Hydraulic machines

Hydraulic machines (Turbo Machinery) plays a great role in hydraulics; pumps and turbines are among them. Fluid machines either add energy in to a flow or extract from it. A machine that adds energy to a flow is known as a **pump.** In common usage the term pump is used when the flow is liquid. For gases the terms used are: fans and propellers (low pressure rise), and blowers and compressors (if pressure rise involved is high). Machines that convert the energy of flow into some other form (mechanical) of energy are called **turbines**.

The emphasis in this section shall be on pumps and turbines with the fluid being essentially water.

Types of pumps and their selection

Various types of pumps are used in practice. The choice usually depends on the flow rate, Q, the head to be developed, H and the type of fluid. The most important characteristics of the pump are the shape, the size and the speed.

1. Archimedean Screws

In this pump the mechanical energy of the device is converted in to an increase of the potential energy by the continuous lifting of the fluid. There is no pressure added to the fluid. An Archimedean screw consists of an inclined shaft to which one or more blades are helically attached. The unit is closely fitted in a semi circular casing. With the rotation of the screw, the fluid, enclosed by two successive blades, the shaft and casing, is lifted. The speed of this type of this type of pump is relatively low (5-50 rpm). The most important features of the Archimedean screw are:

- only suitable for practically constant heads
- only suitable to pump fluid from one free surface reservoir to another one
- physical size is large compared to the capacity.
- suitable for polluted fluids
- open, easily accessible construction.

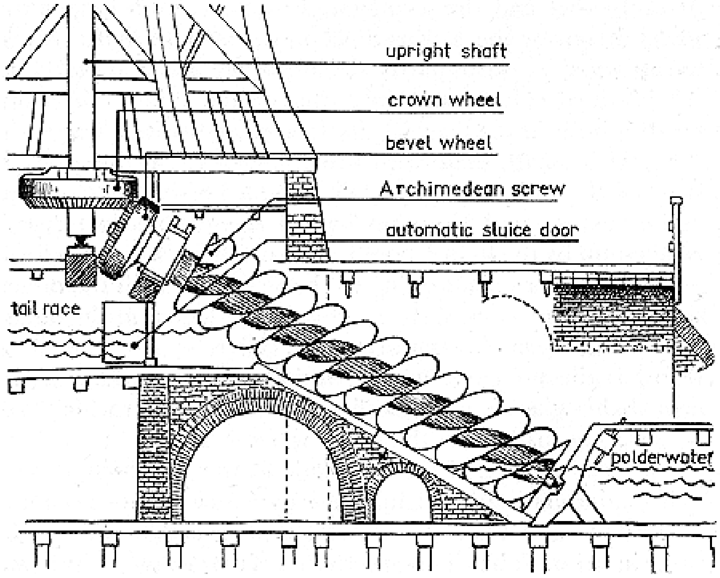


Fig.4.1 Archimedean screw

2. Positive displacement pumps

In positive displacement pumps a moving boundary forces the fluid along by volume change. The pumping proceeds in steps thereby resulting in a pulsating type of flow. The steps involve the admission of the liquid in to a cavity through an inlet (which usually is fitted with a non-return valve). Then the liquid is squeezed so that it leaves the cavity the outlet, figure below.

PDPs have the advantage that they are suitable for any liquid (regardless of viscosity). They do not, however, offer a continuous flow of the fluid.

