upstream, and eventually to the inlet of the duct.

12-108E

Solution Helium enters a constant-area adiabatic duct at a specified state, and leaves at sonic state. The maximum duct length that will not cause the mass flow rate to be reduced is to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor is constant along the duct.

Properties We take the properties of helium to be k = 1.667, c_p = 1.2403 Btu/lbm·R, and R = 0.4961 Btu/lbm·R. The friction factor is given to be f = 0.025.

Analysis The Fanno flow function fL^*/D corresponding to the inlet Mach number of 0.2 is (Table A-16)

$$\frac{fL_1^*}{D} = 14.5333$$

Noting that * denotes sonic conditions, which exist at the exit state, the duct length is determined to be

$$L_1^* = 14.5333D/f = 14.5333(6/12 \text{ ft})/0.025 = \mathbf{291 \text{ ft}}$$

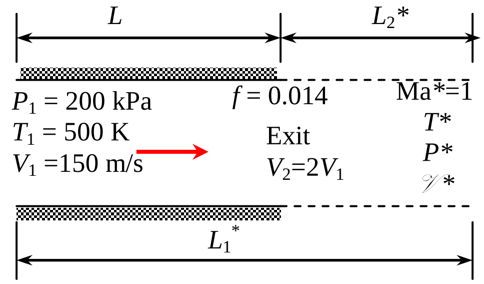
Thus, for the given friction factor, the duct length must be 291 ft for the Mach number to reach Ma = 1 at the duct exit.

Discussion This problem can also be solved using equations instead of tabulated values for the Fanno functions.

12-109

Solution Subsonic airflow in a constant cross-sectional area adiabatic duct is considered. The duct length from the inlet where the inlet velocity doubles and the pressure drop in that section are to be determined.

Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor remains constant along the duct.



Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K. The average friction factor is given to be f = 0.014.

Analysis The inlet Mach number is

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(500\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 448.2\text{ m/s} \quad \rightarrow \quad \text{Ma}_1 = \frac{V_1}{c_1} = \frac{150\text{ m/s}}{448.2\text{ m/s}} = 0.3347$$

The Fanno flow functions corresponding to the inlet and exit Mach numbers are, from Table A-16,

$\text{Ma}_1 = 0.3347$: $(fL^*/D_h)_1 = 3.924$ $P_1/P^* = 3.2373$, $V_1/V^* = 0.3626$

Therefore, $V_1 = 0.3626V^*$. Then the Fanno function V_2/V^* becomes $\dfrac{V_2}{V^*} = \dfrac{2V_1}{V^*} = \dfrac{2\times 0.3626V^*}{V^*} = 0.7252$.

The corresponding Mach number and Fanno flow functions are, from Table A-16,

$\text{Ma}_2 = 0.693$, $(fL^*/D_h)_1 = 0.2220$, and $P_2/P^* = 1.5099$.

Then the duct length where the velocity doubles, the exit pressure, and the pressure drop become

$$L = L_1^* - L_2^* = \left(\frac{fL_1^*}{D_h} - \frac{fL_2^*}{D_h}\right)\frac{D_h}{f} = (3.924 - 0.2220)\frac{0.15\text{ m}}{0.014} = \mathbf{39.7\text{ m}}$$

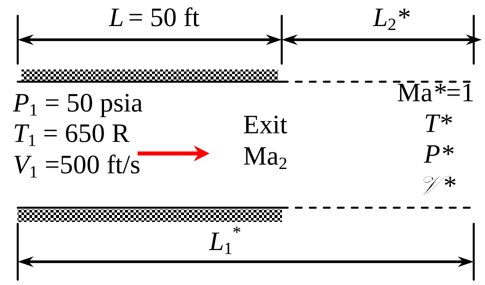
$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1.5099}{3.2373} = 0.4664 \quad \rightarrow \quad P_2 = 0.4664P_1 = 0.4664(200\text{ kPa}) = 93.3\text{ kPa}$$

$$\Delta P = P_1 - P_2 = 200 - 93.3 = 106.7\text{ kPa} \cong \mathbf{107\text{ kPa}}$$

Discussion Note that it takes a duct length of 39.7 m for the velocity to double, and the Mach number to increase from 0.3347 to 0.693. The maximum (or sonic) duct lengths at the inlet and exit states in this case are L_1^* = 42.1 m and L_2^* = 2.38 m. Therefore, the flow would reach sonic conditions if there is an additional 2.38 m of duct length.

12-110E

Solution Air enters a constant-area adiabatic duct of given length at a specified state. The velocity, temperature, and pressure at the duct exit are to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor is constant along the duct.

Properties We take the properties of helium to be k = 1.4, c_p = 0.2400 Btu/lbm·R, and R = 0.06855 Btu/lbm·R = 0.3704 psia·ft^3/lbm·R. The friction factor is given to be f = 0.025.

Analysis The first thing we need to know is whether the flow is choked at the exit or not. Therefore, we first determine the inlet Mach number and the corresponding value of the function fL^*/D_h ,

$$T_1 = T_{01} - \frac{V_1^2}{2c_p} = 650\text{ R} - \frac{(500\text{ ft/s})^2}{2\times 0.2400\text{ Btu/lbm}\cdot\text{R}}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 629.2\text{ R}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.06855\text{ Btu/lbm}\cdot\text{R})(629.2\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = 1230\text{ ft/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{500\text{ m/s}}{1230\text{ ft/s}} = 0.4066$$

Corresponding to this Mach number we calculate (or read) from Table A-16), $(fL^*/D_h)_1$ = 2.1911. Also, using the actual duct length L, we have

$$\frac{fL}{D_h} = \frac{(0.02)(50\text{ ft})}{6/12\text{ ft}} = 2 \;<\; 2.1911$$

Therefore, the flow is *not* choked and exit Mach number is less than 1. Noting that $L = L_1^* - L_2^*$, the function fL^*/D_h at the exit state is calculated from

$$\left(\frac{fL^*}{D_h}\right)_2 = \left(\frac{fL^*}{D_h}\right)_1 - \frac{fL}{D_h} = 2.1911 - 2 = 0.1911$$

The Mach number corresponding to this value of fL^*/D is obtained from Table A-16 to be Ma$_2$ = 0.7091. The Fanno flow functions corresponding to the inlet and exit Mach numbers are, from Table A-16,

Ma$_1$ = 0.4066: T_1/T^* = 1.1616, P_1/P^* = 2.6504, V_1/V^* = 0.4383
Ma$_2$ = 0.7091: T_2/T^* = 1.0903, P_2/P^* = 1.4726, V_2/V^* = 0.7404

Then the temperature, pressure, and velocity at the duct exit are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1.0903}{1.1616} = 0.9386 \quad\rightarrow\quad T_2 = 0.9386T_1 = 0.9386(629.2\text{ R}) = \mathbf{591R}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1.4726}{2.6504} = 0.5556 \quad\rightarrow\quad P_2 = 0.5556P_1 = 0.5556(50\text{ psia}) = \mathbf{27.8psia}$$

$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{0.7404}{0.4383} = 1.6893 \quad\rightarrow\quad V_2 = 1.6893V_1 = 1.6893(500\text{ ft/s}) = \mathbf{845\text{ ft/s}}$$

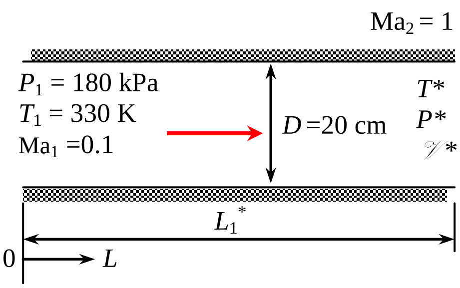
Discussion It can be shown that L_2^* = 4.8 ft, indicating that it takes a duct length of 50 ft for the Mach number to increase from 0.4066 to 0.7091, but only 4.8 ft to increase from 0.7091 to 1. Therefore, the Mach number rises at a much higher rate as sonic conditions are approached.

12-111

Solution Choked subsonic airflow in a constant cross-sectional area adiabatic duct is considered. The variation of duct length with Mach number is to be investigated, and the results are to be plotted.

Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor remains constant along the duct.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K. The average friction factor is given to be f = 0.02.

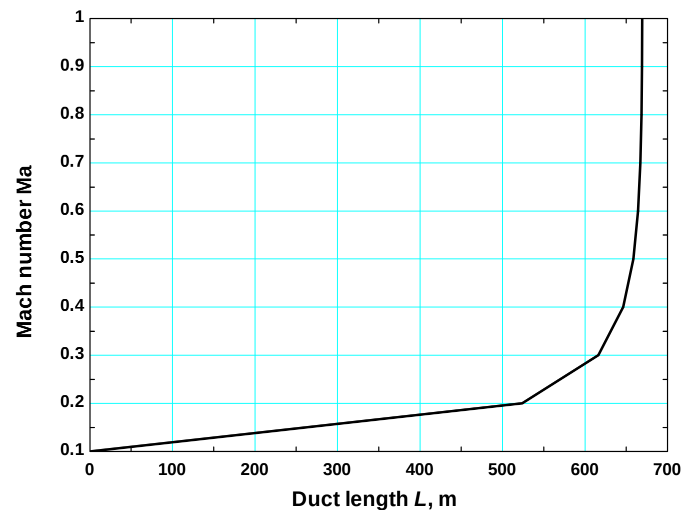


Analysis The flow is choked, and thus Ma_2 =1. Corresponding to the inlet Mach number of Ma_1 = 0.1 we have, from Table A-16, fL^*/D_h = 66.922, Therefore, the original duct length is

$$L_1^* = 66.922\frac{D}{f} = 66.922\frac{0.20\text{ m}}{0.02} = 669\text{ m}$$

Repeating the calculations for different Ma_2 as it varies from 0.1 to 1 results in the following table for the location on the duct from the inlet. The EES *Equations* window is printed below, along with the plotted results.

Mach number, Ma	Duct length L, m
0.10	0
0.20	524
0.30	616
0.40	646
0.50	659
0.60	664
0.70	667
0.80	668
0.90	669
1.00	669



EES program:

```
k=1.4
cp=1.005
R=0.287

P1=180
T1=330
Ma1=0.1
"Ma2=1"
f=0.02
D=0.2

C1=sqrt(k*R*T1*1000)
Ma1=V1/C1
T01=T02
T01=T1*(1+0.5*(k-1)*Ma1^2)
T02=T2*(1+0.5*(k-1)*Ma2^2)
P01=P1*(1+0.5*(k-1)*Ma1^2)^(k/(k-1))

rho1=P1/(R*T1)
```

```
Ac=pi*D^2/4
mair=rho1*Ac*V1

P01Ps=((2+(k-1)*Ma1^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma1
P1Ps=((k+1)/(2+(k-1)*Ma1^2))^0.5/Ma1
T1Ts=(k+1)/(2+(k-1)*Ma1^2)
R1Rs=((2+(k-1)*Ma1^2)/(k+1))^0.5/Ma1
V1Vs=1/R1Rs
fLs1=(1-Ma1^2)/(k*Ma1^2)+(k+1)/(2*k)*ln((k+1)*Ma1^2/(2+(k-1)*Ma1^2))
Ls1=fLs1*D/f

P02Ps=((2+(k-1)*Ma2^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma2
P2Ps=((k+1)/(2+(k-1)*Ma2^2))^0.5/Ma2
T2Ts=(k+1)/(2+(k-1)*Ma2^2)
R2Rs=((2+(k-1)*Ma2^2)/(k+1))^0.5/Ma2
V2Vs=1/R2Rs
fLs2=(1-Ma2^2)/(k*Ma2^2)+(k+1)/(2*k)*ln((k+1)*Ma2^2/(2+(k-1)*Ma2^2))
Ls2=fLs2*D/f
L=Ls1-Ls2

P02=P02Ps/P01Ps*P01
P2=P2Ps/P1Ps*P1
V2=V2Vs/V1Vs*V1
```

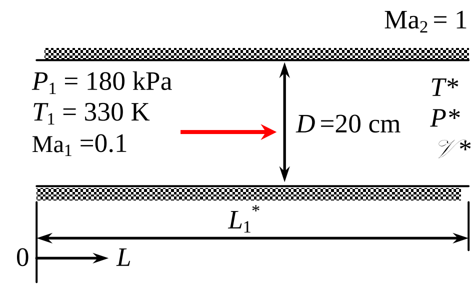
Discussion Note that the Mach number increases very mildly at the beginning, and then rapidly near the duct outlet. It takes 262 m of duct length for Mach number to increase from 0.1 to 0.2, but only 1 m to increase from 0.7 to 1.

12-112

Solution Choked subsonic airflow in a constant cross-sectional area adiabatic duct is considered. The variation of duct length with Mach number is to be investigated, and the results are to be plotted.

Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor remains constant along the duct.

Properties We take the properties of helium to be k = 1.667, c_p = 5.193 kJ/kg·K, and R = 2.077 kJ/kg·K. The average friction factor is given to be f = 0.02.

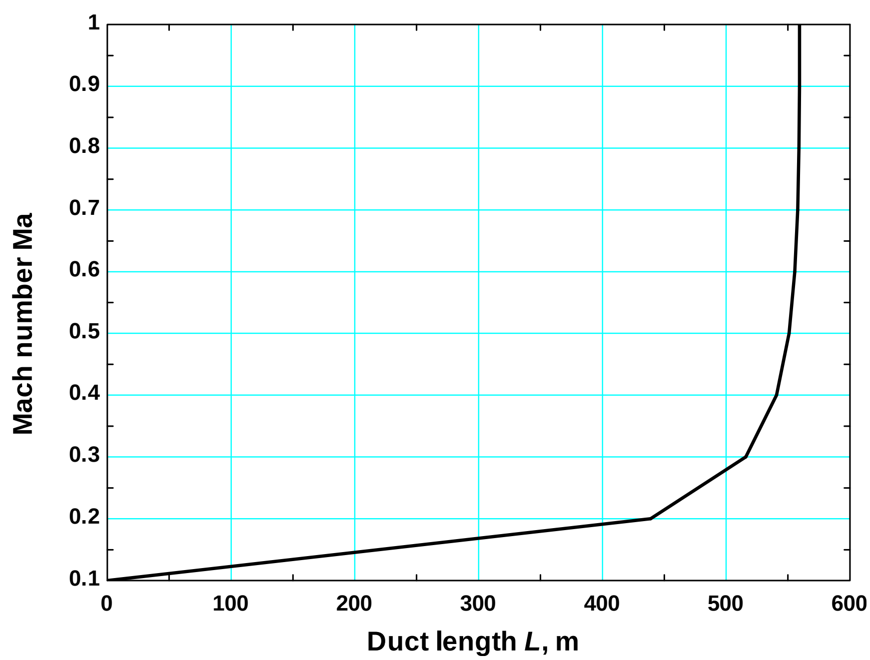


Analysis The flow is choked, and thus Ma_2 =1. Corresponding to the inlet Mach number of Ma_1 = 0.1 we have, from Table A-16, fL^*/D_h = 66.922, Therefore, the original duct length is

$$L_1^* = 66.922\frac{D}{f} = 66.922\frac{0.20\text{ m}}{0.02} = 669\text{ m}$$

Repeating the calculations for different Ma_2 as it varies from 0.1 to 1 results in the following table for the location on the duct from the inlet. The EES *Equations* window is printed below, along with the plotted results.

Mach number, Ma	Duct length L, m
0.10	0
0.20	439
0.30	516
0.40	541
0.50	551
0.60	555
0.70	558
0.80	559
0.90	559
1.00	559



EES program:

```
k=1.667
cp=5.193
R=2.077

P1=180
T1=330
Ma1=0.1
"Ma2=1"
f=0.02
D=0.2
```

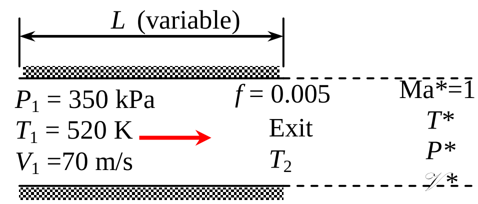
```
C1=sqrt(k*R*T1*1000)
Ma1=V1/C1
T01=T02
T01=T1*(1+0.5*(k-1)*Ma1^2)
T02=T2*(1+0.5*(k-1)*Ma2^2)
P01=P1*(1+0.5*(k-1)*Ma1^2)^(k/(k-1))

rho1=P1/(R*T1)
Ac=pi*D^2/4
mair=rho1*Ac*V1

P01Ps=((2+(k-1)*Ma1^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma1
P1Ps=((k+1)/(2+(k-1)*Ma1^2))^0.5/Ma1
T1Ts=(k+1)/(2+(k-1)*Ma1^2)
R1Rs=((2+(k-1)*Ma1^2)/(k+1))^0.5/Ma1
V1Vs=1/R1Rs
fLs1=(1-Ma1^2)/(k*Ma1^2)+(k+1)/(2*k)*ln((k+1)*Ma1^2/(2+(k-1)*Ma1^2))
Ls1=fLs1*D/f

P02Ps=((2+(k-1)*Ma2^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma2
P2Ps=((k+1)/(2+(k-1)*Ma2^2))^0.5/Ma2
T2Ts=(k+1)/(2+(k-1)*Ma2^2)
R2Rs=((2+(k-1)*Ma2^2)/(k+1))^0.5/Ma2
V2Vs=1/R2Rs
fLs2=(1-Ma2^2)/(k*Ma2^2)+(k+1)/(2*k)*ln((k+1)*Ma2^2/(2+(k-1)*Ma2^2))
Ls2=fLs2*D/f
L=Ls1-Ls2

P02=P02Ps/P01Ps*P01
P2=P2Ps/P1Ps*P1
V2=V2Vs/V1Vs*V1
```

Discussion Note that the Mach number increases very mildly at the beginning, and then rapidly near the duct outlet. It takes 262 m of duct length for Mach number to increase from 0.1 to 0.2, but only 1 m to increase from 0.7 to 1.

12-113

Solution The flow of argon gas in a constant cross-sectional area adiabatic duct is considered. The variation of entropy change with exit temperature is to be investigated, and the calculated results are to be plotted on a *T*-*s* diagram.

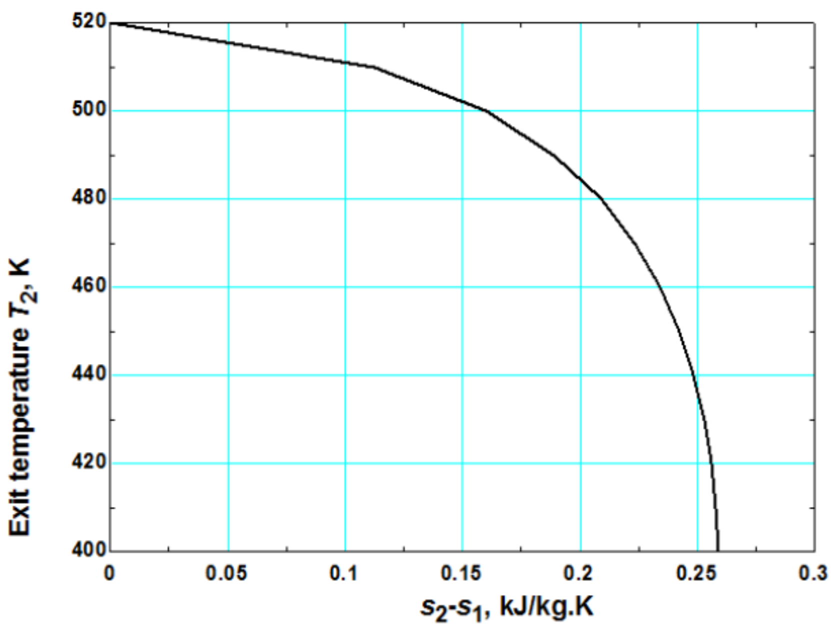
Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor remains constant along the duct.



Properties The properties of argon are given to be k = 1.667, c_p = 0.5203 kJ/kg·K, and R = 0.2081 kJ/kg·K. The average friction factor is given to be f = 0.005.

Analysis Using EES, we determine the entropy change and tabulate and plot the results as follows:

Exit temp. T_2, K	Mach umber Ma_2	Entropy change Δs, kg/kg·K
520	0.165	0.000
510	0.294	0.112
500	0.385	0.160
490	0.461	0.189
480	0.528	0.209
470	0.591	0.224
460	0.649	0.234
450	0.706	0.242
440	0.760	0.248
430	0.813	0.253
420	0.865	0.256
410	0.916	0.258
400	0.967	0.259



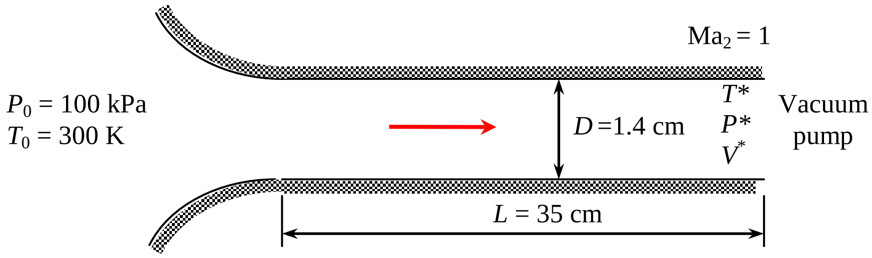
EES Program:

```
k=1.667
cp=0.5203
R=0.2081
P1=350
T1=520
V1=70
"T2=400"
f=0.005
D=0.08
C1=sqrt(k*R*T1*1000)
Ma1=V1/C1
T01=T02
T01=T1*(1+0.5*(k-1)*Ma1^2)
P01=P1*(1+0.5*(k-1)*Ma1^2)^(k/(k-1))
rho1=P1/(R*T1)
Ac=pi*D^2/4
mair=rho1*Ac*V1
P01Ps=((2+(k-1)*Ma1^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma1
P1Ps=((k+1)/(2+(k-1)*Ma1^2))^0.5/Ma1
T1Ts=(k+1)/(2+(k-1)*Ma1^2)
R1Rs=((2+(k-1)*Ma1^2)/(k+1))^0.5/Ma1
V1Vs=1/R1Rs
fLs1=(1-Ma1^2)/(k*Ma1^2)+(k+1)/(2*k)*ln((k+1)*Ma1^2/(2+(k-1)*Ma1^2))
Ls1=fLs1*D/f

P02Ps=((2+(k-1)*Ma2^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma2
P2Ps=((k+1)/(2+(k-1)*Ma2^2))^0.5/Ma2
T2Ts=(k+1)/(2+(k-1)*Ma2^2)
R2Rs=((2+(k-1)*Ma2^2)/(k+1))^0.5/Ma2
V2Vs=1/R2Rs
fLs2=(1-Ma2^2)/(k*Ma2^2)+(k+1)/(2*k)*ln((k+1)*Ma2^2/(2+(k-1)*Ma2^2))
Ls2=fLs2*D/f
L=Ls1-Ls2

P02=P02Ps/P01Ps*P01
P2=P2Ps/P1Ps*P1
T2=T2Ts/T1Ts*T1
V2=V2Vs/V1Vs*V1
Del_s=cp*ln(T2/T1)-R*ln(P2/P1)
```

Discussion Note that entropy increases with increasing duct length and Mach number (and thus decreasing temperature). It reached a maximum value of 0.259 kJ/kg.K when the Mach number reaches Ma_2 =1 and thus the flow is choked.

12-114

Solution Air enters a constant-area adiabatic duct at a specified state, and leaves at sonic state. The maximum duct length that will not cause the mass flow rate to be reduced is to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor is constant along the duct.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K. The friction factor is given to be f = 0.018.

Analysis The mass flow rate will be maximum when the flow is choked, and thus the exit Mach number is Ma_2 = 1. In that case we have

$$\frac{fL_1^*}{D}=\frac{fL_1}{D}=\frac{(0.018)(0.35\ \mathrm{m})}{0.014\ \mathrm{m}}=0.45$$

The Mach number corresponding to this value of fL^*/D at the tube inlet is obtained from Table A-16 to be Ma_1 = 0.6107 ≈ **0.611**. This value is obtained using the analytical relation. An interpolation on Table 16 gives 0.614. Noting that the flow in the nozzle section is isentropic, the thermodynamic temperature, pressure, and density at the tube inlet become

$$T_1=T_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-1}=(300\ \mathrm{K})\left(1+\frac{1.4\text{-}1}{2}(0.6107)^2\right)^{-1}=279.2\ \mathrm{K}$$

$$P_1=P_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-k/(k-1)}=(100\ \mathrm{kPa})\left(1+\frac{1.4\text{-}1}{2}(0.6107)^2\right)^{-1.4/0.4}=77.74\ \mathrm{kPa}$$

$$\rho_1=\frac{P_1}{RT_1}=\frac{77.74\ \mathrm{kPa}}{(0.287\ \mathrm{kJ/kgK})(279.2\ \mathrm{K})}=0.9702\ \mathrm{kg/m^3}$$

Then the inlet velocity and the mass flow rate become

$$c_1=\sqrt{kRT_1}=\sqrt{(1.4)(0.287\ \mathrm{kJ/kg\cdot K})(279.2\ \mathrm{K})\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)}=334.9\ \mathrm{m/s}$$

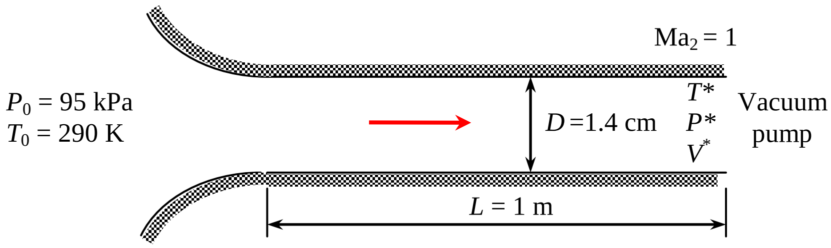
$$V_1=\mathrm{Ma}_1c_1=0.6107(334.9\ \mathrm{m/s})=204.5\ \mathrm{m/s}$$

$$\dot m_{air}=\rho_1A_{c1}V_1=(0.9702\ \mathrm{kg/m^3})[\pi(0.014\ \mathrm{m})^2/4](204.5\ \mathrm{m/s})=\mathbf{0.0305\ kg/s}$$

Discussion This is the maximum mass flow rate through the tube for the specified stagnation conditions at the inlet. The flow rate will remain at this level even if the vacuum pump drops the pressure even further.

12-115

Solution Air enters a constant-area adiabatic duct at a specified state, and leaves at sonic state. The maximum duct length that will not cause the mass flow rate to be reduced is to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor is constant along the duct.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K. The friction factor is given to be $f = 0.025$.

Analysis The mass flow rate will be maximum when the flow is choked, and thus the exit Mach number is $Ma_2 = 1$. In that case we have

$$\frac{fL_1^*}{D} = \frac{fL_1}{D} = \frac{(0.025)(1\,\text{m})}{0.014\,\text{m}} = 1.786$$

The Mach number corresponding to this value of fL^*/D at the tube inlet is obtained from Table A-16 to be $Ma_1 =$ **0.4422**. Noting that the flow in the nozzle section is isentropic, the thermodynamic temperature, pressure, and density at the tube inlet become

$$T_1 = T_{01}\left(1 + \frac{k-1}{2}\text{Ma}_1^2\right)^{-1} = (290\,\text{K})\left(1 + \frac{1.4\text{-}1}{2}(0.4422)^2\right)^{-1} = 279.1\,\text{K}$$

$$P_1 = P_{01}\left(1 + \frac{k-1}{2}\text{Ma}_1^2\right)^{-k/(k-1)} = (95\,\text{kPa})\left(1 + \frac{1.4\text{-}1}{2}(0.4422)^2\right)^{-1.4/0.4} = 83.06\,\text{kPa}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{83.06\,\text{kPa}}{(0.287\,\text{kJ/kgK})(279.1\,\text{K})} = 1.037\,\text{kg/m}^3$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\,\text{kJ/kg}\cdot\text{K})(279.1\,\text{K})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = 334.9\,\text{m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 0.4422(334.9\,\text{m/s}) = 148.1\,\text{m/s}$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (1.037\,\text{kg/m}^3)[\pi(0.014\,\text{m})^2/4](148.1\,\text{m/s}) = \mathbf{0.0236\,kg/s}$$

Discussion This is the maximum mass flow rate through the tube for the specified stagnation conditions at the inlet. The flow rate will remain at this level even if the vacuum pump drops the pressure even further.

Review Problems

12-116

Solution The thrust developed by the engine of a Boeing 777 is about 380 kN. The mass flow rate of gases through the nozzle is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow of combustion gases through the nozzle is isentropic. **3** Choked flow conditions exist at the nozzle exit. **4** The velocity of gases at the nozzle inlet is negligible.

Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K, and it can also be used for combustion gases. The specific heat ratio of combustion gases is k = 1.33.

Analysis The velocity at the nozzle exit is the sonic speed, which is determined to be

$$V = c = \sqrt{kRT} = \sqrt{(1.33)(0.287\ \text{kJ/kg}\cdot\text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)(220\ \text{K})} = 289.8\ \text{m/s}$$

Noting that thrust F is related to velocity by $F = \dot{m}V$, the mass flow rate of combustion gases is determined to be

$$\dot{m} = \frac{F}{V} = \frac{380{,}000\ \text{N}}{289.8\ \text{m/s}}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 1311\ \text{kg/s} \cong \mathbf{1310\ kg/s}$$

Discussion The combustion gases are mostly nitrogen (due to the 78% of N_2 in air), and thus they can be treated as air with a good degree of approximation.

12-117

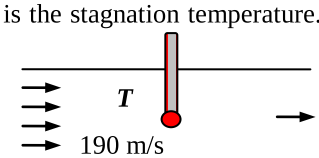
Solution A stationary temperature probe is inserted into an air duct reads 85°C. The actual temperature of air is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** The stagnation process is isentropic.

Properties The specific heat of air at room temperature is c_p = 1.005 kJ/kg·K.

Analysis The air that strikes the probe will be brought to a complete stop, and thus it will undergo a stagnation process. The thermometer will sense the temperature of this stagnated air, which is the stagnation temperature. The actual air temperature is determined from

$$T = T_0 - \frac{V^2}{2c_p} = 85°\text{C} - \frac{(190\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{67.0°C}$$



Discussion Temperature rise due to stagnation is very significant in high-speed flows, and should always be considered when compressibility effects are not negligible.

12-118

Solution Nitrogen flows through a heat exchanger. The stagnation pressure and temperature of the nitrogen at the inlet and the exit states are to be determined.

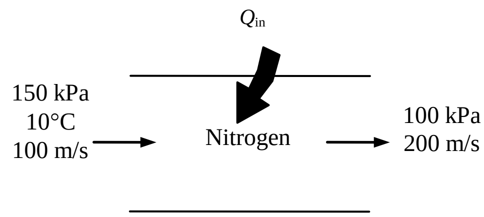
Assumptions **1** Nitrogen is an ideal gas with constant specific heats. **2** Flow of nitrogen through the heat exchanger is isentropic.

Properties The properties of nitrogen are c_p = 1.039 kJ/kg·K and k = 1.4.

Analysis The stagnation temperature and pressure of nitrogen at the inlet and the exit states are determined from

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 10°\text{C} + \frac{(100\text{ m/s})^2}{2 \times 1.039\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{14.8°C}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (150\text{ kPa})\left(\frac{288.0\text{ K}}{283.2\text{ K}}\right)^{1.4/(1.4-1)} = \mathbf{159\ kPa}$$



From the energy balance relation $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ with w = 0

$$q_{\text{in}} = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2} + \Delta pe^{\nearrow 0}$$

$$150\text{ kJ/kg} = (1.039\text{ kJ/kg}\cdot°\text{C})(T_2 - 10°\text{C}) + \frac{(200\text{ m/s})^2 - (100\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

$$T_2 = 139.9°\text{C}$$

and

$$T_{02} = T_2 + \frac{V_2^2}{2c_p} = 139.9°\text{C} + \frac{(200\text{ m/s})^2}{2 \times 1.039\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{159°C}$$

$$P_{02} = P_2\left(\frac{T_{02}}{T_2}\right)^{k/(k-1)} = (100\text{ kPa})\left(\frac{432.3\text{ K}}{413.1\text{ K}}\right)^{1.4/(1.4-1)} = \mathbf{117\ kPa}$$

Discussion Note that the stagnation temperature and pressure can be very different than their thermodynamic counterparts when dealing with compressible flow.

12-119

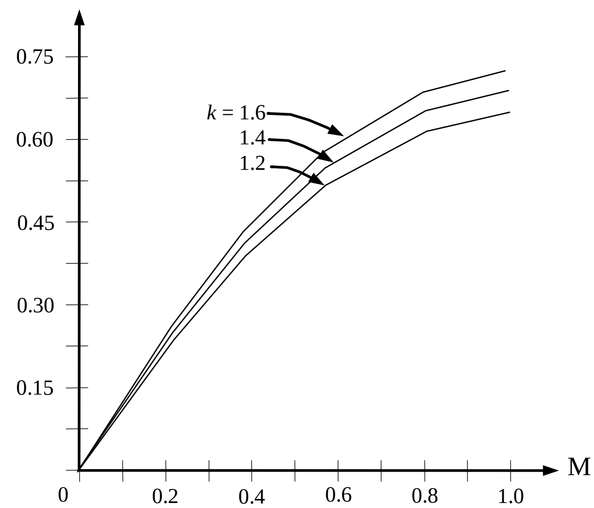
Solution The mass flow parameter $\dot{m}\sqrt{RT_0}/(AP_0)$ versus the Mach number for k = 1.2, 1.4, and 1.6 in the range of $0 \le \text{Ma} \le 1$ is to be plotted.

Analysis The mass flow rate parameter $(\dot{m}\sqrt{RT_0})/P_0A$ can be expressed as

$$\frac{\dot{m}\sqrt{RT_0}}{P_0 A} = \text{Ma}\sqrt{k}\left(\frac{2}{2+(k-1)M^2}\right)^{(k+1)/2(k-1)}$$

Thus,

Ma	***k* = 1.2**	***k* = 1.4**	***k* = 1.6**
0.0	0	0	0
0.1	0.1089	0.1176	0.1257
0.2	0.2143	0.2311	0.2465
0.3	0.3128	0.3365	0.3582
0.4	0.4015	0.4306	0.4571
0.5	0.4782	0.5111	0.5407
0.6	0.5411	0.5763	0.6077
0.7	0.5894	0.6257	0.6578
0.8	0.6230	0.6595	0.6916
0.9	0.6424	0.6787	0.7106
1.0	0.6485	0.6847	0.7164



Discussion Note that the mass flow rate increases with increasing Mach number and specific heat ratio. It levels off at Ma = 1, and remains constant (choked flow).

12-120

Solution The equivalent relation for the speed of sound is to be verified using thermodynamic relations.

Analysis The two relations are $c^2 = \left(\frac{\partial P}{\partial \rho}\right)_s$ and $c^2 = k\left(\frac{\partial P}{\partial \rho}\right)_T$

From $r = 1/v \longrightarrow dr = -dv/v^2$. Thus, $c^2 = \left(\frac{\partial P}{\partial r}\right)_s = -v^2\left(\frac{\partial P}{\partial v}\right)_s = -v^2\left(\frac{\partial P}{\partial T}\frac{\partial T}{\partial v}\right)_s = -v^2\left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial T}{\partial v}\right)_s$

From the cyclic rule,

$$(P,T,s): \left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial T}{\partial s}\right)_P\left(\frac{\partial s}{\partial P}\right)_T = -1 \longrightarrow \left(\frac{\partial P}{\partial T}\right)_s = -\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial P}{\partial s}\right)_T$$

$$(T,v,s): \left(\frac{\partial T}{\partial v}\right)_s\left(\frac{\partial v}{\partial s}\right)_T\left(\frac{\partial s}{\partial T}\right)_v = -1 \longrightarrow \left(\frac{\partial T}{\partial v}\right)_s = -\left(\frac{\partial s}{\partial v}\right)_T\left(\frac{\partial T}{\partial s}\right)_v$$

Substituting,

$$c^2 = -v^2\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial P}{\partial s}\right)_T\left(\frac{\partial s}{\partial v}\right)_T\left(\frac{\partial T}{\partial s}\right)_v = -v^2\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial T}{\partial s}\right)_v\left(\frac{\partial P}{\partial v}\right)_T$$

Recall that $\frac{c_p}{T} = \left(\frac{\partial s}{\partial T}\right)_P$ and $\frac{c_v}{T} = \left(\frac{\partial s}{\partial T}\right)_v$. Substituting,

$$c^2 = -v^2\left(\frac{c_p}{T}\right)\left(\frac{T}{c_v}\right)\left(\frac{\partial P}{\partial v}\right)_T = -v^2 k\left(\frac{\partial P}{\partial v}\right)_T$$

Replacing $-dv/v^2$ by $d\rho$, we get $\boxed{c^2 = k\left(\frac{\partial P}{\partial \rho}\right)_T}$, which is the desired expression

Discussion Note that the differential thermodynamic property relations are very useful in the derivation of other property relations in differential form.

12-121

Solution For ideal gases undergoing isentropic flows, expressions for P/P^*, T/T^*, and ρ/ρ^* as functions of k and Ma are to be obtained.

Analysis Equations 12-18 and 12-21 are given to be $\frac{T_0}{T} = \frac{2+(k-1)\text{Ma}^2}{2}$ and $\frac{T^*}{T_0} = \frac{2}{k+1}$

Multiplying the two, $\left(\frac{T_0}{T}\frac{T^*}{T_0}\right) = \left(\frac{2+(k-1)\text{Ma}^2}{2}\right)\left(\frac{2}{k+1}\right)$

Simplifying and inverting, $\boxed{\frac{T}{T^*} = \frac{k+1}{2+(k-1)\text{Ma}^2}}$ (1)

From $\frac{P}{P^*} = \left(\frac{T}{T^*}\right)^{k/(k-1)} \longrightarrow \boxed{\frac{P}{P^*} = \left(\frac{k+1}{2+(k-1)\text{Ma}^2}\right)^{k/(k-1)}}$ (2)

From $\frac{\rho}{\rho^*} = \left(\frac{\rho}{\rho^*}\right)^{k/(k-1)} \longrightarrow \boxed{\frac{\rho}{\rho^*} = \left(\frac{k+1}{2+(k-1)\text{Ma}^2}\right)^{k/(k-1)}}$ (3)

Discussion Note that some very useful relations can be obtained by very simple manipulations.

12-122

Solution It is to be verified that for the steady flow of ideal gases $dT_0/T = dA/A + (1\text{-Ma}^2)\ dV/V$. The effect of heating and area changes on the velocity of an ideal gas in steady flow for subsonic flow and supersonic flow are to be explained.

Analysis We start with the relation $$\frac{V^2}{2} = c_p(T_0 - T) \qquad (1)$$

Differentiating, $$V\,dV = c_p(dT_0 - dT) \qquad (2)$$

We also have $$\frac{d\rho}{\rho} + \frac{dA}{A} + \frac{dV}{V} = 0 \qquad (3)$$

and $$\frac{dP}{\rho} + V\,dV = 0 \qquad (4)$$

Differentiating the ideal gas relation $P = \rho RT$, $$\frac{dP}{P} = \frac{d\rho}{\rho} + \frac{dT}{T} = 0 \qquad (5)$$

From the speed of sound relation, $$c^2 = kRT = (k-1)c_pT = kP/\rho \qquad (6)$$

Combining Eqs. (3) and (5), $$\frac{dP}{P} - \frac{dT}{T} + \frac{dA}{A} + \frac{dV}{V} = 0 \qquad (7)$$

Combining Eqs. (4) and (6), $$\frac{dP}{\rho} = \frac{dP}{kP/c^2} = -VdV$$

or, $$\frac{dP}{P} = -\frac{k}{C^2}V\,dV = -k\frac{V^2}{C^2}\frac{dV}{V} = -k\text{Ma}^2\frac{dV}{V} \qquad (8)$$

Combining Eqs. (2) and (6), $$dT = dT_0 - V\frac{dV}{c_p}$$

or, $$\frac{dT}{T} = \frac{dT_0}{T} - \frac{V^2}{C_pT}\frac{dV}{V} = \frac{dT}{T} = \frac{dT_0}{T} - \frac{V^2}{C^2/(k-1)}\frac{dV}{V} = \frac{dT_0}{T} - (k-1)\text{Ma}^2\frac{dV}{V} \qquad (9)$$

Combining Eqs. (7), (8), and (9), $$-(k-1)\text{Ma}^2\frac{dV}{V} - \frac{dT_0}{T} + (k-1)\text{Ma}^2\frac{dV}{V} + \frac{dA}{A} + \frac{dV}{V} = 0$$

or, $$\frac{dT_0}{T} = \frac{dA}{A} + \left[-k\text{Ma}^2 + (k-1)\text{Ma}^2 + 1\right]\frac{dV}{V}$$

Thus, $$\boxed{\frac{dT_0}{T} = \frac{dA}{A} + (1-\text{Ma}^2)\frac{dV}{V}} \qquad (10)$$

Differentiating the steady-flow energy equation $q = h_{02} - h_{01} = c_p(T_{02} - T_{01})$

$$\delta q = c_p dT_0 \qquad (11)$$

Eq. (11) relates the stagnation temperature change dT_0 to the net heat transferred to the fluid. Eq. (10) relates the velocity changes to area changes dA, and the stagnation temperature change dT_0 or the heat transferred.

(*a*) When Ma < 1 (subsonic flow), **the fluid accelerates if the duct converges** ($dA < 0$) **or the fluid is heated** ($dT_0 > 0$ or $\delta q > 0$). **The fluid decelerates if the duct converges** (dA < 0) **or the fluid is cooled** ($dT_0 < 0$ or δq < 0).

(*b*) When Ma > 1 (supersonic flow), **the fluid accelerates if the duct diverges** ($dA > 0$) **or the fluid is cooled** ($dT_0 < 0$ or $\delta q < 0$). **The fluid decelerates if the duct converges** ($dA < 0$) **or the fluid is heated** ($dT_0 > 0$ or $\delta q > 0$).

Discussion Some of these results are not intuitively obvious, but come about by satisfying the conservation equations.

12-123

Solution A Pitot-static probe measures the difference between the static and stagnation pressures for a subsonic airplane. The speed of the airplane and the flight Mach number are to be determined.

Assumptions **1** Air is an ideal gas with a constant specific heat ratio. **2** The stagnation process is isentropic.

Properties The properties of air are R = 0.287 kJ/kg·K and k = 1.4.

Analysis The stagnation pressure of air at the specified conditions is

$$P_0 = P + \Delta P = 54 + 16 = 70\ \text{kPa}$$

Then,

$$\frac{P_0}{P} = \left(1 + \frac{(k-1)\text{Ma}^2}{2}\right)^{k/k-1} \longrightarrow \frac{70}{54} = \left(1 + \frac{(1.4-1)\text{Ma}^2}{2}\right)^{1.4/0.4}$$

It yields **Ma = 0.620**

The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(256\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 320.7\ \text{m/s}$$

Thus,

$$V = \text{Ma} \times c = (0.620)(320.7\ \text{m/s}) = \mathbf{199\ m/s}$$

Discussion Note that the flow velocity can be measured in a simple and accurate way by simply measuring pressure.

12-124

Solution An expression for the speed of sound based on van der Waals equation of state is to be derived. Using this relation, the speed of sound in carbon dioxide is to be determined and compared to that obtained by ideal gas behavior.

Properties The properties of CO_2 are R = 0.1889 kJ/kg·K and k = 1.279 at T = 50°C = 323.2 K.

Analysis Van der Waals equation of state can be expressed as $P = \dfrac{RT}{v-b} - \dfrac{a}{v^2}$.

Differentiating, $$\left(\frac{\partial P}{\partial v}\right)_T = \frac{RT}{(v-b)^2} + \frac{2a}{v^3}$$

Noting that $\rho = 1/v \longrightarrow d\rho = -dv/v^2$, the speed of sound relation becomes

$$c^2 = k\left(\frac{\partial P}{\partial r}\right)_T = v^2 k\left(\frac{\partial P}{\partial v}\right)_T$$

Substituting,

$$c^2 = \frac{v^2 kRT}{(v-b)^2} - \frac{2ak}{v}$$

Using the molar mass of CO_2 (M = 44 kg/kmol), the constant a and b can be expressed per unit mass as

$$a = 0.1882\ \text{kPa}\cdot\text{m}^6/\text{kg}^2 \quad \text{and} \qquad b = 9.70\times10^{-4}\ \text{m}^3/\text{kg}$$

The specific volume of CO_2 is determined to be

$$200\ \text{kPa} = \frac{(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323.2\ \text{K})}{\text{v} - 0.000970\ \text{m}^3/\text{kg}} - \frac{2\times0.1882\ \text{kPa}\cdot\text{m}^6/\text{kg}^2}{v^2} \rightarrow v = 0.300\ \text{m}^3/\text{kg}$$

Substituting,

$$\text{c} = \left(\left(\frac{(0.300\ \text{m}^3/\text{kg})^2(1.279)(0.1889\ \text{kJ/kg}\cdot\text{K})(323.2\ \text{K})}{(0.300-0.000970\ \text{m}^3/\text{kg})^2} - \frac{2(0.1882\ \text{kPa}\cdot\text{m}^6/\text{kg}^3)(1.279)}{(0.300\ \text{m}^3/\text{kg})^2}\right)\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kPa}\cdot\text{m}^3/\text{kg}}\right)\right)^{1/2}$$
$$= \mathbf{271 m/s}$$

If we treat CO_2 as an ideal gas, the speed of sound becomes

$$c = \sqrt{kRT} = \sqrt{(1.279)(0.1889\ \text{kJ/kg}\cdot\text{K})(323.2\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{279\ m/s}$$

Discussion Note that the ideal gas relation is the simplest equation of state, and it is very accurate for most gases encountered in practice. At high pressures and/or low temperatures, however, the gases deviate from ideal gas behavior, and it becomes necessary to use more complicated equations of state.

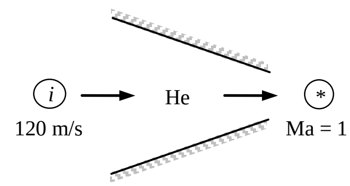
12-125

Solution Helium gas is accelerated in a nozzle. The pressure and temperature of helium at the location where Ma = 1 and the ratio of the flow area at this location to the inlet flow area are to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of helium are R = 2.0769 kJ/kg·K, c_p = 5.1926 kJ/kg·K, and k = 1.667.

Analysis The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.



$$T_0 = T_i + \frac{V_i^2}{2c_p} = 560\text{ K} + \frac{(120\text{ m/s})^2}{2\times 5.1926\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 561.4\text{ K}$$

and

$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (0.6\text{ MPa})\left(\frac{561.4\text{ K}}{560\text{ K}}\right)^{1.667/(1.667-1)} = 0.6037\text{ MPa}$$

The Mach number at the nozzle exit is given to be Ma = 1. Therefore, the properties at the nozzle exit are the *critical properties* determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (561.4\text{ K})\left(\frac{2}{1.667+1}\right) = 421.0\text{ K} = \mathbf{421K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.6037\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = 0.2941\text{ MPa} \cong \mathbf{0.294MPa}$$

The speed of sound and the Mach number at the nozzle inlet are

$$c_i = \sqrt{kRT}_i = \sqrt{(1.667)(2.0769\text{ kJ/kg}\cdot\text{K})(560\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 1392\text{ m/s}$$

$$\text{Ma}_i = \frac{V_i}{c_i} = \frac{120\text{ m/s}}{1392\text{ m/s}} = 0.08618$$

The ratio of the entrance-to-throat area is

$$\frac{A_i}{A^*} = \frac{1}{\text{Ma}_i}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\text{Ma}_i^2\right)\right]^{(k+1)/[2(k-1)]}$$

$$= \frac{1}{0.08618}\left[\left(\frac{2}{1.667+1}\right)\left(1+\frac{1.667-1}{2}(0.08618)^2\right)\right]^{2.667/(2\times 0.667)}$$

$$= 8.745$$

Then the ratio of the throat area to the entrance area becomes

$$\frac{A^*}{A_i} = \frac{1}{8.745} = 0.1144 \cong \mathbf{0.114}$$

Discussion The compressible flow functions are essential tools when determining the proper shape of the compressible flow duct.

12-126

Solution Helium gas enters a nozzle with negligible velocity, and is accelerated in a nozzle. The pressure and temperature of helium at the location where Ma = 1 and the ratio of the flow area at this location to the inlet flow area are to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The entrance velocity is negligible.

Properties The properties of helium are R = 2.0769 kJ/kg·K, c_p = 5.1926 kJ/kg·K, and k = 1.667.

Analysis We treat helium as an ideal gas with k = 1.667. The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *.

The stagnation temperature and pressure in this case are identical to the inlet temperature and pressure since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.

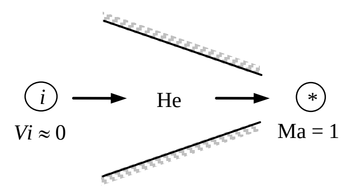
$$T_0 = T_i = 560 \text{ K}$$

$$P_0 = P_i = 0.6 \text{ MPa}$$

The Mach number at the nozzle exit is given to be Ma = 1. Therefore, the properties at the nozzle exit are the *critical properties* determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (560\text{ K})\left(\frac{2}{1.667+1}\right) = \mathbf{420 K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.6\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{0.292 MPa}$$



The ratio of the nozzle inlet area to the throat area is determined from

$$\frac{A_i}{A^*} = \frac{1}{\text{Ma}_i}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\text{Ma}_i^2\right)\right]^{(k+1)/[2(k-1)]}$$

But the Mach number at the nozzle inlet is Ma = 0 since $V_i \cong 0$. Thus the ratio of the throat area to the nozzle inlet area is

$$\frac{A^*}{A_i} = \frac{1}{\infty} = \mathbf{0}$$

Discussion The compressible flow functions are essential tools when determining the proper shape of the compressible flow duct.

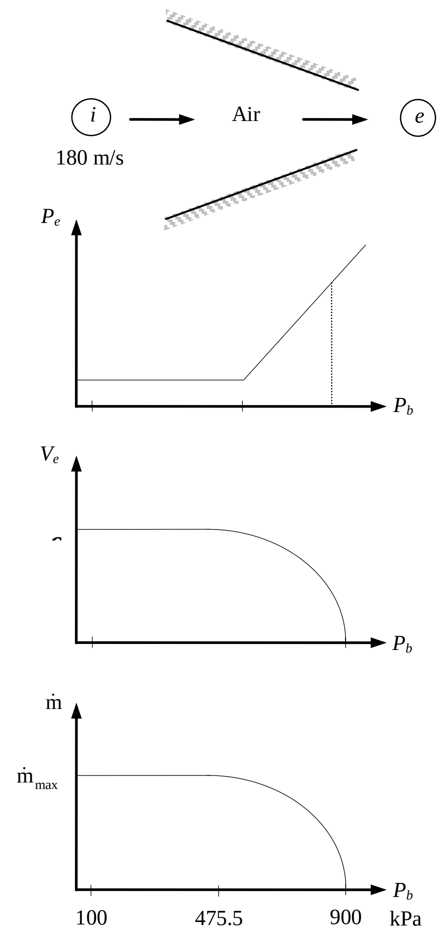
12-127

Solution Air enters a converging nozzle. The mass flow rate, the exit velocity, the exit Mach number, and the exit pressure-stagnation pressure ratio versus the back pressure-stagnation pressure ratio for a specified back pressure range are to be calculated and plotted.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Properties The properties of air at room temperature are R = 0.287 kJ/kg·K, c_p = 1.005 kJ/kg·K, and k = 1.4.

Analysis We use EES to tabulate and plot the results. The stagnation properties remain constant throughout the nozzle since the flow is isentropic. They are determined from



$$T_0 = T_i + \frac{V_i^2}{2c_p} = 400\text{ K} + \frac{(180\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 416.1\text{ K}$$

$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (900\text{ kPa})\left(\frac{416.1\text{ K}}{400\text{ K}}\right)^{1.4/(1.4-1)} = 1033.3\text{ kPa}$$

The critical pressure is determined to be

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (1033.3\text{ kPa})\left(\frac{2}{1.4+1}\right)^{1.4/0.4} = 545.9\text{ kPa}$$

Then the pressure at the exit plane (throat) is

$P_e = P_b$ for $P_b \geq 545.9$ kPa

$P_e = P^* = 545.9$ kPa for $P_b < 545.9$ kPa (choked flow)

Thus the back pressure does not affect the flow when $100 < P_b < 545.9$ kPa. For a specified exit pressure P_e, the temperature, velocity, and mass flow rate are

Temperature $\quad T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (416.1\text{ K})\left(\frac{\text{P}_\text{e}}{1033.3}\right)^{0.4/1.4}$

Velocity $\quad V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.005\text{ kJ/kg}\cdot\text{K})(416.1 - T_e)\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)}$

Speed of sound $\quad c_e = \sqrt{kRT}_e = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)}$

Mach number $\quad \text{Ma}_e = V_e / c_e$

Density $\quad \rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_e}$

Mass flow rate $\quad \dot{m} = \rho_e V_e A_e = \rho_e V_e (0.001\text{ m}^2)$

P_b, kPa	P_b, P_0	P_e, kPa	P_b, P_0	T_e, K	V_e, m/s	Ma	ρ_e, kg/m^3	$\dot{m}$, kg/s
900	0.871	900	0.871	400.0	180.0	0.45	7.840	0
800	0.774	800	0.774	386.8	162.9	0.41	7.206	1.174
700	0.677	700	0.677	372.3	236.0	0.61	6.551	1.546
600	0.581	600	0.581	356.2	296.7	0.78	5.869	1.741
545.9	0.528	545.9	0.528	333.3	366.2	1.00	4.971	1.820
500	0.484	545.9	0.528	333.2	366.2	1.00	4.971	1.820
400	0.387	545.9	0.528	333.3	366.2	1.00	4.971	1.820
300	0.290	545.9	0.528	333.3	366.2	1.00	4.971	1.820
200	0.194	545.9	0.528	333.3	366.2	1.00	4.971	1.820
100	0.097	545.9	0.528	333.3	366.2	1.00	4.971	1.820

Discussion Once the back pressure drops below 545.0 kPa, the flow is choked, and $\dot{m}$ remains constant from then on.

12-128

Solution Nitrogen gas enters a converging nozzle. The properties at the nozzle exit are to be determined.

Assumptions **1** Nitrogen is an ideal gas with k = 1.4. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Analysis The schematic of the duct is shown in Fig. 12–25. For isentropic flow through a duct, the area ratio A/A^* (the flow area over the area of the throat where Ma = 1) is also listed in Table A–13. At the initial Mach number of Ma = 0.3, we read

$$\frac{A_1}{A^*} = 2.0351, \qquad \frac{T_1}{T_0} = 0.9823, \quad \text{and} \qquad \frac{P_1}{P_0} = 0.9395$$

With a 20 percent reduction in flow area, $A_2 = 0.8A_1$, and

$$\frac{A_2}{A^*} = \frac{A_2}{A_1}\frac{A_1}{A^*} = (0.8)(2.0351) = 1.6281$$

For this value of A_2/A^* from Table A–13, we read

$$\frac{T_2}{T_0} = 0.9791, \qquad \frac{P_2}{P_0} = 0.8993, \quad \text{and} \quad \text{Ma}_2 = \mathbf{0.391}$$

Here we chose the subsonic Mach number for the calculated A_2/A^* instead of the supersonic one because the duct is converging in the flow direction and the initial flow is subsonic. Since the stagnation properties are constant for isentropic flow, we can write

$$\frac{T_2}{T_1} = \frac{T_2/T_0}{T_1/T_0} \quad \rightarrow \quad T_2 = T_1\left(\frac{T_2/T_0}{T_1/T_0}\right) = (400\text{ K})\left(\frac{0.9791}{0.9823}\right) = \mathbf{399K}$$

$$\frac{P_2}{P_1} = \frac{P_2/P_0}{P_1/P_0} \quad \rightarrow \quad P_2 = P_1\left(\frac{P_2/P_0}{P_1/P_0}\right) = (100\text{ kPa})\left(\frac{0.8993}{0.9395}\right) = \mathbf{95.7K}$$

which are the temperature and pressure at the desired location.

Discussion Note that the temperature and pressure drop as the fluid accelerates in a converging nozzle.

12-129

Solution Nitrogen gas enters a converging nozzle. The properties at the nozzle exit are to be determined.

Assumptions **1** Nitrogen is an ideal gas with k = 1.4. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

Analysis The schematic of the duct is shown in Fig. 12–25. For isentropic flow through a duct, the area ratio A/A^* (the flow area over the area of the throat where Ma = 1) is also listed in Table A–13. At the initial Mach number of Ma = 0.5, we read

$$\frac{A_1}{A^*}=1.3398, \qquad \frac{T_1}{T_0}=0.9524, \quad \text{and} \qquad \frac{P_1}{P_0}=0.8430$$

With a 20 percent reduction in flow area, A_2 = 0.8A_1, and

$$\frac{A_2}{A^*}=\frac{A_2}{A_1}\frac{A_1}{A^*}=(0.8)(1.3398)=1.0718$$

For this value of A_2/A^* from Table A–13, we read

$$\frac{T_2}{T_0}=0.9010, \qquad \frac{P_2}{P_0}=0.6948, \quad \text{and} \quad \mathrm{Ma}_2=\mathbf{0.740}$$

Here we chose the subsonic Mach number for the calculated A_2/A^* instead of the supersonic one because the duct is converging in the flow direction and the initial flow is subsonic. Since the stagnation properties are constant for isentropic flow, we can write

$$\frac{T_2}{T_1}=\frac{T_2/T_0}{T_1/T_0} \quad \rightarrow \quad T_2=T_1\left(\frac{T_2/T_0}{T_1/T_0}\right)=(400\text{ K})\left(\frac{0.9010}{0.9524}\right)=\mathbf{378K}$$

$$\frac{P_2}{P_1}=\frac{P_2/P_0}{P_1/P_0} \quad \rightarrow \quad P_2=P_1\left(\frac{P_2/P_0}{P_1/P_0}\right)=(100\text{ kPa})\left(\frac{0.6948}{0.8430}\right)=\mathbf{82.4K}$$

which are the temperature and pressure at the desired location.

Discussion Note that the temperature and pressure drop as the fluid accelerates in a converging nozzle.

12-130

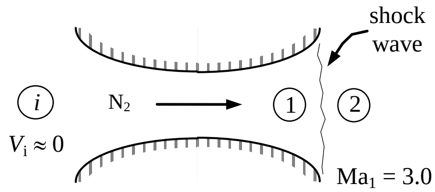
Solution Nitrogen entering a converging-diverging nozzle experiences a normal shock. The pressure, temperature, velocity, Mach number, and stagnation pressure downstream of the shock are to be determined. The results are to be compared to those of air under the same conditions.

Assumptions **1** Nitrogen is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

Properties The properties of nitrogen are R = 0.297 kJ/kg·K and k = 1.4.

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Assuming the flow before the shock to be isentropic,

$$P_{01} = P_i = 620\ \text{kPa}$$
$$T_{01} = T_i = 310\ \text{K}$$



Then,

$$T_1 = T_{01}\left(\frac{2}{2+(k-1)\text{Ma}_1^2}\right) = (310\ \text{K})\left(\frac{2}{2+(1.4-1)3^2}\right) = 110.7\ \text{K}$$

and

$$P_1 = P_{01}\left(\frac{T_1}{T_{01}}\right)^{k/(k-1)} = (620\ \text{kPa})\left(\frac{110.7}{310}\right)^{1.4/0.4} = 16.88\ \text{kPa}$$

The fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-14. For Ma_1 = 3.0 we read

$$\text{Ma}_2 = 0.4752 \cong \mathbf{0.475}, \quad \frac{P_{02}}{P_{01}} = 0.32834, \quad \frac{P_2}{P_1} = 10.333, \quad \text{and} \quad \frac{T_2}{T_1} = 2.679$$

Then the stagnation pressure P_{02}, static pressure P_2, and static temperature T_2, are determined to be

$$P_{02} = 0.32834 P_{01} = (0.32834)(620\ \text{kPa}) = 203.6\ \text{kPa} \cong \mathbf{204\ kPa}$$
$$P_2 = 10.333 P_1 = (10.333)(16.88\ \text{kPa}) = 174.4\ \text{kPa} \cong \mathbf{174\ kPa}$$
$$T_2 = 2.679 T_1 = (2.679)(110.7\ \text{K}) = 296.6\ \text{K} \cong \mathbf{297\ K}$$

The velocity after the shock can be determined from $V_2 = \text{Ma}_2 c_2$, where c_2 is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.4752)\sqrt{(1.4)(0.297\ \text{kJ/kg}\cdot\text{K})(296.6\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 166.9\ \text{m/s} \cong \mathbf{167\ m/s}$$

Discussion For *air* at specified conditions k = 1.4 (same as nitrogen) and R = 0.287 kJ/kg·K. Thus the only quantity which will be different in the case of air is the velocity after the normal shock, which happens to be 164.0 m/s.

12-131

Solution The diffuser of an aircraft is considered. The static pressure rise across the diffuser and the exit area are to be determined.

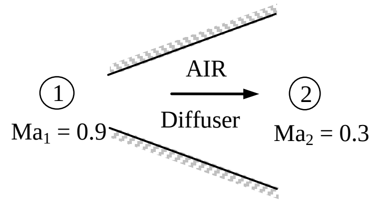
Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the diffuser is steady, one-dimensional, and isentropic. **3** The diffuser is adiabatic.

Properties Air properties at room temperature are R = 0.287 kJ/kg·K, c_p = 1.005 kJ/kg·K, and k = 1.4.

Analysis The inlet velocity is

$$V_1 = \text{Ma}_1 c_1 = M_1\sqrt{kRT_1} = (0.9)\sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(242.7\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 281.0\text{ m/s}$$

Then the stagnation temperature and pressure at the diffuser inlet become



$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 242.7 + \frac{(281.0\text{ m/s})^2}{2(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 282.0\text{ K}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (41.1\text{ kPa})\left(\frac{282.0\text{ K}}{242.7\text{ K}}\right)^{1.4/(1.4-1)} = 69.50\text{ kPa}$$

For an adiabatic diffuser, the energy equation reduces to $h_{01} = h_{02}$. Noting that $h = c_pT$ and the specific heats are assumed to be constant, we have

$$T_{01} = T_{02} = T_0 = 282.0\text{ K}$$

The isentropic relation between states 1 and 02 gives

$$P_{02} = P_{02} = P_1\left(\frac{T_{02}}{T_1}\right)^{k/(k-1)} = (41.1\text{ kPa})\left(\frac{282.0\text{ K}}{242.7\text{ K}}\right)^{1.4/(1.4-1)} = 69.50\text{ kPa}$$

The exit velocity can be expressed as

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.3)\sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})\,T_2\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 6.01\sqrt{T_2}$$

Thus $$T_2 = T_{02} - \frac{V_2^2}{2c_p} = (282.0) - \frac{6.01^2\,\text{T}_2\text{ m}^2/\text{s}^2}{2(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 277.0\text{ K}$$

Then the static exit pressure becomes

$$P_2 = P_{02}\left(\frac{T_2}{T_{02}}\right)^{k/(k-1)} = (69.50\text{ kPa})\left(\frac{277.0\text{ K}}{282.0\text{ K}}\right)^{1.4/(1.4-1)} = 65.28\text{ kPa}$$

Thus the static pressure rise across the diffuser is

$$\Delta P = P_2 - P_1 = 65.28 - 41.1 = \mathbf{24.2kPa}$$

Also, $$\rho_2 = \frac{P_2}{RT_2} = \frac{65.28\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(277.0\text{ K})} = 0.8211\text{ kg/m}^3$$

$$V_2 = 6.01\sqrt{T_2} = 6.01\sqrt{277.0} = 100.0\text{ m/s}$$

Thus $$A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{38\text{ kg/s}}{(0.8211\text{ kg/m}^3)(100.0\text{ m/s})} = \mathbf{0.463m^2}$$

Discussion The pressure rise in actual diffusers will be lower because of the irreversibilities. However, flow through well-designed diffusers is very nearly isentropic.

12-132

Solution The critical temperature, pressure, and density of an equimolar mixture of oxygen and nitrogen for specified stagnation properties are to be determined.

Assumptions Both oxygen and nitrogen are ideal gases with constant specific heats at room temperature.

Properties The specific heat ratio and molar mass are k = 1.395 and M = 32 kg/kmol for oxygen, and k = 1.4 and M = 28 kg/kmol for nitrogen.

Analysis The gas constant of the mixture is

$$M_m = y_{O_2} M_{O_2} + y_{N_2} M_{N_2} = 0.5 \times 32 + 0.5 \times 28 = 30 \text{ kg/kmol}$$

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{30 \text{ kg/kmol}} = 0.2771 \text{ kJ/kg} \cdot \text{K}$$

The specific heat ratio is 1.4 for nitrogen, and nearly 1.4 for oxygen. Therefore, the specific heat ratio of the mixture is also 1.4. Then the critical temperature, pressure, and density of the mixture become

$$T^* = T_0 \left(\frac{2}{k+1} \right) = (550 \text{ K}) \left(\frac{2}{1.4+1} \right) = 458.3 \text{ K} \cong \mathbf{458 K}$$

$$P^* = P_0 \left(\frac{2}{k+1} \right)^{k/(k-1)} = (350 \text{ kPa}) \left(\frac{2}{1.4+1} \right)^{1.4/(1.4-1)} = 184.9 \text{ kPa} \cong \mathbf{185 kPa}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{184.9 \text{ kPa}}{(0.2771 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(458.3 \text{ K})} = \mathbf{1.46 kg/m^3}$$

Discussion If the specific heat ratios k of the two gases were different, then we would need to determine the k of the mixture from $k = c_{p,m}/c_{v,m}$ where the specific heats of the mixture are determined from

$$C_{p,m} = \text{mf}_{O_2} c_{p,O_2} + \text{mf}_{N_2} c_{p,N_2} = (y_{O_2} M_{O_2} / M_m) c_{p,O_2} + (y_{N_2} M_{N_2} / M_m) c_{p,N_2}$$
$$C_{v,m} = \text{mf}_{O_2} c_{v,O_2} + \text{mf}_{N_2} c_{v,N_2} = (y_{O_2} M_{O_2} / M_m) c_{v,O_2} + (y_{N_2} M_{N_2} / M_m) c_{v,N_2}$$

where mf is the mass fraction and y is the mole fraction. In this case it would give

$$c_{p,m} = (0.5 \times 32/30) \times 0.918 + (0.5 \times 28/30) \times 1.039 = 0.974 \text{ kJ/kg.K}$$

$$c_{p,m} = (0.5 \times 32/30) \times 0.658 + (0.5 \times 28/30) \times 0.743 = 0.698 \text{ kJ/kg.K}$$

and

k = 0.974/0.698 = 1.40

12-133E

Solution Helium gas is accelerated in a nozzle. For a specified mass flow rate, the throat and exit areas of the nozzle are to be determined for the case of isentropic nozzle.

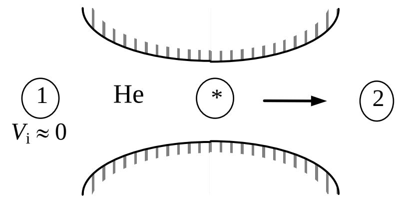
Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

Properties The properties of helium are R = 0.4961 Btu/lbm·R = 2.6809 psia·ft^3/lbm·R, c_p = 1.25 Btu/lbm·R, and k = 1.667.

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible,

$$T_{01} = T_1 = 740 \text{ R}$$

$$P_{01} = P_1 = 220 \text{ psia}$$



The flow is assumed to be isentropic, thus the stagnation temperature and pressure remain constant throughout the nozzle,

$$T_{02} = T_{01} = 740 \text{ R}$$

$$P_{02} = P_{01} = 220 \text{ psia}$$

The critical pressure and temperature are determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (740 \text{ R})\left(\frac{2}{1.667+1}\right) = 554.9 \text{ R}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (220 \text{ psia})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = 107.2 \text{ psia}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{107.2 \text{ psia}}{(2.6809 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(554.9 \text{ R})} = 0.072031 \text{bm/ft}^3$$

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.667)(0.4961 \text{ Btu/lbm}\cdot\text{R})(554.9 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/1bm}}\right)} = 3390 \text{ ft/s}$$

and $$A^* = \frac{\dot m}{\rho^* V^*} = \frac{0.2 \ \text{1bm/s}}{(0.072031 \text{bm/ft}^3)(3390 \text{ ft/s})} = \mathbf{8.19 \times 10^{-4} \ ft^2}$$

At the nozzle exit the pressure is P_2 = 15 psia. Then the other properties at the nozzle exit are determined to be

$$\frac{p_0}{p_2} = \left(1 + \frac{k-1}{2}\text{Ma}_2^2\right)^{k/(k-1)} \longrightarrow \frac{220 \text{ psia}}{15 \text{ psia}} = \left(1 + \frac{1.667-1}{2}\text{Ma}_2^2\right)^{1.667/0.667}$$

It yields Ma_2 = 2.405, which is greater than 1. Therefore, the nozzle must be converging-diverging.

$$T_2 = T_0\left(\frac{2}{2+(k-1)\text{Ma}_2^2}\right) = (740 \text{ R})\left(\frac{2}{2+(1.667-1)\times 2.405^2}\right) = 252.6 \text{ R}$$

$$\rho_2 = \frac{P_2}{RT_2} = \frac{15 \text{ psia}}{(2.6809 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(252.6 \text{ R})} = 0.02215 \ \text{1bm/ft}^3$$

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (2.405)\sqrt{(1.667)(0.4961 \text{ Btu/lbm}\cdot\text{R})(252.6 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/1bm}}\right)} = 5500 \text{ ft/s}$$

Thus the exit area is

$$A_2 = \frac{\dot m}{\rho_2 V_2} = \frac{0.2 \text{ lbm/s}}{(0.02215 \text{ lbm/ft}^3)(5500 \text{ ft/s})} = \mathbf{0.00164 ft^2}$$

Discussion Flow areas in actual nozzles would be somewhat larger to accommodate the irreversibilities.

12-134

Solution Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-13 for an ideal gas with k = 1.667.

Properties The specific heat ratio of the ideal gas is given to be k = 1.667.

Analysis The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\text{Ma}^* = \text{Ma}\sqrt{\frac{k+1}{2+(k-1)\text{Ma}^2}}$$

$$\frac{A}{A^*} = \frac{1}{\text{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\text{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\text{Ma}^2\right)^{-k/(k-1)}$$

$$\frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\text{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\text{Ma}^2\right)^{-1}$$

```
k=1.667
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M
```

Ma	Ma*	A/A^*	P/P_0	ρ/ρ_0	T/T_0
0.0	0	∞	1.0000	1.0000	1.0000
0.1	0.1153	5.6624	0.9917	0.9950	0.9967
0.2	0.2294	2.8879	0.9674	0.9803	0.9868
0.3	0.3413	1.9891	0.9288	0.9566	0.9709
0.4	0.4501	1.5602	0.8782	0.9250	0.9493
0.5	0.5547	1.3203	0.8186	0.8869	0.9230
0.6	0.6547	1.1760	0.7532	0.8437	0.8928
0.7	0.7494	1.0875	0.6850	0.7970	0.8595
0.8	0.8386	1.0351	0.6166	0.7482	0.8241
0.9	0.9222	1.0081	0.5501	0.6987	0.7873
1.0	1.0000	1.0000	0.4871	0.6495	0.7499
1.2	1.1390	1.0267	0.3752	0.5554	0.6756
1.4	1.2572	1.0983	0.2845	0.4704	0.6047
1.6	1.3570	1.2075	0.2138	0.3964	0.5394
1.8	1.4411	1.3519	0.1603	0.3334	0.4806
2.0	1.5117	1.5311	0.1202	0.2806	0.4284
2.2	1.5713	1.7459	0.0906	0.2368	0.3825
2.4	1.6216	1.9980	0.0686	0.2005	0.3424
2.6	1.6643	2.2893	0.0524	0.1705	0.3073
2.8	1.7007	2.6222	0.0403	0.1457	0.2767
3.0	1.7318	2.9990	0.0313	0.1251	0.2499
5.0	1.8895	9.7920	0.0038	0.0351	0.1071
∝	1.9996	∞	0	0	0

Discussion The tabulated values are useful for quick calculations, but be careful – they apply only to one specific value of k, in this case k = 1.667.

12-135

Solution Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-14 for an ideal gas with k = 1.667.

Properties The specific heat ratio of the ideal gas is given to be k = 1.667.

Analysis The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2/P_1}{T_2/T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

```
k=1.667
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)
```

Ma_1	Ma_2	P_2/P_1	ρ_2/ρ_1	T_2/T_1	P_{02}/P_{01}	P_{02}/P_1
1.0	1.0000	1.0000	1.0000	1.0000	1	2.0530
1.1	0.9131	1.2625	1.1496	1.0982	0.999	2.3308
1.2	0.8462	1.5500	1.2972	1.1949	0.9933	2.6473
1.3	0.7934	1.8626	1.4413	1.2923	0.9813	2.9990
1.4	0.7508	2.2001	1.5805	1.3920	0.9626	3.3838
1.5	0.7157	2.5626	1.7141	1.4950	0.938	3.8007
1.6	0.6864	2.9501	1.8415	1.6020	0.9085	4.2488
1.7	0.6618	3.3627	1.9624	1.7135	0.8752	4.7278
1.8	0.6407	3.8002	2.0766	1.8300	0.8392	5.2371
1.9	0.6227	4.2627	2.1842	1.9516	0.8016	5.7767
2.0	0.6070	4.7503	2.2853	2.0786	0.763	6.3462
2.1	0.5933	5.2628	2.3802	2.2111	0.7243	6.9457
2.2	0.5814	5.8004	2.4689	2.3493	0.6861	7.5749
2.3	0.5708	6.3629	2.5520	2.4933	0.6486	8.2339
2.4	0.5614	6.9504	2.6296	2.6432	0.6124	8.9225
2.5	0.5530	7.5630	2.7021	2.7989	0.5775	9.6407
2.6	0.5455	8.2005	2.7699	2.9606	0.5442	10.3885
2.7	0.5388	8.8631	2.8332	3.1283	0.5125	11.1659
2.8	0.5327	9.5506	2.8923	3.3021	0.4824	11.9728
2.9	0.5273	10.2632	2.9476	3.4819	0.4541	12.8091
3.0	0.5223	11.0007	2.9993	3.6678	0.4274	13.6750
4.0	0.4905	19.7514	3.3674	5.8654	0.2374	23.9530
5.0	0.4753	31.0022	3.5703	8.6834	0.1398	37.1723
∞	0.4473	∞	3.9985	∞	0	∞

Discussion The tabulated values are useful for quick calculations, but be careful – they apply only to one specific value of k, in this case k = 1.667.

12-136

Solution Helium gas is accelerated in a nozzle isentropically. For a specified mass flow rate, the throat and exit areas of the nozzle are to be determined.

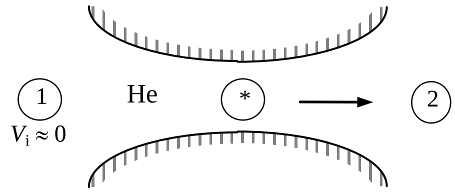
Assumptions **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

Properties The properties of helium are R = 2.0769 kJ/kg.K, c_p = 5.1926 kJ/kg.K, and k = 1.667.

Analysis The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible,

$$T_{01} = T_1 = 500 \text{ K}$$

$$P_{01} = P_1 = 1.0 \text{ MPa}$$



The flow is assumed to be isentropic, thus the stagnation temperature and pressure remain constant throughout the nozzle,

$$T_{02} = T_{01} = 500 \text{ K}$$

$$P_{02} = P_{01} = 1.0 \text{ MPa}$$

The critical pressure and temperature are determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (500 \text{ K})\left(\frac{2}{1.667+1}\right) = 375.0 \text{ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (1.0 \text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = 0.487 \text{ MPa}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{487 \text{ kPa}}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(375 \text{ K})} = 0.625 \text{ kg/m}^3$$

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.667)(2.0769 \text{ kJ/kg}\cdot\text{K})(375 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 1139.4 \text{ m/s}$$

Thus the throat area is

$$A^* = \frac{\dot{m}}{\rho^* V^*} = \frac{0.46 \text{ kg/s}}{(0.625 \text{ kg/m}^3)(1139.4 \text{ m/s})} = 6.460\times10^{-4} \text{ m}^2 = \mathbf{6.46\,cm^2}$$

At the nozzle exit the pressure is P_2 = 0.1 MPa. Then the other properties at the nozzle exit are determined to be

$$\frac{P_0}{P_2} = \left(1+\frac{k-1}{2}\text{Ma}_2^2\right)^{k/(k-1)} \longrightarrow \frac{1.0 \text{ MPa}}{0.1 \text{MPa}} = \left(1+\frac{1.667-1}{2}\text{Ma}_2^2\right)^{1.667/0.667}$$

It yields Ma_2 = 2.130, which is greater than 1. Therefore, the nozzle must be converging-diverging.

$$T_2 = T_0\left(\frac{2}{2+(k-1)\text{Ma}_2^2}\right) = (500 \text{ K})\left(\frac{2}{2+(1.667-1)\times 2.13^2}\right) = 199.0 \text{ K}$$

$$\rho_2 = \frac{P_2}{RT_2} = \frac{100 \text{ kPa}}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(199 \text{ K})} = 0.242 \text{ kg/m}^3$$

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (2.13)\sqrt{(1.667)(2.0769 \text{ kJ/kg}\cdot\text{K})(199 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 1768.0 \text{ m/s}$$

Thus the exit area is

$$A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.46 \text{ kg/s}}{(0.242 \text{ kg/m}^3)(1768 \text{ m/s})} = 0.1075\times10^{-3} \text{ m}^2 \cong \mathbf{10.8\,cm^2}$$

Discussion Flow areas in actual nozzles would be somewhat larger to accommodate the irreversibilities.

12-137

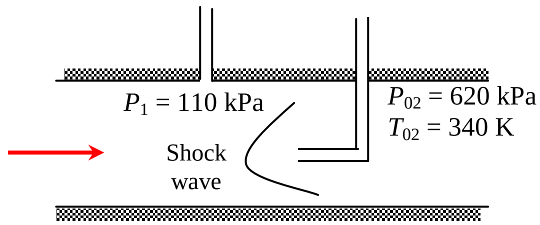
Solution The flow velocity of air in a channel is to be measured using a Pitot-static probe, which causes a shock wave to occur. For measured values of static pressure before the shock and stagnation pressure and temperature after the shock, the flow velocity before the shock is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady and one-dimensional.

Properties The specific heat ratio of air at room temperature is k= 1.4.

Analysis The nose of the probe is rounded (instead of being pointed), and thus it will cause a bow shock wave to form. Bow shocks are difficult to analyze. But they are normal to the body at the nose, and thus we can approximate them as normal shocks in the vicinity of the probe. It is given that the static pressure before the shock is P_1 = 110 kPa, and the stagnation pressure and temperature after the shock are P_{02} = 620 kPa, and T_{02} = 340 K. Noting that the stagnation temperature remains constant, we have

$$T_{01} = T_{02} = 340\text{ K}$$



Also, $$\frac{P_{02}}{P_1} = \frac{620\text{ kPa}}{110\text{ kPa}} = 5.6364 \approx 5.64$$

The fluid properties after the shock are related to those before the shock through the functions listed in Table A-14.
For $P_{02}/P_1 = 5.64$ we read

$$\text{Ma}_1 = 2.0, \quad \text{Ma}_2 = 0.5774, \quad \frac{P_{02}}{P_{01}} = 0.7209, \quad \frac{V_1}{V_2} = \frac{\rho_2}{\rho_1} = 2.6667,$$

Then the stagnation pressure and temperature before the shock become

$P_{01} = P_{02}/0.7209$ = (620 kPa)/0.7209 = 860 kPa

$$T_1 = T_{01}\left(\frac{P_1}{P_{01}}\right)^{(k-1)/k} = (340\text{ K})\left(\frac{110\text{ kPa}}{860\text{ kPa}}\right)^{(1.4-1)/1.4} = 188.9\text{ K}$$

The flow velocity before the shock can be determined from $V_1 = \text{Ma}_1 c_1$, where c_1 is the speed of sound before the shock,

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(188.9\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 275.5\text{ m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 2(275.5\text{ m/s}) = \mathbf{551\text{ m/s}}$$

Discussion The flow velocity after the shock is $V_2 = V_1/2.6667$ = 551/2.6667 = 207 m/s. Therefore, the velocity measured by a Pitot-static probe would be very different that the flow velocity.

12-138

Solution Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-14 for air.

Properties The specific heat ratio is given to be k = 1.4 for air.

Analysis The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2/P_1}{T_2/T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

Air:

```
k=1.4
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)
```

Ma_1	Ma_2	P_2/P_1	ρ_2/ρ_1	T_2/T_1	P_{02}/P_{01}	P_{02}/P_1
1.0	1.0000	1.0000	1.0000	1.0000	1	1.8929
1.5	0.7011	2.4583	1.8621	1.3202	0.9298	3.4133
2.0	0.5774	4.5000	2.6667	1.6875	0.7209	5.6404
2.5	0.5130	7.1250	3.3333	2.1375	0.499	8.5261
3.0	0.4752	10.3333	3.8571	2.6790	0.3283	12.0610
3.5	0.4512	14.1250	4.2609	3.3151	0.2129	16.2420
4.0	0.4350	18.5000	4.5714	4.0469	0.1388	21.0681
4.5	0.4236	23.4583	4.8119	4.8751	0.0917	26.5387
5.0	0.4152	29.0000	5.0000	5.8000	0.06172	32.6535
5.5	0.4090	35.1250	5.1489	6.8218	0.04236	39.4124
6.0	0.4042	41.8333	5.2683	7.9406	0.02965	46.8152
6.5	0.4004	49.1250	5.3651	9.1564	0.02115	54.8620
7.0	0.3974	57.0000	5.4444	10.4694	0.01535	63.5526
7.5	0.3949	65.4583	5.5102	11.8795	0.01133	72.8871
8.0	0.3929	74.5000	5.5652	13.3867	0.008488	82.8655
8.5	0.3912	84.1250	5.6117	14.9911	0.006449	93.4876
9.0	0.3898	94.3333	5.6512	16.6927	0.004964	104.7536
9.5	0.3886	105.1250	5.6850	18.4915	0.003866	116.6634
10.0	0.3876	116.5000	5.7143	20.3875	0.003045	129.2170

Discussion The tabulated values are useful for quick calculations, but be careful – they apply only to one specific value of k, in this case k = 1.4.

12-139

Solution Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-14 for methane.

Properties The specific heat ratio is given to be k = 1.3 for methane.

Analysis The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2/P_1}{T_2/T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

Methane:

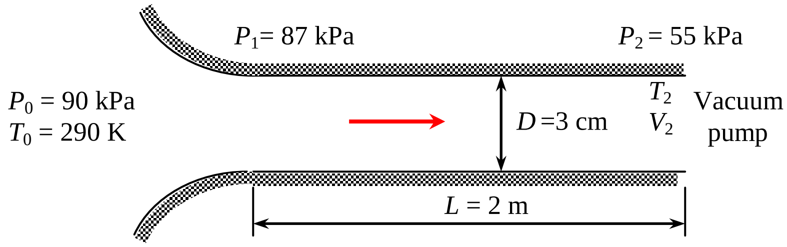
```
k=1.3
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)
```

Ma_1	Ma_2	P_2/P_1	ρ_2/ρ_1	T_2/T_1	P_{02}/P_{01}	P_{02}/P_1
1.0	1.0000	1.0000	1.0000	1.0000	1	1.8324
1.5	0.6942	2.4130	1.9346	1.2473	0.9261	3.2654
2.0	0.5629	4.3913	2.8750	1.5274	0.7006	5.3700
2.5	0.4929	6.9348	3.7097	1.8694	0.461	8.0983
3.0	0.4511	10.0435	4.4043	2.2804	0.2822	11.4409
3.5	0.4241	13.7174	4.9648	2.7630	0.1677	15.3948
4.0	0.4058	17.9565	5.4118	3.3181	0.09933	19.9589
4.5	0.3927	22.7609	5.7678	3.9462	0.05939	25.1325
5.0	0.3832	28.1304	6.0526	4.6476	0.03613	30.9155
5.5	0.3760	34.0652	6.2822	5.4225	0.02243	37.3076
6.0	0.3704	40.5652	6.4688	6.2710	0.01422	44.3087
6.5	0.3660	47.6304	6.6218	7.1930	0.009218	51.9188
7.0	0.3625	55.2609	6.7485	8.1886	0.006098	60.1379
7.5	0.3596	63.4565	6.8543	9.2579	0.004114	68.9658
8.0	0.3573	72.2174	6.9434	10.4009	0.002827	78.4027
8.5	0.3553	81.5435	7.0190	11.6175	0.001977	88.4485
9.0	0.3536	91.4348	7.0837	12.9079	0.001404	99.1032
9.5	0.3522	101.8913	7.1393	14.2719	0.001012	110.367
10.0	0.3510	112.9130	7.1875	15.7096	0.000740	122.239

Discussion The tabulated values are useful for quick calculations, but be careful – they apply only to one specific value of k, in this case k = 1.3.

12-140

Solution Air enters a constant-area adiabatic duct at a specified state, and leaves at a specified pressure. The mass flow rate of air, the exit velocity, and the average friction factor are to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor is constant along the duct.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K. The friction factor is given to be f = 0.025.

Analysis Noting that the flow in the nozzle section is isentropic, the Mach number, thermodynamic temperature, and density at the tube inlet become

$$P_1 = P_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-k/(k-1)} \quad\rightarrow\quad 87\ \mathrm{kPa} = (90\ \mathrm{kPa})\left(1+\frac{1.4\text{-}1}{2}\mathrm{Ma}_1^2\right)^{-1.4/0.4} \quad\rightarrow\quad \mathrm{Ma}_1 = 0.2206$$

$$T_1 = T_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-1} = (290\ \mathrm{K})\left(1+\frac{1.4\text{-}1}{2}(0.2206)^2\right)^{-1} = 287.2\ \mathrm{K}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{87\ \mathrm{kPa}}{(0.287\ \mathrm{kJ/kgK})(287.2\ \mathrm{K})} = 1.055\ \mathrm{kg/m^3}$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\ \mathrm{kJ/kg\cdot K})(287.2\ \mathrm{K})\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = 339.7\ \mathrm{m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 0.2206(339.7\ \mathrm{m/s}) = 74.94\ \mathrm{m/s}$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (1.055\ \mathrm{kg/m^3})[\pi(0.03\ \mathrm{m})^2/4](74.94\ \mathrm{m/s}) = \mathbf{0.0559\ kg/s}$$

The Fanno flow functions corresponding to the inlet Mach number are, from Table A-16 (we used analytical relations),

$\mathrm{Ma}_1 = 0.2206$: $(fL^*/D_h)_1$ =11.520 T_1/T^* = 1.1884, P_1/P^* = 4.9417, V_1/V^* = 0.2405

Therefore, $P_1 = 4.9417P^*$. Then the Fanno function P_2/P^* becomes

$$\frac{P_2}{P^*} = \frac{P_2}{P_1/5.2173} = \frac{4.9417(55\ \mathrm{kPa})}{87\ \mathrm{kPa}} = 3.124$$

The corresponding Mach number and Fanno flow functions are, from Table A-16,

$\mathrm{Ma}_2 = 0.3465$, $(fL^*/D_h)_2 = 3.5536$, and V_2/V^* =0.3751.

Then the air velocity at the duct exit and the average friction factor become

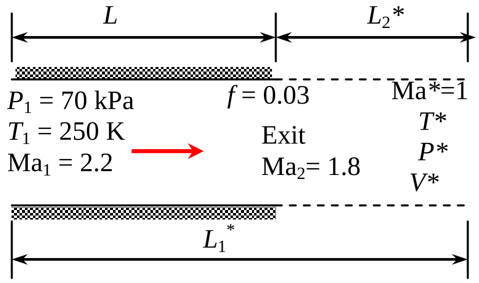
$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{0.3751}{0.2405} = 1.5597 \qquad\rightarrow\quad V_2 = 1.5597V_1 = 1.5597(74.94\ \mathrm{m/s}) = \mathbf{117\ m/s}$$

$$L = L_1^* - L_2^* = \left(\frac{fL_1^*}{D_h} - \frac{fL_2^*}{D_h}\right)\frac{D_h}{f} \quad\rightarrow\quad 2\ \mathrm{m} = (11.520 - 3.5536)\frac{0.03\ \mathrm{m}}{f} \quad\rightarrow\quad f = \mathbf{0.120}$$

Discussion Note that the mass flow rate and the average friction factor can be determined by measuring static pressure, as in incompressible flow.

12-141

Solution Supersonic airflow in a constant cross-sectional area adiabatic duct is considered. For a specified exit Mach number, the temperature, pressure, and velocity at the duct exit are to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor remains constant along the duct.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K. The average friction factor is given to be $f = 0.03$.

Analysis The inlet velocity is

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(250 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 316.9 \text{ m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 2.2(316.9 \text{ m/s}) = 697.3 \text{ m/s}$$

The Fanno flow functions corresponding to the inlet and exit Mach numbers are, from Table A-16,

$\text{Ma}_1 = 2.2$: $(fL^*/D_h)_1 = 0.3609$ $T_1/T^* = 0.6098$, $P_1/P^* = 0.3549$, $V_1/V^* = 1.7179$

$\text{Ma}_2 = 1.8$: $(fL^*/D_h)_2 = 0.2419$ $T_2/T^* = 0.7282$, $P_2/P^* = 0.4741$, $V_2/V^* = 1.5360$

Then the temperature, pressure, and velocity at the duct exit are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.7282}{0.6098} = 1.1942 \quad \rightarrow \quad T_2 = 1.1942T_1 = 1.1942(250 \text{ K}) = \mathbf{299 K}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{0.4741}{0.3549} = 1.3359 \quad \rightarrow \quad P_2 = 1.3359P_1 = 1.3359(70 \text{ kPa}) = \mathbf{93.5 kPa}$$

$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{1.5360}{1.7179} = 0.8941 \quad \rightarrow \quad V_2 = 0.8941V_1 = 0.8941(697.3 \text{ m/s}) = \mathbf{623 m/s}$$

Discussion The duct length is determined to be

$$L = L_1^* - L_2^* = \left(\frac{fL_1^*}{D_h} - \frac{fL_2^*}{D_h}\right)\frac{D_h}{f} = (0.3609 - 0.2419)\frac{0.055 \text{ m}}{0.03} = \mathbf{0.218 m}$$

Note that it takes a duct length of only 0.218 m for the Mach number to decrease from 2.2 to 1.8. The maximum (or sonic) duct lengths at the inlet and exit states in this case are $L_1^* = 0.662$ m and $L_2^* = 0.443$ m. Therefore, the flow would reach sonic conditions if a 0.443-m long section were added to the existing duct.

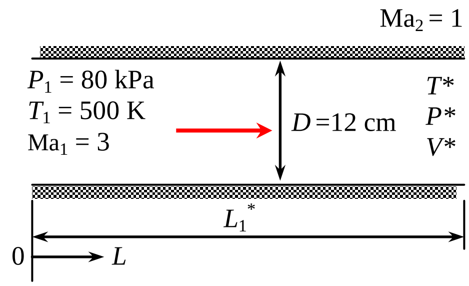
12-142

Solution Choked supersonic airflow in a constant cross-sectional area adiabatic duct is considered. The variation of duct length with Mach number is to be investigated, and the results are to be plotted.

Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor remains constant along the duct.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K. The average friction factor is given to be f = 0.03.

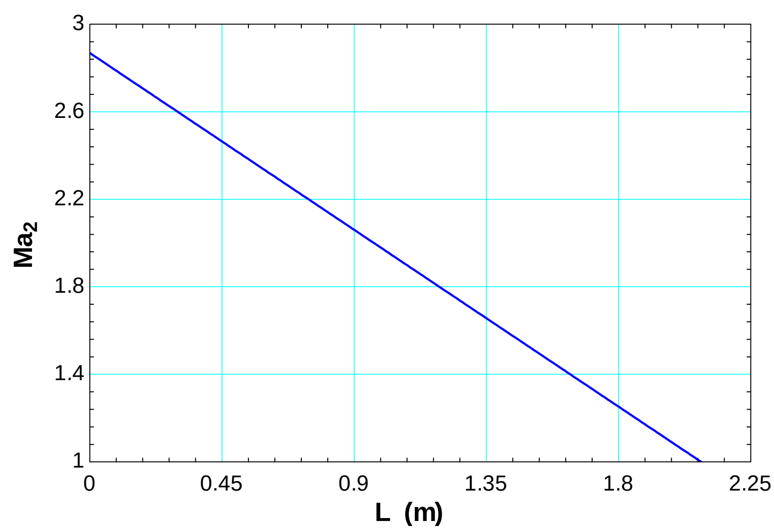
Analysis We use EES to solve the problem. The flow is choked, and thus Ma_2 =1. Corresponding to the inlet Mach number of Ma_1 = 3 we have, from Table A-16, fL^*/D_h = 0.5222, Therefore, the original duct length is



$$L_1^* = 0.5222\frac{D}{f} = 0.5222\frac{0.18\,\mathrm{m}}{0.03} = 3.13\,\mathrm{m}$$

Repeating the calculations for different Ma_2 as it varies from 3 to 1 results in the following table for the location on the duct from the inlet:

Mach number, Ma	Duct length L, m
1.00	2.09
1.25	1.89
1.50	1.54
1.75	1.19
2.00	0.87
2.25	0.59
2.50	0.36
2.75	0.16
3.00	0.00



EES program:

```
k=1.4
cp=1.005
R=0.287

P1=80
T1=500
Ma1=3
"Ma2=1"
f=0.03
D=0.12

C1=sqrt(k*R*T1*1000)
Ma1=V1/C1
T01=T02
T01=T1*(1+0.5*(k-1)*Ma1^2)
```

```
T02=T2*(1+0.5*(k-1)*Ma2^2)
P01=P1*(1+0.5*(k-1)*Ma1^2)^(k/(k-1))

rho1=P1/(R*T1)
Ac=pi*D^2/4
mair=rho1*Ac*V1

P01Ps=((2+(k-1)*Ma1^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma1
P1Ps=((k+1)/(2+(k-1)*Ma1^2))^0.5/Ma1
T1Ts=(k+1)/(2+(k-1)*Ma1^2)
R1Rs=((2+(k-1)*Ma1^2)/(k+1))^0.5/Ma1
V1Vs=1/R1Rs
fLs1=(1-Ma1^2)/(k*Ma1^2)+(k+1)/(2*k)*ln((k+1)*Ma1^2/(2+(k-1)*Ma1^2))
Ls1=fLs1*D/f

P02Ps=((2+(k-1)*Ma2^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma2
P2Ps=((k+1)/(2+(k-1)*Ma2^2))^0.5/Ma2
T2Ts=(k+1)/(2+(k-1)*Ma2^2)
R2Rs=((2+(k-1)*Ma2^2)/(k+1))^0.5/Ma2
V2Vs=1/R2Rs
fLs2=(1-Ma2^2)/(k*Ma2^2)+(k+1)/(2*k)*ln((k+1)*Ma2^2/(2+(k-1)*Ma2^2))
Ls2=fLs2*D/f
L=Ls1-Ls2

P02=P02Ps/P01Ps*P01
P2=P2Ps/P1Ps*P1
V2=V2Vs/V1Vs*V1
```

Discussion Note that the Mach number decreases nearly linearly along the duct.

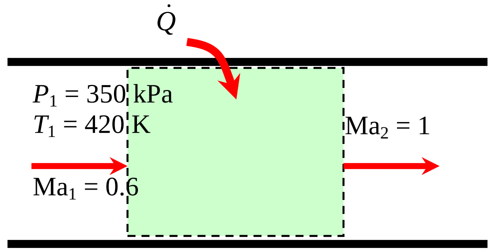
12-143

Solution Air flowing at a subsonic velocity in a duct is accelerated by heating. The highest rate of heat transfer without affecting the inlet conditions is to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K.

Analysis Heat transfer will stop when the flow is choked, and thus $\mathrm{Ma}_2 = V_2/c_2 = 1$. The inlet density and stagnation temperature are



$$\rho_1 = \frac{P_1}{RT_1} = \frac{350\text{ kPa}}{(0.287\text{ kJ/kgK})(420\text{ K})} = 2.904\text{ kg/m}^3$$

$$T_{01} = T_1\left(1 + \frac{k-1}{2}\mathrm{Ma}_1^2\right) = (420\text{ K})\left(1 + \frac{1.4\text{-}1}{2}0.6^2\right) = 450.2\text{ K}$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(420\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 410.8\text{ m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 0.6(410.8\text{ m/s}) = 246.5\text{ m/s}$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (2.904\text{ kg/m}^3)(0.1\times 0.1\text{m}^2)(246.5\text{ m/s}) = 7.157\text{ kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are $T_{02}/T_0^* = 1$ (since $\mathrm{Ma}_2 = 1$) .

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\mathrm{Ma}_1^2[2+(k-1)\mathrm{Ma}_1^2]}{(1+k\mathrm{Ma}_1^2)^2} = \frac{(1.4+1)0.6^2[2+(1.4-1)0.6^2]}{(1+1.4\times 0.6^2)^2} = 0.8189$$

Therefore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.8189} \qquad \rightarrow \quad T_{02} = T_{01}/0.8189 = (450.2\text{ K})/0.8189 = 549.8\text{ K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\mathrm{air}} c_p (T_{02} - T_{01}) = (7.157\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(549.8 - 450.2)\text{ K} = \mathbf{716\,kW}$$

Discussion It can also be shown that T_2 = 458 K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. We can also solve this problem using the Rayleigh function values listed in Table A-15.

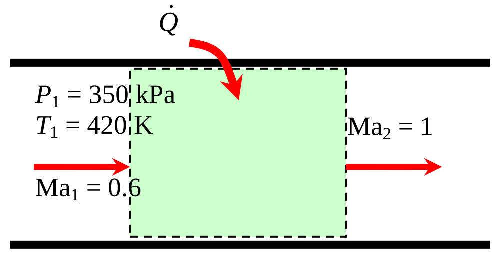
12-144

Solution Helium flowing at a subsonic velocity in a duct is accelerated by heating. The highest rate of heat transfer without affecting the inlet conditions is to be determined.

Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

Properties We take the properties of helium to be k = 1.667, c_p = 5.193 kJ/kg·K, and R = 2.077 kJ/kg·K.

Analysis Heat transfer will stop when the flow is choked, and thus $\text{Ma}_2 = V_2/c_2 = 1$. The inlet density and stagnation temperature are



$$\rho_1 = \frac{P_1}{RT_1} = \frac{350\ \text{kPa}}{(2.077\ \text{kJ/kgK})(420\ \text{K})} = 0.4012\ \text{kg/m}^3$$

$$T_{01} = T_1\left(1 + \frac{k-1}{2}\text{Ma}_1^2\right) = (420\ \text{K})\left(1 + \frac{1.667 - 1}{2}0.6^2\right) = 470.4\ \text{K}$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.667)(2.077\ \text{kJ/kg}\cdot\text{K})(420\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 1206\ \text{m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 0.6(1206\ \text{m/s}) = 723.5\ \text{m/s}$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (0.4012\ \text{kg/m}^3)(0.1\times 0.1\text{m}^2)(723.5\ \text{m/s}) = 2.903\ \text{kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are $T_{02}/T_0^* = 1$ (since $\text{Ma}_2 = 1$)

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\text{Ma}_1^2[2+(k-1)\text{Ma}_1^2]}{(1+k\text{Ma}_1^2)^2} = \frac{(1.667+1)0.6^2[2+(1.667-1)0.6^2]}{(1+1.667\times 0.6^2)^2} = 0.8400$$

Therefore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.8400} \quad \rightarrow \quad T_{02} = T_{01}/0.8400 = (470.4\ \text{K})/0.8400 = 560.0\ \text{K}$$

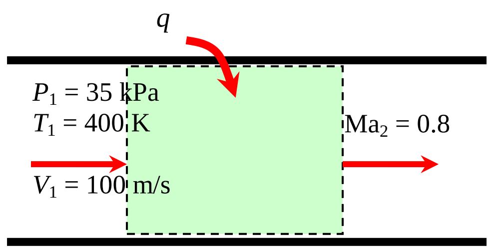
Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (2.903\ \text{kg/s})(5.193\ \text{kJ/kg}\cdot\text{K})(560.0 - 470.4)\ \text{K} = \mathbf{1350\ kW}$$

Discussion It can also be shown that T_2 = 420 K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. Also, in the solution of this problem, we cannot use the values of Table A-15 since they are based on k = 1.4.

12-145

Solution Air flowing at a subsonic velocity in a duct is accelerated by heating. For a specified exit Mach number, the heat transfer for a specified exit Mach number as well as the maximum heat transfer are to be determined.



Assumptions **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K.

Analysis The inlet Mach number and stagnation temperature are

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(400\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 400.9\text{ m/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{100\text{ m/s}}{400.9\text{ m/s}} = 0.2494$$

$$T_{01} = T_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right) = (400\text{ K})\left(1+\frac{1.4\text{-}1}{2}0.2494^2\right) = 405.0\text{ K}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-15):

$\text{Ma}_1 = 0.2494$: $T_{01}/T^* = 0.2559$
$\text{Ma}_2 = 0.8$: $T_{02}/T^* = 0.9639$

Then the exit stagnation temperature and the heat transfer are determined to be

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{0.9639}{0.2559} = 3.7667 \quad\rightarrow\quad T_{02} = 3.7667T_{01} = 3.7667(405.0\text{ K}) = 1526\text{ K}$$

$$q = c_p\left(T_{02}-T_{01}\right) = \left(1.005\text{ kJ/kg}\cdot\text{K}\right)\left(1526-405\right)\text{ K} = 1126\text{ kJ/kg} \cong \mathbf{1130\ kJ/kg}$$

Maximum heat transfer will occur when the flow is choked, and thus $\text{Ma}_2 = 1$ and thus $T_{02}/T^* = 1$. Then,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.2559} \quad\rightarrow\quad T_{02} = T_{01}/0.2559 = (405\text{ K})/0.2559 = 1583\text{ K}$$

$$q_{max} = c_p(T_{02}-T_{01}) = (1.005\text{ kJ/kg}\cdot\text{K})(1583-405)\text{ K} = 1184\text{ kJ/kg} \cong \mathbf{1180\ kJ/kg}$$

Discussion This is the maximum heat that can be transferred to the gas without affecting the mass flow rate. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease.

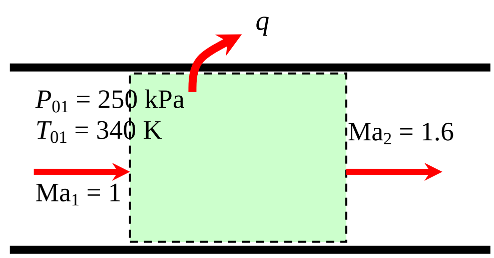
12-146

Solution Air flowing at sonic conditions in a duct is accelerated by cooling. For a specified exit Mach number, the amount of heat transfer per unit mass is to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K.

Analysis Noting that Ma_1 = 1, the inlet stagnation temperature is



$$T_{01} = T_1\left(1 + \frac{k-1}{2}\mathrm{Ma}_1^2\right) = (340\text{ K})\left(1 + \frac{1.4\text{-}1}{2}1^2\right) = 408\text{ K}$$

The Rayleigh flow functions T_0/T_0^* corresponding to the inlet and exit Mach numbers are (Table A-15):

Ma_1 = 1: T_{01}/T_0^* = 1

Ma_2 = 1.6: T_{02}/T_0^* = 0.8842

Then the exit stagnation temperature and heat transfer are determined to be

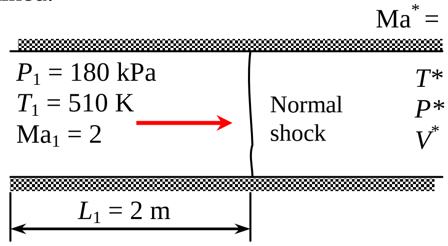
$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{0.8842}{1} = 0.8842 \qquad \rightarrow \qquad T_{02} = 0.8842T_{01} = 0.8842(408\text{ K}) = 360.75\text{ K}$$

$$q = c_p(T_{02} - T_{01}) = (1.005\text{ kJ/kg}\cdot\text{K})(360.75 - 408)\text{ K} = -47.49\text{ kJ/kg} \cong \mathbf{-47.5\text{ kJ/kg}}$$

Discussion The negative sign confirms that the gas needs to be cooled in order to be accelerated.

12-147

Solution Combustion gases enter a constant-area adiabatic duct at a specified state, and undergo a normal shock at a specified location. The exit velocity, temperature, and pressure are to be determined.



Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor is constant along the duct.

Properties The specific heat ratio and gas constant of combustion gases are given to be $k = 1.33$ and $R = 0.280$ kJ/kg·K. The friction factor is given to be $f = 0.010$.

Analysis The Fanno flow functions corresponding to the inlet Mach number of 2 are calculated from the relations in Table A-16 for $k = 1.33$ to be

$Ma_1 = 2$: $(fL^*/D_h)_1 = 0.3402$ $T_1/T^* = 0.7018$, $P_1/P^* = 0.4189$

First we check to make sure that the flow everywhere upstream the shock is supersonic. The required duct length from the inlet L_1^* for the flow to reach sonic conditions is $L_1^* = 0.3402\dfrac{D}{f} = 0.3402\dfrac{0.10\text{ m}}{0.010} = 3.40\text{ m}$, which is greater than the actual length of 2 m. Therefore, the flow is indeed supersonic when the normal shock occurs at the indicated location. Also, using the actual duct length L_1, we have $\dfrac{fL_1}{D_h} = \dfrac{(0.010)(2\text{ m})}{0.10\text{ m}} = 0.2000$. Noting that $L_1 = L_1^* - L_2^*$, the function fL^*/D_h at the exit state and the corresponding Mach number are $\left(\dfrac{fL^*}{D_h}\right)_2 = \left(\dfrac{fL^*}{D_h}\right)_1 - \dfrac{fL_1}{D_h} = 0.3402 - 0.2000 = 0.1402 \;\rightarrow\; Ma_2 = 1.476.$

From the relations in Table A-16, at $Ma_2 = 1.476$: $T_2/T^* = 0.8568$, $P_2/P^* = 0.6270$. Then the temperature, pressure, and velocity before the shock are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.8568}{0.7018} = 1.2209 \quad \rightarrow \quad T_2 = 1.2209T_1 = 1.2209(510\text{ K}) = 622.7\text{ K}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{0.6270}{0.4189} = 1.4968 \quad \rightarrow \quad P_2 = 1.4968P_1 = 1.4968(180\text{ kPa}) = 269.4\text{ kPa}$$

The normal shock functions corresponding to a Mach number of 1.476 are, from the relations in Table A-14,

$Ma_2 = 1.476$: $Ma_3 = 0.7052$, $T_3/T_2 = 1.2565$, $P_3/P_2 = 2.3466$

Then the temperature and pressure after the shock become

$$T_3 = 1.2565T_2 = 1.2565(622.7\text{ K}) = 782.4\text{ K}$$

$$P_3 = 2.3466P_2 = 2.3466(269.4\text{ kPa}) = 632.3\text{ kPa}$$

Sonic conditions exist at the duct exit, and the flow downstream of the shock is still Fanno flow. From the relations in Table A-16,

$Ma_3 = 0.7052$: $T_3/T^* = 1.0767$, $P_3/P^* = 1.4713$

$Ma_4 = 1$: $T_4/T^* = 1$, $P_4/P^* = 1$

Then the temperature, pressure, and velocity at the duct exit are determined to be

$$\frac{T_4}{T_3} = \frac{T_4/T^*}{T_3/T^*} = \frac{1}{1.0767} \quad \rightarrow \quad T_4 = T_3/1.0767 = (782.4\text{ K})/1.0767 = \mathbf{727\,K}$$

$$\frac{P_4}{P_3} = \frac{P_4/P^*}{P_3/P^*} = \frac{1}{1.4713} \quad \rightarrow \quad P_4 = P_3/1.4713 = (632.3\text{ kPa})/1.4713 = \mathbf{430\,kPa}$$

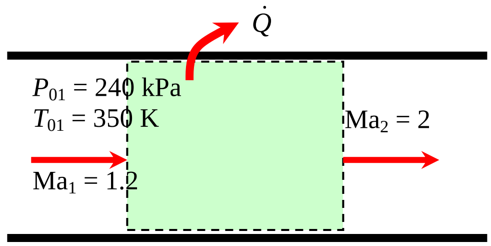
$$V_4 = Ma_4 c_4 = (1)\sqrt{kRT_4} = \sqrt{(1.33)(0.280\text{ kJ/kg}\cdot\text{K})(727\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{520\,m/s}$$

Discussion It can be shown that $L_3^* = 2.13$ m, and thus the total length of this duct is 4.13 m. If the duct is extended, the normal shock will move farther upstream, and eventually to the inlet of the duct.

12-148

Solution Air flowing at a supersonic velocity in a duct is accelerated by cooling. For a specified exit Mach number, the rate of heat transfer is to be determined.

Assumptions The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.



Properties We take the properties of air to be k = 1.4, c_p = 1.005 kJ/kg·K, and R = 0.287 kJ/kg·K.

Analysis Knowing stagnation properties, the static properties are determined to be

$$T_1 = T_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-1} = (350\text{ K})\left(1+\frac{1.4\text{-}1}{2}1.2^2\right)^{-1} = 271.7\text{ K}$$

$$P_1 = P_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-k/(k-1)} = (240\text{ kPa})\left(1+\frac{1.4\text{-}1}{2}1.2^2\right)^{-1.4/0.4} = 98.97\text{ kPa}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{98.97\text{ kPa}}{(0.287\text{ kJ/kgK})(271.7\text{ K})} = 1.269\text{ kg/m}^3$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(271.7\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 330.4\text{ m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 1.2(330.4\text{ m/s}) = 396.5\text{ m/s}$$

$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (1.269\text{ kg/m}^3)[\pi(0.20\text{ m})^2/4](396.5\text{ m/s}) = 15.81\text{ kg/s}$$

The Rayleigh flow functions T_0/T_0^* corresponding to the inlet and exit Mach numbers are (Table A-15):

Ma_1 = 1.8: T_{01}/T_0^* = 0.9787
Ma_2 = 2: T_{02}/T_0^* = 0.7934

Then the exit stagnation temperature is determined to be

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{0.7934}{0.9787} = 0.8107 \quad \rightarrow \quad T_{02} = 0.8107T_{01} = 0.8107(350\text{ K}) = 283.7\text{ K}$$

Finally, the rate of heat transfer is

$$\dot{Q} = \dot{m}_{\text{air}} c_p \left(T_{02} - T_{01}\right) = (15.81\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(283.7-350)\text{ K} = -1053\text{ kW} \cong \mathbf{-1050\ kW}$$

Discussion The negative sign confirms that the gas needs to be cooled in order to be accelerated. Also, it can be shown that the thermodynamic temperature drops to 158 K at the exit, which is extremely low. Therefore, the duct may need to be heavily insulated to maintain indicated flow conditions.

12-149

Solution Choked subsonic airflow in a constant cross-sectional area adiabatic duct is considered. The effect of duct length on the mass flow rate and the inlet conditions is to be investigated as the duct length is doubled.

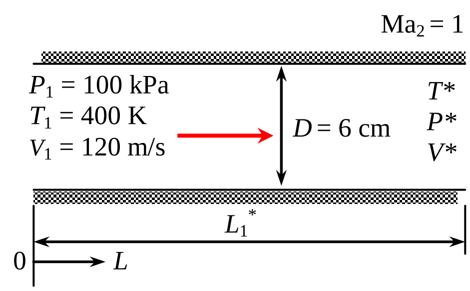
Assumptions **1** The assumptions associated with Fanno flow (i.e., steady, frictional flow of an ideal gas with constant properties through a constant cross-sectional area adiabatic duct) are valid. **2** The friction factor remains constant along the duct.

Properties We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K. The average friction factor is given to be $f = 0.02$.

Analysis We use EES to solve the problem. The flow is choked, and thus $Ma_2 = 1$. The inlet Mach number is

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(400\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 400.9\text{ m/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{120\text{ m/s}}{400.9\text{ m/s}} = 0.2993$$



Corresponding to this Mach number we have, from Table A-16, $fL^*/D_h = 5.3312$, Therefore, the original duct length is

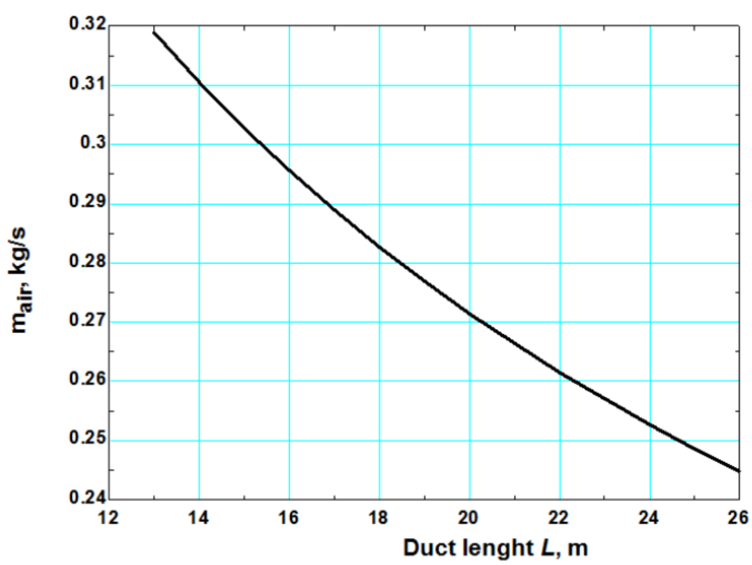
$$L = L_1^* = 5.3312\frac{D}{f} = 5.3312\frac{0.06\text{ m}}{0.02} = 16.0\text{ m}$$

Then the initial mass flow rate becomes

$$\rho_1 = \frac{P_1}{RT_1} = \frac{100\text{ kPa}}{(0.287\text{ kJ/kgK})(400\text{ K})} = 0.8711\text{ kg/m}^3$$

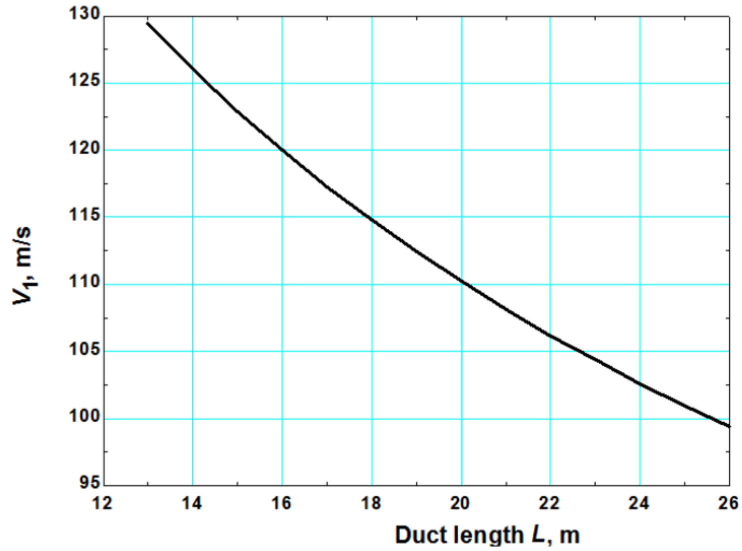
$$\dot{m}_{air} = \rho_1 A_{c1} V_1 = (0.8711\text{ kg/m}^3)[\pi(0.06\text{ m})^2/4](120\text{ m/s}) = 0.296\text{ kg/s}$$

Duct length L, m	Inlet velocity V_1, m/s	Mass flow rate $\dot{m}_{air}$, kg/s
13	129	0.319
14	126	0.310
15	123	0.303
16	120	0.296
17	117	0.289
18	115	0.283
19	112	0.277
20	110	0.271
21	108	0.266
22	106	0.262
23	104	0.257
24	103	0.253
25	101	0.249
26	99	0.245



The EES program is listed below, along with a plot of inlet velocity vs. duct length:

```
k=1.4
cp=1.005
R=0.287

P1=100
T1=400
"L=26"
Ma2=1
f=0.02
D=0.06

C1=sqrt(k*R*T1*1000)
Ma1=V1/C1
T01=T02
T01=T1*(1+0.5*(k-1)*Ma1^2)
T02=T2*(1+0.5*(k-1)*Ma2^2)
P01=P1*(1+0.5*(k-1)*Ma1^2)^(k/(k-1))

rho1=P1/(R*T1)
Ac=pi*D^2/4
mair=rho1*Ac*V1
```




```
P01Ps=((2+(k-1)*Ma1^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma1
P1Ps=((k+1)/(2+(k-1)*Ma1^2))^0.5/Ma1
T1Ts=(k+1)/(2+(k-1)*Ma1^2)
R1Rs=((2+(k-1)*Ma1^2)/(k+1))^0.5/Ma1
V1Vs=1/R1Rs
fLs1=(1-Ma1^2)/(k*Ma1^2)+(k+1)/(2*k)*ln((k+1)*Ma1^2/(2+(k-1)*Ma1^2))
Ls1=fLs1*D/f

P02Ps=((2+(k-1)*Ma2^2)/(k+1))^(0.5*(k+1)/(k-1))/Ma2
P2Ps=((k+1)/(2+(k-1)*Ma2^2))^0.5/Ma2
T2Ts=(k+1)/(2+(k-1)*Ma2^2)
R2Rs=((2+(k-1)*Ma2^2)/(k+1))^0.5/Ma2
V2Vs=1/R2Rs
fLs2=(1-Ma2^2)/(k*Ma2^2)+(k+1)/(2*k)*ln((k+1)*Ma2^2/(2+(k-1)*Ma2^2))
Ls2=fLs2*D/f
L=Ls1-Ls2

P02=P02Ps/P01Ps*P01
P2=P2Ps/P1Ps*P1
V2=V2Vs/V1Vs*V1
```

Discussion Note that once the flow is choked, any increase in duct length results in a decrease in the mass flow rate and the inlet velocity.

12-150

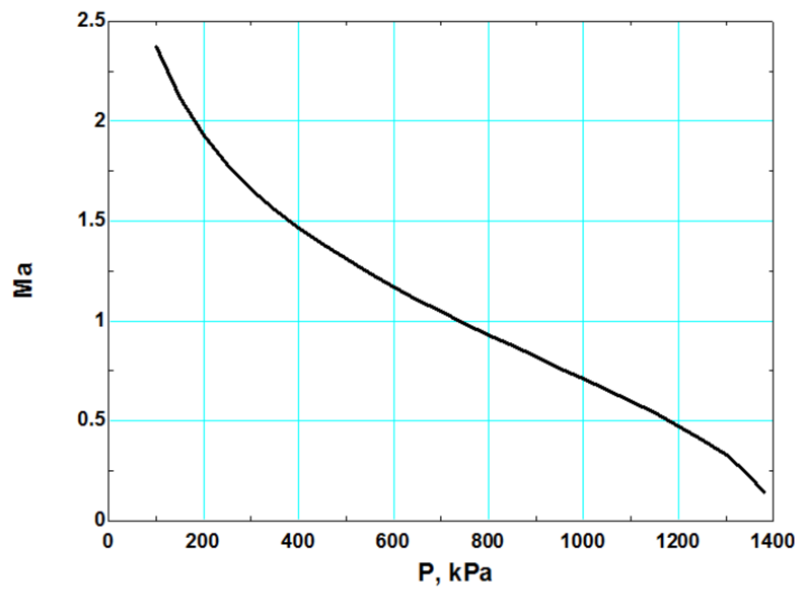
Solution Using EES (or other) software, the shape of a converging-diverging nozzle is to be determined for specified flow rate and stagnation conditions. The nozzle and the Mach number are to be plotted.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

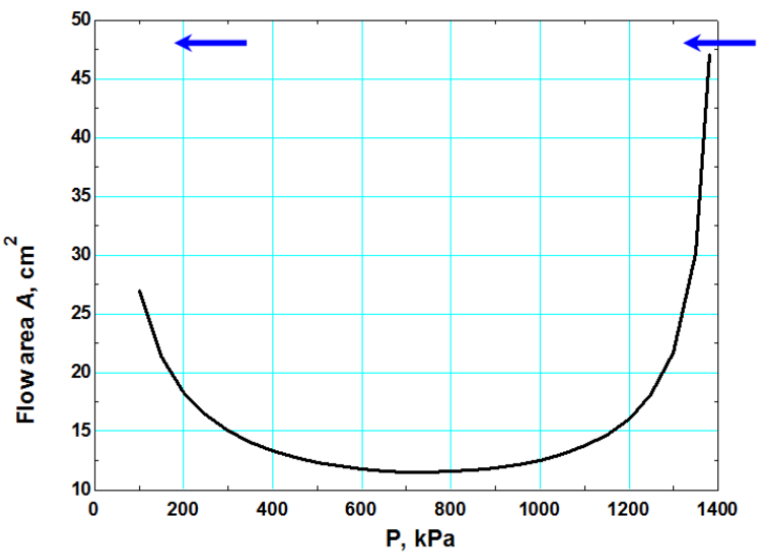
Properties The specific heat ratio of air at room temperature is 1.4.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
P0=1400 "kPa"
T0=200+273 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
T=T0*(P/P0)^((k-1)/k)
V=SQRT(2*Cp*(T0-T)*1000)
A=m/(rho*V)*10000 "cm2"
C=SQRT(k*R*T*1000)
Ma=V/C
```



Pressure P, kPa	Flow area A, cm^2	Mach number Ma
1400	∞	0
1350	30.1	0.229
1300	21.7	0.327
1250	18.1	0.406
1200	16.0	0.475
1150	14.7	0.538
1100	13.7	0.597
1050	13.0	0.655
1000	12.5	0.710
950	12.2	0.766
900	11.9	0.820
850	11.7	0.876
800	11.6	0.931
750	11.5	0.988
700	11.5	1.047
650	11.6	1.107
600	11.8	1.171
550	12.0	1.237
500	12.3	1.308
450	12.8	1.384
400	13.3	1.467
350	14.0	1.559
300	15.0	1.663
250	16.4	1.784
200	18.3	1.929
150	21.4	2.114
100	27.0	2.373



Discussion The shape is not actually to scale since the horizontal axis is pressure rather than distance. If the pressure decreases linearly with distance, then the shape *would* be to scale.

12-151

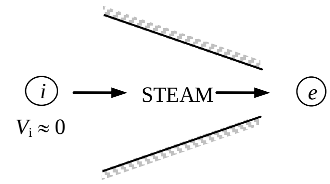
Solution Steam enters a converging nozzle. The exit pressure, the exit velocity, and the mass flow rate versus the back pressure for a specified back pressure range are to be plotted.

Assumptions **1** Steam is to be treated as an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

Properties The ideal gas properties of steam are R = 0.462 kJ/kg.K, c_p = 1.872 kJ/kg.K, and k = 1.3.

Analysis We use EES to solve the problem. The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Since the flow is isentropic, they remain constant throughout the nozzle,

$P_0 = P_\mathrm{i}$ = 6 MPa and $T_0 = T_i$ = 700 K



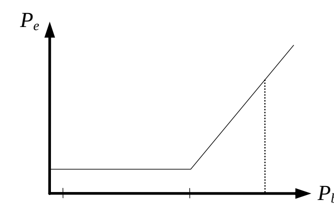
The critical pressure is determined to be

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (6\,\mathrm{MPa})\left(\frac{2}{1.3+1}\right)^{1.3/0.3} = 3.274\,\mathrm{MPa}$$

Then the pressure at the exit plane (throat) is

$P_e = P_b$ for $P_b \geq 3.274$ MPa

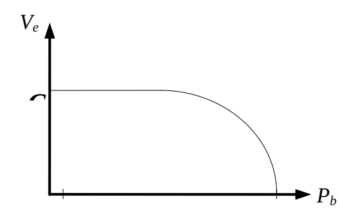
P_e = P* = 3.274 MPa for $P_b < 3.274$ MPa (choked flow)



Thus the back pressure does not affect the flow when 3 < P_b < 3.274 MPa. For a specified exit pressure P_e, the temperature, velocity, and mass flow rate are

Temperature $$T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (700\,\mathrm{K})\left(\frac{P_e}{6}\right)^{0.3/1.3}$$

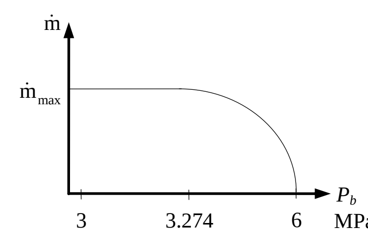
Velocity $$V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.872\,\mathrm{kJ/kg\cdot K})(700 - T_e)\left(\frac{1000\,\mathrm{m^2/s^2}}{1\,\mathrm{kJ/kg}}\right)}$$



Density $$\rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.462\,\mathrm{kPa\cdot m^3/kg\cdot K})T_e}$$

Mass flow rate $$\dot{m} = \rho_e V_e A_e = \rho_e V_e (0.0008\,\mathrm{m^2})$$

The results of the calculations are tabulated as follows:



P_b, MPa	P_e, MPa	T_e, K	V_e, m/s	ρ_e, kg/m^3	$\dot{m}$, kg / s
6.0	6.0	700	0	18.55	0
5.5	5.5	686.1	228.1	17.35	3.166
5.0	5.0	671.2	328.4	16.12	4.235
4.5	4.5	655.0	410.5	14.87	4.883
4.0	4.0	637.5	483.7	13.58	5.255
3.5	3.5	618.1	553.7	12.26	5.431
3.274	3.274	608.7	584.7	11.64	5.445
3.0	3.274	608.7	584.7	11.64	5.445

Discussion Once the back pressure drops below 3.274 MPa, the flow is choked, and $\dot{m}$ remains constant from then on.

12-152

Solution An expression for the ratio of the stagnation pressure after a shock wave to the static pressure before the shock wave as a function of k and the Mach number upstream of the shock wave is to be found.

Analysis The relation between P_1 and P_2 is

$$\frac{P_2}{P_1}=\frac{1+k\mathrm{Ma}_2^2}{1+k\mathrm{Ma}_1^2}\longrightarrow P_2=P_1\left(\frac{1+k\mathrm{Ma}_1^2}{1+k\mathrm{Ma}_2^2}\right)$$

We substitute this into the isentropic relation

$$\frac{P_{02}}{P_2}=\left(1+\left(k-1\right)\mathrm{Ma}_2^2/2\right)^{k/(k-1)}$$

which yields

$$\frac{P_{02}}{P_1}=\left(\frac{1+k\mathrm{Ma}_1^2}{1+k\mathrm{Ma}_2^2}\right)\left(1+(k-1)\mathrm{Ma}_2^2/2\right)^{k/(k-1)}$$

where

$$\mathrm{Ma}_2^2=\frac{\mathrm{Ma}_1^2+2/(k-1)}{2k\mathrm{Ma}_2^2/(k-1)-1}$$

Substituting,

$$\boxed{\frac{P_{02}}{P_1}=\left(\frac{(1+k\mathrm{Ma}_1^2)(2k\mathrm{Ma}_1^2-k+1)}{k\mathrm{Ma}_1^2(k+1)-k+3}\right)\left(1+\frac{(k-1)\mathrm{Ma}_1^2/2+1}{2k\mathrm{Ma}_1^2/(k-1)-1}\right)^{k/(k-1)}}$$

Discussion Similar manipulations of the equations can be performed to get the ratio of other parameters across a shock.

12-153

Solution Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-13 for air.

Properties The specific heat ratio is given to be k = 1.4 for air.

Analysis The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}^* = \mathrm{Ma}\sqrt{\frac{k+1}{2+(k-1)\mathrm{Ma}^2}} \qquad \frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-k/(k-1)} \qquad \frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1}$$

Air:

```
k=1.4
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M
```

Ma	Ma*	A/A^*	P/P_0	ρ/ρ_0	T/T_0
1.0	1.0000	1.0000	0.5283	0.6339	0.8333
1.5	1.3646	1.1762	0.2724	0.3950	0.6897
2.0	1.6330	1.6875	0.1278	0.2300	0.5556
2.5	1.8257	2.6367	0.0585	0.1317	0.4444
3.0	1.9640	4.2346	0.0272	0.0762	0.3571
3.5	2.0642	6.7896	0.0131	0.0452	0.2899
4.0	2.1381	10.7188	0.0066	0.0277	0.2381
4.5	2.1936	16.5622	0.0035	0.0174	0.1980
5.0	2.2361	25.0000	0.0019	0.0113	0.1667
5.5	2.2691	36.8690	0.0011	0.0076	0.1418
6.0	2.2953	53.1798	0.0006	0.0052	0.1220
6.5	2.3163	75.1343	0.0004	0.0036	0.1058
7.0	2.3333	104.1429	0.0002	0.0026	0.0926
7.5	2.3474	141.8415	0.0002	0.0019	0.0816
8.0	2.3591	190.1094	0.0001	0.0014	0.0725
8.5	2.3689	251.0862	0.0001	0.0011	0.0647
9.0	2.3772	327.1893	0.0000	0.0008	0.0581
9.5	2.3843	421.1314	0.0000	0.0006	0.0525
10.0	2.3905	535.9375	0.0000	0.0005	0.0476

Discussion The tabulated values are useful for quick calculations, but be careful – they apply only to one specific value of k, in this case k = 1.4.

12-154

Solution Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-13 for methane.

Properties The specific heat ratio is given to be k = 1.3 for methane.

Analysis The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}^* = \mathrm{Ma}\sqrt{\frac{k+1}{2+(k-1)\mathrm{Ma}^2}} \qquad \frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-k/(k-1)} \qquad \frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1}$$

Methane:

```
k=1.3
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M
```

Ma	Ma*	A/A^*	P/P_0	ρ/ρ_0	T/T_0
1.0	1.0000	1.0000	0.5457	0.6276	0.8696
1.5	1.3909	1.1895	0.2836	0.3793	0.7477
2.0	1.6956	1.7732	0.1305	0.2087	0.6250
2.5	1.9261	2.9545	0.0569	0.1103	0.5161
3.0	2.0986	5.1598	0.0247	0.0580	0.4255
3.5	2.2282	9.1098	0.0109	0.0309	0.3524
4.0	2.3263	15.9441	0.0050	0.0169	0.2941
4.5	2.4016	27.3870	0.0024	0.0095	0.2477
5.0	2.4602	45.9565	0.0012	0.0056	0.2105
5.5	2.5064	75.2197	0.0006	0.0033	0.1806
6.0	2.5434	120.0965	0.0003	0.0021	0.1563
6.5	2.5733	187.2173	0.0002	0.0013	0.1363
7.0	2.5978	285.3372	0.0001	0.0008	0.1198
7.5	2.6181	425.8095	0.0001	0.0006	0.1060
8.0	2.6350	623.1235	0.0000	0.0004	0.0943
8.5	2.6493	895.5077	0.0000	0.0003	0.0845
9.0	2.6615	1265.6040	0.0000	0.0002	0.0760
9.5	2.6719	1761.2133	0.0000	0.0001	0.0688
10.0	2.6810	2416.1184	0.0000	0.0001	0.0625

Discussion The tabulated values are useful for quick calculations, but be careful – they apply only to one specific value of k, in this case k = 1.3.

Fundamentals of Engineering (FE) Exam Problems

12-155

An aircraft is cruising in still air at 5°C at a velocity of 400 m/s. The air temperature at the nose of the aircraft where stagnation occurs is

(a) 5°C (b) 25°C (c) 55°C (d) 80°C (e) 85°C

Answer (e) 85°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
T1=5 "C"
Vel1= 400 "m/s"
T1_stag=T1+Vel1^2/(2*Cp*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Tstag=T1 "Assuming temperature rise"
W2_Tstag=Vel1^2/(2*Cp*1000) "Using just the dynamic temperature"
W3_Tstag=T1+Vel1^2/(Cp*1000) "Not using the factor 2"
```

12-156

Air is flowing in a wind tunnel at 25°C, 80 kPa, and 250 m/s. The stagnation pressure at a probe inserted into the flow stream is

(a) 87 kPa (b) 93 kPa (c) 113 kPa (d) 119 kPa (e) 125 kPa

Answer (c) 113 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
T1=25 "K"
P1=80 "kPa"
Vel1= 250 "m/s"
T1_stag=(T1+273)+Vel1^2/(2*Cp*1000) "C"
T1_stag/(T1+273)=(P1_stag/P1)^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
T11_stag/T1=(W1_P1stag/P1)^((k-1)/k);   T11_stag=T1+Vel1^2/(2*Cp*1000)  "Using deg. C for temperatures"
T12_stag/(T1+273)=(W2_P1stag/P1)^((k-1)/k);   T12_stag=(T1+273)+Vel1^2/(Cp*1000)  "Not using the factor 2"
T13_stag/(T1+273)=(W3_P1stag/P1)^(k-1);   T13_stag=(T1+273)+Vel1^2/(2*Cp*1000)  "Using wrong isentropic relation"
```

12-157

An aircraft is reported to be cruising in still air at -20°C and 40 kPa at a Mach number of 0.86. The velocity of the aircraft is

(a) 91 m/s (b) 220 m/s (c) 186 m/s (d) 280 m/s (e) 378 m/s

Answer (d) 280 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
T1=-20+273 "K"
P1=40 "kPa"
Mach=0.86
VS1=SQRT(k*R*T1*1000)
Mach=Vel1/VS1

"Some Wrong Solutions with Common Mistakes:"
W1_vel=Mach*VS2; VS2=SQRT(k*R*T1) "Not using the factor 1000"
W2_vel=VS1/Mach "Using Mach number relation backwards"
W3_vel=Mach*VS3; VS3=k*R*T1 "Using wrong relation"
```

12-158

Air is flowing in a wind tunnel at 12°C and 66 kPa at a velocity of 230 m/s. The Mach number of the flow is

(a) 0.54 (b) 0.87 (c) 3.3 (d) 0.36 (e) 0.68

Answer (e) 0.68

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
T1=12+273 "K"
P1=66 "kPa"
Vel1=230 "m/s"
VS1=SQRT(k*R*T1*1000)
Mach=Vel1/VS1

"Some Wrong Solutions with Common Mistakes:"
W1_Mach=Vel1/VS2; VS2=SQRT(k*R*(T1-273)*1000) "Using C for temperature"
W2_Mach=VS1/Vel1 "Using Mach number relation backwards"
W3_Mach=Vel1/VS3; VS3=k*R*T1 "Using wrong relation"
```

12-159

Consider a converging nozzle with a low velocity at the inlet and sonic velocity at the exit plane. Now the nozzle exit diameter is reduced by half while the nozzle inlet temperature and pressure are maintained the same. The nozzle exit velocity will

(a) remain the same (b) double (c) quadruple (d) go down by half (e) go down to one-fourth

Answer (a) remain the same

12-160

Air is approaching a converging-diverging nozzle with a low velocity at 12°C and 200 kPa, and it leaves the nozzle at a supersonic velocity. The velocity of air at the throat of the nozzle is

(a) 338 m/s (b) 309 m/s (c) 280 m/s (d) 256 m/s (e) 95 m/s

Answer (b) 309 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
"Properties at the inlet"
T1=12+273 "K"
P1=200 "kPa"
Vel1=0 "m/s"
To=T1 "since velocity is zero"
Po=P1
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Vthroat=SQRT(k*R*T1*1000) "Using T1 for temperature"
W2_Vthroat=SQRT(k*R*T2_throat*1000); T2_throat=2*(To-273)/(k+1) "Using C for temperature"
W3_Vthroat=k*R*T_throat "Using wrong relation"
```

12-161

Argon gas is approaching a converging-diverging nozzle with a low velocity at 20°C and 120 kPa, and it leaves the nozzle at a supersonic velocity. If the cross-sectional area of the throat is 0.015 m^2, the mass flow rate of argon through the nozzle is

(a) 0.41 kg/s (b) 3.4 kg/s (c) 5.3 kg/s (d) 17 kg/s (e) 22 kg/s

Answer (c) 5.3 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=0.5203 "kJ/kg.K"
R=0.2081 "kJ/kg.K"
A=0.015 "m^2"
"Properties at the inlet"
T1=20+273 "K"
P1=120 "kPa"
Vel1=0 "m/s"
To=T1 "since velocity is zero"
Po=P1
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
rho_throat=P_throat/(R*T_throat)
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)
m=rho_throat*A*V_throat

"Some Wrong Solutions with Common Mistakes:"
W1_mass=rho_throat*A*V1_throat;  V1_throat=SQRT(k*R*T1_throat*1000); T1_throat=2*(To-273)/(k+1) "Using C for temp"
W2_mass=rho2_throat*A*V_throat; rho2_throat=P1/(R*T1)  "Using density at inlet"
```

12-162

Carbon dioxide enters a converging-diverging nozzle at 60 m/s, 310°C, and 300 kPa, and it leaves the nozzle at a supersonic velocity. The velocity of carbon dioxide at the throat of the nozzle is

(a) 125 m/s (b) 225 m/s (c) 312 m/s (d) 353 m/s (e) 377 m/s

Answer (d) 353 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
"Properties at the inlet"
T1=310+273 "K"
P1=300 "kPa"
Vel1=60 "m/s"
To=T1+Vel1^2/(2*Cp*1000)
To/T1=(Po/P1)^((k-1)/k)
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Vthroat=SQRT(k*R*T1*1000) "Using T1 for temperature"
W2_Vthroat=SQRT(k*R*T2_throat*1000); T2_throat=2*(T_throat-273)/(k+1) "Using C for temperature"
W3_Vthroat=k*R*T_throat "Using wrong relation"
```

12-163

Consider gas flow through a converging-diverging nozzle. Of the five statements below, select the one that is incorrect:

(a) The fluid velocity at the throat can never exceed the speed of sound.

(b) If the fluid velocity at the throat is below the speed of sound, the diversion section will act like a diffuser.

(c) If the fluid enters the diverging section with a Mach number greater than one, the flow at the nozzle exit will be supersonic.

(d) There will be no flow through the nozzle if the back pressure equals the stagnation pressure.

(e) The fluid velocity decreases, the entropy increases, and stagnation enthalpy remains constant during flow through a normal shock.

Answer (c) If the fluid enters the diverging section with a Mach number greater than one, the flow at the nozzle exit will be supersonic.

12-164

Combustion gases with k = 1.33 enter a converging nozzle at stagnation temperature and pressure of 350°C and 400 kPa, and are discharged into the atmospheric air at 20°C and 100 kPa. The lowest pressure that will occur within the nozzle is

(a) 13 kPa (b) 100 kPa (c) 216 kPa (d) 290 kPa (e) 315 kPa

Answer (c) 216 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.33
Po=400 "kPa"
"The critical pressure is"
P_throat=Po*(2/(k+1))^(k/(k-1))
"The lowest pressure that will occur in the nozzle is the higher of the critical or atmospheric pressure."

"Some Wrong Solutions with Common Mistakes:"
W2_Pthroat=Po*(1/(k+1))^(k/(k-1)) "Using wrong relation"
W3_Pthroat=100 "Assuming atmospheric pressure"
```

Design and Essay Problems

12-165 to 12-167

Solution Students' essays and designs should be unique and will differ from each other.

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

Third Edition

Yunus A. Çengel & John M. Cimbala

McGraw-Hill, 2013

Chapter 13
OPEN-CHANNEL FLOW

Classification, Froude Number, and Wave Speed

13-1C

Solution We are to define normal depth and how it is established.

Analysis In open channels of constant slope and constant cross-section, the fluid accelerates until the head loss due to frictional effects equals the elevation drop. The fluid at this point reaches its terminal velocity, and uniform flow is established. The flow remains uniform as long as the slope, cross-section, and the surface roughness of the channel remain unchanged. **The flow depth in uniform flow** is called the *normal depth* y_n, which is an important characteristic parameter for open-channel flows.

Discussion The normal depth is a fairly strong function of surface roughness.

13-2C

Solution We are to discuss how pressure changes along the free surface in open-channel flow.

Analysis The free surface **coincides with the hydraulic grade line** (HGL), and the **pressure is constant along the free surface**.

Discussion At a free surface of a liquid, the pressure must be equal to the pressure of the gas above it.

13-3C

Solution We are to determine if the slope of the free surface is equal to the slope of the channel bottom.

Analysis **No in general**. The slope of the free surface is **not necessarily equal** to the slope of the bottom surface even during steady fully developed flow.

Discussion However, there are situations called *uniform flow* in which the conditions here are met.

13-4C

Solution We are to discuss some reasons for nonuniform flow in open channels, and the difference between rapidly varied flow and gradually varied flow.

Analysis The **presence of an obstruction in a channel such as a gate or a change in slope or cross-section** causes the flow depth to vary, and thus the flow to become varied or nonuniform. The varied flow is called *rapidly varied flow* (RVF) if the flow depth changes markedly over a relatively short distance in the flow direction (such as the flow of water past a partially open gate or shortly before a falls), and *gradually varied flow* (GVF) if the flow depth changes gradually over a long distance along the channel.

Discussion The equations of GVF are simplified because of the slow changes in the flow direction.

13-5C

Solution We are to discuss the driving force in open-channel flow and how flow rate is determined.

Analysis Flow in a channel is driven **naturally by gravity**. Water flow in a river, for example, is driven by the elevation difference between the source and the sink. The flow rate in an open channel is established by the **dynamic balance between gravity and friction**. Inertia of the flowing fluid also becomes important in unsteady flow.

Discussion In pipe flow, on the other hand, there may be an additional driving force of pressure due to pumps.

13-6C

Solution We are to discuss the difference between uniform and nonuniform flow.

Analysis The flow in a channel is said to be *uniform* if the **flow depth (and thus the average velocity) remains constant**. Otherwise, the flow is said to be *nonuniform* or *varied*, indicating that the flow depth varies with distance in the flow direction. Uniform flow conditions are commonly encountered in practice in long straight sections of channels with constant slope and constant cross-section.

Discussion In uniform open-channel flow, the head loss due to frictional effects equals the elevation drop.

13-7C

Solution We are to explain how to determine if a flow is tranquil, critical, or rapid.

Analysis Knowing the average flow velocity and flow depth, the Froude number is determined from $\mathrm{Fr} = V/\sqrt{gy}$. Then the flow is classified as

Fr < 1	**Subcritical or tranquil flow**
Fr = 1	**Critical flow**
Fr > 1	**Supercritical or rapid flow**

Discussion The Froude number is the most important parameter in open-channel flow.

13-8C

Solution We are to discuss whether the flow upstream of a hydraulic jump must be supercritical, and whether the flow downstream of a hydraulic jump must be subcritical.

Analysis Upstream of a hydraulic jump, the **upstream flow must be supercritical**. Downstream of a hydraulic jump, the **downstream flow must be subcritical**.

Discussion Otherwise, the second law of thermodynamics would be violated. Note that a hydraulic jump is analogous to a normal shock wave – in that case, the flow upstream must be supersonic and the flow downstream must be subsonic.

13-9C

Solution We are to define critical length, and discuss how it is determined.

Analysis The **flow depth y_c corresponding to a Froude number of Fr = 1** is the *critical depth*, and it is determined from $\boxed{V=\sqrt{gy_c}}$ or $\boxed{y_c = V^2/g}$.

Discussion Critical depth is a useful parameter, even if the depth does not actually equal y_c anywhere in the flow.

13-10C

Solution We are to define and discuss the usefulness of the Froude number.

Analysis *Froude number*, defined as $\boxed{\mathrm{Fr} = V/\sqrt{gy}}$, is a **dimensionless parameter that governs the character of flow in open channels**. Here, g is the gravitational acceleration, V is the mean fluid velocity at a cross-section, and L_c is a characteristic length (L_c = flow depth y for wide rectangular channels). Fr represents the **ratio of inertia forces to viscous forces in open-channel flow**. The Froude number is also the **ratio of the flow speed to wave speed**, Fr = V/c_o.

Discussion The Froude number is the most important parameter in open-channel flow.

13-11

Solution A single wave is initiated in a sea by a strong jolt during an earthquake. The speed of the resulting wave is to be determined.

Assumptions The depth of water is constant,

Analysis Surface wave speed is determined the wave-speed relation to be

$$c_0 = \sqrt{gh} = \sqrt{(9.81\,\mathrm{m/s^2})(2000\,\mathrm{m})} = \mathbf{140 m/s}$$

Discussion Note that wave speed depends on the water depth, and the wave speed increases as the water depth increases. Also, the waves eventually die out because of the viscous effects.

13-12

Solution The flow of water in a wide channel is considered. The speed of a small disturbance in flow for two different flow depths is to be determined for both water and oil.

Assumptions The distance across the wave is short and thus friction at the bottom surface and air drag at the top are negligible,

Analysis Surface wave speed can be determined directly from the relation $c_0 = \sqrt{gh}$.

(a) $c_0 = \sqrt{gh} = \sqrt{(9.81\,\mathrm{m/s^2})(0.25\,\mathrm{m})} = \mathbf{1.57\ m/s}$

(b) $c_0 = \sqrt{gh} = \sqrt{(9.81\,\mathrm{m/s^2})(0.8\,\mathrm{m})} = \mathbf{2.80\ m/s}$

Therefore, a disturbance in the flow will travel at a speed of 0.990 m/s in the first case, and 2.80 m/s in the second case.

Discussion Note that wave speed depends on the water depth, and the wave speed increases as the water depth increases as long as the water remains shallow. Results would not change if the fluid were oil, because the wave speed depends only on the fluid depth.

13-13

Solution Water flows uniformly in a wide rectangular channel. For given values of flow depth and velocity, it is to be determined whether the flow is subcritical or supercritical.

Assumptions **1** The flow is uniform. **2** The channel is wide and thus the side wall effects are negligible.

Analysis The Froude number is $\mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{1.5\,\mathrm{m/s}}{\sqrt{(9.81\,\mathrm{m/s^2})(0.24\,\mathrm{m})}} = 0.978$, which is lower than 1.

Therefore, the flow is **subcritical**.

Discussion Note that the Froude Number is not function of any temperature-dependent properties, and thus temperature.

13-14

Solution Rain water flows on a concrete surface. For given values of flow depth and velocity, it is to be determined whether the flow is subcritical or supercritical.

Assumptions **1** The flow is uniform. **2** The thickness of water layer is constant.

Analysis The Froude number is $\mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{1.3\,\mathrm{m/s}}{\sqrt{(9.81\,\mathrm{m/s^2})(0.02\,\mathrm{m})}} = 2.93$, which is greater than 1.

Therefore, the flow is **supercritical**.

Discussion This water layer will undergo a hydraulic jump when the ground slope decreases or becomes adverse.

13-15E

Solution Water flows uniformly in a wide rectangular channel. For given flow depth and velocity, it is to be determined whether the flow is laminar or turbulent, and whether it is subcritical or supercritical.

Assumptions The flow is uniform.

Properties The density and dynamic viscosity of water at 70ºF are ρ = 62.30 lbm/ft^3 and μ = 6.556×10^{-4} lbm/ft·s.

Analysis (*a*) The Reynolds number of the flow is $\mathrm{Re} = \dfrac{\rho V y}{\mu} = \dfrac{(62.30\,\mathrm{lbm/ft^3})(6\,\mathrm{ft/s})(0.5\,\mathrm{ft})}{6.556\times10^{-4}\,\mathrm{lbm/ft\cdot s}} = 2.85\times10^5$, which is greater than the critical value of 500. Therefore, the flow is **turbulent**.

(*b*) The Froude number is $\mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{6\,\mathrm{ft/s}}{\sqrt{(32.2\,\mathrm{ft/s^2})(0.5\,\mathrm{ft})}} = 1.50$, which is greater than 1.

Therefore, the flow is **supercritical**.

Discussion The result in (*a*) is expected since almost all open channel flows are turbulent. Also, hydraulic radius for a wide rectangular channel approaches the water depth *y* as the ratio *y/*b approaches zero.

13-16

Solution Water flows uniformly in a wide rectangular channel. For given flow depth and velocity, it is to be determined whether the flow is laminar or turbulent, and whether it is subcritical or supercritical.

Assumptions The flow is uniform.

Properties The density and dynamic viscosity of water at 20ºC are ρ = 998.0 kg/m^3 and μ = 1.002×10^{-3} kg/m⋅s.

Analysis (a) The Reynolds number of the flow is $\mathrm{Re} = \dfrac{\rho V y}{\mu} = \dfrac{(998.0\ \mathrm{kg/m^3})(1.5\ \mathrm{m/s})(0.16\ \mathrm{m})}{1.002\times10^{-3}\ \mathrm{kg/m\cdot s}} = 2.390\times10^5$, which is greater than the critical value of 500. Therefore, the flow is **turbulent**.

(*b*) The Froude number is $\mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{1.5\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m/s^2})(0.16\ \mathrm{m})}} = 1.20$, which is greater than 1.

Therefore, the flow is **supercritical**.

Discussion The result in (*a*) is expected since almost all open channel flows are turbulent. Also, hydraulic radius for a wide rectangular channel approaches the water depth *y* as the ratio *y/*b approaches zero.

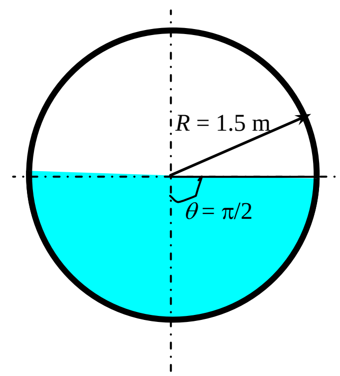
13-17

Solution Water flows uniformly through a half-full circular channel. For a given average velocity, the hydraulic radius, the Reynolds number, and the flow regime are to be determined.

Assumptions The flow is uniform.

Properties The density and dynamic viscosity of water at 10ºC are ρ = 999.7 kg/m^3 and μ = 1.307×10^{-3} kg/m⋅s.

Analysis From geometric considerations, the hydraulic radius is



$$R_h = \frac{A_c}{p} = \frac{\pi R^2/2}{\pi R} = \frac{R}{2} = \frac{1.5\ \mathrm{m}}{2} = \mathbf{0.75\ m}$$

The Reynolds number of the flow is

$$\mathrm{Re} = \frac{\rho V R_h}{\mu} = \frac{(999.7\ \mathrm{kg/m^3})(2.5\ \mathrm{m/s})(0.75\ \mathrm{m})}{1.307\times10^{-3}\ \mathrm{kg/m\cdot s}} = \mathbf{1.43\times10^6}$$, which is greater than the critical value of 500. Therefore, **the flow is turbulent**.

When calculating the Froude number, the hydraulic depth should be used rather than the maximum depth or the hydraulic radius. For a non-rectangular channel, hydraulic depth is defined as the ratio of the flow area to top width,

$$y_h = \frac{A_c}{\text{Top width}} = \frac{\pi R^2/2}{2R} = \frac{\pi R}{4} = \frac{\pi(1.5\ \mathrm{m})}{4} = 1.178\ \mathrm{m}$$

$\mathrm{Fr} = \dfrac{V}{\sqrt{g y_h}} = \dfrac{2.5\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m/s^2})(1.178\ \mathrm{m})}} = 0.735$, which is lower than 1. Therefore, the flow is **subcritical**.

Discussion If the maximum flow depth were used instead of the hydraulic depth, the result would still be subcritical flow, but this is not always the case.

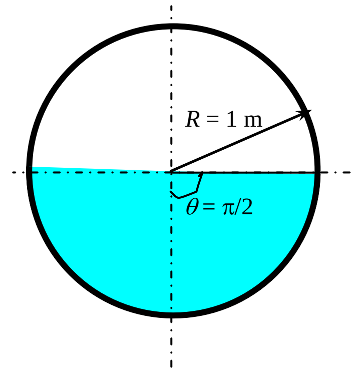
13-18

Solution Water flows uniformly through a half-full circular channel. For a given average velocity, the hydraulic radius, the Reynolds number, and the flow regime are to be determined.

Assumptions The flow is uniform.

Properties The density and dynamic viscosity of water at 10ºC are ρ = 999.7 kg/m^3 and μ = 1.307×10^{-3} kg/m·s.

Analysis From geometric considerations, the hydraulic radius is



$$R_h = \frac{A_c}{p} = \frac{\pi R^2/2}{\pi R} = \frac{R}{2} = \frac{1\,\text{m}}{2} = \mathbf{0.50\ m}$$

The Reynolds number of the flow is

$$\text{Re} = \frac{\rho V R_h}{\mu} = \frac{(999.7\ \text{kg/m}^3)(2.5\ \text{m/s})(0.50\ \text{m})}{1.307\times 10^{-3}\ \text{kg/m}\cdot\text{s}} = \mathbf{9.56\times 10^5}$$, which is

greater than the critical value of 500. Therefore, **the flow is turbulent**.

When calculating the Froude number, the hydraulic depth should be used rather than the maximum depth or the hydraulic radius. For a non-rectangular channel, hydraulic depth is defined as the ratio of the flow area to top width,

$$y_h = \frac{A_c}{\text{Top width}} = \frac{\pi R^2/2}{2R} = \frac{\pi R}{4} = \frac{\pi(1.0\,\text{m})}{4} = 0.7854\,\text{m}$$

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{2.5\,\text{m/s}}{\sqrt{(9.81\,\text{m/s}^2)(0.7854\,\text{m})}} = 0.901$$, which is lower than 1. Therefore, the flow is **subcritical**.

Discussion If the maximum flow depth were used instead of the hydraulic depth, the result would still be subcritical flow, but this is not always the case.

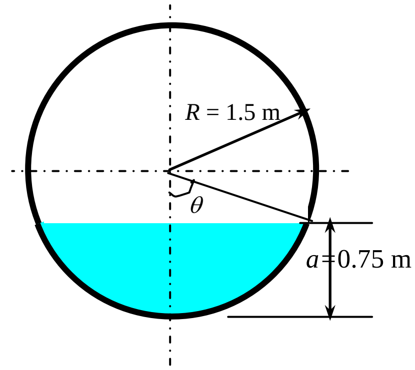
13-19

Solution Water flow in a partially full circular channel is considered. For given water depth and average velocity, the hydraulic radius, Reynolds number, and the flow regime are to be determined.

Assumptions **1** The flow is uniform.

Properties The density and dynamic viscosity of water at 20ºC are ρ = 998.0 kg/m^3 and μ = 1.002×10^{-3} kg/m⋅s.

Analysis From geometric considerations,



$$\cos\theta = \frac{R-a}{R} = \frac{1.5-0.75}{1} = 0.75 \quad \rightarrow \quad \theta = 60° = 60\frac{2\pi}{360} = \frac{\pi}{3}$$

Then the hydraulic radius becomes

$$R_h = \frac{A_c}{p} = \frac{\theta - \sin\theta\cos\theta}{2\theta}R = \frac{\pi/3 - \sin(\pi/3)\cos(\pi/3)}{2\pi/3}(1.5\text{ m}) = \mathbf{0.440\ m}$$

The Reynolds number of the flow is

$$\text{Re} = \frac{\rho V R_h}{\mu} = \frac{(998.0\text{ kg/m}^3)(2\text{ m/s})(0.440\text{ m})}{1.002\times10^{-3}\text{ kg/m}\cdot\text{s}} = \mathbf{8.76\times10^5}$$

which is greater than the critical value of 500. Therefore, the flow is turbulent.

When calculating the Froude number, the hydraulic depth should be used rather than the maximum depth or the hydraulic radius. For a non-rectangular channel, hydraulic depth is defined as the ratio of the flow area to top width,

$$A_c = R^2(\theta - \sin\theta\cos\theta) = (1.5\text{ m})^2[\pi/3 - \sin(\pi/3)\cos(\pi/3)] = 1.382\text{ m}^2$$

$$y_h = \frac{A_c}{\text{Top width}} = \frac{A_c}{2R\sin\theta} = \frac{1.382\text{ m}^2}{2(1.5\text{ m})\sin 60°} = 0.5319\text{ m} \quad \rightarrow \quad \text{Fr} = \frac{V}{\sqrt{gy}} = \frac{2\text{ m/s}}{\sqrt{(9.81\text{m/s}^2)(0.5319\text{ m})}} = 0.876$$

which is lower than 1. Therefore, the flow is **subcritical**.

Specific Energy and the Energy Equation

13-20C

Solution We are to compare the specific energy in two flows – one subcritical and one supercritical.

Analysis A plot of E_s versus y for constant $\dot{V}$ through a rectangular channel of width b reveals that there are two y values corresponding to a fixed value of E_s: one for subcritical flow and one for supercritical flow. Therefore, **the specific energies of water in those two channels can be identical**.

Discussion If the flow is varied (not uniform), however, E_s is not *necessarily* identical in the two channels.

13-21C

Solution We are to define and discuss specific energy.

Analysis The *specific energy* E_s of a fluid flowing in an open channel is the **sum of the pressure and dynamic heads of a fluid**, and is expressed as $\boxed{E_s = y + \frac{V^2}{2g}}$.

Discussion Specific energy is very useful when analyzing varied flows.

13-22C

Solution We are to examine a claim that during steady flow in a wide rectangular channel, the energy line of the flow is parallel to the channel bottom when the frictional losses are negligible.

Analysis **No, the claim is not correct**. The energy line is a distance $E_s = y + V^2/2g$ (total mechanical energy of the fluid) above a horizontal reference datum. When there is no head loss, the energy line is horizontal even when the channel is not. The elevation and velocity heads ($z + y$ and $V^2/2g$) may convert to each other during flow in this case, but their sum remains constant.

Discussion Keep in mind that in real life, there is no such thing as frictionless flow. However, there are situations in which the frictional effects are negligible compared to other effects in the flow.

13-23C

Solution We are to examine a claim that during steady 1-D flow through a wide rectangular channel, the total mechanical energy of the fluid at the free surface is equal to that of the fluid at the channel bottom.

Analysis **Yes, the claim is correct**. During steady one-dimensional flow, the total mechanical energy of a fluid at any point of a cross-section is given by $H = z + y + V^2/2g$.

Discussion The physical elevation of the point under consideration does not appear in the above equation for *H*.

13-24C

Solution We are to express the total mechanical energy in steady 1-D flow in terms of heads.

Analysis The total mechanical energy of a fluid at any point of a cross-section is expressed as $\boxed{H = z + y + V^2/2g}$ where *y* is the flow depth, *z* is the elevation of the channel bottom, and *V* is the average flow velocity. **It is related to the specific energy of the fluid by** $\boxed{H = z + E_s}$.

Discussion Because of irreversible frictional head losses, *H* must decrease in the flow direction in open-channel flow.

13-25C

Solution We are to express the 1-D energy equation for open-channel flow and discuss head loss.

Analysis The one-dimensional energy equation for open channel flow between an upstream section 1 and downstream section 2 is written as $\boxed{z_1 + y_1 + \frac{V_1^2}{2g} = z_2 + y_2 + \frac{V_2^2}{2g} + h_L}$ where *y* is the flow depth, *z* is the elevation of the channel bottom, and *V* is the average flow velocity. The head loss h_L due to frictional effects can be determined from $\boxed{h_L = f\frac{L}{R_h}\frac{V^2}{8g}}$ where *f* is the average friction factor and *L* is the length of channel between sections 1 and 2.

Discussion Head loss is always positive – it can never be negative since this would violate the second law of thermodynamics. Thus, the total mechanical energy must decrease downstream in open-channel flow.

13-26C

Solution We are to examine claims about the minimum value of specific energy.

Analysis The point of minimum specific energy is the critical point, and thus **the first person is correct**.

Discussion The specific energy cannot go below the critical point for a given volume flow rate, as is clear from the plot of specific energy as a function of flow depth.

13-27C

Solution We are to examine a claim about supercritical flow of water in an open channel, namely, that the larger the flow depth, the larger the specific energy.

Analysis **No, the claim is incorrect**. A plot of E_s versus y for constant $\dot{V}$ reveals that the *specific energy decreases as the flow depth increases during supercritical channel flow*.

Discussion This may go against our intuition, since a larger flow depth seems to imply greater energy, but this is not necessarily the case (we cannot always trust our intuition).

13-28C

Solution We are to examine a claim that specific energy remains constant in steady uniform flow.

Analysis The **first person (who claims that specific energy remains constant) is correct** since in uniform flow, the flow depth and the flow velocity, and thus the specific energy, remain constant since $E_s = y + V^2/2g$. The head loss is made up by the decline in elevation (the channel is sloped downward in the flow direction).

Discussion In uniform flow, the flow depth and the average velocity do not change downstream, since the elevation drop exactly overcomes the frictional losses.

13-29C

Solution We are to define and discuss friction slope.

Analysis The *friction slope* is related to head loss h_L, and is defined as $\boxed{S_f = h_L / L}$ where L is the channel length. The **friction slope is equal to the bottom slope when the head loss is equal to the elevation drop**. That is, $S_f = S_0$ when $h_L = z_1 - z_2$.

Discussion Friction slope is a useful concept when analyzing uniform or varied flow in open channels.

13-30

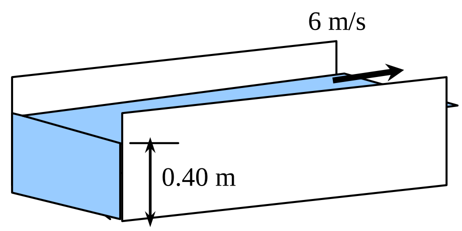
Solution Water flows in a rectangular channel. The critical depth, the alternate depth, and the minimum specific energy are to be determined.

Assumptions The channel is sufficiently wide so that the edge effects are negligible.

Analysis For convenience, we take the channel width to be b = 1 m. Then the volume flow rate and the critical depth for this flow become

$$\dot{V} = VA_c = Vyb = (6\ \text{m/s})(0.4\ \text{m})(1\ \text{m}) = 2.40\ \text{m}^3/\text{s}$$

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(2.40\ \text{m}^3/\text{s})^2}{(9.81\ \text{m/s}^2)(1\ \text{m})^2}\right)^{1/3} = \mathbf{0.837\ m}$$



(b) The flow is *supercritical* since the actual flow depth is y = 0.4 m, and $y < y_c$. The specific energy for given conditions is

$$E_{s1} = y_1 + \frac{\dot{V}^2}{2gb^2 y_1^2} = y_1 + \frac{V^2}{2g} = (0.4\ \text{m}) + \frac{(6\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = 2.23\ \text{m}$$

Then the alternate depth is determined from $E_{s1} = E_{s2}$ to be

$$E_{s2} = y_2 + \frac{\dot{V}^2}{2gb^2 y_2^2} \quad \rightarrow \quad 2.23\ \text{m} = y_2 + \frac{0.240\ \text{m}^3/\text{s}}{y_2^2}$$

Solving for y_2 gives the alternate depth to be y_2 = **2.17 m**. Therefore, if the character of flow is changed from supercritical to subcritical while holding the specific energy constant, the flow depth will rise from 0.4 m to 2.17 m.

(c) the minimum specific energy is

$$E_{s,\min} = y_c + \frac{V_c^2}{2g} = y_c + \frac{gy_c}{2g} = \frac{3}{2}y_c = \frac{3}{2}(0.837\ \text{m}) = \mathbf{1.26\ m}$$

Discussion Note that minimum specific energy is observed when the flow depth is critical.

13-31

Solution Water flows in a rectangular channel. The critical depth, the alternate depth, and whether the flow is subcritical or supercritical are to be determined.

Assumptions The flow is uniform and thus the specific energy is constant.

Analysis (*a*) The critical depth is calculated to be $y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(12\text{ m}^3/\text{s})^2}{(9.81\text{ m/s}^2)(6\text{ m})^2}\right)^{1/3} = \mathbf{0.742 m}$

(*b*) The average flow velocity and the Froude number are

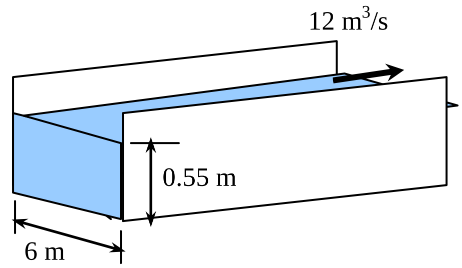
$$V = \frac{\dot{V}}{by} = \frac{12\text{ m}^3/\text{s}}{(6\text{ m})(0.55\text{ m})} = 3.636\text{ m/s} \quad \text{and} \quad \text{Fr}_1 = \frac{V}{\sqrt{gy}} = \frac{3.636\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(0.55\text{ m})}} = 1.565$$

, which is greater than 1.

Therefore, the flow is **supercritical**.

(*c*) Specific energy for this flow is

$$E_{s1} = y_1 + \frac{\dot{V}^2}{2gb^2y_1^2} = (0.55\text{ m}) + \frac{(12\text{ m}^3/\text{s})^2}{2(9.81\text{ m/s}^2)(6\text{ m})^2(0.55\text{ m})^2} = 1.224\text{ m}$$



Then the alternate depth is determined from $E_{s1}=E_{s2}$,

$$E_{s2} = y_2 + \frac{\dot{V}^2}{2gb^2y_2^2} \quad \rightarrow \quad 1.224\text{ m} = y_2 + \frac{(12\text{ m}^3/\text{s})^2}{2(9.81\text{ m/s}^2)(6\text{ m})^2 y_2^2}$$

The alternate depth is calculated to be y_2 = **1.03 m** which is the subcritical depth for the same value of specific energy.

Discussion The depths 0.55 m and 1.03 are alternate depths for the given discharge and specific energy. The flow conditions determine which one is observed.

13-32E

Solution Water flows in a wide rectangular channel. For specified values of flow depth and average velocity, the Froude number, critical depth, and whether the flow is subcritical or supercritical are to be determined.

Assumptions The flow is uniform and thus the specific energy is constant.

Analysis (*a*) The Froude number is $\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{20\text{ ft/s}}{\sqrt{(32.2\text{ ft/s}^2)(1.4\text{ ft})}} = \mathbf{2.98}$

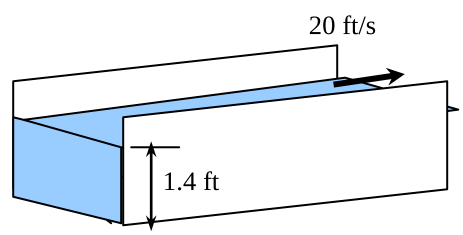
(*b*) The critical depth is calculated to be $y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{V^2y^2b^2}{gb^2}\right)^{1/3} = \left(\frac{(20\text{ ft/s})^2(1.4\text{ ft})^2}{(32.2\text{ ft/s}^2)}\right)^{1/3} = \mathbf{2.90 ft}$

(*c*) The flow is **supercritical** since Fr > 1.

For the case of *y* = 0.2 ft:
Replacing 1.4 ft in above calculations by 0.2 ft gives

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{20\text{ ft/s}}{\sqrt{(32.2\text{ ft/s}^2)(0.2\text{ ft})}} = \mathbf{7.88}$$

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{V^2y^2b^2}{gb^2}\right)^{1/3} = \left(\frac{(20\text{ ft/s})^2(0.2\text{ ft})^2}{(32.2\text{ ft/s}^2)}\right)^{1/3} = \mathbf{0.792 ft}$$



The flow is supercritical in this case also since Fr > 1.

Discussion Note that the value of critical depth depends on flow rate, and it decreases as the flow rate decreases.

13-33E

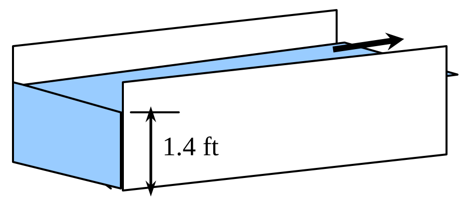
Solution Water flows in a wide rectangular channel. For specified values of flow depth and average velocity, the Froude number, critical depth, and whether the flow is subcritical or supercritical are to be determined.

Assumptions The flow is uniform and thus the specific energy is constant.

Analysis (*a*) The Froude number is $\mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{10 \text{ ft/s}}{\sqrt{(32.2 \text{ ft/s}^2)(1.4 \text{ ft})}} = \mathbf{1.49}$

(*b*) The critical depth is calculated to be $y_c = \left(\dfrac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\dfrac{V^2y^2b^2}{gb^2}\right)^{1/3} = \left(\dfrac{(10 \text{ ft/s})^2(1.4 \text{ ft})^2}{(32.2 \text{ ft/s}^2)}\right)^{1/3} = \mathbf{1.83\ ft}$

(*c*) The flow is **supercritical** since Fr > 1.



For the case of *y* = 0.2 ft:
Replacing 0.8 ft in above calculations by 0.2 ft gives

$$\mathrm{Fr} = \frac{V}{\sqrt{gy}} = \frac{10 \text{ ft/s}}{\sqrt{(32.2 \text{ ft/s}^2)(0.2 \text{ ft})}} = \mathbf{3.94}$$

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{V^2y^2b^2}{gb^2}\right)^{1/3} = \left(\frac{(14 \text{ ft/s})^2(0.2 \text{ ft})^2}{(32.2 \text{ ft/s}^2)}\right)^{1/3} = \mathbf{0.50\ ft}$$

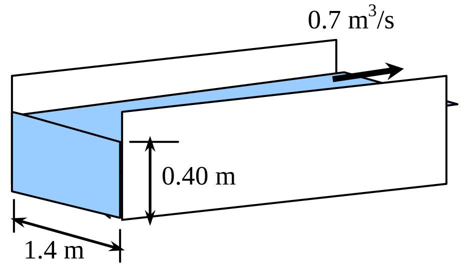
The flow is supercritical in this case also since Fr > 1.

Discussion Note that the value of critical depth depends on flow rate, and it decreases as the flow rate decreases.

13-34

Solution Water flow in a rectangular channel is considered. The character of flow, the flow velocity, and the alternate depth are to be determined.

Assumptions The specific energy is constant.



Analysis The average flow velocity is determined from

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{yb} = \frac{0.7\ \text{m}^3/\text{s}}{(0.40\ \text{m})(1.4\ \text{m})} = \mathbf{1.25\ m/s}$$

The critical depth for this flow is

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(0.7\ \ \text{m}^3/\text{s})^2}{(9.81\ \text{m/s}^2)(1.4\ \text{m})^2}\right)^{1/3} = 0.294\ \text{m}$$

Therefore, the flow is *supercritical* since the actual flow depth is y = 0.40 m, and $y < y_c$. The specific energy for given conditions is

$$E_{s1} = y_1 + \frac{\dot{V}^2}{2gb^2 y_1^2} = (0.40\ \text{m}) + \frac{(0.7\ \text{m}^3/\text{s})^2}{2(9.81\ \text{m/s}^2)(1.4\ \text{m})^2(0.40\ \text{m})^2} = 0.4796\ \text{m}$$

Then the alternate depth is determined from $E_{s1} = E_{s2}$ to be

$$E_{s2} = y_2 + \frac{\dot{V}^2}{2gb^2 y_2^2} \quad \rightarrow \quad 0.4796\ \text{m} = y_2 + \frac{(0.7\ \text{m}^3/\text{s})^2}{2(9.81\ \text{m/s}^2)(1.4\ \text{m})^2\, y_2^2}$$

Solving for y_2 gives the alternate depth to be y_2 **= 0.223 m**. There are three roots of this equation; one for subcritical, one for supercritical and third one as a negative root. Therefore, if the character of flow is changed from supercritical to subcritical while holding the specific energy constant, the flow depth will drop from 0.40 m to 0.223 m.

Discussion Two alternate depths show two possible flow conditions for a given specific energy. If the energy is not the minimum specific energy, there are two water depths corresponding to subcritical and supercritical states of flow. As an example, these two depths may be observed before and after a sluice gate as alternate depths, if the losses are disregarded.

13-35

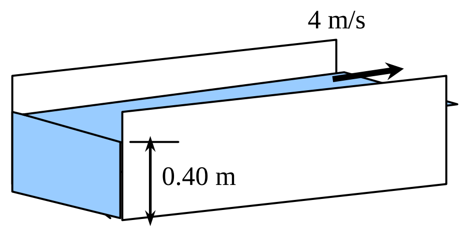
Solution Water flows in a rectangular channel. The specific energy and whether the flow is subcritical or supercritical are to be determined.

Assumptions The flow is uniform and thus the specific energy is constant.

Analysis For convenience, we take the channel width to be b = 1 m. Then the volume flow rate and the critical depth for this flow become

$$\dot{V} = VA_c = Vyb = (4\ \text{m/s})(0.4\text{m})(1\ \text{m}) = 1.60\ \text{m}^3/\text{s}$$

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(1.60\ \text{m}^3/\text{s})^2}{(9.81\ \text{m/s}^2)(1\ \text{m})^2}\right)^{1/3} = \mathbf{0.639\ m}$$



The flow is **supercritical** since the actual flow depth is y = 0.4 m, and $y < y_c$. The specific energy for given conditions is

$$E_{s1} = y_1 + \frac{\dot{V}^2}{2gb^2 y_1^2} = y_1 + \frac{V^2}{2g} = (0.4\ \text{m}) + \frac{(4\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = \mathbf{1.22\ m}$$

Discussion Note that the flow may also exist as subcritical flow at the same value of specific energy,

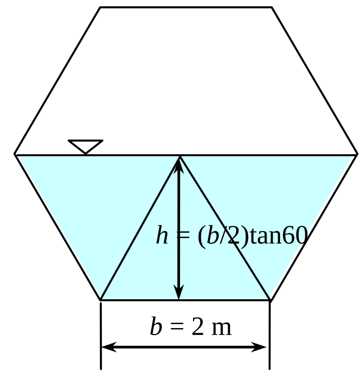
---

13-36

Solution Water flows uniformly through a half-full hexagon channel. For a given flow rate, the average velocity and whether the flow is subcritical or supercritical are to be determined.

Assumptions The flow is uniform.

Analysis (*a*) The flow area is determined from geometric considerations to be



$$A_c = \frac{(b+2b)}{2}\frac{b}{2}\tan 60° = \frac{(2+2\times 2)\text{ m}}{2}\frac{2\text{ m}}{2}\tan 60° = 5.196\text{ m}^2$$

Then the average velocity becomes

$$V = \frac{\dot{V}}{A_c} = \frac{60\text{ m}^3/\text{s}}{5.196\text{ m}^2} = 11.55\text{ m/s} \cong \mathbf{11.6\ m/s}$$

(*b*) When calculating the Froude number, the hydraulic depth should be used rather than the maximum depth or the hydraulic radius. For a non-rectangular channel, hydraulic depth is defined as the ratio of the flow area to top width,

$$y = y_h = \frac{A_c}{\text{Top width}} = \frac{A_c}{2b} = \frac{5.196\text{ m}}{2\times 2\text{ m}} = 1.299\text{ m}$$

Then the Froude number becomes

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{11.55\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(1.299\text{ m})}} = 3.23$$

which is greater than 1. Therefore, the flow is **supercritical**.

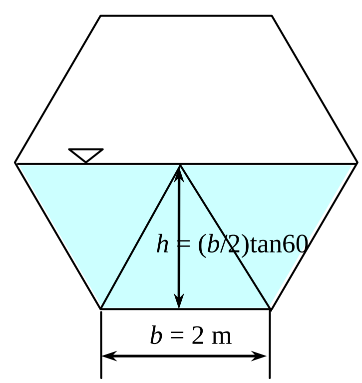
Discussion The analysis is approximate since the edge effects are significant here compared to a wide rectangular channel, and thus the results should be interpreted accordingly.

13-37

Solution Water flows uniformly through a half-full hexagon channel. For a given flow rate, the average velocity and whether the flow is subcritical or supercritical are to be determined.

Assumptions The flow is uniform.

Analysis The flow area is determined from geometric considerations to be



$$A_c = \frac{(b+2b)}{2}\frac{b}{2}\tan 60° = \frac{(2+2\times 2)\text{ m}}{2}\frac{2\text{ m}}{2}\tan 60° = 5.196\text{ m}^2$$

Then the average velocity becomes

$$V = \frac{\dot{V}}{A_c} = \frac{30\text{ m}^3/\text{s}}{5.196\text{ m}^2} = \mathbf{5.77\ m/s}$$

When calculating the Froude number, the hydraulic depth should be used rather than the maximum depth or the hydraulic radius. For a non-rectangular channel, hydraulic depth is defined as the ratio of the flow area to top width,

$$y = y_h = \frac{A_c}{\text{Top width}} = \frac{A_c}{2b} = \frac{5.196\text{ m}^2}{2\times 2\text{ m}} = 1.299\text{ m}$$

Then the Froude number becomes

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{5.77\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(1.299\text{ m})}} = 1.62$$

which is greater than 1. Therefore, the flow is **supercritical**.

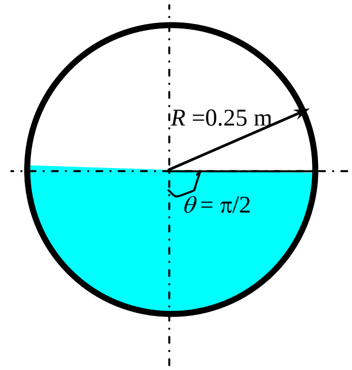
Discussion The analysis is approximate since the edge effects are significant here compared to a wide rectangular channel, and thus the results should be interpreted accordingly.

13-38

Solution Water flows uniformly through a half-full circular steel channel. For a given average velocity, the volume flow rate, critical slope, and the critical depth are to be determined.

Assumptions The flow is uniform.

Analysis The volume flow rate is determined from



$$\dot{V} = VA_c = V\frac{\pi R^2}{2} = (2.8\,\mathrm{m/s})\frac{\pi(0.25\,\mathrm{m})^2}{2} = \mathbf{0.275\,m^3/s}$$

When calculating the Froude number, the hydraulic depth should be used rather than the maximum depth or the hydraulic radius. For a non-rectangular channel, hydraulic depth is defined as the ratio of the flow area to top width,

$$y_h = \frac{A_c}{\text{Top width}} = \frac{\pi R^2/2}{2R} = \frac{\pi R}{4} = \frac{\pi(0.25\,\mathrm{m})}{4} = 0.1963\,\mathrm{m}$$

$$\mathrm{Fr} = \frac{V}{\sqrt{gy}} = \frac{2.8\,\mathrm{m/s}}{\sqrt{(9.81\,\mathrm{m/s^2})(0.1963\,\mathrm{m})}} = 2.02$$

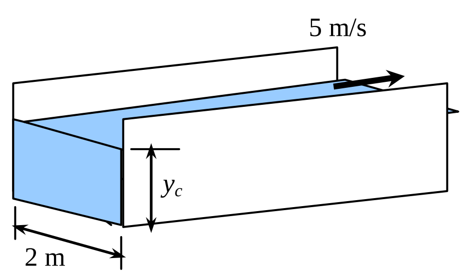
which is greater than 1. Therefore, the flow is **supercritical**.

Discussion Note that if the maximum flow depth were used instead of the hydraulic depth, the result could be different, especially when the Froude number is close to 1.

13-39

Solution Critical flow of water in a rectangular channel is considered. For a specified average velocity, the flow rate of water is to be determined.

Assumptions The flow is uniform and thus the specific energy is constant.



Analysis The Froude number must be unity since the flow is critical, and thus $\mathrm{Fr} = V/\sqrt{gy} = 1$. Therefore,

$$y = y_c = \frac{V^2}{g} = \frac{(5\,\mathrm{m/s})^2}{9.81\,\mathrm{m/s^2}} = 2.548\,\mathrm{m}$$

Then the flow rate becomes

$$\dot{V} = VA_c = Vby = (5\,\mathrm{m/s})(2\,\mathrm{m})(2.548\,\mathrm{m}) = \mathbf{25.5\,m^3/s}$$

Discussion Critical flow is not a stable type of flow and can be observed for short intervals. Occurrence of critical depth is important as boundary condition most of the time. For example it can be used as a flow rate computation mechanism for a channel ending with a drawdown.

Uniform Flow and Best Hydraulic Cross Sections

13-40C

Solution We are to discuss when flow in an open channel is uniform, and how it remains uniform.

Analysis Flow in a channel is called *uniform flow* if the **flow depth (and thus the average flow velocity) remains constant**. **The flow remains uniform as long as the slope, cross-section, and the surface roughness of the channel remain unchanged**.

Discussion Uniform flow in open-channel flow is somewhat analogous to fully developed pipe flow in internal flow.

13-41C

Solution We are to determine which cross section is better – one with a small or large hydraulic radius.

Analysis The best hydraulic cross-section for an open channel is the one with the **maximum hydraulic radius**, or equivalently, the one with the minimum wetted perimeter for a specified cross-sectional area.

Discussion Frictional losses occur at the wetted perimeter walls of the channel, so it makes sense to minimize the wetted perimeter in order to minimize the frictional losses.

13-42C

Solution We are to determine which cross section shape is best for an open channel.

Analysis The best hydraulic cross-section for an open channel is a (*a*) **circular** one.

Discussion Circular channels are often more difficult to construct, however, so they are often not used in practice.

13-43C

Solution We are to determine the best hydraulic cross section for a rectangular channel.

Analysis The best hydraulic cross section for a rectangular channel is one whose fluid height is (*a*) **half the channel width**.

Discussion It turns out that for this case, the wetted perimeter, and thus the frictional losses, are smallest.

13-44C

Solution We are to determine the best hydraulic cross section for a trapezoidal channel.

Analysis The best hydraulic cross section for a trapezoidal channel of base width b is (*a*) one for which **the length of the side edge of the flow section is *b***.

Discussion It turns out that for this case, the wetted perimeter, and thus the frictional losses, are smallest.

13-45C

Solution We are to examine a claim that head loss can be determined by multiplying bottom slope by channel length.

Analysis **Yes, the claim is correct**. The head loss in uniform flow is $h_L = S_0 L$ since the head loss must equal elevation loss.

Discussion In uniform flow, frictional head losses are exactly balanced by elevation loss, which is directly proportional to bottom slope.

13-46C

Solution We are to discuss how flow depth changes when the bottom slope is increased.

Analysis The flow depth **decreases** when the bottom slope is increased.

Discussion You can think of it in simple terms this way: As the slope increases, the liquid flows faster, and faster flow requires lower depth.

13-47

Solution We are to determine how the flow rate changes when the Manning coefficient doubles.

Analysis The flow rate in uniform flow is given as $\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2}$, and thus the flow rate is inversely proportional to the Manning coefficient. Therefore, if the Manning coefficient doubles as a result of some algae growth on surfaces while the flow cross section remains constant, the flow rate will (*d*) **decrease by half**.

Discussion In an actual case, the cross section may also change due to flow depth changes as well.

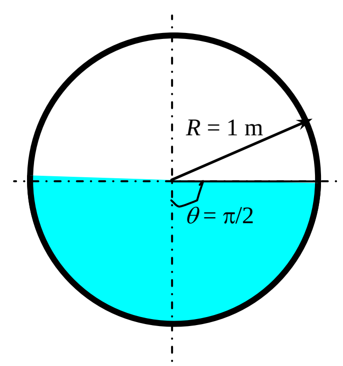
13-48

Solution Water flows uniformly half-full in a circular finished-concrete channel. For a given bottom slope, the flow rate is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties Manning coefficient for an open channel of finished concrete is n = 0.012 (Table 13-1).

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are



$$A_c = \frac{\pi R^2}{2} = \frac{\pi (1\,\text{m})^2}{2} = 1.571\,\text{m}^2$$

$$p = \frac{2\pi R}{2} = \frac{2\pi (1\,\text{m})}{2} = 3.142\,\text{m}$$

$$R_h = \frac{A_c}{P} = \frac{\pi R^2/2}{\pi R} = \frac{R}{2} = \frac{1\,\text{m}}{2} = 0.50\ \text{m}$$

Then the flow rate can be determined from Manning's equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{1\,\text{m}^{1/3}/s}{0.012}(1.571\,\text{m}^2)(0.50\,\text{m})^{2/3}(1.5/1000)^{1/2} = \mathbf{3.19\ m^3/s}$$

Discussion Note that the flow rate in a given channel is a strong function of the bottom slope.

13-49

Solution The flow of water in a trapezoidal finished-concrete channel is considered. For a given flow depth and bottom slope, the flow rate is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

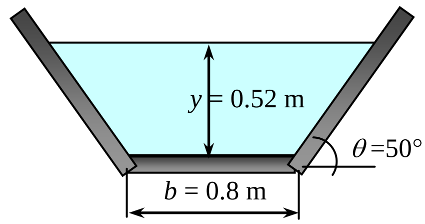
Properties Manning coefficient for an open channel of finished concrete is n = 0.012 (Table 13-1).

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are

$$A_c = y\left(b + \frac{y}{\tan\theta}\right) = (0.52\,\text{m})\left(0.80\,\text{m} + \frac{0.52\,\text{m}}{\tan 50°}\right) = 0.6429\ \text{m}^2$$

$$p = b + \frac{2y}{\sin\theta} = 0.8\,\text{m} + \frac{2(0.52\,\text{m})}{\sin 50°} = 2.158\,\text{m}$$

$$R_h = \frac{A_c}{p} = \frac{0.6429\,\text{m}^2}{2.158\,\text{m}} = 0.2980\ \text{m}$$



Bottom slope of the channel is

$$S_0 = \tan 0.4^\circ = 0.006981$$

Then the flow rate can be determined from Manning's equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{1\,\text{m}^{1/3}/s}{0.012}(0.6429\,\text{m}^2)(0.2980\,\text{m})^{2/3}(0.006981)^{1/2} = 1.997\,\text{m}^3/s \cong \mathbf{2.00\ m^3/s}$$

Discussion Note that the flow rate in a given channel is a strong function of the bottom slope.

13-50E

Solution Water is to be transported uniformly in a full semi-circular unfinished-concrete channel. For a specified flow rate, the elevation difference across the channel is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

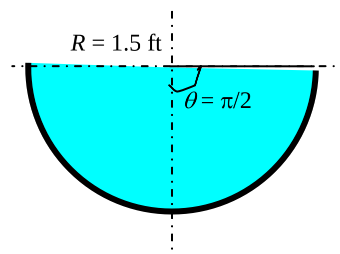
Properties Manning coefficient for an open channel of unfinished concrete is n = 0.014 (Table 13-1).

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are

$$A_c = \frac{\pi R^2}{2} = \frac{\pi(1.5\,\text{ft})^2}{2} = 3.534\,\text{ft}^2$$

$$p = \frac{2\pi R}{2} = \frac{2\pi(1.5\,\text{ft})}{2} = 4.712\,\text{ft}$$

$$R_h = \frac{A_c}{P} = \frac{\pi R^2/2}{\pi R} = \frac{R}{2} = \frac{1.5\,\text{ft}}{2} = 0.75\,\text{ft}$$



Substituting the given quantities into Manning's equation,

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} \rightarrow \quad 90\,\text{ft}^3/\text{s} = \frac{1.486\,\text{ft}^{1/3}/s}{0.014}(3.534\,\text{ft}^2)(0.75\,\text{ft})^{2/3} S_0^{1/2}$$

It gives the slope to be S_0 = 0.08448. Therefore, the *elevation difference* Δz across a pipe length of L = 1 mile = 5280 ft must be

$$\Delta z = S_0 L = 0.08448(5280\,\text{ft}) = \mathbf{446\,ft}$$

Discussion Note that when transporting water through a region of fixed elevation drop, the only way to increase the flow rate is to use a channel with a larger cross-section.

13-51

Solution We are to discuss the constants and coefficients in the Manning equation.

Analysis The value of the factor a in SI units is $\boldsymbol{a}$ **= 1 m$^{1/3}$/s**. Combining the relations $C = \sqrt{8g/f}$ and $C = \frac{a}{n}R_h^{1/6}$ and solving them for n gives the desired relation to be $\boxed{n = \frac{a}{\sqrt{8g/f}}R_h^{1/6}}$. In practice, n is usually determined experimentally.

Discussion The value of n varies greatly with surface roughness.

13-52

Solution It is to be shown that for uniform critical flow, the general critical slope relation $S_c = \frac{gn^2 y_c}{a^2 R_h^{4/3}}$ reduces to $S_c = \frac{gn^2}{a^2 y_c^{1/3}}$ for film flow with $b >> y_c$.

Analysis For critical flow, the flow depth is $y = y_c$. For film flow, the hydraulic radius is $R_h = y = y_c$. Substituting into the critical slope relation gives the desired result, $\boxed{S_c = \frac{gn^2 y_c}{a^2 R_h^{4/3}} = \frac{gn^2 y_c}{a^2 y_c^{4/3}} = \frac{gn^2}{a^2 y_c^{1/3}}}$.

Discussion The reduced equation is valid for film flow only – be careful not to apply it to channels of other shapes.

13-53

Solution Water is to be transported uniformly in a trapezoidal asphalt-lined channel. For a specified flow rate, the required elevation drop per km channel length is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

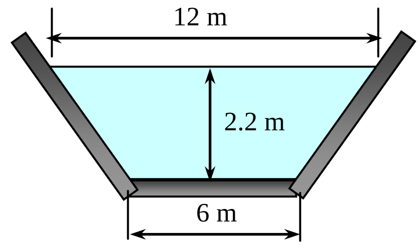
Properties Manning coefficient for an asphalt-lined open channel is n = 0.016 (Table 13-1).

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are

$$A_c = \frac{12\,\text{m} + 6\,\text{m}}{2}(2.2\,\text{m}) = 19.8\,\text{m}^2$$

$$p = 6\,\text{m} + 2\sqrt{(2.2\,\text{m})^2 + (3\,\text{m})^2} = 13.4404\,\text{m}$$

$$R_h = \frac{A_c}{p} = \frac{19.8\ \text{m}^2}{13.4404\ \text{m}} = 1.4732\,\text{m}$$



Substituting the given quantities into Manning's equation,

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} \quad \rightarrow \quad S_0 = \left(\frac{\dot{V} n}{a A_c R_h^{2/3}} \right)^2 = \left(\frac{\left(120\,\text{m}^3/\text{s}\right)\left(0.016\right)}{\left(1\ \text{m}^{1/3}/s\right)(19.8\,\text{m}^2)(1.4732\ \text{m})^{2/3}} \right)^2 = 0.0056097$$

Therefore, the *elevation drop* Δz across a pipe length of L = 1 km must be

$$\Delta z = S_0 L = 0.0056097(1000\,\text{m}) = \mathbf{5.61\ m}$$

Discussion Note that when transporting water through a region of fixed elevation drop, the only way to increase the flow rate is to use a channel with a larger cross-section.

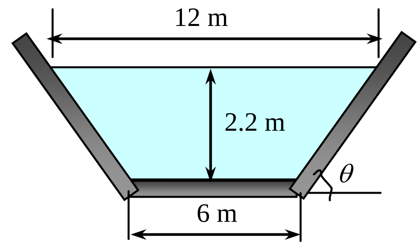
13-54

Solution The flow of water through the trapezoidal asphalt-lined channel in the previous problem is reconsidered. The maximum flow rate corresponding to a given maximum channel height is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Analysis We denote the flow conditions in the previous problem by subscript 1 and the conditions for the maximum case in this problem by subscript 2. Using the Manning's equation $\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2}$ and noting that the Manning coefficient and the channel slope remain constant, the flow rate in case 2 can be expressed in terms of flow rate in case 1 as

$$\frac{\dot{V}_2}{\dot{V}_1} = \frac{(a/n)A_{c2}R_{h2}^{2/3}}{(a/n)A_{c1}R_{h1}^{2/3}} \quad \rightarrow \quad \dot{V}_2 = \frac{A_{c2}}{A_{c1}}\left(\frac{R_{h2}}{R_{h1}}\right)^{2/3}\dot{V}_1$$

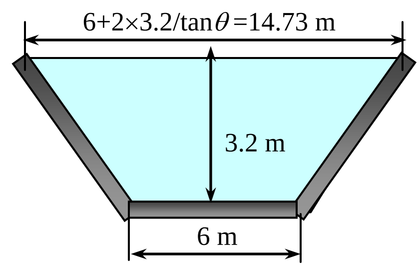


The trapezoid angle is $\tan\theta = 2.2/3 = 0.733 \rightarrow \theta = 2.2/3 = 36.25°$.
From geometric considerations,

$$A_{c1} = \frac{12\,\text{m} + 6\,\text{m}}{2}(2.2\,\text{m}) = 19.8\,\text{m}^2$$

$$p_1 = (6\,\text{m}) + 2\sqrt{(2.2\,\text{m})^2 + (3\,\text{m})^2} = 13.44\,\text{m}$$

$$R_{h1} = \frac{A_{c1}}{p_1} = \frac{19.8\,\text{m}^2}{13.44\,\text{m}} = 1.473\,\text{m}$$



and

$$A_{c2} = \frac{14.73\,\text{m} + 6\,\text{m}}{2}(3.2\,\text{m}) = 33.17\,\text{m}^2$$

$$p_2 = (6\,\text{m}) + 2\sqrt{(3.2\ \text{m})^2 + (14.73 - 6)/2\,\text{m})^2} = 16.82\,\text{m}$$

$$R_{h2} = \frac{A_{c2}}{p_2} = \frac{33.17\,\text{m}^2}{16.82\,\text{m}} = 1.972\,\text{m}$$

Substituting,

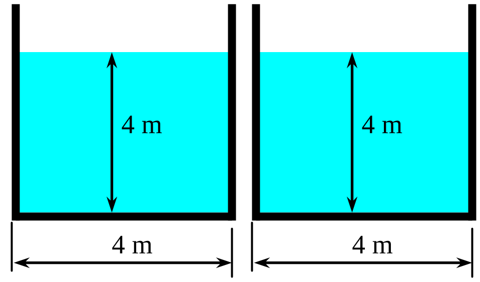
$$\dot{V}_2 = \frac{A_{c2}}{A_{c1}}\left(\frac{R_{h2}}{R_{h1}}\right)^{2/3}\dot{V}_1 = \frac{33.17\,\text{m}^2}{19.8\,\text{m}^2}\left(\frac{1.972\,\text{m}}{1.473\,\text{m}}\right)^{2/3}(120\,\text{m}^3/\text{s}) = \mathbf{244\,m^3/s}$$

Discussion Note that a 45% increase in flow depth results in a 103% increase in flow rate.

13-55

Solution The flow of water through two identical channels with square flow sections is considered. The percent increase in flow rate as a result of combining the two channels while the flow depth remains constant is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.



Analysis We denote the flow conditions for two separate channels by subscript 1 and the conditions for the combined wide channel by subscript 2. Using the Manning's equation $\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2}$ and noting that the Manning coefficient, channel slope, and the flow area A_c remain constant, the flow rate in case 2 can be expressed in terms of flow rate in case 1 as

$$\frac{\dot{V}_2}{\dot{V}_1} = \frac{(a/n)A_{c2}R_{h2}^{2/3}}{(a/n)A_{c1}R_{h1}^{2/3}} = \left(\frac{R_{h2}}{R_{h1}}\right)^{2/3} = \left(\frac{A_{c2}/p_2}{A_{c1}/p_1}\right)^{2/3} = \left(\frac{p_1}{p_2}\right)^{2/3}$$

where p is the wetted perimeter. Substituting,

$$\frac{\dot{V}_2}{\dot{V}_1} = \left(\frac{p_2}{p_2}\right)^{2/3} = \left(\frac{6\times 4\,\text{m}}{4\times 4\,\text{m}}\right)^{2/3} = \left(\frac{3}{2}\right)^{2/3} = \mathbf{1.31} \qquad \textbf{(31\% increase)}$$

Discussion This is a very significant increase, and shows the importance of eliminating unnecessary surfaces in flow systems, including pipe flow.

13-56

Solution The flow of water in a V-shaped cast iron channel is considered. For a given flow depth and bottom slope, the flow rate is to be determined.

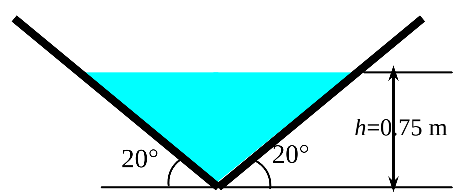
Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties Manning coefficient for an open channel of cast iron is $n = 0.013$ (Table 13-1).

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are

$$A_c = \frac{2h\times h}{2} = h^2 = (0.75\,\text{m})^2 = 0.5625\,\text{m}^2 \qquad p = 2h/\sin\theta = 2(0.75\,\text{m})/\sin 20° = 4.386\,\text{m}$$

$$R_h = \frac{A_c}{p} = \frac{0.5625\,\text{m}^2}{4.386\,\text{m}} = 0.1283\,\text{m}$$



The bottom slope of the channel is

$$S_0 = \tan 0.5° = 0.008727$$

Then the flow rate is determined from Manning's equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{1\,\text{m}^{1/3}/\text{s}}{0.013}(0.5625\,\text{m}^2)(0.1283\,\text{m})^{2/3}(0.008727)^{1/2} = \mathbf{1.03\,m^3/s}$$

Discussion Note that the flow rate in a given channel is a strong function of the bottom slope.

13-57E

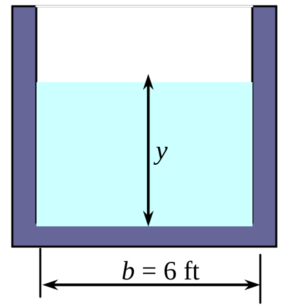
Solution The flow of water in a rectangular cast iron channel is considered. For given flow rate and bottom slope, the flow depth is to be determined.

Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The roughness coefficient is constant.

Properties Manning coefficient for a cast iron open channel is n = 0.013 (Table 13-1).

Analysis From the geometry, the flow area, wetted perimeter, and hydraulic radius are

$$A_c = by = (6\text{ ft})\,y = 6y \qquad p = (6\text{ ft}) + 2y = 6 + 2y \qquad R_h = \frac{A_c}{p} = \frac{6y}{6+2y}$$



The channel bottom slope is S_0 = 1.5/1000 = 0.0015.
Substituting the given quantities into Manning's equation,

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} \quad \rightarrow \quad 70\text{ ft}^3/\text{s} = \frac{1.486\text{ ft}^{1/3}/s}{0.013}(6y)\left(\frac{6y}{6+2y}\right)^{2/3}(0.0015)^{1/2}$$

Solution of the above equation gives the flow depth to be h **= 2.24 ft**.

Discussion Non-linear equations frequently arise in the solution of open channel flow problems. They are best handled by equation solvers such as EES.

13-58

Solution Water is to be transported uniformly in a clean-earth trapezoidal channel. For a specified flow rate, the required elevation drop per km channel length is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

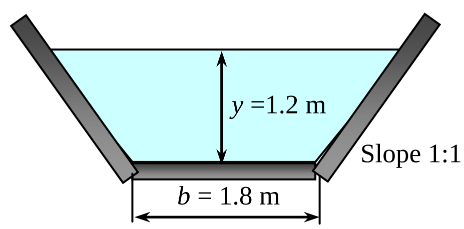
Properties Manning coefficient for the clean-earth lined open channel is n = 0.022 (Table 13-1).

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are

$$A_c = \frac{(1.8+1.8+2.4)\text{ m}}{2}(1.2\text{ m}) = 3.6\text{ m}^2$$

$$p = (1.8\text{ m}) + 2\sqrt{(1.2\text{ m})^2 + (1.2\text{ m})^2} = 5.194\text{ m}$$

$$R_h = \frac{A_c}{p} = \frac{3.6\text{ m}^2}{5.194\text{ m}} = 0.6931\text{ m}$$



Substituting the given quantities into Manning's equation,

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} \quad \rightarrow \quad 8\text{ m}^3/\text{s} = \frac{1\text{ m}^{1/3}/s}{0.022}(3.6\text{ m}^2)(0.6931\text{ m})^{2/3} S_0^{1/2}$$

It gives the slope to be S_0 = 0.003897. Therefore, the *elevation drop* Δz across a pipe length of L = 1 km must be

$$\Delta z = S_0 L = 0.003897(1000\text{ m}) = \mathbf{3.90\text{ m}}$$

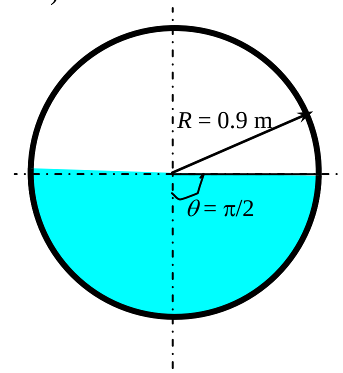
Discussion Note that when transporting water through a region of fixed elevation drop, the only way to increase the flow rate is to use a channel with a larger cross-section.

13-59

Solution A water draining system consists of three circular channels, two of which draining into the third one. If all channels are to run half-full, the diameter of the third channel is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel. **4** Losses at the junction are negligible.

Properties The Manning coefficient for asphalt lined open channels is n = 0.016 (Table 13-1).



Analysis The flow area, wetted perimeter, and hydraulic radius of the two pipes upstream are

$$A_c = \frac{\pi R^2}{2} = \frac{\pi(0.9\,\text{m})^2}{2} = 1.272\,\text{m}^2 \qquad p = \frac{2\pi R}{2} = \frac{2\pi(0.9\,\text{m})}{2} = 2.827\,\text{m}$$

$$R_h = \frac{A_c}{P} = \frac{\pi R^2/2}{\pi R} = \frac{R}{2} = \frac{0.9\,\text{m}}{2} = 0.45\ \text{m}$$

Then the flow rate through the 2 pipes becomes, from Manning's equation,

$$\dot{V} = 2\frac{a}{n}A_c R_h^{2/3} S_0^{1/2} = 2\frac{1\,\text{m}^{1/3}/s}{0.016}(1.272\,\text{m}^2)(0.45\,\text{m})^{2/3}(0.0025)^{1/2} = 4.669\,\text{m}^3/\text{s}$$

The third channel is half-full, and the flow rate through it remains the same. Noting that the flow area is $\pi R^2/2$ and the hydraulic radius is $R/2$, we have

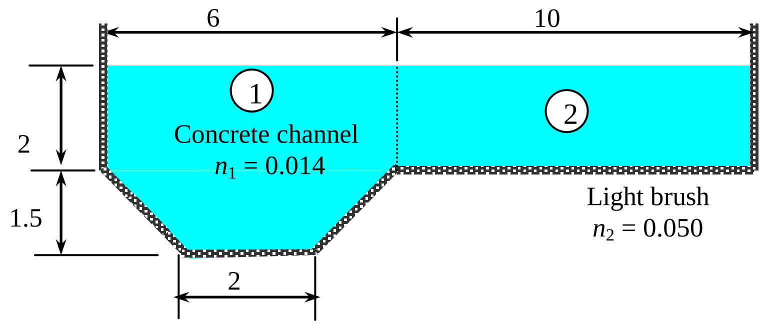
$$4.669\,\text{m}^3/\text{s} = \frac{1\,\text{m}^{1/3}/s}{0.016}(\pi R^2/2\,\text{m}^2)(R/2\,\text{m})^{2/3}(0.0025)^{1/2}$$

Solving for R gives R = 1.167 m. Therefore, the diameter of the third channel is D_3 = **2.33 m** .

Discussion Note that if the channel diameter were larger, the channel would have been less than half full.

13-60

Solution Water is flowing through a channel with nonuniform surface properties. The flow rate and the effective Manning coefficient are to be determined.



Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The Manning coefficients do not vary along the channel.

Analysis The channel involves two parts with different roughness, and thus it is appropriate to divide the channel into two subsections. The flow rate for each subsection can be determined from the Manning equation, and the total flow rate can be determined by adding them up.

The flow area, perimeter, and hydraulic radius for each subsection and the entire channel are:

Subsection 1: $A_{c1} = 18\text{ m}^2$, $\quad p_1 = 9\text{ m}$, $\quad R_{h1} = \dfrac{A_{c1}}{p_1} = \dfrac{18\text{ m}^2}{9\text{ m}} = 2.00\text{ m}$

Subsection 2: $A_{c2} = 20\text{ m}^2$, $\quad p_2 = 12\text{ m}$, $\quad R_{h2} = \dfrac{A_{c2}}{p_2} = \dfrac{20\text{ m}^2}{12\text{ m}} = 1.67\text{ m}$

Entire channel: $A_c = 38\text{ m}^2$, $\quad p = 21\text{ m}$, $\quad R_h = \dfrac{A_c}{p} = \dfrac{38\text{ m}^2}{21\text{ m}} = 1.81\text{ m}$

Applying the Manning equation to each subsection, the total flow rate through the channel is determined to be

$$\dot{V} = \dot{V}_1 + \dot{V}_2 = \frac{a}{n_1} A_1 R_1^{2/3} S_0^{1/2} + \frac{a}{n_1} A_1 R_1^{2/3} S_0^{1/2} = (1\text{ m}^{1/3}/\text{s}) \left(\frac{(18\text{ m}^2)\,(2\text{ m})^{2/3}}{0.014} + \frac{(20\text{ m}^2)\,(1.67\text{ m})^{2/3}}{0.05} \right) (0.002)^{1/2} = \mathbf{116\ m^3/s}$$

Knowing the total flow rate, the effective Manning coefficient for the entire channel can be determined from the Manning equation to be

$$n_{\text{eff}} = \frac{a A_c R_h^{2/3} S_0^{1/2}}{\dot{V}} = \frac{(1\text{ m}^{1/3}/\text{s})(38\text{ m}^2)(1.81\text{ m})^{2/3}(0.002)^{1/2}}{116\text{ m}^3/\text{s}} = \mathbf{0.0217}$$

Discussion The effective Manning coefficient n_{eff} lies between the two n values as expected. The weighted average of the Manning coefficient of the channel is $n_{\text{ave}} = (n_1 p_1 + n_2 p_2)/p$ = 0.035, which is quite different than n_{eff}. Therefore, using a weighted average Manning coefficient for the entire channel may be tempting, but it would not be accurate.

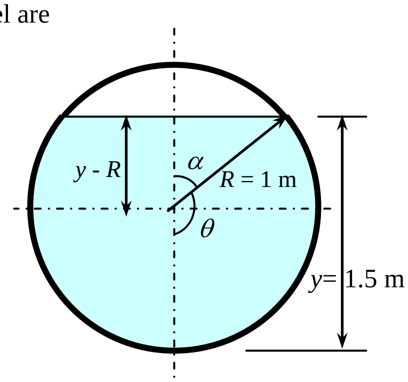
13-61

Solution The flow of water in a circular open channel is considered. For given flow depth and flow rate, the elevation drop per km length is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties The Manning coefficient for the steel channel is given to be n = 0.012.

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are



$$\cos\alpha = \frac{y-R}{R} = \frac{1.5-1}{1} = 0.5 \quad \rightarrow \quad \alpha = 60° = 60\frac{2\pi}{360} = \frac{\pi}{3}$$

$$\theta = \pi - \alpha = \pi - \frac{\pi}{3} = \frac{2\pi}{3} = 120°$$

$$A_c = R^2(\theta - \sin\theta\cos\theta) = (1\,\text{m})^2[2\pi/3 - \sin(2\pi/3)\cos(2\pi/3)] = 2.527\,\text{m}^2$$

$$R_h = \frac{A_c}{p} = \frac{\theta - \sin\theta\cos\theta}{2\theta}R = \frac{2\pi/3 - \sin(2\pi/3)\cos(2\pi/3)}{2\times 2\pi/3}(1\,\text{m}) = 0.6034\,\text{m}$$

Substituting the given quantities into Manning's equation,

$$\dot{V} = \frac{a}{n}A_c R_h^{2/3} S_0^{1/2} \quad \rightarrow \quad 12\,\text{m}^3/\text{s} = \frac{1\,\text{m}^{1/3}/s}{0.012}(2.527\,\text{m}^2)(0.6034\,\text{m})^{2/3} S_0^{1/2}$$

It gives the slope to be S_0 = 0.00637. Therefore, the *elevation drop* Δz across a pipe length of L = 1 km must be

$$\Delta z = S_0 L = 0.00637(1000\,\text{m}) = \mathbf{6.37\,m}$$

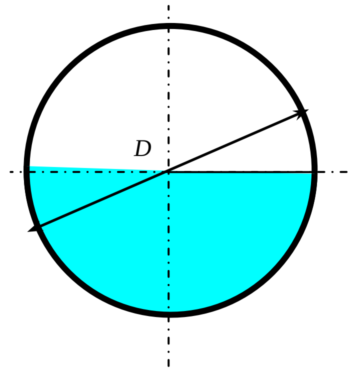
Discussion Note that when transporting water through a region of fixed elevation drop, the only way to increase the flow rate is to use a channel with a larger cross-section.

13-62

Solution Water is transported in an asphalt lined open channel at a specified rate. The dimensions of the best cross-section for various geometric shapes are to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

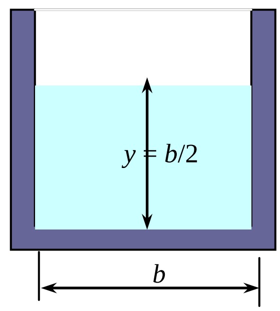
Properties The Manning coefficient for asphalt lined open channels is n = 0.016 (Table 13-1).



Analysis (a) Circular channel of Diameter D: Best cross-section occurs when the channel is half-full, and thus the flow area is $\pi D^2/8$ and the hydraulic radius is $D/4$. Then from Manning's equation, $\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2}$,

$$10\,\mathrm{m^3/s} = \frac{1\,\mathrm{m^{1/3}}/s}{0.016}(\pi D^2/8\,\mathrm{m^2})(D/4\,\mathrm{m})^{2/3}(0.0015)^{1/2}$$

which gives D **= 3.42 m.**

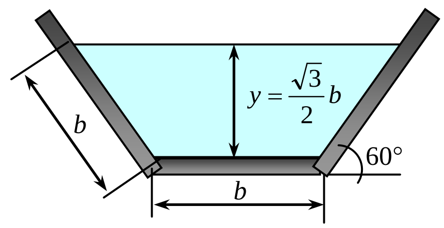


(b) Rectangular channel of bottom width b: For best cross-section, $y = b/2$. Then $A_c = yb = b^2/2$ and $R_h = b/4$. From the Manning equation,

$$10\,\mathrm{m^3/s} = \frac{1\,\mathrm{m^{1/3}}/s}{0.016}(b^2/2\,\mathrm{m^2})(b/4\,\mathrm{m})^{2/3}(0.0015)^{1/2}$$

which gives b **= 3.12 m,** and $y = b/2$ = **1.56 m.**

(c) Trapezoidal channel of bottom width b: For best cross-section, θ = 60° and $y = b\sqrt{3}/2$. Then, $A_c = y(b + b\cos\theta) = 0.5\sqrt{3}b^2(1+\cos 60°) = 0.75\sqrt{3}b^2$, $p = 3b$, $R_h = \frac{y}{2} = \frac{\sqrt{3}}{4}b$. From the Manning equation,



$$10\,\mathrm{m^3/s} = \frac{1\,\mathrm{m^{1/3}}/s}{0.016}(0.75\sqrt{3}b^2\,\mathrm{m^2})(\sqrt{3}b/4\,\mathrm{m})^{2/3}(0.0015)^{1/2}$$

which gives b **= 1.90 m,** and y **= 1.65 m** and θ = **60°.**

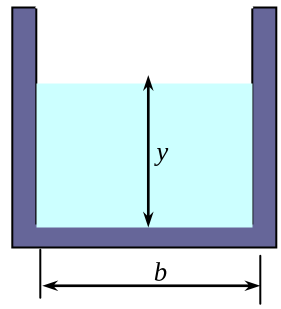
Discussion The perimeters for the circular, rectangular, and trapezoidal channels are 5.37 m, 6.24 m, and 5.70 m, respectively. Therefore, the circular cross-section has the smallest perimeter.

13-63

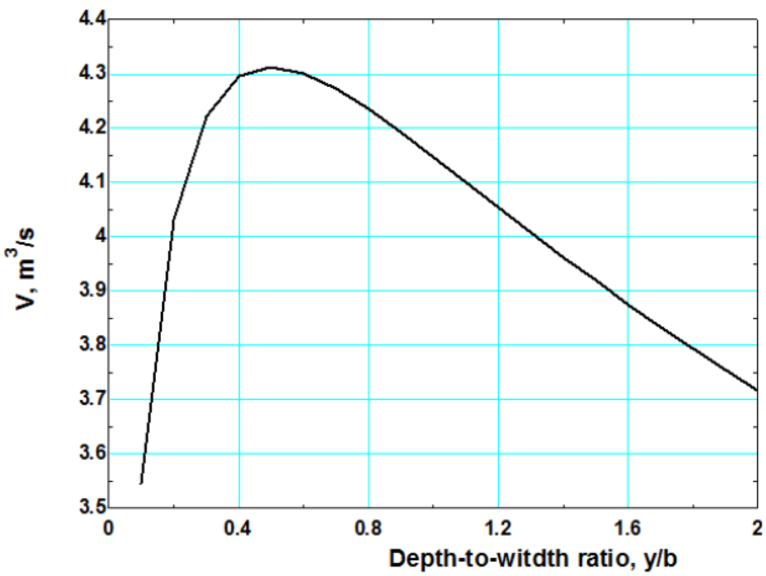
Solution Uniform flow in an asphalt-lined rectangular channel is considered. By varying the depth-to-width ratio from 0.1 to 2 in increments of 0.1 for a fixed value of flow area, it is the to be shown that the best hydraulic cross section occurs when y/b = 0.5, and the results are to be plotted.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
a=1
n=0.016 "Manning coefficient"
s=0.003 "Bottom slope is constant"
Ac=2 "Flow area remains constant at 2 m2"
Ratio=y/b
Ac=b*y
p=b+2*y
Rh=Ac/p "Hydraulic radius"
Vdot=(a/n)*Ac*Rh^(2/3)*SQRT(s) "Volume flow rate"
```