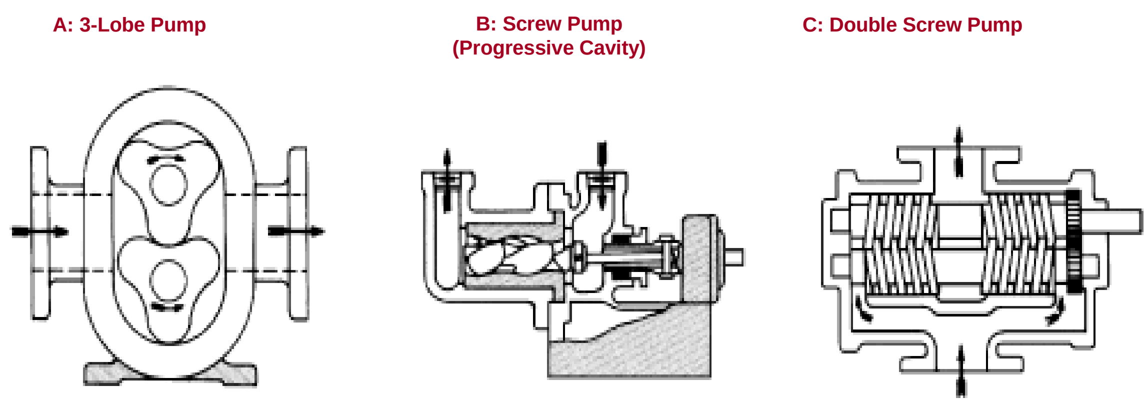


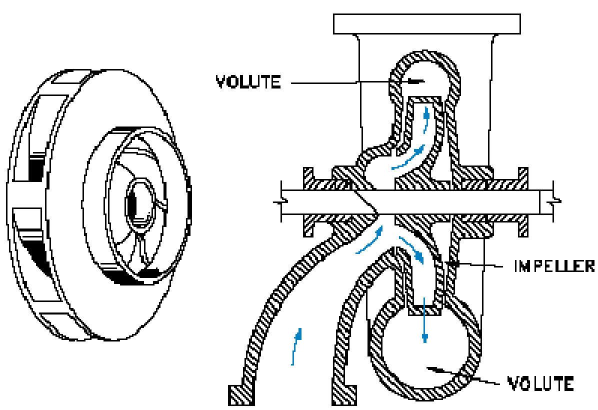
Fig.4.2 positive displacement pumps

3. Dynamic (impeller) pumps

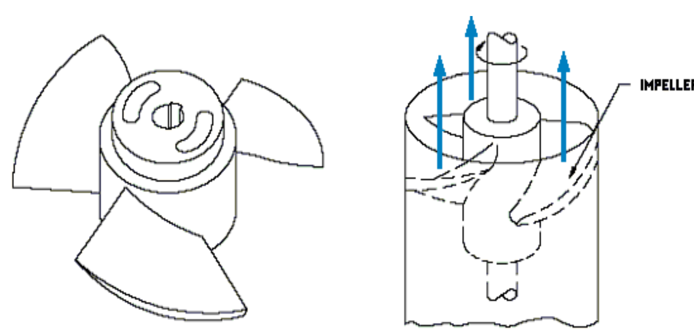
Such pumps deliver the fluid by adding momentum to it by means of fast moving blades or vanes. If the motion of the blades is a rotational one the machines are known as

rotodynamic pumps. The most common types of such pumps are classified based on the direction of flow relative to the axis of rotation. They can then be either radial exit flow (Centrifugal), or axial flow or mixed flow (some where between axial and radial).
For the same power input and efficiency the centrifugal type would generate a relatively large pressure head with a low discharge, the axial flow type a relatively large discharge at low head with the mixed flow having characteristics somewhere in between the other two.

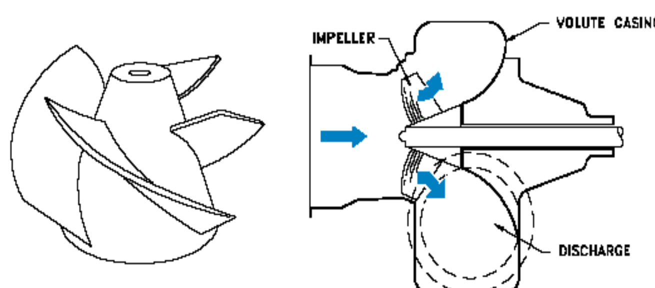
Dynamic pumps generally offer steadier and higher discharge than PDPs. The emphasis in this section shall be on rotodynamic pumps. Impeller pumps are by far the most widely used type of pumps in practice.



a. Centrifugal pump



b. Axial flow pump



c. Mixed flow pump

Pump- pipeline systems and basic definitions
A typical installation of a pump is shown in the figure below:

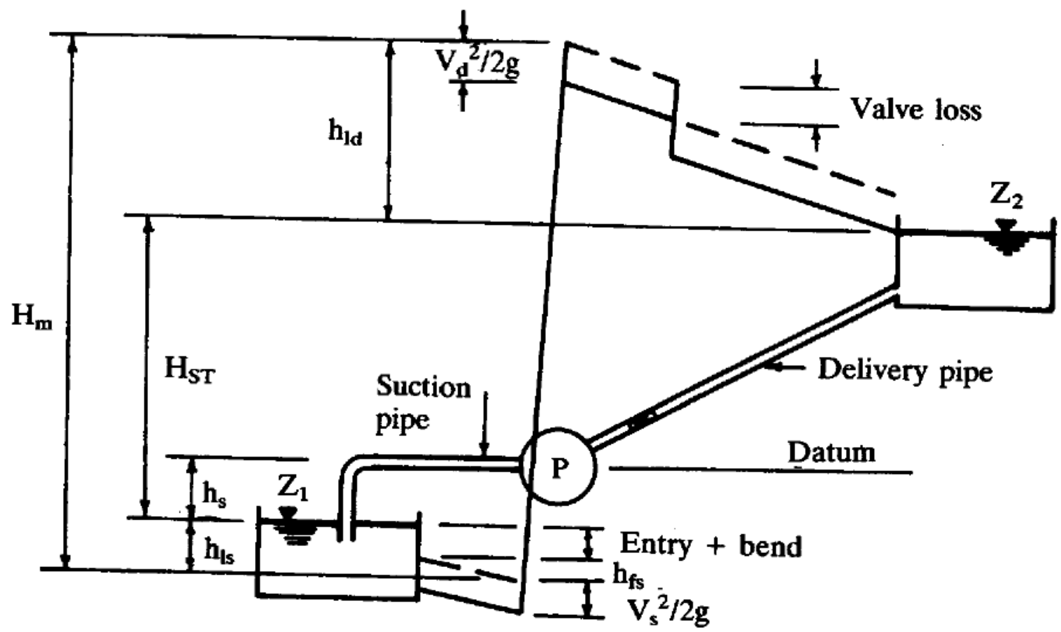


Fig.4.4 pump –pipe line

If one writes Bernoulli's equation between the pump inlet (suction side) and outlet (deliver side),

$$\frac{P_s}{\gamma} + Z_s + \frac{V_s^2}{2g} + H_m = \frac{P_d}{\gamma} + Z_d + \frac{V_d}{2g}$$

$$\rightarrow H_m = \frac{P_d}{\gamma} - \frac{P_s}{\gamma}$$

where the differences in velocity and elevation between inlet and outlet are ignored.
By applying Bernoulli's equation between lower tank and inlet, and outlet and upper reservoir one gets the realtion

$$H_m = H_{st} + h_{ls} + h_{ld}$$

Thus the pump has to deliver the static head plus the losses in the suction and delivery pipes.

Pump Characteristics

The essential hydraulic properties of a pump are laid down in the following basic pump characteristics, refer to the figure below.

- the Q-H characteristics, which indicates the relation between the flow Q and the head H at constant speed of the pump.
- the Q-η characteristics, which indicates the relation between the flow Q and the hydraulic efficiency η. The hydraulic efficiency η is the relation between the hydraulic energy absorbed by the fluid and the mechanical energy supplied by the motor via the draft shaft. $\eta = \frac{\gamma QH}{\omega T} = \frac{\gamma QH}{T(2\pi n/60)}$

  where n= speed of the shaft(1/min)

  T = shaft torque (Nm)

The power required for the drive is: $P_d = \frac{\gamma QH}{\eta \eta_d}$

where P_d = the power required for the drive (watt)

η_d = efficiency of the drive

- the Q-P characteristics, which indicates the relation between the flow Q and the power P that is supplied to the pump shaft.
- the net positive suction head (NPSH) characteristics, which indicates the relation between the flow Q and the margin that is required between the energy level at the suction side of the pump and the vapor pressure level of the pumped fluid, so that a certain amount of cavitation is not exceeded.