Depth-to-width ratio, y/b	Channel width, b, m	Flow rate, $\dot{V}$, m^3/s
0.1	4.47	3.546
0.2	3.16	4.031
0.3	2.58	4.221
0.4	2.24	4.295
0.5	2.00	4.313
0.6	1.83	4.301
0.7	1.69	4.273
0.8	1.58	4.235
0.9	1.49	4.192
1.0	1.41	4.147
1.1	1.35	4.101
1.2	1.29	4.054
1.3	1.24	4.008
1.4	1.20	3.963
1.5	1.15	3.919
1.6	1.12	3.876
1.7	1.08	3.834
1.8	1.05	3.794
1.9	1.03	3.755
2.0	1.00	3.717



Discussion It is clear from the table and the chart that the depth-to-width ratio of y/b = 0.5 corresponds to the best cross-section for an open channel of rectangular cross-section.

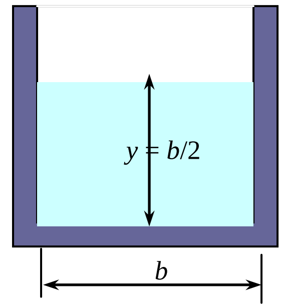
13-64E

Solution Water is to be transported in a rectangular channel at a specified rate. The dimensions for the best cross-section if the channel is made of unfinished and finished concrete are to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties The Manning coefficient is n = 0.014 for unfinished concrete (part a) and n = 0.012 for finished conrete (part b), respectively (Table 13-1). In English units, a = 1.486 ft$^{1/3}$/s.

Analysis For best cross-section of a rectangular cross-section, $y = b/2$. Then $A_c = yb = b^2/2$ and $R_h = b/4$. The flow rate is determined from the Manning equation, $\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2}$. Plugging in and solving for dimension b we get



$$\boxed{b = \left(\frac{2\dot{V} n \left(4^{2/3}\right)}{a\sqrt{S_0}} \right)^{3/8}}$$ (This is the answer in variable form)

(*a*) Unfinished concrete, n = 0.014:

$$b = \left(\frac{2\left(750\,\frac{\text{ft}^3}{\text{s}}\right)(0.014)\left(4^{2/3}\right)}{\left(1.486\,\frac{\text{ft}^{1/3}}{\text{s}}\right)\sqrt{0.0004}} \right)^{3/8} = 16.556 \text{ ft}$$

Thus, b **= 16.6 ft,** and $y = b/2$ = **8.28 ft** (to three significant digits).

(*b*) Finished concrete, n = 0.012:

$$b = \left(\frac{2\left(750\,\frac{\text{ft}^3}{\text{s}}\right)(0.012)\left(4^{2/3}\right)}{\left(1.486\,\frac{\text{ft}^{1/3}}{\text{s}}\right)\sqrt{0.0004}} \right)^{3/8} = 15.626 \text{ ft}$$

Thus, b **= 15.6 ft,** and $y = b/2$ = **7.81 ft** (to three significant digits).

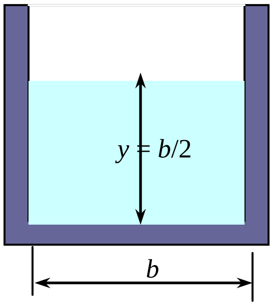
Discussion Note that channels with rough surfaces require a larger cross-section to transport the same amount of water.

13-65E

Solution Water is to be transported in a rectangular channel at a specified rate. The dimensions for the best cross-section if the channel is made of unfinished and finished concrete are to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties The Manning coefficient is n = 0.012 and n = 0.014 for finished and unfinished concrete, respectively (Table 13-1).



Analysis For best cross-section of a rectangular cross-section, $y = b/2$. Then $A_c = yb = b^2/2$ and $R_h = b/4$. The flow rate is determined from the Manning equation, $\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2}$,

(a) Finished concrete, n = 0.012:

$$650 \text{ ft}^3/\text{s} = \frac{1.486 \text{ ft}^{1/3}/s}{0.012}(b^2/2 \text{ ft}^2)(b/4 \text{ ft})^{2/3}(0.0004)^{1/2}$$

It gives b **= 14.8 ft,** and $y = b/2$ = **7.41 ft**

(b) Unfinished concrete, n = 0.014:

$$650 \text{ ft}^3/\text{s} = \frac{1.486 \text{ ft}^{1/3}/s}{0.014}(b^2/2 \text{ ft}^2)(b/4 \text{ ft})^{2/3}(0.0004)^{1/2}$$

It gives b **= 15.7 ft,** and $y = b/2$ = **7.85 ft**

Discussion Note that channels with rough surfaces require a larger cross-section to transport the same amount of water.

13-66

Solution The flow of water in a trapezoidal channel made of unfinished-concrete is considered. For given flow rate and bottom slope, the flow depth is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

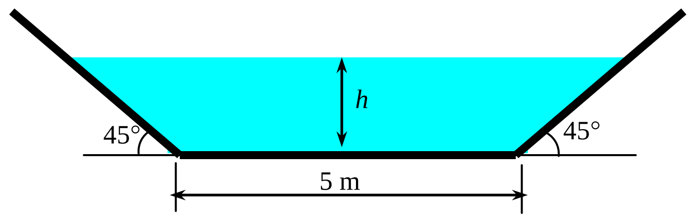
Properties Manning coefficient for an open channel of unfinished concrete is n = 0.014 (Table 13-1).

Analysis From geometric considerations, the flow area, wetted perimeter, and hydraulic radius are

$$A_c = \frac{5\,\text{m} + 5\,\text{m} + 2h}{2}h = (5+h)h$$

$$p = (5\,\text{m}) + 2h/\sin 45° = 5 + 2.828h$$

$$R_h = \frac{A_c}{p} = \frac{(5+h)h}{5+2h/\sin 45°}$$



Substituting the given quantities into Manning's equation,

$$\dot{V} = \frac{a}{n}A_c R_h^{2/3} S_0^{1/2} \quad \rightarrow \quad 25\,\text{m}^3/\text{s} = \frac{1\,\text{m}^{1/3}/s}{0.014}(5+h)h\left(\frac{(5+h)h}{5+2h/\sin 45°}\right)^{2/3}(\tan 1°)^{1/2}$$

It gives the flow depth to be h = **0.685 m**.

Discussion Non-linear equations frequently arise in the solution of open channel flow problems. They are best handled by equation solvers such as EES.

13-67

Solution The flow of water in a weedy excavated trapezoidal channel is considered. For given flow rate and bottom slope, the flow depth is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

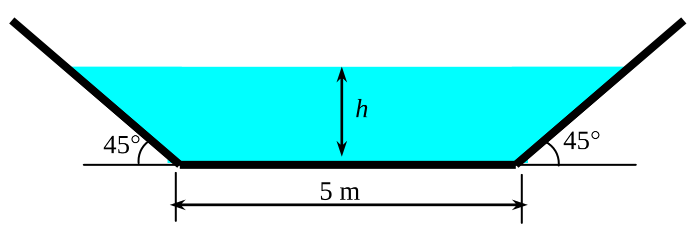
Properties Manning coefficient for the channel is given to be n = 0.030.

Analysis From geometric considerations, the flow area, wetted perimeter, and hydraulic radius are

$$A_c = \frac{5\,\text{m} + 5\,\text{m} + 2h}{2}h = (5+h)h$$

$$p = (5\,\text{m}) + 2h/\sin 45° = 5 + 2.828h$$

$$R_h = \frac{A_c}{p} = \frac{(5+h)h}{5+2h/\sin 45°}$$



Substituting the given quantities into Manning's equation,

$$\dot{V} = \frac{a}{n}A_c R_h^{2/3} S_0^{1/2} \quad \rightarrow \quad 25\,\text{m}^3/\text{s} = \frac{1\,\text{m}^{1/3}/s}{0.030}(5+h)h\left(\frac{(5+h)h}{5+2h/\sin 45°}\right)^{2/3}(\tan 1°)^{1/2}$$

It gives the flow depth to be y = **1.07 m**.

Discussion Note that as the Manning coefficient increases because of the increased surface roughness of the channel, the flow depth required to maintain the same flow rate also increases.

Gradually and Rapidly Varied Flows and Hydraulic Jump

13-68C

Solution We are to discuss the differences between GVF and RVF.

Analysis *Gradually varied flow* (GVF) is characterized by **gradual variations in flow depth and velocity** (small slopes and no abrupt changes) and **a free surface that always remains smooth** (no discontinuities or zigzags). *Rapidly varied flow* (RVF) involves **rapid changes in flow depth and velocity**. A change in the bottom slope or cross-section of a channel or an obstruction on the path of flow may cause the uniform flow in a channel to become gradually or rapidly varied flow. **Analytical relations for the profile of the free surface can be obtained in GVF, but this is not the case for RVF because of the intense agitation**.

Discussion In many situations, the shape of the free surface must be solved numerically, even for GVF.

13-69C

Solution We are to discuss the difference between uniform and nonuniform (varied) flow.

Analysis Both uniform and varied flows are steady, and thus neither involves any change with time at a specified location. In *uniform flow*, **the flow depth *y* and the flow velocity *V* remain constant** whereas in *nonuniform* or *varied flow*, **the flow depth and velocity vary in the streamwise direction of the flow**. In *uniform flow*, **the slope of the energy line is equal to the slope of the bottom surface**. Therefore, the friction slope equals the bottom slope, $S_f = S_0$. In *varied flow*, however, **these slopes are different**.

Discussion Varied flows are further classified into gradually varied flow (GVF) and rapidly varied flow (RVF).

13-70C

Solution We are to analyze a claim that wall shear is negligible in RVF but important in GVF.

Analysis **Yes, we agree with this claim**. Rapidly varied flows occur over a short section of the channel with relatively small surface area, and thus frictional losses associated with wall shear are negligible compared with losses due to intense agitation and turbulence. Losses in GVF, on the other hand, are primarily due to frictional effects along the channel, and should be considered.

Discussion There is somewhat of an analogy here with internal flows. In long pipe sections with entrance lengths and/or gradually changing pipe diameter, wall shear is important. However, in short sections of piping with rapid change of diameter or a blockage or turn, etc (minor loss), friction along the wall is typically negligible compared to other losses.

13-71C

Solution We are to analyze what happens to flow depth in an upward-sloped rectangular channel during supercritical flow.

Analysis The flow depth *y* **(*a*) *increases* in the flow direction**.

Discussion Since the flow is supercritical, this increase in flow depth may occur via a hydraulic jump.

13-72C

Solution We are to determine if it is possible for subcritical flow to undergo a hydraulic jump.

Analysis **No. It is impossible for subcritical flow to undergo a hydraulic jump**. Such a process would require the head loss h_L to become negative, which is impossible. It would correspond to negative entropy generation, which would be a violation of the second law of thermodynamics. Therefore, the upstream flow must be supercritical ($\mathrm{Fr}_1 > 1$) for a hydraulic jump to occur.

Discussion This is analogous to **normal shock waves** in gases – the only way a shock wave can occur is if the flow upstream of the shock wave is supersonic with $\mathrm{Ma}_1 > 1$ (analogous to supercritical in open-channel flow with $\mathrm{Fr}_1 > 1$).

13-73C

Solution We are to define the energy dissipation ratio for a hydraulic jump and discuss why a hydraulic jump is sometimes used to dissipate energy.

Analysis Hydraulic jumps are often designed in conjunction with stilling basins and spillways of dams in order to waste as much of the mechanical energy as possible **to minimize the mechanical energy of the fluid and thus its potential to cause damage**. In such cases, a measure of performance of a hydraulic jump is the *energy dissipation ratio*, which is the **fraction of energy dissipated** through a hydraulic jump, defined as

$$\boxed{\text{Dissipation ratio} = \frac{h_L}{E_{s1}} = \frac{h_L}{y_1 + V_1^2/(2g)} = \frac{h_L}{y_1\left(1+\mathrm{Fr}_1^2/2\right)}}.$$

Discussion Since the head loss is always positive, the dissipation ratio is also always positive.

13-74C

Solution We are to analyze what happens to flow depth in a horizontal rectangular channel during subcritical flow.

Analysis The flow depth y must **(*c*) *decrease* in the flow direction**.

Discussion Since the flow is subcritical, there is no possibility of a hydraulic jump.

13-75C

Solution We are to analyze what happens to flow depth in a sloped rectangular channel during subcritical flow.

Analysis The flow depth y must **(*a*) *increase* in the flow direction**.

Discussion Since the flow is subcritical, there is no possibility of a hydraulic jump.

13-76C

Solution We are to analyze what happens to flow depth in a horizontal rectangular channel during supercritical flow.

Analysis The flow depth y (*a*) ***increases* in the flow direction**.

Discussion Since the flow is supercritical, this increase in flow depth may occur via a hydraulic jump.

13-77C

Solution We are to analyze what happens to flow depth in a sloped rectangular channel during subcritical flow.

Analysis The flow depth *y* **(*c*) *decreases* in the flow direction**.

Discussion Since the flow is subcritical, there is no possibility of a hydraulic jump.

13-78

Solution Water is flowing in a V-shaped open channel with a specified bottom slope at a specified rate. It is to be determined whether the slope of this channel should be classified as mild, critical, or steep.

Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

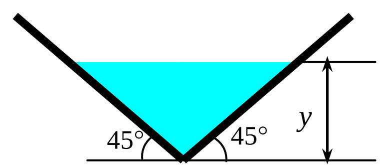
Properties The Manning coefficient for a cast iron channel is n = 0.013 (Table 13-1).

Analysis From geometric considerations, the cross-sectional area, perimeter, and hydraulic radius are

$$A_c = y(2y)/2 = y^2 \qquad p = 2\sqrt{y^2 + y^2} = 2\sqrt{2}y \qquad R_h = \frac{A_c}{p} = \frac{y^2}{2\sqrt{2}y} = \frac{y}{2\sqrt{2}}$$

Substituting the known quantities into the Manning equation,

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} \quad \rightarrow \quad 3\,\text{m}^3/\text{s} = \frac{1\,\text{m}^{1/3}/\text{s}}{0.013}(y^2)\left(\frac{y}{2\sqrt{2}}\right)^{2/3}(0.002)^{1/2}$$



Solving for the flow depth y gives y = 1.23 m. The critical depth for this flow is

$$y_c = \frac{\dot{V}^2}{gA_c^2} = \frac{(3\,\text{m}^3/\text{s})^2}{(9.81\,\text{m/s}^2)(1.23\,\text{m})^2} = 0.61\,\text{m}$$

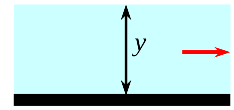
This channel at these flow conditions is classified as **mild** since $y > y_c$, and the flow is subcritical.

Discussion If the flow depth were smaller than 0.61 m, the channel slope would be said to be *steep*. Therefore, the bottom slope alone is not sufficient to classify a downhill channel as being mild, critical, or steep.

13-79

Solution Water is flowing in an wide brick open channel uniformly. The range of flow depth for which the channel can be classified as "steep" is to be determined.

Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.



Properties The Manning coefficient for a brick open channel is n = 0.015 (Table 13-1).

Analysis The slope of the channel is $S_0 = \tan\alpha = \tan 0.4° = 0.006981$.

The hydraulic radius for a wide channel is equal to the flow depth, $R_h = y$. Now assume the flow in the channel to be critical, The channel flow in this case would be critical slope S_c, and the flow depth would be the critical flow depth, which is determined from

$$S_c = \frac{g\,n^2}{a^2 y_c^{1/3}} \quad \rightarrow \quad y_c = \left(\frac{g\,n^2}{a^2 S_c}\right)^3$$

Substituting,

$$y_c = \left(\frac{g\,n^2}{a^2 S_c}\right)^3 = \left(\frac{(9.81\text{m/s}^2)(0.015)^2}{(1\text{ m}^{1/3}/s)^2(0.006981)}\right)^3 = 0.03160\,\text{m}$$

Therefore, this channel can be classified as *steep* for uniform flow depths less than y_c, i.e., ***y* < 0.03160 m**.

Discussion Note that two channels of the same slope can be classified as differently (one mild and the other steep) if they have different roughness and thus different values of *n*.

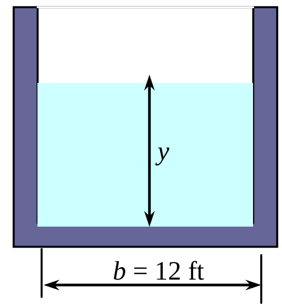
13-80E

Solution Water is flowing in a rectangular open channel with a specified bottom slope at a specified flow rate. It is to be determined whether the slope of this channel should be classified as mild, critical, or steep. The surface profile is also to be classified for a specified flow depth of 2 m.

Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

Properties The Manning coefficient of a channel with unfinished concrete surfaces is n = 0.014 (Table 13-1).

Analysis The cross-sectional area, perimeter, and hydraulic radius are



$$A_c = yb = y(12\text{ ft}) = 12y\text{ ft}^2 \qquad p = b + 2y = 12\text{ ft} + 2y = 12 + 2y\text{ ft}$$

$$R_h = \frac{A_c}{p} = \frac{12y\text{ ft}^2}{(12+2\text{y})\text{ ft}}$$

Substituting the known quantities into the Manning equation,

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} \quad \rightarrow \quad 300\text{ ft}^3/\text{s} = \frac{1.486\text{ ft}^{1/3}/s}{0.014}(12y)\left(\frac{12y}{12+2y}\right)^{2/3}(\tan 0.5°)^{1/2}$$

Solving for the flow depth y gives y = 1.95 ft. The critical depth for this flow is

$$y_c = \frac{\dot{V}^2}{gA_c^2} = \frac{(300\text{ ft}^3/\text{s})^2}{(32.2\text{ ft/s}^2)(12\text{ ft}\times 1.95\text{ ft})^2} = 5.10\text{ ft}$$

This channel at these flow conditions is classified as **steep** since $y < y_c$, and the flow is supercritical. Alternately, we could solve for Froude number and show that Fr > 1 and reach the same conclusion. The given flow is uniform, and thus $y = y_n$ = 1.95 ft. Therefore, the given value of y = 3 ft during development is between y_c and y_n, and the **flow profile is S2** (Table 13-3).

Discussion If the flow depth were larger than 5.10 ft, the channel slope would be said to be *mild*. Therefore, the bottom slope alone is not sufficient to classify a downhill channel as being mild, critical, or steep.

13-81

Solution Water is flowing in an open channel uniformly. It is to be determined whether the channel slope is mild, critical, or steep for this flow.

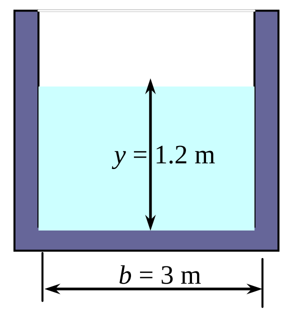
Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

Properties The Manning coefficient for an open channel with finished concrete surfaces is $n = 0.012$ (Table 13-1).

Analysis The cross-sectional area, perimeter, and hydraulic radius are

$$A_c = yb = (1.2\text{ m})(3\text{ m}) = 3.6\text{ m}^2 \qquad p = b + 2y = 3\text{ m} + 2(1.2\text{ m}) = 5.4\text{ m}$$

$$R_h = \frac{A_c}{p} = \frac{3.6\text{ m}^2}{5.4\text{ m}} = 0.6667\text{ m}$$



The flow rate is determined from the Manning equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{1\text{ m}^{1/3}/\text{s}}{0.012}(3.6\text{ m}^2)(0.6667\text{ m})^{2/3}(0.002)^{1/2} = 10.2\text{ m}^3/\text{s}$$

Noting that the flow is uniform, the specified flow rate is the normal depth and thus $y = y_n = 1.2$ m. The critical depth for this flow is

$$y_c = \left(\frac{\dot{V}^2}{g b^2}\right)^{1/3} = \left(\frac{(10.2\text{ m}^3/\text{s})^2}{(9.81\text{ m/s}^2)(3\text{ m})^2}\right)^{1/3} = 1.06\text{ m}$$

This channel at these flow conditions is classified as **mild** since $y > y_c$, and the flow is subcritical.

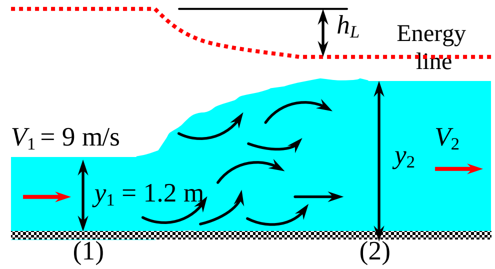
Discussion If the flow depth were smaller than 1.06 m, the channel slope would be said to be *steep*. Therefore, the bottom slope alone is not sufficient to classify a downhill channel as being mild, critical, or steep.

13-82

Solution Water at a specified depth and velocity undergoes a hydraulic jump. The depth and Froude number after the jump, the head loss and dissipation ratio, and dissipated mechanical power are to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The channel is horizontal.

Properties The density of water is 1000 kg/m^3.



Analysis (*a*) The Froude number before the hydraulic jump is

$$\mathrm{Fr}_1 = \frac{V_1}{\sqrt{gy_1}} = \frac{9\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(1.2\text{ m})}} = 2.62$$

which is greater than 1. Therefore, the flow is supercritical before the jump. The flow depth, velocity, and Froude number after the jump are

$$y_2 = 0.5y_1\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right) = 0.5(1.2\text{ m})\left(-1+\sqrt{1+8\times 2.62^2}\right) = \mathbf{3.89\,m}$$

$$V_2 = \frac{y_1}{y_2}V_1 = \frac{1.2\text{ m}}{3.89\text{ m}}(9\text{ m/s}) = 2.78\text{ m/s}$$

$$\mathrm{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{2.78\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(3.89\text{ m})}} = \mathbf{0.449}$$

(*b*) The head loss is determined from the energy equation to be

$$h_L = y_1 - y_2 + \frac{V_1^2 - V_2^2}{2g} = (1.2\text{ m}) - (3.89\text{ m}) + \frac{(9\text{ m/s})^2 - (2.78\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = \mathbf{1.05\,m}$$

The specific energy of water before the jump and the dissipation ratio are

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = (1.2\text{ m}) + \frac{(9\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 5.33\text{ m}$$

$$\text{Dissipation ratio} = \frac{h_L}{E_{s1}} = \frac{1.04\text{ m}}{5.33\text{ m}} = \mathbf{0.195}$$

Therefore, 19.5% of the available head (or mechanical energy) of the liquid is wasted (converted to thermal energy) as a result of frictional effects during this hydraulic jump.

(*c*) The mass flow rate of water is

$$\dot{m} = \rho\dot{\boldsymbol{V}} = \rho b y_1 V_1 = (1000\text{ kg/m}^3)(1.2\text{ m})(8\text{ m})(9\text{ m/s}) = 86{,}400\text{ kg/s}$$

Then the dissipated mechanical power becomes

$$\dot{E}_{\text{dissipated}} = \dot{m}gh_L = (86{,}400\text{ kg/s})(9.81\text{ m/s}^2)(1.04\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 881{,}000\text{ Nm/s} = \mathbf{881\,kW}$$

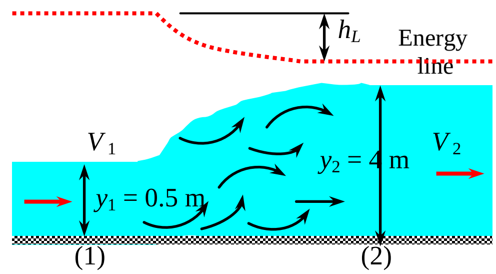
Discussion The results show that the hydraulic jump is a highly dissipative process, wasting 881 kW of power production potential in this case. That is, if the water is routed to a hydraulic turbine instead of being released from the sluice gate, up to 881 kW of power could be produced.

13-83

Solution Water flowing in a wide channel at a specified depth and flow rate undergoes a hydraulic jump. The mechanical power wasted during this process is to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The channel is horizontal.

Properties The density of water is 1000 kg/m^3.



Analysis Average velocities before and after the jump are

$$V_1 = \frac{70\ \text{m}^3/\text{s}}{(10\ \text{m})(0.5\ \text{m})} = 14\ \text{m/s}$$

$$V_2 = \frac{70\ \text{m}^3/\text{s}}{(10\ \text{m})(4\ \text{m})} = 1.75\ \text{m/s}$$

The head loss is determined from the energy equation to be

$$h_L = y_1 - y_2 + \frac{V_1^2 - V_2^2}{2g} = (0.5\ \text{m}) - (4\ \text{m}) + \frac{(14\ \text{m/s})^2 - (1.75\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = 6.33\ \text{m}$$

The mass flow rate of water is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1000\ \text{kg/m}^3)(70\ \text{m}^3/\text{s}) = 70{,}000\ \text{kg/s}$$

Then the dissipated mechanical power becomes

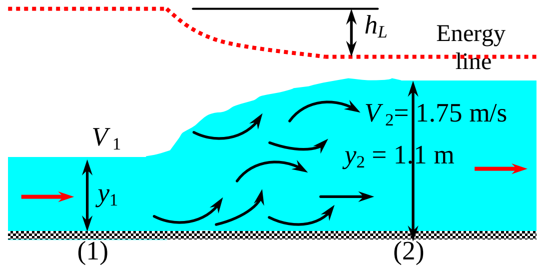
$$\dot{E}_{\text{dissipated}} = \dot{m} g h_L = (70{,}000\ \text{kg/s})(9.81\ \text{m/s}^2)(6.33\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right) = 4350\ \text{kNm/s} = \mathbf{4.35\ MW}$$

Discussion The results show that the hydraulic jump is a highly dissipative process, wasting 4.35 MW of power production potential in this case.

13-84

Solution The flow depth and average velocity of water after a hydraulic jump are measured. The flow depth and velocity before the jump as well as the fraction of mechanical energy dissipated are to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The channel is horizontal.



Analysis The Froude number after the hydraulic jump is

$$\mathrm{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{1.75\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m/s^2})(1.1\ \mathrm{m})}} = 0.5327$$

It can be shown that the subscripts in the relation $y_2 = 0.5y_1\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right)$ are interchangeable. Thus,

$$y_1 = 0.5y_2\left(-1+\sqrt{1+8\mathrm{Fr}_2^2}\right) = 0.5(1.1\mathrm{m})\left(-1+\sqrt{1+8\times 0.5327^2}\right) = 0.4446\ \mathrm{m}$$

$$V_1 = \frac{y_2}{y_1}V_2 = \frac{1.1\ \mathrm{m}}{0.4446\ \mathrm{m}}(1.75\ \mathrm{m/s}) = 4.329\ \mathrm{m/s}$$

The Froude number before the jump is

$$\mathrm{Fr}_1 = \frac{V_1}{\sqrt{gy_1}} = \frac{4.329\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m/s^2})(0.4446\ \mathrm{m})}} = 2.073$$

which is greater than 1. Therefore, the flow is indeed supercritical before the jump. The head loss is determined from the energy equation to be

$$h_L = y_1 - y_2 + \frac{V_1^2 - V_2^2}{2g} = (0.4446\ \mathrm{m}) - (1.1\ \mathrm{m}) + \frac{(4.329\ \mathrm{m/s})^2 - (1.75\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})} = 0.1437\ \mathrm{m}$$

The specific energy of water before the jump and the dissipation ratio is

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = (0.4446\ \mathrm{m}) + \frac{(4.329\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})} = 1.400\ \mathrm{m}$$

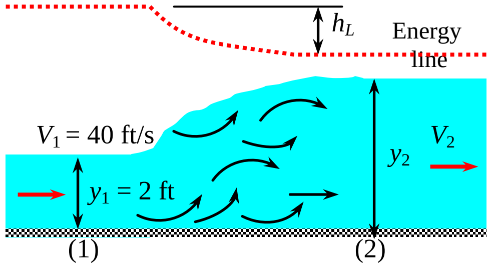
$$\text{Dissipation ratio} = \frac{h_L}{E_{s1}} = \frac{0.1437\ \mathrm{m}}{1.400\ \mathrm{m}} = \mathbf{0.103}$$

Discussion Note that as a result of this jump, 10.3% of the available energy is wasted.

13-85E

Solution Water at a specified depth and velocity undergoes a hydraulic jump, and dissipates a known fraction of its energy. The flow depth, velocity, and Froude number after the jump and the head loss associated with the jump are to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The channel is horizontal.



Analysis The Froude number before the hydraulic jump is

$$\mathrm{Fr}_1 = \frac{V_1}{\sqrt{gy_1}} = \frac{40\ \mathrm{ft/s}}{\sqrt{(32.2\ \mathrm{m/s^2})(2\ \mathrm{ft})}} = 4.984$$

which is greater than 1. Therefore, the flow is indeed supercritical before the jump. The flow depth, velocity, and Froude number after the jump are

$$y_2 = 0.5y_1\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right) = 0.5(2\ \mathrm{ft})\left(-1+\sqrt{1+8\times 4.984^2}\right) = \mathbf{13.1\ ft}$$

$$V_2 = \frac{y_1}{y_2}V_1 = \frac{2\ \mathrm{ft}}{13.1\ \mathrm{ft}}(40\ \mathrm{ft/s}) = \mathbf{6.09\ ft/s}$$

$$\mathrm{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{6.091\ \mathrm{ft/s}}{\sqrt{(32.2\ \mathrm{ft/s^2})(13.13\ \mathrm{m})}} = \mathbf{0.296}$$

The head loss is determined from the energy equation to be

$$h_L = y_1 - y_2 + \frac{V_1^2 - V_2^2}{2g} = (2\ \mathrm{ft}) - (13.1\ \mathrm{ft}) + \frac{(40\ \mathrm{ft/s})^2 - (6.09\ \mathrm{ft/s})^2}{2(32.2\ \mathrm{ft/s^2})} = \mathbf{13.2\ ft}$$

Discussion The results show that the hydraulic jump is a highly dissipative process, wasting 13.2 ft of head in the process.

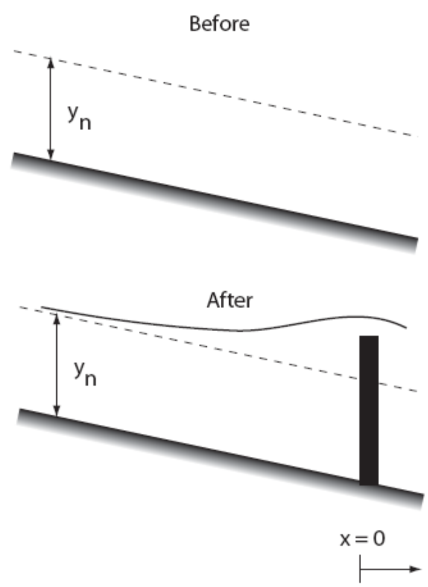
13-86

Solution A dam is built downstream of a wide rectangular channel in which water is flowing uniformly. The normal and critical flow depths upstream, the flow type, and how far upstream of the dam the reservoir extends are to be determined.

Assumptions **1** The channel is wide. **2** The flow is initially uniform, and becomes gradually varied as the effect of the dam is felt. **3** The bottom slope is constant. **4** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

Properties The Manning coefficient of the channel is given to be n = 0.03.

Analysis (a) The channel is said to be wide, and thus the hydraulic radius is equal to the flow depth, $R_h \cong y$. Knowing the flow rate per unit width (b = 1 m), the normal depth is determined from the Manning equation to be



$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n}(yb)y^{2/3} S_0^{1/2} = \frac{a}{n} b y^{5/3} S_0^{1/2}$$

$$y_n = \left(\frac{(\dot{V}/b)n}{a S_0^{1/2}}\right)^{3/5} = \left(\frac{(1.5\,\text{m}^2/\text{s})(0.03)}{(1\,\text{m}^{1/3}/\text{s})(0.0005)^{1/2}}\right)^{3/5} = \mathbf{1.52\,m}$$

The critical depth for this flow is

$$y_c = \frac{\dot{V}^2}{g A_c^2} = \frac{\dot{V}^2}{g(by)^2} = \left(\frac{(\dot{V}/b)^2}{g}\right)^{1/3} = \left(\frac{(1.5\,\text{m}^2/\text{s})^2}{(9.81\,\text{m/s}^2)}\right)^{1/3} = \mathbf{0.61\,m}$$

Noting that $y_n > y_c$, the uniform flow upstream the channel **subcritical**.

(b) Knowing the initial condition $y(0)$ = 2.5 m, the flow depth y at any x location can be determined by numerical integration of the GVF equation

$$\frac{dy}{dx} = \frac{S_0 - S_f}{1 - \text{Fr}^2}$$

where the Froude number for a wide rectangular channel is

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{\dot{V}/by}{\sqrt{gy}} = \frac{\dot{V}/b}{\sqrt{gy^3}}$$

and the friction slope is determined from the uniform-flow equation by setting $S_0 = S_f$,

$$\dot{V} = \frac{a}{n} b y^{5/3} S_f^{1/2} \;\rightarrow\; S_f = \left(\frac{(\dot{V}/b)n}{a y^{5/3}}\right)^2 = \frac{(\dot{V}/b)^2 n^2}{a^2 y^{10/3}}$$

Substituting, the GVF equation for a wide rectangular channel becomes

$$\frac{dy}{dx} = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1 - (\dot{V}/b)^2/(gy^3)}$$

which is highly nonlinear, and thus difficult to integrate analytically. The solution of the nonlinear first order differential equation subject to the initial condition $y(x_2) = y_2$ can be expressed as

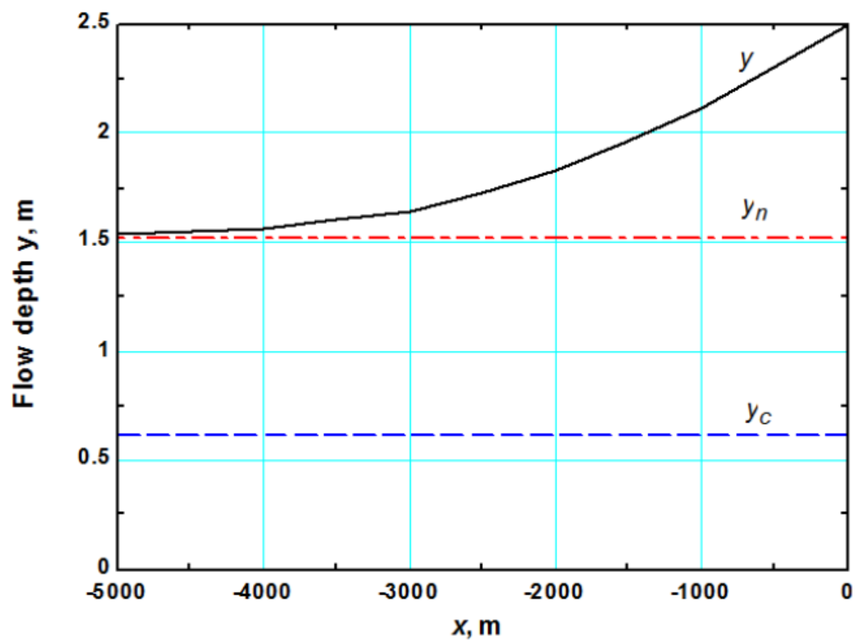
$$y = y_2 - \int_{x_1}^{x_2} f(x,y)dx \qquad \text{where} \qquad f(x,y) = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1 - (\dot{V}/b)^2/(gy^3)}$$

Where $x_2 = 0$ and $y = y(x)$ is the water depth at the specified location x negative value). For given numerical values and taking x_1 = - 500 m, this problem can be solved using EES as follows:

```
Vol=1.5 "m^3/s, volume flow rate per unit width, b = 1 m"
b=1 "m. width of channel"
n=0.03 "Manning coefficient"
S_0=0.0005 "Slope of channel"
g=9.81 "gravitational acceleration, m/s^2"

y_c=(Vol^2/(g*b^2))^(1/3) "Critical depth"
y_n=(Vol*n/(b*S_0^0.5))^(3/5) "Normal depth"

x2=0; y2=2.5 "m, initial condition"
x1=-500 "m, length of channel"

f_xy=(S_0-((Vol/b)^2*n^2/y^(10/3)))/(1-(Vol/b)^2/(g*y^3)) "the GVF equation to be integrated"
y=y2-integral(f_xy, x, x1, x2) "integral equation, auto step: Press F2 to solve."
```

Copying and pasting the mini program above into a blank EES screen gives the water depth at a location of x_1 = - 500 m to be 2.30 m, which is considerably higher than 1.60 m (5% above the normal depth of 1.52 m). Repeating calculations for different x_1 values and tabulating, we get

Distance along channel *x*, m	Flow depth *y*, m
0	2.50
-500	2.30
-1000	2.12
-1500	1.96
-2000	1.83
-2500	1.72
-3000	1.65
-3500	1.60
-4000	1.57
-4500	1.55
-5000	1.54



Therefore, the x value corresponding to a flow depth of y = 1.60 m is -3500 m. Finally, **the reservoir extends 3500 m upstream.**

Discussion This problem solves the GVF equation in the 'backwards' direction in order to determine the extent of the backwater created by a dam or obstruction. The surface profile is also plotted above using the tabulated values and the plot feature of EES. From the dam, looking upstream, the water surface profile is an M1 type. The water depth decreases with distance upstream, and the uniform flow depth is steadily approached.

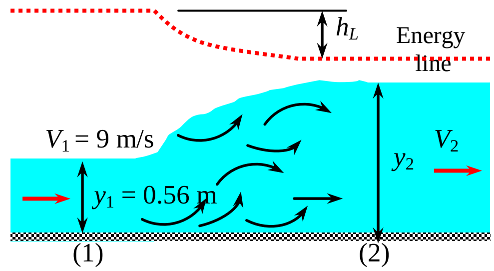
13-87

Solution Water at a specified depth and velocity undergoes a hydraulic jump. The head loss associated with this process is to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The channel is horizontal.

Analysis The Froude number before the hydraulic jump is $\mathrm{Fr}_1 = \dfrac{V_1}{\sqrt{gy_1}} = \dfrac{9 \text{ m/s}}{\sqrt{(9.81\,\text{m/s}^2)(0.56\,\text{m})}} = 3.840$, which is greater than 1. Therefore, the flow is indeed supercritical before the jump. The flow depth, velocity, and Froude number after the jump are

$$y_2 = 0.5y_1\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right) = 0.5(0.56\,\text{m})\left(-1+\sqrt{1+8\times 3.840^2}\right) = 2.774 \text{ m}$$



$$V_2 = \frac{y_1}{y_2}V_1 = \frac{0.56\,\text{m}}{2.774\,\text{m}}(9 \text{ m/s}) = 1.817 \text{ m/s}$$

$$\mathrm{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{1.817 \text{ m/s}}{\sqrt{(9.81\,\text{m/s}^2)(2.774\,\text{m})}} = 0.3483$$

The head loss is determined from the energy equation to be

$$h_L = y_1 - y_2 + \frac{V_1^2 - V_2^2}{2g} = (0.56\,\text{m}) - (2.774\,\text{m}) + \frac{(9\text{ m/s})^2 - (1.817\text{ m/s})^2}{2(9.81\,\text{m/s}^2)} = \mathbf{1.75m}$$

Discussion The results show that the hydraulic jump is a highly dissipative process, wasting 1.75 m of head in the process.

13-88

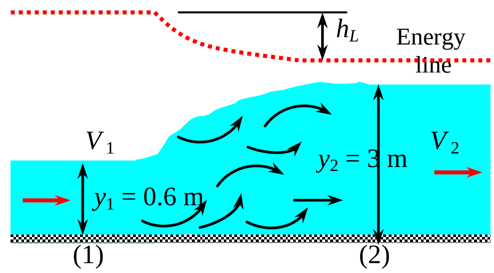
Solution The increase in flow depth during a hydraulic jump is given. The velocities and Froude numbers before and after the jump, and the energy dissipation ratio are to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The channel is horizontal.

Analysis The Froude number before the jump is determined from

$$y_2 = 0.5y_1\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right) \rightarrow \quad 3\,\mathrm{m} = 0.5\times(0.6\,\mathrm{m})\left(-1+\sqrt{1+8\mathrm{Fr}^2}\right)$$

which gives Fr_1 = **3.873.** Then,



$$V_1 = \mathrm{Fr}_1\sqrt{gy_1} = 3.873\sqrt{(9.81\,\mathrm{m/s^2})(0.6\,\mathrm{m})} = \mathbf{9.40\,m/s}$$

$$V_2 = \frac{y_1}{y_2}V_1 = \frac{0.6\,\mathrm{m}}{3\,\mathrm{m}}(9.40\,\mathrm{m/s}) = \mathbf{1.88\,m/s}$$

$$\mathrm{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{1.88\,\mathrm{m/s}}{\sqrt{(9.81\,\mathrm{m/s^2})(3\,\mathrm{m})}} = \mathbf{0.347}$$

The head loss is determined from the energy equation to be

$$h_L = y_1 - y_2 + \frac{V_1^2 - V_2^2}{2g} = (0.6\,\mathrm{m}) - (3\,\mathrm{m}) + \frac{(9.40\,\mathrm{m/s})^2 - (1.88\,\mathrm{m/s})^2}{2(9.81\,\mathrm{m/s^2})} = \mathbf{1.92\,m}$$

The specific energy of water before the jump and the dissipation ratio are

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = (0.6\,\mathrm{m}) + \frac{(9.40\,\mathrm{m/s})^2}{2(9.81\,\mathrm{m/s^2})} = 5.10\,\mathrm{m}$$

$$\text{Dissipation ratio} = \frac{h_L}{E_{s1}} = \frac{1.92\,\mathrm{m}}{5.10\,\mathrm{m}} = \mathbf{0.376}$$

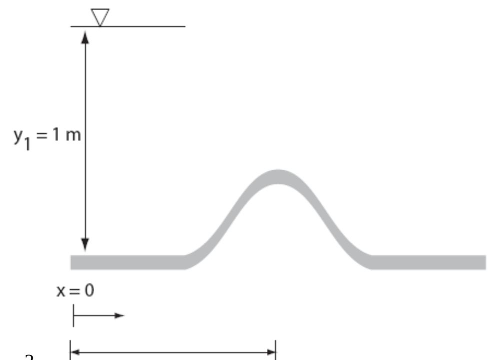
Therefore, 37.6% of the available head (or mechanical energy) of water is wasted (converted to thermal energy) as a result of frictional effects during this hydraulic jump.

Discussion The results show that the hydraulic jump is a highly dissipative process, wasting over one-third of the available head.

13-89

Solution Gradually varied flow over a bump in a wide channel is considered. The normal and critical flow depths are to be calculated and plotted, and the behavior of the free surface is to be investigated.

Assumptions **1** The channel is wide, and the flow is gradually varied. **2** The bottom slope is constant. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.



Properties The Manning coefficient of the channel is n = 0.02 (given).

Analysis (a) The channel is wide, and thus the hydraulic radius is equal to the flow depth, $R_h \cong y$. The flow rate per unit width (b = 1 m) is

$$\dot{V} = V_1 A_{c1} = V_1 b y_1 \quad \rightarrow \quad \dot{V}/b = V_1 y_1 = (0.75\,\text{m})(1\,\text{m}) = 0.75\,\text{m}^3/\text{s}\cdot\text{m}$$

Then the critical depth becomes

$$y_c = \frac{\dot{V}^2}{gA_c^2} = \frac{\dot{V}^2}{g(by)^2} = \left(\frac{(\dot{V}/b)^2}{g}\right)^{1/3} = \left(\frac{(0.75\,\text{m}^2/\text{s})^2}{(9.81\,\text{m/s}^2)}\right)^{1/3} = \mathbf{0.386\,m}$$

Noting that $y_1 > y_c$, the initial flow is **subcritical**.

The elevation of the channel bottom is given as $z_b = \Delta z_b \exp[-0.001(x-100)^2]$. Noting that S_0 is the negative of the bottom slope,

$$S_0(x) = -\frac{dz_b}{dx} = -\frac{d\left(\Delta z_b \exp[-0.001(x-100)^2]\right)}{dx} = 0.002(x-100)\exp[-0.001(x-100)^2]$$

which varies along the channel. Note that S_0 is negative (adverse flow) for x < 100 m. Then the normal depth is determined from the Manning equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n}(yb)y^{2/3}S_0^{1/2} = \frac{a}{n} b y^{5/3} S_0^{1/2}$$

$$y_n = \left(\frac{(\dot{V}/b)n}{aS_0^{1/2}}\right)^{3/5} = \left(\frac{(\dot{V}/b)n}{a\left(0.002(x-100)e^{-0.001(x-100)^2}\right)^{1/2}}\right)^{3/5} = \left(\frac{(0.75\,\text{m}^2/\text{s})(0.02)}{(1\,\text{m}^{1/3}/\text{s})\left(0.002(x-100)e^{-0.001(x-100)^2}\right)^{1/2}}\right)^{3/5}$$

Normal flow cannot exist for x < 100 m since $S_0 < 0$, and $y_n \rightarrow \propto$ for $S_0 = 0$. Therefore, y_n is undefined for x < 100 m, infinity for x = 0, and first decreases and then increases for x > 100 m as the slope S_0 increases and then decreases. This is shown in the figure.

(b) Knowing the initial condition $y(0)$ = 1 m, the flow depth y at any x location can be determined by numerical integration of the GVF equation

$$\frac{dy}{dx} = \frac{S_0 - S_f}{1-\text{Fr}^2}$$

where the Froude number for a wide rectangular channel is

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{\dot{V}/by}{\sqrt{gy}} = \frac{\dot{V}/b}{\sqrt{gy^3}}$$

and the friction slope is determined from the uniform-flow equation by setting $S_0 = S_f$,

$$\dot{V} = \frac{a}{n} b y^{5/3} S_f^{1/2} \quad \rightarrow \quad S_f = \left(\frac{(\dot{V}/b)n}{ay^{5/3}}\right)^2 = \frac{(\dot{V}/b)^2 n^2}{a^2 y^{10/3}}$$

Substituting, the GVF equation for a wide rectangular channel becomes

$$\frac{dy}{dx} = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1-(\dot{V}/b)^2/(gy^3)}$$

which is highly nonlinear, and thus difficult to integrate analytically. The solution of the nonlinear first order differential equation subject to the initial condition $y(x_1) = y_1$ can be expressed as

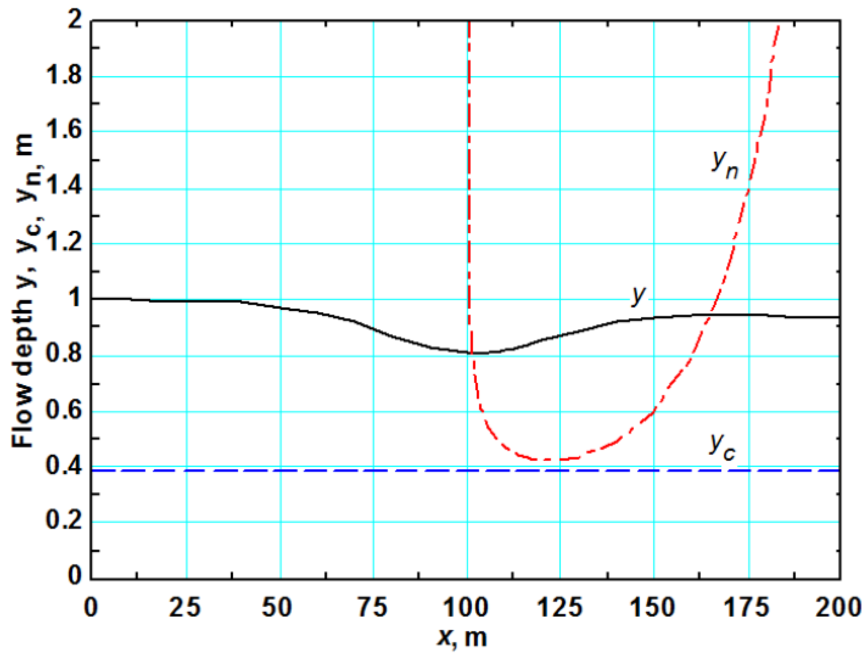
$$y = y_1 + \int_{x_1}^{x_2} f(x,y)dx \qquad \text{where} \qquad f(x,y) = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1-(\dot{V}/b)^2/(gy^3)}$$

where $y = y(x)$ is the water depth at the specified location x. For given numerical values, this problem can be solved using EES as follows:

```
Vol=0.75 "m^3/s, volume flow rate per unit width, b = 1 m"
b=1 "m. width of channel"
n=0.02 "Manning coefficient"
S_0=0.15*0.002*(x-100)*exp(-0.001*(x-100)^2) "Slope of channel"
g=9.81 "gravitational acceleration, m/s^2"
y_c=(Vol^2/(g*b^2))^(1/3) "Critical depth"
y_n=(Vol*n/(b*ABS(0.15*0.002*(x2-100)*exp(-0.001*(x2-100)^2))^0.5))^(3/5) "Normal depth"
x1=0; y1=1 "m, initial condition"
x2=110 "m, length of channel"
f_xy=(S_0-((Vol/b)^2*n^2/y^(10/3)))/(1-(Vol/b)^2/(g*y^3)) "the GVF equation to be integrated"
y=y1+integral(f_xy, x, x1, x2) "integral equation, auto step: Press F2 to solve."
```

Copying and pasting the mini program above into a blank EES screen gives the normal and actual water depth at a location of x_2 = 110 m to be $y_n(x_2)$ = 0.47 m and $y(x_2)$ = 0.82 m. Repeating calculations for different x_2 values and tabulating and plotting, we get

Distance along channel x, m	Flow depth y, m	Normal depth y_n, m
0	1.00	-
10	1.00	-
20	0.99	-
30	0.99	-
40	0.99	-
50	0.97	-
60	0.95	-
70	0.92	-
80	0.87	-
90	0.83	-
100	0.81	∞
110	0.82	0.47
120	0.85	0.42
130	0.89	0.43
140	0.92	0.49
150	0.94	0.60
160	0.94	0.79
170	0.94	1.12
180	0.94	1.68
190	0.94	2.70
200	0.94	4.63



In this problem, the GVF equation for the case of a frictional flow over a Gaussian bump is solved. Note that the local slope must be computed at each integration step since the bathymetry is changing continuously and smoothly,

Discussion From the subcritical state of our initial flow, we note that we are on an H2 profile at the start. As soon as the leading edge of the bump is encountered, this turns into an A2 profile. For this portion of the flow, y_n is undefined and Table 13-3 predicts a decrease in water depth. We note that our knowledge of inviscid flows over bumps (Section 13-9) also predicts that subcritical flows will decrease in depth over the leading edge of a bump. The graphical results confirm this. On the downstream portion of the bump, y_n is real, and we see that we are briefly on an M1 profile, with increasing water depth. Finally, once the channel bottom again becomes horizontal, $y_n \to \infty$ and we are on an M2 profile with very slightly decreasing water depth. Downstream of the bump, the flow depth continues to decrease on an H2 profile. If friction had been omitted, the water surface would return to the initial elevation.

13-90

Solution Gradually varied flow of water in a wide rectangular channel with a break in channel slope is considered. The normal and critical flow depths in the two segments are to be determined, and the water surface profile is to be plotted and classified.

Assumptions **1** The channel is wide, and the flow is gradually varied. **2** The bottom slope is constant in each of the two segments. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

Properties The Manning coefficient of the channel is given to be $n = 0.02$.

Analysis (*a*) The channel is said to be wide, and thus the hydraulic radius is equal to the flow depth, $R_h \cong y$. Knowing the flow rate per unit width ($b = 1$ m), the normal depth is determined from the Manning equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n}(yb) y^{2/3} S_0^{1/2} = \frac{a}{n} b y^{5/3} S_0^{1/2}$$

Mild segment: $$y_{n1} = \left(\frac{(\dot{V}/b)n}{aS_{01}^{1/2}}\right)^{3/5} = \left(\frac{(5\text{ m}^2/\text{s})(0.02)}{(1\text{ m}^{1/3}/\text{s})(0.01)^{1/2}}\right)^{3/5} = \mathbf{1.00\,m}$$

Steep segment: $$y_{n2} = \left(\frac{(\dot{V}/b)n}{aS_{02}^{1/2}}\right)^{3/5} = \left(\frac{(5\text{ m}^2/\text{s})(0.02)}{(1\text{ m}^{1/3}/\text{s})(0.02)^{1/2}}\right)^{3/5} = \mathbf{0.81m}$$

The critical depth for this flow is

$$y_c = \frac{\dot{V}^2}{gA_c^2} = \frac{\dot{V}^2}{g(by)^2} \quad \rightarrow \quad y_c = \left(\frac{(\dot{V}/b)^2}{g}\right)^{1/3} = \left(\frac{(5\text{ m}^2/\text{s})^2}{(9.81\text{ m/s}^2)}\right)^{1/3} = \mathbf{1.37m}$$

Comparing these three depth values, we see that our open channel flow must be classified as **steep** for both channel segments, since $y_n < y_c$.

(*b*) Knowing the initial condition $y(0) = 1.25$ m, the flow depth y at any x location can be determined by numerical integration of the GVF equation

$$\frac{dy}{dx} = \frac{S_0 - S_f}{1 - \text{Fr}^2}$$

where the Froude number for a wide rectangular channel is

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{\dot{V}/by}{\sqrt{gy}} = \frac{\dot{V}/b}{\sqrt{gy^3}}$$

and the friction slope is determined from the uniform-flow equation by setting $S_0 = S_f$,

$$\dot{V} = \frac{a}{n} b y^{5/3} S_f^{1/2} \quad \rightarrow \quad S_f = \left(\frac{(\dot{V}/b)n}{ay^{5/3}}\right)^2 = \frac{(\dot{V}/b)^2 n^2}{a^2 y^{10/3}}$$

Substituting, the GVF equation for a wide rectangular channel becomes

$$\frac{dy}{dx} = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1 - (\dot{V}/b)^2/(gy^3)}$$

which is highly nonlinear, and thus difficult to integrate analytically. The solution of the nonlinear first order differential equation subject to the initial condition $y(x_1) = y_1$ can be expressed as

$$y = y_1 + \int_{x_1}^{x_2} f(x,y)dx \qquad \text{where} \qquad f(x,y) = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1 - (\dot{V}/b)^2/(gy^3)}$$

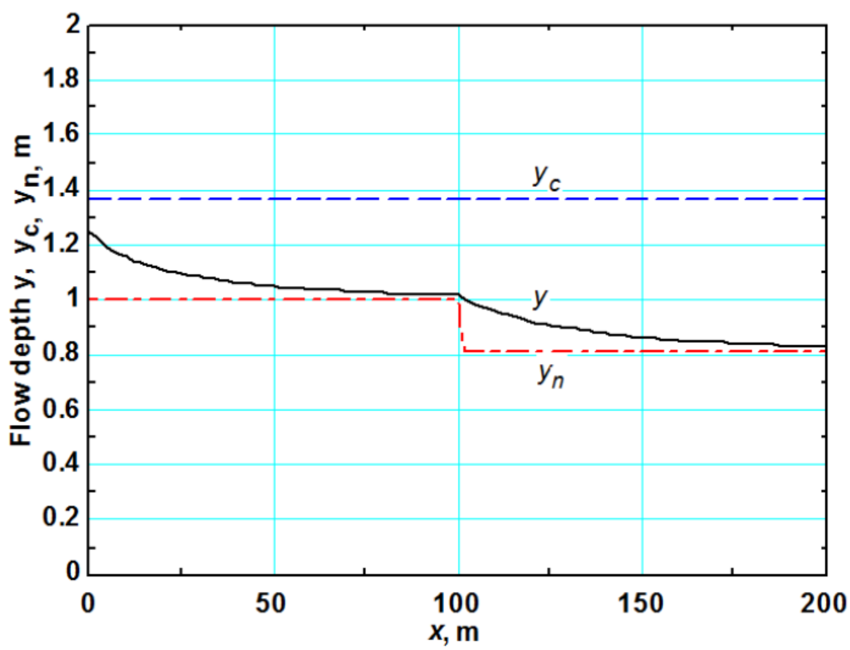
where $y = y(x)$ is the water depth at the specified location x. For given numerical values, this problem can be solved using EES as follows:

```
Function Slope(x,S01,S02)
```

```
If (x<= 100) Then Slope:=S01 Else Slope:=S02
End

Function Yn(x2,y_n1,y_n2)
If (x2<= 100) Then Yn:=y_n1 Else Yn:=y_n2
End

Vol=5 "m^3/s, volume flow rate per unit width, b = 1 m"
b=1 "m. width of channel"
n=0.02 "Manning coefficient for the channel"
S01=0.01 "Channel slope for mild segment"
S02=0.02 "Channel slope for steep segment"
g=9.81 "gravitational acceleration, m/s^2"

y_c=(Vol^2/(g*b^2))^(1/3) "Critical depth"
y_n1=(Vol*n/(b*S01^0.5))^(3/5) "Normal depth for channel section 1"
y_n2=(Vol*n/(b*S02^0.5))^(3/5) "Normal depth for channel section 1"
y_n=Yn(x2,y_n1,y_n2)

x1=0; y1=1.25 "m, initial condition"
x2=10 "m, length of channel"

S_0=Slope(x,S01,S02)
f_xy=(S_0-((Vol/b)^2*n^2/y^(10/3)))/(1-(Vol/b)^2/(g*y^3)) "the GVF equation to be integrated"
y=y1+integral(f_xy, x, x1, x2) "integral equation, auto step: Press F2 to solve."
```

Copying and pasting the mini program above into a blank EES screen gives the water depth at a location of 10 m to be $y(x_2) = y(10\text{ m}) = 1.16$ m. Repeating calculations for different x_2 values and tabulating and plotting, we get

Distance along channel x, m	Flow depth y, m	Normal depth y_n, m
0	1.25	1.00
10	1.16	1.00
20	1.11	1.00
30	1.08	1.00
40	1.06	1.00
50	1.05	1.00
60	1.04	1.00
70	1.03	1.00
80	1.02	1.00
90	1.02	1.00
100	1.02	1.00
110	0.96	0.81
120	0.92	0.81
130	0.90	0.81
140	0.88	0.81
150	0.86	0.81
160	0.85	0.81
170	0.84	0.81
180	0.84	0.81
190	0.83	0.81
200	0.83	0.81



Discussion This problem deals with the GVF equation for the case where there is a break in channel slope. The flow behavior depends strongly upon the initial depth. The calculated results agree with our understanding of flow behavior as illustrated in Table 13-3. For the initial water depth of 1.25 m, we are on an S2 profile and the flow depth will decrease towards the normal depth of the first channel segment. At the change in slope, the normal depth changes, but the critical depth does not. The water surface profile will remain on an S2 curve, and the flow depth will continue to decrease as the new normal depth is approached.

13-91

Solution Gradually varied flow of water in a wide rectangular channel with a break in channel slope is considered. The normal and critical flow depths in the two segments are to be determined, and the water surface profile is to be plotted and classified.

Assumptions **1** The channel is wide, and the flow is gradually varied. **2** The bottom slope is constant in each of the two segments. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

Properties The Manning coefficient of the channel is given to be $n = 0.02$.

Analysis (*a*) The channel is said to be wide, and thus the hydraulic radius is equal to the flow depth, $R_h \cong y$. Knowing the flow rate per unit width (b = 1 m), the normal depth is determined from the Manning equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n}(yb)y^{2/3} S_0^{1/2} = \frac{a}{n} by^{5/3} S_0^{1/2}$$

Mild segment: $$y_{n1} = \left(\frac{(\dot{V}/b)n}{aS_{01}^{1/2}}\right)^{3/5} = \left(\frac{(5\text{ m}^2/\text{s})(0.02)}{(1\text{ m}^{1/3}/\text{s})(0.01)^{1/2}}\right)^{3/5} = \mathbf{1.00\,m}$$

Steep segment: $$y_{n2} = \left(\frac{(\dot{V}/b)n}{aS_{02}^{1/2}}\right)^{3/5} = \left(\frac{(5\text{ m}^2/\text{s})(0.02)}{(1\text{ m}^{1/3}/\text{s})(0.02)^{1/2}}\right)^{3/5} = \mathbf{0.81\,m}$$

The critical depth for this flow is

$$y_c = \frac{\dot{V}^2}{gA_c^2} = \frac{\dot{V}^2}{g(by)^2} \quad \rightarrow \quad y_c = \left(\frac{(\dot{V}/b)^2}{g}\right)^{1/3} = \left(\frac{(5\text{ m}^2/\text{s})^2}{(9.81\text{ m/s}^2)}\right)^{1/3} = \mathbf{1.37\,m}$$

Comparing these three depth values, we see that our open channel flow must be classified as **steep** for both channel segments, since $y_n < y_c$.