Pumps, once installed may have to operate under different combinations of Q and H. Hence the choice of pumps and drives always has to be based on the most unfavorable operating conditions, both with regard to the required power, head and efficiency and with regard to NPSH (cavitation). When a pump has to operate under a wide range of discharges a flat curve is to be preferred.

Pump characteristics of impeller pumps are generally determined by measurements at the pumps concerned, measurements at a small scale model of the pump or by extrapolation of the characteristics of an already existing pump that is identical in shape, with the help of hydraulic scaling laws.

If pumps a and b are two geometrically similar pumps operating under identical flow conditions, then the following similarity rules apply to them

$$\frac{P_b}{P_a} = \frac{\rho_b}{\rho_a}\left(\frac{n_b}{n_a}\right)^3\left(\frac{D_b}{D_a}\right)^5$$

$$\frac{Q_b}{Q_a} = \frac{n_b}{n_a}\left(\frac{D_b}{D_a}\right)^3$$

where Q flow rate (m^3/s)

n speed

D diameter of impeller

P power

Index a: pump whose characteristics are available

Index b: pump that is identical in shape, the characteristics of which are derived from existing pump.

Most pump applications involve a known head and discharge for the particular system, plus a speed range dictated by electric motor speeds or cavitation requirements. The designer then selects the best size and shape of the pump. Pumps exhibit varying efficiencies over a range of discharges and operating conditions. One of the most important parameters that are used in the selection of appropriate type of pump is specific speed. The specific speed (n_s) is a dimensionless ratio involving the rotational speed, the discharge (or power output for turbines) and the head the pump delivers. It is given by the expression: $n_s = \frac{N\sqrt{Q}}{(gH)^{3/4}}$

where N is the rotational speed in rad/s. In engineering practice, however, the specific speed is commonly defined as $N_s = \frac{N\sqrt{Q}}{H^{3/4}}$

where N is the speed of the pump shaft in rev/min,
Q is the pump flow with the maximum efficiency (best efficiency point) in m^3/s and
H is the head with maximum efficiency in meters.
Such definition of specific speed does not result in a dimensionless constant; hence its unit depends on the system of units used.
The specific speed is also used to classify pumps, the reason why the specific speed is also called **type number**.

Low specific speed	$N_s \leq 2600$	(centrifugal pumps)
Medium specific speed	$2600 < N_s \leq 5000$	(mixed flow pumps)
High specific speed	$5000 < N_s < 10000$	(axial flow pumps)

Note: the N_s is computed with Q in l/s and H in m.
A pump with low specific speed will generally render a relatively small flow with a relatively high head; where as a pump with a high specific speed will generally render a large flow with a relatively low head.

Suction and cavitation, the Net Positive Suction Head (NPSH)

Cavitation consists of local vaporization od a liquid due to a drop in its pressure below its vapour pressure. In a pump installation if the pump is located above the liquid surface of the reservoir from which it delivers then there could be a danger of cavitation. Cavitation causes physical damage, reduction in discharge and makes undesired noise.
Thus for cavitation not to occur the suction pressure (P_s) should not equal the vapour pressure under the given conditions.

If the suction pressure head is expressed in absolute terms (i.e. wrt absolute zero), then the Net Positive Suction Head (NPSH) is defined as

$$NPSH = \frac{P_s}{\gamma} - \frac{P_v}{\gamma} = \frac{P_a}{\gamma} - H_s - \frac{P_v}{\gamma}$$

where P_a is the local atmospheric pressure,

P_v = vapor pressure of the liquid at the given temperature,

H_s = the manometric suction head = $h_s + h_{ls} + \frac{V_s^2}{2g}$

V_s = velocity in the suction pipe.