(*b*) Knowing the initial condition $y(0) = 0.75$ m, the flow depth y at any x location can be determined by numerical integration of the GVF equation

$$\frac{dy}{dx} = \frac{S_0 - S_f}{1 - \text{Fr}^2}$$

where the Froude number for a wide rectangular channel is

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{\dot{V}/by}{\sqrt{gy}} = \frac{\dot{V}/b}{\sqrt{gy^3}}$$

and the friction slope is determined from the uniform-flow equation by setting $S_0 = S_f$,

$$\dot{V} = \frac{a}{n} by^{5/3} S_f^{1/2} \quad \rightarrow \quad S_f = \left(\frac{(\dot{V}/b)n}{ay^{5/3}}\right)^2 = \frac{(\dot{V}/b)^2 n^2}{a^2 y^{10/3}}$$

Substituting, the GVF equation for a wide rectangular channel becomes

$$\frac{dy}{dx} = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1 - (\dot{V}/b)^2/(gy^3)}$$

which is highly nonlinear, and thus difficult to integrate analytically. The solution of the nonlinear first order differential equation subject to the initial condition $y(x_1) = y_1$ can be expressed as

$$y = y_1 + \int_{x_1}^{x_2} f(x, y)dx \qquad \text{where} \qquad f(x, y) = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1 - (\dot{V}/b)^2/(gy^3)}$$

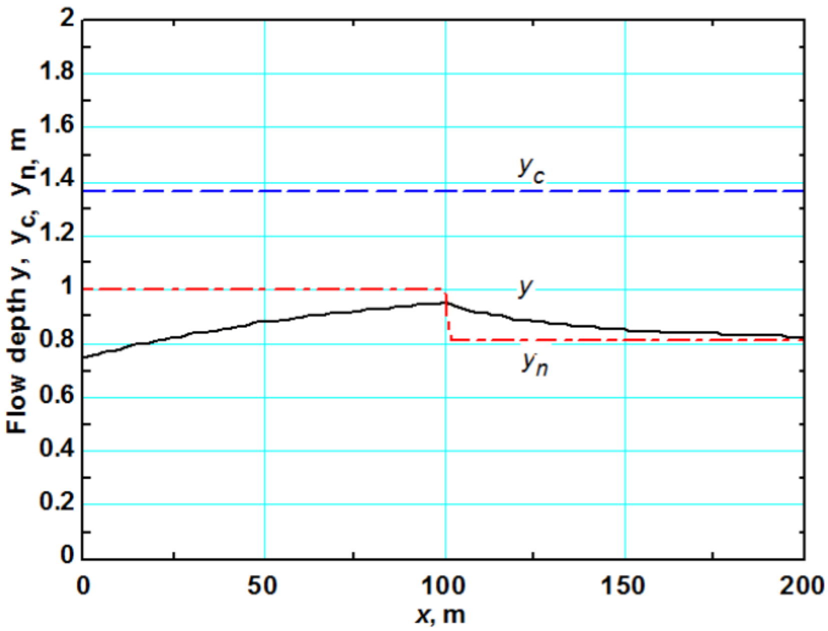
where $y = y(x)$ is the water depth at the specified location x. For given numerical values, this problem can be solved using EES as follows:

```
Function Slope(x,S01,S02)
If (x<= 100) Then Slope:=S01 Else Slope:=S02
End
```

```
Function Yn(x2,y_n1,y_n2)
If (x2<= 100) Then Yn:=y_n1 Else Yn:=y_n2
End

Vol=5 "m^3/s, volume flow rate per unit width, b = 1 m"
b=1 "m. width of channel"
n=0.02 "Manning coefficient for the channel"
S01=0.01 "Channel slope for mild segment"
S02=0.02 "Channel slope for steep segment"
g=9.81 "gravitational acceleration, m/s^2"

y_c=(Vol^2/(g*b^2))^(1/3) "Critical depth"
y_n1=(Vol*n/(b*S01^0.5))^(3/5) "Normal depth for channel section 1"
y_n2=(Vol*n/(b*S02^0.5))^(3/5) "Normal depth for channel section 1"
y_n=Yn(x2,y_n1,y_n2)

x1=0; y1=0.75 "m, initial condition"
x2=10 "m, length of channel"

S_0=Slope(x,S01,S02)
f_xy=(S_0-((Vol/b)^2*n^2/y^(10/3)))/(1-(Vol/b)^2/(g*y^3)) "the GVF equation to be integrated"
y=y1+integral(f_xy, x, x1, x2) "integral equation, auto step: Press F2 to solve."
```

Copying and pasting the mini program above into a blank EES screen gives the water depth at a location of 10 m to be $y(x_2) = y(10\text{ m}) = 1.16$ m. Repeating calculations for different x_2 values and tabulating and plotting, we get

Distance along channel x, m	Flow depth y, m	Normal depth y_n, m
0	0.75	1.00
10	0.78	1.00
20	0.81	1.00
30	0.83	1.00
40	0.86	1.00
50	0.88	1.00
60	0.90	1.00
70	0.91	1.00
80	0.93	1.00
90	0.94	1.00
100	0.95	1.00
110	0.91	0.81
120	0.89	0.81
130	0.87	0.81
140	0.86	0.81
150	0.85	0.81
160	0.84	0.81
170	0.84	0.81
180	0.83	0.81
190	0.83	0.81
200	0.82	0.81



Discussion This problem deals with the GVF equation for the case where there is a break in channel slope. The flow behavior depends strongly upon the initial depth. The calculated results agree with our understanding of flow behavior as illustrated in Table 13-3. For the initial water depth of 0.75 m, we begin on an S3 curve. Provided the depth has increased to at least 0.81 m (y_n on segment 2) by the time the change in slope is encountered, we then will be on an S2 curve.

13-92

Solution A hydraulic jump that occurs in a wide rectangular channel is considered. The critical flow depth is to be determined, and it is to be verifed that the initial and final flows are supercritical and subcritical, respectively, and the location of the jump is to be predicted.

In this problem, the GVF equation is solved for the case of a hydraulic jump in a horizontal channel. The inlet and outlet depths are specified, and we are to predict where the hydraulic jump will occur.

Assumptions **1** The channel is wide, and the flow is gradually varied upstream and downstream of the jump. **2** The hydraulic jump has zero streamwise length, i.e. it is a discontinuity. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

Properties The Manning coefficient of the channel is given to be n = 0.009.

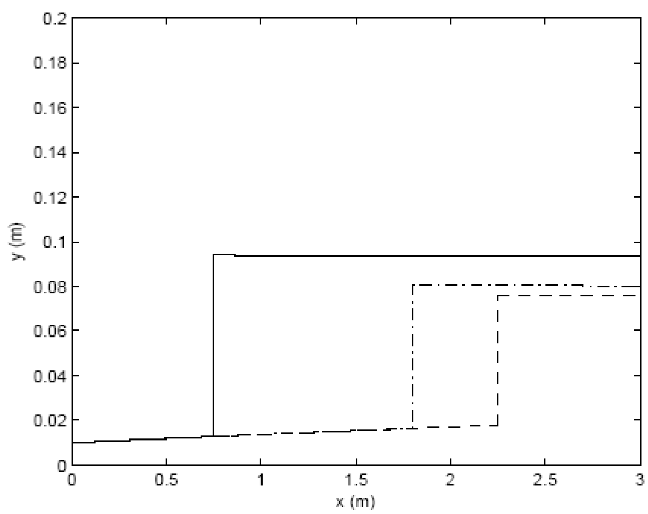
Analysis (a) The channel is said to be wide, and thus the hydraulic radius is equal to the flow depth, $R_h \cong y$. Knowing the flow rate per unit width (b = 1 m), the critical depth of this flow is determined to be

$$y_c = \frac{\dot{V}^2}{gA_c^2} = \frac{\dot{V}^2}{g(by)^2} \quad \rightarrow \quad y_c = \left(\frac{(\dot{V}/b)^2}{g}\right)^{1/3} = \left(\frac{(0025\,\mathrm{m}^2/\mathrm{s})^2}{(9.81\,\mathrm{m/s}^2)}\right)^{1/3} = \mathbf{0.040\,m}$$

Thus, we see that our initial flow is indeed supercritical and our final flow is subcritical.

(b) Now we try several jump locations, for example, at x = 0.75 m. In each case, we integrate the GVF equation from the head gate to the jump location and determine the flow depth y_1 before the jump, calculate the inflow Froude number of the jump at that point from $\mathrm{Fr}_1 = \frac{\dot{V}/b}{\sqrt{gy_1^3}}$, use the hydraulic jump equation $\frac{y_2}{y_1} = 0.5\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right)$ to get the downstream (subcritical) flow depth y_2, and continue integrating the GVF equation from the jump location to the tail gate, and compare the calculated flow depth y_3 at x = 3 m to the measured value of 0.08 m. As summarized in the table and figure below, the jump should be located at x = 1.8 m. For given numerical values, this problem can be solved using EES as follows:

```
Vol=0.025 "m^3/s, volume flow rate per unit width, b = 1 m"
b=1 "m. width of channel"
n=0.009 "Manning coefficient for channel"
S_0=0 "Slope of channel"
g=9.81 "gravitational acceleration, m/s^2"

y_c=(Vol^2/(g*b^2))^(1/3) "Critical depth"

x0=0; y0=0.01 "m, initial condition at gate"
x1=0.75 "m, guessed location of jump"
x2=3 "m, total length of channel"

"GVF solution before the jump:"
f_xy1=(S_0-((Vol/b)^2*n^2/y^(10/3)))/(1-(Vol/b)^2/(g*y^3))
y=y0+integral(f_xy1, x, x0, x1) "integral equation-auto step."

y1=y
Fr1=(Vol/b)/SQRT(g*y1^3)
y2=0.5*y1*(-1+SQRT(1+8*Fr1^2))
Fr2=(Vol/b)/SQRT(g*y2^3)

"GVF solution after the jump:"
f_xy2=(S_0-((Vol/b)^2*n^2/yy^(10/3)))/(1-(Vol/b)^2/(g*yy^3))
yy=y2+integral(f_xy2, xx, x1, x2)
y3=yy "Flow depth at end of channel"
```



Jump location x_1, m	Flow depth before jump y_1, m	Froude number Fr_1	Flow depth after jump y_2, m	Froude number Fr_2	Flow depth at channel end y_3, m
0.0	0.010	7.98	0.108	0.22	0.108
0.2	0.011	7.17	0.104	0.24	0.103
0.4	0.011	6.50	0.100	0.25	0.100
0.6	0.012	5.93	0.096	0.27	0.096
0.8	0.013	5.45	0.093	0.28	0.093
1.0	0.014	5.04	0.090	0.29	0.090
1.2	0.014	4.68	0.088	0.31	0.087
1.4	0.015	4.36	0.085	0.32	0.085
1.6	0.016	4.08	0.083	0.34	0.082
1.8	**0.016**	**3.83**	**0.081**	**0.35**	**0.080**
2.0	0.017	3.61	0.079	0.36	0.078
2.2	0.018	3.40	0.077	0.38	0.076
2.4	0.018	3.22	0.075	0.39	0.075
2.6	0.019	3.05	0.073	0.40	0.073
2.8	0.020	2.90	0.071	0.42	0.071
3.0	0.020	2.75	0.070	0.43	0.070

Thus, **the location of the hydraulic jump is at 1.80 m**.

Discussion In all cases, the water depth initially increases on an H3 profile. After the jump, the subcritical flow is characterized by (very slightly) decreasing water depth on an H2 profile. As the jump is positioned closer to the sluice gate, the stronger subcritical flow leads to greater depth ratios across the jump. Alternatively, raising the tailgate will have the effect of pushing the jump closer to the head gate.

13-93

Solution Gradually varied flow of water in a wide rectangular channel is considered. It is to be shown that the slope of the surface is a function of S_0, y, y_n, and y_c alone.

Assumptions **1** The channel is wide, and the flow is gradually varied. **2** The bottom slope is constant. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

Analysis The channel is said to be wide, and thus the hydraulic radius is equal to flow depth, $R_h \cong y$. First we consider the numerator $S_0 - S_f$. Here S_0 is the actual channel slope and would produce a uniform flow depth of y_n. The friction slope S_f, on the other hand, is the slope that would produce uniform flow at the actual flow depth y. Noting that R_h is equivalent to flow depth for a wide rectangular channel, yhe Manning equation simplifies to

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n}(yb)y^{2/3} S_0^{1/2} = \frac{a}{n} b y^{5/3} S_0^{1/2} \rightarrow S_0 = \frac{a^2 \dot{V}^2 n^2}{b^2 y_n^{10/3}} \quad \text{and} \quad S_f = \frac{a^2 \dot{V}^2 n^2}{b^2 y^{10/3}}$$

Therefore,

$$S_0 - S_f = S_0\left(1 - \frac{S_f}{S_0}\right) = S_0\left(1 - \frac{(a^2\dot{V}^2 n^2)/(b^2 y^{10/3})}{(a^2\dot{V}^2 n^2)/(b^2 y_n^{10/3})}\right) = S_0\left[1 - \left(\frac{y_n}{y}\right)^{10/3}\right] \qquad (1)$$

The critical depth for flow in a wide rectangular channel is

$$y_c = \frac{\dot{V}^2}{gA_c^2} = \frac{\dot{V}^2}{g(by_c)^2} = \frac{\dot{V}^2}{gb^2 y_c^2} \quad \rightarrow \quad \frac{\dot{V}^2}{gb^2} = y_c^3$$

Then the Froude number for a wide rectangular channel becomes

$$\mathrm{Fr} = \frac{V}{\sqrt{gy}} = \frac{\dot{V}/by}{\sqrt{gy}} = \frac{\dot{V}/b}{\sqrt{gy^3}} \quad \rightarrow \quad 1\text{-}\mathrm{Fr}^2 = 1 - \frac{\dot{V}^2}{gb^2 y^3} = 1 - \frac{y_c^3}{y^3} = 1 - \left(\frac{y_c}{y}\right)^3 \qquad (2)$$

Substituting Eqs. (1) and (2) in the GVF equation gives the desired result,

$$\boxed{\frac{dy}{dx} = \frac{S_0 - S_f}{1 - \mathrm{Fr}^2} = \frac{S_0\left[1 - (y_n/y)^{10/3}\right]}{1 - (y_c/y)^3}}$$

Discussion This problem simplifies the GVF equation for the special case of a wide rectangular channel. The simplified equation makes explicit the importance of the relationship between y, y_n, and y_c in terms of determining the behavior of the flow. From this modified GVF equation, we now see the explicit relationship between y, y_n, and y_c. The relative magnitudes of these terms determine the signs of the numerator and denominator in Eq. 13-65 and therefore the overall sign of *dy/dx* as discussed in Table 13-3.

13-94E

Solution Gradually varied flow of water in a wide rectangular channel is considered. The classification the flow type and the flow depths at specified locations are to be determined.

Assumptions **1** The channel is wide, and the flow is gradually varied. **2** The bottom slope is constant. **3** The roughness of the wetted surface of the channel and thus the friction coefficient are constant.

Properties The Manning coefficient of the channel is given to be $n = 0.008$.

Analysis The channel is said to be wide, and thus the hydraulic radius is equal to the flow depth, $R_h \cong y$. The normal depth is determined from the Manning equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n}(yb)y^{2/3} S_0^{1/2} = \frac{a}{n} b y^{5/3} S_0^{1/2}$$

$$y_n = \left(\frac{\dot{V}n}{abS_0^{1/2}}\right)^{3/5} = \left(\frac{(300 \text{ ft}^3/\text{s})(0.008)}{(1.486 \text{ ft}^{1/3}/\text{s})(20 \text{ ft})(0.01)^{1/2}}\right)^{3/5} = 1.52 \text{ ft}$$

The critical depth for this flow is

$$y_c = \frac{\dot{V}^2}{gA_c^2} = \frac{\dot{V}^2}{g(by)^2} \rightarrow y_c = \left(\frac{\dot{V}^2}{b^2 g}\right)^{1/3} = \left(\frac{(300 \text{ ft}^3/\text{s})^2}{(20 \text{ ft})^2 (32.2 \text{ ft/s}^2)}\right)^{1/3} = 1.91 \text{ ft}$$

The flow depth at $x = 0$ is

$$y_1 = \frac{\dot{V}}{bV_1} = \frac{300 \text{ ft}^3/\text{s}}{(20 \text{ ft})(5.2 \text{ ft/s})} = 2.89 \text{ ft}$$

Noting that $y_c < y_n < y$ at $x = 0$, we see from Table 13-3 that the water surface profile during this GVF is classified as **M1**.

(a) Knowing the initial condition $y(0) = y_1 = 2.89$ ft, the flow depth y at any x location can be determined by numerical integration of the GVF equation

$$\frac{dy}{dx} = \frac{S_0 - S_f}{1 - \text{Fr}^2}$$

where the Froude number for a wide rectangular channel is

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{\dot{V}/by}{\sqrt{gy}} = \frac{\dot{V}/b}{\sqrt{gy^3}}$$

and the friction slope is determined from the uniform-flow equation by setting $S_0 = S_f$,

$$\dot{V} = \frac{a}{n} b y^{5/3} S_f^{1/2} \rightarrow S_f = \left(\frac{(\dot{V}/b)n}{ay^{5/3}}\right)^2 = \frac{(\dot{V}/b)^2 n^2}{a^2 y^{10/3}}$$

Substituting, the GVF equation for a wide rectangular channel becomes

$$\frac{dy}{dx} = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1 - (\dot{V}/b)^2/(gy^3)}$$

which is highly nonlinear, and thus difficult to integrate analytically. The solution of the nonlinear first order differential equation subject to the initial condition $y(x_1) = y_1$ can be expressed as

$$y = y_1 + \int_{x_1}^{x_2} f(x, y)dx \qquad \text{where} \qquad f(x, y) = \frac{S_0 - (\dot{V}/b)^2 n^2/(a^2 y^{10/3})}{1 - (\dot{V}/b)^2/(gy^3)}$$

where $y = y(x)$ is the water depth at the specified location x. For given numerical values, this problem can be solved using EES as follows:

```
Vol=300 "ft^3/s, volume flow rate"
b=20 "ft. width of channel"
n=0.02 "Manning coefficient"
a=1.486
S_0=0.01 "Slope of channel"
g=32.2 "gravittational acceleration, ft/s^2"
Vel1=5.2 "ft/s"

y_c=(Vol^2/(g*b^2))^(1/3) "Critical depth"
y_n=(Vol*n/(a*b*S_0^0.5))^(3/5) "Normal depth"

x1=0; y1=Vol/(Vel1*b) "ft, initial condition"
x2=500 "ft, lenght of channel"

f_xy=(S_0-((Vol/b)^2*n^2/a^2/y^(10/3)))/(1-(Vol/b)^2/(g*y^3)) "the GVF equation to be integrated"
y=y1+integral(f_xy, x, x1, x2) "integral equation, auto step: Press F2 to solve."
```

Copying and pasting the mini program above into a blank EES screen gives the water depth at a location of 500 ft to be

$y(x_2) = y(500\text{ ft}) = \mathbf{8.13\ ft}$

(*b*), (*c*) Similarly, the flow depths at x = 1000 ft and x = 2000 ft are determined to be

$y(x_3) = y(1000\text{ ft}) = \mathbf{13.2\ ft}$

$y(x_4) = y(2000\text{ ft}) = \mathbf{23.2\ ft}$

Discussion The above results confirm the quantitative prediction from Table 13-3 that an M1 profile should yield increasing water depth in the downstream direction.

13-95

Solution Gradually varied flow of water in a wide rectangular irrigation channel with a rough flow section is considered. The normal and critical flow depths in both smooth and rough segments are to be determined, and the water surface profile is to be plotted.

Assumptions **1** The channel is wide, and the flow is gradually varied. **2** The bottom slope is constant. **3** The roughness of the wetted surface of the channel and thus the friction coefficient in each segment are constant.

Properties The Manning coefficient of the channel is given to be n = 0.02 in the smooth section, and 0.03 in the rough section.

Analysis (a) The channel is said to be wide, and thus the hydraulic radius is equal to the flow depth, $R_h \cong y$. Knowing the flow rate per unit width (b = 1 m), the normal depth is determined from the Manning equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n}(yb) y^{2/3} S_0^{1/2} = \frac{a}{n} b y^{5/3} S_0^{1/2}$$

Rough: $$y_{n1} = \left(\frac{(\dot{V}/b) n_1}{a S_0^{1/2}} \right)^{3/5} = \left(\frac{(5\ \text{m}^2/\text{s})(0.03)}{(1\ \text{m}^{1/3}/\text{s})(0.01)^{1/2}} \right)^{3/5} = \mathbf{1.28\ m}$$

Smooth: $$y_{n2} = \left(\frac{(\dot{V}/b) n_2}{a S_0^{1/2}} \right)^{3/5} = \left(\frac{(5\ \text{m}^2/\text{s})(0.02)}{(1\ \text{m}^{1/3}/\text{s})(0.01)^{1/2}} \right)^{3/5} = \mathbf{1.00\ m}$$

The critical depth for this flow is

$$y_c = \frac{\dot{V}^2}{g A_c^2} = \frac{\dot{V}^2}{g (by)^2} \quad \rightarrow \quad y_c = \left(\frac{(\dot{V}/b)^2}{g} \right)^{1/3} = \left(\frac{(5\ \text{m}^2/\text{s})^2}{(9.81\ \text{m/s}^2)} \right)^{1/3} = \mathbf{1.37\ m}$$

The flow is initially uniform, and thus $y(0) = y_{n2}$ = 1.0 m at x = 0.

(b) Knowing the initial condition $y(0)$ = 1.0 m, the flow depth y at any x location can be determined by numerical integration of the GVF equation

$$\frac{dy}{dx} = \frac{S_0 - S_f}{1 - \text{Fr}^2}$$

where the Froude number for a wide rectangular channel is

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{\dot{V}/by}{\sqrt{gy}} = \frac{\dot{V}/b}{\sqrt{gy^3}}$$

and the friction slope is determined from the uniform-flow equation by setting $S_0 = S_f$,

$$\dot{V} = \frac{a}{n} b y^{5/3} S_f^{1/2} \quad \rightarrow \quad S_f = \left(\frac{(\dot{V}/b) n}{a y^{5/3}} \right)^2 = \frac{(\dot{V}/b)^2 n^2}{a^2 y^{10/3}}$$

Substituting, the GVF equation for a wide rectangular channel becomes

$$\frac{dy}{dx} = \frac{S_0 - (\dot{V}/b)^2 n^2 / (a^2 y^{10/3})}{1 - (\dot{V}/b)^2 / (g y^3)}$$

which is highly nonlinear, and thus difficult to integrate analytically. The solution of the nonlinear first order differential equation subject to the initial condition $y(x_1) = y_1$ can be expressed as

$$y = y_1 + \int_{x_1}^{x_2} f(x, y) dx \qquad \text{where} \qquad f(x, y) = \frac{S_0 - (\dot{V}/b)^2 n^2 / (a^2 y^{10/3})}{1 - (\dot{V}/b)^2 / (g y^3)}$$

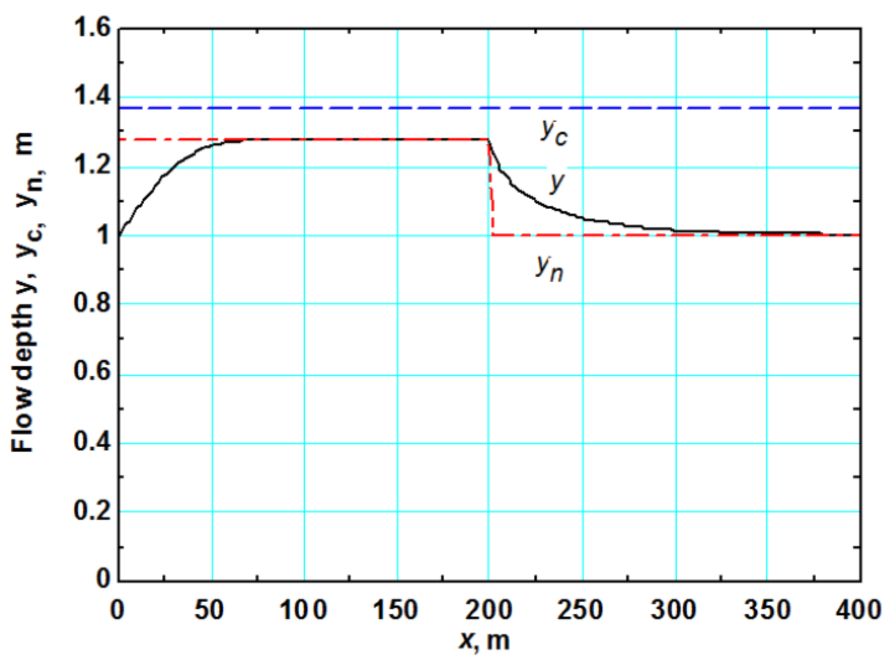
where $y = y(x)$ is the water depth at the specified location x. For given numerical values, this problem can be solved using EES as follows:

```
Function Manning(x,n1,n2)
If (x<= 200) Then Manning:=n1 Else Manning:=n2
```

```
End

Function Yn(x2,y_n1,y_n2)
If (x2<= 200) Then Yn:=y_n1 Else Yn:=y_n2
End

Vol=5 "m^3/s, volume flow rate per unit width, b = 1 m"
b=1 "m. width of channel"
n1=0.03 "Manning coefficient for rough channel segment"
n2=0.02 "Manning coefficient for smoother channel segment"
S_0=0.01 "Slope of channel"
g=9.81 "gravitational acceleration, m/s^2"

y_c=(Vol^2/(g*b^2))^(1/3) "Critical depth"
y_n1=(Vol*n1/(b*S_0^0.5))^(3/5) "Normal depth for channel section 1"
y_n2=(Vol*n2/(b*S_0^0.5))^(3/5) "Normal depth for channel section 1"
y_n=Yn(x2,y_n1,y_n2)

x1=0; y1=y_n2 "m, initial condition - uniform flow depth for smooth section"
x2=220 "m, length of channel"

n=Manning(x,n1,n2)
f_xy=(S_0-((Vol/b)^2*n^2/y^(10/3)))/(1-(Vol/b)^2/(g*y^3)) "the GVF equation to be integrated"
y=y1+integral(f_xy, x, x1, x2) "integral equation, auto step: Press F2 to solve."
```

Copying and pasting the mini program above into a blank EES screen gives the water depth at a location of 220 m to be $y(x_2) = y(220 \text{ m}) = 1.11$ m. Repeating calculations for different x_2 values and tabulating and plotting, we get

Distance along channel x, m	Flow depth y, m
0	1.00
20	1.14
40	1.24
60	1.27
80	1.27
100	1.28
120	1.28
140	1.28
160	1.28
180	1.28
200	1.28
220	1.11
240	1.07
260	1.04
280	1.02
300	1.02
320	1.01
340	1.01
360	1.00
380	1.00
400	1.00



Discussion The graphical result shows that the flow is supercritical over the entire domain. Upon beginning the rough section of channel, the normal depth jumps upward and the water surface climbs toward this new value on an S3 curve. Upon returning to smoother conditions, the water surface descends on an S2 curve toward the original normal depth.

Flow Control and Measurement in Channels

13-96C

Solution We are to define and classify sharp-crested weirs.

Analysis A *sharp-crested weir* is a **vertical plate placed in a channel that forces the fluid to flow through an opening to measure the flow rate**. They are **characterized by the shape of the opening**. For example, a weir with a triangular opening is referred to as a triangular weir.

Discussion Similar to the broad-crested weir, this type of flow measurement is quite obtrusive, but requires no special measuring equipment or probes.

13-97C

Solution We are to discuss how flow rate is measured with a broad-crested weir.

Analysis The operation of broad crested weir is based on **blocking the flow in the channel with a rectangular block, and establishing critical flow over the block**. Then the flow rate is determined by measuring flow depths.

Discussion This technique is quite obtrusive, but requires no special measuring equipment or probes.

13-98C

Solution We are to define the discharge coefficient for sluice gates, and discuss some typical values.

Analysis For sluice gates, the *discharge coefficient* C_d is defined as **the ratio of the actual velocity through the gate to the maximum velocity** as determined by the Bernoulli equation for the idealized frictionless flow case. **For ideal flow,** $\boldsymbol{C_d = 1}$. Typical values of C_d for sluice gates with free outflow are in the range of **0.55 to 0.60**.

Discussion Actual values of the discharge coefficient must be less than one or else the second law would be violated.

13-99C

Solution We are to analyze whether the free surface of flow over a bump will increase, decrease, or remain constant.

Analysis In the case of *subcritical flow*, the **flow depth *y* will decrease during flow over the bump**.

Discussion This may be contrary to our intuition at first, but if we think in terms of increasing velocity and decreasing pressure over the bump (a Bernoulli type of analysis), it makes sense that the surface will decrease over the bump.

13-100C

Solution We are to analyze what happens in subcritical flow over a bump when the bump height increases.

Analysis When the specific energy reaches its minimum value, **the flow is critical, and the flow at this point is said to be choked**. If the bump height is increased even further, **the flow remains critical and thus choked. The flow will not become supercritical**.

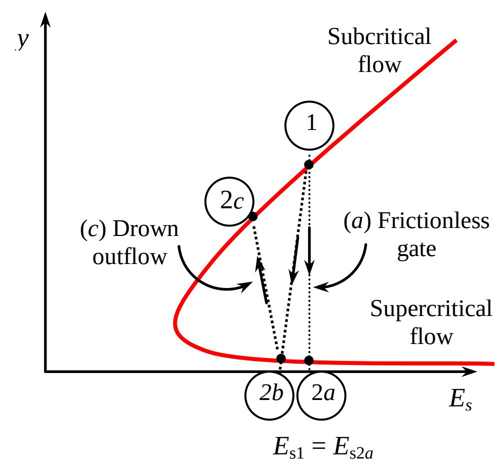
Discussion This is somewhat analogous to compressible flow in a converging nozzle – the flow cannot become supersonic at the nozzle exit unless there is a diverging section of the nozzle downstream of the throat.

13-101C

Solution We are to draw a flow depth-specific energy diagram for several types of flow.

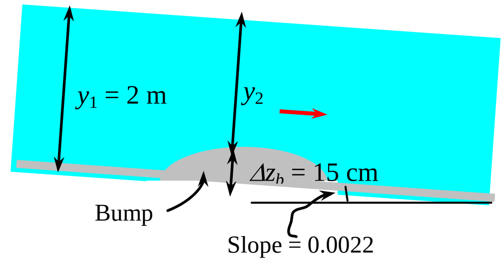
Analysis On the figure, diagram 1-2a is for frictionless gate, **1-2b is for sluice gate with free outflow**, and **1-2b-2c is for sluice gate with drown outflow, including the hydraulic jump back to subcritical flow**.

Discussion A plot of flow depth as a function of specific energy, as shown here, is quite useful in the analysis of varied open-channel flow because the states upstream and downstream of a change must jump between the two branches.



13-102

Solution The flow of water in a wide channel with a bump is considered. The flow rate of water without the bump and the effect of the bump on the flow rate for the case of a flat surface are to be determined.



Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel. **4** The channel is sufficiently wide so that the end effects are negligible. **5** Frictional effects during flow over the bump are negligible.

Properties Manning coefficient for an open channel of unfinished concrete is n = 0.014 (Table 13-1).

Analysis For a wide channel, the hydraulic radius is equal to the flow depth, and thus R_h = 2 m. Then the flow rate *before the bump* per m width (i.e., b = 1 m) can be determined from Manning's equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{1\,\text{m}^{1/3}/s}{0.014}(1\times 2\,\text{m}^2)(2\,\text{m})^{2/3}(0.0022)^{1/2} = 10.64\,\text{m}^3/\text{s}$$

The average flow velocity is $V = \dfrac{\dot{V}}{A_c} = \dfrac{10.64\,\text{m}^3/\text{s}}{1\times 2\,\text{m}^2} = 5.32\,\text{m/s}$.

When a bump is placed, it is said that the flow depth remains the same and there is no rise/drop, and thus $y_2 = y_1 - \Delta z_b$. But the energy equation is given as

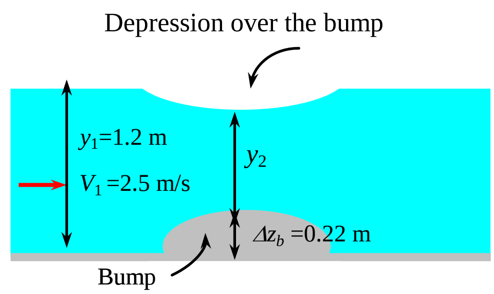
$$E_{s2} = E_{s1} - \Delta z_b \quad \rightarrow \quad y_2 + \frac{V_2^2}{2g} = y_1 + \frac{V_1^2}{2g} - \Delta z_b \quad \rightarrow \quad \frac{V_2^2}{2g} = \frac{V_1^2}{2g}$$

since $y_2 = y_1 - \Delta z_b$, and thus $V_1 = V_2$. But from the continuity equation $y_2 V_2 = y_1 V_1$, this is possible only if the flow depth over the bump remains constant, i.e., $y_1 = y_2$, which is a contradiction since y_2 cannot be equal to both y_1 and $y_1 - \Delta z_b$ while Δz_b remains nonzero. Therefore, the second part of the problem can have **no solution** since it is physically impossible.

Discussion Note that sometimes it is better to investigate whether there is really a solution before spending a lot of time trying to find a solution.

13-103

Solution Water flowing in a horizontal open channel encounters a bump. It will be determined if the flow over the bump is choked.



Assumptions **1** The flow is steady. **2** Frictional effects are negligible so that there is no dissipation of mechanical energy. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The upstream Froude number and the critical depth are

$$\mathrm{Fr}_1 = \frac{V_1}{\sqrt{gy_1}} = \frac{2.5\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m^2/s})(1.2\ \mathrm{m})}} = 0.729$$

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(by_1V_1)^2}{gb^2}\right)^{1/3} = \left(\frac{y_1^2V_1^2}{g}\right)^{1/3} = \left(\frac{(1.2\ \mathrm{m})^2(2.5\ \mathrm{m/s})^2}{9.81\ \mathrm{m/s^2}}\right)^{1/3} = 0.972\ \mathrm{m}$$

The flow is subcritical since Fr < 1, and the flow depth decreases over the bump. The upstream, over the bump, and critical specific energies are

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = (1.2\ \mathrm{m}) + \frac{(2.5\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})} = 1.52\ \mathrm{m}$$

$$E_{s2} = E_{s1} - \Delta z_b = 1.52 - 0.22 = 1.30\ \mathrm{m}$$

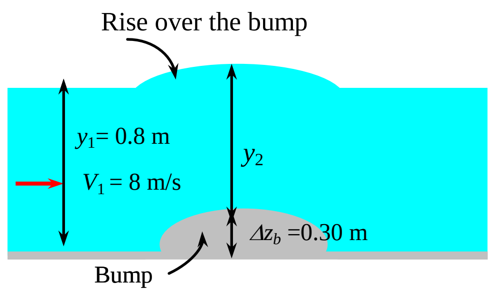
$$E_c = \frac{3}{2}y_c = 1.46\ \mathrm{m}$$

We have an interesting situation: The calculations show that $E_{s2} < E_c$. That is, the specific energy of the fluid decreases below the level of energy at the critical point, which is the minimum energy, and this is impossible. Therefore, the flow at specified conditions cannot exist. **The flow is choked** when the specific energy drops to the minimum value of 1.46 m, which occurs at a bump-height of $\Delta z_{b,\max} = E_{s1} - E_c = 1.52 - 1.46 = 0.06\ \mathrm{m}$.

Discussion A bump-height over 6 cm results in a reduction in the flow rate of water, or a rise of upstream water level. Therefore, a 22-cm high bump alters the upstream flow. On the other hand, a bump less than 6 cm high will not affect the upstream flow.

13-104

Solution Water flowing in a horizontal open channel encounters a bump. The change in the surface level over the bump and the type of flow (sub- or supercritical) over the bump are to be determined.



Assumptions **1** The flow is steady. **2** Frictional effects are negligible so that there is no dissipation of mechanical energy. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The upstream Froude number and the critical depth are

$$\mathrm{Fr}_1 = \frac{V_1}{\sqrt{gy_1}} = \frac{8\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m^2/s})(0.8\ \mathrm{m})}} = 2.856$$

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(by_1V_1)^2}{gb^2}\right)^{1/3} = \left(\frac{y_1^2V_1^2}{g}\right)^{1/3} = \left(\frac{(0.8\ \mathrm{m})^2(8\ \mathrm{m/s})^2}{9.81\ \mathrm{m/s^2}}\right)^{1/3} = 1.61\ \mathrm{m}$$

The upstream flow is supercritical since Fr > 1, and the flow depth increases over the bump. The upstream, over the bump, and critical specific energies are

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = (0.8\ \mathrm{m}) + \frac{(8\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})} = 4.06\ \mathrm{m}$$

$$E_{s2} = E_{s1} - \Delta z_b = 4.06 - 0.30 = 3.76\ \mathrm{m}$$

$$E_c = \frac{3}{2}y_c = 2.42\ \mathrm{m}$$

The flow depth over the bump is determined from

$$y_2^3 - (E_{s1} - \Delta z_b)y_2^2 + \frac{V_1^2}{2g}y_1^2 = 0 \quad \rightarrow \quad y_2^3 - (4.06 - 0.30\ \mathrm{m})y_2^2 + \frac{(8\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})}(0.80\ \mathrm{m})^2 = 0$$

Using an equation solver, the physically meaningful root of this equation is determined to be 0.846 m. Therefore, there is a rise of

$$\text{Rise over bump} = y_2 - y_1 + \Delta z_b = 0.846 - 0.80 + 0.30 = \mathbf{0.346 m}$$

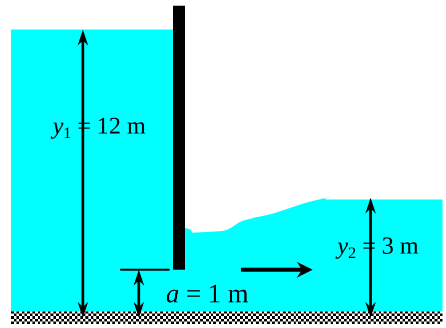
over the surface relative to the upstream water surface. The specific energy decreases over the bump from, 4.06 to 3.76 m, but it is still over the minimum value of 2.42 m. Therefore, the flow over the bump is still **supercritical**.

Discussion The actual value of surface rise may be different than 4.6 cm because of frictional effects that are neglected in this simplified analysis.

13-105

Solution Water is released from a reservoir through a sluice gate into an open channel. For specified flow depths, the rate of discharge is to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.



Analysis The depth ratio y_1/a and the contraction coefficient y_2/a are

$$\frac{y_1}{a}=\frac{12\text{ m}}{1\text{ m}}=12 \quad \text{and} \quad \frac{y_2}{a}=\frac{3\text{ m}}{1\text{ m}}=3$$

The corresponding discharge coefficient is determined from Fig. 13-44 to be C_d = 0.59. Then the discharge rate becomes

$$\dot{V}=C_d ba\sqrt{2gy_1}=0.59(6\text{ m})(1\text{ m})\sqrt{2(9.81\text{ m/s}^2)(12\text{ m})}=\mathbf{54.3\ m^3/s}$$

Discussion Discharge coefficient is the same as free flow because of small depth ratio after the gate. So, the flow rate would not change if it were not drowned.

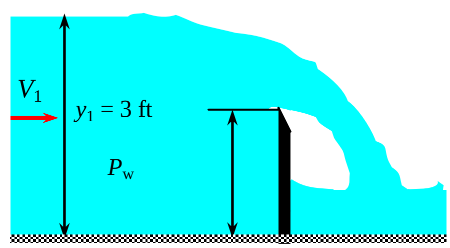
13-106E

Solution The flow rate in an open channel is to be measured using a sharp-crested rectangular weir. For specified upper limits of flow rate and flow depth, the appropriate height of the weir is to be determined.

Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The weir head is $H=y_1-P_w=3-P_w$. The discharge coefficient of the weir is

$$C_{wd,\text{rec}}=0.598+0.0897\frac{H}{P_w}=0.598+0.0897\frac{3-P_w}{P_w}$$



Sharp-crested rectangular weir

The water flow rate through the channel can be expressed as

$$\dot{V}_\text{rec}=C_{wd,\text{rec}}\frac{2}{3}b\sqrt{2g}H^{3/2}$$

Substituting the known quantities,

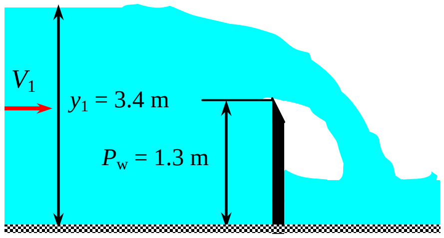
$$180\text{ ft}^3/\text{s}=\left(0.598+0.0897\frac{3-P_w}{P_w}\right)\frac{2}{3}(7\text{ ft})\sqrt{2(32.2\text{ ft/s}^2)}(3-P_w)^{3/2}$$

Solution of the above equation yields the weir height as P_w = **0.415 ft**

Discussion Nonlinear equations of this kind can be solved easily using equation solvers like EES.

13-107

Solution The flow rate in an open channel is to be measured using a sharp-crested rectangular weir. For a measured value of flow depth upstream, the flow rate is to be determined.



Sharp-crested rectangular weir

Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The weir head is

$$H = y_1 - P_w = 3.4 - 1.3 = 2.1\ \text{m}$$

The discharge coefficient of the weir is

$$C_{wd,\text{rec}} = 0.598 + 0.0897\frac{H}{P_w} = 0.598 + 0.0897\frac{2.1\ \text{m}}{1.3\ \text{m}} = 0.7429$$

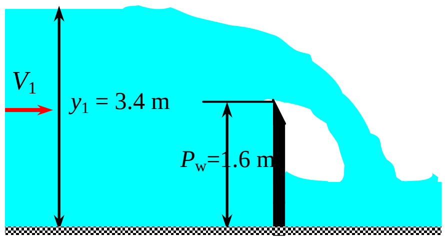
The condition $H/P_w < 2$ is satisfied since 2.1/1.3 = 1.62. Then the water flow rate through the channel becomes

$$\dot{V}_{\text{rec}} = C_{wd,\text{rec}}\frac{2}{3}b\sqrt{2g}H^{3/2} = (0.7429)\frac{2}{3}(10\ \text{m})\sqrt{2(9.81\ \text{m/s}^2)}(2.1\ \text{m})^{3/2} = \mathbf{66.8\ m^3/s}$$

Discussion The upstream velocity and the upstream velocity head are $V_1 = \dfrac{\dot{V}}{by_1} = \dfrac{66.8\ \text{m}^3\text{/s}}{(10\ \text{m})(3.4\ \text{m})} = 1.96\ \text{m/s}$ and $\dfrac{V_1^2}{2g} = \dfrac{(1.96\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = 0.197\ \text{m}$ respectively. This is 9.4% of the weir head, which is significant. When the upstream velocity head is considered, the flow rate becomes 77.8 m^3/s, which is about 16 percent higher than the value determined above. Therefore, it is good practice to consider the upstream velocity head unless the weir height P_w is very large relative to the weir head H.

13-108

Solution The flow rate in an open channel is to be measured using a sharp-crested rectangular weir. For a measured value of flow depth upstream, the flow rate is to be determined.



Sharp-crested rectangular weir

Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The weir head is $H = y_1 - P_w = 3.4 - 1.6 = 1.8\ \text{m}$. The discharge coefficient of the weir is

$$C_{wd,\text{rec}} = 0.598 + 0.0897\frac{H}{P_w} = 0.598 + 0.0897\frac{1.8\ \text{m}}{1.6\ \text{m}} = 0.6989$$

The condition $H/P_w < 2$ is satisfied since 1.8/1.6 = 1.125. Then the water flow rate through the channel becomes

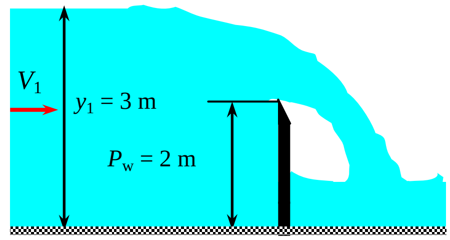
$$\dot{V}_{\text{rec}} = C_{wd,\text{rec}}\frac{2}{3}b\sqrt{2g}H^{3/2} = (0.6989)\frac{2}{3}(10\ \text{m})\sqrt{2(9.81\ \text{m/s}^2)}(1.8\ \text{m})^{3/2} = \mathbf{49.8\ m^3/s}$$

Discussion The upstream velocity and the upstream velocity head are $V_1 = \dfrac{\dot{V}}{by_1} = \dfrac{49.8\ \text{m}^3\text{/s}}{(10\ \text{m})(3.4\ \text{m})} = 1.47\ \text{m/s}$ and $\dfrac{V_1^2}{2g} = \dfrac{(1.47\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = 0.110\ \text{m}$, respectively. This is 6.1% of the weir head, which may be significant. When the upstream velocity head is considered, the flow rate becomes 55.0 m^3/s, which is about 10 percent higher than the value determined above. Therefore, it is good practice to consider the upstream velocity head unless the weir height P_w is very large relative to the weir head H.

13-109

Solution Water flowing over a sharp-crested rectangular weir is discharged into a channel where uniform flow conditions are established. The maximum slope of the downstream channel to avoid hydraulic jump is to be determined.

Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.



Sharp-crested rectangular weir

Properties Manning coefficient for an open channel of unfinished concrete is n = 0.014 (Table 13-1).

Analysis The weir head is $H = y_1 - P_w = 3.0\,\text{m} - 2.0\text{ m} = 1.0\,\text{m}$. The condition $H/P_w < 2$ is satisfied since 1.0/2.0 = 0.5. The discharge coefficient of the weir is

$$C_{wd,\text{rec}} = 0.598 + 0.0897\frac{H}{P_w} = 0.598 + 0.0897\frac{1.0\text{ m}}{2.0\text{ m}} = 0.6429$$

Then the water flow rate through the channel per meter width (i.e., taking b = 1 m) becomes

$$\dot{V}_{\text{rec}} = C_{wd,\text{rec}}\frac{2}{3}b\sqrt{2g}H^{3/2} = (0.6429)\frac{2}{3}(1\text{ m})\sqrt{2(9.81\text{ m/s}^2)}(1.0\text{ m})^{3/2} = 1.898\text{ m}^3\text{/s}$$

To avoid hydraulic jump, we must avoid supercritical flow in the channel. Therefore, the bottom slope should not be higher than the critical slope, in which case the flow depth becomes the critical depth,

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(1.898\text{ m}^3\text{/s})^2}{(9.81\text{m/s}^2)(1\text{m}^2)}\right)^{1/3} = 0.7162\text{ m}$$

Noting that the hydraulic radius of a wide channel is equal to the flow depth, the bottom slope is determined from the Manning equation to be

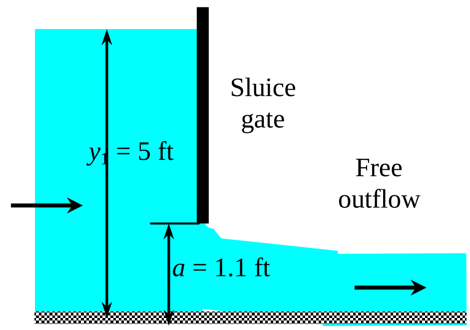
$$\dot{V} = \frac{a}{n}A_c R_h^{2/3} S_0^{1/2} \quad\rightarrow\quad 1.898\text{ m}^3\text{/s} = \frac{1\text{m}^{1/3}/s}{0.014}(0.7162\times 1\text{m}^2)(0.7162\text{ m})^{2/3}S_0^{1/2}$$

Solution gives the slope to be S_0 = 0.00215. Therefore, $S_{0,\,\text{max}}$ = **0.00215**.

Discussion For a bottom slope smaller than calculated value, downstream channel would have a mild slope, that will force the flow to remain subcritical.

13-110E

Solution Water is released from a reservoir through a sluice gate with free outflow. For specified flow depths, the flow rate per unit width and the downstream Froude number are to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Analysis For free outflow, we only need the depth ratio y_1/a to determine the discharge coefficient (for drowned outflow, we also need to know y_2/a and thus the flow depth y_2 downstream the gate)

$$\frac{y_1}{a}=\frac{5\text{ ft}}{1.1\text{ ft}}=4.55$$

The corresponding discharge coefficient is determined from Fig. 13-41 to be $C_d = 0.55$. Then the discharge rate becomes

$$\dot{V}=C_d ba\sqrt{2gy_1}=0.55(1\text{ ft})(1.1\text{ ft})\sqrt{2(32.2\text{ ft/s}^2)(5\text{ ft})}=\mathbf{10.9\ ft^3/s}$$

The specific energy of a fluid remains constant during horizontal flow when the frictional effects are negligible, $E_{s1}=E_{s2}$. With these approximations, the flow depth past the gate and the Froude number are determined to be

$$E_{s1}=y_1+\frac{V_1^2}{2g}=y_1+\frac{\dot{V}^2}{2g(by_1)^2}=5\text{ ft}+\frac{(10.9\text{ ft/s}^2)^2}{2(32.2\text{ ft/s}^2)[(1\text{ ft})(5\text{ ft})]^2}=5.074\text{ ft}$$

$$E_{s2}=y_2+\frac{V_2^2}{2g}=y_2+\frac{\dot{V}^2}{2g(by_2)^2}=E_{s1}\ \rightarrow\ y_2+\frac{(10.9\text{ ft/s}^2)^2}{2(32.2\text{ ft/s}^2)[(1\text{ ft})(y_2)]^2}=5.074\text{ ft}$$

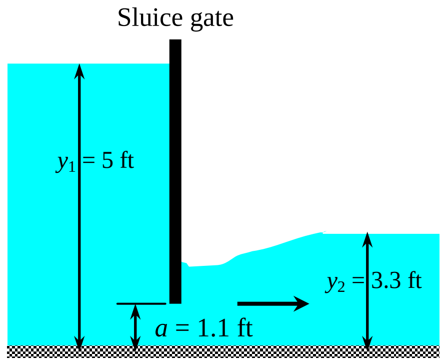
Solution yields y_2 = 0.643 ft as the physically meaningful root (positive and less than 5 ft). Then,

$$V_2=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{by_2}=\frac{10.9\text{ ft}^3/\text{s}}{(1\text{ ft})(0.643\text{ ft})}=16.9\text{ ft/s}\qquad\text{and}\qquad \text{Fr}_2=\frac{V_2}{\sqrt{gy_2}}=\frac{16.9\text{ ft/s}}{\sqrt{(32.2\text{ ft/s}^2)(0.643\text{ ft})}}=\mathbf{3.71}$$

Discussion In actual gates some frictional losses are unavoidable, and thus the actual velocity and Froude number downstream will be lower.

13-111E

Solution Water is released from a reservoir through a drowned sluice gate into an open channel. For specified flow depths, the rate of discharge is to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Analysis The depth ratio y_1/a and the contraction coefficient y_2/a are

$$\frac{y_1}{a} = \frac{5\text{ ft}}{1.1\text{ ft}} = 4.55 \quad \text{and} \quad \frac{y_2}{a} = \frac{3.3\text{ ft}}{1.1\text{ ft}} = 3$$

The corresponding discharge coefficient is determined from Fig. 13-41 to be $C_d = 0.44$. Then the discharge rate becomes

$$\dot{V} = C_d ba\sqrt{2gy_1} = 0.44(1\text{ ft})(1.1\text{ ft})\sqrt{2(32.2\text{ ft/s}^2)(5\text{ ft})} = \mathbf{8.69\ ft^3/s}$$

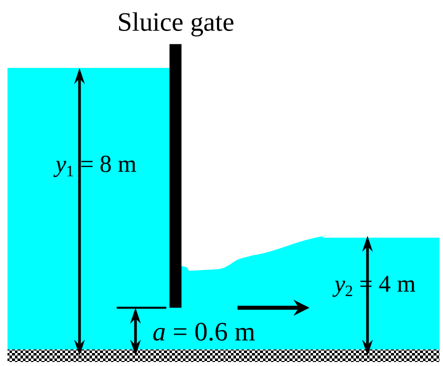
Then the Froude number downstream the gate becomes

$$V_2 = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{by_2} = \frac{8.69\text{ ft}^3\text{/s}}{(1\text{ ft})(3.3\text{ ft})} = 2.63\text{ ft/s} \quad \rightarrow \quad \text{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{2.63\text{ ft/s}}{\sqrt{(32.2\text{ ft/s}^2)(3.3\text{ ft})}} = \mathbf{0.255}$$

Discussion Note that the flow past the gate becomes subcritical when the outflow is drowned.

13-112

Solution Water is released from a lake through a drowned sluice gate into an open channel. For specified flow depths, the rate of discharge through the gate is to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Analysis The depth ratio y_1/a and the contraction coefficient y_2/a are

$$\frac{y_1}{a} = \frac{8\text{ m}}{0.6\text{ m}} = 13.3 \quad \text{and} \quad \frac{y_2}{a} = \frac{4\text{ m}}{0.6\text{ m}} = 6.7$$

The corresponding discharge coefficient is determined from Fig. 13-41 to be $C_d = 0.47$. Then the discharge rate becomes

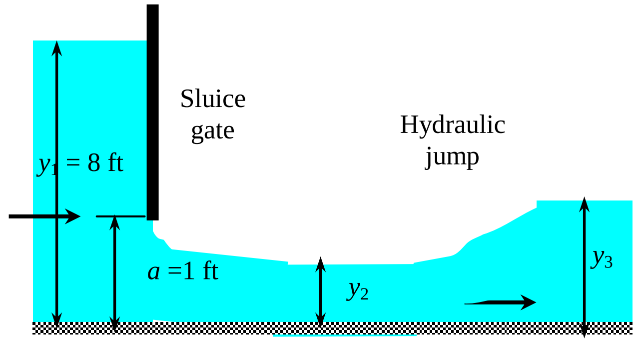
$$\dot{V} = C_d ba\sqrt{2gy_1} = 0.47(5\text{ m})(0.6\text{ m})\sqrt{2(9.81\text{ m/s}^2)(8\text{ m})} = \mathbf{17.7\ m^3/s}$$

Discussion Note that the use of the discharge coefficient enables us to determine the flow rate through sluice gates by measuring 3 flow depths only.

13-113E

Solution Water discharged through a sluice gate undergoes a hydraulic jump. The flow depth and velocities before and after the jump and the fraction of mechanical energy dissipated are to be determined.

Assumptions **1** The flow is steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** Frictional effects associated with sluice gate are negligible. **4** The channel is horizontal.



Analysis For free outflow, we only need the depth ratio y_1/a to determine the discharge coefficient,

$$\frac{y_1}{a} = \frac{8\text{ ft}}{1\text{ ft}} = 8$$

The corresponding discharge coefficient is determined from Fig. 13-41 to be C_d = 0.58. Then the discharge rate becomes

$$\dot{V} = C_d ba\sqrt{2gy_1} = 0.58(1\text{ ft})(1\text{ ft})\sqrt{2(32.2\text{ ft/s}^2)(8\text{ ft})} = \mathbf{13.16\ ft^3/s}$$

The specific energy of a fluid remains constant during horizontal flow when the frictional effects are negligible, $E_{s1} = E_{s2}$. With these approximations, the flow depth past the gate and the Froude number are determined to be

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = y_1 + \frac{\dot{V}^2}{2g(by_1)^2} = 8\text{ ft} + \frac{(13.16\text{ ft}^3/\text{s})^2}{2(32.2\text{ ft/s}^2)[(1\text{ ft})(8\text{ ft})]^2} = 8.042\text{ ft}$$

$$E_{s2} = y_2 + \frac{V_2^2}{2g} = y_2 + \frac{\dot{V}^2}{2g(by_2)^2} = E_{s1} \rightarrow y_2 + \frac{(13.16\text{ ft}^3/\text{s})^2}{2(32.2\text{ ft/s}^2)[(1\text{ ft})(y_2)]^2} = 8.042\text{ ft}$$

It gives $\boldsymbol{y_2}$ **= 0.601** ft as the physically meaningful root (positive and less than 8 ft). Then,

$$V_2 = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{by_2} = \frac{13.16\text{ ft}^3/\text{s}}{(1\text{ ft})(0.601\text{ ft})} = \mathbf{21.9\ ft/s}$$

$$\text{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{21.9\text{ ft/s}}{\sqrt{(32.2\text{ ft/s}^2)(0.601\text{ ft})}} = 4.97$$

Then the flow depth and velocity after the jump (state 3) become

$$y_3 = 0.5y_2\left(-1+\sqrt{1+8\text{Fr}_2^2}\right) = 0.5(0.601\text{ ft})\left(-1+\sqrt{1+8\times 4.97^2}\right) = \mathbf{3.94\ ft}$$

$$V_3 = \frac{y_2}{y_3}V_2 = \frac{0.601\text{ ft}}{3.94\text{ ft}}(21.9\text{ ft/s}) = \mathbf{3.34\ ft/s}$$

The head loss and the fraction of mechanical energy dissipated during the jump are

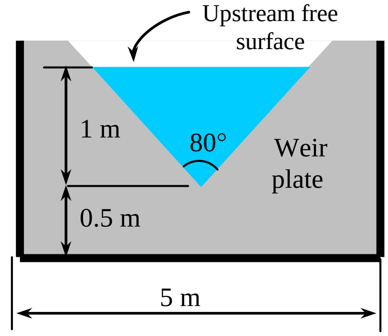
$$h_L = y_2 - y_3 + \frac{V_2^2 - V_3^2}{2g} = (0.601\text{ ft}) - (3.94\text{ ft}) + \frac{(21.9\text{ ft/s})^2 - (3.34\text{ ft/s})^2}{2(32.2\text{ ft/s}^2)} = 3.93\text{ ft}$$

$$\text{Dissipation ratio} = \frac{h_L}{E_{s2}} = \frac{h_L}{y_2(1+\text{Fr}_2^2/2)} = \frac{3.93\text{ ft}}{(0.601\text{ ft})(1+4.97^2/2)} = \mathbf{0.488}$$

Discussion Note that almost half of the mechanical energy of the fluid is dissipated during hydraulic jump.

13-114

Solution The flow rate of water in an open channel is to be measured with a sharp-crested triangular weir. For a given flow depth upstream the weir, the flow rate is to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Properties The weir discharge coefficient is given to be 0.60.

Analysis The discharge rate of water is determined directly from

$$\dot{V} = C_{wd,\mathrm{tri}} \frac{8}{15} \tan\left(\frac{\theta}{2}\right) \sqrt{2g} H^{5/2}$$

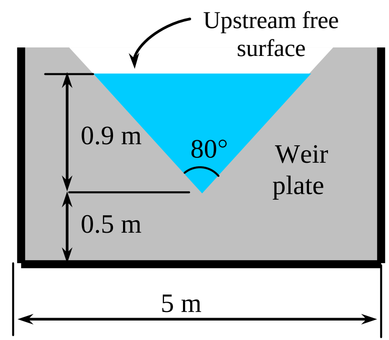
where C_{wd} = 0.60, θ = 60°, and H = 1 m. Substituting,

$$\dot{V} = (0.60) \frac{8}{15} \tan\left(\frac{80°}{2}\right) \sqrt{2(9.81\,\mathrm{m/s^2})}(1\,\mathrm{m})^{5/2} = \mathbf{1.19\ m^3/s}$$

Discussion Note that the use of the discharge coefficient enables us to determine the flow rate in a channel by measuring a single flow depth. Triangular weirs are best-suited to measure low discharge rates as they are more accurate than the other weirs for small heads.

13-115

Solution The flow rate of water in an open channel is to be measured with a sharp-crested triangular weir. For a given flow depth upstream the weir, the flow rate is to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Properties The weir discharge coefficient is given to be 0.60.

Analysis The discharge rate of water is determined directly from

$$\dot{V} = C_{wd,\mathrm{tri}} \frac{8}{15} \tan\left(\frac{\theta}{2}\right) \sqrt{2g} H^{5/2}$$

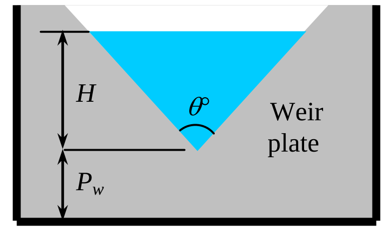
where C_{wd} = 0.60, θ = 80°, and H = 0.9 m. Substituting,

$$\dot{V} = (0.60) \frac{8}{15} \tan\left(\frac{80°}{2}\right) \sqrt{2(9.81\,\mathrm{m/s^2})}(0.9\,\mathrm{m})^{5/2} = \mathbf{0.914 m^3/s}$$

Discussion Note that the use of the discharge coefficient enables us to determine the flow rate in a channel by measuring a single flow depth. Triangular weirs are best-suited to measure low discharge rates as they are more accurate than the other weirs for small heads.

13-116

Solution The notch angle of a sharp-crested triangular weir used to measure the discharge rate of water from a lake is reduced by half. The percent reduction in the discharge rate is to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The water depth in the lake and the weir discharge coefficient remain unchanged.

Analysis The discharge rate through a triangular weir is given as

$$\dot{V} = C_{wd,\text{tri}} \frac{8}{15} \tan\left(\frac{\theta}{2}\right)\sqrt{2g} H^{5/2}$$

Therefore, the discharge rate is proportional to the tangent of the half notch angle, and the ratio of discharge rates is calculated to be

$$\dot{V} = \frac{\dot{V}_{50°}}{\dot{V}_{100°}} = \frac{\tan(50°/2)}{\tan(100°/2)} = 0.391$$

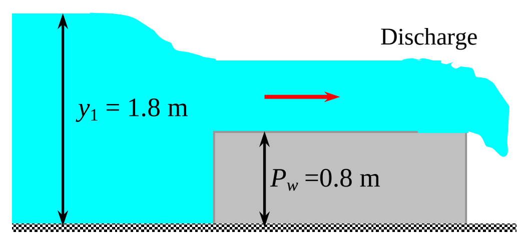
When the notch angle is reduced by half, the discharge rate drops to 39.1% of the original level. Therefore, the percent reduction in the discharge rate is

Percent reduction = 1-0.391 =0.609= **60.9%**

Discussion Note that triangular weirs with small notch angles can be used to measure small discharge rates while weirs with large notch angles can be used to measure for large discharge rates.

13-117

Solution The flow rate in an open channel is to be measured using a broad-crested rectangular weir. For a measured value of flow depth upstream, the flow rate is to be determined.



Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The weir head is $H = y_1 - P_w = 1.8 - 0.8 = 1.0\text{ m}$. The discharge coefficient of the weir is

$$C_{wd,\text{broad}} = \frac{0.65}{\sqrt{1+H/P_w}} = \frac{0.65}{\sqrt{1+(1.0\text{ m})/(0.8\text{ m})}} = 0.4333$$

Then the water flow rate through the channel becomes

$$\dot{V}_{\text{rec}} = C_{wd,\text{broad}} b\sqrt{g}\left(\frac{2}{3}\right)^{3/2} H^{3/2} = (0.4333)(5\text{ m})(2/3)^{3/2}\sqrt{9.81\text{ m/s}^2}(1.0\text{ m})^{3/2} = 3.694\text{ m}^3\text{/s} \cong \mathbf{3.69\ m^3/s}$$

The minimum flow depth above the weir is the critical depth, which is determined from

$$y_{\min} = y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(3.694\text{ m}^3\text{/s})^2}{(9.81\text{ m/s}^2)(5\text{ m})^2}\right)^{1/3} = \mathbf{0.382\ m}$$

Discussion The upstream velocity and the upstream velocity head are

$$V_1 = \frac{\dot{V}}{by_1} = \frac{3.694\text{ m}^3\text{/s}}{(5\text{ m})(1.8\text{ m})} = 0.4104\text{ m/s} \quad \text{and} \quad \frac{V_1^2}{2g} = \frac{(0.4104\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 0.00859\text{ m}$$

This is 0.9% of the weir head, which is negligible. When the upstream velocity head is considered (by replacing H in the flow rate relation by $H + V_1^2/2g$), the flow rate becomes 3.74 m^3/s, which is practically identical to the value determined above.

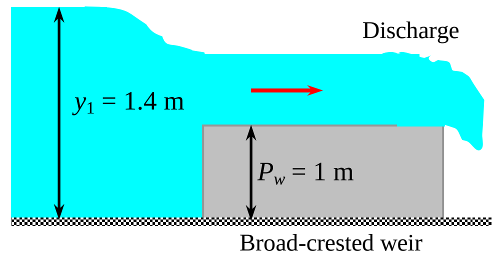
13-118

Solution The flow rate in an open channel is to be measured using a broad-crested rectangular weir. For a measured value of flow depth upstream, the flow rate is to be determined.

Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The weir head is $H = y_1 - P_w = 1.4 - 1.0 = 0.4\text{ m}$. The discharge coefficient of the weir is

$$C_{wd,\text{broad}} = \frac{0.65}{\sqrt{1 + H/P_w}} = \frac{0.65}{\sqrt{1 + (0.4\text{ m})/(1.0\text{ m})}} = 0.5494$$



Then the water flow rate through the channel becomes

$$\dot{V}_{\text{rec}} = C_{wd,\text{broad}} b\sqrt{g}\left(\frac{2}{3}\right)^{3/2} H^{3/2}$$

$$= (0.5494)(5\text{ m})(2/3)^{3/2}\sqrt{9.81\text{ m/s}^2}\,(0.4\text{ m})^{3/2}$$

$$= 1.185\text{ m}^3\text{/s} \cong \mathbf{1.18\ m^3/s}$$

The minimum flow depth above the weir is the critical depth, which is determined from

$$y_{\min} = y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(1.185\text{ m}^3\text{/s})^2}{(9.81\text{ m/s}^2)(5\text{ m})^2}\right)^{1/3} = \mathbf{0.179\ m}$$

Discussion The upstream velocity and the upstream velocity head are

$$V_1 = \frac{\dot{V}}{by_1} = \frac{1.185\text{ m}^3\text{/s}}{(5\text{ m})(1.4\text{ m})} = 0.169\text{ m/s} \qquad \text{and} \qquad \frac{V_1^2}{2g} = \frac{(0.169\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 0.00146\text{ m}$$

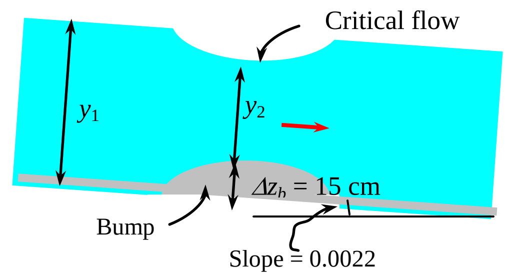
This is 0.4% of the weir head, which is negligible. When the upstream velocity head is considered (by replacing H in the flow rate relation by $H + V_1^2/2g$, the flow rate becomes 1.19 m^3/s, which is practically identical to the value determined above.

13-119

Solution Uniform subcritical water flow of water in a wide channel with a bump is considered. For critical flow over the bump, the flow rate of water and the flow depth over the bump are to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel. **4** The channel is sufficiently wide so that the end effects are negligible. **5** Frictional effects during flow over the bump are negligible.

Properties Manning coefficient for an open channel of unfinished concrete is $n = 0.014$ (Table 13-1).



Analysis Let subscript 1 denote the upstream conditions (uniform flow) in the channel, and 2 denote the critical conditions over the bump. For a wide channel, the hydraulic radius is equal to the flow depth, and thus $R_h = y_1$. Then the flow rate per m width (i.e., b = 1 m) can be determined from Manning's equation,

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{1\,\text{m}^{1/3}/s}{0.014} y_1 (y_1)^{2/3} (0.0022)^{1/2} = 3.350 y_1^{5/3} \text{ m}^3/\text{s}$$

The critical depth corresponding to this flow rate is (note that b = 1 m),

$$y_2 = y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(3.350 y_1^{5/3})^2}{g}\right)^{1/3} = \left(\frac{11.224 y_1^{10/3}}{9.81\,\text{m/s}^2}\right)^{1/3} = 1.046 y_1^{10/9}$$

The average flow velocity is $V_1 = \dot{V}/A_c = 3.350 y_1^{5/3}/y_1 = 3.350 y_1^{2/3}$ m/s. Also,

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = y_1 + \frac{(3.350 y_1^{2/3})^2}{2(9.81\,\text{m/s}^2)} = y_1 + 0.5720 y_1^{4/3}$$

$$E_{s2} = E_c = \frac{3}{2} y_c = \frac{3}{2}(1.046 y_1^{10/9}) = 1.569 y_1^{10/9}$$

Substituting these two relations into $E_{s2} = E_{s1} - \Delta z_b$ where $\Delta z_b = 0.15$ m gives

$$1.569 y_1^{10/9} = y_1 + 0.5720 y_1^{4/3} - 0.15$$

Using an equation solver such as EES or an iterative approach, the flow depth upstream is determined to be

$y_1 = 2.947$ m

Then the flow rate and the flow depth over the bump becomes

$$\dot{V} = 3.350 y_1^{5/3} = 3.350(2.947)^{5/3} = \mathbf{20.3\,m^3/s}$$

$$y_2 = y_c = 1.046 y_1^{10/9} = 1.046(2.947)^{10/9} = \mathbf{3.48\,m}$$

Discussion Note that when critical flow is established and the flow is "choked", the flow rate calculations become very easy, and it required minimal measurements. Also, $V_1 = 3.350(2.947)^{2/3} = 6.89$ m/s and

$\text{Fr}_1 = V_1/\sqrt{gy_1} = (6.89\,\text{m/s})/\sqrt{(9.81\,\text{m}^2/\text{s})(2.947\,\text{m})} = 1.28$, and thus the upstream flow is supercritical.

13-120

Solution The flow rate in an open channel is measured using a broad-crested rectangular weir. For a measured value of minimum flow depth over the weir, the flow rate and the upstream flow depth are to be determined.

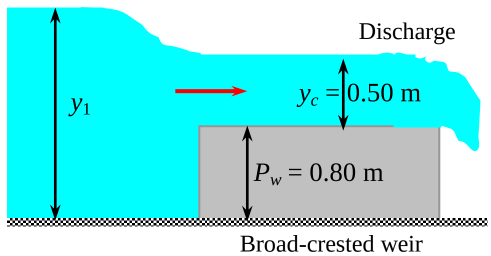
Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The flow depth over the reaches its minimum value when the flow becomes critical. Therefore, the measured minimum depth is the critical depth y_c. Then the flow rate is determined from the critical depth relation to be

$$y_{\min} = y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} \rightarrow \dot{V} = \sqrt{y_c^3 g b^2} = \sqrt{(0.50\text{ m})^3(9.81\text{ m/s}^2)(1\text{ m})^2} = \mathbf{1.11\ m^3/s}$$

This is the flow rate per m width of the channel since we have taken b = 1 m. Disregarding the upstream velocity head and noting that the discharge coefficient of the weir is $C_{wd,\text{broad}} = 0.65/\sqrt{1+H/P_w}$, the flow rate for a broad-crested weir can be expressed as

$$\dot{V}_{\text{rec}} = \frac{0.65}{\sqrt{1+H/P_w}} b\sqrt{g}\left(\frac{2}{3}\right)^{3/2} H^{3/2}$$



Substituting,

$$1.11\text{ m}^3/\text{s} = \frac{0.65\text{ m}}{\sqrt{1+\text{H}/(0.8\text{ m})}}(1\text{ m})(2/3)^{3/2}\sqrt{9.81\text{ m/s}^2}\, H^{3/2}$$
$$= \mathbf{4.91 m^3/s}$$

Its solution is H = 1.40 m. Then the flow depth upstream the weir becomes

$$y_1 = H + P_w = 1.40 + 0.80 = \mathbf{2.20 m}$$

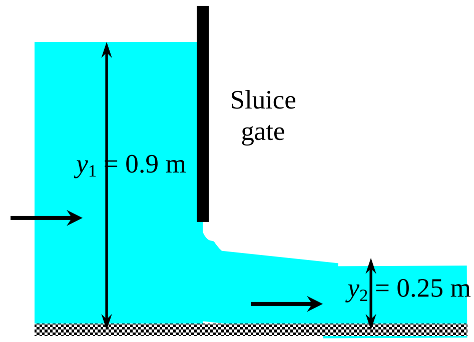
Discussion The upstream velocity and the upstream velocity head are

$$V_1 = \frac{\dot{V}}{by_1} = \frac{1.11\text{ m}^3/\text{s}}{(1\text{ m})(2.2\text{ m})} = 0.503\text{ m/s} \quad \text{and} \quad \frac{V_1^2}{2g} = \frac{(0.503\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 0.013\text{ m}$$

This is 0.9% of the weir head, which is negligible. When the upstream velocity head is considered (by replacing H in the flow rate relation by $H + V_1^2/2g$, the flow rate becomes 1.12 m^3/s, which is practically identical to the value determined above.

13-121

Solution A sluice gate is used to control the flow rate of water in a channel. For specified flow depths upstream and downstream from the gate, the flow rate of water and the downstream Froude number are to be determined.



Assumptions **1** The flow is steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** Frictional effects associated with sluice gate are negligible. **4** The channel is horizontal.

Analysis When frictional effects are negligible and the flow section is horizontal, the specific energy remains constant, $E_{s1} = E_{s2}$. Then,

$$y_1 + \frac{V_1^2}{2g} = y_2 + \frac{V_2^2}{2g} \quad \rightarrow \quad 0.9\text{ m} + \frac{\dot{V}^2}{2(9.81\text{ m/s}^2)[(8\text{ m})(0.9\text{ m})]^2} = 0.25\text{ m} + \frac{\dot{V}^2}{2(9.81\text{ m/s}^2)[(8\text{ m})(0.25\text{ m})]^2}$$

Solving for the flow rate gives $\dot{V} = 7.435\text{ m}^3\text{/s} \cong \mathbf{7.44\ m^3/s}$. The downstream velocity and Froude number are

$$V_2 = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{by_2} = \frac{7.435\text{ m}^3\text{/s}}{(8\text{ m})(0.25\text{ m})} = 3.718\text{ m/s} \qquad \text{and} \qquad \text{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{3.718\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(0.25\text{ m})}} = \mathbf{2.37}$$

Discussion The actual values will be somewhat different because of frictional effects.

Review Problems

13-122

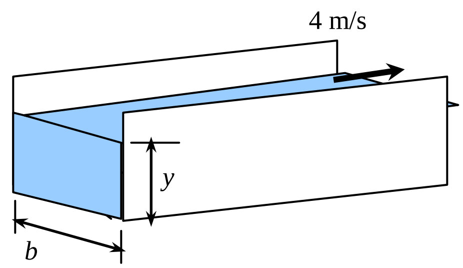
Solution Water flows in a canal at a specified average velocity. For various flow depths, it is to be determined whether the flow is subcritical or supercritical.

Assumptions The flow is uniform.

Analysis For each depth, we determine the Froude number and compare it to the critical value of 1:

(***a***) y = 0.2 m: $\quad \mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{4\ \mathrm{m/s}}{\sqrt{(9.81\,\mathrm{m/s^2})\,(0.2\,\mathrm{m})}} = 2.86 > 1$

which is greater than 1. Therefore, the flow is **supercritical**.



(***b***) y = 2 m: $\quad \mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{4\ \mathrm{m/s}}{\sqrt{(9.81\,\mathrm{m/s^2})\,(2\,\mathrm{m})}} = 0.903 < 1$

which is less than 1. Therefore, the flow is **subcritical**.

(***c***) y = 1.63 m: $\quad \mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{4\ \mathrm{m/s}}{\sqrt{(9.81\,\mathrm{m/s^2})\,(1.63\,\mathrm{m})}} = 1$

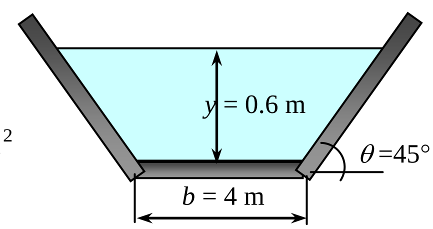
which is equal to 1. Therefore, the flow is **critical**.

Discussion Note that a flow is more likely to exist as supercritical when the flow depth is low and thus the flow velocity is high. Also, the type of flow can be determined easily by checking Froude number.

13-123

Solution Water flows uniformly in a trapezoidal channel. For a given flow depth, it is to be determined whether the flow is subcritical or supercritical.

Assumptions The flow is uniform.



Analysis The flow area and the average velocity are

$$A_c = y\frac{(b+b+2y/\tan\theta)}{2} = (0.60\ \mathrm{m})\frac{[4+4+2(0.60\ \mathrm{m})/\tan 45°]\ \mathrm{m}}{2} = 2.76\ \mathrm{m}^2$$

$$V = \frac{\dot{V}}{A_c} = \frac{18\ \mathrm{m}^3/\mathrm{s}}{2.76\ \mathrm{m}^2} = 6.522\ \mathrm{m/s}$$

When calculating the Froude number, the hydraulic depth should be used rather than the maximum depth or the hydraulic radius. For a non-rectangular channel, hydraulic depth is defined as the ratio of the flow area to top width,

$$y = y_h = \frac{A_c}{\text{Top width}} = \frac{A_c}{b+2y/\tan\theta} = \frac{2.76\ \mathrm{m}^2}{(4+2\times 0.60/\tan 45°)\ \mathrm{m}} = 0.5308\ \mathrm{m}$$

Then the Froude number becomes $\mathrm{Fr} = \dfrac{V}{\sqrt{gy}} = \dfrac{6.522\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m/s^2})(0.5308\ \mathrm{m})}} = 2.86$, which is greater than 1.

Therefore, the flow is **supercritical**.

Discussion The analysis is approximate since the edge effects are significant here compared to a wide rectangular channel, and thus the results should be interpreted accordingly.

13-124

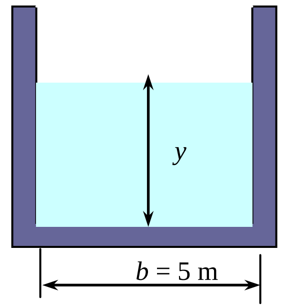
Solution The flow of water in a rectangular channel is considered. The effect of bottom slope on the flow rate is to be investigated as the bottom angle varies from 0.5 to 10°.

Assumptions **1** The flow is steady and uniform. **2** Roughness coefficient is constant along the channel.

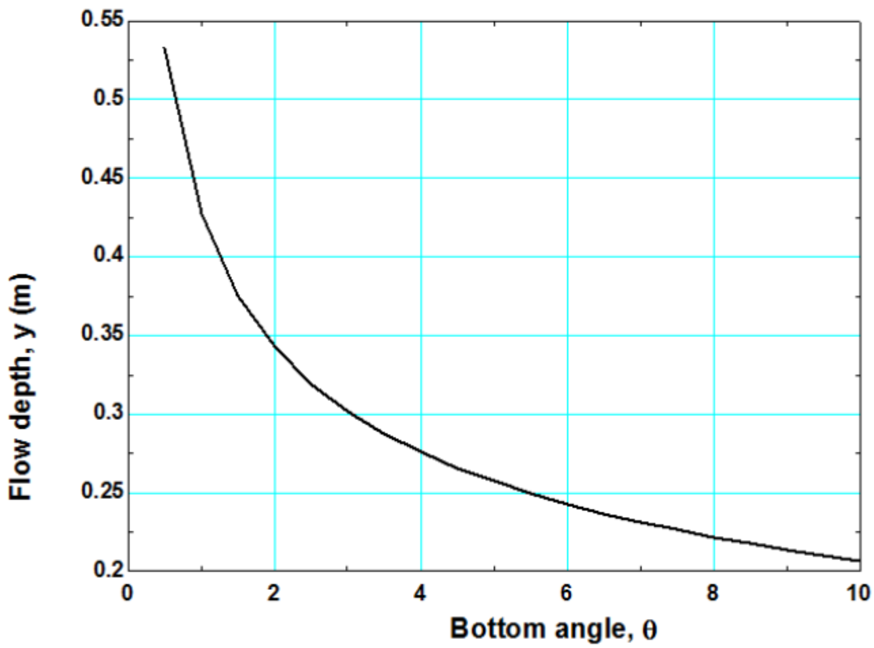
Properties Manning coefficient for an open channel made of finished concrete is n = 0.012 (Table 13-1).

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
a=1
b=5
Vdot = 12 "m3/s"
n=0.012

s=tan(teta)
Ac=b*y
p=b+2*y
Rh=Ac/p
Vdot=(a/n)*Ac*Rh^(2/3)*SQRT(s)
```



Bottom angle, θ°	Flow depth, y, m
0.5	0.533
1.0	0.427
1.5	0.375
2.0	0.343
2.5	0.320
3.0	0.302
3.5	0.287
4.0	0.276
4.5	0.266
5.0	0.257
5.5	0.250
6.0	0.243
6.5	0.237
7.0	0.231
7.5	0.226
8.0	0.222
8.5	0.218
9.0	0.214
9.5	0.210
10.0	0.207



Discussion Note that the flow depth decreases as the bottom angle increases, as expected.

13-125

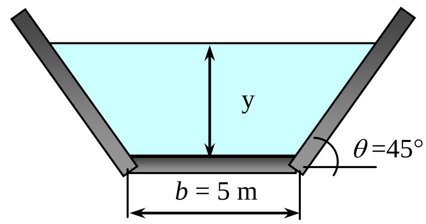
Solution The flow of water in a trapezoidal channel is considered. The effect of bottom slope on the flow rate is to be investigated as the bottom angle varies from 0.5 to 10°.

Assumptions **1** The flow is steady and uniform. **2** Roughness coefficient is constant along the channel.

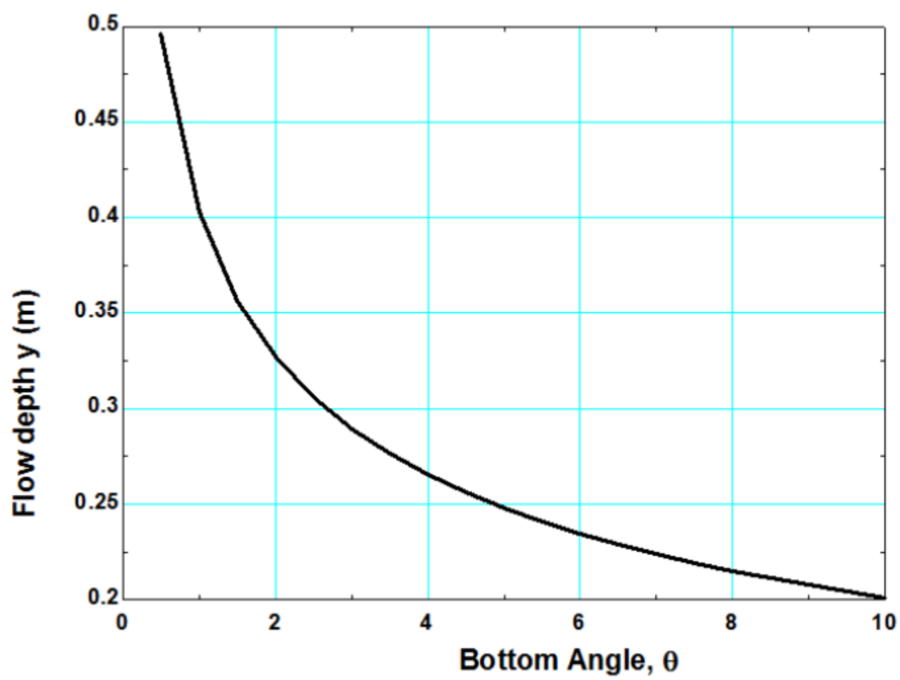
Properties Manning coefficient for an open channel made of finished concrete is n = 0.012 (Table 13-1).

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
a=1
b=5
Vdot = 12 "m3/s"
n=0.012

s=tan(teta)
Ac=y*(b+y/tan(45))
p=b+2*y/sin(45)
Rh=Ac/p
Vdot=(a/n)*Ac*Rh^(2/3)*SQRT(s)
```



Bottom angle, θ°	Flow depth, y, m
0.5	0.496
1.0	0.403
1.5	0.357
2.0	0.327
2.5	0.306
3.0	0.290
3.5	0.276
4.0	0.266
4.5	0.256
5.0	0.248
5.5	0.241
6.0	0.235
6.5	0.229
7.0	0.224
7.5	0.220
8.0	0.215
8.5	0.211
9.0	0.208
9.5	0.204
10.0	0.201



Discussion As expected, flow depth decreases with increasing bottom angle, but the relationship is far from linear.

13-126

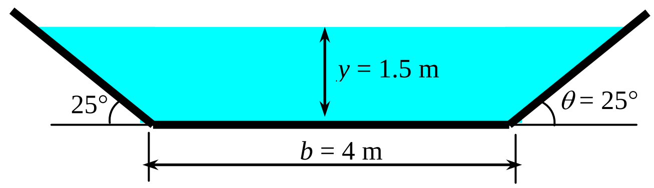
Solution The flow of water in a trapezoidal channel is considered. For a given flow depth and bottom slope, the flow rate is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties The Manning coefficient for a brick-lined open channel is n = 0.015 (Table 13-1).

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are

$$A_c = y\left(b + \frac{y}{\tan\theta}\right) = (1.5\,\text{m})\left(4\,\text{m} + \frac{1.5\,\text{m}}{\tan 25°}\right) = 10.83\,\text{m}^2$$



$$p = b + \frac{2y}{\sin\theta} = 4\,\text{m} + \frac{2(1.5\,\text{m})}{\sin 25°} = 11.10\,\text{m}$$

$$R_h = \frac{A_c}{p} = \frac{10.83\,\text{m}^2}{11.10\,\text{m}} = 0.9758\,\text{m}$$

Bottom slope of the channel is S_o = 0.001.Then the flow rate can be determined from Manning's equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{1\,\text{m}^{1/3}/s}{0.015}(10.83\,\text{m}^2)(0.9758\,\text{m})^{2/3}(0.001)^{1/2} = \mathbf{22.5\ m^3/s}$$

Discussion Note that the flow rate in a given channel is a strong function of the bottom slope.

13-127

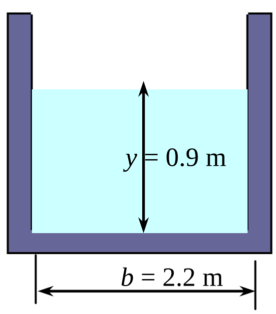
Solution The flow of water in a rectangular channel is considered. For a given flow depth and bottom slope, the flow rate is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties The Manning coefficient is given to be n = 0.012 (Table 13-1).

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are

$$A_c = by = (2.2\,\text{m})(0.9\,\text{m}) = 1.98\,\text{m}^2 \qquad p = 2.2\,\text{m} + 2\times 0.9\,\text{m} = 4.0\,\text{m}$$



$$R_h = \frac{A_c}{p} = \frac{1.98\,\text{m}^2}{4.0\,\text{m}} = 0.495\,\text{m}$$

Bottom slope of the channel is

$$S_0 = \tan 0.6^\circ = 0.01047$$

Then the flow rate can be determined from Manning's equation to be

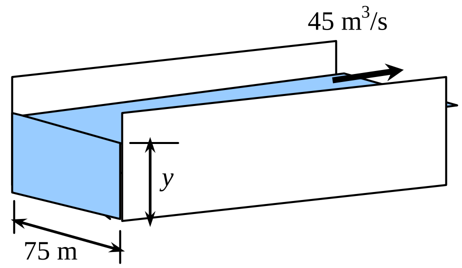
$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{1\,\text{m}^{1/3}/s}{0.012}(1.98\,\text{m}^2)(0.495\,\text{m})^{2/3}(0.01047)^{1/2} = \mathbf{10.6\ m^3/s}$$

Discussion Note that the flow rate in a given channel is a strong function of the bottom slope.

13-128

Solution Water flows in a rectangular channel. The flow depth below which the flow is supercritical is to be determined.

Assumptions The flow is uniform.



Analysis The flow depth below which the flow is super critical is the critical depth y_c determined from

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(45\,\text{m}^3/\text{s})^2}{(9.81\,\text{m/s}^2)(7\,\text{m})^2}\right)^{1/3} = \mathbf{1.62\,m}$$

Therefore, flow is **supercritical** for $y < 1.62$ m.

Discussion Note that a flow is more likely to exist as supercritical when the flow depth is low and thus the flow velocity is high.

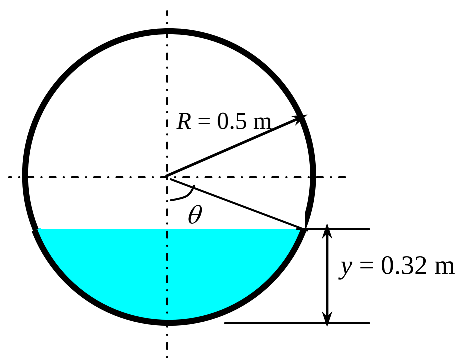
13-129

Solution Waters flows in a partially filled circular channel made of finished concrete. For a given flow depth and bottom slope, the flow rate is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties Manning coefficient for an open channel made of finished concrete is $n = 0.012$ (Table 13-1).

Analysis From geometric considerations,



$$\cos\theta = \frac{R-y}{R} = \frac{0.5-0.32}{0.5} = 0.36 \quad \rightarrow \quad \theta = 68.9° = 68.9\frac{2\pi}{360} = 1.203$$

$$A_c = R^2(\theta - \sin\theta\cos\theta) = (0.5\,\text{m})^2[1.203 - \sin(1.203)\cos(1.203)] = 0.2169\,\text{m}^2$$

$$R_h = \frac{A_c}{p} = \frac{\theta - \sin\theta\cos\theta}{2\theta}R = \frac{1.203 - \sin(1.203)\cos(1.203)}{2(1.203)}(0.5\,\text{m}) = 0.1803\,\text{m}$$

Then the flow rate can be determined from Manning's equation to be

$$\dot{V} = \frac{a}{n}A_c R_h^{2/3} S_0^{1/2} = \frac{1\,\text{m}^{1/3}/s}{0.012}(0.2169\,\text{m}^2)(0.1803\,\text{m})^{2/3}(0.002)^{1/2} = \mathbf{0.258\,m^3/s}$$

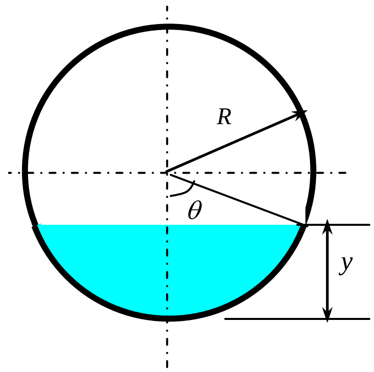
Discussion Note that the flow rate in a given channel is a strong function of the bottom slope.

13-130

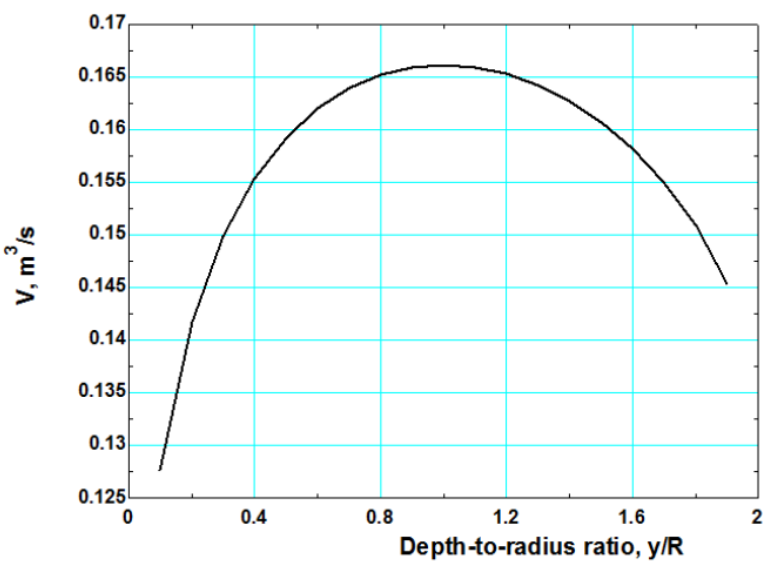
Solution The previous problem is reconsidered. By varying the flow depth-to-radius ratio from 0.1 to 1.9 for a fixed value of flow area, it is the to be shown that the best hydraulic cross section occurs when the circular channel is half-full, and the results are to be plotted.

Analysis The EES *Equations* window is printed below, along with the tabulated and plotted results.

```
a=1
n=0.012
s=0.002
Ac=0.1536 "Flow area kept constant"
ratio=y/R "This ratio is varied from 0.1 to 1.9"
bdeg=arcsin((R-y)/R)
tetadeg=90-bdeg
teta=tetadeg*2*pi/360
Ac=R^2*(teta-sin(tetadeg)*cos(tetadeg))
p=2*teta*R
Rh=Ac/p
Vdot=(a/n)*Ac*Rh^(2/3)*SQRT(s)
```



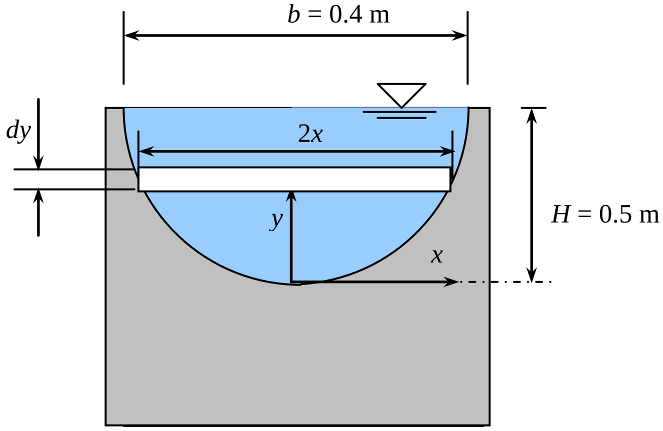
Depth-to-radius ratio, y/R	Channel radius, R, m	Flow rate, $\dot{V}$, m^3/s
0.1	1.617	0.1276
0.2	0.969	0.1417
0.3	0.721	0.1498
0.4	0.586	0.1553
0.5	0.500	0.1592
0.6	0.440	0.1620
0.7	0.396	0.1639
0.8	0.362	0.1652
0.9	0.335	0.1659
1.0	0.313	**0.1661**
1.1	0.295	0.1659
1.2	0.279	0.1653
1.3	0.267	0.1642
1.4	0.256	0.1627
1.5	0.247	0.1607
1.6	0.239	0.1582
1.7	0.232	0.1550
1.8	0.227	0.1509
1.9	0.223	0.1453



Discussion The depth-to-radius ratio of $y/R = 1$ corresponds to a half-full circular channel, and it is clear from the table and the chart that, for a fixed flow area, the flow rate becomes maximum when the channel is half-full.