Values of NPSH are delivered from the manufacturer of the pumps and should not be exceeded

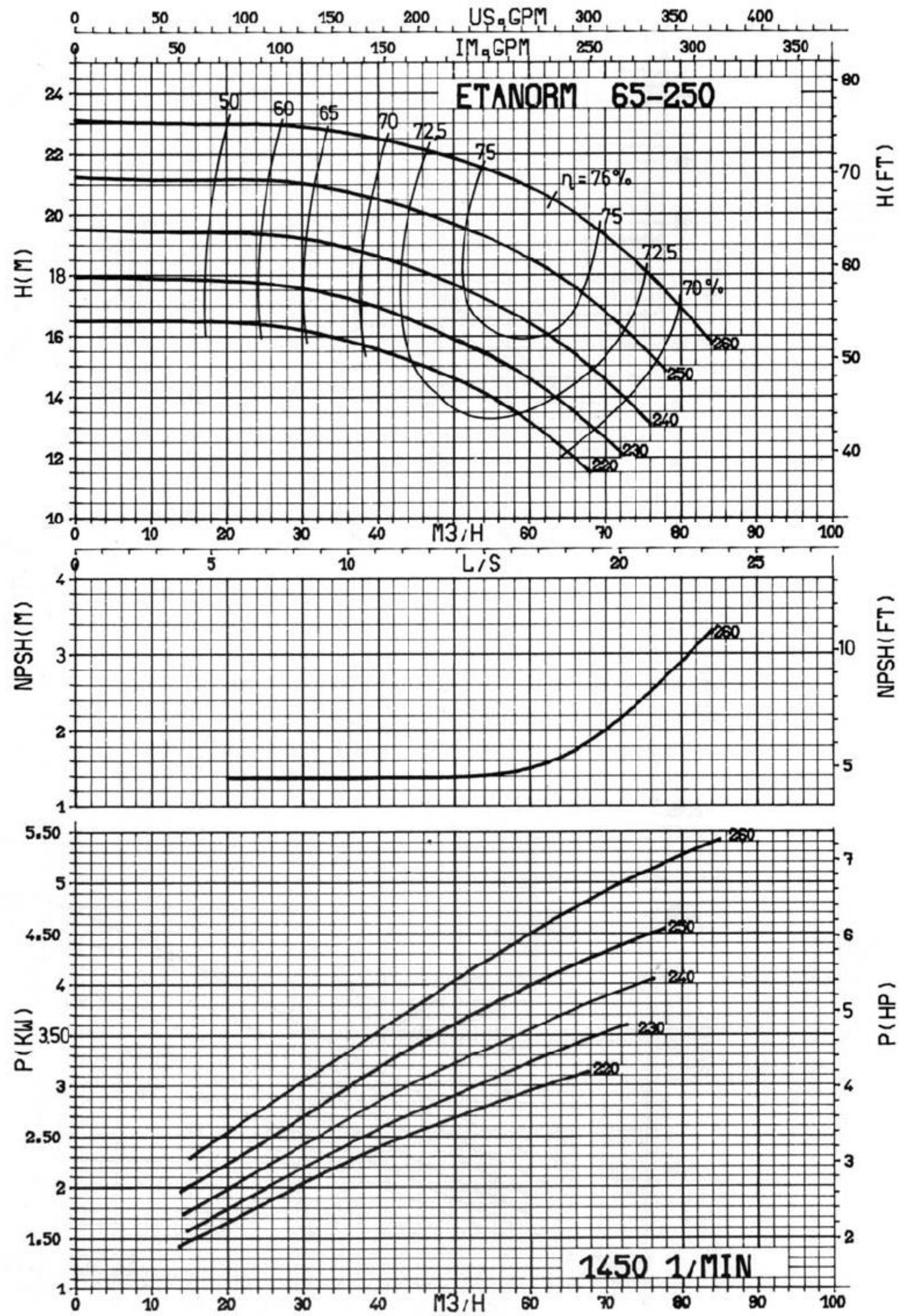

ETANORM 65-250
US. GPM
IM. GPM
H(M)
H(FT)
η=76%
70%
260
250
240
230
220
M3/H
L/S
NPSH(M)
NPSH(FT)
P(KW)
P(HP)
1450 1/MIN

Fig 4.5 pump characteristic curves

Pump- pipe line systems design

For effective design of a pump - pipe line system, basic knowledge of the hydrodynamic characteristics of pumps is required. Particularly the head curve (Q-H), the efficiency curve, (Q-η) and the power curve, (Q-P).

In order to transport a fluid through a pipe line, energy may have to be added to the fluid to overcome losses and to increase the potential energy of the fluid. The device that is most often used to achieve this is the pump.

In order to reach the optimal operation, a good consideration of the following items is required:

a. Accurate calculations of the required operating point or points (Q, H).
b. Possibility of deviation of actual operating point from the calculated operating point should be accurately assessed and catered for. This could happen as a result of bad prediction of the hydraulic losses of the pipeline system.
c. Determine the minimum available suction pressure with in the system and the vapor pressure of the pumped fluid, which will influence the choice of pump with regard to its cavitation problems.
d. optimal pump arrangement according to the available capabilities
e. select the most suitable pump type according to the system requirements

Consider the system shown in fig 4.4. The total energy the pump has to supply is the sum of the static lift (the elevation difference between the water surfaces in the two reservoirs) and the head loss in the suction and delivery pipes (h_{ls} +h_{ld}). This can be expressed as

$$E = H_{st} + kQ^2$$

Where the static lift is the elevation difference H_{st} and the head losses can be expressed by kQ^2. This equation (or its graphical form) is known as the **system characteristics**. It should be noted that the term kQ^2 includes all minor as well as major losses in the pipe system leading the water from the source to the destination.

For rotodynamic pumps, the head generated is not constant but is a function of discharge, as expressed by the Q-H curve. If such a pump is installed in a pipe system, then the two must handle the same flow rate, i.e. the flow through the pump is the same as that through the connected pipeline system. On the other hand the head generated by the pump must equal to the system energy requirement at that flow rate. Hence, the solution of

$$f(Q) = \Delta Z+kQ^2$$

gives the actual discharge for the particular installation. The solution is usually obtained graphically. The intersection point of the two curves is known as **the operating point**. Pump matching usually means the process of selecting a pump to operate in conjunction

with a given system so that it delivers the required flow rate, operating at its best efficiency, which corresponds to the pump's design point.
The point on the system characteristic which corresponds to the required rate through the system is known as the **duty required**. Thus, for correct matching the operating point should coincide with the system duty required.

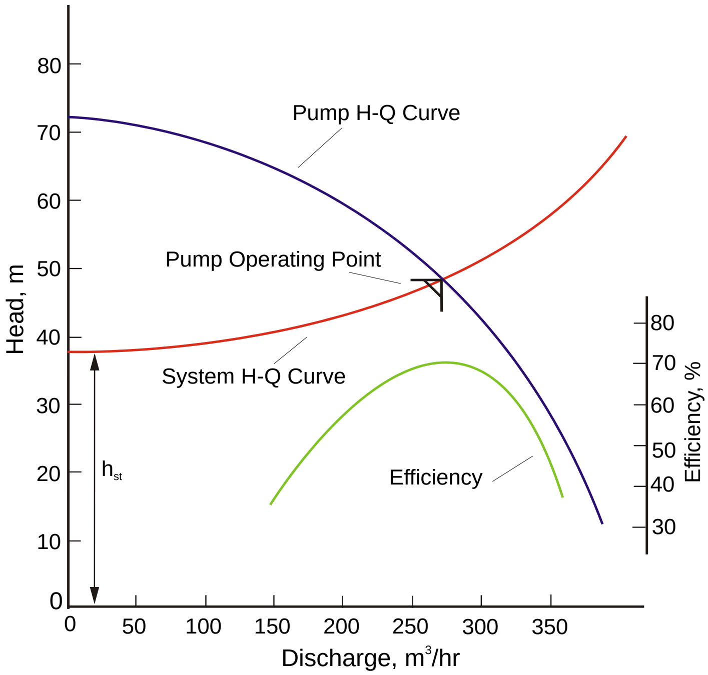


Fig. 4.6 pump operating point

Multiple pump systems
a. parallel connection
pumping stations usually contain several pumps connected in "parallel". This arrangement allows to deliver large range of discharges. In such a connection the head across all pumps is assumed to be equal and the discharges in the individual pumps is added up.