13-131

Solution The flow of water through a parabolic notch is considered. A relation is to be developed for the flow rate, and its numerical value is to be calculated.



Assumptions **1** The flow is steady. **2** All frictional effects are negligible, and Toricelli's equation can be used for the velocity.

Analysis The notch is parabolic with $y = 0$ at $x = 0$, and thus it can be expressed analytically as $y = Cx^2$. Using the coordinates of the upper right corner, the value of the constant is determined to be

$$C = y/x^2 = H/(b/2)^2 = 4H/b^2 = 4(0.5\text{ m})/(0.4\text{ m})^2 = 12.5\text{ m}^{-1}.$$

A differential area strip can be expressed as

$$dA = 2xdy = 2\sqrt{y/C}\,dy$$

Noting that the flow velocity is $V = \sqrt{2g(H-y)}$, the flow rate through this differential area is

$$VdA = V\left(2\sqrt{y/C}\,dy\right) = \sqrt{2g(H-y)}\,2\sqrt{y/C}\,dy = 2\sqrt{2g/C}\sqrt{y(H-y)}\,dy$$

Then the flow rate through the entire notch is determined by integration to be

$$\dot{V} = \int_A VdA = 2\sqrt{2g/C}\int_{y=0}^{H}\sqrt{y(H-y)}\,dy$$

where

$$\int_{y=0}^{H}\sqrt{y(H-y)}\,dy = \left[\frac{1}{4}(2y-H)\sqrt{Hy-y^2} + \frac{H^2}{8}Arc\tan\left(\frac{2y-H}{2\sqrt{Hy-y^2}}\right)\right]_0^H = \frac{\pi}{16}H^2$$

Then the expression for the volume flow rate and its numerical value become

$$\boxed{\dot{V} = \frac{\pi}{8}\sqrt{\frac{2g}{C}}H^2 = \frac{\pi}{8}\sqrt{\frac{2(9.81\text{ m/s}^2)}{12.5\text{ m}^{-1}}}H^2 = (0.4920\text{ m/s})H^2 = (0.492\text{ m/s})(0.5\text{ m})^2 = \mathbf{0.123\ m^3/s}}$$

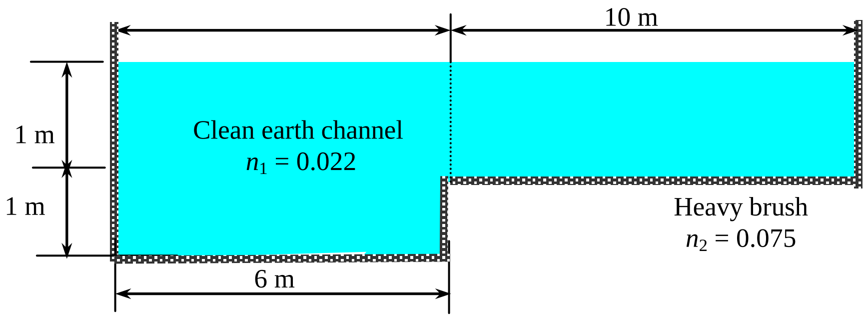
Discussion Note that a general flow rate equation for parabolic notch would be in the form of $\dot{V} = KH^2$, where $K = C_d\frac{\pi}{8}\sqrt{\frac{2g}{C}}$ and C_d is the discharge coefficient whose value is determined experimentally to account for nonideal effects.

13-132

Solution Water is flowing through a channel with nonuniform surface properties. The flow rate through the channel and the effective Manning coefficient are to be determined.

Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The Manning coefficients do not vary along the channel.

Analysis The channel involves two parts with different roughness, and thus it is appropriate to divide the channel into two subsections. The flow rate for each subsection can be determined from the Manning equation, and the total flow rate can be determined by adding them up.



The flow area, perimeter, and hydraulic radius for each subsection and the entire channel are:

Subsection 1: $A_{c1} = 6\text{ m}^2$, $p_1 = 6\text{ m}$, $R_{h1} = \dfrac{A_{c1}}{p_1} = \dfrac{6\text{ m}^2}{6\text{ m}} = 1.00\text{ m}$

Subsection 2: $A_{c2} = 10\text{ m}^2$, $p_2 = 11\text{ m}$, $R_{h2} = \dfrac{A_{c2}}{p_2} = \dfrac{10\text{ m}^2}{11\text{ m}} = 0.909\text{ m}$

Entire channel: $A_c = 16\text{ m}^2$, $p = 17\text{ m}$, $R_h = \dfrac{A_c}{p} = \dfrac{16\text{ m}^2}{17\text{ m}} = 0.941\text{ m}$

Applying the Manning equation to each subsection, the total flow rate through the channel becomes

$$\dot{V} = \dot{V}_1 + \dot{V}_2 = \frac{a}{n_1} A_1 R_1^{2/3} S_0^{1/2} + \frac{a}{n_2} A_2 R_2^{2/3} S_0^{1/2}$$

$$= (1\text{ m}^{1/3}/\text{s})\left(\frac{(6\text{ m}^2)\ (1\text{ m})^{2/3}}{0.022} + \frac{(10\text{ m}^2)(0.909\text{ m})^{2/3}}{0.075}\right)(\tan 0.5°)^{1/2}$$

$$= \mathbf{37.2 m^3/s}$$

Knowing the total flow rate, the effective Manning coefficient for the entire channel can be determined from the Manning equation to be

$$n_{\text{eff}} = \frac{aA_c R_h^{2/3} S_0^{1/2}}{\dot{V}} = \frac{(1\text{ m}^{1/3}/\text{s})(16\text{ m}^2)(0.941\text{ m})^{2/3}(0.00873)^{1/2}}{37.2\text{ m}^3/\text{s}} = \mathbf{0.0386}$$

Discussion The effective Manning coefficient n_{eff} of the channel turns out to lie between the two n values, as expected. The weighted average of the Manning coefficient of the channel is $n_{\text{ave}}=(n_1p_1+ n_2p_2)/p = 0.056$, which is quite different than n_{eff}. Therefore, using a weighted average Manning coefficient for the entire channel may be tempting, but it would not be so accurate.

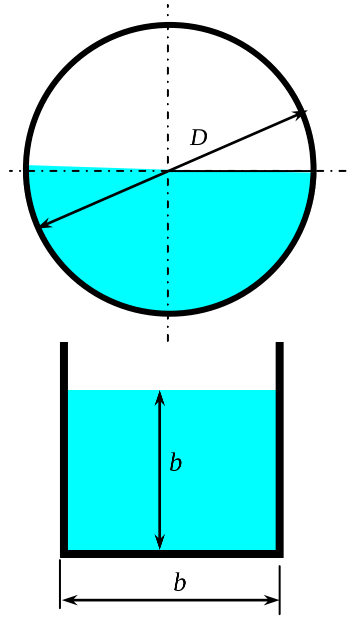
13-133

Solution Two identical channels, one rectangular of bottom width b and one circular of diameter D, with identical flow rates, bottom slopes, and surface linings are considered. The relation between b and D is to be determined for the case of the flow height $y = b$ and the circular channel is flowing half full.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Analysis The cross-sectional area, perimeter, and hydraulic radius of the rectangular channel are

$$A_c = b^2, \quad p = 3b, \text{ and } \quad R_h = \frac{A_c}{p} = \frac{b^2}{3\text{b}} = \frac{b}{3}$$



Then using the Manning equation, the flow rate can be expressed as

$$\dot{V}_{\text{rec}} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n} b^2 \left(\frac{b}{3}\right)^{2/3} S_0^{1/2} = \frac{a}{n} S_0^{1/2} \frac{b^{8/3}}{3^{2/3}}$$

The corresponding relations for the semi-circular channel are

$$A_c = \frac{\pi D^2}{8}, \quad p = \frac{\pi D}{2}, \text{ and } \quad R_h = \frac{A_c}{p} = \frac{D}{4}$$

and

$$\dot{V}_{\text{cir}} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} = \frac{a}{n} \pi \frac{D^2}{8} \left(\frac{D}{4}\right)^{2/3} S_0^{1/2} = \frac{a}{n} S_0^{1/2} \frac{\pi D^{8/3}}{8 \times 4^{2/3}}$$

Setting the flow rates in the two channels equal to each other $\dot{V}_{\text{cir}} = \dot{V}_{\text{rec}}$ gives

$$\frac{a}{n} S_0^{1/2} \frac{b^{8/3}}{3^{2/3}} = \frac{a}{n} \frac{\pi D^{8/3}}{8 \times 4^{2/3}} S_0^{1/2} \quad \rightarrow \quad \frac{b^{8/3}}{3^{2/3}} = \frac{\pi D^{8/3}}{8 \times 4^{2/3}} \quad \rightarrow \quad \frac{b}{D} = \left(\frac{\pi 3^{2/3}}{8 \times 4^{2/3}}\right)^{3/8} = 0.655$$

Therefore, the desired relation is **b = 0.655D**.

Discussion Note that the wetted perimeters in this case are $p_{\text{rec}} = 3b = 2.0D$ and $p_{\text{cir}} = \pi D/2 = 1.57D$. Therefore, the semi-circular channel is a more efficient channel than the rectangular one.

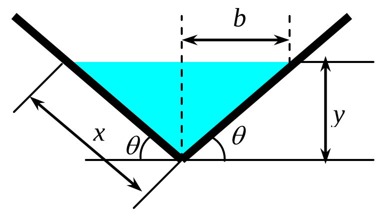
13-134

Solution The flow of water through a V-shaped open channel is considered. The angle θ the channel makes from the horizontal is to be determined for the case of most efficient flow.

Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The roughness coefficient is constant.

Analysis We let the length of the sidewall of the channel be *x*. From trigonometry,

$$\sin\theta = \frac{y}{x} \quad \rightarrow \quad y = x\sin\theta \qquad \cos\theta = \frac{b}{x} \quad \rightarrow \quad b = x\cos\theta$$



Then the cross-sectional area and the perimeter of the flow section become

$$A_c = by = x\cos\theta\sin\theta = \frac{x^2}{2}\sin 2\theta \quad \rightarrow \quad x = \sqrt{\frac{2A_c}{\sin 2\theta}}$$

$$p = 2x = 2\sqrt{\frac{2A_c}{\sin 2\theta}} \quad \rightarrow \quad p = 2\sqrt{2A_c}(\sin 2\theta)^{-1/2}$$

Now we apply the criterion that the best hydraulic cross-section for an open channel is the one with the minimum wetted perimeter for a given cross-section. Taking the derivative of p with respect to θ while holding A_c constant gives

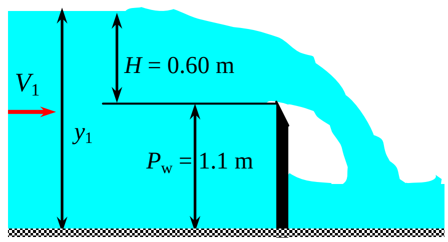
$$\frac{dp}{d\theta} = 2\sqrt{2A_c}\frac{d[(\sin 2\theta)^{-1/2}]}{d\theta} = 2\sqrt{2A_c}\frac{d[(\sin 2\theta)^{-1/2}]}{d(\sin 2\theta)}\frac{d(\sin 2\theta)}{d\theta} = 2\sqrt{2A_c}\frac{-3}{2(\sin 2\theta)^{3/2}}2\cos 2\theta$$

Setting $dp/d\theta = 0$ gives $\cos 2\theta = 0$, which is satisfied when $2\theta = 90°$. Therefore, the criterion for the best hydraulic cross-section for a triangular channel is determined to be $\boldsymbol{\theta = 45°}$.

Discussion The procedure used here can be used to determine the best hydraulic cross-section for any geometric shape.

13-135

Solution The flow rate in an open channel is to be measured using a sharp-crested rectangular weir. For a measured value of flow depth upstream, the flow rate is to be determined.



Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The weir head is given to be H = 0.60 m. The discharge coefficient of the weir is

$$C_{wd,\mathrm{rec}} = 0.598 + 0.0897\frac{H}{P_w} = 0.598 + 0.0897\frac{0.60\ \mathrm{m}}{1.1\ \mathrm{m}} = 0.6469$$

The condition $H/P_w < 2$ is satisfied since 0.60/1.1 = 0.55. Then the water flow rate through the channel becomes

$$\dot{V} = C_{wd,\mathrm{rec}}\frac{2}{3}b\sqrt{2g}H^{3/2}$$
$$= (0.6469)\frac{2}{3}(6\ \mathrm{m})\sqrt{2(9.81\ \mathrm{m/s^2})}(0.60\ \mathrm{m})^{3/2}$$
$$= \mathbf{5.33\ m^3/s}$$

Discussion The upstream velocity and the upstream velocity head are

$$V_1 = \frac{\dot{V}}{by_1} = \frac{5.33\ \mathrm{m^3/s}}{(6\ \mathrm{m})(1.70\ \mathrm{m})} = 0.522\ \mathrm{m/s} \quad \text{and} \quad \frac{V_1^2}{2g} = \frac{(0.522\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})} = 0.014\ \mathrm{m}$$

This is 2.3% of the weir head, which is negligible. When the upstream velocity head is considered, the flow rate becomes 5.50 m^3/s, which is about 3 percent higher than the value determined above. Therefore, the assumption of negligible velocity head is reasonable in this case.

13-136E

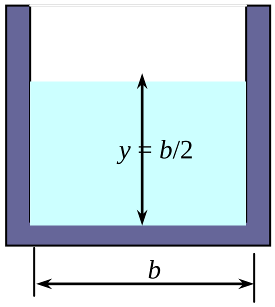
Solution Water is to be transported in a rectangular channel at a specified rate. The dimensions for the best cross-section if the channel is made of unfinished concrete are to be determined.

Assumptions **1** The flow is steady and uniform. **2** The bottom slope is constant. **3** The roughness coefficient is constant.

Properties The Manning coefficient is n = 0.014 for channels made of unfinished concrete (Table 13-1).

Analysis For best cross-section of a rectangular cross-section, $y = b/2$. Then $A_c = yb = b^2/2$, and $R_h = b/4$.

The flow rate is determined from the Manning equation, $\dot{V} = \dfrac{a}{n} A_c R_h^{2/3} S_0^{1/2}$.



(a) Bottom drop of 5 ft per mile: $s = (5\text{ ft})/(5280\text{ ft}) = 0.0009470$

$$200\text{ ft}^3/\text{s} = \frac{1.486\text{ ft}^{1/3}/s}{0.014}(b^2/2)(b/4)^{2/3}(0.0009470)^{1/2}$$

Solving the above equation gives b **= 8.58 ft,** and $y = b/2$ = 4.29 ft.

(b) Bottom drop of 10 ft per mile: $s = (10\text{ ft})/(5280\text{ ft}) = 0.001894$

$$200\text{ ft}^3/\text{s} = \frac{1.486\text{ ft}^{1/3}/s}{0.014}(b^2/2)(b/4)^{2/3}(0.001894)^{1/2}$$

Solving the above equation gives b **= 7.54 ft,** and $y = b/2$ = 3.77 ft.

Discussion The concept of best cross-section is an important consideration in the design of open channels because it directly affects the construction costs.

13-137E

Solution Water is to be transported in a trapezoidal channel at a specified rate. The dimensions for the best cross-section if the channel is made of unfinished concrete are to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Properties The Manning coefficient is n = 0.014 for channels made of unfinished concrete (Table 13-1).

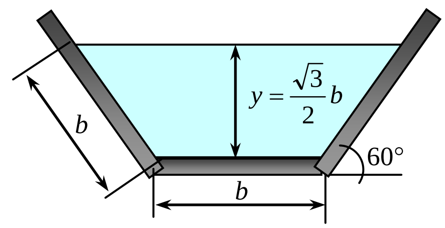
Analysis For best cross-section of a trapezoidal channel of bottom width b, θ = 60° and $y = b\sqrt{3}/2$. Then,

$$A_c = y(b + b\cos\theta) = 0.5\sqrt{3}b^2(1+\cos 60°) = 0.75\sqrt{3}b^2,\ \ p = 3b,\ \text{and}\ \ R_h = \frac{y}{2} = \frac{\sqrt{3}}{4}b\,.$$

The flow rate is determined from the Manning equation, $\dot{V} = \dfrac{a}{n} A_c R_h^{2/3} S_0^{1/2}$,

(*a*) Bottom drop of 5 ft per mile:

$$s = (5\text{ ft})/(5280\text{ ft}) = 0.0009470$$



$$200\text{ ft}^3/\text{s} = \frac{1.486\text{ ft}^{1/3}/s}{0.014}(0.75\sqrt{3}b^2)(\sqrt{3}b/4)^{2/3}(0.0009470)^{1/2}$$

Solving for b yields b = **5.23 ft,** and y = 4.53 ft

(*b*) Bottom drop of 10 ft per mile:

$$s = (10\text{ ft})/(5280\text{ ft}) = 0.001894$$

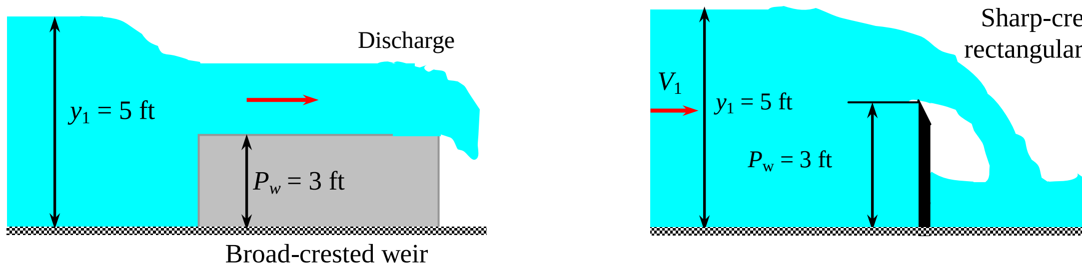
$$200\text{ ft}^3/\text{s} = \frac{1.486\text{ ft}^{1/3}/s}{0.014}(0.75\sqrt{3}b^2)(\sqrt{3}b/4)^{2/3}(0.001894)^{1/2}$$

Solving for b yields b = **4.59 ft,** and y = 3.98 ft

Discussion The concept of best cross-section is an important consideration in the design of open channels because it directly affects the construction costs.

13-138E

Solution The flow rates in two open channels are to be measured using a sharp-crested weir in one and a broad-crested rectangular weir in the other. For identical flow depths, the flow rates through both channels are to be determined.



Assumptions **1** The flow is steady. **2** The upstream velocity head is negligible. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The weir head is

$$H = y_1 - P_w = 5.0\text{ ft} - 3.0\text{ ft} = 2.0\text{ ft}$$

The condition $H/P_w < 2$ is satisfied since 2.0/3.0 = 0.667. The discharge coefficients of the weirs are

Sharp-crested weir:

$$C_{wd,\text{sharp}} = 0.598 + 0.0897\frac{H}{P_w} = 0.598 + 0.0897\frac{2.0\text{ ft}}{3.0\text{ ft}} = 0.6578$$

$$\dot{V}_{\text{sharp}} = C_{wd,sharp}\frac{2}{3}b\sqrt{2g}H^{3/2} = (0.6578)\frac{2}{3}(15\text{ ft})\sqrt{2(32.2\text{ ft/s}^2)}(2.0\text{ ft})^{3/2} = \mathbf{149\ ft^3/s}$$

Broad-crested weir:

$$C_{wd,\text{broad}} = \frac{0.65}{\sqrt{1 + H/P_w}} = \frac{0.65}{\sqrt{1 + (2.0\text{ ft})/(3.0\text{ ft})}} = 0.5035$$

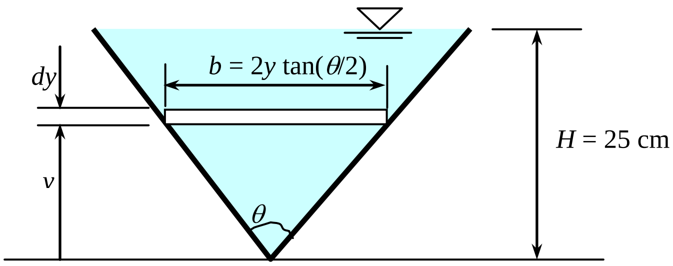
$$\dot{V}_{\text{broad}} = C_{wd,\text{broad}}b\sqrt{g}\left(\frac{2}{3}\right)^{3/2}H^{3/2} = (0.5035)(15\text{ ft})(2/3)^{3/2}\sqrt{32.2\text{ ft/s}^2}(2.0\text{ ft})^{3/2} = \mathbf{66.0\ ft^3/s}$$

Discussion Note that the flow rate in the channel with the broad-crested weir is much less than the channel with the sharp-crested weir. Also, if the upstream velocity is taken into consideration, the flow rate would be 155 ft^3/s (4% difference) for the channel with the sharp-crested weir, and 66.6 ft^3/s (0.9% difference) for the one with broad-crested weir. Therefore, the assumption of negligible dynamic head is not quite appropriate for the channel with the sharp-crested weir.

13-139

Solution The flow of water through a parabolic notch is considered. A relation is to be developed for the flow rate, and its numerical value is to be calculated.

Assumptions **1** The flow is steady. **2** All frictional effects are negligible, and Toricelli's equation can be used for the velocity.



Analysis Consider a differential strip area shown on the sketch. It can be expressed as

$$dA = bdy = 2y\tan(\theta/2)dy$$

Noting that the flow velocity is $V = \sqrt{2g(H-y)}$, the flow rate through this differential area is

$$VdA = V\left(2y\tan(\theta/2)dy\right) = \sqrt{2g(H-y)}\,2y\tan(\theta/2)dy = 2\sqrt{2g}\,\tan(\theta/2)y\sqrt{H-y}dy$$

Then the flow rate through the entire notch is determined by integration to be

$$\dot{V} = \int_A VdA = 2\sqrt{2g}\,\tan(\theta/2)\int_{y=0}^{H} y\sqrt{H-y}dy$$

where

$$\int_{y=0}^{H} y\sqrt{H-y}dy = \left[-\frac{2}{5}y^{5/2} + \frac{2}{3}Hy^{3/2}\right]\Bigg|_0^H = \frac{4}{15}H^{5/2}$$

Then the expression for the volume flow rate and its numerical value become

$$\dot{V} = \frac{8\sqrt{2g}}{15}\tan(\theta/2)H^{5/2} = \frac{8\sqrt{2(9.81\,\text{m/s}^2)}}{15}\tan(\theta/2)(0.25)^{5/2} = 0.07382\tan(\theta/2) \quad (\text{m}^3/\text{s})$$

$\theta = \mathbf{25°}$: $\dot{V} = 0.07382\tan\left(25°/2\right) = \mathbf{0.0164\ m^3/s}$

$\theta = \mathbf{40°}$: $\dot{V} = 0.07382\tan\left(40°/2\right) = \mathbf{0.0269\ m^3/s}$

$\theta = \mathbf{60°}$: $\dot{V} = 0.07382\tan\left(60°/2\right) = \mathbf{0.0426\ m^3/s}$

$\theta = \mathbf{75°}$: $\dot{V} = 0.07382\tan\left(75°/2\right) = \mathbf{0.0566\ m^3/s}$

These results are plotted, using EES.

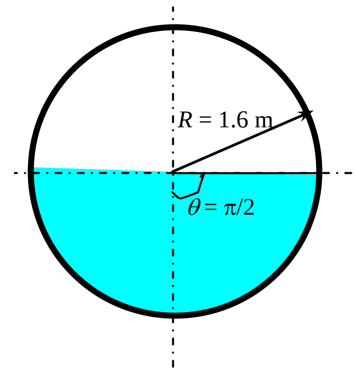
Discussion Note that a general flow rate equation for the V-notch would be in the form of $\dot{V} = K\tan(\theta/2)H^{5/2}$, where $K = C_d 8\sqrt{2g}/15$ and C_d is the discharge coefficient whose value is determined experimentally to account for nonideal effects.

13-140

Solution Water flows uniformly half-full in a circular channel. For specified flow rate and bottom slope, the Manning coefficient is to be determined.

Assumptions **1** The flow is steady and uniform. **2** Bottom slope is constant. **3** Roughness coefficient is constant along the channel.

Analysis The flow area, wetted perimeter, and hydraulic radius of the channel are



$$A_c = \frac{\pi R^2}{2} = \frac{\pi(1.6\,\text{m})^2}{2} = 4.021\,\text{m}$$

$$p = \frac{2\pi R}{2} = \frac{2\pi(1.6\,\text{m})}{2} = 5.027\,\text{m}$$

$$R_h = \frac{A_c}{p} = \frac{\pi R^2/2}{\pi R} = \frac{R}{2} = \frac{1.6\,\text{m}}{2} = 0.8\,\text{m}$$

Then the Manning coefficient can be determined from Manning's equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2}$$

from which we solve for n,

$$n = \frac{a A_c R_h^{2/3} S_0^{1/2}}{\dot{V}} = \frac{(1\,\text{m}^{1/3}/\text{s})(4.021\,\text{m}^2)(0.8\,\text{m})^{2/3}(0.004)^{1/2}}{4.5\,\text{m}^3/\text{s}} = \mathbf{0.0487}$$

When calculating the Froude number, the hydraulic depth should be used rather than the maximum depth or the hydraulic radius. For a non-rectangular channel, hydraulic depth is defined as the ratio of the flow area to top width,

$$y_h = \frac{A_c}{\text{Top width}} = \frac{\pi R^2/2}{2R} = \frac{\pi R}{4} = \frac{\pi(1.6\,\text{m})}{4} = 1.257\,\text{m}$$

$$V = \frac{\dot{V}}{A_c} = \frac{4.5\,\text{m}^3/\text{s}}{4.021\,\text{m}^2} = 1.119\,\text{m/s}$$

$$\text{Fr} = \frac{V}{\sqrt{gy}} = \frac{1.119\,\text{m/s}}{\sqrt{(9.81\,\text{m/s}^2)(1.257\,\text{m})}} = \mathbf{0.319}$$

which is lower than 1. Therefore, **the flow is *subcritical***.

Discussion It appears that this channel is made of cast iron or unplaned wood .

13-141

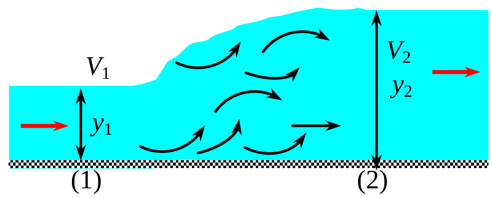
Solution Water flow through a wide rectangular channel undergoing a hydraulic jump is considered. It is to be shown that the ratio of the Froude numbers before and after the jump can be expressed in terms of flow depths y_1 and y_2 before and after the jump, respectively, as $\mathrm{Fr}_1/\mathrm{Fr}_2=\sqrt{(y_2/y_1)^3}$.

Assumptions **1** The flow is steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Analysis The Froude number for a wide channel of width *b* and flow depth *y* is given as

$$\mathrm{Fr}=\frac{V}{\sqrt{gy}}=\frac{\dot{V}/by}{\sqrt{gy}}=\frac{\dot{V}}{by\sqrt{gy}}=\frac{\dot{V}}{b\sqrt{gy^3}}$$

Expressing the Froude number before and after the jump and taking their ratio gives



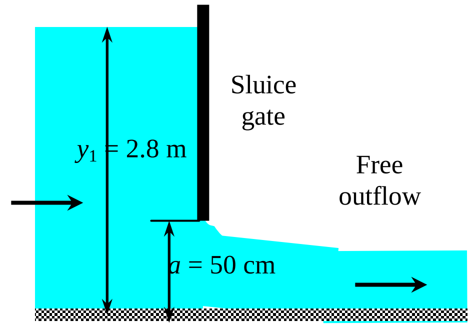
$$\boxed{\frac{\mathrm{Fr}_1}{\mathrm{Fr}_2}=\frac{\dot{V}/\left(b\sqrt{gy_1^3}\right)}{\dot{V}/\left(b\sqrt{gy_2^3}\right)}=\frac{\sqrt{gy_2^3}}{\sqrt{gy_1^3}}=\sqrt{\left(\frac{y_2}{y_1}\right)^3}}$$

which is the desired result.

Discussion Using the momentum equation, other relations such as $y_2=0.5y_1\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right)$ can also be developed.

13-142

Solution A sluice gate with free outflow is used to control the flow rate of water. For specified flow depths, the flow rate per unit width and the downstream flow depth and velocity are to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Analysis For free outflow, we only need the depth ratio y_1/a to determine the discharge coefficient (for drowned outflow, we also need to know y_2/a and thus the flow depth y_2 downstream the gate),

$$\frac{y_1}{a} = \frac{2.8\,\text{m}}{0.50\,\text{m}} = 5.6$$

The corresponding discharge coefficient is determined from Fig. 13-44 to be $C_d = 0.56$. Then the discharge rate per m width becomes

$$\dot{V} = C_d ba\sqrt{2gy_1} = 0.56\,(1\,\text{m})(0.50\,\text{m})\sqrt{2\,(9.81\,\text{m/s}^2)(2.8\,\text{m})} = 2.075\,\text{m}^3/\text{s} \cong \mathbf{2.08\,m^3/s}$$

The specific energy of a fluid remains constant during horizontal flow when the frictional effects are negligible, $E_{s1} = E_{s2}$. With these approximations, the flow depth and velocity past the gate become

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = y_1 + \frac{\dot{V}^2}{2g(by_1)^2} = 2.8\,\text{m} + \frac{(2.075\,\text{m}^3/\text{s})^2}{2(9.81\,\text{m/s}^2)[(1\,\text{m})(2.8\,\text{m})]^2} = 2.828\,\text{m}$$

$$E_{s2} = y_2 + \frac{V_2^2}{2g} = y_2 + \frac{\dot{V}^2}{2g(by_2)^2} = E_{s1} \rightarrow \quad y_2 + \frac{(2.075\,\text{m}^3/\text{s})^2}{2(9.81\,\text{m/s}^2)[(1\,\text{m})(y_2)]^2} = 2.828\,\text{m}$$

It gives y_2 = **0.294** m for flow depth as the physically meaningful root (positive and less than 2.2 m). Also,

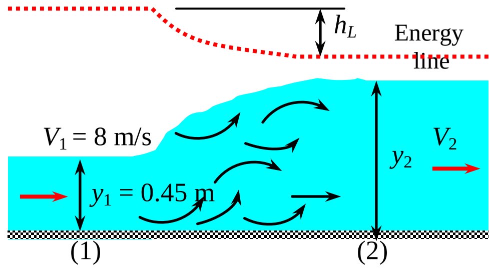
$$V_2 = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{by_2} = \frac{2.075\,\text{m}^3/\text{s}}{(1\,\text{m})(0.294\,\text{m})} = \mathbf{7.06\,m/s}$$

Discussion In actual gates some frictional losses are unavoidable, and thus the actual velocity downstream will be lower.

13-143

Solution Water at a specified depth and velocity undergoes a hydraulic jump. The fraction of mechanical energy dissipated is to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The channel is horizontal.



Analysis The Froude number before the hydraulic jump is

$$\mathrm{Fr}_1 = \frac{V_1}{\sqrt{gy_1}} = \frac{8\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(0.45\text{ m})}} = 3.8076$$

which is greater than 1. Therefore, the flow is indeed supercritical before the jump. The flow depth, velocity, and Froude number after the jump are

$$y_2 = 0.5y_1\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right) = 0.5(0.45\text{ m})\left(-1+\sqrt{1+8\times 3.8076^2}\right) = 2.2086\text{ m}$$

$$V_2 = \frac{y_1}{y_2}V_1 = \frac{0.45\text{ m}}{2.2086\text{ m}}(8\text{ m/s}) = 1.6300\text{ m/s} \qquad \mathrm{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{1.6300\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(2.2086\text{ m})}} = 0.35019$$

The head loss and the fraction of mechanical energy dissipated during the jump are

$$h_L = y_1 - y_2 + \frac{V_1^2 - V_2^2}{2g} = (0.45\text{ m})\text{-}(2.2086\text{ m}) + \frac{(8\text{ m/s})^2\text{-}(1.6300\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 1.3680\text{ m}$$

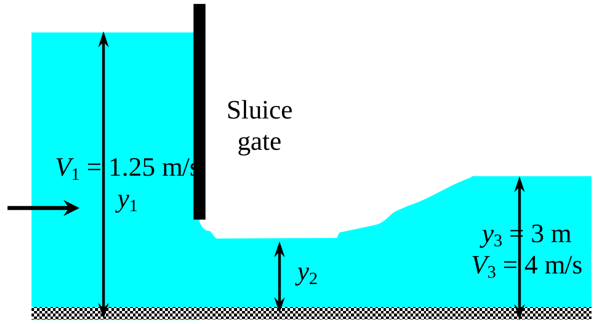
$$\text{Dissipation ratio} = \frac{h_L}{E_{s1}} = \frac{h_L}{y_1(1+\mathrm{Fr}_1^2/2)} = \frac{1.3680\text{ m}}{(0.45\text{ m})(1+3.8076^2/2)} = 0.36853$$

or, in terms of percentage, **the dissipation ratio is 36.9%**.

Discussion Note that almost over one-third of the mechanical energy of the fluid is dissipated during hydraulic jump.

13-144

Solution The flow depth and average velocity of water after a hydraulic jump together with approach velocity to sluice gate are given. The flow rate per m width, the flow depths before and after the gate, and the energy dissipation ratio are to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Analysis The flow rate per m width of channel, flow depth before the sluice gate, and the Froude number after the jump is

$$\dot{V} = V_3 A_{c3} = V_3 b y_3 = (4\text{ m/s})(1\text{ m})(3\text{ m}) = \mathbf{12\ m^3/s}$$

$$y_1 = \frac{V_3}{V_1} y_3 = \frac{4\text{ m/s}}{1.25\text{ m/s}}(3\text{ m}) = \mathbf{9.60\ m}$$

$$\text{Fr}_3 = \frac{V_3}{\sqrt{g y_3}} = \frac{4\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(3\text{ m})}} = 0.7373$$

The flow dept, velocity, and Froude number before the jump are

$$y_2 = 0.5 y_3\left(-1+\sqrt{1+8\text{Fr}_3^2}\right) = 0.5(3\text{ m})\left(-1+\sqrt{1+8\times 0.7373^2}\right) = 1.969\text{ m} \cong \mathbf{1.97\ m}$$

$$V_2 = \frac{y_3}{y_2} V_3 = \frac{3\text{ m}}{1.969\text{ m}}(4\text{ m/s}) = 6.094\text{ m/s}$$

$$\text{Fr}_2 = \frac{V_2}{\sqrt{g y_2}} = \frac{6.094\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(1.969\text{ m})}} = 1.387$$

which is greater than 1, and thus the flow before the jump is indeed supercritical. The head loss and the fraction of mechanical energy dissipated during hydraulic jump are

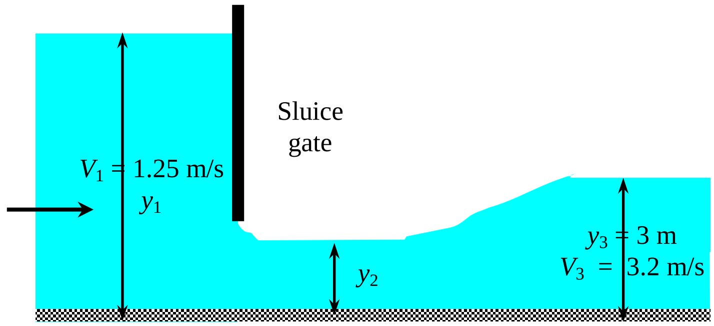
$$h_L = y_2 - y_3 + \frac{V_2^2 - V_3^2}{2g} = (1.969\text{ m}) - (3\text{ m}) + \frac{(6.094\text{ m/s})^2 - (4\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 0.0463\text{ m}$$

$$\text{Dissipation ratio} = \frac{h_L}{E_{s2}} = \frac{h_L}{y_2(1+\text{Fr}_2^2/2)} = \frac{0.0463\text{ m}}{(1.969\text{ m})(1+1.387^2/2)} = \mathbf{0.0120}$$

Discussion Note that this is a "mild" hydraulic jump, and only 1.2% of the mechanical energy is wasted.

13-145

Solution The flow depth and average velocity of water after a hydraulic jump together with approach velocity to sluice gate are given. The flow rate per m width, the flow depths before and after the gate, and the energy dissipation ratio are to be determined.



Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Analysis The flow rate per m width of channel, flow depth before the sluice gate, and the Froude number after the jump is

$$\dot{V} = V_3 A_{c3} = V_3 b y_3 = (3.2\text{ m/s})(1\text{ m})(3\text{ m}) = \mathbf{9.6\ m^3/s}$$

$$y_1 = \frac{V_3}{V_1} y_3 = \frac{3.2\text{ m/s}}{1.25\text{ m/s}}(3\text{ m}) = \mathbf{7.68\ m}$$

$$\text{Fr}_3 = \frac{V_3}{\sqrt{g y_3}} = \frac{3.2\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(3\text{ m})}} = 0.5899$$

The flow depth, velocity, and Froude number before the jump are

$$y_2 = 0.5 y_3\left(-1 + \sqrt{1 + 8\text{Fr}_3^2}\right) = 0.5(3\text{ m})\left(-1 + \sqrt{1 + 8 \times 0.5899^2}\right) = 1.418\text{ m} \cong \mathbf{1.42\ m}$$

$$V_2 = \frac{y_3}{y_2} V_3 = \frac{3\text{ m}}{1.418\text{ m}}(3.2\text{ m/s}) = 6.771\text{ m/s}$$

$$\text{Fr}_2 = \frac{V_2}{\sqrt{g y_2}} = \frac{6.771\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(1.418\text{ m})}} = 1.815$$

which is greater than 1, and thus the flow before the jump is indeed supercritical. The head loss and the fraction of mechanical energy dissipated during hydraulic jump are

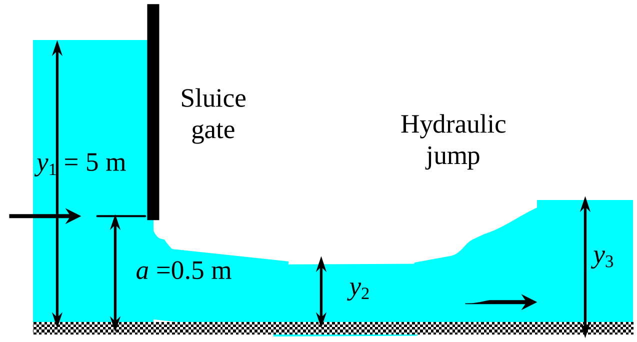
$$h_L = y_2 - y_3 + \frac{V_2^2 - V_3^2}{2g} = (1.418\text{ m}) - (3\text{ m}) + \frac{(6.771\text{ m/s})^2 - (3.2\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 0.2328\text{ m}$$

$$\text{Dissipation ratio} = \frac{h_L}{E_{s2}} = \frac{h_L}{y_2(1 + \text{Fr}_2^2/2)} = \frac{0.2328\text{ m}}{(1.418\text{ m})(1 + 1.815^2/2)} = \mathbf{0.0620}$$

Discussion Note that this hydraulic jump wastes 6.2% of the mechanical energy of the fluid.

13-146

Solution Water from a lake is discharged through a sluice gate into a channel where uniform flow conditions are established, and then undergoes a hydraulic jump. The flow depth, velocity, and Froude number after the jump are to be determined.



Assumptions **1** The flow is steady. **2** The channel is sufficiently wide so that the end effects are negligible. **3** The effects of channel slope on hydraulic jump are negligible.

Properties The Manning coefficient for an open channel made of finished concrete is n = 0.012 (Table 13-1).

Analysis For free outflow, we only need the depth ratio y_1/a to determine the discharge coefficient,

$$\frac{y_1}{a} = \frac{5\,\text{m}}{0.5\,\text{m}} = 10$$

The corresponding discharge coefficient is determined from Fig. 13-41 to be C_d = 0.58. Then the discharge rate per m width (b = 1 m) becomes

$$\dot{V} = C_d ba\sqrt{2gy_1} = 0.58(1\,\text{m})(0.5\,\text{m})\sqrt{2(9.81\,\text{m/s}^2)(5\,\text{m})} = 2.872\,\text{m}^3/\text{s}$$

For wide channels, hydraulic radius is the flow depth and thus $R_h = y_2$. Then the flow depth in uniform flow after the gate is determined from the Manning's equation to be

$$\dot{V} = \frac{a}{n} A_c R_h^{2/3} S_0^{1/2} \quad\rightarrow\quad 2.872\,\text{m}^3/\text{s} = \frac{1\,\text{m}^{1/3}/s}{0.012}[(1\,\text{m})y_2](y_2)^{2/3} 0.004^{1/2}$$

It gives y_2 = 0.6948 m, which is also the flow depth before water undergoes a hydraulic jump. The flow velocity and Froude number in uniform flow are

$$V_2 = \frac{\dot{V}}{b\,y_2} = \frac{2.872\,\text{m}^3/\text{s}}{(1\,\text{m})(0.6948\,\text{m})} = 4.134\,\text{m/s}$$

$$\text{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{4.134\,\text{m/s}}{\sqrt{(9.81\,\text{m/s}^2)(0.6948\,\text{m})}} = 1.584$$

Then the flow depth, velocity, and Froude number after the jump (state 3) become

$$y_3 = 0.5y_2\left(-1+\sqrt{1+8\text{Fr}_2^2}\right) = 0.5(0.6948\,\text{m})\left(-1+\sqrt{1+8\times 1.584^2}\right) = \mathbf{1.25\,m}$$

$$V_3 = \frac{y_2}{y_3}V_2 = \frac{0.6948\,\text{m}}{1.25\,\text{m}}(4.134\,\text{m/s}) = \mathbf{2.30\,m/s}$$

$$\text{Fr}_3 = \frac{V_3}{\sqrt{gy_{23}}} = \frac{2.30\,\text{m/s}}{\sqrt{(9.81\,\text{m/s}^2)(1.25\,\text{m})}} = \mathbf{0.659}$$

Discussion This is a relatively "mild" jump. It can be shown that the head loss during hydraulic jump is 0.049 m, which corresponds to an energy dissipation ratio of 3.1%.

13-147

Solution Water is discharged from a dam into a wide spillway to reduce the risk of flooding by dissipating a large fraction of mechanical energy via hydraulic jump. For specified flow depths, the velocities before and after the jump, and the mechanical power dissipated per meter with of the spillway are to be determined.

Assumptions **1** The flow is steady or quasi-steady. **2** The channel is sufficiently wide so that the end effects are negligible.

Properties The density of water is 1000 kg/m^3.

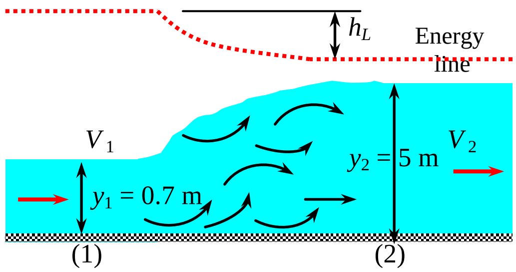
Analysis The Froude number and velocity before the jump are

$$\frac{y_2}{y_1} = 0.5\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right) \rightarrow \frac{5\text{ m}}{0.7\text{ m}} = 0.5\left(-1+\sqrt{1+8\mathrm{Fr}_1^2}\right)$$

which gives Fr_1 = 5.393. Also, from the definition of Froude number,

$$V_1 = \mathrm{Fr}_1\sqrt{gy_1} = (5.393)\sqrt{(9.81\text{ m/s}^2)(0.7\text{ m})} = 14.13\text{ m/s} \cong \mathbf{14.1\ m/s}$$

Velocity and Froude number after the jump are



$$V_2 = \frac{y_1}{y_2}V_1 = \frac{0.7\text{ m}}{5\text{ m}}(14.13\text{ m/s}) = 1.978\text{ m/s} \cong \mathbf{1.98\ m/s}$$

$$\mathrm{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{1.978\text{ m/s}}{\sqrt{(9.81\text{ m/s}^2)(5\text{ m})}} = 0.2825$$

The head loss is determined from the energy equation to be

$$h_L = y_1 - y_2 + \frac{V_1^2 - V_2^2}{2g} = (0.7\text{ m}) - (5\text{ m}) + \frac{(14.13\text{ m/s})^2 - (1.978\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 5.679\text{ m}$$

The volume and mass flow rates of water per m width are

$$\dot{V} = V_1 A_{c1} = V_1 b y_1 = (14.13\text{ m/s})(1\text{ m})(0.7\text{ m}) = 9.892\text{ m}^3\text{/s}$$

$$\dot{m} = \rho\dot{V} = (1000\text{ kg/m}^3)(9.892\text{ m}^3\text{/s}) = 9892\text{ kg/s}$$

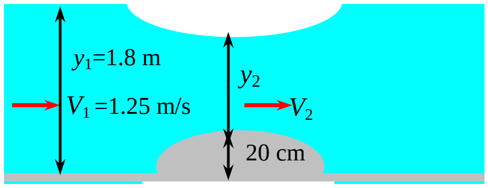
Then the dissipated mechanical power becomes

$$\dot{E}_{\text{dissipated}} = \dot{m}gh_L = (9892\text{ kg/s})(9.81\text{ m/s}^2)(5.679\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 551.1\text{ kN}\cdot\text{m/s} \cong \mathbf{551\ kW}$$

Discussion The results show that the hydraulic jump is a highly dissipative process, wasting 551 kW of power in this case.

13-148

Solution Water flowing in a horizontal open channel encounters a bump. Flow properties over the bump are to be determined.



Assumptions **1** The flow is steady. **2** Frictional effects are negligible so that there is no dissipation of mechanical energy. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The upstream Froude number and the critical depth are

$$\mathrm{Fr}_1 = \frac{V_1}{\sqrt{gy_1}} = \frac{1.25\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m^2/s})(1.8\ \mathrm{m})}} = 0.297$$

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(by_1V_1)^2}{gb^2}\right)^{1/3} = \left(\frac{y_1^2V_1^2}{g}\right)^{1/3} = \left(\frac{(1.8\ \mathrm{m})^2(1.25\ \mathrm{m/s})^2}{9.81\ \mathrm{m/s^2}}\right)^{1/3} = 0.802\ \mathrm{m}$$

The upstream flow is subcritical since Fr < 1, and the flow depth decreases over the bump. The upstream, over the bump, and critical specific energy are

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = (1.80\ \mathrm{m}) + \frac{(1.25\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})} = 1.88\ \mathrm{m}$$

The flow depth over the bump can be determined from

$$y_2^3 - (E_{s1} - \Delta z_b)y_2^2 + \frac{V_1^2}{2g}y_1^2 = 0 \quad \rightarrow \quad y_2^3 - (1.88 - 0.20\ \mathrm{m})y_2^2 + \frac{(1.25\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})}(1.80\ \mathrm{m})^2 = 0$$

Using an equation solver, the physically meaningful root of this equation is determined to be y_2 = **1.576 m**. Then,

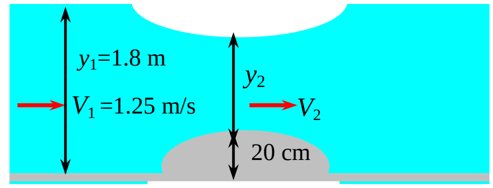
$$V_2 = \frac{y_1}{y_2}V_1 = \frac{1.8\ \mathrm{m}}{1.576\ \mathrm{m}}(1.25\ \mathrm{m/s}) = \mathbf{1.43\ m/s}$$

$$\mathrm{Fr}_2 = \frac{V_2}{\sqrt{gy_2}} = \frac{1.428\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m/s^2})(1.576\ \mathrm{m})}} = \mathbf{0.363}$$

Discussion The actual values may be somewhat different than those given above because of the frictional effects that are neglected in the analysis.

13-149

Solution Water flowing in a horizontal open channel encounters a bump. The bump height for which the flow over the bump is critical is to be determined.



Assumptions **1** The flow is steady. **2** Frictional effects are negligible so that there is no dissipation of mechanical energy. **3** The channel is sufficiently wide so that the end effects are negligible.

Analysis The upstream Froude number and the critical depth are

$$\mathrm{Fr}_1 = \frac{V_1}{\sqrt{gy_1}} = \frac{1.25\ \mathrm{m/s}}{\sqrt{(9.81\ \mathrm{m^2/s})(1.8\ \mathrm{m})}} = 0.297$$

$$y_c = \left(\frac{\dot{V}^2}{gb^2}\right)^{1/3} = \left(\frac{(by_1V_1)^2}{gb^2}\right)^{1/3} = \left(\frac{y_1^2V_1^2}{g}\right)^{1/3} = \left(\frac{(1.8\ \mathrm{m})^2(1.25\ \mathrm{m/s})^2}{9.81\ \mathrm{m/s^2}}\right)^{1/3} = 0.802\ \mathrm{m}$$

The upstream flow is subcritical since Fr < 1, and the flow depth decreases over the bump. The upstream specific energy is

$$E_{s1} = y_1 + \frac{V_1^2}{2g} = (1.80\ \mathrm{m}) + \frac{(1.25\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})} = 1.88\ \mathrm{m}$$

Noting that the flow over the bump is critical and that $E_{s2} = E_{s1} - \Delta z_b$,

$$E_{s2} = E_c = \frac{3}{2}y_c = \frac{3}{2}(0.802\ \mathrm{m}) = 1.20\ \mathrm{m}$$

and

$$\Delta z_b = E_{s1} - E_{s2} = 1.88 - 1.20 = \mathbf{0.68\,m}$$

Discussion If a higher bump is used, the flow will remain critical but the flow rate will decrease (the choking effect).

Fundamentals of Engineering (FE) Exam Problems

13-150

Which ones are examples of open-channel flow?

I. Flow of water in rivers

II. Draining of rainwater off highways

III. Upward draft of rain and snow

IV. Sewer lines

(*a*) I and II (*b*) I and III (*c*) II and III (*d*) I, II, and IV (*e*) I, II, III, and IV

Answer (*d*) I, II, and IV

13-151

If the flow depth remains constant in an open-channel flow, the flow is called

(*a*) Uniform flow (*b*) Steady flow (*c*) Varied flow (*d*) Unsteady flow

(*e*) Laminar flow

Answer (*a*) Uniform flow

13-152

Consider water flow in a rectangular open channel of height 2 m and width 5 m containing water of depth 1.5 m. The hydraulic radius for this flow is

(*a*) 0.47 m (*b*) 0.94 m (*c*) 1.5 m (*d*) 3.8 m (*e*) 5 m

Answer (*b*) 0.94 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h=2 [m]
b=5 [m]
y=1.5 [m]
A_c=y*b
p=b+2*y
R_h=A_c/p
```

13-153

Water flows in a rectangular open channel of width 5 m at a rate of 7.5 m^3/s. The critical depth for this flow is

(*a*) 5 m (*b*) 2.5 m (*c*) 1.5 m (*d*) 0.96 m (*e*) 0.61 m

Answer (*e*) 0.61 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=5 [m]
V_dot=7.5 [m^3/s]
g=9.81 [m/s^2]
y_c=(V_dot^2/(g*b^2))^(1/3)
```

13-154

Water flows in a rectangular open channel of width 0.6 m at a rate of 0.25 m^3/s. If the flow depth is 0.2 m, what is the alternate flow depth if the character of flow were to change?

(*a*) 0.2 m (*b*) 0.26 m (*c*) 0.35 m (*d*) 0.6 m (*e*) 0.8 m

Answer (*c*) 0.35 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=0.6 [m]
V_dot=0.25 [m^3/s]
y1=0.2 [m]
g=9.81 [m/s^2]
E_s1=y1+V_dot^2/(2*g*b^2*y1^2)
E_s2=y2+V_dot^2/(2*g*b^2*y2^2)
E_s1=E_s2
```

13-155

Water flows in a 6-m-wide rectangular open channel at a rate of 55 m^3/s. If the flow depth is 2.4 m, the Froude number is

(*a*) 0.531 (*b*) 0.787 (*c*) 1.0 (*d*) 1.72 (*e*) 2.65

Answer (*b*) 0.787

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=6 [m]
V_dot=55 [m^3/s]
y=2.4 [m]
g=9.81 [m/s^2]
A_c=y*b
V=V_dot/A_c
Fr=V/sqrt(g*y)
```

13-156

Water flows in a clean and straight natural channel of rectangular cross section with a bottom width of 0.75 m and a bottom slope angle of 0.6°. If the flow depth is 0.15 m, the flow rate of water through the channel is

(*a*) 0.0317 m^3/s (*b*) 0.05 m^3/s (*c*) 0.0674 m^3/s (*d*) 0.0866 m^3/s (*e*) 1.14 m^3/s

Answer (*d*) 0.0866 m^3/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=0.75 [m]
alpha=0.6 [degrees]
y=0.15 [m]
a=1 [m^(1/3)/s]
n=0.030 "from Table 13-1"
g=9.81 [m/s^2]
A_c=y*b
p=b+2*y
R_h=A_c/p
S_0=tan(alpha)
V_dot=a/n*A_c*R_h^(2/3)*S_0^(1/2)
```

13-157

Water is to be transported in a finished-concrete rectangular channel with a bottom width of 1.2 m at a rate of 5 m^3/s. The channel bottom drops 1 m per 500 m length. The minimum height of the channel under uniform-flow conditions is

(*a*) 1.9 m (*b*) 1.5 m (*c*) 1.2 m (*d*) 0.92 m (*e*) 0.60 m

Answer (*a*) 1.9 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=1.2 [m]
V_dot=5 [m^3/s]
S_0=1/500
a=1 [m^(1/3)/s]
n=0.012 "from Table 13-1"
g=9.81 [m/s^2]
A_c=y*b
p=b+2*y
R_h=A_c/p
V_dot=a/n*A_c*R_h^(2/3)*S_0^(1/2)
```

13-158

Water is to be transported in a 4-m-wide rectangular open channel. The flow depth to maximize the flow rate is

(*a*) 1 m (*b*) 2 m (*c*) 4 m (*d*) 6 m (*e*) 8 m

Answer (*b*) 2 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=4 [m]
y=b/2
```

13-159

Water is to be transported in a clay tile lined rectangular channel at a rate of 0.8 m^3/s. The channel bottom slope is 0.0015. The width of the channel for the best cross section is

(*a*) 0.68 m (*b*) 1.33 m (*c*) 1.63 m (*d*) 0.98 m (*e*) 1.15 m

Answer (*e*) 1.15 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_dot=0.8 [m^3/s]
S_0=0.0015
a=1 [m^(1/3)/s]
n=0.014 "from Table 13-1"
g=9.81 [m/s^2]
A_c=y*b
p=b+2*y
R_h=A_c/p
y=b/2
V_dot=a/n*A_c*R_h^(2/3)*S_0^(1/2)
```

13-160

Water is to be transported in a clay tile lined trapezoidal channel at a rate of 0.8 m^3/s. The channel bottom slope is 0.0015. The width of the channel for the best cross section is

(*a*) 0.48 m (*b*) 0.70 m (*c*) 0.84 m (*d*) 0.95 m (*e*) 1.22 m

Answer (*b*) 0.70 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_dot=0.8 [m^3/s]
S_0=0.0015
a=1 [m^(1/3)/s]
n=0.014 "from Table 13-1"
g=9.81 [m/s^2]
theta=60 [degrees]
A_c=y*(b+b*cos(theta))
p=3*b
R_h=y/2
y=sqrt(3)/2*b
V_dot=a/n*A_c*R_h^(2/3)*S_0^(1/2)
```

13-161

Water flows uniformly in a finished-concrete rectangular channel with a bottom width of 0.85 m. The flow depth is 0.4 m and the bottom slope is 0.003. The channel should be classified as

(*a*) Steep (*b*) Critical (*c*) Mild (*d*) Horizontal (*e*) Adverse

Answer (*c*) Mild

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=0.85 [m]
y=0.4 [m]
S_0=0.003
a=1 [m^(1/3)/s]
n=0.012 "from Table 13-1"
g=9.81 [m/s^2]
A_c=y*b
p=b+2*y
R_h=A_c/p
V_dot=a/n*A_c*R_h^(2/3)*S_0^(1/2)
y_c=(V_dot^2/(g*b^2))^(1/3)
"Since y_n = y = 0.4 m is greater than y_c = 0.35 m, the flow is mild"
```

13-162

Water discharges into a rectangular horizontal channel from a sluice gate and undergoes a hydraulic jump. The channel is 25-m-wide and the flow depth and velocity before the jump are 2 m and 9 m/s, respectively. The flow depth after the jump is

(*a*) 1.26 m (*b*) 2 m (*c*) 3.61 m (*d*) 4.83 m (*e*) 6.55 m

Answer (*d*) 4.83 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=25 [m]
y1=2 [m]
V1=9 [m/s]
g=9.81 [m/s^2]
Fr_1=V1/sqrt(g*y1)
y2=0.5*y1*(-1+sqrt(1+8*Fr_1^2))
```

13-163

Water discharges into a rectangular horizontal channel from a sluice gate and undergoes a hydraulic jump. The flow depth and velocity before the jump are 1.25 m and 6 m/s, respectively. The percentage available head loss due to the hydraulic jump is

(*a*) 4.7% (*b*) 6.2% (*c*) 8.5% (*d*) 13.9% (*e*) 17.4%

Answer (*a*) 4.7%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
y1=1.25 [m]
V1=6 [m/s]
g=9.81 [m/s^2]
Fr_1=V1/sqrt(g*y1)
y2=0.5*y1*(-1+sqrt(1+8*Fr_1^2))
V2=y1/y2*V1
h_L=y1-y2+(V1^2-V2^2)/(2*g)
E_s1=y1+V1^2/(2*g)
DR=h_L/E_s1
PercentLoss=DR*Convert(,%)
```

13-164

Water discharges into a 7-m-wide rectangular horizontal channel from a sluice gate and undergoes a hydraulic jump. The flow depth and velocity before the jump are 0.65 m and 5 m/s, respectively. The wasted power potential due to the hydraulic jump is

(*a*) 158 kW (*b*) 112 kW (*c*) 67.3 kW (*d*) 50.4 kW (*e*) 37.6 kW

Answer (*e*) 37.6 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=7 [m]
y1=0.65 [m]
V1=5 [m/s]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
Fr_1=V1/sqrt(g*y1)
y2=0.5*y1*(-1+sqrt(1+8*Fr_1^2))
V2=y1/y2*V1
h_L=y1-y2+(V1^2-V2^2)/(2*g)
m_dot=rho*b*y1*V1
E_dot_wasted=m_dot*g*h_L*Convert(W, kW)
```

13-165

Water is released from a 0.8-m-deep reservoir into a 4-m-wide open channel through a sluice gate with a 0.1-m-high opening at the channel bottom. The flow depth after all turbulence subsides is 0.5 m. The rate of discharge is

(*a*) 0.92 m^3/s (*b*) 0.79 m^3/s (*c*) 0.66 m^3/s (*d*) 0.47 m^3/s (*e*) 0.34 m^3/s

Answer (*c*) 0.66 m^3/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
y1=0.8 [m]
b=4 [m]
a=0.1 [m]
y2=0.5 [m]
g=9.81 [m/s^2]
y1\a=y1/a
y2\a=y2/a
C_d=0.415 "from Fig. 13-44 at y1/a and y2/a"
V_dot=C_d*b*a*sqrt(2*g*y1)
```

13-166

The flow rate of water in a 3-m-wide horizontal open channel is being measured with a 0.4-m-high sharp-crested rectangular weir of equal width. If the water depth upstream is 0.9 m, the flow rate of water is

(*a*) 1.37 m^3/s (*b*) 2.22 m^3/s (*c*) 3.06 m^3/s (*d*) 4.68 m^3/s (*e*) 5.11 m^3/s

Answer (*b*) 2.22 m^3/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
b=3 [m]
y1=0.9 [m]
P_w=0.4 [m]
g=9.81 [m/s^2]
H=y1-P_w
C_wd_rec=0.598+0.0897*H/P_w
V_dot_rec=C_wd_rec*2/3*b*sqrt(2*g)*H^(3/2)
```

Design and Essay Problems

13-167 to 13-168

Solution Students' essays and designs should be unique and will differ from each other.

Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

Third Edition

Yunus A. Çengel & John M. Cimbala

McGraw-Hill, 2013

Chapter 14
TURBOMACHINERY

General Problems

14-1C

Solution We are to list examples of fans, blowers, and compressors.

Analysis Common examples of fans are **window fans, ceiling fans, fans in computers and other electronics equipment, radiator fans in cars**, etc. Common examples of blowers are **leaf blowers, hair dryers, air blowers in furnaces and automobile ventilation systems**. Common examples of compressors are **tire pumps, refrigerator and air conditioner compressors**.

Discussion Students should come up with a diverse variety of examples.

14-2C

Solution We are to discuss the differences between fans, blowers, and compressors.

Analysis A *fan* is a gas pump with relatively l**ow pressure rise** and **high flow rate**. A ***blower*** is a gas pump with relatively **moderate to high pressure rise** and **moderate to high flow rate**. A ***compressor*** is a gas pump designed to deliver a very **high pressure rise**, typically at **low to moderate flow rates**.

Discussion The boundaries between these three types of pump are not always clearly defined.

14-3C

Solution We are to discuss energy producing and energy absorbing devices.

Analysis A more common term for an energy producing turbomachine is a **turbine**. **Turbines extract energy from the moving fluid, and convert that energy into useful mechanical energy in the surroundings**, usually in the form of a rotating shaft. Thus, the phrase "energy producing" is from a frame of reference of the fluid – the fluid loses energy as it drives the turbine, producing energy to the surroundings. On the other hand, a more common term for an energy absorbing turbomachine is a **pump**. **Pumps absorb mechanical energy from the surroundings, usually in the form of a rotating shaft, and increase the energy of the moving fluid**. Thus, the phrase "energy absorbing" is from a frame of reference of the fluid – the fluid gains or absorbs energy as it flows through the pump.

Discussion From the frame of reference of the surroundings, a pump absorbs energy from the surroundings, while a turbine produces energy to the surroundings. Thus, you may argue that the terminology also holds for the frame of reference of the surroundings. This alternative explanation is also acceptable.

14-4C

Solution We are to discuss the difference between a positive-displacement turbomachine and a dynamic turbomachine.

Analysis **A *positive-displacement turbomachine* is a device that contains a closed volume; energy is transferred to the fluid (pump) or from the fluid (turbine) via movement of the boundaries of the closed volume**. On the other hand, **a *dynamic turbomachine* has no closed volume; instead, energy is transferred to the fluid (pump) or from the fluid (turbine) via rotating blades**. Examples of positive-displacement pumps include well **pumps, hearts, some aquarium pumps, and pumps designed to release precise volumes of medicine**. Examples of positive-displacement turbines include **water meters and gas meters in the home**. Examples of dynamic pumps include **fans, centrifugal blowers, airplane propellers, centrifugal water pumps** (like in a car engine), etc. Examples of dynamic turbines include **windmills, wind turbines, turbine flow meters**, etc.

Discussion Students should come up with a diverse variety of examples.

14-5C

Solution We are to explain the "extra" term in the Bernoulli equation in a rotating reference frame.

Analysis A rotating reference frame is not an inertial reference frame. When we move outward in the radial direction, the absolute velocity at this location is faster due to the rotating body, since v_θ is equal to ωr. When solving a turbomachinery problem in a rotating reference frame, we use the *relative* fluid velocity (velocity relative to the rotating reference frame). Thus, **in order for the Bernoulli equation to be physically correct, we must subtract the absolute velocity of the rotating body so that the equation applies to an inertial reference frame. This accounts for the "extra" term**.

Discussion The Bernoulli equation is the same physical equation in either the absolute or the rotating reference frame, but it is more convenient to use the form with the extra term in turbomachinery applications.

14-6C

Solution We are to discuss the difference between brake horsepower and water horsepower, and then discuss turbine efficiency.

Analysis In turbomachinery terminology, **brake horsepower is the power actually delivered by the turbine to the shaft.** (One may also call it "shaft power".) On the other hand, **water horsepower is the power extracted from the water flowing through the turbine**. Water horsepower is always greater than brake horsepower; because of inefficiencies; hence **turbine efficiency is defined as the ratio of brake horsepower to water horsepower**.

Discussion For a pump, efficiency is defined in the opposite way, since brake horsepower is *greater* than water horsepower.

14-7C

Solution We are to discuss the difference between brake horsepower and water horsepower, and then discuss pump efficiency.

Analysis In turbomachinery terminology, **brake horsepower is the power actually delivered to the pump through the shaft**. (One may also call it "shaft power".) On the other hand, **water horsepower is the useful portion of the brake horsepower that is actually delivered to the fluid**. Water horsepower is always less than brake horsepower; hence **pump efficiency is defined as the ratio of water horsepower to brake horsepower**.

Discussion For a turbine, efficiency is defined in the opposite way, since brake horsepower is *less* than water horsepower.

14-8

Solution For an air compressor with equal inlet and outlet areas, and with both density and pressure increasing, we are to determine how the average speed at the outlet compares to the average speed at the inlet.

Assumptions **1** The flow is steady.

Analysis Conservation of mass requires that the mass flow rate in equals the mass flow rate out. The cross-sectional areas of the inlet and outlet are the same. Thus,

Conservation of mass: $\dot{m}_{\text{in}} = \rho_{\text{in}} V_{\text{in}} A_{\text{in}} = \dot{m}_{\text{out}} = \rho_{\text{out}} V_{\text{out}} A_{\text{out}}$

or

$$V_{\text{out}} = V_{\text{in}} \frac{\rho_{\text{in}}}{\rho_{\text{out}}} \qquad (1)$$

Since $\rho_{\text{in}} < \rho_{\text{out}}$, V_{out} must be **less than** V_{in}.

Discussion A compressor does not necessarily increase the speed of the fluid passing through it. In fact, the average speed through the pump can actually *decrease*, as it does here.

14-9

Solution We are to determine how the average speed at the outlet compares to the average speed at the inlet of a water pump.

Assumptions **1** The flow is steady (in the mean). **2** The water is incompressible.

Analysis Conservation of mass requires that the mass flow rate in equals the mass flow rate out. Thus,

Conservation of mass: $\dot{m}_{\text{in}} = \rho_{\text{in}} V_{\text{in}} A_{\text{in}} = \dot{m}_{\text{out}} = \rho_{\text{out}} V_{\text{out}} A_{\text{out}}$

or, since the cross-sectional area is proportional to the square of diameter,

$$V_{\text{out}} = V_{\text{in}} \frac{\rho_{\text{in}}}{\rho_{\text{out}}} \left(\frac{D_{\text{in}}}{D_{\text{out}}} \right)^2 = V_{\text{in}} \left(\frac{D_{\text{in}}}{D_{\text{out}}} \right)^2 \qquad (1)$$

(a) For the case where $D_{\text{out}} < D_{\text{in}}$, V_{out} must be **greater than** V_{in}.

(b) For the case where $D_{\text{out}} = D_{\text{in}}$, V_{out} must be **equal to** V_{in}.

(c) For the case where $D_{\text{out}} > D_{\text{in}}$, V_{out} must be **less than** V_{in}.

Discussion A pump does not necessarily increase the speed of the fluid passing through it. In fact, the average speed through the pump can actually *decrease*, as it does here in part (c).

Pumps

14-10C

Solution We are to define and discuss NPSH and $\text{NPSH}_{\text{required}}$.

Analysis Net positive suction head (NPSH) is defined as **the difference between the pump's inlet stagnation pressure head and the vapor pressure head**,

$$\boxed{\text{NPSH} = \left(\frac{P}{\rho g} + \frac{V^2}{2g}\right)_{\text{pump inlet}} - \frac{P_v}{\rho g}}$$

We may think of NPSH as the actual or available net positive suction head. On the other hand, required net positive suction head ($\text{NPSH}_{\text{required}}$) is defined as the **minimum NPSH necessary to avoid cavitation in the pump**. As long as the actual NPSH is greater than $\text{NPSH}_{\text{required}}$, there should be no cavitation in the pump.

Discussion Although NPSH and $\text{NPSH}_{\text{required}}$ are measured at the pump inlet, cavitation (if present) happens somewhere inside the pump, typically on the suction surface of the rotating pump impeller blades.

14-11C

Solution

(*a*) ***False***: Actually, backward-inclined blades yield the highest efficiency.
(*b*) ***True***: The pressure rise is higher, but at the cost of less efficiency.
(*c*) ***True***: In fact, this is the primary reason for choosing forward-inclined blades.
(*d*) ***False***: Actually, the opposite is true – a pump with forward-inclined blades usually has more blades, but they are usually smaller.

14-12C

Solution We are to choose which pump location is better and explain why.

Analysis The two systems are identical except for the location of the pump (and some minor differences in pipe layout). The overall length of pipe, number of elbows, elevation difference between the two reservoir free surfaces, etc. are the same. **Option (a) is better because it has the pump at a lower elevation, increasing the net positive suction head, and lowering the possibility of pump cavitation**. In addition, the length of pipe from the lower reservoir to the pump inlet is smaller in Option (a), and there is one less elbow between the lower reservoir and the pump inlet, thereby decreasing the head loss upstream of the pump – both of which also increase NPSH, and reduce the likelihood of cavitation.

Discussion Another point is that if the pump is not self-priming, Option (b) may run into start-up problems if the free surface of the lower reservoir falls below the elevation of the pump inlet. Since the pump in Option (a) is below the reservoir, self-priming is not an issue.

14-13C

Solution We are to list and define the three categories of dynamic pumps.

Analysis The three categories are: **Centrifugal flow pump** – fluid enters axially (in the same direction as the axis of the rotating shaft) in the center of the pump, but is discharged radially (or tangentially) along the outer radius of the pump casing. **Axial-flow pump** – fluid enters and leaves axially, typically only along the outer portion of the pump because of blockage by the shaft, motor, hub, etc. **Mixed-flow pump** – intermediate between centrifugal and axial, with the flow entering axially, not necessarily in the center, but leaving at some angle between radially and axially.

Discussion There are also some non-rotary dynamic pumps, such as jet pumps and electromagnetic pumps, that are not discussed in this text.

14-14C

Solution

(*a*) ***True***: As volume flow rate increases, not only does $\text{NPSH}_{\text{required}}$ increase, but the available NPSH decreases, increasing the likelihood that NPSH will drop below $\text{NPSH}_{\text{required}}$ and cause cavitation to occur.

(*b*) ***False***: $\text{NPSH}_{\text{required}}$ is *not* a function of water temperature, although available NPSH does depend on water temperature.

(*c*) ***False***: Available NPSH actually *decreases* with increasing water temperature, making cavitation more likely to occur.

(*d*) ***False***: Actually, warmer water causes cavitation to be *more likely*. The best way to think about this is that warmer water is already closer to its boiling point, so cavitation is more likely to happen in warm water than in cold water.

14-15C

Solution We are to discuss ways to improve the cavitation performance of a pump, based on the equation for NPSH.

Analysis NPSH is defined as

$$\boxed{\text{NPSH} = \left(\frac{P}{\rho g} + \frac{V^2}{2g}\right)_{\text{pump inlet}} - \frac{P_v}{\rho g}} \qquad (1)$$

To avoid cavitation, NPSH must be increased as much as possible. For a given liquid at a given temperature, the vapor pressure head (last term on the right side of Eq. 1) is constant. Hence, the only way to increase NPSH is to increase the stagnation pressure head at the pump inlet. We list several ways to increase the available NPSH: (1) **Lower the pump or raise the inlet reservoir level. (2) Use a larger diameter pipe upstream of the pump. (3) Re-route the piping system such that fewer minor losses (elbows, valves, etc.) are encountered upstream of the pump. (4) Shorten the length of pipe upstream of the pump. (5) Use a smoother pipe. (6) Use elbows, valves, inlets, etc. that have smaller minor loss coefficients**. Suggestion (1) raises NPSH by increasing the hydrostatic component of pressure at the pump inlet. Suggestions (2) through (6) raise NPSH by lowering the irreversible head losses, thereby increasing the pressure at the pump inlet.

Discussion By definition, when the available NPSH falls below the required NPSH, the pump is prone to cavitation, which should be avoided if at all possible.

14-16C

Solution

(*a*) ***True***: The maximum volume flow rate occurs when the net head is zero, and this "free delivery" flow rate is typically much higher than that at the BEP.

(*b*) ***True***: By definition, there is no flow rate at the shutoff head. Thus the pump is not doing any useful work, and the efficiency must be zero.

(*c*) ***False***: Actually, the net head is typically greatest near the shutoff head, at zero volume flow rate, not near the BEP.

(*d*) ***True***: By definition, there is no head at the pump's free delivery. Thus, the pump is working against no "resistance", and is therefore not doing any useful work, and the efficiency must be zero.

14-17C

Solution We are to explain why dissimilar pumps should not be arranged in series or in parallel.

Analysis Arranging dissimilar **pumps in series can create problems because the volume flow rate through each pump must be the same, but the overall pressure rise is equal to the pressure rise of one pump plus that of the other**. If the pumps have widely different performance curves, the smaller pump may be forced to operate beyond its free delivery flow rate, whereupon it acts like a head *loss*, reducing the total volume flow rate. Arranging dissimilar **pumps in parallel can create problems because the overall pressure rise must be the same, but the net volume flow rate is the sum of that through each branch**. If the pumps are not sized properly, the smaller pump may not be able to handle the large head imposed on it, and the flow in its branch could actually be *reversed*; this would inadvertently reduce the overall pressure rise. In either case, the power supplied to the smaller pump would be wasted.

Discussion If the pumps are not significantly dissimilar, a series or parallel arrangement of the pumps might be wise.

14-18C

Solution

(a) ***False***: Since the pumps are in series, the volume flow rate through each pump must be the same: $\dot{V} = \dot{V}_1 = \dot{V}_2$.

(b) ***True***: The net head increases by H_1 through the first pump, and then by H_2 through the second pump. The overall rise in net head is thus the sum of the two.

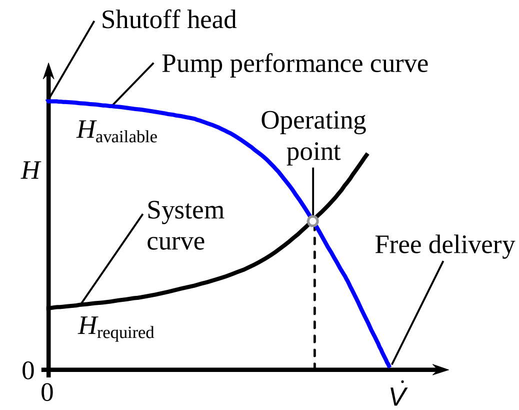
(c) ***True***: Since the pumps are in parallel, the total volume flow rate is the sum of the individual volume flow rates.

(d) ***False***: For pumps in parallel, the change in pressure from the upstream junction to the downstream junction is the same regardless of which parallel branch is under consideration. Thus, even though the volume flow rate may not be the same in each branch, the net head must be the same: $H = H_1 = H_2$.

14-19C

Solution We are to label several items on the provided plot.

Analysis The figure is re-drawn here, and the requested items are labeled.



Discussion Also labeled are the available net head, corresponding to the pump performance curve, and the required net head, corresponding to the system curve. The intersection of these two curves is the operating point of the pump.

14-20

Solution We are to determine which free surface is at higher elevation, and justify our answer with the energy equation.

Analysis It is simplest to consider zero-flow conditions ($\dot{V}$ = 0), at which we see that the required net head is positive. This implies that, even when there is no flow between the two tanks, the pump would need to provide some net head just to overcome the pressure differences. Since there is no flow, pressure differences can come only from gravity. Hence, **the outlet tank's free surface must be higher than that of the inlet tank**. Mathematically, we apply the energy equation in head form between the inlet tank's free surface (1) and the outlet tank's free surface (2),

Energy eq. at zero flow conditions:

$$H_{\text{required}} = h_{\text{pump,u}} = \cancel{\frac{P_2 - P_1}{\rho g}} + \frac{\alpha_2 \cancel{V_2^2} - \alpha_1 \cancel{V_1^2}}{2g} + \left(z_2 - z_1\right) + \cancel{h_{\text{turbine}}} + \cancel{h_{L,\text{total}}} \qquad (1)$$

Since both free surfaces are at atmospheric pressure, $P_1 = P_2 = P_{\text{atm}}$, and the first term on the right side of Eq. 1 vanishes. Furthermore, since there is no flow, $V_1 = V_2 = 0$, and the second term vanishes. There is no turbine in the control volume, so the second-to-last term is zero. Finally, there are no irreversible head losses since there is no flow, and the last term is also zero. Equation 1 reduces to

$$H_{\text{required}} = h_{\text{pump}} = \left(z_2 - z_1\right) \qquad (2)$$

Since H_{required} is positive on Fig. P14-19 at $\dot{V}$ = 0, the quantity ($z_2 - z_1$) must also be positive by Eq. 2. Thus we have shown mathematically that **the outlet tank's free surface is higher in elevation than that of the inlet tank**.

Discussion If the reverse were true (outlet tank free surface lower than inlet tank free surface), H_{required} at $\dot{V}$ = 0 would be *negative*, implying that the pump would need to supply enough *negative* net head to hold back the natural tendency of the water to flow from higher to lower elevation. In reality, the pump would not be able to do this unless it were spun backwards.

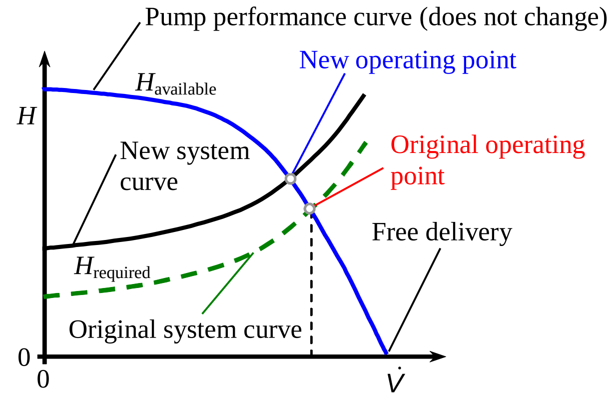
14-21

Solution We are to discuss what would happen to the pump performance curve, the system curve, and the operating point if the free surface of the outlet tank were raised to a higher elevation.

Analysis The pump is the same pump regardless of the locations of the inlet and outlet tanks' free surfaces; thus, **the pump performance curve does not change**. The energy equation is

$$H_{\text{required}} = h_{\text{pump,u}} = \frac{P_2 - P_1}{\rho g} + \frac{\alpha_2 V_2^{\,2} - \alpha_1 V_1^{\,2}}{2g} + \left(z_2 - z_1\right) + \cancel{h_{\text{turbine}}} + h_{L,\text{total}} \qquad (1)$$

Since the only thing that changes is the elevation difference, Eq. 1 shows that H_{required} shifts up as $(z_2 - z_1)$ increases. Thus, **the system curve rises linearly with elevation increase**. A plot of H versus $\dot{V}$ is plotted, and the new operating point is labeled. Because of the upward shift of the system curve, **the operating point moves to a lower value of volume flow rate**.



Discussion The shift of operating point to lower $\dot{V}$ agrees with our physical intuition. Namely, as we raise the elevation of the outlet, the pump has to do more work to overcome gravity, and we expect the flow rate to decrease accordingly.

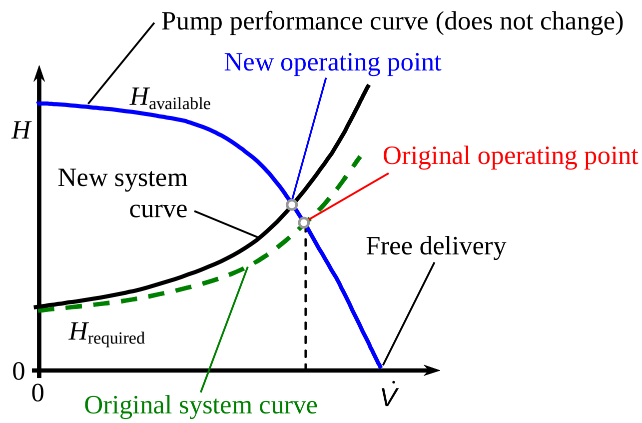
14-22

Solution We are to discuss what would happen to the pump performance curve, the system curve, and the operating point if a valve changes from 100% to 50% open.

Analysis The pump is the same pump regardless of the locations of the inlet and outlet tanks' free surfaces; thus, **the pump performance curve does not change**. The energy equation is

$$H_{\text{required}} = h_{\text{pump,u}} = \frac{P_2 - P_1}{\rho g} + \frac{\alpha_2 V_2^{\,2} - \alpha_1 V_1^{\,2}}{2g} + \left(z_2 - z_1\right) + \cancel{h_{\text{turbine}}} + h_{L,\text{total}} \qquad (1)$$

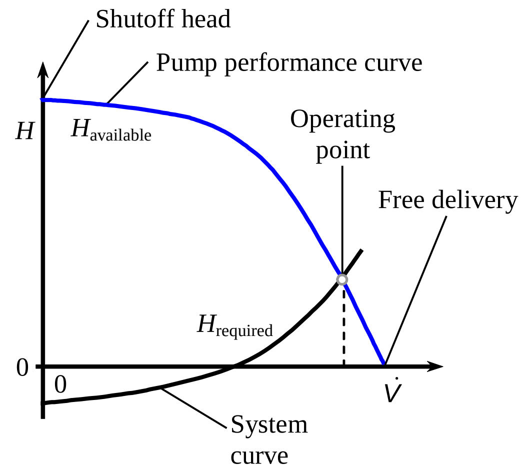
Since both free surfaces are open to the atmosphere, the pressure term vanishes. Since both V_1 and V_2 are negligibly small at the free surface (the tanks are large), the second term on the right also vanishes. The elevation difference $(z_2 - z_1)$ does not change, and so the only term in Eq. 1 that is changed by closing the valve is the irreversible head loss term. We know that the minor loss associated with a valve increases significantly as the valve is closed. Thus, **the system curve** (the curve of H_{required} versus $\dot{V}$) **increases more rapidly with volume flow rate (has a larger slope) when the valve is partially closed**. A sketch of H versus $\dot{V}$ is plotted, and the new operating point is labeled. Because of the higher system curve, ***the operating point moves to a lower value of volume flow rate***, as indicated on the figure. I.e., **the volume flow rate decreases**.



Discussion The shift of operating point to lower $\dot{V}$ agrees with our physical intuition. Namely, as we close the valve somewhat, the pump has to do more work to overcome the losses, and we expect the flow rate to decrease accordingly.

14-23

Solution We are to create a qualitative plot of pump net head versus pump capacity.



Analysis The result is shown in the figure, and the requested items are labeled. Also labeled are the available net head, corresponding to the pump performance curve, and the required net head, corresponding to the system curve. The intersection of these two curves is the operating point of the pump. Note that since the elevation of the outlet is lower than that of the free surface of the inlet tank, the required net head must be *negative* at zero flow rate conditions, as sketched, implying that the pump holds back the natural tendency of the water to flow from higher to lower elevation. Only at higher flow rates does the system curve rise to positive values of $H_{required}$.

Discussion A real pump cannot produce negative net head at zero volume flow rate unless its blades are spun in the opposite direction than that for which they are designed.

14-24

Solution We are to estimate the volume flow rate through a piping system.

Assumptions **1** Since the reservoir is large, the flow is nearly steady. **2** The water is incompressible. **3** The water is at room temperature. **4** The flow in the pipe is fully developed and turbulent, with $\alpha = 1.05$.

Properties The density and viscosity of water at $T = 20^{\circ}$C are 998.0 kg/m^3 and 1.002×10^{-3} kg/m·s respectively.

Analysis By definition, at free delivery conditions, the net head across the pump is zero. Thus, there is no loss or gain of pressure across the pump, and we can essentially ignore it in the calculations here. We apply the head form of the steady energy equation from location 1 to location 2,

$$H_{\text{required}} = h_{\text{pump}} = 0 = \cancel{\frac{P_2 - P_1}{\rho g}} + \frac{\alpha_2 V_2^2 - \cancel{V_1^2}}{2g} + (z_2 - z_1) + \cancel{h_{\text{turbine}}} + h_{L,\text{total}} \tag{1}$$

where the pressure term vanishes since the free surface at location 1 and at the exit (location 2) are both open to the atmosphere. The inlet velocity term disappears since V_1 is negligibly small at the free surface. Thus, Eq. 1 reduces to a balance between supplied potential energy head ($z_1 - z_2$), kinetic energy head at the exit $\alpha_2 V_2^2/2g$, and irreversible head losses,

$$(z_1 - z_2) = \frac{\alpha_2 V_2^2}{2g} + h_{L,\text{total}} \tag{2}$$

The total irreversible head loss in Eq. 2 consists of both major and minor losses. We split the minor losses into those associated with the mean velocity V through the pipe, and the minor loss associated with the contraction, based on exit velocity V_2,

$$(z_1 - z_2) = \frac{\alpha_2 V_2^2}{2g} + \frac{V^2}{2g}\left(f\frac{L}{D} + \sum_{\text{pipe}} K_L \right) + \frac{V_2^2}{2g} K_{L,\text{contraction}} \tag{3}$$

where $\sum_{\text{pipe}} K_L$ = 0.50 + 2(2.4) + 3(0.90) = 8.0, and $K_{L,\text{contraction}}$ = 0.15.

By conservation of mass,

$$VA = V_2 A_2 \quad \rightarrow \quad V_2 = V\frac{A}{A_2} = V\left(\frac{D}{D_2}\right)^2 \tag{4}$$

Substitution of Eq. 4 into Eq. 3 yields

$$(z_1 - z_2) = \frac{V^2}{2g}\left(f\frac{L}{D} + \sum_{\text{pipe}} K_L + \left(\frac{D}{D_2}\right)^4 \left(\alpha_2 + K_{L,\text{contraction}}\right) \right) \tag{5}$$

Equation 5 is an implicit equation for V since the Darcy friction factor is a function of Reynolds number Re = $\rho VD/\mu$, as obtained from either the Moody chart or the Colebrook equation. The solution can be obtained by an iterative method, or through use of a mathematical equation solver like EES. The result is V = 1.82548 m/s, or to three significant digits, V = 1.83 m/s, from which the volume flow rate is

$$\dot{V} = V\frac{\pi D^2}{4} = (1.82548 \text{ m/s})\frac{\pi (0.020 \text{ m})^2}{4} = 5.73491\times 10^{-4} \text{ m}^3/\text{s} \cong 5.73\times 10^{-4} \text{ m}^3/\text{s} \tag{6}$$

In more common units, $\dot{V}$ = **34.4 Lpm** (liters per minute). The Reynolds number is 3.64×10^4.

Discussion Since there is no net head across the pump at free delivery conditions, the pump could be removed (inlet and outlet pipes connected together without the pump), and the flow rate would be the same. Another way to think about this is that the pump's efficiency is zero at the free delivery operating point, so it is doing no useful work.

14-25

Solution We are to calculate the volume flow rate through a piping system in which the pipe is rough.

Assumptions **1** Since the reservoir is large, the flow is nearly steady. **2** The water is incompressible. **3** The water is at room temperature. **4** The flow in the pipe is fully developed and turbulent, with α = 1.05.

Properties The density and viscosity of water at T = 20°C are 998.0 kg/m^3 and 1.002 × 10^{-3} kg/m·s respectively.

Analysis The relative pipe roughness is ε/D = (0.012 cm)/(2.0 cm) = 0.006 (very rough, as seen on the Moody chart). The calculations are identical to that of the previous problem, except for the pipe roughness. The result is V = 1.6705 m/s, or to three significant digits, V = 1.67 m/s, from which the volume flow rate is 5.25 × 10^{-4} m^3/s, or $\dot{V}$ = **31.5 Lpm**. The Reynolds number is 3.33 × 10^4. **The volume flow rate is lower by about 8.4% compared to the smooth pipe case**. This agrees with our intuition, since pipe roughness leads to more pressure drop at a given flow rate.

Discussion If the calculations of the previous problem are done on a computer, it is trivial to change ε for the present calculations.

14-26

Solution For a given pump and piping system, we are to calculate the volume flow rate and compare with that calculated for Problem 14-24.

Assumptions **1** Since the reservoir is large, the flow is nearly steady. **2** The water is incompressible. **3** The water is at room temperature. **4** The flow in the pipe is fully developed and turbulent, with α = 1.05.

Properties The density and viscosity of water at T = 20°C are 998.0 kg/m^3 and 1.002 × 10^{-3} kg/m·s respectively.

Analysis The calculations are identical to those of the previous problem except that the pump's net head is not zero, but instead is given in the problem statement. At the operating point, we match $H_{\text{available}}$ to H_{required}, yielding

$$H_{\text{available}} = H_{\text{required}} \quad \rightarrow \quad H_0 - a\dot{V}^2 = \frac{V^2}{2g}\left(f\frac{L}{D} + \sum_{\text{pipe}} K_L + \left(\frac{D}{D_2}\right)^4 \left(\alpha_2 + K_{L,\text{contraction}}\right)\right) - \left(z_1 - z_2\right) \qquad (1)$$

We re-write the second term on the left side of Eq. 1 in terms of average pipe velocity V instead of volume flow rate, since $\dot{V} = V\pi D^2/4$, and solve for V,

$$V = \sqrt{\frac{H_0 + \left(z_1 - z_2\right)}{\frac{1}{2g}\left(f\frac{L}{D} + \sum_{\text{pipe}} K_L + \left(\frac{D}{D_2}\right)^4 \left(\alpha_2 + K_{L,\text{contraction}}\right)\right) + a\frac{\pi^2 D^4}{16}}} \qquad (2)$$

Equation 2 is an implicit equation for V since the Darcy friction factor is a function of Reynolds number Re = $\rho VD/\mu$, as obtained from either the Moody chart or the Colebrook equation. The solution can be obtained by an iterative method, or through use of a mathematical equation solver like EES. The result is V = 2.9293 m/s, from which the volume flow rate is

$$\dot{V} = V\frac{\pi D^2}{4} = \left(2.9293 \text{ m/s}\right)\frac{\pi\left(0.020 \text{ m}\right)^2}{4} = 9.203 \times 10^{-4}\ \frac{\text{m}^3}{\text{s}} \qquad (3)$$

In more common units, $\dot{V}$ = **55.2 Lpm, an increase of about 60% compared to the flow rate with the pump operating at free delivery**. This agrees with our expectations – adding a pump in the line produces a higher flow rate.

Discussion Although there was a pump in the previous problem as well, it was operating at free delivery conditions, implying that it was not contributing anything to the flow – that pump could be removed from the system with no change in flow rate. Here, however, the net head across the pump is about 6.82 m, implying that it is contributing useful head to the flow (in addition to the gravity head already present).

14-27

Solution We are to calculate the volume flow rate through a piping system in which the pipe is rough.

Assumptions **1** Since the reservoir is large, the flow is nearly steady. **2** The water is incompressible. **3** The water is at room temperature. **4** The flow in the pipe is fully developed and turbulent, with $\alpha = 1.05$.

Properties The density and viscosity of water at $T = 20^{\circ}\text{C}$ are 998.0 kg/m^3 and 1.002×10^{-3} kg/m·s respectively.

Analysis The relative pipe roughness is ε/D = (0.012 cm)/(2.0 cm) = 0.006 (very rough, as seen on the Moody chart). The calculations are identical to that of the previous problem, except for the pipe roughness. The result is V = 2.786 m/s, from which the volume flow rate is 8.753×10^{-4} m^3/s, or $\dot{V}$ = **52.5 Lpm**. The Reynolds number is 5.55×10^4. **The volume flow rate is lower by about 4.9% compared to the smooth pipe case**. This agrees with our intuition, since pipe roughness leads to more pressure drop at a given flow rate.

Discussion If the calculations of the previous problem are done on a computer, it is trivial to change ε for the present calculations.

14-28

Solution We are to calculate pump efficiency and estimate the BEP conditions.

Properties The density of water at 20°C is 998.0 kg/m^3.

Analysis (*a*) Pump efficiency is

Pump efficiency:
$$\eta_{\text{pump}} = \frac{\rho g \dot{V} H}{bhp} \qquad (1)$$

We show the second row of data (at $\dot{V}$ = 6.0 Lpm) as an example – the rest are calculated in a spreadsheet for convenience,

TABLE 1
Pump performance data for water at 20°C.

$\dot{V}$ (Lpm)	***H*** (m)	***bhp*** (W)	η_{pump} (%)
0.0	47.5	133	**0.0**
6.0	46.2	142	**31.9**
12.0	42.5	153	**54.4**
18.0	36.2	164	**64.8**
24.0	26.2	172	**59.7**
30.0	15.0	174	**42.2**
36.0	0.0	174	**0.0**

$$\eta_{\text{pump}} = \frac{\left(998.0\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)\left(6.0\ \text{L/min}\right)\left(46.2\ \text{m}\right)}{142\ \text{W}}\left(\frac{1\ \text{m}^3}{1000\ \text{L}}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right)\times\left(\frac{\text{N}\cdot\text{s}^2}{\text{kg}\cdot\text{m}}\right)\left(\frac{\text{W}\cdot\text{s}}{\text{N}\cdot\text{m}}\right) = 0.319 = 31.9\%$$

The results for all rows are shown in Table 1.

(*b*) The best efficiency point (BEP) occurs at approximately the fourth row of data: $\dot{V}^*$ = **18.0 Lpm,** H^* = **36.2 m of head,** bhp^* = **164. W**, and η_{pump}^* **= 64.8%**.

Discussion A more precise BEP could be obtained by curve-fitting the data, as in the next problem.

14-29

Solution We are to generate least-squares polynomial curve fits of a pump's performance curves, plot the curves, and calculate the BEP.

Properties The density of water at 20°C is 998.0 kg/m³.

Analysis The efficiencies for each data point in Table P14-31 were calculated in the previous problem. We use Regression analysis to generate the least-squares fits. The equation and coefficients for H are

$$H = H_0 - a\dot{V}^2 \qquad H_0 = 47.6643 \text{ m} \qquad a = 0.0366453 \text{ m/Lpm}^2$$

Or, to 3 significant digits, $H_0 = \mathbf{47.7\ m}$ $\quad a = \mathbf{0.0366\ m/Lpm^2}$

The equation and coefficients for *bhp* are

$$bhp = bhp_0 + a_1\dot{V} + a_2\dot{V}^2 \qquad bhp_0 = 131.\text{ W}$$

$$a_1 = \mathbf{2.37\ W/Lpm} \qquad a_2 = \mathbf{-0.0317\ W/Lpm^2}$$

The equation and coefficients for η_{pump} are

$$\eta_{\text{pump}} = \eta_{\text{pump},0} + a_1\dot{V} + a_2\dot{V}^2 + a_3\dot{V}^3 \qquad \eta_{\text{pump},0} = 0.152\%$$

$$a_1 = \mathbf{5.87\ \%/Lpm} \quad a_2 = \mathbf{-0.0905\ \%/Lpm^2} \quad a_3 = \mathbf{-0.00201\ \%/Lpm^3}$$

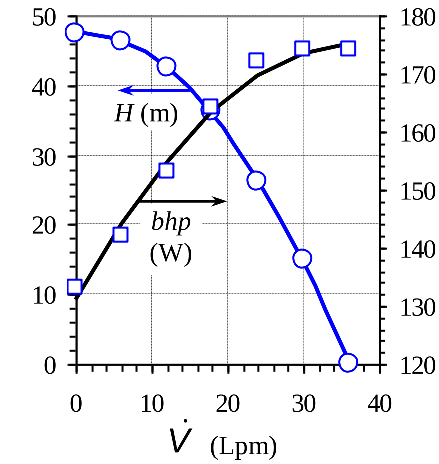
The tabulated data are plotted in Fig. 1 as symbols only. The fitted data are plotted on the same plots as lines only. The agreement is excellent.

The best efficiency point is obtained by differentiating the curve-fit expression for η_{pump} with respect to volume flow rate, and setting the derivative to zero (solving the resulting quadratic equation for $\dot{V}*$),

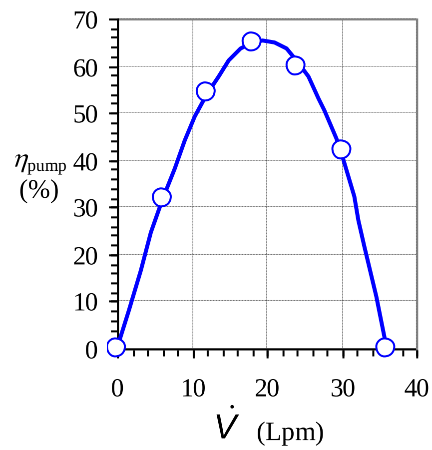
$$\frac{d\eta_{\text{pump}}}{d\dot{V}} = a_1 + 2a_2\dot{V} + 3a_3\dot{V}^2 = 0 \qquad \rightarrow \qquad \dot{V}* = 19.6 \text{ Lpm}$$

At this volume flow rate, the curve-fitted expressions for H, *bhp*, and η_{pump} yield the operating conditions at the best efficiency point (to three digits each):

$$\dot{V}* = \mathbf{19.6\ Lpm}, \quad H* = \mathbf{33.6\ m}, \quad bhp* = \mathbf{165\ W}, \quad \eta* = \mathbf{65.3\%}$$



(a)



(b)

FIGURE 1
Pump performance curves: (a) H and *bhp* versus $\dot{V}$, and (b) η_{pump} versus $\dot{V}$.

Discussion This BEP is more precise than that of the previous problem because of the curve fit. The other root of the quadratic is negative – obviously not the correct choice.

14-30

Solution For a given pump and system requirement, we are to estimate the operating point.

Assumptions **1** The flow is steady. **2** The water is at 20°C and is incompressible.

Analysis The operating point is the volume flow rate at which $H_{\text{required}} = H_{\text{available}}$. We set the given expression for H_{required} to the curve fit expression of the previous problem, $H_{\text{available}} = H_0 - a\dot{V}^2$, and obtain

Operating point:
$$\dot{V} = \sqrt{\frac{H_0 - (z_2 - z_1)}{a + b}} = \sqrt{\frac{47.6643\text{ m} - 21.7\text{ m}}{(0.0366453 + 0.0185)\text{ m/Lpm}^2}} = 21.6987\text{ Lpm} \cong \mathbf{21.7\,Lpm}$$

At this volume flow rate, we use the curve fit to estimate the head,

Operating point:
$$H_0 = H_0 \text{ - } a\dot{V} = 47.6643\text{ m} - (0.0366453\text{ m/Lpm}^2)(21.6987\text{ Lpm})^2 = 30.4105\text{ m} \cong \mathbf{30.4\,m}$$

Discussion At this operating point, the flow rate is higher than that at the BEP.

14-31E

Solution We are to calculate pump efficiency and estimate the BEP conditions.

Properties The density of water at 77°F is 62.24 lbm/ft^3.

Analysis (*a*) Pump efficiency is

Pump efficiency:
$$\eta_{\text{pump}} = \frac{\rho g \dot{V} H}{bhp} \qquad (1)$$

We show the second row of data (at $\dot{V}$ = 4.0 gpm) as an example – the rest are calculated in a spreadsheet for convenience,

Pump performance data for water at 77°F.

$\dot{V}$ (gpm)	*H* (ft)	***bhp*** (hp)	η_{pump} (%)
0.0	19.0	0.06	**0.0**
4.0	18.5	0.064	**29.2**
8.0	17.0	0.069	**49.7**
12.0	14.5	0.074	**59.3**
16.0	10.5	0.079	**53.6**
20.0	6.0	0.08	**37.8**
24.0	0	0.078	**0.0**

$$\eta_{\text{pump}} = \frac{\left(62.24\text{ lbm/ft}^3\right)\left(32.2\frac{\text{ft}}{\text{s}^2}\right)\left(4.0\frac{\text{gal}}{\text{min}}\right)(18.5\text{ ft})}{0.064\text{ hp}}\left(\frac{0.1337\text{ ft}^3}{\text{gal}}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right)\times\left(\frac{\text{lbf}\cdot\text{s}^2}{32.2\text{ lbm}\cdot\text{ft}}\right)\left(\frac{\text{hp}\cdot\text{s}}{550\text{ ft}\cdot\text{lbf}}\right) = 0.292$$

or 29.2%. The results for all rows are shown in the table.

(*b*) The best efficiency point (BEP) occurs at approximately the fourth row of data: $\dot{V}^* =$ **12.0 gpm, *H** = 14.5 ft of head, *bhp** = 0.074 hp, and η_{pump}* = 59.3%**.

Discussion A more precise BEP could be obtained by curve-fitting the data, as in Problem 14-29.

14-32E

Solution We are to convert the pump performance data to metric units and calculate pump efficiency.

Properties The density of water at T = 20°C is 998.0 kg/m³.

Analysis The conversions are straightforward, and the results are shown in the table. A sample calculation of the pump efficiency for the second row of data is shown below:

Pump performance data for water at 77°F.

$\dot{V}$ (Lpm)	***H*** (m)	***bhp*** (W)	η_{pump} (%)
0.0	5.79	44.7	**0.0**
15.1	5.64	47.7	**29.2**
30.3	5.18	51.5	**49.7**
45.4	4.42	55.2	**59.3**
60.6	3.20	58.9	**53.6**
75.7	1.83	59.7	**37.8**
90.9	0.00	58.2	**0.0**

$$\eta_{\text{pump}} = \frac{\left(998.0\text{ kg/m}^3\right)\left(9.81\text{ m/s}^2\right)\left(15.1\text{ L/min}\right)\left(5.64\text{ m}\right)}{47.7\text{ W}}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right)\times\left(\frac{\text{N}\cdot\text{s}^2}{\text{kg}\cdot\text{m}}\right)\left(\frac{\text{W}\cdot\text{s}}{\text{N}\cdot\text{m}}\right) = 0.292 = \mathbf{29.2\%}$$

The pump efficiency data are identical to those of the previous problem, as they must be, regardless of the system of units.

Discussion If the calculations of the previous problem are done on a computer, it is trivial to convert to metric units in the present calculations.

14-33E

Solution We are to generate least-squares polynomial curve fits of a pump's performance curves, plot the curves, and calculate the BEP.

Properties The density of water at 77°F is 62.24 lbm/ft³.

Analysis The efficiencies for each data point in Table P14-27 are calculated in Problem 14-27. We use regression analysis to generate the least-squares fits. The equation and coefficients for H are

$$H = H_0 - a\dot{V}^2 \qquad H_0 = 19.0774 \text{ ft} \qquad a = 0.032996 \text{ ft/gpm}^2$$

Or, to 3 digits of precision, $H_0 = \mathbf{19.1\ ft}$ $\quad a = \mathbf{0.0330\ ft/gpm^2}$

The equation and coefficients for *bhp* are

$$bhp = bhp_0 + a_1\dot{V} + a_2\dot{V}^2 \qquad bhp_0 = 0.0587 \text{ hp}$$

$$a_1 = \mathbf{0.00175\ hp/gpm} \qquad a_2 = \mathbf{-3.72\times10^{-5}\ hp/gpm^2}$$

The equation and coefficients for η_{pump} are

$$\eta_{\text{pump}} = \eta_{\text{pump},0} + a_1\dot{V} + a_2\dot{V}^2 + a_3\dot{V}^3 \qquad \eta_{\text{pump},0} = 0.0523\%$$

$$a_1 = \mathbf{8.21\ \%/gpm} \qquad a_2 = \mathbf{-0.210\ \%/gpm^2} \qquad a_3 = \mathbf{-0.00546\ \%/gpm^3}$$

The tabulated data are plotted in Fig. 1 as symbols only. The fitted data are plotted on the same plots as lines only. The agreement is excellent.

The best efficiency point is obtained by differentiating the curve-fit expression for η_{pump} with respect to volume flow rate, and setting the derivative to zero (solving the resulting quadratic equation for $\dot{V}*$),

$$\frac{d\eta_{\text{pump}}}{d\dot{V}} = a_1 + 2a_2\dot{V} + 3a_3\dot{V}^2 = 0 \qquad \rightarrow \qquad \dot{V}* = 12.966 \text{ GPM} \approx 13.0 \text{ gpm}$$

At this volume flow rate, the curve-fitted expressions for H, *bhp*, and η_{pump} yield the operating conditions at the best efficiency point (to three digits each):

$$\dot{V}* = \mathbf{13.0\ gpm}, \qquad H* = \mathbf{13.5\ ft}, \qquad bhp* = \mathbf{0.0752\ hp}, \qquad \eta* = \mathbf{59.2\%}$$

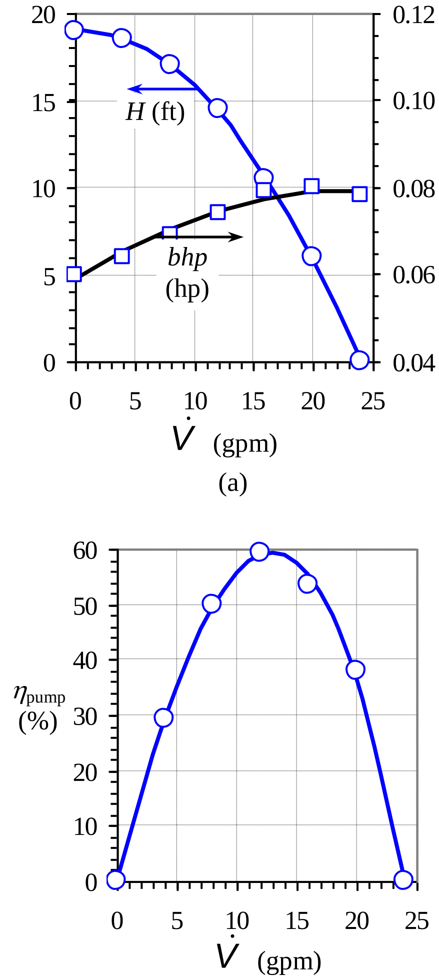


FIGURE 1
Pump performance curves: (a) H and *bhp* versus $\dot{V}$, and (b) η_{pump} versus $\dot{V}$.

Discussion This BEP is more precise than that of Problem 14-27 because of the curve fit. The other root of the quadratic is negative – obviously not the correct choice.

14-34E

Solution For a given pump and system requirement, we are to estimate the operating point.

Assumptions **1** The flow is steady. **2** The water is at 77°F and is incompressible.

Analysis The operating point is the volume flow rate at which $H_{\text{required}} = H_{\text{available}}$. We set the given expression for H_{required} to the curve fit expression of Problem 14-26, $H_{\text{available}} = H_0 - a\dot{V}^2$, and obtain

Operating point:
$$\dot{V} = \sqrt{\frac{H_0 - (z_2 - z_1)}{a+b}} = \sqrt{\frac{19.0774\text{ ft} - 11.3\text{ ft}}{(0.032996 + 0.00986)\text{ ft/gpm}^2}} = \mathbf{13.5\ gpm}$$

At this volume flow rate, the net head of the pump is **13.1 ft**.

Discussion At this operating point, the flow rate is lower than that at the BEP.

14-35

Solution We are to perform a regression analysis to estimate the shutoff head and free delivery of a pump, and then we are to determine if this pump is adequate for the system requirements.

Assumptions **1** The water is incompressible. **2** The water is at room temperature.

Properties The density of water at T = 20°C is 998.0 kg/m³.

Analysis (*a*) We perform a regression analysis, and obtain $\boldsymbol{H_0}$ **= 23.9 m** and $\boldsymbol{a}$ **= 0.00642 m/Lpm²**. The curve fit is reasonable, as seen in Fig. 1. **The shutoff head is estimated as 23.9 m of water column**. At the pump's free delivery, the net head is zero. Setting $H_{\text{available}}$ to zero in Eq. 1 gives

Free delivery:
$$\dot{V}_{\max}^{\ 2} = \frac{H_0}{a} \quad \rightarrow \quad \dot{V}_{\max} = \sqrt{\frac{H_0}{a}} = \sqrt{\frac{23.9\text{ m}}{0.00642\text{ m/(Lpm)}^2}} = 61.0\text{ Lpm}$$

The free delivery is estimated as **61.0 Lpm**.

(*b*) At the required operating conditions, $\dot{V}$ = 57.0 Lpm, and the net head is converted to meters of water column for analysis,

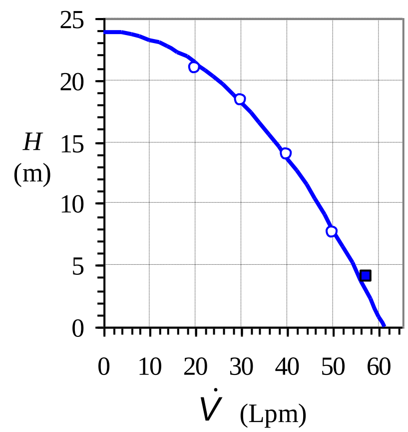


FIGURE 1
Tabulated data (circles) and curve-fitted data (line) for $H_{\text{available}}$ versus $\dot{V}$ for the given pump. The filled, square data point is the required operating point.

Required operating head:
$$H_{\text{required}} = \frac{(\Delta P)_{\text{required}}}{\rho g} = \frac{5.8\text{ psi}}{(998.\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{6{,}894.8\text{ N/m}^2}{\text{psi}}\right)\left(\frac{\text{kg}\cdot\text{m}}{\text{s}^2\cdot\text{N}}\right) = 4.08\text{ m}$$

As seen in Fig. 1, this operating point lies *above* the pump performance curve. Thus, **this pump is not quite adequate for the job at hand**.

Discussion The operating point is also very close to the pump's free delivery, and therefore the pump efficiency would be low even if it *could* put out the required head.

14-36

Solution We are to calculate the operating point of a given pipe/pump system.

Assumptions **1** The water is incompressible. **2** The flow is steady since the reservoirs are large. **3** The water is at room temperature.

Properties The density and viscosity of water at T = 20°C are 998.0 kg/m^3 and 1.002 × 10^{-3} kg/m·s respectively, but these properties are not actually needed in the analysis.

Analysis The operating point is obtained by matching the pump's performance curve to the system curve,

Operating point:
$$H_{\text{available}} = H_0 - a\dot{V}^2 = H_{\text{required}} = (z_2 - z_1) + b\dot{V}^2$$

from which we solve for the volume flow rate (capacity) at the operating point,

$$\dot{V}_{\text{operating}} = \sqrt{\frac{H_0 - (z_2 - z_1)}{a+b}} = \sqrt{\frac{7.46\text{ m} - 3.52\text{ m}}{(0.0453 + 0.0261)\text{ m/Lpm}^2}} = \mathbf{7.43\ Lpm}$$

and for the net pump head at the operating point,

$$H_{\text{operating}} = \frac{H_0 b + a(z_2 - z_1)}{a+b} = \frac{(7.46\text{ m})(0.0261\text{ m}) + (0.0453\text{ m})(3.52\text{ m})}{(0.0453\text{ m}) + (0.0261\text{ m})} = \mathbf{4.96\ m}$$

Discussion The water properties ρ and μ are not needed because the system curve (H_{required} versus $\dot{V}$) is provided here.

14-37

Solution We are to calculate the operating point of a given pipe/pump system.

Assumptions **1** The water is incompressible. **2** The flow is steady since the reservoirs are large. **3** The water is at room temperature.

Properties The density and viscosity of water at T = 20°C are 998.0 kg/m^3 and 1.002 × 10^{-3} kg/m·s respectively, but these properties are not actually needed in the analysis.

Analysis The operating point is obtained by matching the pump's performance curve to the system curve,

Operating point:
$$H_{\text{available}} = H_0 - a\dot{V}^2 = H_{\text{required}} = (z_2 - z_1) + b\dot{V}^2$$

from which we solve for the volume flow rate (capacity) at the operating point,

$$\dot{V}_{\text{operating}} = \sqrt{\frac{H_0 - (z_2 - z_1)}{a+b}} = \sqrt{\frac{8.13\text{ m} - 3.52\text{ m}}{(0.0297 + 0.0261)\text{ m/Lpm}^2}} = \mathbf{9.09\ Lpm}$$

and for the net pump head at the operating point,

$$H_{\text{operating}} = \frac{H_0 b + a(z_2 - z_1)}{a+b} = \frac{(8.13\text{ m})(0.0261\text{ m}) + (0.0297\text{ m})(3.52\text{ m})}{(0.0297\text{ m}) + (0.0261\text{ m})} = \mathbf{5.68\ m}$$

This represents **an improvement in flow rate of about 22%, and YES, it meets the requirement**.

Discussion The water properties ρ and μ are not needed because the system curve (H_{required} versus $\dot{V}$) is provided here.

14-38E

Solution We are to find the units of coefficient a, write $\dot{V}_{max}$ in terms of H_0 and a, and calculate the operating point of the pump.

Assumptions **1** The flow is steady. **2** The water is incompressible.

Analysis (a) Solving the given expression for a gives

Coefficient a:
$$a = \frac{H_0 - H_{\text{available}}}{\dot{V}^2} \quad \rightarrow \quad \boxed{\text{units of } a = \frac{\text{ft}}{\text{gpm}^2}} \qquad (1)$$

(b) At the pump's free delivery, the net head is zero. Setting $H_{\text{available}}$ to zero in the given expression gives

Free delivery:
$$\dot{V}_{max}{}^2 = \frac{H_0}{a} \quad \rightarrow \quad \boxed{\dot{V}_{max} = \sqrt{\frac{H_0}{a}}} \qquad (2)$$

(c) The operating point is obtained by matching the pump's performance curve to the system curve. Equating these gives

$$H_{\text{available}} = H_0 - a\dot{V}^2 = H_{\text{required}} = \left(z_2 - z_1\right) + b\dot{V}^2 \qquad (3)$$

After some algebra, Eq. 3 reduces to

Operating point capacity:
$$\boxed{\dot{V}_{\text{operating}} = \sqrt{\frac{H_0 - \left(z_2 - z_1\right)}{a+b}}} \qquad (4)$$

and the net pump head at the operating point is obtained by plugging Eq. 4 into the given expression,

Operating point pump head:
$$\boxed{H_{\text{operating}} = \frac{H_0 b + a\left(z_2 - z_1\right)}{a+b}} \qquad (5)$$

Discussion Equation 4 reveals that H_0 must be greater than elevation difference ($z_2 - z_1$) in order to have a valid operating point. This agrees with our intuition, since the pump must be able to overcome the gravitational head between the tanks.

14-39E

Solution For a given pump and system, we are to calculate the capacity.

Assumptions **1** The water is incompressible. **2** The flow is nearly steady since the reservoirs are large. **3** The water is at room temperature.

Properties The kinematic viscosity of water at T = 68°F is 1.055 × 10^{-5} ft^2/s.

Analysis We apply the energy equation in head form between the inlet reservoir's free surface (1) and the outlet reservoir's free surface (2),

$$H_{\text{required}} = h_{\text{pump,u}} = \cancel{\frac{P_2 - P_1}{\rho g}} + \frac{\alpha_2 \cancel{V_2^2} - \alpha_1 \cancel{V_1^2}}{2g} + \left(z_2 - z_1\right) + \cancel{h_{\text{turbine}}} + h_{L,\text{total}} \qquad (1)$$

Since both free surfaces are at atmospheric pressure, $P_1 = P_2 = P_{\text{atm}}$, and the first term on the right side of Eq. 1 vanishes. Furthermore, since there is no flow, $V_1 = V_2 = 0$, and the second term also vanishes. There is no turbine in the control volume, so the second-to-last term is zero. Finally, the irreversible head losses are composed of both major and minor losses, but the pipe diameter is constant throughout. Equation 1 therefore reduces to

$$H_{\text{required}} = \left(z_2 - z_1\right) + h_{L,\text{total}} = \left(z_2 - z_1\right) + \left(f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g} \qquad (2)$$

The dimensionless roughness factor is ε/D = 0.0011/1.20 = 9.17 × 10^{-4}, and the sum of all the minor loss coefficients is

$$\sum K_L = 0.5 + 2.0 + 6.8 + \left(3 \times 0.34\right) + 1.05 = 11.37$$

The pump/piping system operates at conditions where the available pump head equals the required system head. Thus, we equate the given expression and Eq. 2 to find the operating point,

$$H_{\text{available}} = H_{\text{required}} \quad \rightarrow \quad H_0 - a\frac{\pi^2 D^4}{16}V^2 = \left(z_2 - z_1\right) + \left(f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g} \qquad (3)$$

where we have written the volume flow rate in terms of average velocity through the pipe,

Volume flow rate in terms of average velocity: $$\dot{V} = V\frac{\pi D^2}{4} \qquad (4)$$

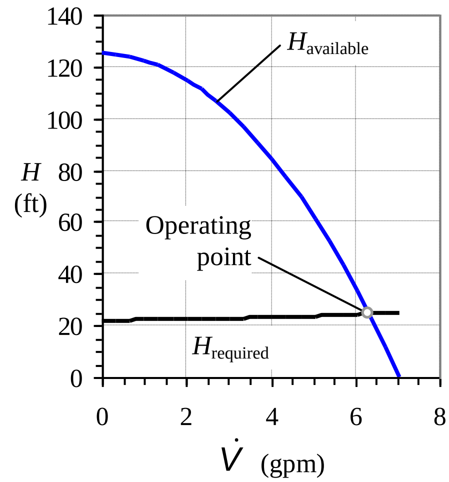
Equation 3 is an implicit equation for V since the Darcy friction factor f is a function of Reynolds number Re = $\rho VD/\mu$ = VD/ν, as obtained from either the Moody chart or the Colebrook equation. The solution can be obtained by an iterative method, or through use of a mathematical equation solver like EES. The result is V = 1.80 ft/s, from which the volume flow rate is $\dot{V}$ = **6.34 gpm**. The Reynolds number is 1.67 × 10^4.

Discussion We verify our results by comparing $H_{\text{available}}$ (given) and H_{required} (Eq. 2) at this flow rate: $H_{\text{available}}$ = 24.4 ft and H_{required} = 24.4 ft.

14-40E

Solution We are to plot $H_{required}$ and $H_{available}$ versus $\dot{V}$, and indicate the operating point.

Analysis We use the equations of the previous problem, with the same constants and parameters, to generate the plot shown. The operating point is the location where the two curves intersect. The values of H and $\dot{V}$ at the operating point match those of the previous problem, as they should.



Discussion A plot like this, in fact, is an alternate method of obtaining the operating point. In this case, the curve of $H_{required}$ is fairly flat, indicating that the majority of the required pump head is attributed to elevation change, while a small fraction is attributed to major and minor head losses through the piping system.

14-41E

Solution We are to re-calculate volume flow rate for a piping system with a much longer pipe, and we are to compare with the previous results.

Analysis All assumptions, properties, dimensions, and parameters are identical to those of the previous problem, except that total pipe length L is longer. We repeat the calculations and find that V = 1.63 ft/s, from which the volume flow rate is $\dot{V}$ = **5.75 gpm**, and the net head of the pump is 42.3 ft. The Reynolds number for the flow in the pipe is 1.55×10^4. **The volume flow rate has decreased by about 9.3%**.

Discussion The decrease in volume flow rate is smaller than we may have suspected. This is because the majority of the pump work goes into raising the elevation of the water. In addition, as seen in the plot from the previous problem, the pump performance curve is quite steep near these flow rates – a significant change in required net head leads to a much less significant change in volume flow rate.

14-42E

Solution We are to perform a regression analysis to translate tabulated pump performance data into an analytical expression, and then use this expression to predict the volume flow rate through a piping system.

Assumptions **1** The water is incompressible. **2** The flow is nearly steady since the reservoirs are large. **3** The water is at room temperature.

Properties For water at T = 68°F, μ = 6.572 × 10^{-4} lbm/ft·s, and ρ = 62.31 lbm/ft^3, from which ν = 1.055 × 10^{-5} ft^2/s.

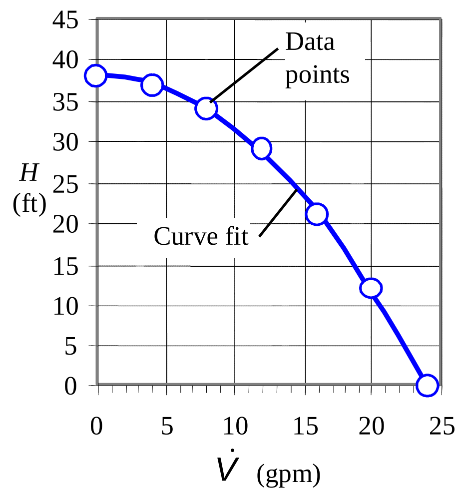


FIGURE 1
Tabulated data (symbols) and curve-fitted data (line) for $H_{available}$ versus $\dot{V}$ for the proposed pump.

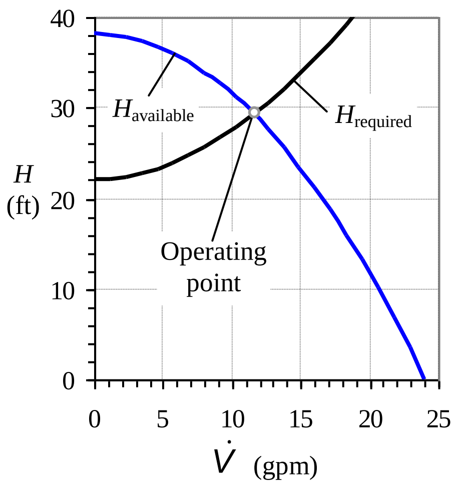


FIGURE 2
$H_{available}$ and $H_{required}$ versus $\dot{V}$ for a piping system with pump; the operating point is also indicated, where the two curves meet.

Analysis (*a*) We perform a regression analysis, and obtain $\boldsymbol{H_0}$ **= 38.15 ft** and $\boldsymbol{a}$ **= 0.06599 ft/gpm^2**. The curve fit is very good, as seen in Fig. 1.

(*b*) We repeat the calculations of Problem 14-37 with the new pump performance coefficients, and find that V = 3.29 ft/s, from which the volume flow rate is $\dot{V}$ = **11.6 gpm**, and the net head of the pump is 29.3 ft. The Reynolds number for the flow in the pipe is 3.05 × 10^4. **The volume flow rate has increased by about 83%**. **Paul is correct** – this pump performs much better, nearly doubling the flow rate.

(*c*) A plot of net head versus volume flow rate is shown in Fig. 2.

Discussion This pump is more appropriate for the piping system at hand.

14-43

Solution For a given pump and system, we are to calculate the capacity.

Assumptions **1** The water is incompressible. **2** The flow is nearly steady since the reservoirs are large. **3** The water is at room temperature.

Properties The density and viscosity of water at T = 20°C are 998.0 kg/m^3 and 1.002 × 10^{-3} kg/m·s respectively.

Analysis We apply the energy equation in head form between the inlet reservoir's free surface (1) and the outlet reservoir's free surface (2),

$$H_{\text{required}} = h_{\text{pump,u}} = \cancel{\frac{P_2 - P_1}{\rho g}} + \frac{\alpha_2 \cancel{V_2^2} - \alpha_1 \cancel{V_1^2}}{2g} + (z_2 - z_1) + \cancel{h_{\text{turbine}}} + h_{L,\text{total}} \tag{1}$$

Since both free surfaces are at atmospheric pressure, $P_1 = P_2 = P_{\text{atm}}$, and the first term on the right side of Eq. 1 vanishes. Furthermore, since there is no flow, $V_1 = V_2 = 0$, and the second term also vanishes. There is no turbine in the control volume, so the second-to-last term is zero. Finally, the irreversible head losses are composed of both major and minor losses, but the pipe diameter is constant throughout. Equation 1 therefore reduces to

$$H_{\text{required}} = (z_2 - z_1) + h_{L,\text{total}} = (z_2 - z_1) + \left(f\frac{L}{D} + \sum K_L \right) \frac{V^2}{2g} \tag{2}$$

The dimensionless roughness factor is

$$\frac{\varepsilon}{D} = \frac{0.25 \text{ mm}}{2.03 \text{ cm}} \left(\frac{1 \text{ cm}}{10 \text{ mm}} \right) = 0.0123$$

The sum of all the minor loss coefficients is

$$\sum K_L = 0.5 + 17.5 + (5 \times 0.92) + 1.05 = 23.65$$

The pump/piping system operates at conditions where the available pump head equals the required system head. Thus, we equate the given expression and Eq. 2 to find the operating point,

$$H_{\text{available}} = H_{\text{required}} \quad \rightarrow \quad H_0 - a\frac{\pi^2 D^4}{16} V^2 = (z_2 - z_1) + \left(f\frac{L}{D} + \sum K_L \right) \frac{V^2}{2g} \tag{3}$$

where we have written the volume flow rate in terms of average velocity through the pipe,

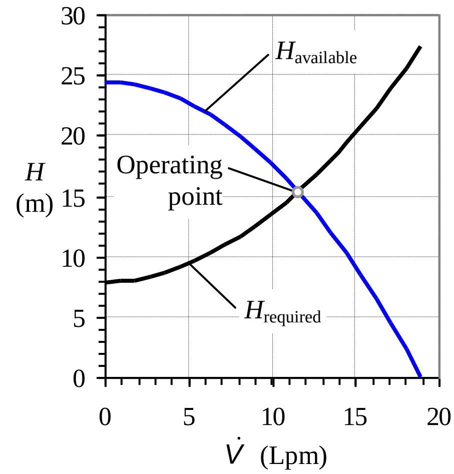
$$\dot{V} = V\frac{\pi D^2}{4}$$

Equation 3 is an implicit equation for V since the Darcy friction factor f is a function of Reynolds number Re = $\rho VD/\mu$, as obtained from either the Moody chart or the Colebrook equation. The solution can be obtained by an iterative method, or through use of a mathematical equation solver like EES. The result is V = 0.59603 ≈ 0.596 m/s, from which the volume flow rate is $\dot{V}$ = **11.6 Lpm**. The Reynolds number is 1.21 × 10^4.

Discussion We verify our results by comparing $H_{\text{available}}$ (given) and H_{required} (Eq. 2) at this flow rate: $H_{\text{available}}$ = 15.3 m and H_{required} = 15.3 m.

14-44

Solution We are to plot $H_{required}$ and $H_{available}$ versus $\dot{V}$, and indicate the operating point.



Analysis We use the equations of the previous problem, with the same constants and parameters, to generate the plot shown. The operating point is the location where the two curves intersect. The values of H and $\dot{V}$ at the operating point match those of the previous problem, as they should.

Discussion A plot like this, in fact, is an alternate method of obtaining the operating point.

14-45

Solution We are to re-calculate volume flow rate for a piping system with a smaller elevation difference, and we are to compare with the previous results.

Analysis All assumptions, properties, dimensions, and parameters are identical to those of the previous problem, except that the elevation difference between reservoir surfaces ($z_2 - z_1$) is smaller. We repeat the calculations and find that V = 0.538 m/s, from which the volume flow rate is $\dot{V}$ = **10.5 Lpm**, and the net head of the pump is 17.0 m. The Reynolds number for the flow in the pipe is 1.09×10^4. **The volume flow rate has increased by about 9.5%**.

Discussion The increase in volume flow rate is modest. This is because only about half of the pump work goes into raising the elevation of the water – the other half goes into overcoming irreversible losses.

14-46

Solution We are to perform a regression analysis to translate tabulated pump performance data into an analytical expression, and then use this expression to predict the volume flow rate through a piping system.

Assumptions **1** The water is incompressible. **2** The flow is nearly steady since the reservoirs are large. **3** The water is at room temperature.

Properties The density and viscosity of water at T = 20°C are 998.0 kg/m^3 and 1.002 × 10^{-3} kg/m·s respectively.

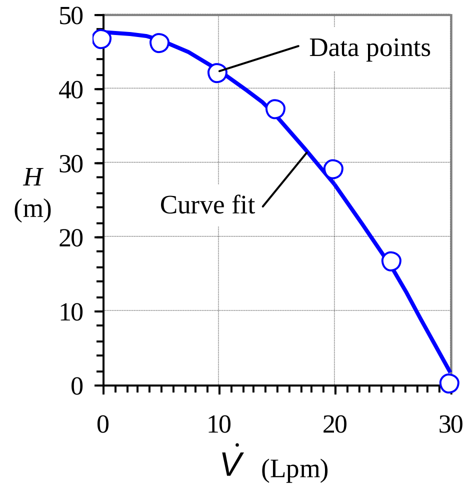


FIGURE 1
Tabulated data (symbols) and curve-fitted data (line) for $H_{available}$ versus $\dot{V}$ for the proposed pump.

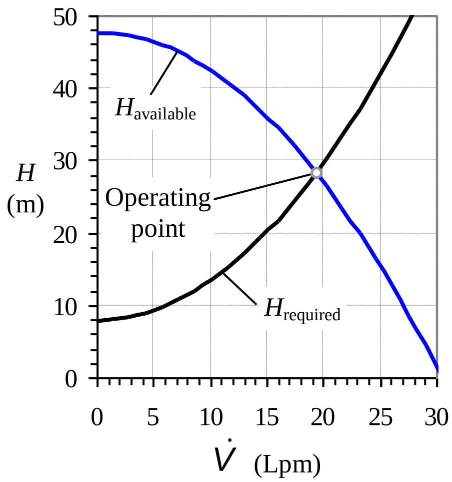


FIGURE 2
$H_{available}$ and $H_{required}$ versus $\dot{V}$ for a piping system with pump; the operating point is also indicated, where the two curves meet.

Analysis (*a*) We perform a regression analysis, and obtain $\boldsymbol{H_0}$ **= 47.6 m** and $\boldsymbol{a}$ **= 0.05119 m/Lpm^2**. The curve fit is reasonable, as seen in Fig. 1.

(*b*) We repeat the calculations of Problem 14-41 with the new pump performance coefficients, and find that V = 1.00 m/s, from which the volume flow rate is $\dot{V}$ = **19.5 Lpm**, and the net head of the pump is 28.3 m. The Reynolds number for the flow in the pipe is 2.03 × 10^4. **The volume flow rate has increased by about 69%**. **April's goal has *not* been reached**. She will need to search for an even stronger pump.

(*c*) A plot of net head versus volume flow rate is shown in Fig. 2.

Discussion As is apparent from Fig. 2, the required net head increases rapidly with increasing volume flow rate. Thus, doubling the flow rate would require a significantly heftier pump.

14-47

Solution We are to calculate the volume flow rate when the pipe diameter of a piping/pump system is doubled.

Analysis The analysis is identical to that of Problem 14-41 except for the diameter change. The calculations yield V = 0.19869 ≈ 0.199 m/s, from which the volume flow rate is $\dot{V}$ = **15.4 Lpm**, and the net head of the pump is 8.25 m. The Reynolds number for the flow in the pipe is 8.03×10^3. **The volume flow rate has increased by about 33%**. This agrees with our intuition since irreversible head losses go down significantly by increasing pipe diameter.

Discussion The gain in volume flow rate is significant because the irreversible head losses contribute to about half of the total pump head requirement in the original problem.

14-48

Solution We are to compare Reynolds numbers for a pipe flow system – the second case having a pipe diameter twice that of the first case.

Properties The density and viscosity of water at T = 20°C are 998.0 kg/m³ and 1.002×10^{-3} kg/m·s respectively.

Analysis From the results of the two problems, the Reynolds number of the first case is

Case 1 (D = 2.03 cm):

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{\left(998\text{ kg/m}^3\right)\left(0.59603\text{ m/s}\right)\left(0.0203\text{ m}\right)}{1.002\times10^{-3}\text{ kg/m}^3} = 1.21\times10^4$$

and that of the second case is

Case 2 (D = 4.06 cm):

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{\left(998\text{ kg/m}^3\right)\left(0.19869\text{ m/s}\right)\left(0.0406\text{ m}\right)}{1.002\times10^{-3}\text{ kg/m}^3} = 0.803\times10^4$$

Thus, **the Reynolds number of the larger diameter pipe is *smaller* than that of the smaller diameter pipe**. This may be somewhat surprising, but since average pipe velocity scales as the inverse of pipe diameter squared, Reynolds number increases linearly with pipe diameter due to the D in the numerator, but decreases quadratically with pipe diameter due to the V in the numerator. The net effect is a decrease in Re with pipe diameter when $\dot{V}$ is the same. In this problem, $\dot{V}$ increases somewhat as the diameter is doubled, but not enough to increase the Reynolds number.

Discussion At first glance, most people would think that Reynolds number increases as both diameter and volume flow rate increase, but this is not always the case.

14-49

Solution We are to compare the volume flow rate in a piping system with and without accounting for minor losses.

Analysis The analysis is identical to that of Problem 14-41, except we ignore all the minor losses. The calculations yield V = 0.604 m/s, from which the volume flow rate is $\dot{V}$ = **11.7 Lpm**, and the net head of the pump is 15.1 m. The Reynolds number for the flow in the pipe is 1.22×10^4. **The volume flow rate has increased by about 1.3%**. Thus, **minor losses are nearly negligible in this calculation**. This agrees with our intuition since the pipe is very long.

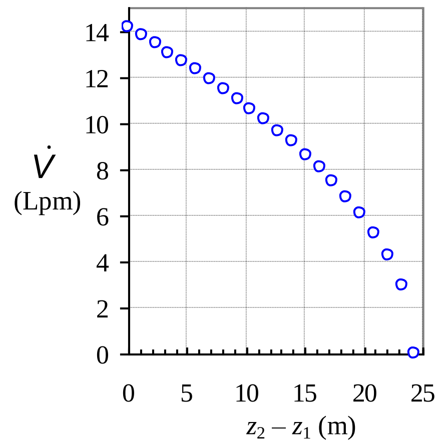
Discussion Since the Colebrook equation is accurate to at most 5%, a 1.3% change is well within the error. Nevertheless, it is not excessively difficult to include the minor losses, especially when solving the problem on a computer.

14-50

Solution We are to examine how increasing ($z_2 - z_1$) affects the volume flow rate of water pumped by the water pump.

Assumptions **1** The flow at any instant of time is still considered quasi-steady, since the surface level of the upper reservoir rises very slowly. **2** The minor losses, dimensions, etc., fluid properties, and all other assumptions are identical to those of Problem 14-43 except for the elevation difference ($z_2 - z_1$).

Analysis We repeat the calculations of Problem 14-43 for several values of ($z_2 - z_1$), ranging from 0 to H_0, the shutoff head of the pump, since above the shutoff head, the pump cannot overcome the elevation difference. **The volume flow rate is zero at the shutoff head of the pump**. The data are plotted here. As expected, the volume flow rate decreases as ($z_2 - z_1$) increases, starting at a maximum flow rate of about 14.1 Lpm when there is no elevation difference, and reaching zero (no flow) when ($z_2 - z_1$) = H_0 = 24.4 m. **The curve is *not* linear**, since neither the Darcy friction factor nor the pump performance curve are linear. If ($z_2 - z_1$) were increased beyond H_0, the pump would not be able to handle the elevation difference. Despite its valiant efforts, with blades spinning as hard as they could, **the water would flow backwards through the pump**.



Discussion You may wish to think of the backward-flow through the pump as a case in which the pump efficiency is *negative*. In fact, at ($z_2 - z_1$) = H_0, the pump could be replaced by a closed valve to keep the water from draining from the upper reservoir to the lower reservoir.

14-51

Solution We are to estimate the operating point of a given fan and duct system.

Assumptions **1** The flow is steady and incompressible. **2** The concentration of contaminants is low; the fluid properties are those of air alone. **3** The air is at 25°C and 101,300 Pa. **4** The air flowing in the duct is turbulent with $\alpha = 1.05$.

Properties For air at 25°C, $\mu = 1.849 \times 10^{-5}$ kg/m·s, $\rho = 1.184$ kg/m³, and $\nu = 1.562 \times 10^{-5}$ m²/s. The density of water at STP (for conversion to water head) is 997.0 kg/m³.

Analysis We apply the steady energy equation along a streamline from point 1 in the stagnant air region in the room to point 2 at the duct outlet,

$$H_{\text{required}} = \cancel{\frac{P_2 - P_1}{\rho g}} + \frac{\alpha_2 V_2^{\,2} - \cancel{\alpha_1 V_1^{\,2}}}{2g} + \cancel{(z_2 - z_1)} + h_{L,\text{total}} \tag{1}$$

P_1 is equal to P_{atm}, and P_2 is also equal to P_{atm} since the jet discharges into outside air on the roof of the building. Thus the pressure terms cancel out in Eq. 1. We ignore the air speed at point 1 since it is chosen (wisely) far enough away from the hood inlet so that the air is nearly stagnant. Finally, the elevation difference is neglected for gases. Equation 1 reduces to

Required net head:
$$H_{\text{required}} = \frac{\alpha_2 V_2^{\,2}}{2g} + h_{L,\text{total}} \tag{2}$$

The total head loss in Eq. 2 is a combination of major and minor losses. Since the duct diameter is constant,

Total irreversible head loss:
$$h_{L,\text{total}} = \left(f\frac{L}{D} + \sum K_L \right)\frac{V^2}{2g} \tag{3}$$

The required net head of the fan is thus

$$H_{\text{required}} = \left(\alpha_2 + f\frac{L}{D} + \sum K_L \right)\frac{V^2}{2g} \tag{4}$$

To find the operating point, we equate $H_{\text{available}}$ and H_{required}, being careful to keep consistent units. Note that the required head in Eq. 4 is expressed naturally in units of equivalent column height of the pumped fluid, which is air in this case. However, the available net head (given) is in terms of equivalent *water* column height. We convert constants H_0 and a in Eq. 1 to mm of *air* column for consistency by multiplying by the ratio of water density to air density,

$$H_{0,\text{ mm water}}\rho_{\text{water}} = H_{0,\text{ mm air}}\rho_{\text{air}} \quad \rightarrow \quad H_{0,\text{ mm air}} = H_{0,\text{ mm water}}\frac{\rho_{\text{water}}}{\rho_{\text{air}}} \quad \text{and} \quad a_{(\text{mm air})/\text{LPM}^2} = a_{(\text{mm water})/\text{LPM}^2}\frac{\rho_{\text{water}}}{\rho_{\text{air}}}$$

We re-write the given expression in terms of average duct velocity rather than volume flow rate,

Available net head:
$$H_{\text{available}} = H_0 - a\frac{\pi^2 D^4}{16}V^2 \tag{5}$$

Equating Eqs. 4 and 5 yields the operating point,

$$H_{\text{available}} = H_{\text{required}} \quad \rightarrow \quad H_0 - a\frac{\pi^2 D^4}{16}V^2 = \left(\alpha_2 + f\frac{L}{D} + \sum K_L \right)\frac{V^2}{2g} \tag{6}$$

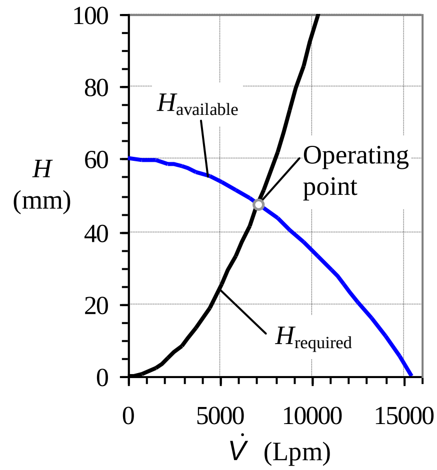
The dimensionless roughness factor is ε/D = 0.15/150 = 1.00×10^{-3}, and the sum of all the minor loss coefficients is $\sum K_L = 3.3 + (3\times 0.21) + 1.8 + 0.36 + 6.6 = 12.69$. Note that there is no minor loss associated with the exhaust, since point 2 is at the exit plane of the duct, and does not include irreversible losses associated with the turbulent jet. Equation 6 is an implicit equation for V since the Darcy friction factor is a function of Reynolds number Re = $\rho VD/\mu = VD/\nu$, as obtained from the Moody chart or the Colebrook equation. The solution can be obtained by an iterative method, or through use of a mathematical equation solver like EES. The result is V = 6.71 m/s, from which the volume flow rate is $\dot{V}$ = **7090 Lpm**.

Discussion We verify our results by comparing $H_{\text{available}}$ (given) and H_{required} (Eq. 5) at this flow rate: $H_{\text{available}}$ = 47.4 mm of water and H_{required} = 47.4 mm of water, both of which are equivalent to 40.0 m of air column.

14-52

Solution We are to plot H_{required} and $H_{\text{available}}$ versus $\dot{V}$, and indicate the operating point.

Analysis We use the equations of the previous problem, with the same constants and parameters, to generate the plot shown. The operating point is the location where the two curves intersect. The values of H and $\dot{V}$ at the operating point match those of the previous problem, as they should.



Discussion A plot like this, in fact, is an alternate method of obtaining the operating point. The operating point is at a volume flow rate near the center of the plot, indicating that the fan efficiency is probably reasonably high.

14-53

Solution We are to estimate the volume flow rate at the operating point without accounting for minor losses, and then we are to compare with the previous results.

Analysis All assumptions and properties are the same as those of Problem 14-52, except that we ignore all minor losses (we set $\Sigma K_L = 0$). The resulting volume flow rate at the operating point is $\dot{V}$ **= 10,900 Lpm** (to three significant digits), approximately 54% higher than for the case with minor losses taken into account. In this problem, minor losses are not "minor", and are by no means negligible. Even though the duct is fairly long (L/D is about 163), the minor losses are large, especially those through the damper and the one-way valve.

Discussion An error of 54% is not acceptable in this type of problem. Furthermore, since it is not difficult to account for minor losses, especially if the calculations are performed on a computer, it is wise not to ignore these terms.

14-54

Solution We are to calculate pressure at two locations in a blocked duct system.

Assumptions **1** The flow is steady. **2** The concentration of contaminants in the air is low; the fluid properties are those of air alone. **3** The air is at standard temperature and pressure (STP: 25°C and 101,300 Pa), and is incompressible.

Properties The density of water at 25°C is 997.0 kg/m^3.

Analysis Since the air is completely blocked by the one-way valve, there is no flow. Thus, there are no major or minor losses – just a pressure gain across the fan. Furthermore, the fan is operating at its shutoff head conditions. Since the pressure in the room is atmospheric, the gage pressure anywhere in the stagnant air region in the duct between the fan and the one-way valve is therefore equal to H_0 = 60.0 mm of water column. We convert to pascals as follows:

Gage pressure at both locations: $$P_{\text{gage}} = \rho_{\text{water}} g H_0 = \left(998.0\ \frac{\text{kg}}{\text{m}^3}\right)\left(9.81\ \frac{\text{m}}{\text{s}^2}\right)(0.060\ \text{m})\left(\frac{\text{N}\cdot\text{s}^2}{\text{kg}\cdot\text{m}}\right)\left(\frac{\text{Pa}\cdot\text{m}^2}{\text{N}}\right) = 587\ \text{Pa}$$

Thus, at either location, the gage pressure is **60.0 mm of water column, or 587 Pa**.

Discussion The answer depends only on the shutoff head of the fan – duct diameter, minor losses, etc, are irrelevant for this case since there is no flow. The fan should not be run for long time periods under these conditions, or it may burn out.

14-55E

Solution We are to estimate the operating point of a given fan and duct system.

Assumptions **1** The flow is steady. **2** The concentration of contaminants in the air is low; the fluid properties are those of air alone. **3** The air is at standard temperature and pressure (STP), and is incompressible. **4** The air flowing in the duct is turbulent with $\alpha = 1.05$.

Properties For air at STP (T = 77°F, P = 14.696 psi = 2116.2 lbf/ft^2), $\mu = 1.242 \times 10^{-5}$ lbm/ft·s, ρ = 0.07392 lbm/ft^3, and $\nu = 1.681 \times 10^{-4}$ ft^2/s. The density of water at STP (for conversion to inches of water head) is 62.24 lbm/ft^3.

Analysis We apply the steady energy equation along a streamline from point 1 in the stagnant air region in the room to point 2 at the duct outlet,

Required net head:
$$H_{\text{required}} = \cancel{\frac{P_2 - P_1}{\rho g}} + \frac{\alpha_2 V_2^{\,2} - \alpha_1 \cancel{V_1^{\,2}}}{2g} + \cancel{(z_2 - z_1)} + h_{L,\text{total}} \tag{1}$$

At point 1, P_1 is equal to P_{atm}, and at point 2, P_2 is also equal to P_{atm} since the jet discharges into the outside air on the roof of the building. Thus the pressure terms cancel out in Eq. 1. We ignore the air speed at point 1 since it is chosen (wisely) far enough away from the hood inlet so that the air is nearly stagnant. Finally, the elevation difference is neglected for gases. Equation 1 reduces to

$$H_{\text{required}} = \frac{\alpha_2 V_2^{\,2}}{2g} + h_{L,\text{total}} \tag{2}$$

The total head loss in Eq. 2 is a combination of major and minor losses, and depends on volume flow rate. Since the duct diameter is constant,

Total irreversible head loss:
$$h_{L,\text{total}} = \left(f\frac{L}{D} + \sum K_L \right)\frac{V^2}{2g} \tag{3}$$

The required net head of the fan is thus

$$H_{\text{required}} = \left(\alpha_2 + f\frac{L}{D} + \sum K_L \right)\frac{V^2}{2g} \tag{4}$$

To find the operating point, we equate $H_{\text{available}}$ and H_{required}, being careful to keep consistent units. Note that the required head in Eq. 4 is expressed naturally in units of equivalent column height of the pumped fluid, which is air in this case. However, the available net head (given) is in terms of equivalent water column height. We convert constants H_0 and a to inches of *air* column for consistency by multiplying by the ratio of water density to air density,

$$H_{0,\text{ inch water}}\rho_{\text{water}} = H_{0,\text{ inch air}}\rho_{\text{air}} \quad \rightarrow \quad H_{0,\text{ inch air}} = H_{0,\text{ inch water}}\frac{\rho_{\text{water}}}{\rho_{\text{air}}}$$

and similarly,

$$a_{(\text{inch air})/\text{SCFM}^2} = a_{(\text{inch water})/\text{SCFM}^2}\frac{\rho_{\text{water}}}{\rho_{\text{air}}}$$

We re-write the given expression in terms of average duct velocity rather than volume flow rate,

Available net head:
$$H_{\text{available}} = H_0 - a\frac{\pi^2 D^4}{16}V^2 \tag{5}$$

again taking care to keep consistent units. Equating Eqs. 4 and 5 yields

Operating point:
$$H_{\text{available}} = H_{\text{required}} \quad \rightarrow \quad H_0 - a\frac{\pi^2 D^4}{16}V^2 = \left(\alpha_2 + f\frac{L}{D} + \sum K_L \right)\frac{V^2}{2g} \tag{6}$$

The dimensionless roughness factor is ε/D = 0.0059/9.06 = 6.52 × 10^{-4}, and the sum of all the minor loss coefficients is

Minor losses: $$\sum K_L = 4.6 + (3\times 0.21) + 1.8 = 7.03$$

Note that there is no minor loss associated with the exhaust, since point 2 is taken at the exit plane of the duct, and does not include irreversible losses associated with the turbulent jet. Equation 6 is an implicit equation for V since the Darcy friction factor is a function of Reynolds number Re = $\rho VD/\mu$ = VD/ν, as obtained from either the Moody chart or the Colebrook equation. The solution can be obtained by an iterative method, or through use of a mathematical equation solver like EES. The result is V = 16.8 ft/s, from which the volume flow rate is $\dot{V}$ = **452 SCFM**. The Reynolds number is 7.63 × 10^4.

Discussion We verify our results by comparing $H_{\text{available}}$ (Eq. 1) and H_{required} (Eq. 5) at this flow rate: $H_{\text{available}}$ = 0.566 inches of water and H_{required} = 0.566 inches of water.

14-56E

Solution We are to calculate the value of $K_{L,\text{ damper}}$ such that the volume flow rate through the duct decreases by a factor of 3.

Analysis All assumptions and properties are the same as those of the previous problem. We set the volume flow rate to $\dot{V}$ = 452/3 = 150.7 SCFM, one-half of the previous result, and solve for $K_{L,\text{ damper}}$. The result is $\boldsymbol{K_{L,\text{ damper}} = 295}$, significantly higher than the value of 1.8 for the fully open case.

Discussion Because of the nonlinearity of the problem, we cannot simply double the damper's loss coefficient in order to decrease the flow rate by a factor of three. Indeed, the minor loss coefficient must be increased by a factor of more than 160. If a computer was used for the calculations of the previous problem, the solution here is most easily obtained by trial and error.

14-57E

Solution We are to estimate the volume flow rate at the operating point without accounting for minor losses, and then we are to compare with the previous results.

Analysis All assumptions and properties are the same as those of Problem 14-49, except that we ignore all minor losses (we set $\Sigma K_L = 0$). The resulting volume flow rate at the operating point is $\dot{V}$ **= 503 SCFM**, approximately 11% higher than for the case with minor losses taken into account. In this problem, minor losses are indeed "minor", although they are not negligible. We should not be surprised at this result, since there are several minor losses, and the duct is not extremely long (L/D is only 45.0).

Discussion An error of 11% may be acceptable in this type of problem. However, since it is not difficult to account for minor losses, especially if the calculations are performed on a computer, it is wise not to ignore these terms.

14-58E

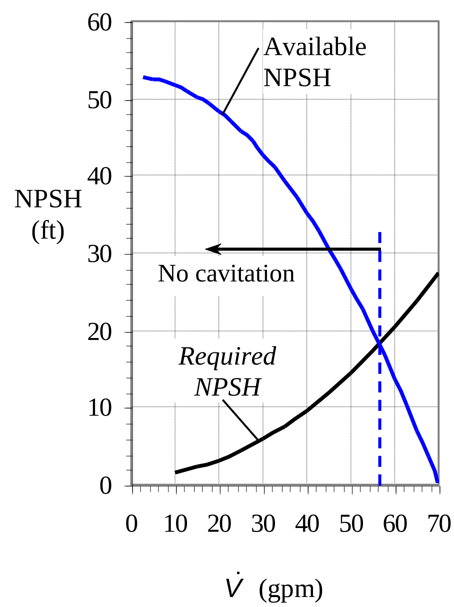
Solution For a given pump and piping system we are to estimate the maximum volume flow rate that can be pumped without cavitation.

Assumptions **1** The flow is steady. **2** The water is incompressible. **3** The flow is turbulent.

Properties P_{atm} = 14.696 psi = 2116.2 lbf/ft^2. For water at T = 77°F, $\mu = 6.002 \times 10^{-4}$ lbm/(ft·s), ρ = 62.24 lbm/ft^3, and P_v = 66.19 lbf/ft^2.

Analysis We apply the steady energy equation in head form along a streamline from point 1 at the reservoir surface to point 2 at the pump inlet,

$$\frac{P_1}{\rho g}+\cancel{\frac{\alpha_1 V_1^2}{2g}}+z_1+\cancel{h_{\text{pump,u}}}=\frac{P_2}{\rho g}+\frac{\alpha_2 V_2^2}{2g}+z_2+\cancel{h_{\text{turbine}}}+h_{L,\text{total}} \qquad (1)$$



In Eq. 1 we have ignored the water speed at the reservoir surface ($V_1 \approx 0$). There is no turbine in the piping system. Also, although there is a pump in the system, there is no pump between point 1 and point 2; hence the pump head term also drops out. We solve Eq. 1 for $P_2/(\rho g)$, which is the pump inlet pressure expressed as a head,

$$\frac{P_2}{\rho g}=\frac{P_{\text{atm}}}{\rho g}+\left(z_1-z_2\right)-\frac{\alpha_2 V_2^2}{2g}-h_{L,\text{total}} \qquad (2)$$

Note that in Eq. 2, we have recognized that $P_1 = P_{atm}$ since the reservoir surface is exposed to atmospheric pressure.

The available net positive suction head at the pump inlet is obtained from Eq. 14-8. After substitution of Eq. 2, approximating $\alpha_2 \approx 1$, we get an expression for the *available* NPSH:

$$\text{NPSH}=\frac{P_{\text{atm}}-P_v}{\rho g}+\left(z_1-z_2\right)-h_{L,\text{total}} \qquad (3)$$

Since we know P_{atm}, P_v, and the elevation difference, all that remains is to estimate the total irreversible head loss through the piping system from the reservoir surface (1) to the pipe inlet (2), which depends on volume flow rate. Since the pipe diameter is constant, the total irreversible head loss becomes

$$h_{L,\text{total}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g} \qquad (4)$$

The rest of the problem is most easily solved on a computer. For a given volume flow rate, we calculate speed V and Reynolds number Re. From Re and the known pipe roughness, we use the Moody chart (or the Colebrook equation) to obtain friction factor f. The sum of all the minor loss coefficients is

$$\sum K_L = 0.5+0.3+6.0=6.8 \qquad (5)$$

where we have not included the minor losses downstream of the pump, since they are irrelevant to the present analysis.

We make one calculation by hand for illustrative purposes. At $\dot{V}$ = 40.0 gpm, the average speed of water through the pipe is

$$V=\frac{\dot{V}}{A}=\frac{4\dot{V}}{\pi D^2}=\frac{4\left(40.0\text{ gal/min}\right)}{\pi\left(1.2\text{ in}\right)^2}\left(\frac{231\text{ in}^3}{\text{gal}}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right)\left(\frac{1\text{ ft}}{12\text{ in}}\right)=11.35\text{ ft/s} \qquad (6)$$

which produces a Reynolds number of Re = $\rho VD/\mu = 1.17 \times 10^5$. At this Reynolds number, and with roughness factor ε/D = 0, the Colebrook equation yields f = 0.0174. After substitution of the given variables along with f, D, L, and Eqs. 4, 5, and 6 into Eq. 3, we get

$$\text{NPSH} = \frac{(2116.2 - 66.19)\ \text{lbf/ft}^2}{\left(62.24\ \text{lbm/ft}^3\right)\left(32.174\ \text{ft/s}^2\right)}\left(\frac{32.174\ \text{lbm ft}}{\text{s}^2\text{lbf}}\right) + 20.0\ \text{ft} - \left(0.0174\frac{12.0\ \text{ft}}{0.10\ \text{ft}} + 6.8\right)\frac{(11.35\ \text{ft/s})^2}{2\left(32.174\ \text{ft/s}^2\right)} = 35.1\ \text{ft}$$

The required net positive suction head is obtained from the given expression. At our example flow rate of 40.0 gpm we see that $\text{NPSH}_{\text{required}}$ is about 9.6 ft. Since the actual NPSH is much higher than this, we need not worry about cavitation at this flow rate. We use a spreadsheet to calculate NPSH as a function of volume flow rate, and the results are plotted. It is clear from the plot that **cavitation occurs at flow rates above about 56 gallons per minute**.

Discussion $\text{NPSH}_{\text{required}}$ rises with volume flow rate, but the actual or available NPSH decreases with volume flow rate.

14-59E

Solution We are to calculate the volume flow rate below which cavitation in a pump is avoided.

Assumptions **1** The flow is steady. **2** The water is incompressible.

Properties P_{atm} = 14.696 psi = 2116.2 lbf/ft^2. For water at T = 113^{o}F, we interpolate in the property tables of the Appendix to get $\mu = 4.013 \times 10^{-4}$ lbm/ft·s, ρ = 61.82 lbm/ft^3, and P_v = 201.9 lbf/ft^2.

Analysis The procedure is identical to that of the previous problem, except for the water properties. The calculations predict that **the pump cavitates at volume flow rates greater than about 55 gpm**. This is somewhat lower than the result of the previous problem, as expected, since cavitation occurs more readily in warmer water.

Discussion Note that NPSH$_{required}$ does not depend on water temperature, but the actual or available NPSH decreases with temperature.

14-60

Solution For a given pump and piping system we are to estimate the maximum volume flow rate that can be pumped without cavitation.

Assumptions **1** The flow is steady. **2** The water is incompressible. **3** The flow is turbulent.

Properties Standard atmospheric pressure is P_{atm} = 101.3 kPa. For water at T = 25°C, ρ = 997.0 kg/m^3, $\mu = 8.91\times10^{-4}$ kg/m·s, and P_v = 3.169 kPa.

Analysis We apply the steady energy equation in head form along a streamline from point 1 at the reservoir surface to point 2 at the pump inlet,

Energy equation:
$$\frac{P_1}{\rho g}+\cancel{\frac{\alpha_1 V_1^2}{2g}}+z_1+\cancel{h_{\text{pump,u}}}=\frac{P_2}{\rho g}+\frac{\alpha_2 V_2^2}{2g}+z_2+\cancel{h_{\text{turbine}}}+h_{L,\text{total}} \tag{1}$$

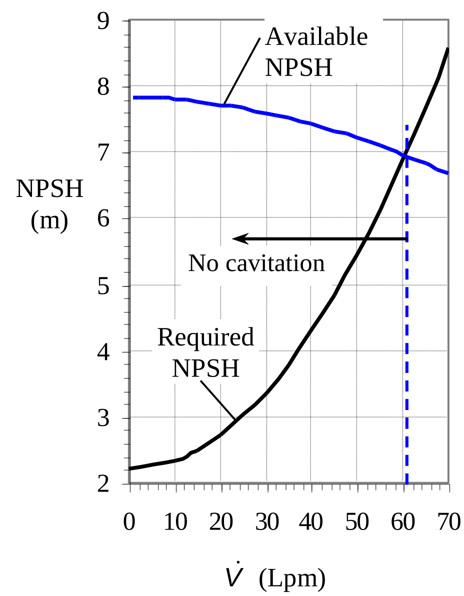
In Eq. 1 we have ignored the water speed at the reservoir surface ($V_1 \approx 0$). There is no turbine in the piping system. Also, although there is a pump in the system, there is no pump between point 1 and point 2; hence the pump head term also drops out. We solve Eq. 1 for $P_2/(\rho g)$, which is the pump inlet pressure expressed as a head,

Pump inlet pressure head:
$$\frac{P_2}{\rho g}=\frac{P_{\text{atm}}}{\rho g}+\left(z_1-z_2\right)-\frac{\alpha_2 V_2^2}{2g}-h_{L,\text{total}} \tag{2}$$

Note that in Eq. 2, we have recognized that $P_1 = P_{atm}$ since the reservoir surface is exposed to atmospheric pressure.

The available net positive suction head at the pump inlet is obtained from Eq. 14-8. After substitution of Eq. 2, and approximating α_2 as 1.0, we get

Available NPSH:
$$\text{NPSH}=\frac{P_{\text{atm}}-P_v}{\rho g}+\left(z_1-z_2\right)-h_{L,\text{total}} \tag{3}$$



Since we know P_{atm}, P_v, and the elevation difference, all that remains is to estimate the total irreversible head loss through the piping system from the reservoir surface (1) to the pipe inlet (2), which depends on volume flow rate. Since the pipe diameter is constant, the total irreversible head loss is

$$h_{L,\text{total}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g} \tag{4}$$

The rest of the problem is most easily solved on a computer. For a given volume flow rate, we calculate speed V and Reynolds number Re. From Re and the known pipe roughness, we use the Moody chart (or the Colebrook equation) to obtain friction factor f. The sum of all the minor loss coefficients is

$$\sum K_L = 0.85+0.3=1.15 \tag{5}$$

where we have not included the minor losses downstream of the pump, since they are irrelevant to the present analysis.

We make one calculation by hand for illustrative purposes. At $\dot{V}$ = 40.0 Lpm, the average speed of water through the pipe is

$$V=\frac{\dot{V}}{A}=\frac{4\dot{V}}{\pi D^2}=\frac{4\left(40.0\text{ L/min}\right)}{\pi\left(0.024\text{ m}\right)^2}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right)=1.474\text{ m/s} \tag{6}$$

which produces a Reynolds number of Re = $\rho VD/\mu = 3.96 \times 10^4$. At this Reynolds number, and with roughness factor ε/D = 0, the Colebrook equation yields f = 0.0220. After substitution of the given variables, along with f, D, L, and Eqs. 4, 5, and 6 into Eq. 3, we calculate the available NPSH,

$$\text{NPSH} = \frac{\left(101{,}300 - 3{,}169\right)\ \text{N/m}^2}{\left(997.0\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)}\left(\frac{\text{kg m}}{\text{s}^2\text{N}}\right) - 2.2\ \text{m} - \left(0.0220\frac{2.8\ \text{m}}{0.024\ \text{m}} + 1.15\right)\frac{\left(1.474\ \text{m/s}\right)^2}{2\left(9.81\ \text{m/s}^2\right)} = 7.42\ \text{m} \qquad (7)$$

The required net positive suction head is obtained from the given expression. At our example flow rate of 40.0 Lpm we see that $\text{NPSH}_{\text{required}}$ is about 4.28 m. Since the actual NPSH is higher than this, the pump does not cavitate at this flow rate. We use a spreadsheet to calculate NPSH as a function of volume flow rate, and the results are plotted. It is clear from this plot that **cavitation occurs at flow rates above 60.5 liters per minute**.

Discussion $\text{NPSH}_{\text{required}}$ rises with volume flow rate, but the actual or available NPSH decreases with volume flow rate.

14-61

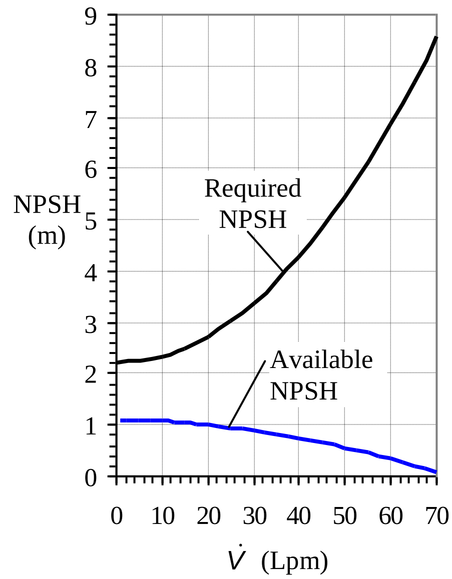
Solution We are to calculate the volume flow rate below which cavitation in a pump is avoided, at two temperatures.

Assumptions **1** The flow is steady. **2** The water is incompressible.

Properties Standard atmospheric pressure is P_{atm} = 101.3 kPa. For water at T = 80°C, ρ = 971.9 kg/m^3, μ = 3.55×10^{-4} kg/m·s, and P_v = 47.35 kPa. At T = 90°C, ρ = 965.3 kg/m^3, μ = 3.15×10^{-4} kg/m·s, and P_v = 70.11 kPa.

Analysis The procedure is identical to that of the previous problem, except for the water properties. The calculations predict that **at T = 80°C, the pump cavitates at volume flow rates greater than 28.0 Lpm**. This is substantially lower than the result of the previous problem, as expected, since cavitation occurs more readily in warmer water.

At 90°C, the vapor pressure is very high since the water is near boiling (at atmospheric pressure, water boils at 100°C). For this case, the curves of NPSH$_{available}$ and NPSH$_{required}$ do not cross at all as seen in the plot, implying that **the pump cavitates at any flow rate when T = 90°C**.



Discussion Note that NPSH$_{required}$ does not depend on water temperature, but the actual or available NPSH decreases with temperature.

14-62

Solution We are to calculate the volume flow rate below which cavitation in a pump is avoided, and compare with a previous case with a smaller pipe diameter.

Assumptions **1** The flow is steady. **2** The water is incompressible.

Properties Standard atmospheric pressure is P_{atm} = 101.3 kPa. For water at T = 25°C, ρ = 997.0 kg/m^3, μ = 8.91×10^{-4} kg/m·s, and P_v = 3.169 kPa.

Analysis The analysis is identical to that of Problem 14-58, except that the pipe diameter is 48.0 mm instead of 24.0 mm. Compared to problem 14-58, at a given volume flow rate, the average speed through the pipe decreases by a factor of four since the pipe area increases by a factor of four. The Reynolds number goes down by a factor of two, but the flow is still turbulent (At our sample flow rate of 40.0 Lpm, Re = 1.98 × 10^4). For a smooth pipe at this Reynolds number, f = 0.0260. and the available NPSH is 7.81 m, slightly higher than the previous case with the smaller diameter pipe. After repeating the calculations at several flow rates, we find that **the pump cavitates at $\dot{V}$ = 65.5 Lpm**. This represents an increase of about 8.3%. **Cavitation occurs at a higher volume flow rate when the pipe diameter is increased because the irreversible head losses in the piping system upstream of the pump are decreased**.

Discussion If a computer program like EES was used for Problem 14-58, it is a trivial matter to change the pipe diameter and re-do the calculations.

14-63E

Solution We are to calculate the mass flow rate of slurry through a two-lobe rotary pump for given values of lobe volume and rotation rate.

Assumptions **1** The flow is steady in the mean. **2** There are no leaks in the gaps between lobes or between lobes and the casing. **3** The slurry is incompressible.

Analysis By studying the figure provided with this problem, we see that for each 360° rotation of the two counter-rotating shafts (n = 1 rotation), the total volume of pumped fluid is

Closed volume pumped per rotation: $$V_{\text{closed}} = 4V_{\text{lobe}} \quad (1)$$

The volume flow rate is then calculated from Eq. 14-11,

$$\dot{V} = \dot{n}\frac{V_{closed}}{n} = (175\text{ rot/min})\frac{4(0.110\text{ gal})}{1\text{ rot}} = \mathbf{77.0\,gal/min} \quad (2)$$

Thus, **the volume flow rate is 77.0 gpm**.

Discussion If there were leaks in the pump, the volume flow rate would be lower. The fluid density was not used in the calculation of volume flow rate. However, the higher the density, the higher the required shaft torque and brake horsepower.

14-64E

Solution We are to calculate the mass flow rate of slurry through a three-lobe rotary pump for given values of lobe volume and rotation rate.

Assumptions **1** The flow is steady in the mean. **2** There are no leaks in the gaps between lobes or between lobes and the casing. **3** The slurry is incompressible.

Analysis When there are three lobes, three lobe volumes are pumped for each 360° rotation of each rotor (n = 1 rotation). Thus, the total volume of pumped fluid is

Closed volume pumped per rotation: $$V_{\text{closed}} = 6V_{\text{lobe}} \quad (1)$$

The volume flow rate is then calculated from Eq. 14-11,

$$\dot{V} = \dot{n}\frac{V_{closed}}{n} = (175\text{ rot/min})\frac{6(0.0825\text{ gal})}{1\text{ rot}} = 86.63\text{ gal/min} \cong \mathbf{86.6\,gal/min} \quad (2)$$

Thus, **the volume flow rate is 86.6 gpm** (to three significant digits).

Discussion If there were leaks in the pump, the volume flow rate would be lower. This flow rate is slightly lower than that of the previous problem. Why? For the same overall diameter, it is clear from geometry that the more lobes, the less the volume per lobe, and the more “wasted” volume inside the pump – but the flow may be smoother (less unsteady) with more lobes.

14-65

Solution We are to calculate the volume flow rate of tomato paste through a positive-displacement pump for given values of lobe volume and rotation rate.

Assumptions **1** The flow is steady in the mean. **2** There are no leaks in the gaps between lobes or between lobes and the casing. **3** The fluid is incompressible.

Analysis By studying Fig. 14-27 or 14-30, we see that for each 360° rotation of the two counter-rotating shafts (n = 1 rotation), the total volume of pumped fluid is

Closed volume pumped per rotation:
$$V_{\text{closed}} = 4V_{\text{lobe}} \tag{1}$$

The volume flow rate is then calculated from Eq. 14-11,

$$\dot{V} = \dot{n}\frac{V_{\text{closed}}}{n} = (336 \text{ rot/min})\frac{4(3.64 \text{ cm}^3)}{1 \text{ rot}} = 4892.2 \text{ cm}^3/\text{min} \cong \mathbf{4890\ cm^3/min} \tag{2}$$

Thus, **the volume flow rate is 4890 cm^3/min** (to three significant digits).

Discussion If there were leaks in the pump, the volume flow rate would be lower. The fluid density was not used in the calculation of volume flow rate. However, the higher the fluid density, the higher the required shaft torque and brake horsepower.

14-66

Solution We are to calculate the volume flow rate per rotation of a gear pump.

Assumptions **1** The flow is steady in the mean. **2** There are no leaks in the gaps between lobes or between lobes and the casing. **3** The fluid is incompressible.

Analysis From Fig. 14-26c, we count 14 teeth per gear. Thus, for each 360° rotation of each gear (n = 1 rotation), 14·(0.350 cm^3) of fluid is pumped. Sine there are two gears, the total volume of fluid pumped per rotation is 2(14)(0.350 cm^3) = **9.80 cm^3**.

Discussion The actual volume flow rate will be lower than this due to leakage in the gaps.

14-67

Solution We are to calculate the brake horsepower and net head of an idealized centrifugal pump at a given volume flow rate and rotation rate.

Assumptions **1** The flow is steady in the mean. **2** The water is incompressible. **3** The efficiency of the pump is 100% (no irreversible losses).

Properties We take the density of water to be ρ = 998.0 kg/m^3.

Analysis Since the volume flow rate (capacity) is given, we calculate the normal velocity components at the inlet and the outlet using Eq. 14-12,

Normal velocity, inlet:
$$V_{1,\mathrm{n}} = \frac{\dot{V}}{2\pi r_1 b_1} = \frac{0.573\ \mathrm{m^3/s}}{2\pi(0.120\ \mathrm{m})(0.180\ \mathrm{m})} = 4.222\ \mathrm{m/s}$$

$V_1 = V_{1,\mathrm{n}}$, and $V_{1,\mathrm{t}} = 0$, since $\alpha_1 = 0^\circ$. Similarly, $V_{2,\mathrm{n}}$ = 2.3456 m/s and

Tangential component of absolute velocity at impeller outlet:
$$V_{2,\mathrm{t}} = V_{2,\mathrm{n}} \tan\alpha_2 = (2.3456\ \mathrm{m/s})\tan(35^\circ) = 1.6424\ \mathrm{m/s}$$

Now we use Eq. 14-17 to predict the net head,

$$H = \frac{\omega}{g}\left(r_2 V_{2,\mathrm{t}} - r_1 \underbrace{\cancel{V_{1,\mathrm{t}}}}_{0}\right) = \frac{78.54\ \mathrm{rad/s}}{9.81\ \mathrm{m/s^2}}(0.240\ \mathrm{m})(1.6424\ \mathrm{m/s}) = 3.1558\ \mathrm{m} \cong \mathbf{3.16\ m}$$

Finally, we use Eq. 14-16 to predict the required brake horsepower,

$$bhp = \rho g \dot{V} H = (998.\ \mathrm{kg/m^3})(9.81\ \mathrm{m/s^2})(0.573\ \mathrm{m^3/s})(3.1558\ \mathrm{m})\left(\frac{\mathrm{W\cdot s^3}}{\mathrm{kg\cdot m^2}}\right) = \mathbf{17{,}700\ W}$$

Discussion The actual net head delivered to the water will be lower than this due to inefficiencies. Similarly, actual brake horsepower will be higher than that predicted here due to inefficiencies in the pump, friction on the shaft, etc.

14-68

Solution We are to calculate the brake horsepower and net head of an idealized centrifugal pump at a given volume flow rate and rotation rate.

Assumptions **1** The flow is steady in the mean. **2** The water is incompressible. **3** The efficiency of the pump is 100% (no irreversible losses).

Properties We take the density of water to be $\rho = 998.0\ \text{kg/m}^3$.

Analysis Since the volume flow rate (capacity) is given, we calculate the normal velocity components at the inlet and the outlet using Eq. 14-12,

Normal velocity, inlet:
$$V_{1,\text{n}} = \frac{\dot{V}}{2\pi r_1 b_1} = \frac{0.573\ \text{m}^3/\text{s}}{2\pi\left(0.120\ \text{m}\right)\left(0.180\ \text{m}\right)} = 4.2220\ \text{m/s}$$

We calculate the tangential component of absolute velocity at the impeller inlet,

Tangential velocity, inlet:
$$V_{1,t} = V_{1,n}\tan\alpha_2 = (4.2220\ \text{m/s})\tan(7°) = 0.51840\ \text{m/s}$$

Similarly, $V_{2,\text{n}}$ = 2.3456 m/s and $V_{2,\text{t}}$ = 1.6424 m/s. Now we use Eq. 14-17 to predict the net head,

Net head:
$$H = \frac{\omega}{g}(r_2V_{2,t} - r_1V_{1,t}) = \frac{78.54\ \text{rad/s}}{9.81\ \text{m/s}^2}\left[(0.240\ \text{m})(1.6424\ \text{m/s}) - (0.120\ \text{m})(0.51840\ \text{m/s})\right]$$
$$= 2.6578\ \text{m} \cong \mathbf{2.66\ m}$$

Finally, we use Eq. 14-16 to predict the required brake horsepower,

Brake horsepower:
$$bhp = \rho g\dot{V}H = (998.0\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.573\ \text{m}^3/\text{s})(2.6578\ \text{m})\left(\frac{\text{W}\cdot\text{s}^3}{\text{kg}\cdot\text{m}^2}\right) = 14{,}910\ \text{W} \cong \mathbf{14{,}900\ W}$$

The brake horsepower and net head are considerably lower than those of the previous problem (by about 16%). So, we conclude that the angle at which the fluid impinges on the impeller blade *is* a critical parameter in the design of centrifugal pumps.

Discussion The actual net head delivered to the water will be lower than this due to inefficiencies. Similarly, actual brake horsepower will be higher than that predicted here due to inefficiencies in the pump, friction on the shaft, etc.

14-69

Solution We are to calculate the brake horsepower and net head of an idealized centrifugal pump at a given volume flow rate and rotation rate.

Assumptions **1** The flow is steady in the mean. **2** The water is incompressible. **3** The efficiency of the pump is 100% (no irreversible losses).

Properties We take the density of water to be ρ = 998.0 kg/m^3.

Analysis Since the volume flow rate (capacity) is given, we calculate the normal velocity components at the inlet and the outlet using Eq. 14-12,

Normal velocity, inlet:
$$V_{1,\mathrm{n}} = \frac{\dot{V}}{2\pi r_1 b_1} = \frac{0.573\ \mathrm{m^3/s}}{2\pi\left(0.120\ \mathrm{m}\right)\left(0.180\ \mathrm{m}\right)} = 4.2220\ \mathrm{m/s}$$

We calculate the tangential component of absolute velocity at the impeller inlet,

Tangential velocity, inlet:
$$V_{1,\mathrm{t}} = V_{1,\mathrm{n}} \tan\alpha_2 = \left(4.2220\ \mathrm{m/s}\right)\tan(-10^{\mathrm{o}}) = -0.74446\ \mathrm{m/s}$$

Similarly, $V_{2,\mathrm{n}}$ = 2.3456 m/s and $V_{2,\mathrm{t}}$ = 1.6424 m/s. Now we use Eq. 14-17 to predict the net head,

Net head:
$$H = \frac{\omega}{g}\left(r_2 V_{2,\mathrm{t}} - r_1 V_{1,\mathrm{t}}\right) = \frac{78.54\ \mathrm{rad/s}}{9.81\ \mathrm{m/s^2}}\left(\left(0.240\ \mathrm{m}\right)\left(1.6424\ \mathrm{m/s}\right) - \left(0.120\ \mathrm{m}\right)\left(-0.74446\ \mathrm{m/s}\right)\right)$$
$$= 3.87104\ \mathrm{m} \cong \mathbf{3.87\ m}$$

Finally, we use Eq. 14-16 to predict the required brake horsepower,

Brake horsepower:
$$bhp = \rho g \dot{V} H = \left(998.\ \mathrm{kg/m^3}\right)\left(9.81\ \mathrm{m/s^2}\right)\left(0.573\ \mathrm{m^3/s}\right)\left(3.87104\ \mathrm{m}\right)\left(\frac{\mathrm{W\cdot s^3}}{\mathrm{kg\cdot m^2}}\right) = \mathbf{21,700\ W}$$

The brake horsepower and net head are considerably higher than those of the original problem (by about 23%). So, we conclude that the angle at which the fluid impinges on the impeller blade *is* a critical parameter in the design of centrifugal pumps. Here, reverse swirl has improved (increased the net head), but at the expense of higher bhp. So, **a little reverse flow *is* desirable in terms of increasing the net head, but the increase in net head comes at a cost – more required power**. Also, keep in mind that we are neglecting losses (assuming 100% efficiency). In real life, **losses at the inlet may be significantly greater when the flow does not impinge parallel to the blades**.

Discussion The actual net head delivered to the water will be lower than this due to inefficiencies. Similarly, actual brake horsepower will be higher than that predicted here due to inefficiencies in the pump, friction on the shaft, etc.

14-70

Solution For given flow conditions and stator blade shape at a given radius, we are to design the rotor blade. Specifically, we are to calculate the leading and trailing edge angles of the rotor blade and sketch its shape. We are also to decide how many rotor blades to construct.

Assumptions **1** The air is nearly incompressible. **2** The flow area between hub and tip is constant. **3** Two-dimensional blade row analysis is appropriate.

Analysis First we analyze flow through the stator from an absolute reference frame, using the two-dimensional approximation of a cascade (blade row) of stator blades as sketched in Fig. 1. Flow enters axially (horizontally), and is turned 26.6° downward. Since the axial component of velocity must remain constant to conserve mass, the magnitude of the velocity leaving the trailing edge of the stator, $\vec{V}_{st}$ is calculated,

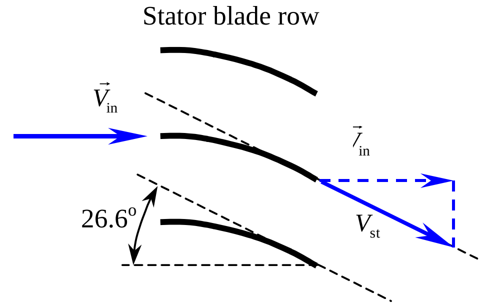


FIGURE 1
Analysis of the stator of a vaneaxial flow fan as a two-dimensional cascade of stator blades; absolute reference frame.

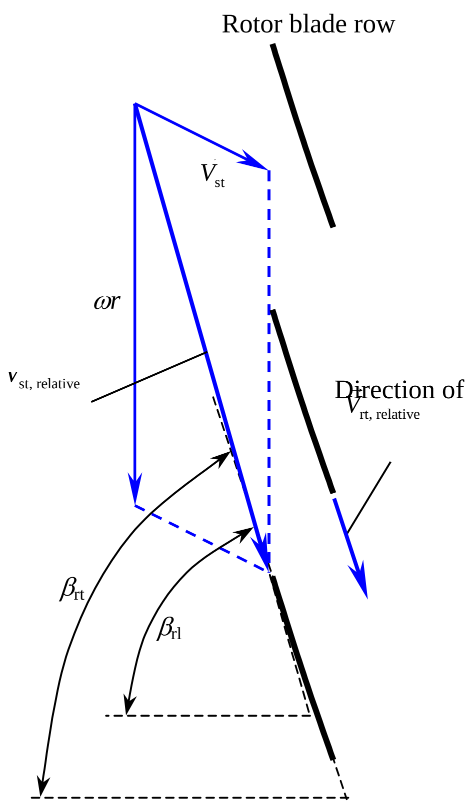


FIGURE 2
Analysis of the stator trailing edge velocity as it impinges on the rotor leading edge; relative reference frame.

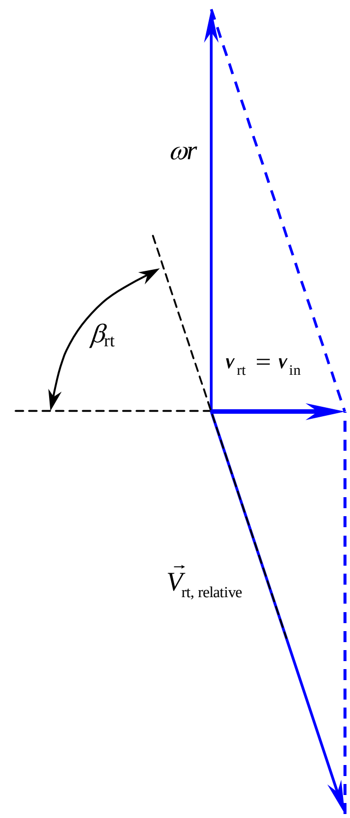


FIGURE 3
Analysis of the rotor trailing edge velocity; absolute reference frame.

$$V_{st} = \frac{V_{in}}{\cos\beta_{st}} = \frac{31.4 \text{ m/s}}{\cos\left(26.6^{\circ}\right)} = 35.12 \text{ m/s} \qquad (1)$$

The direction of $\vec{V}_{st}$ is assumed to be that of the stator trailing edge. In other words we assume that the flow turns nicely through the blade row and exits parallel to the trailing edge of the blade, as shown in the sketch.

We convert $\vec{V}_{st}$ to the *relative* reference frame moving with the rotor blades. At a radius of 0.50 m, the tangential velocity of the rotor blades is

$$u_\theta = \omega r = \left[(1800 \text{ rot/min}) \left(\frac{2\pi \text{ rad}}{\text{rot}} \right) \left(\frac{1 \text{ min}}{60 \text{ s}} \right) \right] (0.50 \text{ m}) = 94.25 \text{ m/s} \tag{2}$$

Since the rotor blade row moves upward in the figure provided with this problem, we add a *downward* velocity with magnitude given by Eq. 2 to translate $\vec{V}_{\text{st}}$ into the rotating reference frame sketched in Fig. 2. The angle of the leading edge of the rotor, β_{rl}, can be calculated. After some trig,

$$\beta_{\text{rl}} = \arctan \frac{\omega r + V_{\text{in}} \tan \beta_{\text{st}}}{V_{\text{in}}} = \arctan \frac{94.25 \text{ m/s} + (31.4 \text{ m/s}) \tan(26.6^\circ)}{31.4 \text{ m/s}} = 74.06^\circ \tag{3}$$

The air must now be turned by the rotor blade row in such a way that it leaves the trailing edge of the rotor blade at zero angle (axially – no swirl) from an absolute reference frame. This determines the rotor's trailing edge angle, β_{rt}. Specifically, when we add an *upward* velocity of magnitude ωr (Eq. 2) to the relative velocity exiting the trailing edge of the rotor, $\vec{V}_{\text{rt, relative}}$, we convert back to the absolute reference frame, and obtain $\vec{V}_{\text{rt}}$, the velocity leaving the rotor trailing edge. It is this velocity, $\vec{V}_{\text{rt}}$, which must be axial (horizontal). Furthermore, to conserve mass $\vec{V}_{\text{rt}}$ must equal $\vec{V}_{\text{in}}$ since we are assuming incompressible flow. Working "backwards" we construct $\vec{V}_{\text{rt, relative}}$ in Fig. 3. Some trigonometry reveals that

$$\beta_{\text{rt}} = \arctan \frac{\omega r}{V_{\text{in}}} = \arctan \frac{94.25 \text{ m/s}}{31.4 \text{ m/s}} = 71.57^\circ \tag{4}$$

We conclude that the rotor blade at this radius has a leading edge angle of about **74.1°** (Eq. 3) and a trailing edge angle of about **71.6°** (Eq. 4). A sketch of the rotor blade is provided in Fig. 2; it is clear that the blade is nearly straight, at least at this radius.

Finally, to avoid interaction of the stator blade wakes with the rotor blade leading edges, we choose the number of rotor blades such that it has no common denominator with the number of stator blades. Since there are 18 stator blades, we pick a number like **13**, **17**, or **19** rotor blades. 16 would not be appropriate since it shares a common denominator of 2 with the number 18.

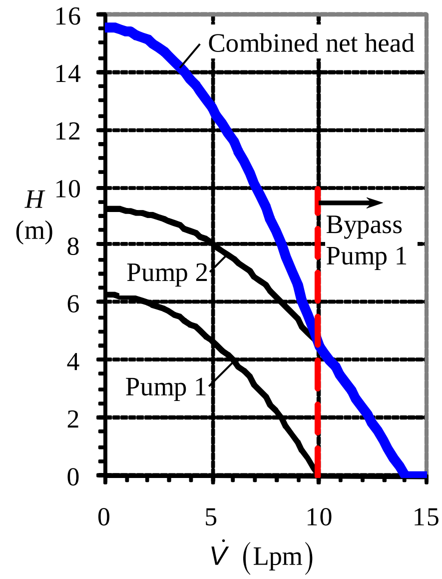
Discussion We can easily repeat the calculation for all radii from hub to tip, completing the design of the entire rotor.

14-71

Solution We are to calculate the combined shutoff head and free delivery for two pumps in series, and discuss why the weaker pump should be shut off and bypassed above some flow rate.

Assumptions **1** The water is incompressible. **2** The flow is steady.

Analysis The pump performance curves for both pumps individually and for their combination in series are plotted here. At zero flow rate, the shutoff head of the two pumps in series is equal to the sum of their individual shutoff heads: $H_{0,\text{combined}} = H_{0,1} + H_{0,2}$ = 6.33 m + 9.25 m. Thus **the combined shutoff head is 15.6 m** (to three significant digits). At free delivery conditions (zero net head), the free delivery of the two pumps in series is limited to that of the stronger pump, in this case Pump 2: $\dot{V}_{\text{max,combined}} = \text{MAX}(\dot{V}_{\text{max},1}, \dot{V}_{\text{max},2})$ = 13.999 Lpm. Thus **the combined free delivery is 14.0 Lpm** (to three significant digits).



As volume flow rate increases, the combined net pump head is equal to the sum of the net pump heads of the individual pumps, as seen on the plot. However, the free delivery of Pump 1 is

$$0 = H_0 - a\dot{V}^2 \quad \rightarrow \quad \dot{V}_{\text{max}} = \sqrt{\frac{H_0}{a}} = \sqrt{\frac{6.33 \text{ m}}{0.0633 \text{ m/(Lpm)}^2}} = 10.0 \text{ Lpm}$$

Above this flow rate, Pump 1 no longer contributes to the flow. In fact, it becomes a liability to the system since its net head would be *negative* at flow rates above 10.0 Lpm. For this reason, **it is wise to shut off and bypass Pump 1 above 10.0 Lpm**. We are assuming no additional losses in the bypass line around pump 1.

Discussion The free delivery of Pump 2 is 14.0 Lpm. The free delivery of the two-pump system is no better than that of Pump 2 alone, since Pump 1 is shut off and bypassed at flow rates above 10.0 Lpm.

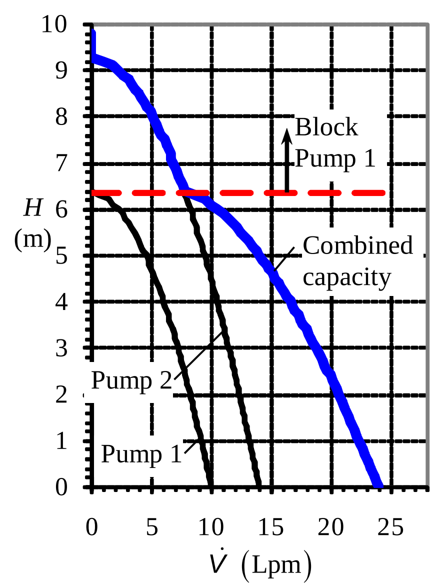
14-72

Solution We are to calculate the free delivery and shutoff head for two pumps in parallel, and discuss why the weaker pump should be shut off and bypassed above some net head.

Assumptions **1** The water is incompressible. **2** The flow is steady.

Analysis The pump performance curves for both pumps and for their combination in parallel are plotted. At zero flow rate, the shutoff head of the two pumps in parallel is equal to that of the stronger pump, in this case Pump 2: $H_{0,\text{combined}} = \text{MAX}(H_{0,1}, H_{0,2})$ = 9.25 m. Thus **the combined shutoff head is 9.25 m**. At free delivery conditions (zero net head), the free delivery of the two pumps in parallel is the sum of their individual free deliveries: $\dot{V}_{\text{max,combined}} = \dot{V}_{\text{max},1} + \dot{V}_{\text{max},2}$ = 10.0 Lpm + 14.0 Lpm. Thus **the combined free delivery is 24.0 Lpm**.

As net head increases, the combined capacity is equal to the sum of the capacities of the individual pumps, as seen in the plot. However, the shutoff head of Pump 1 (6.33 m) is lower than that of Pump 2 (9.25 m). Above the shutoff head of Pump 1, that pump no longer contributes to the flow. In fact, it becomes a liability to the system since it cannot sustain such a high head. If not shut off and blocked, the volume flow rate through Pump 1 would be *negative* at net heads above 6.33 m. For this reason, **it is wise to shut off and block Pump 1 above 6.33 m**.



Discussion The shutoff head of Pump 2 is 9.25 m. The shutoff head of the two-pump system is no better than that of Pump 2 alone, since Pump 1 is shut off and blocked at net heads above 9.25 m.

Turbines

14-73C

Solution We are to discuss the meaning and purpose of draft tubes.

Analysis **A draft tube is a diffuser that also turns the flow downstream of a turbine. Its purpose is to turn the flow horizontally and recover some of the kinetic energy leaving the turbine runner. If the draft tube is not designed carefully, much of the kinetic energy leaving the runner would be wasted, reducing the overall efficiency of the turbine system**.

Discussion Students' answers should be in their own words.

14-74C

Solution We are to name and describe the two types of dynamic turbine.

Analysis There are two basic types of dynamic turbine – **impulse** and **reaction**. In an impulse turbine, fluid is sent through a nozzle so that most of its available mechanical energy is converted into kinetic energy. The high-speed jet then impinges on bucket-shaped vanes that transfer energy to the turbine shaft. In a reaction turbine, the fluid completely fills the casing, and the runner is rotated by momentum exchange due to pressure differences across the blades, rather than by kinetic energy impingement. Impulse turbines require a higher head, but can operate with a smaller volume flow rate. Reaction turbines can operate with much less head, but require higher volume flow rate.

Discussion Students' answers should be in their own words.

14-75C

Solution We are to discuss reverse swirl in reaction turbines.

Analysis **Reverse swirl is when the runner blades turn the flow so much that the swirl at the runner outlet is in the direction opposite to runner rotation**. A small amount of reverse swirl may be desirable so that more power is absorbed from the water. We can easily see this from the Euler turbomachine equation,

$$\dot{W}_{\text{shaft}} = \omega \mathrm{T}_{\text{shaft}} = \rho \omega \dot{V}\left(r_2 V_{2,\text{t}} - r_1 V_{1,\text{t}}\right) \qquad (1)$$

Namely, since there is a negative sign on the last term, the shaft power increases if $V_{1,t}$ is *negative*, in other words, if there is reverse swirl at the runner outlet. If there is too much reverse swirl, a lot of extra kinetic energy gets wasted downstream of the runner.

Discussion A well designed draft tube can recover a good portion of the streamwise kinetic energy of the water leaving the runner. However, the swirling kinetic energy cannot be recovered.

14-76C

Solution We are to discuss why turbines have higher efficiencies than pumps.

Analysis There are several reasons for this. First, **pumps normally operate at higher rotational speeds than do turbines; therefore, shear stresses and frictional losses are higher**. Second, **conversion of kinetic energy into flow energy (pumps) has inherently higher losses than does the reverse (turbines)**. You can think of it this way: Since pressure *rises* across a pump (adverse pressure gradient), but *drops* across a turbine (favorable pressure gradient), boundary layers are less likely to separate in a turbine than in a pump. Third, **turbines (especially hydroturbines) are often much larger than pumps, and viscous losses become less important as size increases**. Finally, while pumps often operate over a wide range of flow rates, **most electricity-generating turbines run within a narrower operating range and at a controlled constant speed**; they can therefore be designed to operate very efficiently at those conditions.

Discussion Students' answers should be in their own words.

14-77C

Solution We are to discuss the classification of dynamic pumps and reaction turbines.

Analysis **Dynamic pumps are classified according to the angle at which the flow *exits* the impeller blade** – centrifugal, mixed-flow, or axial. **Reaction turbines, on the other hand, are classified according to the angle that the flow *enters* the runner** – radial, mixed-flow, or axial. This is the main difference between how dynamic pumps and reaction turbines are classified.

Discussion Students' answers should be in their own words.

14-78

Solution We are to estimate the power production from a hydroelectric plant.

Properties The density of water at T = 25°C is 997.0 kg/m^3.

Analysis The ideal power produced by one hydroturbine is

$$\dot{W}_{\text{ideal}} = \rho g \dot{V} H_{\text{gross}}$$

$$= \left(997.0\ \text{kg/m}^3\right)\left(9.807\ \text{m/s}^2\right)\left(13.6\ \text{m}^3\text{/s}\right)\left(284\ \text{m}\right)\times\left(\frac{\text{N}}{\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{\text{W}}{\text{N}\cdot\text{m/s}}\right)\left(\frac{1\ \text{MW}}{10^6\ \text{W}}\right) = 37.765\ \text{MW}$$

But inefficiencies in the turbine, the generator, and the rest of the system reduce the actual electrical power output. For each turbine,

$$\dot{W}_{\text{electrical}} = \dot{W}_{\text{ideal}}\eta_{\text{turbine}}\eta_{\text{generator}}\eta_{\text{other}} = \left(37.765\ \text{MW}\right)\left(0.959\right)\left(0.942\right)\left(0.956\right) = 32.615\ \text{MW}$$

Finally, since there are 14 turbines in parallel, the total power produced is

$$\dot{W}_{\text{total, electrical}} = 14\dot{W}_{\text{electrical}} = 14\left(32.615\ \text{MW}\right) = 456.61\ \text{MW} \cong \mathbf{457\ MW}$$

Discussion A small improvement in any of the efficiencies ends up increasing the power output, and increases the power company's profitability.

14-79

Solution We are to calculate several performance parameters for a Pelton wheel turbine.

Assumptions **1** Frictional losses are negligible, so that the Euler turbomachine equation applies, and the relative exit speed of the jet is the same as its relative inlet speed. **2** The turning angle is sufficient to prevent the exiting fluid from striking the next bucket. **3** The water is at 20°C.

Properties The density of water at T = 20°C is 998.0 kg/m³.

Analysis (*a*) The volume flow rate of the jet is equal to jet area times jet velocity,

$$\dot{V} = V_j \pi D_j^2/4 = (102 \text{ m/s})\pi(0.100 \text{ m})^2/4 = \mathbf{0.801\ m^3/s}$$

(*b*) The maximum output shaft power occurs when the bucket moves at half the jet speed ($\omega r = V_j/2$). Thus,

$$\omega = \frac{V_j}{2r} = \frac{102 \text{ m/s}}{2(1.83 \text{ m})} = 27.87 \text{ rad/s}$$

$$\rightarrow \quad \dot{n} = (27.87 \text{ rad/s})\left(\frac{\text{rot}}{2\pi \text{ rad}}\right)\left(\frac{60 \text{ s}}{\text{min}}\right) = \mathbf{266\ rpm}$$

(*c*) The ideal shaft power is found from Eq. 14-40,

$$\dot{W}_{\text{ideal}} = \rho\omega r\dot{V}(V_j - \omega r)(1-\cos\beta)$$

$$= (998.0 \text{ kg/m}^3)\left(27.87\frac{\text{rad}}{\text{s}}\right)(1.83 \text{ m})\left(0.801\frac{\text{m}^3}{\text{s}}\right)\left(\frac{102 \text{ m/s}}{2}\right)(1-\cos 165^\circ)$$

$$\times\left(\frac{\text{N}\cdot\text{s}^2}{\text{kg}\cdot\text{m}}\right)\left(\frac{\text{W}\cdot\text{s}}{\text{N}\cdot\text{m}}\right)\left(\frac{1 \text{ MW}}{10^6 \text{ W}}\right) = 4.09 \text{ MW}$$

where we have substituted $\omega r = V_j/2$ for convenience in the calculations. Since the turbine efficiency is given, we calculate the actual output shaft power, or brake horsepower,

$$\dot{W}_{\text{actual}} = bhp = \dot{W}_{\text{ideal}}\eta_{\text{turbine}} = (4.09 \text{ MW})(0.82) = \mathbf{3.35\ MW}$$

Discussion At other rotation speeds, the turbine would not be as efficient.

14-80

Solution We are to estimate the ideal power production from a hydroturbine.

Assumptions **1** Frictional losses are negligible. **2** The water is at 20°C.

Properties The density of water at T = 20°C is 998.0 kg/m³.

Analysis The ideal power produced by a hydroturbine is

$$\dot{W}_{\text{ideal}} = \rho g\dot{V}H_{\text{gross}} = (998.0 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.95 \text{ m}^3/\text{s})(340 \text{ m})\left(\frac{\text{N}\cdot\text{s}^2}{\text{kg}\cdot\text{m}}\right)\left(\frac{\text{W}\cdot\text{s}}{\text{N}\cdot\text{m}}\right)\left(\frac{1 \text{MW}}{10^6 \text{ W}}\right) = \mathbf{3.16 MW}$$

Discussion If a hydroelectric dam were to be built on this site, the actual power output per turbine would be smaller than this, of course, due to inefficiencies.

14-81

Solution We are to prove that the maximum power of a Pelton wheel occurs when $\omega r = V_j/2$ (bucket moving at half the jet speed).

Assumptions **1** Frictional losses are negligible, so that the Euler turbomachine equation applies, and the relative exit speed of the jet is the same as its relative inlet speed. **2** The turning angle is sufficient to prevent the exiting fluid from striking the next bucket. **3** The fluid, jet speed, volume flow rate, turning angle, and wheel radius are fixed – rotation rate is the only variable about which we are concerned.

Analysis With the stated assumptions, Eq. 14-40 applies, namely,

Output shaft power:
$$\dot{W}_{\text{shaft}} = \rho\omega r\dot{V}\left(V_j - \omega r\right)\left(1-\cos\beta\right) \qquad (1)$$

We differentiate Eq. 1 with respect to ω, and set the derivative equal to zero,

Maximum power:
$$\frac{d\dot{W}_{\text{shaft}}}{d\omega} = 0 \quad \rightarrow \quad \frac{d}{d\omega}\left(\omega V_j - \omega^2 r\right) = 0 \quad \rightarrow \quad V_j - 2\omega r = 0 \qquad (2)$$

Solution of Eq. 2 yields the desired result, namely, the maximum power of a Pelton wheel occurs when $\boxed{\boldsymbol{\omega r = V_j/2}}$ (bucket moving at half the jet speed).

Discussion To be sure that we have not identified a *minimum* instead of a maximum, we could substitute some numerical values and plot $\dot{W}_{\text{shaft}}$ versus ω. It turns out that we have indeed found the maximum value of $\dot{W}_{\text{shaft}}$.

14-82

Solution We are to estimate the power generated by a wind turbine.

Assumptions **1** The power coefficient is 0.42 and the combined gearbox/generator efficiency is 0.88.

Properties The air density is given as 1.204 kg/m^3.

Analysis (*a*) From the definition of power coefficient,

$$\dot{W}_{\text{rotor shaft output}} = C_P \tfrac{1}{2}\rho V^3 A = C_P \tfrac{1}{2}\rho V^3\left(\pi D^2/4\right)$$

But the actual electrical power produced is lower than this because of generator inefficiencies,

$$\begin{aligned}\dot{W}_{\text{electrical output}} &= \eta_{\text{gearbox/generator}}\frac{C_p \pi\rho V^3 D^2}{8}\\ &= (0.88)\frac{(0.42)\pi(1.204\text{ kg/m}^3)(7.8\text{ m/s})^3(45\text{ m})^2}{8}\left(\frac{\text{N}}{\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{\text{W}}{\text{N}\cdot\text{m/s}}\right)\\ &= 167{,}930\text{ W} \cong \mathbf{168\ kW}\end{aligned}$$

(*b*) If we could achieve the Betz limit, the shaft power output would increase by a factor of $C_{P,\text{ Betz}}/C_{P,\text{ real}}$. Assuming the same gearbox/generator efficiency,

$$\begin{aligned}\dot{W}_{\text{electrical output ideal}} &= \eta_{\text{gearbox/generator}}\frac{C_{p,\text{Betz}} \pi\rho V^3 D^2}{8}\\ &= (0.88)\frac{\left(\frac{16}{27}\right)\pi(1.204\text{ kg/m}^3)(7.8\text{ m/s})^3(45\text{ m})^2}{8}\left(\frac{\text{N}}{\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{\text{W}}{\text{N}\cdot\text{m/s}}\right)\\ &= 236{,}938\text{ W} \cong \mathbf{237\ kW}\end{aligned}$$

Discussion We give the final answers to three significant digits since we cannot expect any better than that. The Betz limit is unachievable in practice, so the solution of Part (*b*) is academic.

14-83

Solution We are to calculate runner blade angles, required net head, and power output for a Francis turbine.

Assumptions **1** The flow is steady. **2** The fluid is water at 20°C. **3** The blades are infinitesimally thin. **4** The flow is everywhere tangent to the runner blades. **5** We neglect irreversible losses through the turbine.

Properties For water at 20°C, ρ = 998.0 kg/m^3.

Analysis We solve for the normal component of velocity at the inlet,

$$V_{2,\mathrm{n}} = \frac{\dot{V}}{2\pi r_2 b_2} = \frac{340\ \mathrm{m^3/s}}{2\pi(2.00\ \mathrm{m})(0.731\ \mathrm{m})} = 37.0\ \mathrm{m/s} \tag{1}$$

Using the figure provided with this problem as a guide, the tangential velocity component at the inlet is

$$V_{2,\mathrm{t}} = V_{2,\mathrm{n}} \tan\alpha_2 = (37.0\ \mathrm{m/s})\tan 30^\mathrm{o} = 21.4\ \mathrm{m/s} \tag{2}$$

The angular velocity is $\omega = 2\pi\dot{n}/60$ = 18.85 rad/s.

The tangential velocity component of the absolute velocity at the inlet is obtained from trigonometry to be (see Eq. 14-45)

$$V_{2,\mathrm{t}} = \omega r_2 - \frac{V_{2,\mathrm{n}}}{\tan\beta_2}$$

From the above relationship, we solve for the runner leading edge angle β_2,

$$\beta_2 = \arctan\left[\frac{V_{2,\mathrm{n}}}{\omega r_2 - V_{2,\mathrm{t}}}\right] = \arctan\left[\frac{37.0\ \mathrm{m/s}}{(18.85\ \mathrm{rad/s})(2.00\ \mathrm{m}) - 21.4\ \mathrm{m/s}}\right] = \mathbf{66.2^o} \tag{3}$$

Equations 1 through 3 are repeated for the runner outlet, with the following results:

Runner outlet: $$V_{1,\mathrm{n}} = 17.3\ \mathrm{m/s},\quad V_{1,\mathrm{t}} = 3.05\ \mathrm{m/s},\quad \beta_1 = \mathbf{36.1^o} \tag{4}$$

Using Eqs. 2 and 4, the shaft output power is estimated from the Euler turbomachine equation,

$$\begin{aligned}\dot{W}_{\text{shaft}} &= \rho\omega\dot{V}\left(r_2V_{2,\mathrm{t}} - r_1V_{1,\mathrm{t}}\right) = \left(998.0\ \mathrm{kg/m^3}\right)(18.85\ \mathrm{rad/s})\left(340\ \mathrm{m^3/s}\right)\times\left[(2.00\ \mathrm{m})(21.4\ \mathrm{m/s}) - (1.42\ \mathrm{m})(3.05\ \mathrm{m/s})\right] \\ &= 2.46\times10^8\ \frac{\mathrm{kg\cdot m^2}}{\mathrm{s^3}}\left(\frac{\mathrm{MW\cdot s^3}}{10^6\ \mathrm{kg\cdot m^2}}\right) = 246.055\ \mathrm{MW} \cong \mathbf{246\ MW}\end{aligned} \tag{5}$$

Finally, we calculate the required net head using Eq. 14-44, assuming that η_{turbine} = 100% since we are ignoring irreversibilities,

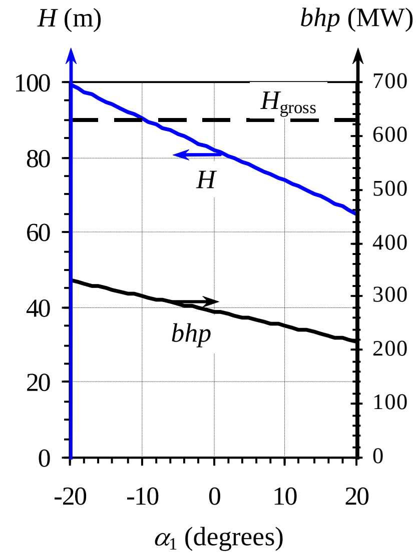
$$H = \frac{bhp}{\rho g\dot{V}} = \frac{246.055\ \mathrm{MW}}{\left(998.0\ \mathrm{kg/m^3}\right)\left(9.81\ \mathrm{m/s^2}\right)\left(340\ \mathrm{m^3/s}\right)}\left(\frac{10^6\ \mathrm{kg\cdot m^2}}{\mathrm{MW\cdot s^3}}\right) = \mathbf{73.9\ m} \tag{6}$$

Since the required net head is less than the gross net head, **the design is feasible**.

Discussion This is a preliminary design in which we are neglecting irreversibilities. Actual output power will be lower, and actual required net head will be higher than the values predicted here.

14-84

Solution We are to examine the effect of runner outlet angle α_1 on the required net head and the output power of a hydroturbine.



Assumptions **1** The flow is steady. **2** The fluid is water at 20°C. **3** The blades are infinitesimally thin. **4** The flow is everywhere tangent to the runner blades. **5** We neglect irreversible losses through the turbine.

Properties For water at 20°C, ρ = 998.0 kg/m^3.

Analysis We repeat the calculations of the previous problem, but with α_1 varying from -20° to 20°. The results are plotted. We see that the required net head H and the output brake horsepower *bhp* decrease as α_1 increases. This agrees with our expectations, based on the negative sign on the $V_{1,t}$ term in the Euler turbomachine equation. In other words, we can produce greater power by increasing the reverse swirl, but at the cost of increased required net head. However, when α_1 is smaller than about -9°, the required net head rises above H_{gross}, which is impossible, even with no irreversibilities. Thus, **when α_1 < -9°, the predicted net head and brake horsepower are not feasible – they violate the second law of thermodynamics**.

Discussion A small amount of reverse swirl is usually good, but too much reverse swirl is not good.

14-85

Solution We are to calculate flow angles, required net head, and power output for a Francis turbine.

Assumptions **1** The flow is steady. **2** The fluid is water at 20°C. **3** The blades are infinitesimally thin. **4** The flow is everywhere tangent to the runner blades. **5** We neglect irreversible losses through the turbine.

Properties For water at 20°C, ρ = 998.0 kg/m³.

Analysis The angular velocity is $\omega = 2\pi\dot{n}/60$ = 16.7552 rad/s. We solve for the normal component of velocity at the inlet,

$$V_{2,n} = \frac{\dot{V}}{2\pi r_2 b_2} = \frac{80.0\ \text{m}^3/\text{s}}{2\pi(2.00\ \text{m})(0.85\ \text{m})} = 7.4896\ \text{m/s} \tag{1}$$

The tangential velocity component of the absolute velocity at the inlet is obtained from trigonometry to be (see Eq. 14-45)

$$V_{2,t} = \omega r_2 - \frac{V_{2,n}}{\tan\beta_2} = (16.7552\ \text{rad/s})(2.00\ \text{m}) - \frac{7.4896\ \text{m/s}}{\tan 71.4°} = 30.990\ \text{m/s} \tag{2}$$

From these two components of V_2 in the absolute coordinate system, we calculate the angle α_2 through which the wicket gates should turn the flow,

$$\alpha_2 = \tan^{-1}\frac{V_{2,t}}{V_{2,n}} = \tan^{-1}\frac{30.990\ \text{m/s}}{7.4896\ \text{m/s}} = 76.413° \approx \mathbf{76.4°} \tag{3}$$

In exactly similar fashion, we solve for the velocity components and swirl angle at the runner outlet. We get

Runner outlet: $V_{1,n}$ = 4.6639 m/s, $V_{1,t}$ = 4.7334 m/s, α_1 = **45.4°** (4)

Using the tangential velocity components of Eqs. 2 and 4, the shaft output power is estimated from the Euler turbomachine equation,

$$\begin{aligned}\dot{W}_{\text{shaft}} &= \rho\omega\dot{V}(r_2V_{2,t} - r_1V_{1,t})\\ &= (998.0\ \text{kg/m}^3)(16.7552\ \text{rad/s})(80.0\ \text{m}^3/\text{s})\times\left[(2.00\ \text{m})(30.990\ \text{m/s}) - (1.30\ \text{m})(4.7334\ \text{m/s})\right]\\ &= 7.3775\times10^7\ \frac{\text{kg}\cdot\text{m}^2}{\text{s}^3}\left(\frac{\text{MW}\cdot\text{s}^3}{10^6\ \text{kg}\cdot\text{m}^2}\right) = 73.775\ \text{MW} \cong \mathbf{73.8\ MW}\end{aligned} \tag{5}$$

Finally, we calculate the required net head using Eq. 14-44, assuming that η_{turbine} = 100% since we are ignoring irreversibilities,

$$H = \frac{bhp}{\rho g\dot{V}} = \frac{73.775\ \text{MW}}{(998.0\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(80.0\ \text{m}^3/\text{s})}\left(\frac{10^6\ \text{kg}\cdot\text{m}^2}{\text{MW}\cdot\text{s}^3}\right) = 94.194\ \text{m} \cong \mathbf{94.2\ m} \tag{6}$$

Discussion This is a preliminary design in which we are neglecting irreversibilities. Actual output power will be lower, and actual required net head will be higher than the values predicted here. Notice the double negative in the outlet terms of Eq. 5 – reverse swirl leads to greater performance, but requires more head.

14-86E

Solution We are to calculate flow angles, required net head, and power output for a Francis turbine.

Assumptions **1** The flow is steady. **2** The fluid is water at 68°F. **3** The blades are infinitesimally thin. **4** The flow is everywhere tangent to the runner blades. **5** We neglect irreversible losses through the turbine.

Properties For water at 68°F, ρ = 62.32 lbm/ft³.

Analysis The angular velocity is $\omega = 2\pi\dot{n}/60$ = 12.57 rad/s. The volume flow rate is (4.70 × 10⁶ gpm)[ft³/s / (448.83 gpm)] = 10,470 ft³/s. We solve for the normal component of velocity at the inlet,

$$V_{2,\mathrm{n}} = \frac{\dot{V}}{2\pi r_2 b_2} = \frac{10,470\ \mathrm{ft^3/s}}{2\pi(6.60\ \mathrm{ft})(2.60\ \mathrm{ft})} = 97.1\ \mathrm{ft/s} \tag{1}$$

The tangential velocity component of the absolute velocity at the inlet is obtained from trigonometry to be (see Eq. 14-45)

$$V_{2,\mathrm{t}} = \omega r_2 - \frac{V_{2,\mathrm{n}}}{\tan\beta_2} = (12.57\ \mathrm{rad/s})(6.60\ \mathrm{ft}) - \frac{97.12\ \mathrm{ft/s}}{\tan 82^\circ} = 69.3\ \mathrm{ft/s} \tag{2}$$

From these two components of V_2 in the absolute coordinate system, we calculate the angle α_2 through which the wicket gates should turn the flow,

$$\alpha_2 = \tan^{-1}\frac{V_{2,\mathrm{t}}}{V_{2,\mathrm{n}}} = \tan^{-1}\frac{69.3\ \mathrm{ft/s}}{97.1\ \mathrm{ft/s}} = \mathbf{35.5^\circ} \tag{3}$$

In exactly similar fashion, we solve for the velocity components and swirl angle at the runner outlet. We get

Runner outlet: $$V_{1,\mathrm{n}} = 52.6\ \mathrm{ft/s},\quad V_{1,\mathrm{t}} = 4.49\ \mathrm{ft/s},\quad \alpha_1 = \mathbf{4.9^\circ} \tag{4}$$

Since α_1 is positive, **this turbine operates with a small amount of forward swirl**.

Using the tangential velocity components of Eqs. 2 and 4, the shaft output power is estimated from the Euler turbomachine equation,

$$\begin{aligned}\dot{W}_{\mathrm{shaft}} &= \rho\omega\dot{V}\left(r_2V_{2,\mathrm{t}} - r_1V_{1,\mathrm{t}}\right) = \left(62.32\ \mathrm{lbm/ft^3}\right)(12.57\ \mathrm{rad/s})\left(10,470\ \mathrm{ft^3/s}\right)\\ &\quad\times\left[(6.60\ \mathrm{ft})(69.3\ \mathrm{ft/s}) - (4.40\ \mathrm{ft})(4.49\ \mathrm{ft/s})\right]\left(\frac{\mathrm{lbf\cdot s^2}}{32.174\ \mathrm{lbm\cdot ft}}\right)\\ &= 1.116\times10^8\ \frac{\mathrm{ft\cdot lbf}}{\mathrm{s}}\left(\frac{1.8182\times10^{-3}\ \mathrm{hp\cdot s}}{\mathrm{ft\cdot lbf}}\right) = \mathbf{2.03\times10^5\ hp}\end{aligned} \tag{5}$$

Finally, we calculate the required net head using Eq. 14-44, assuming that η_{turbine} = 100% since we are ignoring irreversibilities,

$$H = \frac{bhp}{\rho g\dot{V}} = \frac{1.116\times10^8\ \mathrm{ft\cdot lbf/s}}{\left(62.32\ \mathrm{lbm/ft^3}\right)\left(32.174\ \mathrm{ft/s^2}\right)\left(10,470\ \mathrm{ft^3/s}\right)}\left(\frac{32.174\ \mathrm{lbm\cdot ft}}{\mathrm{lbf\cdot s^2}}\right) = \mathbf{171\,ft} \tag{6}$$

Discussion This is a preliminary design in which we are neglecting irreversibilities. Actual output power will be lower, and actual required net head will be higher than the values predicted here.

14-87E

Solution We are to calculate the runner blade trailing edge angle such that there is no swirl. At this value of β_1, we are to also calculate the shaft power.

Assumptions **1** The flow is steady. **2** The fluid is water at 68°F. **3** The blades are infinitesimally thin. **4** The flow is everywhere tangent to the runner blades. **5** We neglect irreversible losses through the turbine.

Properties For water at 68°F, ρ = 62.32 lbm/ft^3.

Analysis Using EES or other software, we adjust β_1 by trial and error until α_1 = 0. It turns out that β_1 = **43.6°**, at which $\dot{W}_{\text{shaft}}$ = $\mathbf{2.12 \times 10^5}$ **hp**.

Discussion It turns out that the swirl angle at the runner output is a strong function of β_1 – a small change in β_1 leads to a large change in α_1. The shaft power increases, as expected, since the original swirl angle was positive. The increase in shaft power is less than 5%.

14-88

Solution For given flow conditions and stator blade shape at a given radius, we are to design the rotor vane (bucket) for a single-stage turbine. Specifically, we are to calculate the leading and trailing edge angles of the rotor blade and sketch its shape.

Assumptions **1** The water is nearly incompressible. **2** The flow area between hub and tip is constant. **3** Two-dimensional blade row analysis is appropriate.

Analysis First we analyze flow through the stator from an absolute reference frame, using the two-dimensional approximation of a cascade (blade row) of stator blades as sketched in Fig. 1. Flow enters axially (horizontally), and is turned 50.3° upward. Since the axial component of velocity must remain constant to conserve mass, the magnitude of the velocity leaving the trailing edge of the stator, $\vec{V}_{st}$ is calculated,

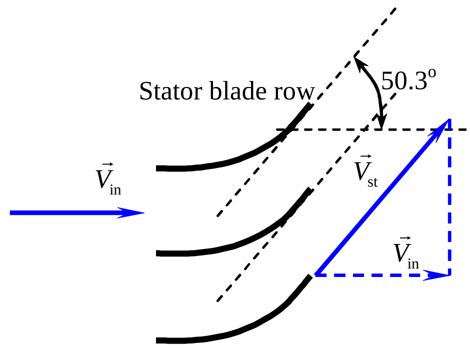


FIGURE 1
Analysis of the stator of an axial turbine as a two-dimensional cascade of stator blades; absolute reference frame.

$$V_{st} = \frac{V_{in}}{\cos\beta_{st}} = \frac{8.31 \text{ m/s}}{\cos\left(50.3^{\circ}\right)} = 13.009 \text{ m/s} \tag{1}$$

The direction of $\vec{V}_{st}$ is assumed to be that of the stator trailing edge. In other words we assume that the flow turns nicely through the blade row and exits parallel to the trailing edge of the blade, as shown in the sketch.

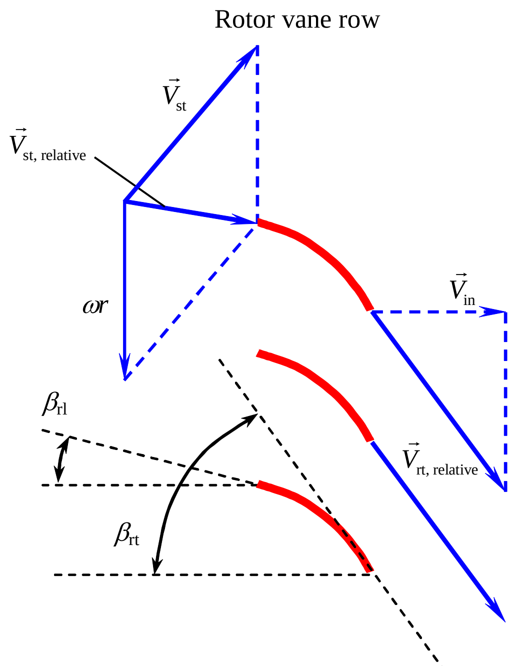


FIGURE 2
Analysis of the stator trailing edge velocity as it impinges on the rotor leading edge; relative reference frame.

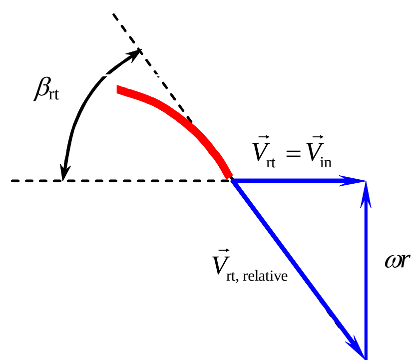


FIGURE 3
Analysis of the rotor trailing edge velocity; absolute reference frame.

We convert $\vec{V}_{st}$ to the *relative* reference frame moving with the rotor vanes. At a radius of 0.324 m, the tangential velocity of the rotor vanes is

$$u_\theta = \omega r = \left[\left(360 \text{ rot/min}\right) \left(\frac{2\pi \text{ rad}}{\text{rot}} \right) \left(\frac{1 \text{ min}}{60 \text{ s}} \right) \right] \left(0.324 \text{ m}\right) = 12.215 \text{ m/s} \tag{2}$$

Since the rotor blade row moves upward in the figure provided with this problem, we add a *downward* velocity with magnitude given by Eq. 2 to translate $\vec{V}_{\text{st}}$ into the rotating reference frame sketched in Fig. 2. The angle of the leading edge of the rotor, β_{rl}, can be calculated. After some trig,

$$\beta_{\text{rl}} = \arctan \frac{\omega r - V_{\text{in}} \tan \beta_{\text{st}}}{V_{\text{in}}} = \arctan \frac{12.215 \text{ m/s} - \left(8.31 \text{ m/s}\right) \tan\left(50.3^{\text{o}}\right)}{8.31 \text{ m/s}} = 14.86^{\text{o}} \tag{3}$$

The water must now be turned by the rotor vane row in such a way that it leaves the trailing edge of the rotor vane at zero angle (axially – no swirl) from an absolute reference frame. This determines the rotor's trailing edge angle, β_{rt}. Specifically, when we add an *upward* velocity of magnitude ωr (Eq. 2) to the relative velocity exiting the trailing edge of the rotor, $\vec{V}_{\text{rt, relative}}$, we convert back to the absolute reference frame, and obtain $\vec{V}_{\text{rt}}$, the velocity leaving the rotor trailing edge. It is this velocity, $\vec{V}_{\text{rt}}$, which must be axial (horizontal). Furthermore, to conserve mass $\vec{V}_{\text{rt}}$ must equal $\vec{V}_{\text{in}}$ since we are assuming incompressible flow. Working "backwards" we construct $\vec{V}_{\text{rt, relative}}$ in Fig. 3. Some trigonometry reveals that

$$\beta_{\text{rt}} = \arctan \frac{\omega r}{V_{\text{in}}} = \arctan \frac{12.215 \text{ m/s}}{8.31 \text{ m/s}} = 55.77^{\text{o}} \tag{4}$$

We conclude that the rotor blade at this radius has a leading edge angle of about **14.9$^{\text{o}}$** (Eq. 3) and a trailing edge angle of about **55.8$^{\text{o}}$** (Eq. 4). A sketch of the rotor vane is provided in Fig. 2; the vane is curved to turn the flow. Some turbine vanes are turned a lot more than this, hence the name "bucket".

Finally, to avoid interaction of the stator blade wakes with the rotor vane leading edges, we choose the number of rotor vanes such that it has no common denominator with the number of stator blades. Since there are 16 stator blades, we pick a number like **13**, **15**, or **17** rotor vanes. 14 or 18 would not be appropriate since they share a common denominator of 2 with the number 16.

Discussion We can easily repeat the calculation for all radii from hub to tip, completing the design of the entire rotor.

14-89

Solution We are to prove that a = 1/3 for maximum wind turbine efficiency.

Analysis From the wind turbine section, the power coefficient, which can also be thought of as an efficiency, is

$$C_P = \frac{\dot{W}_{\text{rotor shaft output}}}{\frac{1}{2}\rho V_1^3 A} = 4a(1-a)^2$$

To find the maximum (or minimum), we take the derivative of C_P and set $dC_P/da = 0$,

$$C_P = 4\left(a - 2a^2 + a^3\right) \quad \rightarrow \quad \frac{dC_P}{da} = 4\left(1 - 4a + 3a^2\right)$$

So, $dC_P/da = 0$ when

$$3a^2 - 4a + 1 = 0$$

We solve using the quadratic rule, yielding

$$a = \left(4 \pm \sqrt{16-12}\right)/6 = 1 \text{ or } 1/3$$

It turns out that a = 1 yields C_P = 0, which is a *minimum*. **The maximum is when a = 1/3, and yields C_P = 16/27**.

Discussion This is the famous Betz limit for wind turbines. No real wind turbine can outperform this limit.

14-90E

Solution We are to estimate the power production from a hydroelectric plant.

Properties The density of water at T = 50°F is 62.41 lbm/ft^3.

Analysis The ideal power produced by one hydroturbine is

$$\begin{aligned}\dot{W}_{\text{ideal}} &= \rho g \dot{V} H_{\text{gross}} \\ &= \left(62.41 \text{ lbm/ft}^3\right)\left(32.2 \text{ ft/s}^2\right)\left(189{,}400 \text{ gal/min}\right)\left(859 \text{ ft}\right)\times\left(\frac{\text{lbf}\cdot\text{s}^2}{32.2 \text{ lbm}\cdot\text{ft}}\right)\left(0.1337\frac{\text{ft}^3}{\text{gal}}\right)\left(\frac{1.356 \text{ W}}{\text{ft}\cdot\text{lbf/s}}\right)\left(\frac{1 \text{ min}}{60 \text{ s}}\right)\left(\frac{1 \text{ MW}}{10^6 \text{ W}}\right) \\ &= 30.681 \text{ MW}\end{aligned}$$

But inefficiencies in the turbine, the generator, and the rest of the system reduce the actual electrical power output. For each turbine,

$$\dot{W}_{\text{electrical}} = \dot{W}_{\text{ideal}}\eta_{\text{turbine}}\eta_{\text{generator}}\eta_{\text{other}} = \left(30.681 \text{ MW}\right)\left(0.963\right)\left(0.939\right)\left(1-0.036\right) = 26.745 \text{ MW}$$

Finally, since there are 10 turbines in parallel, the total power produced is

$$\dot{W}_{\text{total, electrical}} = 10\dot{W}_{\text{electrical}} = 10\left(26.745 \text{ MW}\right) = 267.45 \text{ MW} \cong \mathbf{267\ MW}$$

Discussion A small improvement in any of the efficiencies ends up increasing the power output, and increases the power company's profitability.

14-91

Solution We are to estimate the required disk diameter of a wind turbine and how many homes a wind farm can serve.

Assumptions **1** The power coefficient is 0.41 and the combined gearbox/generator efficiency is 0.928. The power distribution system has an efficiency of 96%.

Properties The air density is given as 1.2 kg/m^3.

Analysis (*a*) From the definition of power coefficient,

$$\dot{W}_{\text{rotor shaft output}} = C_P \tfrac{1}{2}\rho V^3 A = C_P \tfrac{1}{2}\rho V^3 \left(\pi D^2/4\right)$$

But the actual electrical power produced is lower than this because of gearbox and generator inefficiencies,

$$\dot{W}_{\text{electrical output}} = \eta_{\text{gearbox/generator}} \frac{C_P \pi \rho V^3 D^2}{8}$$

which we solve for diameter,

$$D = \sqrt{8\frac{\dot{W}_{\text{electrical output}}}{\eta_{\text{generator}} C_P \pi \rho V^3}} = \sqrt{8\frac{2.5\times10^6\ \text{W}\left(\frac{\text{N}\cdot\text{m/s}}{\text{W}}\right)\left(\frac{\text{kg}\cdot\text{m/s}^2}{\text{N}}\right)}{(0.41)(0.92)\pi\left(1.2\frac{\text{kg}}{\text{m}^3}\right)\left(12.5\frac{\text{m}}{\text{s}}\right)^3}}$$

$$= 84.86\ \text{m} \cong \mathbf{85\ m}$$

(*b*) For 30 machines, the total electrical power produced is 30(2.5 MW) = 75 MW. However, some of that is lost (wasted – turned into heat) due to inefficiencies in the power distribution system. The electrical power that actually makes it to peoples home is thus ($\eta_{\text{power distribution system}}$)(total power) = (0.96)(75 MW) = 72 MW. Since an average home consumes power at a rate of 1500 kW, the number of homes served by this wind farm is calculated as

$$\text{number of homes} = \frac{\left(\eta_{\text{power distribution system}}\right)\left(\text{number of turbines}\right)\left(\dot{W}_{\text{electrical output per turbine}}\right)}{\dot{W}_{\text{electrical usage per home}}}$$

$$= \frac{(0.96)(30\ \text{turbines})\left(2.5\times10^6\ \text{W/turbine}\right)}{1.5\times10^3\ \text{W/home}}$$

$$= 4.8\times10^4\ \text{homes} \cong \mathbf{48{,}000\ homes}$$

Discussion We give the final answers to two significant digits since we cannot expect any better than that. Wind farms of this size and larger are being constructed throughout the world.

Pump and Turbine Scaling Laws

14-92C

Solution We are to discuss the purpose of pump and turbine specific speeds.

Analysis Pump specific speed is used to characterize the operation of a pump at its optimum conditions (best efficiency point), and is useful for **preliminary pump selection**. Likewise, turbine specific speed is used to characterize the operation of a turbine at its optimum conditions (best efficiency point), and is useful for **preliminary turbine selection.**

Discussion Pump specific speed and turbine specific speed are parameters that can be calculated quickly. Based on the value obtained, one can quickly select the type of pump or turbine that is most appropriate for the given application.

14-93C

Solution

(*a*) ***True***: Rotation rate appears with an exponent of 1 in the affinity law for capacity. Thus, the change is linear.
(*b*) ***False***: Rotation rate appears with an exponent of 2 in the affinity law for net head. Thus, if the rpm is doubled, the net head increases by a factor of 4.
(*c*) ***False***: Rotation rate appears with an exponent of 3 in the affinity law for shaft power. Thus, if the rpm is doubled, the shaft power increases by a factor of 8.
(*d*) ***True***: The affinity laws apply to turbines as well as pumps, so this statement is true, as discussed in Part (c).

14-94C

Solution We are to discuss which pump and turbine performance parameters are used as the independent parameter, and explain why

Analysis For pumps, we use C_Q, the **capacity coefficient**, as the independent parameter. The reason is that the goal of a pump is to move fluid from one place to another, and the most important parameter is the pump's capacity (volume flow rate). On the other hand, for most turbines, we use C_P, the **power coefficient**, as the independent parameter. The reason is that the goal of a turbine is to rotate a shaft, and the most important parameter is the turbine's brake horsepower

Discussion There are exceptions. For example, when analyzing a positive-displacement turbine used to measure volume flow rate, capacity is more important than output shaft power, so one might use C_Q, instead of C_P as the independent parameter.

14-95C

Solution We are to give a definition of "affinity", and explain why the scaling laws are called "affinity laws".

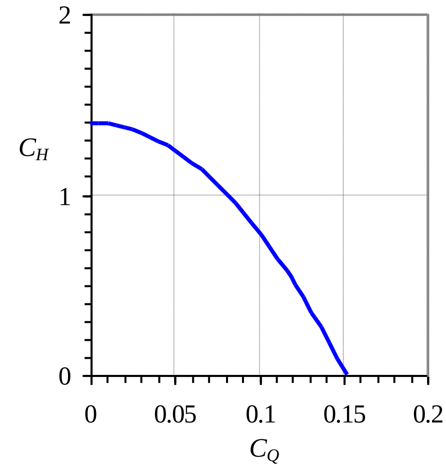
Analysis Among the many definitions of "affinity" is "**inherent likeness or agreement**", and "...**resemblance in general plan or structure**". When two pumps or two turbines are geometrically similar and operate under dynamically similar conditions, they indeed have "inherent likeness". Thus, **the phrase "affinity laws" is appropriate for the scaling laws of turbomachinery**.

Discussion Students will have various definitions, depending on the dictionary they use.

14-96

Solution We are to nondimensionalize a fan performance curve.

Analysis The fan's net head is approximated by the parabolic expression



Fan performance curve: $$H = H_0 - a\dot{V}^2 \quad (1)$$

where shutoff head H_0 = 60.0 mm of water column, coefficient $a = 2.50 \times 10^{-7}$ mm/Lpm^2, available fan head $H_{available}$ is in units of mm of water column, and capacity $\dot{V}$ is in units of liters per minute (Lpm). By definition, head coefficient $C_H = (gH)/(\omega^2 D^2)$, and capacity coefficient $C_Q = (\dot{V})/(\omega D^3)$. To convert, we must be careful with units. For example, we must convert the rotation rate from rpm to radians per second. The rotational speed becomes

$$\omega = 600\frac{\text{rot}}{\text{min}}\left(\frac{2\pi \text{ rad}}{\text{rot}}\right)\left(\frac{1 \text{ min}}{60 \text{ s}}\right) = 62.8319\frac{\text{rad}}{\text{s}}$$

Sample calculations at 13,600 Lpm are shown below.

Capacity coefficient at 13,600 *Lpm*: $$C_Q = \frac{\dot{V}}{\omega D^3} = \frac{13{,}600 \text{ L/min}}{(62.8319 \text{ rad/s})(0.300 \text{ m})^3}\left(\frac{1 \text{ m}^3}{1000 \text{ L}}\right)\left(\frac{1 \text{ min}}{60 \text{ s}}\right) = 0.13361 \cong 0.134$$

At this flow rate, the net head is obtained from Eq. 1,

$$H = 60.0 \text{ mm} - \left(2.50\times10^{-7} \text{ mm/Lpm}^2\right)(13{,}600 \text{ Lpm})^2 = 13.76 \text{ mm H}_2\text{O}$$

from which the head coefficient can be calculated,

Head coefficient at 13,600 *Lpm*: $$C_H = \frac{gH}{\omega^2 D^2} = \frac{\left(9.81 \text{ m/s}^2\right)(0.01376 \text{ m H}_2\text{O})}{(62.8319 \text{ rad/s})^2 (0.300 \text{ m})^2}\frac{998. \text{ kg/m}^3 (\text{m air})}{1.184 \text{ kg/m}^3 (\text{m H}_2\text{O})} = 0.32023 \cong 0.320$$

Note the ratio of water density to air density to convert the head from water column height to air column height. The calculations are repeated for a range of volume flow rates from 0 to $\dot{V}_{max}$. **The nondimensionalized pump performance curve is plotted**.

Discussion Since radians is a dimensionless unit, the units of C_H and C_Q are unity. In other words, C_H and C_Q are nondimensional parameters, as they must be.

14-97

Solution We are to calculate the specific speed of a fan, and determine what kind of fan it is.

Analysis First, we use the values of C_H and C_Q calculated in the previous problem at the BEP (to four significant digits) to calculate the dimensionless form of N_{Sp},

Dimensionless pump specific speed:
$$N_{Sp} = \frac{C_Q^{\ 1/2}}{C_H^{\ 3/4}} = \frac{0.13361^{1/2}}{0.32023^{3/4}} = 0.8587$$

Thus, $\boldsymbol{N_{Sp}}$ **= 0.859**, and is dimensionless. From the conversion given in the text, $\boldsymbol{N_{Sp,\mathrm{US}}}$ **= 0.8587 × 2,734 = 2350**. Alternatively, we can use the dimensional data to calculate $N_{Sp,\mathrm{US}}$, the dimensional pump specific speed in US units,

$$N_{Sp,\mathrm{US}} = \frac{(\dot{n},\ \mathrm{rpm})(\dot{V},\ \mathrm{gpm})^{1/2}}{(H,\ \mathrm{ft})^{3/4}} = \frac{(600\ \mathrm{rpm})\left(13{,}600\ \mathrm{Lpm}\left(\frac{0.2642\ \mathrm{gpm}}{\mathrm{Lpm}}\right)\right)^{1/2}}{\left(11.60\ \mathrm{m}\left(\frac{\mathrm{ft}}{0.3048\ \mathrm{m}}\right)\right)^{3/4}} = 2350$$

Note that we use H = 11.60 m *of air*, since air is the fluid being pumped here. We calculate H as $H_{\mathrm{air}} = H_{\mathrm{water}}(\rho_{\mathrm{water}}/\rho_{\mathrm{air}})$ = (0.01376 m of water)(998/1.184) = 11.60 m of air. From either the dimensionless or the dimensional pump specific speed, Fig. 14-73 shows that **this is most likely a centrifugal fan** (probably a squirrel cage fan).

Discussion We calculated the dimensional pump specific speed two ways as a check of our algebra, and (fortunately) the results agree.

14-98

Solution We are to calculate the specific speed of a water pump, and determine what kind of pump it is.

Analysis First, we use the values of C_H and C_Q calculated in Example 14-11 at the BEP to calculate the dimensionless form of N_{Sp},

Dimensionless pump specific speed:
$$N_{Sp} = \frac{C_Q^{\ 1/2}}{C_H^{\ 3/4}} = \frac{0.0112^{1/2}}{0.133^{3/4}} = 0.481$$

Thus, $\boldsymbol{N_{Sp}}$ **= 0.481**. From the conversion given in the text, $\boldsymbol{N_{Sp,\mathrm{US}}}$ **= 1310**. From Fig. 14-73, **this is most likely a centrifugal pump**.

Discussion From Fig. 14-73, the maximum efficiency one can expect from a centrifugal pump at this value of N_{Sp} is about 0.87 (87%). The actual pump efficiency is only 81.2% (from Example 14-11), so there is room for improvement in the design.

14-99

Solution We are to decide what king of pump should be designed for given performance criteria.

Analysis We calculate the pump specific speed at the given conditions,

$$N_{Sp} = \frac{\omega \dot{V}^{1/2}}{(gH)^{3/4}} = \frac{(1200\ \mathrm{rpm})\left(\frac{2\pi}{60}\ \frac{\mathrm{rad/s}}{\mathrm{rpm}}\right)\left\{(14.0\ \mathrm{L/min})\left(\frac{1\ \mathrm{m}^3}{1000\ \mathrm{L}}\right)\left(\frac{1\ \mathrm{min}}{60\ \mathrm{s}}\right)\right\}^{1/2}}{\left[(9.81\ \mathrm{m/s^2})(1.2\ \mathrm{m})\right]^{3/4}} = 0.302$$

From Fig. 14-73, we see that **Len should design a centrifugal pump**. **The maximum pump efficiency is about 0.81 (81%)** (based on Fig. 14-73 again).

Discussion This value of N_{Sp} is, in fact, on the low end of the curve for centrifugal pumps, so a centrifugal pump is the best Len can do, in spite of the low efficiency.

14-100

Solution We are to calculate the performance of a modified pump.

Assumptions **1** The modified pump operates at the best efficiency point. **2** Pump diameter and fluid properties remain the same.

Analysis At homologous points, the affinity laws are used to estimate the operating conditions of the modified pump. We let the original pump be Pump A, and the modified pump be Pump B. We get

Volume flow rate:

$$\dot{V}_{\mathrm{B}} = \dot{V}_{\mathrm{A}} \frac{\omega_{\mathrm{B}}}{\omega_{\mathrm{A}}} \left(\frac{D_{\mathrm{B}}}{D_{\mathrm{A}}} \right)^3 = (14.0\ \mathrm{L/min}) \frac{1800\ \mathrm{rpm}}{1200\ \mathrm{rpm}} (1)^3 = 21.0\ \mathrm{L/min}$$

and

Net head:

$$H_{\mathrm{B}} = H_{\mathrm{A}} \left(\frac{\omega_{\mathrm{B}}}{\omega_{\mathrm{A}}} \right)^2 \left(\frac{D_{\mathrm{B}}}{D_{\mathrm{A}}} \right) = (1.2\ \mathrm{m}) \left(\frac{1800\ \mathrm{rpm}}{1200\ \mathrm{rpm}} \right)^2 (1) = 2.70\ \mathrm{m}$$

The volume flow rate of the modified pump is 21.0 L/min; the net head is 2.70 m. The pump specific speed of the modified pump is

$$N_{Sp} = \frac{\omega \dot{V}^{1/2}}{(gH)^{3/4}} = \frac{(1800\ \mathrm{rpm}) \left(\frac{2\pi}{60} \frac{\mathrm{rad/s}}{\mathrm{rpm}} \right) \left\{ (21.0\ \mathrm{L/min}) \left(\frac{1\ \mathrm{m}^3}{1000\ \mathrm{L}} \right) \left(\frac{1\ \mathrm{min}}{60\ \mathrm{s}} \right) \right\}^{1/2}}{\left[(9.81\ \mathrm{m/s}^2)(2.70\ \mathrm{m}) \right]^{3/4}} = 0.302$$

Thus, $\boldsymbol{N_{Sp}}$ **= 0.302**, which is the same as that of the original pump. This is not surprising since the two pumps operate at homologous points.

Discussion When the rpm is increased by a factor of 1.5, all else being equal, the volume flow rate of the pump increases by a factor of $(1.5)^1$, while the net head increases by a factor of $(1.5)^2$. This agrees with the mnemonic given in the text: "Very Hard Problems are as easy as 1, 2, 3." We would expect the power to increase by a factor of $(1.5)^3$. The specific speed of the two pumps must match since they operate at homologous points.

14-101E

Solution We are to decide what king of pump should be designed for given performance criteria.

Properties The density of water at 77°F is 62.24 lbm/ft^3.

Analysis We calculate the pump specific speed at the given conditions,

$$N_{Sp} = \frac{\omega \dot{V}^{1/2}}{(gH)^{3/4}} = \frac{(300\text{ rpm})\left(\dfrac{2\pi}{60}\dfrac{\text{rad/s}}{\text{rpm}}\right)\left(2{,}500\text{ gal/min}\left(\dfrac{0.13368\text{ ft}^3}{1\text{ gal}}\right)\left(\dfrac{1\text{ min}}{60\text{ s}}\right)\right)^{1/2}}{\left((32.174\text{ ft/s}^2)(45\text{ ft})\right)^{3/4}} = 0.316$$

From Fig. 14-73, we choose **a centrifugal pump**. **The maximum pump efficiency is about 0.82 (82%)** (based on Fig. 14-73 again). From the definition of pump efficiency, $bhp = \eta_{\text{pump}}\rho\dot{V}gH$. Thus, the required brake horsepower is

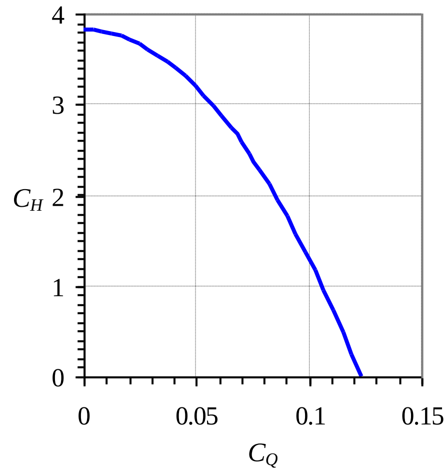
$$bhp = \frac{\rho g H \dot{V}}{\eta_{\text{pump}}} = \frac{(62.24\text{ lbm/ft}^3)(32.174\text{ ft/s}^2)(45\text{ ft})(5.57\text{ ft}^3\text{/s})}{0.82}\left(\frac{\text{lbf}\cdot\text{s}^2}{32.174\text{ lbm}\cdot\text{ft}}\right) = 19{,}025\text{ ft}\cdot\text{lbf/s}\left(\frac{1\text{ hp}}{550\text{ ft}\cdot\text{lbf/s}}\right) = 34.6\text{ hp}$$

Thus, we expect that **the pump will require about 35 hp to turn the shaft**.

Discussion Most large water pumps are of the centrifugal variety. This problem may also be solved in terms of the dimensional pump specific speed in customary US units: $N_{Sp,\text{US}} = 864$.

14-102

Solution We are to nondimensionalize a pump performance curve.



Analysis The pump's net head is approximated by the parabolic expression

Pump performance curve: $H = H_0 - a\dot{V}^2$ (1)

where shutoff head H_0 = 24.4 m of water column, coefficient a = 0.0678 m/Lpm^2, available pump head $H_{available}$ is in units of meters of water column, and capacity $\dot{V}$ is in units of liters per minute (Lpm). By definition, head coefficient $C_H = (gH)/(\omega^2 D^2)$, and capacity coefficient $C_Q = (\dot{V})/(\omega D^3)$. To convert, we must be careful with units. For example, we must convert the rotation rate from rpm to radians per second. The rotational speed is

$$\omega = 4200\frac{\text{rot}}{\text{min}}\left(\frac{2\pi\text{ rad}}{\text{rot}}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right) = 439.823\frac{\text{rad}}{\text{s}}$$

Sample calculations at 14.0 Lpm are shown below.

$$C_Q = \frac{\dot{V}}{\omega D^3} = \frac{14.0\text{ L/min}}{(439.823\text{ rad/s})(0.0180\text{ m})^3}\left(\frac{1\text{ m}^3}{1000\text{ L}}\right)\left(\frac{1\text{ min}}{60\text{ s}}\right) = 0.90966 \cong 0.0910$$

At this flow rate (14.0 Lpm), the net head is obtained from Eq. 1,

$$H = 24.4\text{ m} - \left(0.0678\text{ m/Lpm}^2\right)(14.0\text{ Lpm})^2 = 11.111\text{ m} \cong 11.1\text{ m}$$

from which the head coefficient can be calculated,

$$C_H = \frac{gH}{\omega^2 D^2} = \frac{\left(9.81\text{ m/s}^2\right)(11.111\text{ m})}{(439.823\text{ rad/s})^2(0.0180\text{ m})^2} = 1.7391 \cong 1.74$$

These calculations are repeated for a range of volume flow rates from 0 to $\dot{V}_{max}$. **The nondimensionalized pump performance curve is plotted**.

Discussion Since radians is a dimensionless unit, the units of C_H and C_Q are unity. In other words, C_H and C_Q are nondimensional parameters, as they must be.

14-103

Solution We are to calculate the specific speed of a water pump, and determine what kind of pump it is.

Analysis First, we use the values of C_H and C_Q calculated in the previous problem at the BEP to calculate the dimensionless form of N_{Sp},

Dimensionless pump specific speed:
$$N_{Sp} = \frac{C_Q^{\ 1/2}}{C_H^{\ 3/4}} = \frac{0.090966^{1/2}}{1.7391^{3/4}} = 0.1992$$

Thus, $\boldsymbol{N_{Sp}}$ **= 0.199**, and is dimensionless. From the conversion given in the text, $\boldsymbol{N_{Sp,\mathrm{US}}}$ **= 0.1992 × 2,734 = 545**. Alternatively, we can use the original dimensional data to calculate $N_{Sp,\mathrm{US}}$,

Dimensional pump specific speed in customary US units:
$$N_{Sp,\mathrm{US}} = \frac{(\dot{n},\ \mathrm{rpm})(\dot{V},\ \mathrm{gpm})^{1/2}}{(H,\ \mathrm{ft})^{3/4}} = \frac{(4200\ \mathrm{rpm})\left(14.0\ \mathrm{Lpm}\left(\frac{0.2642\ \mathrm{gpm}}{\mathrm{Lpm}}\right)\right)^{1/2}}{\left(11.111\ \mathrm{m}\left(\frac{\mathrm{ft}}{0.3048\ \mathrm{m}}\right)\right)^{3/4}} = 545$$

From either the dimensionless or the dimensional pump specific speed, Fig. 14-73 shows that **this is definitely a centrifugal pump**.

Discussion We calculated the dimensional pump specific speed two ways as a check of our algebra, and (fortunately) the results agree.

14-104

Solution We are to prove the relationship between turbine specific speed and pump specific speed.

Assumptions **1** The parameters such as net head, volume flow rate, diameter, etc. are defined in similar fashion for the pump and the turbine.

Analysis First we write the definitions of pump specific speed and turbine specific speed,

Pump specific speed:
$$N_{Sp} = \frac{\omega \dot{V}^{1/2}}{(gH)^{3/4}} \qquad (1)$$

and

Turbine specific speed:
$$N_{St} = \frac{\omega (bhp)^{1/2}}{(\rho)^{1/2}(gH)^{5/4}} \qquad (2)$$

After some rearrangement of Eq. 2,

$$N_{St} = \frac{\omega \dot{V}^{1/2}}{(gH)^{3/4}}\left(\frac{bhp}{\rho g H \dot{V}}\right)^{1/2} \qquad (3)$$

We recognize the first grouping of terms in Eq. 3 as N_{Sp} and the second grouping of terms as the square root of turbine efficiency η_{turbine}. Thus,

Final relationship:
$$\boxed{N_{St} = N_{Sp}\sqrt{\eta_{\mathrm{turbine}}}} \qquad (4)$$

Discussion Since turbine efficiency is typically large (90 to 95% for large hydroturbines), pump specific speed and turbine specific speed are nearly equivalent. Note that Eq. 4 does *not* apply to a pump running backwards as a turbine or vice-versa. Such devices, called *pump-turbines*, are addressed in the next problem.

14-105

Solution We are to prove the relationship between turbine specific speed and pump specific speed for the case of a pump-turbine operating at the same volume flow rate and rotational speed when acting as a pump and as a turbine.

Assumptions **1** The parameters such as net head, volume flow rate, diameter, etc. are defined in similar fashion for the pump and the turbine.

Analysis First we write the definitions of pump specific speed and turbine specific speed,

Pump specific speed:
$$N_{Sp} = \frac{\omega \dot{V}^{1/2}}{\left(gH_{\text{pump}}\right)^{3/4}} \tag{1}$$

and

Turbine specific speed:
$$N_{St} = \frac{\omega \left(bhp_{\text{turbine}}\right)^{1/2}}{(\rho)^{1/2}\left(gH_{\text{turbine}}\right)^{5/4}} \tag{2}$$

Note that we have added subscripts "pump" and "turbine" on net head and brake horsepower since $H_{\text{pump}} \neq H_{\text{turbine}}$ and $bhp_{\text{pump}} \neq bhp_{\text{turbine}}$. After some rearrangement of Eq. 2,

$$N_{St} = \frac{\omega \dot{V}^{1/2}}{\left(gH_{\text{turbine}}\right)^{3/4}}\left(\frac{bhp_{\text{turbine}}}{\rho g H_{\text{turbine}}\dot{V}}\right)^{1/2} = \frac{\omega \dot{V}^{1/2}}{\left(gH_{\text{turbine}}\right)^{3/4}}\left(\eta_{\text{turbine}}\right)^{1/2} \tag{3}$$

We also write the definitions of pump efficiency and turbine efficiency,

$$\eta_{\text{pump}} = \frac{\rho g \dot{V} H_{\text{pump}}}{bhp_{\text{pump}}} \qquad \eta_{\text{turbine}} = \frac{bhp_{\text{turbine}}}{\rho g \dot{V} H_{\text{turbine}}} \tag{4}$$

We solve both parts of Eq. 4 for $\dot{V}$ and equate the two, since $\dot{V}$ is the same whether the pump-turbine is operating as a pump or as a turbine. Eliminating $\dot{V}$,

$$H_{\text{turbine}} = \frac{H_{\text{pump}} bhp_{\text{turbine}}}{bhp_{\text{pump}}\eta_{\text{pump}}\eta_{\text{turbine}}} \tag{5}$$

where ρ and g have also dropped out since they are the same for both cases. We plug Eq. 5 into Eq. 3 and rearrange,

$$N_{St} = \frac{\omega \dot{V}^{1/2}}{\left(gH_{\text{pump}}\right)^{3/4}}\left(\eta_{\text{turbine}}\right)^{1/2}\left(\frac{bhp_{\text{pump}}\eta_{\text{pump}}\eta_{\text{turbine}}}{bhp_{\text{turbine}}}\right)^{3/4} \tag{6}$$

We recognize the first grouping of terms in Eq. 6 as N_{Sp} and rearrange,

Final relationship:
$$\boxed{N_{St} = N_{Sp}\left(\eta_{\text{turbine}}\right)^{5/4}\left(\eta_{\text{pump}}\right)^{3/4}\left(\frac{bhp_{\text{pump}}}{bhp_{\text{turbine}}}\right)^{3/4}} \tag{7}$$

An alternate expression is obtained in terms of net heads by substitution of Eq. 5,

Alternate final relationship:
$$\boxed{N_{St} = N_{Sp}\sqrt{\eta_{\text{turbine}}}\left(\frac{H_{\text{pump}}}{H_{\text{turbine}}}\right)^{3/4}} \tag{8}$$

Discussion It is difficult to achieve high efficiency in a pump-turbine during both the pump duty cycle *and* the turbine duty cycle. To achieve the highest possible efficiency, it is critical that both N_{Sp} and N_{St} are appropriate for the chosen design, e.g. radial flow centrifugal pump and radial-flow Francis turbine.

14-106

Solution We are to apply conversion factors to prove a conversion factor.

Properties We set g = 32.174 ft/s^2 and assume water at density ρ = 1.94 slug/ft^3.

Analysis We convert $N_{St,\text{US}}$ to N_{St} by dividing by $g^{5/4}$ and $\rho^{1/2}$, and then using conversion ratios to cancel all units,

$$N_{St} = \underbrace{\frac{(\omega,\ \text{rot/min})(bhp,\ \text{hp})^{1/2}}{(H,\ \text{ft})^{5/4}}}_{N_{St,\text{US}}} \frac{1}{\left(1.94\ \text{slug/ft}^3\right)^{1/2}\left(32.174\ \text{ft/s}^2\right)^{5/4}} \times \left(\frac{1\ \text{min}}{60\ \text{s}}\right)\left(\frac{\text{slug ft}}{\text{s}^2\ \text{lbf}}\right)^{1/2}\left(\frac{550\ \text{ft lbf}}{\text{s hp}}\right)^{1/2}\left(\frac{2\pi\ \text{rad}}{\text{s}}\right)$$

$$\boldsymbol{N_{St} = 0.02301 N_{St,\text{US}}}$$

Finally, the inverse of the above equation yields the desired conversion factor.

Discussion As discussed in the text, some turbomachinery authors do not convert rotations to radians, introducing a confusing factor of 2π into the conversion.

14-107

Solution We are to calculate the specific speed of a turbine, and compare it to the normal range.

Analysis We first calculate the nondimensional form of N_{St},

$$N_{St} = \frac{\omega (bhp)^{1/2}}{\rho^{1/2}(gH)^{5/4}} = \frac{(18.85\ \text{rad/s})\left(2.46\times10^8\ \text{W}\right)^{1/2}}{\left(998.0\ \text{kg/m}^3\right)^{1/2}\left[\left(9.81\ \text{m/s}^2\right)(73.8\ \text{m})\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right)^{1/2} = \mathbf{2.49}$$

From Fig. 14-108, this is much higher than the typical Francis turbine – **the designers should consider a Kaplan turbine instead**. From the conversion given in the text, $\boldsymbol{N_{St,\text{US}}}$ **= 2.49 × 43.46 = 108**. Alternatively, we can use the original dimensional data to calculate $N_{St,\text{US}}$,

$$N_{St,\text{US}} = \frac{(\dot{n},\ \text{rpm})(bhp,\ \text{hp})^{1/2}}{(H,\ \text{ft})^{5/4}} = \frac{(180\ \text{rpm})\left(3.29\times10^5\ \text{hp}\right)^{1/2}}{\left(73.8\ \text{m}\left(\frac{\text{ft}}{0.3048\ \text{m}}\right)\right)^{5/4}} = \mathbf{108}$$

Note that the actual values of brake horsepower and net head will differ from the values calculated here, because we have neglected irreversible losses; thus, the values of N_{St} and $N_{St,\text{US}}$ may change slightly.

Discussion We calculated the dimensional pump specific speed two ways as a check of our algebra, and the results agree.

14-108

Solution We are to calculate turbine specific speed for a given hydroturbine and compare to the guidelines of Fig. 14-116.

Assumptions **1** The fluid is water at 20°C.

Properties The density of water at 20°C is ρ = 998.0 kg/m³.

Analysis First we convert the turbine rotation rate from rpm to radians/s,

Rotational speed:
$$\omega = 100\frac{\text{rot}}{\text{min}}\left(\frac{2\pi\ \text{rad}}{\text{rot}}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = 10.47\ \text{rad/s}$$

With all other parameters already given in SI units, we calculate turbine specific speed,

$$N_{St} = \frac{\omega(bhp)^{1/2}}{(\rho)^{1/2}(gH)^{5/4}} = \frac{(10.47\ \text{rad/s})(194\times10^6\ \text{W})^{1/2}}{(998\ \text{kg/m}^3)^{1/2}\left[(9.81\ \text{m/s}^2)(54.9\ \text{m})\right]^{5/4}}\left(\frac{\text{m}^2\text{kg}}{\text{s}^3\cdot\text{W}}\right)^{1/2} = \mathbf{1.78}$$

Comparing to the turbine specific speed guidelines of Fig. 14-108, we see that this turbine should be of the Francis type, which it is.

Discussion The turbine specific speed of this turbine is close to the crossover point between a Francis turbine and a Kaplan turbine.

14-109

Solution We are to calculate turbine specific speed for a given hydroturbine and compare to the guidelines of Fig. 14-116.

Assumptions **1** The fluid is water at 20°C.

Properties The density of water at 20°C is ρ = 998.0 kg/m³.

Analysis First we convert the turbine rotation rate from rpm to radians/s,

Rotational speed:
$$\omega = 100\frac{\text{rot}}{\text{min}}\left(\frac{2\pi\ \text{rad}}{\text{rot}}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = 10.47\ \text{rad/s}$$

With all other parameters already given in SI units, we calculate turbine specific speed,

$$N_{St} = \frac{\omega(bhp)^{1/2}}{(\rho)^{1/2}(gH)^{5/4}} = \frac{(10.47\ \text{rad/s})(5.37\times10^6\ \text{W})^{1/2}}{(998\ \text{kg/m}^3)^{1/2}\left[(9.81\ \text{m/s}^2)(9.75\ \text{m})\right]^{5/4}}\left(\frac{\text{m}^2\text{kg}}{\text{s}^3\cdot\text{W}}\right)^{1/2} = \mathbf{2.57}$$

Comparing to the turbine specific speed guidelines of Fig. 14-108, we see that this turbine should be of the Kaplan type, which it is.

Discussion In fact, the turbine specific speed of this turbine is close to that which yields the maximum possible efficiency for a Kaplan turbine.

14-110

Solution We are to calculate the specific speed of a turbine, and compare it to the normal range.

Analysis We first calculate the nondimensional form of N_{St},

$$N_{St} = \frac{\omega (bhp)^{1/2}}{\rho^{1/2}(gH)^{5/4}} = \frac{(12.57 \text{ rad/s})(4.61\times10^8 \text{ W})^{1/2}}{(998.0 \text{ kg/m}^3)^{1/2}\left[(9.81 \text{ m/s}^2)(78.6 \text{ m})\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right)^{1/2} = \mathbf{2.10}$$

From Fig. 14-108, this is on the high end for Francis turbines – the designers may wish to consider a Kaplan turbine instead. From the conversion given in the text, $\boldsymbol{N_{St,\text{US}}}$ **= 2.10 × 43.46 = 91.4**. Alternatively, we can use the original dimensional data to calculate $N_{St,\text{US}}$,

$$N_{St,\text{US}} = \frac{(\dot{n}, \text{rpm})(bhp, \text{hp})^{1/2}}{(H, \text{ft})^{5/4}} = \frac{(120 \text{ rpm})(6.18\times10^5 \text{ hp})^{1/2}}{\left(78.6 \text{ m}\left(\frac{\text{ft}}{0.3048 \text{ m}}\right)\right)^{5/4}} = \mathbf{91.3}$$

Note that the actual values of brake horsepower and net head will differ from the values calculated here, because we have neglected irreversible losses; thus, the values of N_{St} and $N_{St,\text{US}}$ may change slightly.

Discussion We calculated the dimensional pump specific speed two ways as a check of our algebra, and (fortunately) the results agree (within round-off error).

14-111

Solution We are to calculate the specific speed of a turbine, and compare it to the normal range.

Analysis We first calculate the nondimensional form of N_{St},

$$N_{St} = \frac{\omega (bhp)^{1/2}}{\rho^{1/2}(gH)^{5/4}} = \frac{(12.5664 \text{ rad/s})(4.6302\times10^7 \text{ W})^{1/2}}{(998.0 \text{ kg/m}^3)^{1/2}\left[(9.81 \text{ m/s}^2)(59.117 \text{ m})\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right)^{1/2} = \mathbf{0.951}$$

From Fig. 14-108, **this is in the range of the typical Francis turbine**. From the conversion given in the text, $\boldsymbol{N_{St,\text{US}}}$ **= 0.951 × 43.46 = 41.3**. Alternatively, we can use the original dimensional data to calculate $N_{St,\text{US}}$,

$$N_{St,\text{US}} = \frac{(\dot{n}, \text{rpm})(bhp, \text{hp})^{1/2}}{(H, \text{ft})^{5/4}} = \frac{(120 \text{ rpm})(6.209\times10^4 \text{ hp})^{1/2}}{(193.95 \text{ ft})^{5/4}} = \mathbf{41.3}$$

Note that the actual values of brake horsepower and net head will differ from the values calculated here, because we have neglected irreversible losses; thus, the values of N_{St} and $N_{St,\text{US}}$ may change slightly.

Discussion We calculated the dimensional pump specific speed two ways as a check of our algebra, and the results agree.

14-112E

Solution We are to calculate the specific speed of a turbine, and compare it to the normal range.

Analysis We first calculate the specific speed in customary US units,

$$N_{St,\text{US}} = \frac{(\dot{n},\ \text{rpm})(bhp,\ \text{hp})^{1/2}}{(H,\ \text{ft})^{5/4}} = \frac{(120\ \text{rpm})(202{,}700\ \text{hp})^{1/2}}{(170.8\ \text{ft})^{5/4}} = \mathbf{87.5}$$

From Fig. 14-108, **this is on the high end of typical Francis turbines, on the border between Francis and Kaplan turbines**. In nondimensional terms, $\boldsymbol{N_{St,\text{US}} = 87.5 / 43.46 = 2.01}$. Note that the actual values of brake horsepower and net head will differ from the values calculated here, because we have neglected irreversible losses; thus, the values of N_{St} and $N_{St,\text{US}}$ may change slightly.

Discussion The actual values of brake horsepower and net head will differ from the values calculated here, because we have neglected irreversible losses; thus, the values of N_{St} and $N_{St,\text{US}}$ may change slightly.

14-113

Solution We are to calculate turbine specific speed for a given hydroturbine and compare to the guidelines of Fig. 14-116.

Assumptions **1** The fluid is water at 20°C.

Properties The density of water at 20°C is ρ = 998.0 kg/m^3.

Analysis First we convert the turbine rotation rate from rpm to radians/s,

Rotational speed:
$$\omega = 180\frac{\text{rot}}{\text{min}}\left(\frac{2\pi\ \text{rad}}{\text{rot}}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = 18.85\ \text{rad/s}$$

With all other parameters already given in SI units, we calculate turbine specific speed,

$$N_{St} = \frac{\omega(bhp)^{1/2}}{(\rho)^{1/2}(gH)^{5/4}} = \frac{(18.85\ \text{rad/s})(119\times10^6\ \text{W})^{1/2}}{(998\ \text{kg/m}^3)^{1/2}\left[(9.81\ \text{m/s}^2)(105\ \text{m})\right]^{5/4}}\left(\frac{\text{m}^2\text{kg}}{\text{s}^3\cdot\text{W}}\right)^{1/2} = \mathbf{1.12}$$

Comparing to the turbine specific speed guidelines of Fig. 14-108, we see that this turbine should be of the Francis type, which it is.

Discussion In fact, the turbine specific speed of this turbine is close to that which yields the maximum possible efficiency for a Francis turbine.

14-114

Solution We are to determine a turbine's efficiency and what kind of turbine is being tested.

Properties The density of water at T = 20°C is 998.0 kg/m³.

Analysis The turbine's efficiency is calculated first,

$$\eta_{\text{turbine}} = \frac{bhp}{\rho g H \dot{V}} = \frac{720\ \text{W}}{(998.0\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(15.0\ \text{m})(25.5\ \text{m}^3/\text{h})}\left(\frac{3600\ \text{s}}{\text{h}}\right)\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = 0.6922 \cong \mathbf{69.2\%}$$

After converting 1500 rpm to 157.1 rad/s, we calculate the nondimensional form of turbine specific speed,

$$N_{St} = \frac{\omega (bhp)^{1/2}}{\rho^{1/2}(gH)^{5/4}} = \frac{(157.1\ \text{rad/s})(720\ \text{W})^{1/2}}{(998.0\ \text{kg/m}^3)^{1/2}\left[(9.81\ \text{m/s}^2)(15.0\ \text{m})\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = 0.260$$

which we convert to customary US units, $N_{St,\text{US}}$ = 0.260 × 43.46 = 11.3. This is most likely an **impulse turbine** (e.g., a Pelton wheel turbine).

Discussion We could instead have used the dimensional units to directly calculate the turbine specific speed in customary US units. The result would be the same.

14-115

Solution We are to scale up the model turbine tests to the prototype turbine.

Assumptions **1** The prototype and model are geometrically similar. **2** The tests are conducted under conditions of dynamic similarity. **3** The water is at the same temperature for both the model and the prototype (20°C).

Properties The density of water at T = 20°C is 998.0 kg/m³.

Analysis We use the turbine scaling laws, starting with the head coefficient, and letting the model be turbine A and the prototype be turbine B,

$$C_{H,\text{A}} = \frac{gH_\text{A}}{\omega_\text{A}^2 D_\text{A}^2} = \frac{gH_\text{B}}{\omega_\text{B}^2 D_\text{B}^2} = C_{H,\text{B}} \quad \rightarrow \quad \omega_\text{B} = \omega_\text{A}\frac{D_\text{A}}{D_\text{B}}\sqrt{\frac{H_\text{B}}{H_\text{A}}} \quad \rightarrow \quad \omega_\text{B} = (1500\ \text{rpm})\frac{1}{5}\sqrt{\frac{50\ \text{m}}{15.0\ \text{m}}} = \mathbf{548\ rpm}$$

We then use the capacity coefficient to calculate the volume flow rate of the prototype,

$$C_{Q,A} = \frac{\dot{V}_\text{A}}{\omega_\text{A} D_A^3} = \frac{\dot{V}_\text{B}}{\omega_\text{B} D_B^3} = C_{Q,B} \rightarrow \dot{V}_\text{B} = \dot{V}_\text{A}\frac{\omega_\text{B}}{\omega_\text{A}}\left(\frac{D_\text{B}}{D_\text{A}}\right)^3 = (25.5\ \text{m}^3/\text{h})\frac{548\ \text{rpm}}{1500\ \text{rpm}}\left(\frac{5}{1}\right)^3 = 1164.5\ \text{m}^3/\text{h} \cong \mathbf{1160\ m^3/h}$$

Finally, we use the power coefficient to calculate the brake horsepower of the prototype,

$$C_{P,A} = \frac{bhp_\text{A}}{\omega_A^3 D_A^5} = \frac{bhp_\text{B}}{\omega_B^3 D_B^5} = C_{P,B} \rightarrow bhp_\text{B} = bhp_\text{A}\left(\frac{\omega_\text{B}}{\omega_\text{A}}\right)^3\left(\frac{D_\text{B}}{D_\text{A}}\right)^5 = (720\ \text{W})\left(\frac{548\ \text{rpm}}{1500\ \text{rpm}}\right)^3\left(\frac{5}{1}\right)^5 = 109{,}711\ \text{W} \cong \mathbf{110{,}000 W}$$

Discussion All results are given to 3 significant digits, but we kept several extra digits in the intermediate calculations to reduce round-off errors.

14-116

Solution We are to compare the model and prototype efficiency and turbine specific speed to prove that they operate at homologous points.

Properties The density of water at T = 20°C is 998.0 kg/m³.

Analysis The model turbine's efficiency was calculated in Problem 14-119 as 69.2%. We calculate the prototype turbine's efficiency as

$$\eta_{\text{turbine}} = \frac{bhp}{\rho g H \dot{V}} = \frac{109{,}711\,\text{W}}{(998.0\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(50\,\text{m})(1164.5\,\text{m}^3/\text{h})}\left(\frac{3600\,\text{s}}{\text{h}}\right)\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = 0.6929 \cong \mathbf{69.3\%}$$

The efficiency of the prototype is practically the same as that of the model. Similarly, the turbine specific speed of the model turbine was calculated previously as 0.260. After converting 548 rpm to 57.36 rad/s, we calculate the nondimensional form of turbine specific speed for the prototype turbine,

$$N_{St} = \frac{\omega (bhp)^{1/2}}{\rho^{1/2}(gH)^{5/4}} = \frac{(57.36\,\text{rad/s})(109{,}711\,\text{W})^{1/2}}{(998.0\,\text{kg/m}^3)^{1/2}\left[(9.81\,\text{m/s}^2)(50\,\text{m})\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = 0.261$$

which is also practically the same as the previous calculation. Comparing to the results of Problem 14-119, we see that both η_{turbine} and N_{st} match between the model and the prototype. Thus, **the model and the prototype operate at homologous points**.

Discussion Other nondimensional turbine parameters, like head coefficient, capacity coefficient, and power coefficient also match between model and prototype. We could instead have used dimensional units to calculate the turbine specific speed in customary US units.

14-117

Solution We are to estimate the actual efficiency of the prototype, and explain why it is higher than the model efficiency.

Analysis We apply the Moody efficiency correction equation,

$$\eta_{\text{turbine,prototype}} \approx 1-(1-\eta_{\text{turbine,model}})\left(\frac{D_{\text{model}}}{D_{\text{protottype}}}\right)^{1/5} = 1-(1-0.692)\left(\frac{1}{5}\right)^{1/5} = 0.7768 \cong \mathbf{77.7\%}$$

This represents an increase of 77.7 – 69.2 = 8.5%. However, as mentioned in the text, we expect only about 2/3 of this increase, or 2(8.5%)/3 = 5.7%. Thus, our best estimate of the actual efficiency of the prototype is

$$\eta_{\text{turbine,prototype}} \approx 62.9+5.7 = \mathbf{68.6\%}$$

There are several reasons for this increase in efficiency: The prototype turbine often operates at high Reynolds numbers that are not achievable in the laboratory. We know from the Moody chart that friction factor decreases with Re, as does boundary layer thickness. Hence, the influence of viscous boundary layers is less significant as turbine size increases, since the boundary layers occupy a less significant percentage of the flow path through the runner. In addition, the relative roughness (ε/D) on the surfaces of the prototype runner blades may be significantly smaller than that on the model turbine unless the model surfaces are micropolished. Finally, large full scale turbines have smaller tip clearances relative to blade diameter; therefore tip losses and leakage are less significant.

Discussion The increase in efficiency between the model and prototype is significant, and helps us to understand why very large hydroturbines can be extremely efficient devices.

Review Problems

14-118C

Solution We are to discuss the definition and usefulness of a pump-turbine.

Analysis A pump-turbine is **a turbomachine that can run both as a *pump* and as a *turbine*** (by running in the opposite direction). A pump-turbine is used by some power plants for **energy storage**; specifically, water is pumped by the pump-turbine during periods of low demand for power, and electricity is generated by the pump-turbine during periods of high demand for power.

Discussion We note that energy is "lost" both ways – when the pump-turbine is acting as a pump, and when it is acting as a turbine. Nevertheless, the energy storage scheme may still be cost-effective and profitable, in spite of the energy losses, because it may enable a power company to delay construction of costly new power-production facilities.

14-119C

Solution We are to discuss a water meter from the point of view of a piping system.

Analysis Although a water meter is a type of turbine, when analyzing pipe flow systems, we would think of the water meter as **a type of minor loss in the system**, just as a valve, elbow, etc. would be a minor loss, since there is a pressure drop associated with flow through the water meter.

Discussion In fact, manufacturers of water meters provide minor loss coefficients for their products.

14-120C

Solution

(*a*) ***True***: As the gears turn, they direct a closed volume of fluid from the inlet to the outlet of the gear pump.

(*b*) ***True*** or ***False***: Rotary pumps can be either positive displacement or dynamic (an unfortunate use of terminology). As a positive-displacement pump, the rotors direct a closed volume of fluid from the inlet to the outlet of the rotary pump. As a dynamic pump, "rotary pump" is sometimes used in place of the more correct term, "rotodynamic pump".

(*c*) ***True***: At a given rotational speed, the volume flow rate of a positive-displacement pump is fairly constant regardless of load because of the fixed closed volume.

(*d*) ***False***: Actually, the net head *increases* with fluid viscosity, because high viscosity fluids cannot penetrate the gaps as easily.

14-121

Solution We are to prove a relationship between two dynamically similar pumps, and discuss its application to turbines.

Assumptions **1** The two pumps are geometrically similar. **2** Reynolds number and roughness effects are not critical in the analysis (the two pumps are not grossly different from each other in size, capacity, etc.).

Analysis Since the two pumps are dynamically similar, dimensionless pump parameter C_H must be the same for both pumps,

$$C_{H,\text{A}} = \frac{gH_\text{A}}{\omega_\text{A}{}^2 D_\text{A}{}^2} = C_{\text{H,B}} = \frac{gH_\text{B}}{\omega_\text{B}{}^2 D_\text{B}{}^2} \quad \rightarrow \quad \frac{\omega_\text{A}}{\omega_\text{B}} = \sqrt{\frac{H_\text{A}}{H_\text{B}}}\frac{D_\text{B}}{D_\text{A}}$$

Similarly, dimensionless pump parameter C_Q must be the same for both pumps,

$$C_{Q,\text{A}} = \frac{\dot{V}_\text{A}}{\omega_\text{A} D_\text{A}{}^3} C_{Q,\text{B}} = \frac{\dot{V}_\text{B}}{\omega_\text{B} D_\text{B}{}^3} \quad \rightarrow \quad \left(\frac{D_\text{B}}{D_\text{A}}\right)^3 = \frac{\omega_\text{A}}{\omega_\text{B}}\frac{\dot{V}_\text{B}}{\dot{V}_\text{A}}$$

Combining the above two equations yields

$$\left(\frac{D_\text{B}}{D_\text{A}}\right)^3 = \sqrt{\frac{H_\text{A}}{H_\text{B}}}\frac{D_\text{B}}{D_\text{A}}\frac{\dot{V}_\text{B}}{\dot{V}_\text{A}}$$

which reduces to

$$\left(\frac{D_\text{B}}{D_\text{A}}\right)^2 = \sqrt{\frac{H_\text{A}}{H_\text{B}}}\frac{\dot{V}_\text{B}}{\dot{V}_\text{A}} \quad \rightarrow \quad \boxed{D_\text{B} = D_\text{A}\left(\frac{H_\text{A}}{H_\text{B}}\right)^{1/4}\left(\frac{\dot{V}_\text{B}}{\dot{V}_\text{A}}\right)^{1/2}}$$

Thus, we have eliminated the angular velocity as a parameter, and the relationship is proven.

Since turbines follow the same affinity laws as pumps, **the relationship also applies to two dynamically similar turbines**.

Discussion In like manner, we can eliminate other parameters through algebraic manipulations to scale a pump or turbine up or down.

14-122

Solution We are to prove a relationship between two dynamically similar turbines, and discuss its application to pumps.

Assumptions **1** The two turbines are geometrically similar. **2** Reynolds number and roughness effects are not critical in the analysis (the two turbines are not grossly different from each other in size, capacity, etc.).

Analysis Since the two turbines are dynamically similar, dimensionless turbine parameter C_H must be the same for both turbines,

$$C_{H,\mathrm{A}} = \frac{gH_\mathrm{A}}{{\omega_\mathrm{A}}^2{D_\mathrm{A}}^2} = C_{\mathrm{H,B}} = \frac{gH_\mathrm{B}}{{\omega_\mathrm{B}}^2{D_\mathrm{B}}^2} \quad \rightarrow \quad \frac{\omega_\mathrm{A}}{\omega_\mathrm{B}} = \sqrt{\frac{H_\mathrm{A}}{H_\mathrm{B}}}\frac{D_\mathrm{B}}{D_\mathrm{A}}$$

Similarly, dimensionless turbine parameter C_P must be the same for both turbines,

$$C_{P,\mathrm{A}} = \frac{bhp_\mathrm{A}}{\rho_\mathrm{A}{\omega_\mathrm{A}}^3{D_\mathrm{A}}^5} = C_{Q,\mathrm{B}} = \frac{bhp_\mathrm{B}}{\rho_\mathrm{B}{\omega_\mathrm{B}}^3{D_\mathrm{B}}^5} \quad \rightarrow \quad \left(\frac{D_\mathrm{B}}{D_\mathrm{A}}\right)^5 = \left(\frac{\omega_\mathrm{A}}{\omega_\mathrm{B}}\right)^3 \frac{\rho_\mathrm{A}}{\rho_\mathrm{B}}\frac{bhp_\mathrm{B}}{bhp_\mathrm{A}}$$

Combining the above two equations yields

$$\left(\frac{D_\mathrm{B}}{D_\mathrm{A}}\right)^5 = \left(\frac{H_\mathrm{A}}{H_\mathrm{B}}\right)^{3/2}\left(\frac{D_\mathrm{B}}{D_\mathrm{A}}\right)^3 \frac{\rho_\mathrm{A}}{\rho_\mathrm{B}}\frac{bhp_\mathrm{B}}{bhp_\mathrm{A}}$$

which reduces to

$$\left(\frac{D_\mathrm{B}}{D_\mathrm{A}}\right)^2 = \left(\frac{H_\mathrm{A}}{H_\mathrm{B}}\right)^{3/2} \frac{\rho_\mathrm{A}}{\rho_\mathrm{B}}\frac{bhp_\mathrm{B}}{bhp_\mathrm{A}} \quad \rightarrow \quad \boxed{D_\mathrm{B} = D_\mathrm{A}\left(\frac{H_\mathrm{A}}{H_\mathrm{B}}\right)^{3/4}\left(\frac{\rho_\mathrm{A}}{\rho_\mathrm{B}}\right)^{1/2}\left(\frac{bhp_\mathrm{B}}{bhp_\mathrm{A}}\right)^{1/2}}$$

Thus, we have eliminated the angular velocity as a parameter, and the relationship is proven.

Since pumps follow the same affinity laws as turbines, **the relationship also applies to two dynamically similar pumps**.

Discussion In like manner, we can eliminate other parameters through algebraic manipulations to scale a pump or turbine up or down.

14-123

Solution We are to design a new hydroturbine by scaling up an existing hydroturbine. Specifically we are to calculate the new turbine diameter, volume flow rate, and brake horsepower.

Assumptions **1** The new turbine is geometrically similar to the existing turbine. **2** Reynolds number effects and roughness effects are negligible. **3** The new penstock is also geometrically similar to the existing penstock so that the flow entering the new turbine (velocity profile, turbulence intensity, etc.) is similar to that of the existing turbine.

Properties The density of water at 20°C is ρ = 998.0 kg/m^3.

Analysis Since the new turbine (B) is dynamically similar to the existing turbine (A), we are concerned with only one particular homologous operating point of both turbines, namely the best efficiency point. We solve Eq. 14-38b for D_B,

$$D_B = D_A\sqrt{\frac{H_B}{H_A}}\frac{\dot{n}_A}{\dot{n}_B} = (1.50\text{ m})\sqrt{\frac{95\text{ m}}{90.0\text{ m}}}\left(\frac{150\text{ rpm}}{105\text{ rpm}}\right) = 2.2016 \cong \mathbf{2.20\text{ m}}$$

We then solve Eq. 14-38a for $\dot{V}_B$,

$$\dot{V}_B = \dot{V}_A\left(\frac{\dot{n}_B}{\dot{n}_A}\right)\left(\frac{D_B}{D_A}\right)^3 = (162\text{ m}^3\text{/s})\left(\frac{105\text{ rpm}}{150\text{ rpm}}\right)\left(\frac{2.2016\text{ m}}{1.50\text{ m}}\right)^3 = 358.5\text{ m}^3\text{/h} \cong \mathbf{359\text{ m}^3\text{/h}}$$

Finally, we solve Eq. 14-38c for bhp_B,

$$bhp_B = bhp_A\left(\frac{\rho_B}{\rho_A}\right)\left(\frac{\dot{n}_B}{\dot{n}_A}\right)^3\left(\frac{D_B}{D_A}\right)^5 = (132\text{ MW})(1)\left(\frac{105\text{ rpm}}{150\text{ rpm}}\right)^3\left(\frac{2.2016\text{ m}}{1.50\text{ m}}\right)^5 = 308.4\text{ MW} \cong \mathbf{308\text{ MW}}$$

Discussion To avoid round-off errors, we save several more significant digits for D_B than are given in the final answer.

14-124

Solution We are to compare the efficiency of two similar turbines, and discuss the Moody efficiency correction.

Analysis We calculate the turbine efficiency for both turbines,

$$\eta_{\text{turbine A}} = \frac{bhp_{\text{A}}}{\rho_{\text{A}} g H_{\text{A}} \dot{V}_{\text{A}}} = \frac{132\times10^{6}\ \text{W}}{\left(998.0\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)\left(90\ \text{m}\right)\left(162\ \text{m}^3/\text{s}\right)}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = \mathbf{92.5\%}$$

$$\eta_{\text{turbine,B}} = \frac{bhp_{\text{B}}}{\rho g H_{\text{B}} \dot{V}_{\text{B}}} = \frac{308.4\times10^{6}\ \text{W}}{(998.0\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(95\ \text{m})(358.5\ \text{m}^3/\text{h})}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = 0.9249 \cong \mathbf{92.5\%}$$

As expected, the two efficiencies are the same, since we have assumed dynamic similarity. However, total dynamic similarity may not actually be achieved between the two turbines because of scale effects (larger turbines generally have higher efficiency). The diameter of the new turbine is about 38% greater than that of the existing turbine, so the increase in efficiency due to turbine size should not be very significant. We verify this by using the Moody efficiency correction equation, considering turbine A as the "model" and B as the "prototype",

Efficiency correction:
$$\eta_{\text{turbine,B}} \approx 1-(1-\eta_{\text{turbine,A}})\left(\frac{D_{\text{A}}}{D_{\text{B}}}\right)^{1/5} = 1-(1-0.925)\left(\frac{1.50\ \text{m}}{2.20\ \text{m}}\right)^{1/5} = 0.9305 \cong \mathbf{93.0\%}$$

or **93.0%**. So, the first-order correction yields a predicted efficiency for the larger turbine that is about half of a percent greater than the smaller turbine. However, as mentioned in the text, we expect only about 2/3 of this increase, or 2(0.5%)/3 = 0.3%. Thus, our best estimate of the actual efficiency of the prototype is $\eta_{\text{turbine B}} \approx 92.5 + 0.3 = \mathbf{92.8\%}$.

Thus, our best estimate indicates that the new larger turbine will be slightly more efficient than its smaller brother, but the increase is only about 0.3%.

Discussion If the flow entering the new turbine from the penstock were not similar to that of the existing turbine (e.g., velocity profile and turbulence intensity), we could not expect exact dynamic similarity.

14-125

Solution The turbine specific speed of two dynamically similar turbines is to be compared, and the most likely type of turbine is to be determined.

Properties The density of water at T = 20°C is ρ = 998.0 kg/m^3.

Analysis We calculate the dimensionless turbine specific speed for turbines A and B,

$$N_{St,\mathrm{A}} = \frac{\omega_\mathrm{A}\left(bhp_\mathrm{A}\right)^{1/2}}{\left(\rho_\mathrm{A}\right)^{1/2}\left(gH_\mathrm{A}\right)^{5/4}} = \frac{\left(15.71\text{ rad/s}\right)\left(132\times10^6\text{ W}\right)^{1/2}}{\left(998.0\text{ kg/m}^3\right)^{1/2}\left[\left(9.81\text{ m/s}^2\right)\left(90.0\text{ m}\right)\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right)^{1/2} = \mathbf{1.19}$$

$$N_{St,B} = \frac{\omega_\mathrm{B}(bhp_\mathrm{B})^{1/2}}{(\rho_\mathrm{B})^{1/2}(gH_\mathrm{B})^{5/4}} = \frac{(11.00\text{ rad/s})(308.4\times10^6\text{ W})^{1/2}}{(998.0\text{ kg/m}^3)^{1/2}\left[(9.81\text{ m/s}^2)(95\text{ m})\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = \mathbf{1.19}$$

We see that the turbine specific speed of the two turbines is the same. In customary US units,

$$N_{St,\mathrm{US,A}} = N_{St,\mathrm{US,B}} = 43.46N_{St} = \left(43.46\right)\left(1.19\right) = \mathbf{51.6}$$

From Fig. 14-108, both of these turbines are most likely **Francis turbines**.

Discussion Since turbine A and turbine B operate at homologous points, it is no surprise that their turbine specific speeds are the same. In fact, if they weren't the same, it would be a sure sign of an algebraic or calculation error.

Fundamentals of Engineering (FE) Exam Problems

14-126

Which turbomachine is designed to deliver a very high pressure rise, typically at low to moderate flow rates?

(*a*) Compressor (*b*) Blower (*c*) Turbine (*d*) Pump (*e*) Fan

Answer (*a*) Compressor

14-127

In the turbomachinery industry, capacity refers to

(*a*) Power (*b*) Mass flow rate (*c*) Volume flow rate (*d*) Net head
(*e*) Energy grade line

Answer (*c*) Volume flow rate

14-128

A pump increases the pressure of water from 100 kPa to 3 MPa at a rate of 0.5 m^3/min. The inlet and outlet diameters are identical and there is no change in elevation across the pump. If the efficiency of the pump is 77 percent, the power supplied to the pump is

(*a*) 18.5 kW (*b*) 21.8 kW (*c*) 24.2 kW (*d*) 27.6 kW (*e*) 31.4 kW

Answer (*e*) 31.4 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P_in=100 [kPa]
P_out=3000 [kPa]
V_dot=0.5 [m^3/min]*Convert(m^3/min, m^3/s)
eta_pump=0.77
W_dot_useful=V_dot*(P_out-P_in)
W_dot_shaft=W_dot_useful/eta_pump
```

14-129

A pump increases the pressure of water from 100 kPa to 900 kPa to an elevation of 35 m. The inlet and outlet diameters are identical. The net head of the pump is

(*a*) 143 m (*b*) 117 m (*c*) 91 m (*d*) 70 m (*e*) 35 m

Answer (*b*) 117 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P_in=100000 [Pa]
P_out=900000 [Pa]
z_in=0 [m]
z_out=35 [m]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
H=P_out/(rho*g)+z_out-P_in/(rho*g)-z_in
```

14-130

The brake horsepower and water horsepower of a pump are determined to be 15 kW and 12 kW, respectively. If the flow rate of water to the pump under these conditions is 0.05 m^3/s, the total head loss of the pump is

(*a*) 11.5 m (*b*) 9.3 m (*c*) 7.7 m (*d*) 6.1 m (*e*) 4.9 m

Answer (*d*) 6.1 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W_dot_in=15000 [W]
W_dot_useful=12000 [W]
V_dot=0.05 [m^3/s]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
W_dot_loss=W_dot_in-W_dot_useful
H=W_dot_loss/(rho*g*V_dot)
```

14-131

In the pump performance curve, the point at which the net head is zero is called

(*a*) Best efficiency point (*b*) Free delivery (*c*) Shutoff head (*d*) Operating point

(*e*) Duty point

Answer (*b*) Free delivery

14-132

A power plant requires 940 L/min of water. The required net head is 5 m at this flow rate. An examination of pump performance curves indicates that two centrifugal pumps with different impeller diameters can deliver this flow rate. The pump with an impeller diameter of 203 mm has a pump efficiency of 73 percent and delivers 10 m of net head. The pump with an impeller diameter of 111 mm has a lower pump efficiency of 67 percent and but delivers 5 m of net head. What is the ratio of the required brake horse power (bhp) of the pump with 203-mm-diameter impeller to that of the pump with 111-mm-diameter impeller?

(*a*) 0.45 (*b*) 0.68 (*c*) 0.86 (*d*) 1.84 (*e*) 2.11

Answer (*d*) 1.84

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_dot=940 [L/min]*Convert(L/min, m^3/s)
H_required=5 [m]
H_1=10 [m]
H_2=5 [m]
eta_pump_1=0.73
eta_pump_2=0.67
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
bhp_1=rho*g*V_dot*H_1/eta_pump_1
bhp_2=rho*g*V_dot*H_2/eta_pump_2
bhp_Ratio=bhp_1/bhp_2
"or"
bhpRatio=(H_1/eta_pump_1)/(H_2/eta_pump_2)
```

14-133

Water enters the pump of a steam power plant at 20 kPa and 50°C at a rate of 0.15 m^3/s. The diameter of the pipe at the pump inlet is 0.25 m. What is the net positive suction head (NPSH) at the pump inlet?

(*a*) 2.14 m (*b*) 1.89 m (*c*) 1.66 m (*d*) 1.42 m (*e*) 1.26 m

Answer (*e*) 1.26 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=20000 [Pa]
T=50 [C]
V_dot=0.15 [m^3/s]
D=0.25 [m]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
A_c=pi*D^2/4
V=V_dot/A_c
P_v=pressure(steam_iapws, T=T, x=1)
NPSH=P/(rho*g)+V^2/(2*g)-P_v/(rho*g)
```

14-134

Which quantities are added when two pumps are connected in series and parallel?

(*a*) Series: Pressure change, Parallel: Net head
(*b*) Series: Net head, Parallel: Pressure change
(*c*) Series: Net head, Parallel: Flow rate
(*d*) Series: Flow rate, Parallel: Net head
(*e*) Series: Flow rate, Parallel: Pressure change

Answer (*c*) Series: Net head, Parallel: Flow rate

14-135

Three pumps are connected in series. According to pump performance curves, the free delivery of each pump is as follows:

Pump 1: 1600 L/min   Pump 2: 2200 L/min   Pump 3: 2800 L/min

If the flow rate for this pump system is 2500 L/min, which pump(s) should be shut off?

(*a*) Pump 1 (*b*) Pump 2 (*c*) Pump 3 (*d*) Pumps 1 and 2 (*e*) Pumps 2 and 3

Answer (*d*) Pumps 1 and 2

14-136

Three pumps are connected in parallel. According to pump performance curves, the shutoff head of each pump is as follows:

Pump 1: 7 m   Pump 2: 10 m   Pump 3: 15 m

If the net head for this pump system is 9 m, which pump(s) should be shut off?

(*a*) Pump 1 (*b*) Pump 2 (*c*) Pump 3 (*d*) Pumps 1 and 2 (*e*) Pumps 2 and 3

Answer (*a*) Pump 1

14-137

A two-lobe rotary positive-displacement pump moves 0.60 cm^3 of motor oil in each lobe volume. For every 90° of rotation of the shaft, one lobe volume is pumped. If the rotation rate is 550 rpm, the volume flow rate of oil is

(*a*) 330 cm^3/min (*b*) 660 cm^3/min (*c*) 1320 cm^3/min (*d*) 2640 cm^3/min

(*e*) 3550 cm^3/min

Answer (*c*) 1320 cm^3/min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_lobe=0.60 [cm^3]
n_dot=550 [1/min]
n=4 "Four lobe volume are pumped for one complete rotation (360 degrees)"
V_dot=n*n_dot*V_lobe
```

14-138

The snail-shaped casing of centrifugal pumps is called

(*a*) Rotor (*b*) Scroll (*c*) Volute (*d*) Impeller (*e*) Shroud

Answer (*b*) Scroll

14-139

A centrifugal blower rotates at 1400 rpm. Air enters the impeller normal to the blades (α_1 = 0°) and exits at an angle of 25° (α_2 = 25°). The inlet radius is r_1 = 6.5 cm, and the inlet blade width b_1 = 8.5 cm. The outlet radius and blade width are r_2 = 12 cm and b_2 = 4.5 cm. The volume flow rate is 0.22 m^3/s. What is the net head produced by this blower in meters of air?

(*a*) 12.3 m (*b*) 3.9 m (*c*) 8.8 m (*d*) 5.4 m (*e*) 16.4 m

Answer (*d*) 5.4 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_dot=1400 [1/min]
alpha_1=0 [degrees]
alpha_2=25 [degrees]
r1=0.065 [m]
b1=0.085 [m]
r2=0.12 [m]
b2=0.045 [m]
V_dot=0.22 [m^3/s]
g=9.81 [m/s^2]
omega=2*pi*n_dot*Convert(1/min, 1/s)
V_1_n=V_dot/(2*pi*r1*b1)
V_1_t=V_1_n*tan(alpha_1)
V_2_n=V_dot/(2*pi*r2*b2)
V_2_t=V_2_n*tan(alpha_2)
H=omega/g*(r2*V_2_t-r1*V_1_t)
```

14-140

A pump is designed to deliver 9500 L/min of water at a required head of 8 m. The pump shaft rotates at 1500 rpm. The pump specific speed in nondimensional form is

(*a*) 0.377 (*b*) 0.540 (*c*) 1.13 (*d*) 1.48 (*e*) 1.84

Answer (*a*) 0.377

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_dot=9500 [L/min]*Convert(L/min, m^3/s)
H=8 [m]
n_dot=1500 [1/min]*Convert(1/min, 1/s)
g=9.81 [m/s^2]
N_Sp=n_dot*V_dot^(1/2)/(g*H)^(3/4)
```

14-141

The net head delivered by a pump at a rotational speed of 1000 rpm is 10 m. If the rotational speed is doubled, the net head delivered will be

(*a*) 5 m (*b*) 10 m (*c*) 20 m (*d*) 40 m (*e*) 80 m

Answer (*d*) 40 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_dot_A=1500 [1/min]
H_A=10 [m]
n_dot_B=2*n_dot_A
H_B/H_A=(n_dot_B/n_dot_A)^2
```

14-142

The rotating part of a turbine is called

(*a*) Propeller (*b*) Scroll (*c*) Blade row (*d*) Impeller (*e*) Runner

Answer (*e*) Runner

14-143

Which one is correct for the comparison of the operation of impulse and reaction turbines?

(*a*) Impulse: Higher flow rate (*b*) Impulse: Higher head

(*c*) Reaction: Higher head (*d*) Reaction: Smaller flow rate

(*e*) None of these

Answer (*b*) Impulse: Higher head

14-144

Which turbine type is an impulse turbine?

(*a*) Kaplan (*b*) Francis (*c*) Pelton (*d*) Propeller (*e*) Centrifugal

Answer (*c*) Pelton

14-145

A turbine is placed at the bottom of a 20-m-high water body. Water flows through the turbine at a rate of 30 m^3/s. If the shaft power delivered by the turbine is 5 MW, the turbine efficiency is

(*a*) 85% (*b*) 79% (*c*) 88% (*d*) 74% (*e*) 82%

Answer (*a*) 85%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
H=20 [m]
V_dot=30 [m^3/s]
W_dot_shaft=5000 [kW]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
W_dot_ideal=rho*g*H*V_dot*Convert(W, kW)
eta_turbine=W_dot_shaft/W_dot_ideal*Convert(,%)
```

14-146

A hydroelectric power plant is to be built in a dam with a gross head of 200 m. The head losses in the head gate and penstock are estimated to be 6 m. The flow rate through the turbine is 18,000 L/min. The efficiencies of the turbine and the generator are 88 percent and 96 percent, respectively. The electricity production from this turbine is

(*a*) 6910 kW (*b*) 6750 kW (*c*) 6430 kW (*d*) 6170 kW (*e*) 5890 kW

Answer (*e*) 5890 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
H_gross=200 [m]
H_loss=6 [m]
V_dot=180000 [L/min]*Convert(L/min, m^3/s)
eta_turbine=0.88
eta_gen=0.96
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
W_dot_ideal=rho*g*H_gross*V_dot*Convert(W, kW)
eta_other=1-H_loss/H_gross
W_dot_electric=W_dot_ideal*eta_turbine*eta_gen*eta_other
```

14-147

In a hydroelectric power plant, water flows through a large tube through the dam is called

(*a*) Tailrace (*b*) Draft tube (*c*) Runner (*d*) Penstock (*e*) Propeller

Answer (*d*) Penstock

14-148

In wind turbines, the minimum wind speed at which useful power can be generated is called

(*a*) Rated speed (*b*) Cut-in speed (*c*) Cut-out speed (*d*) Available speed

(*e*) Betz speed

Answer (*b*) Cut-in speed

14-149

A wind turbine is installed in a location where the wind blows at 8 m/s. The air temperature is 10°C and the diameter of turbine blade is 30 m. If the overall turbine-generator efficiency is 35 percent, the electrical power production is

(*a*) 79 kW (*b*) 109 kW (*c*) 142 kW (*d*) 154 kW (*e*) 225 kW

Answer (*a*) 79 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=8 [m/s]
T=(10+273) [K]
D=30 [m]
eta_overall=0.35
P=101 [kPa]
R=0.287 [kJ/kg-K]
rho=P/(R*T)
A_c=pi*D^2/4
m_dot=rho*A_c*V
W_dot_available=m_dot*V^2/2*Convert(W, kW)
W_dot_electric=eta_overall*W_dot_available
```

14-150

The available power from a wind turbine is calculated to be 50 kW when the wind speed is 5 m/s. If the wind velocity is doubled, the available wind power becomes

(*a*) 50 kW (*b*) 100 kW (*c*) 200 kW (*d*) 400 kW (*e*) 800 kW

Answer (*d*) 400 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W_dot_1=50 [kW]
V1=5 [m/s]
V2=2*V1
W_dot_2/W_dot_1=(V2/V1)^3
```

14-151

A new hydraulic turbine is to be designed to be similar to an existing turbine with following parameters at its best efficiency point: D_A = 3 m, $\dot{n}_A$ = 90 rpm, $\dot{V}_A$ = 200 m^3/s, H_A = 55 m, bhp$_A$ = 100 MW. The new turbine will have a speed of 110 rpm and the net head will be 40 m. What is the bhp of the new turbine such that it operates most efficiently?

(*a*) 17.6 MW (*b*) 23.5 MW (*c*) 30.2 MW (*d*) 40.0 MW (*e*) 53.7 MW

Answer (*c*) 30.2 MW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D_A=3 [m]
n_dot_A=90 [1/min]
V_dot_A=200 [m^3/s]
H_A=55 [m]
bhp_A=100 [MW]
n_dot_B=110 [1/min]
H_B=40 [m]
rho_B\rho_A=1
D_B=D_A*sqrt(H_B/H_A)*(n_dot_A/n_dot_B)
bhp_B=bhp_A*(rho_B\rho_A)*(n_dot_B/n_dot_A)^3*(D_B/D_A)^5
```

14-152

A hydraulic turbine operates at the following parameters at its best efficiency point: $\dot{n}$ = 90 rpm, $\dot{V}$ = 200 m^3/s, H = 55 m, bhp = 100 MW. The turbine specific speed of this turbine is

(*a*) 0.71 (*b*) 0.18 (*c*) 1.57 (*d*) 2.32 (*e*) 1.15

Answer (*e*) 1.15

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_dot=90 [1/min]*Convert(1/min, 1/s)
V_dot=200 [m^3/s]
H=55 [m]
bhp=100000000 [W]
g=9.81 [m/s^2]
rho=1000 [kg/m^3]
omega=2*pi*n_dot
N_St=omega*bhp^(1/2)/(rho^(1/2)*(g*H)^(5/4))
```

Design and Essay Problems

14-153

Solution We are to generate a computer application that uses the affinity laws to design a new pump that is dynamically similar to an existing pump.

Assumptions **1** The two pumps are geometrically similar. **2** Reynolds number and roughness effects are not critical in the analysis (the two pumps are not grossly different from each other in size, capacity, etc.).

Analysis First, we calculate the brake horsepower for pump A,

$$bhp_{\mathrm{A}} = \frac{\rho_{\mathrm{A}}\dot{V}_{\mathrm{A}}gH_{\mathrm{A}}}{\eta_{\mathrm{pump,A}}} = \frac{\left(998.0\ \mathrm{kg/m^3}\right)\left(0.00040\ \mathrm{m^3/s}\right)\left(9.81\ \mathrm{m/s^2}\right)\left(1.20\ \mathrm{m}\right)}{0.81}\left(\frac{\mathrm{W\cdot s^3}}{\mathrm{kg\cdot m^2}}\right) = 5.80\ \mathrm{W}$$

We use the affinity laws to calculate the new pump parameters. Using the given data, we first calculate the new pump diameter,

$$D_{\mathrm{B}} = D_{\mathrm{A}}\left(\frac{H_{\mathrm{A}}}{H_{\mathrm{B}}}\right)^{1/4}\left(\frac{\dot{V}_{\mathrm{B}}}{\dot{V}_{\mathrm{A}}}\right)^{1/2} = \left(5.0\ \mathrm{cm}\right)\left(\frac{120\ \mathrm{cm}}{450\ \mathrm{cm}}\right)^{1/4}\left(\frac{2400\ \mathrm{cm^3/s}}{400\ \mathrm{cm^3/s}}\right)^{1/2} = \mathbf{8.80\ cm}$$

Knowing D_{B}, we calculate the rotation rate of the new pump,

$$\dot{n}_{\mathrm{B}} = \dot{n}_{\mathrm{A}}\frac{D_{\mathrm{A}}}{D_{\mathrm{B}}}\left(\frac{H_{\mathrm{B}}}{H_{\mathrm{A}}}\right)^{1/2} = \left(1725\ \mathrm{rpm}\right)\frac{5.0\ \mathrm{cm}}{8.80\ \mathrm{cm}}\left(\frac{450\ \mathrm{cm}}{120\ \mathrm{cm}}\right)^{1/2} = \mathbf{1898\ rpm}$$

Knowing D_{B} and ω_{B}, we now calculate the required shaft power,

$$bhp_{\mathrm{B}} = bhp_{\mathrm{A}}\frac{\rho_{\mathrm{B}}}{\rho_{\mathrm{A}}}\left(\frac{D_{\mathrm{B}}}{D_{\mathrm{A}}}\right)^5\left(\frac{\omega_{\mathrm{B}}}{\omega_{\mathrm{A}}}\right)^3 = \left(5.80\ \mathrm{W}\right)\frac{1226\ \mathrm{kg/m^3}}{998.0\ \mathrm{kg/m^3}}\left(\frac{8.80\ \mathrm{cm}}{5.0\ \mathrm{cm}}\right)^5\left(\frac{1898\ \mathrm{rpm}}{1725\ \mathrm{rpm}}\right)^3 = \mathbf{160\ W}$$

Our answers agree with those given. Now we are confident that we can apply our computer program to other design problems of similar nature, as in the next problem.

Discussion To avoid round-off errors in the calculations, we saved several more significant digits for D_{B} than are reported here.

14-154

Solution We are to use a computer code and the affinity laws to design a new pump that is dynamically similar to an existing pump.

Assumptions **1** The two pumps are geometrically similar. **2** Reynolds number and roughness effects are not critical in the analysis (the two pumps are not grossly different from each other in size, capacity, etc.). **3** Both pumps operate at their BEP.

Analysis Using our computer code, D_B = **12.8 cm**, $\dot{n}_B$ = **1924 rpm**, and bhp_B = **235 W**. We calculate pump specific speed for the new pump,

Pump B:

$$N_{Sp} = \frac{\omega \dot{V}^{1/2}}{(gH)^{3/4}} = \frac{(1924\text{ rpm})\left(\frac{2\pi}{60}\frac{\text{rad/s}}{\text{rpm}}\right)\left(0.00367\text{ m}^3\text{/s}\right)^{1/2}}{\left(\left(9.81\text{ m/s}^2\right)(5.70\text{ m})\right)^{3/4}} = \mathbf{0.597}$$

and we repeat the calculation for the existing pump,

Pump A:

$$N_{Sp} = \frac{\omega \dot{V}^{1/2}}{(gH)^{3/4}} = \frac{(1500\text{ rpm})\left(\frac{2\pi}{60}\frac{\text{rad/s}}{\text{rpm}}\right)\left(0.00135\text{ m}^3\text{/s}\right)^{1/2}}{\left(\left(9.81\text{ m/s}^2\right)(2.10\text{ m})\right)^{3/4}} = \mathbf{0.597}$$

Our answers agree, as they must, since the two pumps are operating at homologous points. From Fig. 14-73, we can see that these are most likely **centrifugal pumps**.

Discussion The pump performance parameters of pump B can be calculated manually to further verify our computer code.

14-155

Solution We are to generate a computer application that uses the affinity laws to design a new turbine that is dynamically similar to an existing turbine.

Assumptions **1** The two turbines are geometrically similar. **2** Reynolds number and roughness effects are not critical in the analysis (the two turbines are not grossly different from each other in size, capacity, etc.).

Analysis We use the affinity laws to calculate the new turbine parameters. Using the given data, we first calculate the new turbine diameter,

$$D_B = D_A\sqrt{\frac{H_B}{H_A}}\left(\frac{\dot{n}_A}{\dot{n}_B}\right) = (1.40\text{ m})\sqrt{\frac{95.0\text{ m}}{80.0\text{ m}}}\left(\frac{150\text{ rpm}}{120\text{ rpm}}\right) = \mathbf{1.91\ m}$$

We then solve Eq. 14-38a for $\dot{V}_B$,

$$\dot{V}_B = \dot{V}_A\left(\frac{\dot{n}_B}{\dot{n}_A}\right)\left(\frac{D_B}{D_A}\right)^3 = (162\text{ m}^3/\text{s})\left(\frac{120\text{ rpm}}{150\text{ rpm}}\right)\left(\frac{1.91\text{ m}}{1.40\text{ m}}\right)^3 = \mathbf{328\ m^3/s}$$

Finally, we solve Eq. 14-38c for bhp_B,

$$bhp_B = bhp_A\left(\frac{\rho_B}{\rho_A}\right)\left(\frac{\dot{n}_B}{\dot{n}_A}\right)^3\left(\frac{D_B}{D_A}\right)^5 = 118\text{ MW}\left(\frac{998.0\text{ kg/m}^3}{998.0\text{ kg/m}^3}\right)\left(\frac{120\text{ rpm}}{150\text{ rpm}}\right)^3\left(\frac{1.91\text{ m}}{1.40\text{ m}}\right)^5 = \mathbf{283\ MW}$$

Our answers agree with those given. Now we are confident that we can apply our computer program to other design problems of similar nature.

Discussion To avoid round-off errors in the calculations, we saved several more significant digits for D_B than are reported here.

14-156

Solution We are to use a computer program and the affinity laws to design a new turbine that is dynamically similar to an existing turbine.

Assumptions **1** The two pumps are geometrically similar. **2** Reynolds number and roughness effects are not critical in the analysis (the two pumps are not grossly different from each other in size, capacity, etc.).

Analysis Using our computer code, D_B = **2.04 m**, $\dot{V}_B$ = 815 m^3/s, and bhp_B = **577 MW**. We calculate turbine specific speed for the new turbine,

$$N_{St,A} = \frac{\omega_A (bhp_A)^{1/2}}{(\rho_A)^{1/2}(gH_A)^{5/4}} = \frac{(25.13 \text{ rad/s})(11.4\times10^6 \text{ W})^{1/2}}{(998.0 \text{ kg/m}^3)^{1/2}\left[(9.81 \text{ m/s}^2)(22.0 \text{ m})\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right)^{1/2} = \mathbf{3.25}$$

and for turbine B,

$$N_{St,B} = \frac{\omega_B (bhp_B)^{1/2}}{(\rho_B)^{1/2}(gH_B)^{5/4}} = \frac{(21.99 \text{ rad/s})(577\times10^6 \text{ W})^{1/2}}{(998.0 \text{ kg/m}^3)^{1/2}\left[(9.81 \text{ m/s}^2)(95.0 \text{ m})\right]^{5/4}}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right)^{1/2} = \mathbf{3.25}$$

Our answers agree, as they must, since the two turbines are operating at homologous points. From Fig. 14-108, we see that these are most likely **Kaplan turbines**.

Discussion The turbine parameters of turbine B can be calculated manually to further verify our computer code.

14-157

Solution We are to compare efficiencies for two geometrically similar turbines, and discuss the Moody efficiency correction.

Analysis We calculate the turbine efficiency for both turbines,

$$\eta_{\text{turbine A}} = \frac{bhp_{\text{A}}}{\rho_{\text{A}} g H_{\text{A}} \dot{V}_{\text{A}}} = \frac{11.4\times 10^6 \text{ W}}{\left(998.0 \text{ kg/m}^3\right)\left(9.81 \text{ m/s}^2\right)\left(22.0 \text{ m}\right)\left(69.5 \text{ m}^3\text{/s}\right)}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = \mathbf{76.2\%}$$

and

$$\eta_{\text{turbine B}} = \frac{bhp_{\text{B}}}{\rho_{\text{B}} g H_{\text{B}} \dot{V}_{\text{B}}} = \frac{577\times 10^6 \text{ W}}{\left(998.0 \text{ kg/m}^3\right)\left(9.81 \text{ m/s}^2\right)\left(95.0 \text{ m}\right)\left(814.6 \text{ m}^3\text{/s}\right)}\left(\frac{\text{kg}\cdot\text{m}^2}{\text{W}\cdot\text{s}^3}\right) = \mathbf{76.2\%}$$

where we have included an extra digit in the intermediate values to avoid round-off error. As expected, the two efficiencies are the same, since we have assumed dynamic similarity. However, total dynamic similarity may not actually be achieved between the two turbines because of scale effects (larger turbines generally have higher efficiency). The diameter of the new turbine is more than twice that of the existing turbine, so the increase in efficiency due to turbine size may be significant. We account for the size increase by using the Moody efficiency correction equation, considering turbine A as the "model" and B as the "prototype",

Efficiency correction:

$$\eta_{\text{turbine, B}} \approx 1-\left(1-\eta_{\text{turbine, A}}\right)\left(\frac{D_{\text{A}}}{D_{\text{B}}}\right)^{1/5} = 1-\left(1-0.762\right)\left(\frac{0.86 \text{ m}}{2.04 \text{ m}}\right)^{1/5} = \mathbf{0.800}$$

or **80.0%**. So, the first-order correction yields a predicted efficiency for the larger turbine that is about four percent greater than the smaller turbine. However, as mentioned in the text, we expect only about 2/3 of this increase, or 2(0.800 – 0.762)/3 = 0.025 or 2.5%. Thus, our best estimate of the actual efficiency of the prototype is

$$\eta_{\text{turbine B}} \approx 76.2 + 2.5 = \mathbf{78.7\%}$$

The higher efficiency of the new larger turbine is significant because an increase in power production of 2.5% can lead to significant profits for the power company.

Discussion If the flow entering the new turbine from the penstock were not similar to that of the existing turbine (e.g., velocity profile and turbulence intensity), we could not expect exact dynamic similarity.

Solutions Manual for

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Yunus A. Çengel & John M. Cimbala

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Chapter 15

INTRODUCTION TO COMPUTATIONAL FLUID DYNAMICS

Fundamentals, Grid Generation, and Boundary Conditions

15-1C

Solution We are to list the unknowns and the equations for a given flow situation.

Analysis There are only three unknowns in this problem, ***u*, *v*, and *P* (or*P′*)**. Thus, we require three equations: **continuity**, ***x* momentum** (or *x* component of Navier-Stokes), and ***y* momentum** (or *y* component of Navier-Stokes). These equations, when combined with the appropriate boundary conditions, are sufficient to solve the problem.

Discussion The actual equations to be solved by the computer are discretized versions of the differential equations.

15-2C

Solution We are to define several terms or phrases and provide examples.

Analysis

(*a*) **A computational domain is a region in space (either 2-D or 3-D) in which the numerical equations of fluid flow are solved by CFD**. The computational domain is bounded by edges (2-D) or faces (3-D) on which boundary conditions are applied.

(*b*) **A mesh is generated by dividing the computational domain into tiny cells**. The numerical equations are then solved in each cell of the mesh. A mesh is also called a grid.

(*c*) **A transport equation is a differential equation representing how some property is transported through a flow field**. The transport equations of fluid mechanics are conservation equations. For example, the continuity equation is a differential equation representing the transport of mass, and also conservation of mass. The Navier-Stokes equation is a differential equation representing the transport of linear momentum, and also conservation of linear momentum.

(*d*) **Equations are said to be coupled when at least one of the variables (unknowns) appears in more than one equation**. In other words, the equations cannot be solved alone, but must be solved simultaneously with each other. This is the case with fluid mechanics since each component of velocity, for example, appears in the continuity equation and in all three components of the Navier-Stokes equation.

Discussion Students' definitions should be in their own words.

15-3C

Solution We are to discuss the difference between nodes and intervals and analyze a given computational domain in terms of nodes and intervals.

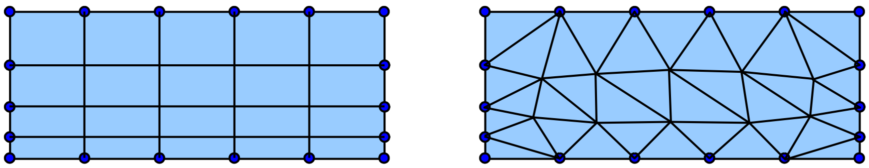
Analysis **Nodes are points along an edge of a computational domain that represent the vertices of cells**. In other words, they are the points where corners of the cells meet. **Intervals, on the other hand, are short line segments between nodes**. Intervals represent the small edges of cells themselves. In Fig. P15-3 there are **6 nodes** and **5 intervals** on the top and bottom edges. There are **5 nodes** and **4 intervals** on the left and right edges.

Discussion We can extend the node and interval concept to three dimensions.

15-4C

Solution For a given computational domain with specified nodes and intervals we are to compare a structured grid and an unstructured grid and discuss.

Analysis We construct the two grids in the figure: (a) structured, and (b) unstructured.



(a) (b)

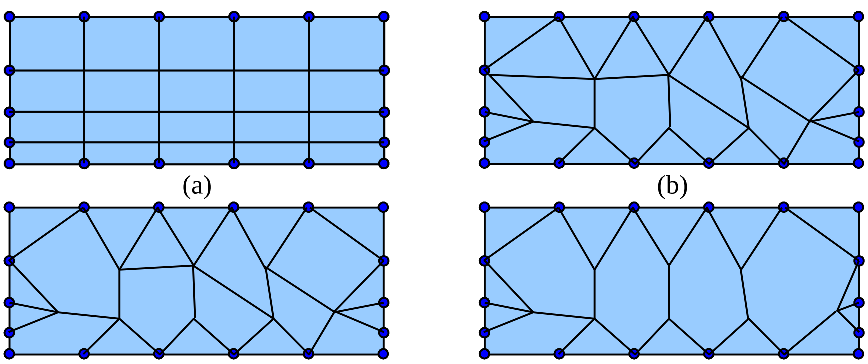
There are 5 × 4 = 20 cells in the structured grid, and there are 36 cells in the unstructured grid.

Discussion Depending on how individual students construct their unstructured grid, the shape, size, and number of cells may differ considerably.

15-5C

Solution For a given computational domain with specified nodes and intervals we are to compare a structured mesh and a polyhedral mesh and discuss.

Analysis We construct the two grids in the figure: (a) structured, and (b) unstructured polyhedral. We show two other options in (c) and (d). There are many possible answers for the polyhedral mesh, depending on how large you want your cells to be.



(a) (b) (c) (d)

There are 5 × 4 = 20 cells in the structured grid. There are 22 cells in polyhedral grid (b). There are some cells with 3 sides, 4 sides, and 5 sides, as required. Compared to the triangular mesh with 36 cells, we have reduced the cell count considerably. In (c) and (d), there are 21 cells and 18 cells respectively. In case (d) we have reduced the cell count below that of even the structured grid. In that case, 3 of the cells have 6 sides each. The cell reduction is particularly useful in large 3-D problems where CPU time and computer memory are important limitations.

Discussion Note that the node distribution along the boundaries is identical in each case, but we have great flexibility in how we create the grid. Depending on how individual students construct their unstructured grid, the shape, size, and number of cells may differ considerably.

15-6C

Solution We are to summarize the eight steps involved in a typical CFD analysis.

Analysis We list the steps in the order presented in this chapter:

1. **Specify a computational domain and generate a grid.**
2. **Specify boundary conditions on all edges or faces.**
3. **Specify the type of fluid and its properties.**
4. **Specify numerical parameters and solution algorithms.**
5. **Apply initial conditions as a starting point for the iteration.**
6. **Iterate towards a solution.**
7. **After convergence, analyze the results (post processing).**
8. **Calculate global and integral properties as needed.**

Discussion The order of some of the steps is interchangeable, particularly Steps 2 through 5.

15-7C

Solution We are to explain why the cylinder should not be centered horizontally in the computational domain.

Analysis Flow separates over bluff bodies, generating a wake with **reverse flow and eddies downstream of the body**. There are no such problems upstream. Hence it is always wise to extend the downstream portion of the domain as far as necessary to avoid reverse flow problems at the outlet boundary.

Discussion The same problems arise at the outlet of ducts and pipes – sometimes we need to extend the duct to avoid reverse flow at the outlet boundary.

15-8C

Solution We are to discuss the significance of several items with respect to iteration.

Analysis

(*a*) In a CFD solution, we typically iterate towards a solution. **In order to get started, we make some initial conditions for all the variables (unknowns) in the problem**. These initial conditions are wrong, of course, but they are necessary as a starting point. Then we begin the iteration process, eventually obtaining the solution.

(*b*) **A residual is a measure of how much our variables differ from the "exact" solution**. We construct a residual by putting all the terms of a transport equation on one side, so that the terms all add to zero if the solution is correct. As we iterate, the terms will *not* add up to zero, and the remainder is called the residual. As the CFD solution iterates further, the residual should (hopefully) decrease.

(*c*) **Iteration is the numerical process of marching towards a final solution**, beginning with initial conditions, and progressively correcting the solution. As the iteration proceeds, the variables converge to their final solution as the residuals decrease.

(*d*) **Once the CFD solution has converged, post processing is performed on the solution**. Examples include plotting velocity and pressure fields, calculating global properties, generating other flow quantities like vorticity, etc. Post processing is performed after the CFD solution has been found, and does not change the results. Post processing is generally not as CPU intensive as the iterative process itself.

Discussion We have assumed steady flow in the above discussions.

15-9C

Solution We are to discuss how the iteration process is made faster.

Analysis

(*a*) With *multigridding*, **solutions of the equations of motion are obtained on a coarse grid first, followed by successively finer grids**. This speeds up convergence because the gross features of the flow are quickly established on the coarse grid, and then the iteration process on the finer grid requires less time.

(*b*) In some CFD codes, **a steady flow is treated as though it were an unsteady flow. Then, an *artificial time* is used to march the solution in time**. Since the solution is steady, however, the solution approaches the steady-state solution as "time" marches on. In some cases, this technique yields faster convergence.

Discussion There are other "tricks" to speed up the iteration process, but CFD solutions often take a long time to converge.

15-10C

Solution We are to list the boundary conditions that are applicable to a given edge, and we are to explain why other boundary conditions are not applicable.

Analysis We may apply the following boundary conditions: **outflow**, **pressure inlet**, **pressure outlet**, **symmetry** (to be discussed), **velocity inlet**, and **wall**. The curved edge cannot be an *axis* because an axis must be a straight line. The edge cannot be a *fan* or *interior* because such edges cannot be at the outer boundary of a computational domain. Finally, **the edge cannot be periodic since there is no other edge along the boundary of the computational domain that is of identical shape** (a periodic boundary must have a "partner"). The symmetry boundary condition merits further discussion. Numerically, gradients of flow variables in the direction normal to a symmetry boundary condition are set to zero, and there is no mathematical reason why the curved right edge of the present computational domain cannot be set as symmetry. However, you would be hard pressed to think of a physical situation in which a curved edge like that of Fig. P15-9 would be a valid symmetry boundary condition.

Discussion Just because you can set a boundary condition and generate a CFD result does not guarantee that the result is physically meaningful.

15-11C

Solution We are to discuss the standard method to test for adequate grid resolution.

Analysis The standard method to test for adequate grid resolution is to **increase the resolution** (by a factor of 2 in all directions if feasible) **and repeat the simulation**. If the results do not change appreciably, the original grid is deemed adequate. If, on the other hand, there are significant differences between the two solutions, the original grid is likely of inadequate resolution. In such a case, an even finer grid should be tried until the grid is adequately resolved.

Discussion Keep in mind that if the boundary conditions are not specified properly, or if the chosen turbulence model is not appropriate for the flow being simulated by CFD, no amount of grid refinement is going to make the solution more physically correct.

15-12C

Solution We are to discuss the difference between a pressure inlet boundary condition and a velocity inlet boundary condition, and we are to explain why both pressure and velocity cannot be specified on the same boundary.

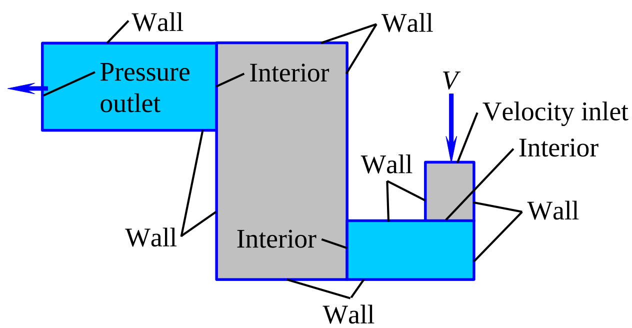
Analysis **At a pressure inlet we specify the pressure but not the velocity**. **At a velocity inlet we specify the opposite – velocity but not pressure**. To specify both pressure and velocity would lead to mathematical **over-specification, since pressure and velocity are coupled in the equations of motion**. When pressure is specified at a pressure inlet (or outlet), the CFD code automatically adjusts the velocity at that boundary. In a similar manner, when velocity is specified at a velocity inlet, the CFD code adjusts the pressure at that boundary.

Discussion Since pressure and velocity are coupled, specification of both at a boundary would lead to inconsistencies in the equations of motion at that boundary.

15-13C

Solution We are to label all the boundary conditions to be applied to a computational domain.

Analysis The inlet is a *velocity inlet*. The outlet is a *pressure outlet*. All other edges that define the outer limits of the computational domain are *walls*. Finally, there are three edges that must be specified as *interior*. These are all labeled in the figure below.



Discussion It is critical that each boundary condition be specified carefully. Otherwise the CFD solution will not be correct.

15-14C

Solution We are to analyze what will happen to inlet pressure and outlet velocity when a fan is turned on in the computational domain of the previous problem.

Analysis Since the fan helps to push air through the channel, the inlet pressure will adjust itself so that less inlet pressure is required. In other words, the **inlet pressure will decrease when the fan is turned on**. Since the inlet velocity is the same in both cases, the mass flow rate (and volume flow rate since the flow is incompressible) must remain the same for either case. Therefore, **outlet velocity will not change**.

Discussion It may seem at first glance that V_{out} should increase because of the fan, but in order to conserve mass, the outlet velocity cannot change. The solution is constrained by the specified inlet boundary condition. In a real physical experiment, there is no such restriction. The fan would cause the inlet pressure to decrease, the inlet velocity to increase, and the outlet velocity to increase.

15-15C

Solution We are to list and briefly describe six boundary conditions, and we are to give an example of each.

Analysis In the chapter we list ten, so any six of these will suffice:

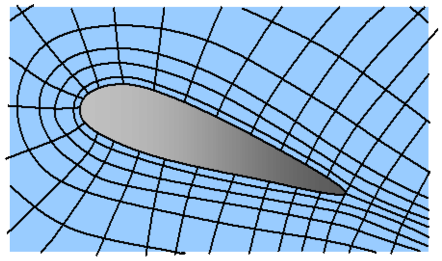
- **Axis**: Used in axisymmetric flows as the axis of rotation. Example: the axis of a torpedo-shaped body.
- **Fan**: An internal edge (2-D) or face (3-D) across which a sudden pressure rise is specified. Example: an axial flow fan in a duct.
- **Interior**: An internal edge (2-D) or face (3-D) across which nothing special happens – the interior boundary condition is used at the interface between two blocks. Example: all of the multiblock problems in this chapter, which require this boundary condition at the interface between any two blocks.
- **Outflow**: An outlet boundary condition in which the gradient of fluid properties is zero normal to the outflow boundary. Outflow is typically useful far away from the object or area of interest in a flow field. Example: the far field of flow over a body.
- **Periodic**: When the physical geometry has periodicity, the periodic boundary condition is used to specify that whatever passes through one face of the periodic pair must simultaneously enter the other face of the periodic pair. Example: in a heat exchanger where there are several rows of tubes.
- **Pressure inlet**: An inflow boundary in which pressure (but not velocity) is known and specified across the face. Example: the high pressure settling chamber of a blow-down wind tunnel facility.
- **Pressure outlet**: An outflow boundary in which pressure (but not velocity) is known and specified across the face. Example: the outlet of a pipe exposed to atmospheric pressure.
- **Symmetry**: A face over which the gradients of all flow variables are set to zero normal to the face – the result is a mirror image across the symmetry plane. Fluid cannot flow *through* a symmetry plane. Example: the mid-plane of flow over a circular cylinder in which the lower half is a mirror image of the upper half.
- **Velocity inlet**: An inflow boundary condition in which velocity (but not pressure) is known and specified across the face. Example: a uniform freestream inlet flow entering a computational domain from one side.
- **Wall**: A boundary through which fluid cannot pass and at which the no-slip condition (or a shear stress condition) is applied. Example: the surface of an airfoil that is being modeled by CFD.

Discussion There are additional boundary conditions used in CFD calculations, but these are the only ones discussed in this chapter.

15-16

Solution We are to sketch a structured and an unstructured grid near the airfoil surface, and discuss advantages and disadvantages of each.

Analysis In either case it is wise to cluster cells close to the airfoil surface since we expect that a thin boundary layer will exist along the surface, and we need many tiny cells within the boundary layer to adequately resolve it. Some simple, coarse meshes are drawn in Fig. 1. We would certainly want much higher resolution for CFD calculations.


(a)

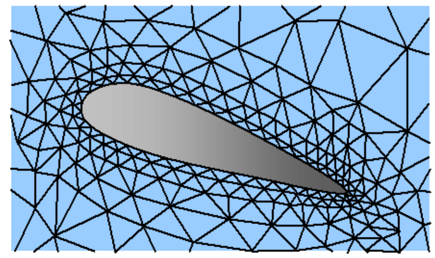

(b)

FIGURE 1
A coarse structured (a) and unstructured (b) grid. Notice that the cells are clustered (more fine) near the surface of the airfoil since there is likely to be large velocity gradients there (in the boundary layer).

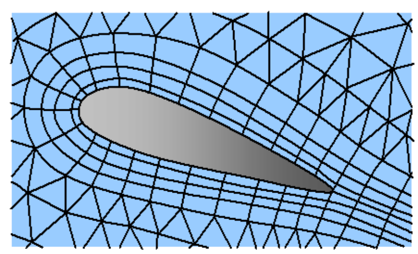
The structured grid in Fig 1a is called a C-grid since it wraps around the airfoil like the letter "C". The main advantage of the structured grid is that we can get high resolution near the surface with few cells. The main advantage of the unstructured grid is that it is somewhat easier to generate when the geometry is complicated (especially for highly curved surfaces). Furthermore, it is easier to transition between curved and straight edges with an unstructured grid. The main disadvantage of an unstructured grid is that more cells are required for the same spatial resolution.

Discussion There are numerous other ways to construct a grid around this airfoil.

15-17

Solution We are to sketch a hybrid grid around an airfoil and explain its advantages.

Analysis We sketch a hybrid grid in the figure. Note that the grid is structured near the airfoil surface, but unstructured beyond the surface. **The advantage of a hybrid grid is that it combines the advantages of both structured and unstructured grids**. Near surfaces we can use a structured grid to finely resolve the boundary layer with a minimum number of cells, and away from surfaces we can use an unstructured grid so that we can rapidly expand the cell size. We can also more easily blend the grid into the edges of the computational domain with an unstructured grid.



Discussion A structured grid is generally the best choice, but a hybrid grid is often a better option than a fully unstructured grid.

15-18

Solution We are to sketch the blocking for a structured grid, sketch a coarse grid, and label all the boundary conditions to be applied to the computational domain.

Analysis First of all, we recognize that because of symmetry, we can split the domain in half vertically. We construct four blocks around the half-cylinder to transform from round to square, and then we add simple rectangular blocks upstream and downstream of the cylinder (Fig. 1). There is a total of **six blocks**.

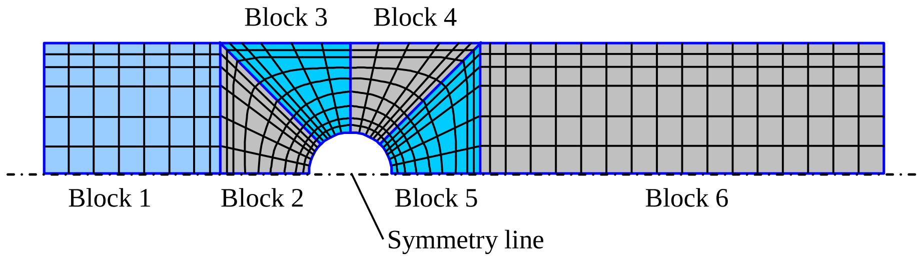


FIGURE 1
A possible blocking topology and coarse structured grid for a 2-D multiblock computational domain.

With the block structure of Fig. 1 no cells are highly skewed, and cells are clustered near the cylinder wall and the upper wall of the duct as desired.

The bottom edge of the computational domain is a line of **symmetry**. The inlet is a **velocity inlet**. The outlet is a **pressure outlet**. The upper edge of the computational domain is a **wall**. The edges that define the cylinder are also **walls**. Finally, there are 5 edges that are specified as **interior**. These are all labeled in Fig. 2.

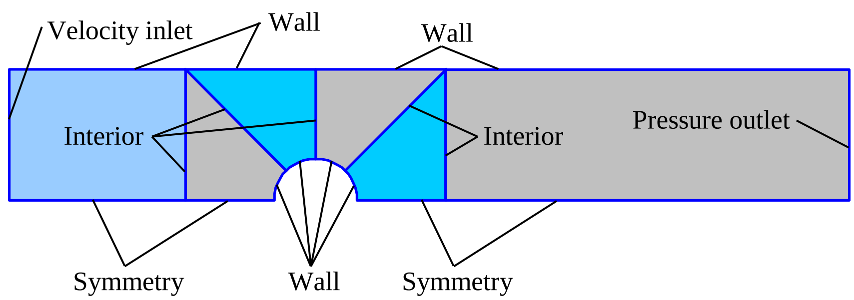


FIGURE 2
Boundary conditions specified on each edge of a 2-D multiblock computational domain.

Discussion There are alternative ways to set up the blocking topology. For example, at the top we may define a thin block (Block 7) that stretches across the entire horizontal domain so that the boundary layer on the top wall of the channel can be more adequately resolved (Fig. 3).

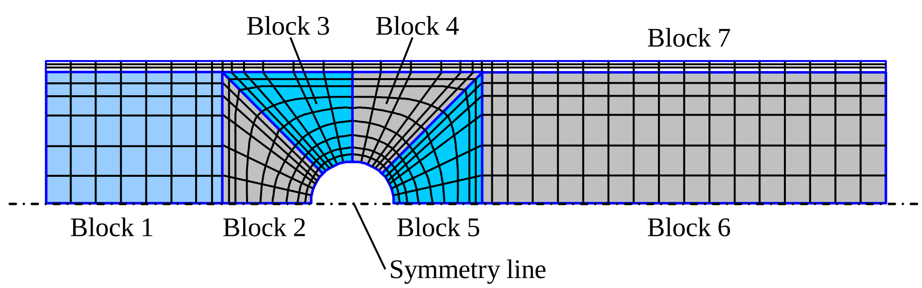


FIGURE 3
An alternative blocking topology and coarse structured grid for the 2-D multiblock computational domain. A seventh block is added at the top for better boundary layer resolution near the top plate.

15-19

Solution We are to sketch the blocking for a structured grid, sketch a coarse grid, and label all the boundary conditions to be applied to the computational domain.

Analysis First of all, we recognize that because of symmetry, we can split the domain in half vertically. We construct four blocks inside the half-circle, and then we add blocks with one curved edge and three straight edges upstream and downstream of the cylinder (Fig. 1).

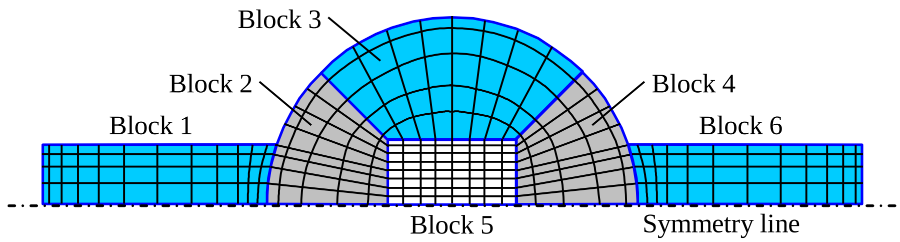


FIGURE 1
The blocking and coarse structured grid for a 2-D multiblock computational domain.

The setup of Fig. 1 contains six blocks. With this block structure, no cells are highly skewed, and cells are clustered near upper wall of the duct as desired. Cells are also clustered at the junctions between Blocks 1 and 2 and Blocks 4 and 6, where flow separation may occur.

The bottom edge of the computational domain is a line of **symmetry**. The inlet is a **velocity inlet**. The outlet is a **pressure outlet**. The upper edge of the computational domain is a **wall**. The edges that define the cylinder are also **walls**. Finally, there are 5 edges that are specified as **interior**. These are all labeled in Fig. 2.

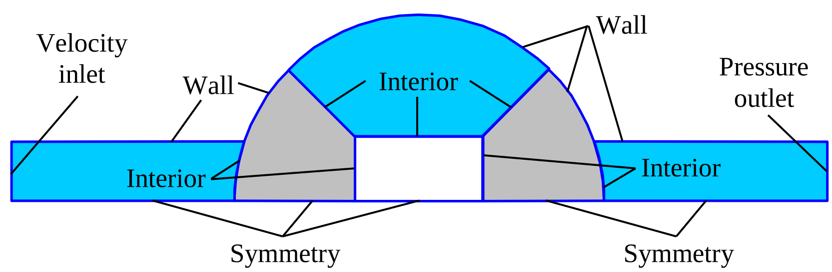


FIGURE 2
Boundary conditions specified on each edge of a 2-D multiblock computational domain.

Discussion There are of course, alternative ways to set up the blocking topology.

15-20

Solution We are to modify an existing grid so that all blocks are elementary blocks. Then we are to verify that the total number of cells does not change.

Analysis The right edge of Block 2 of Fig. 15-11b is split twice to accommodate Block 1. We therefore split Block 2 into three separate elementary blocks. Unfortunately, this process ends up splitting Block 6 twice, which in turn splits Block 4 twice. **We end up with 12 elementary blocks** as shown in Fig. 1.

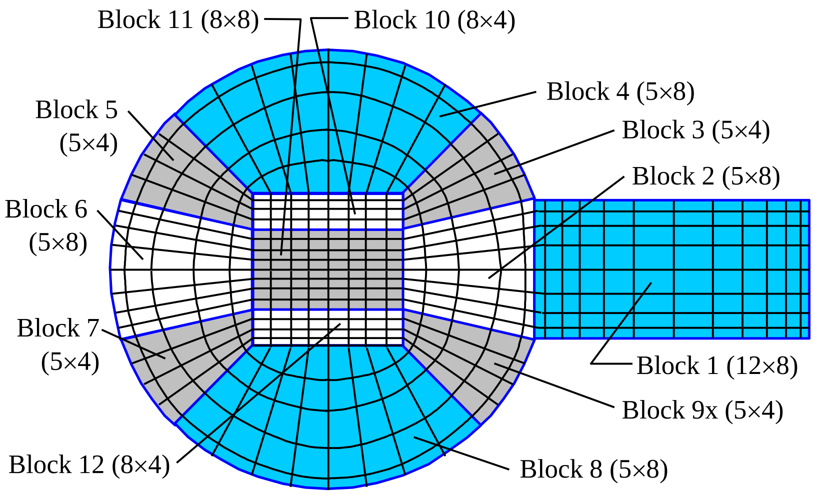


FIGURE 1
The blocking and coarse structured grid for a 2-D multiblock computational domain. Only elementary blocks are used in this grid.

We add up all the cells in these 12 blocks – we get a total of **464 cells**. This agrees with the total of 464 cells for the original 6 blocks in the domain.

Discussion Sometimes it is easier to create a grid with elementary blocks, even if the CFD code can accept blocks with split edges or faces.

15-21

Solution We are to modify an existing grid into a smaller number of non- elementary blocks, and we are to verify that the total number of cells does not change.

Analysis We combine Blocks 2, 3, 4, and 5 of Fig. 15-10b. Together, these produce one structured grid that wraps around the square in the middle – there are still 5 i intervals, but now there are 48 j intervals. **We end up with 3 non-elementary blocks**, as shown in Fig. 1.

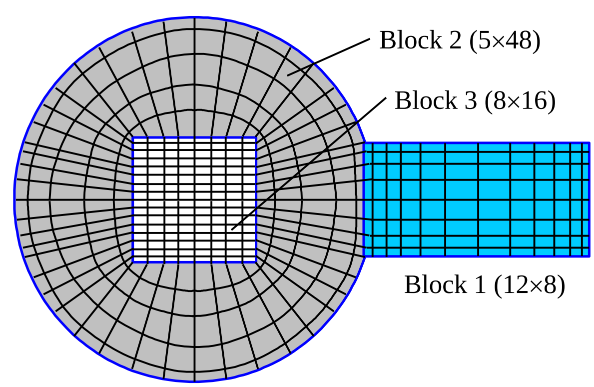


FIGURE 1
The blocking and coarse structured grid for a 2-D multiblock computational domain. Only elementary blocks are used in this grid.

We add up all the cells in these 12 blocks – we get a total of **464 cells**. This agrees with the total of 464 cells for the original 6 blocks in the domain.

Discussion Block 2 in Fig. 1 is called an **O-grid** (for obvious reasons).

15-22

Solution We are to generate a computational domain and label all appropriate boundary conditions for one stage of a new heat exchanger design.

Analysis We take advantage of the periodicity of the geometry. There are several ways to create a periodic grid for this flow. The simplest computational domain consists of a single flow passage between two neighboring tubes. We can make the periodic edge intercept anywhere on the front portion of the tube that we desire. We choose the lower surface for convenience and simplicity. The periodic computational domain is sketched in Fig. 1.

Boundary conditions are also straightforward, and are labeled in Fig. 1. For a known inlet velocity we set the boundary condition at the left edge as a **velocity inlet**. The tube walls are obviously set as **walls**. The outlet can be set as either a **pressure outlet** or an **outflow**, depending on the provided information and how far the outlet region extends beyond the tubes. Finally, we set two pairs of **translationally periodic** boundaries, one fore and one aft of the tubes. We label them separately to avoid confusion.

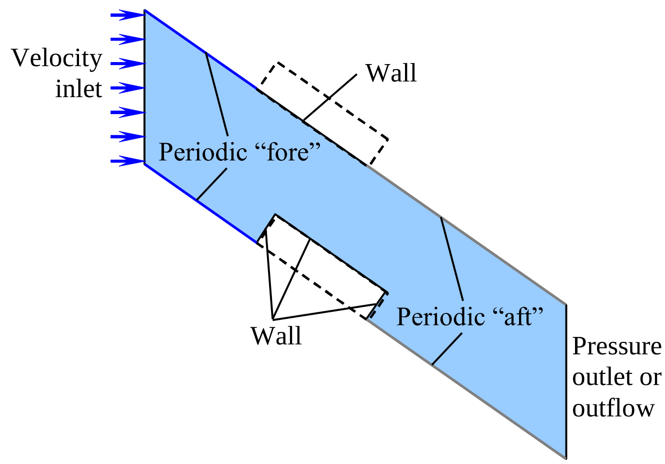


FIGURE 1
A periodic computational domain for a given geometry. Boundary conditions are also labeled.

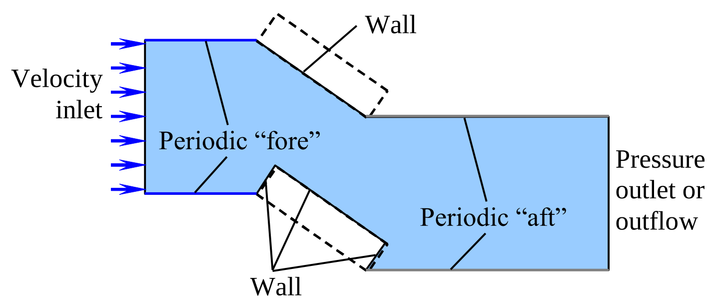


FIGURE 2
An alternative periodic computational domain for a given geometry. Boundary conditions are also labeled.

Discussion The fore and aft periodic edges are not horizontal in Fig. 1. This is not a problem since the periodic boundary condition is not restricted to horizontal or even to flat surfaces. An alternative, equally acceptable computational domain is shown in Fig. 2.

15-23

Solution We are to sketch a structured multiblock grid with four-sided elementary blocks for a given computational domain.

Analysis We choose the computational domain of Fig. 1 of the previous problem. Since all edges are straight, the blocking scheme can be rather simple. We sketch the blocking topology and apply a coarse mesh in Fig. 1 for the case in which the CFD code does not require the node distribution to be exactly the same on periodic pairs.

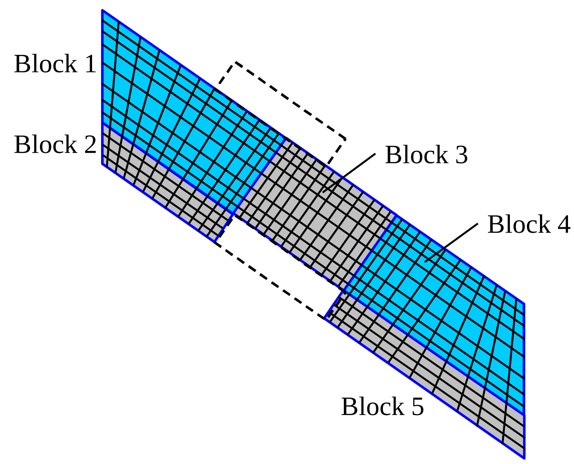


FIGURE 1
The blocking topology and a coarse structured grid for a periodic computational domain. This blocking topology applies to CFD codes that allow a block's edges to be split for application of boundary conditions, and do *not* require periodic edge pairs to have identical node distributions.

Unfortunately, many CFD codes require that the node distribution on periodic pairs of edges be identical (the two edges of a periodic pair are "linked" in the grid generation process). In such a case, the grid of Fig. 1 would not be acceptable. Furthermore, although the edges of the blocks of Fig. 1 are not split with respect to adjacent blocks, the top edges of Block 1 and Block 3 are split with respect to the boundary conditions (part of the edge is periodic and part is a wall). Thus these blocks are not really elementary blocks after all. We construct a more elaborate blocking topology in Fig. 2 to correct these problems. The node distribution on the edges of each periodic pair are identical, at the expense of more complexity (7 instead of 5 blocks) and more cell skewness.

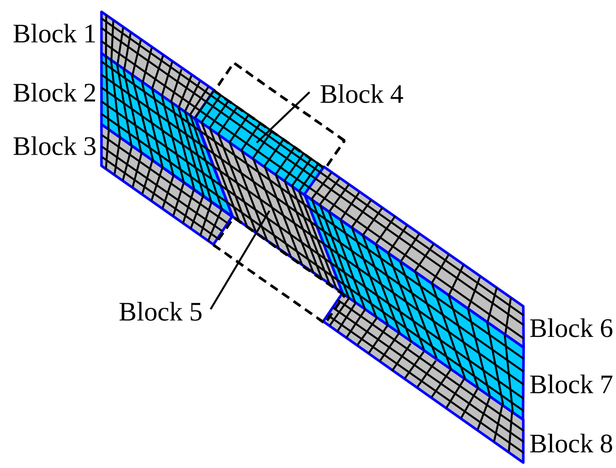


FIGURE 2
The blocking topology and a coarse structured grid for a periodic computational domain. This blocking topology applies to CFD codes that require elementary blocks and require periodic edge pairs to have identical node distributions.

Discussion Some of the cells have moderate skewness with the blocking topology of Fig. 2, especially near the corners of Block 2 and Block 7 and throughout Block 5. A more complicated topology can be devised to reduce the amount of skewness.

15-24

Solution We are to discuss why there is reverse flow in this CFD calculation, and then we are to explain what can be done to correct the problem.

Analysis **Reverse flow at an outlet is usually an indication that the computational domain is not large enough**. In this case the rectangular heat exchanger tubes are inclined at 35^o, and the flow will most certainly separate, leaving large recirculating eddies in the wakes. **Anita should extend the computational domain in the horizontal direction downstream** so that the eddies have a chance to "close" and the flow has a chance to re-develop into a flow without any reverse flow.

Discussion In most commercial CFD codes a warning will pop up on the computer monitor whenever there is reverse flow at an outlet. This is usually an indication that the computational domain should be enlarged.

15-25

Solution We are to generate a computational domain and label all appropriate boundary conditions for two stages of a heat exchanger.

Analysis We look for the smallest computational domain that takes advantage of the periodicity of the geometry. There are several ways to create a periodic grid for this flow. The simplest computational domain consists of a single flow passage between two neighboring tubes. We can make the periodic edge intercept anywhere on the fore and aft portions of the heat exchanger that we desire. We choose the periodic computational domain sketched in Fig. 1.

Boundary conditions are also straightforward, and are labeled in Fig. 1. For a known inlet velocity we set the boundary condition at the left edge as a **velocity inlet**. The tube walls are obviously set as **walls**. The outlet can be set as either a **pressure outlet** or an **outflow**, depending on the provided information and how far the outlet region extends beyond the tubes. Finally, we set three pairs of **translationally periodic** boundaries, one fore, one mid, and one aft of the tubes. We label them separately to avoid confusion.

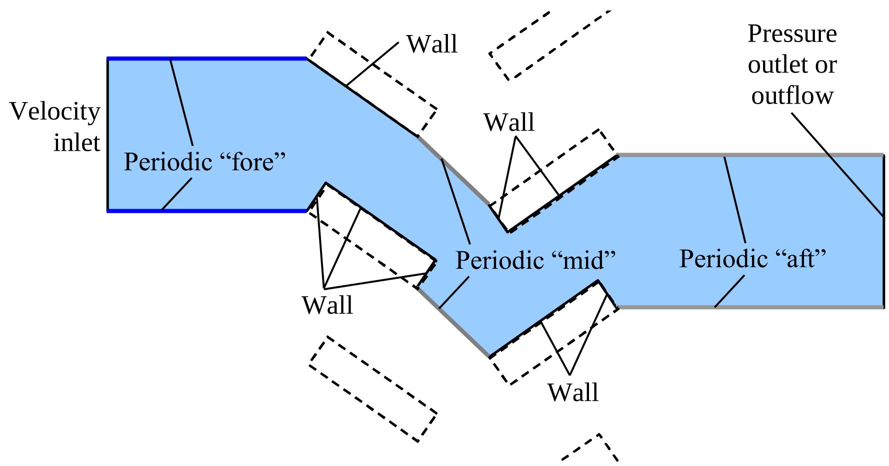


FIGURE 1
A periodic computational domain for a given geometry. Boundary conditions are also labeled.

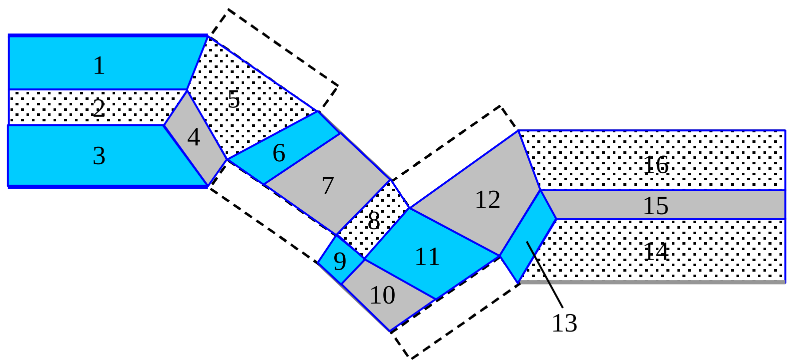
Discussion Many other equally acceptable computational domains are possible.

15-26

Solution We are to sketch a structured multiblock grid with four-sided elementary blocks for a given computational domain.

Analysis We choose the computational domain of Fig. 1 of the previous problem. We sketch one possible elementary blocking topology in Fig. 1 for the case in which the CFD code requires the node distribution to be exactly the same on periodic pairs. We also assume that we cannot split one periodic edge and not its partner. The blocks are numbered. This topology has **16 elementary blocks**.

FIGURE 1
The elementary blocking topology for a periodic computational domain. This blocking topology applies to CFD codes that do not allow a block's edges to be split for application of boundary conditions, and requires periodic edge pairs to have identical node distributions.



Note that with this blocking topology we had to split the periodic "mid" boundary pair into two edges (the tops of blocks 6 and 7 and the bottoms of blocks 9 and 10). As long as both pairs of each segment are the same size and have the same number of nodes, this is not a problem. In the CFD code we would have to name each periodic pair separately, however. The block numbers are labeled. Notice that most of the blocks are nearly rectangular such that none of the computational cells would have to be highly skewed.

Discussion This seems like a rather complicated blocking topology. It would require a bit of work to generate the grid. However, the time spent on developing a good grid is usually well worth the effort. By reducing the amount of cell skewness, we are able to speed up the CFD calculations and obtain more accurate results. This kind of topology also enables us to cluster cells near walls and wakes as needed.

General CFD Problems

15-27

Solution We are to generate three different coarse grids for the same geometry and node distribution, and then compare the cell count and grid quality.

Analysis The three meshes are shown in Fig. 1. The node distributions along the edges of the computational domain are identical in all three cases, and no smoothing of the mesh is performed. The structured multi-block mesh is shown in Fig. 1a. We split the domain into four blocks for convenience, and to achieve cells with minimal skewing. There are 1060 cells. The unstructured triangular mesh is shown in Fig. 1b. There is only one block, and it contains 1996 cells. The unstructured quad mesh is shown in Fig. 1c. It has 833 cells in its one block. Comparing the three meshes, the triangular unstructured mesh has too many cells. The unstructured quad mesh has the least number of cells, but the clustering of cells occurs in undesirable locations, such as at the outlets on the right. The structured quad mesh seems to be the best choice for this geometry – it has only about 27% more cells than the unstructured quad mesh, but we have much more control on the clustering of the cells. Skewness is not a problem with any of the meshes.

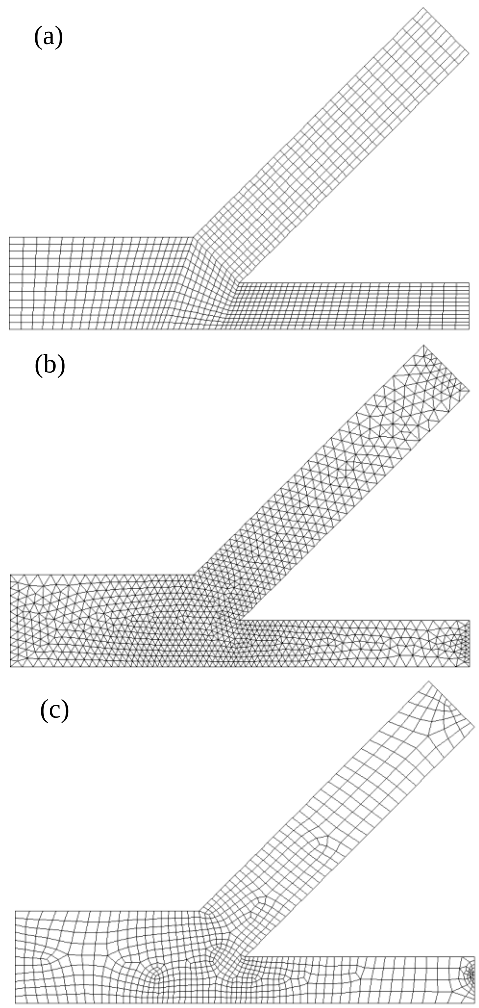


FIGURE 1
Comparison of three meshes: (a) structured multiblock, (b) unstructured triangular, and (c) unstructured quadrilateral.

Discussion Depending on the grid generation software and the specified node distribution, students will get a variety of results.

15-28

Solution We are to run a laminar CFD calculation of flow through a wye, calculate the pressure drop and how the flow splits between the two branches.

Analysis We choose the structured grid for our CFD calculations. The back pressure at both outlets is set to zero gage pressure, and the average pressure at the inlet is calculated to be -8.74×10^{-5} Pa. The pressure drop through the wye is thus only 8.74×10^{-5} Pa (a negligible pressure drop). The streamlines are shown in Fig. 1. For this case, 57.8% of the flow goes out the upper branch, and 42.2% goes out the lower branch.

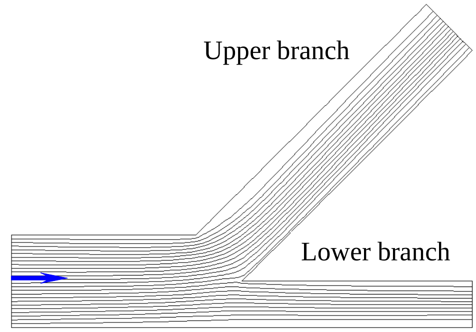


FIGURE 1
Streamlines for laminar flow through a wye.

Discussion There appears to be some tendency for the flow to separate at the upper left corner of the branch, but there is no reverse flow at the outlet of either branch. This case is compared to a turbulent flow case in the following problem.

15-29

Solution We are to run a turbulent CFD calculation of flow through a wye, calculate the pressure drop and how the flow splits between the two branches.

Analysis We choose the structured grid for our CFD calculations. The back pressure at both outlets is set to zero gage pressure, and the average pressure at the inlet is calculated to be -3.295 Pa. The pressure drop through the wye is thus 3.295 Pa (a significantly higher pressure drop than that of the laminar flow, although we note that the inlet velocity for the laminar flow case was 0.002 times that of the turbulent flow case). The streamlines are shown in Fig. 1. For this case, 54.4% of the flow goes out the upper branch, and 45.6% goes out the lower branch. Compared to the laminar case, a greater percentage of the flow goes out the lower branch for the turbulent case. The streamlines at first look similar, but a closer look reveals that the spacing between streamlines in the turbulent case is more uniform, indicating that the velocity distribution is also more uniform (more "full"), as is expected for turbulent flow.

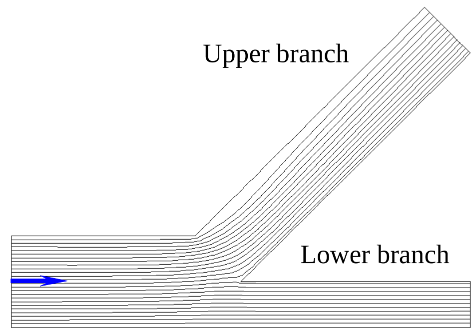


FIGURE 1
Streamlines for turbulent flow through a wye. The k-ε turbulence model is used.

Discussion There appears to be some tendency for the flow to separate at the upper left corner of the branch, but there is no reverse flow at the outlet of either branch.

15-30

Solution We are to keep refining a grid until it becomes grid independent for the case of a laminar boundary layer.

Analysis Students will have varied results, depending on the grid generation code, CFD code, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-31

Solution We are to keep refining a grid until it becomes grid independent for the case of a turbulent boundary layer.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-32

Solution We are to study ventilation in a simple 2-D room using CFD, and using a structured rectangular grid.

Analysis Students will have varied results, depending on the grid generation code, CFD code, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-33

Solution We are to repeat the previous problem except use an unstructured grid, and we are to compare results.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-34

Solution We are to use CFD to analyze the effect of moving the supply and/or return vents in a room.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-35

Solution We are to use CFD to analyze a simple 2-D room with air conditioning and heat transfer.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-36

Solution We are to compare the CFD predictions for 2-D and 3-D ventilation.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-37

Solution We are to use CFD to study compressible flow through a converging nozzle with inviscid walls. Specifically, we are to vary the pressure until we have choked flow conditions.

Analysis Students will have varied results, depending on the grid generation code, CFD code, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-38

Solution We are to repeat the previous problem, but allow friction at the wall, and also use a turbulence model. We are then to compare the results to those of the previous problem to see the effect of wall friction and turbulence on the flow.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-39

Solution We are to generate a low-drag, streamlined, 2-D body, and try to get the smallest drag in laminar flow.

Analysis Students will have varied results, depending on the grid generation code, CFD code, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-40

Solution We are to generate a low-drag, streamlined, axisymmetric body, and try to get the smallest drag in laminar flow. We are also to compare the axisymmetric case to the 2-D case of the previous problem.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-41

Solution We are to generate a low-drag, streamlined, axisymmetric body, and try to get the smallest drag in turbulent flow. We are also to compare the turbulent drag coefficient to the laminar drag coefficient of the previous problem.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-42

Solution We are to use CFD to study Mach waves in supersonic flow. We are also to compare the computed Mach angle with that predicted by theory.

Analysis Students will have varied results, depending on the grid generation code, CFD code, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-43

Solution We are to study the effect of Mach number on the Mach angle in supersonic flow, and we are to compare to theory.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

Review Problems

15-44C

Solution

(*a*) ***False***: If the boundary conditions are not correct, if the computational domain is not large enough, etc., the solution can be erroneous and nonphysical no matter how fine the grid.

(*b*) ***True***: Each component of the Navier-Stokes equation is a transport equation.

(*c*) ***True***: The four-sided cells of a 2-D structured grid require less cells than do the triangular cells of a 2-D unstructured grid. (Note however, that some unstructured cells can be four-sided as well as three-sided.)

(*d*) ***True***: Turbulence models are *approximations* of the physics of a turbulent flow, and unfortunately are not universal in their application.

15-45C

Solution We are to discuss right-left symmetry as applied to a CFD simulation and to a potential flow solution.

Analysis In the time-averaged CFD simulation, we are not concerned about top-bottom fluctuations or periodicity. Thus, top-bottom symmetry can be assumed. However, fluid flows do not have upstream-downstream symmetry in general, even if the geometry is perfectly symmetric fore and aft. In the problem at hand for example, the flow in the channel develops downstream. Also, the flow exiting the left channel enters the circular settling chamber like a jet, separating at the sharp corner. At the opposite end, fluid leaves the settling chamber and enters the duct more like an inlet flow, without significant flow separation. **We certainly cannot expect fore-aft symmetry in a flow such as this**.

On the other hand, potential flow of a symmetric geometry yields a symmetric flow, so it would be okay to cut our grid in half, invoking fore-aft symmetry.

Discussion If unsteady or oscillatory effects were important, we should not even specify top-bottom symmetry in this kind of flow field.

15-46C

Solution We are to discuss improvements to the given computational domain.

Analysis (*a*) Since Gerry is not interested in unsteady fluctuations (which may be unsymmetric), he could eliminate half of the domain. In other words, **he could assume that the axis is a plane of symmetry between the top and bottom of the channel**. Gerry's grid would be cut in size by a factor of two, leading to approximately half the required CPU time, but yielding virtually identical results.

(*b*) The fundamental flaw is that **the outflow boundary is not far enough downstream**. There will likely be flow separation at the corners of the sudden contraction. With a duct that is only about three duct heights long, it is possible that there will be reverse flow at the outlet. Even if there is no reverse flow, the duct is nowhere near long enough for the flow to achieve fully developed conditions. Gerry should extend the outlet duct by many duct heights to allow the flow to develop downstream and to avoid possible reverse flow problems.

Discussion The inlet appears to be perhaps too short as well. If Gerry specifies a fully developed channel flow velocity profile at the inlet, his results may be okay, but again it is better to extend the duct many duct heights beyond what Gerry has included in his computational domain.

15-47C

Solution We are to discuss a feature of modern computer systems for which nearly equal size multiblock grids are desirable.

Analysis The fastest computers are multi-processor computers. In other words, the computer system contains more than one CPU – a **parallel computer**. Modern parallel computers may combine 32, 64, 128, or more CPUs or *nodes*, all working together. In such a situation it is natural to let each node operate on one block. If all the nodes are identical (equal speed and equal RAM), the system is most efficient if the blocks are of similar size.

Discussion In such a situation there must be communication between the nodes. At the interface between blocks, for example, information must pass during the CFD iteration process.

15-48C

Solution We are to discuss the difference between multigridding and multiblocking, and we are to discuss how they may be used to speed up a CFD calculation. Then we are to discuss whether multigridding and multiblocking can be applied together.

Analysis *Multigridding* has to do with the resolution of an established grid during CFD calculations. With multigridding, **solutions of the equations of motion are obtained on a coarse grid first, followed by successively finer grids**. This speeds up convergence because the gross features of the flow are quickly established on the coarse grid (which takes less CPU time), and then the iteration process on the finer grid requires less time.

Multiblocking is something totally different. It refers to the **creation of two or more separate blocks or zones, each with its own grid**. The grids from all the blocks collectively create the overall grid. As discussed in the previous problem, multiblocking can have some speed advantages if using a parallel-processing computer. In addition, some CFD calculations would require too much RAM if the entire computational domain were one large block. In such cases, the grid can be split into multiple blocks, and the CFD code works on one block at a time. This requires less RAM, although information from the dormant blocks must be stored on disk or solid state memory chips, and then swapped into and out of the computer's RAM.

There is no reason why multigridding cannot be used on each block separately. Thus, **multigridding and multiblocking can be used together**.

Discussion Although all the swapping in and out requires more CPU time and I/O time, for large grids multiblocking can sometimes mean the difference between being able to run and not being able to run at all.

15-49C

Solution We are to discuss why we should spend a lot of time developing a multiblock structured grid when we could just use an unstructured grid.

Analysis There are several reasons why a structured grid is "better" than an unstructured grid, even for a case in which the CFD code can handle unstructured grids. First of all **the structured grid can be made to have better resolution *with fewer cells* than the unstructured grid**. This is important if computer memory and CPU time are of concern. Depending on the CFD code, the solution **may *converge more rapidly* with a structured grid**, and the results may be **more accurate**. In addition, by creating multiple blocks, we can more easily **cluster cells** in certain blocks and locations where high resolution is necessary, since we have much more control over the final grid with a structured grid.

Discussion As mentioned in this chapter, time spent creating a good grid is usually time well spent.

15-50

Solution We are to calculate flow through a single-stage heat exchanger.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-51

Solution We are to study the effect of heating element angle of attack on heat transfer through a single-stage heat exchanger.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-52

Solution We are to calculate flow through a single-stage heat exchanger.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-53

Solution We are to study the effect of heating element angle of attack on heat transfer through a two-stage heat exchanger.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-54

Solution We are to study the effect of spin on a cylinder using CFD, and in particular, analyze the lift force.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-55

Solution We are to study the effect of spin speed on a spinning cylinder using CFD, and in particular, analyze the lift force as a function of rotational speed in nondimensional variables.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-56

Solution We are to study flow into a slot along a wall using CFD.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.

15-57

Solution We are to calculate laminar flow into a 2-D slot, compare with irrotational flow theory, and with results of the previous problem, and discuss the vorticity field.

Assumptions **1** The flow is steady and 2-D. **2** The flow is laminar.

Analysis The flow field does not change much from the previous problem, except that a thin boundary layer shows up along the floor. The vorticity is confined to a region close to the floor – vorticity is negligibly small everywhere else, so the irrotational flow approximation is appropriate everywhere except close to the floor.

Discussion The irrotational flow approximation is very useful for suction-type flows, as in air pollution control applications (hoods, etc.).

15-58

Solution We are to model the flow of air into a vacuum cleaner using CFD, and we are to compare the results to those obtained with the potential flow approximation.

Analysis We must include a second length scale in the problem, namely the width *w* of the vacuum nozzle. For the CFD calculations, we set *w* = 2.0 mm and place the inlet plane of the vacuum nozzle at *b* = 2.0 cm above the floor (Fig. 1). Only half of the flow is modeled since we can impose a symmetry boundary condition along the y-axis. We use the same volumetric suction flow rate as in the example problem, i.e., $\dot{V}/L = 0.314$ m^2/s, but in the CFD analysis we specify only *half* of this value since we are modeling half of the flow field.

Results of the CFD calculations are shown in Fig. 2. Fig. 2a shows a view of streamlines in the entire computational plane. Clearly, the streamlines far from the inlet of the nozzle appear as rays into the origin; from "far away" the flow feels the effect of the vacuum nozzle in the same way as it would feel a line sink. In Fig. 2b is shown a close-up view of these same streamlines. Qualitatively, the streamlines appear similar to those predicted by the irrotational flow approximation. In Fig 2c we plot contours of the magnitude of vorticity. Since irrotationality is defined by zero vorticity, these vorticity contours indicate where the irrotational flow approximation is valid – namely in regions where the magnitude of vorticity is negligibly small.

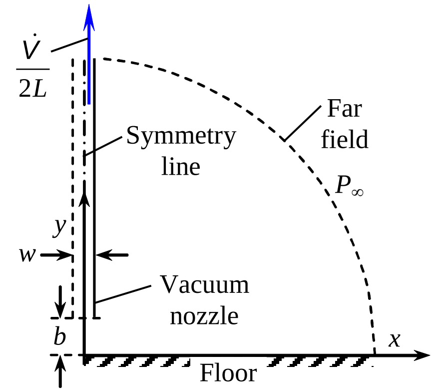


FIGURE 1
CFD model of air being sucked into a vertical vacuum nozzle; the *y*-axis is a line of symmetry (not to scale –the far field is actually much further away from the nozzle than is sketched here).

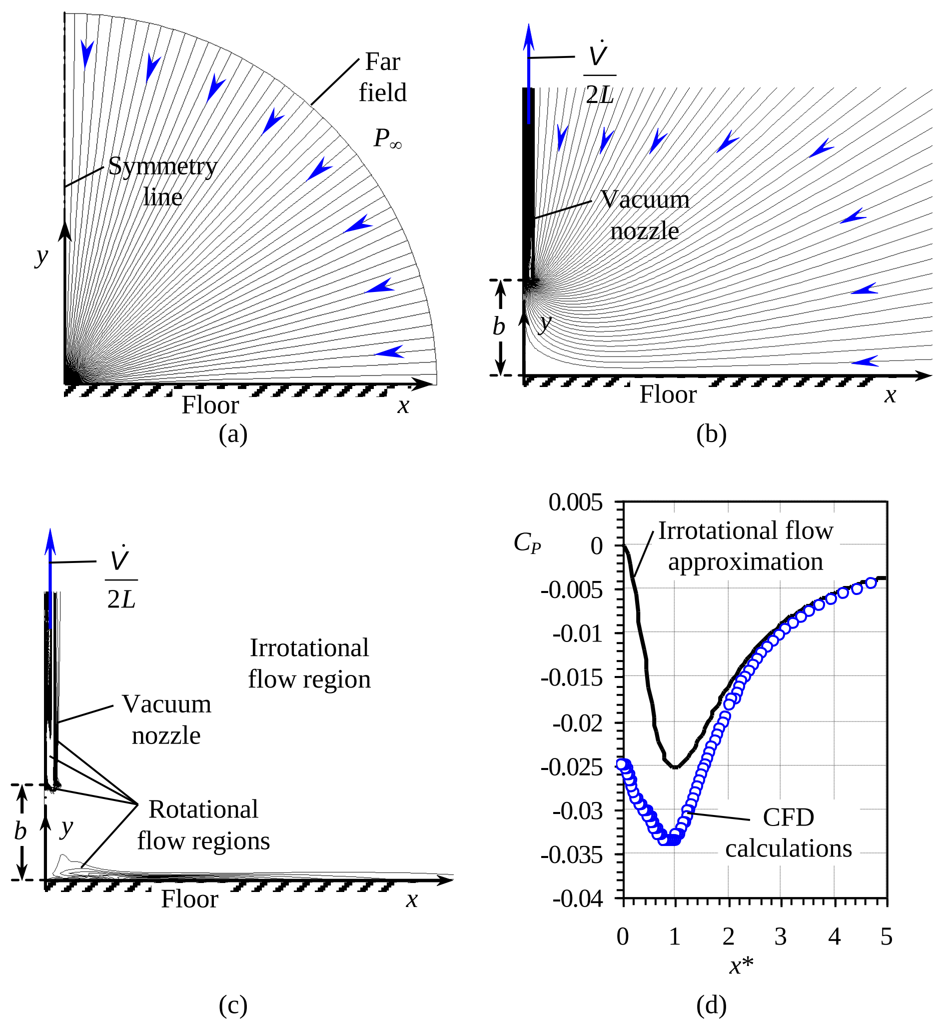


FIGURE 2
CFD calculations of flow into the nozzle of a vacuum cleaner; (a) streamlines in the entire flow domain, (b) close-up view of streamlines, (c) contours of constant magnitude of vorticity illustrating regions where the irrotational flow approximation is valid, and (d) comparison of pressure coefficient with that predicted by the irrotational flow approximation.

We see from Fig. 2c that vorticity is negligibly small everywhere in the flow field except close to the floor, along the vacuum nozzle wall, near the inlet of the nozzle, and inside the nozzle duct. In these regions, net viscous forces are *not* small and fluid particles *rotate* as they move; the irrotational flow approximation is not valid in these regions. Nevertheless, it appears that the irrotational flow approximation is valid throughout the majority of the flow field. Finally, the pressure coefficient predicted by the irrotational flow approximation is compared to that calculated by CFD in Fig. 2d.

Discussion For x^* greater than about 2, the agreement is excellent. However, the irrotational flow approximation is not very reliable close to the nozzle inlet. Note that the irrotational flow prediction that the minimum pressure occurs at $x^* \approx 1$ is verified by CFD.

15-59

Solution We are to compare CFD calculations of flow into a vacuum cleaner for the case of laminar flow versus the inviscid flow approximation.

Analysis Students will have varied results, depending on the grid generation code, CFD code, turbulence model, and their choice of computational domain, etc.

Discussion Instructors can add more details to the problem statement, if desired, to ensure consistency among the students' results.