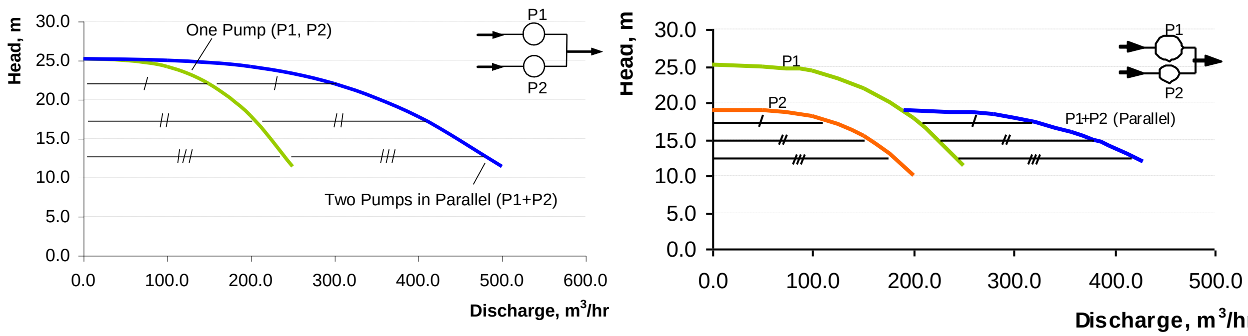


fig.4.7 H-Q curves for two pumps connected in parallel

b. series connection

This connection is the basis for multi-stage pumping. The discharge from the first pump is delivered to the inlet of the second pump and so on in order to lift the water over a high elevation which would have otherwise been difficult to achieve by using a single pump. The same discharge passes through each pump receiving a pressure boost in doing so. The resulting H-Q curve will be obtained by adding individual pump heads for the given discharges.

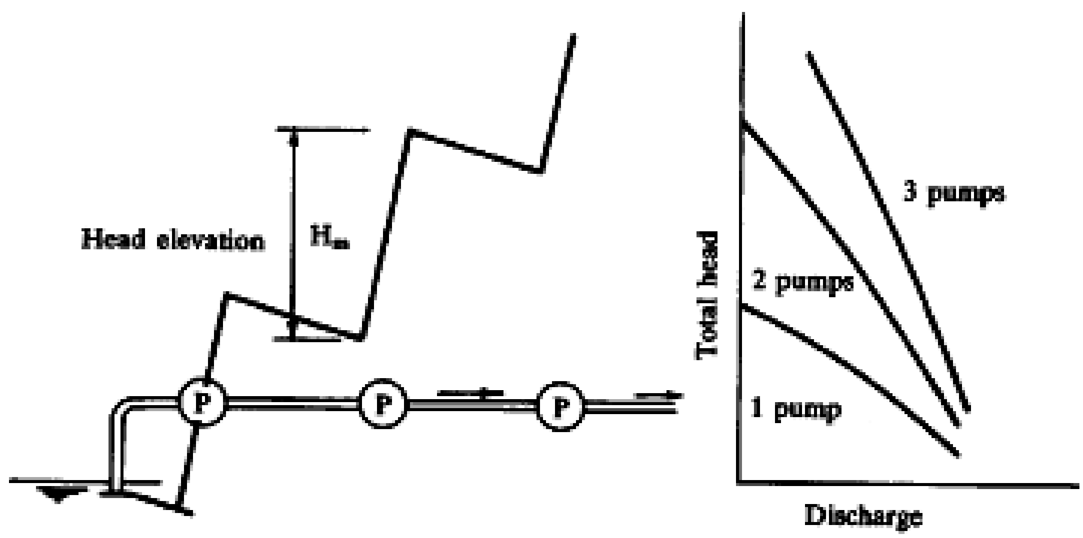


Fig.4.7 Pumps operating in series connection

Turbines

Turbines extract energy from a flowing fluid. Hydraulic turbines extract energy of flowing water and coupled wit generators convert it to electric power.

Hydraulic features

Classification based on head

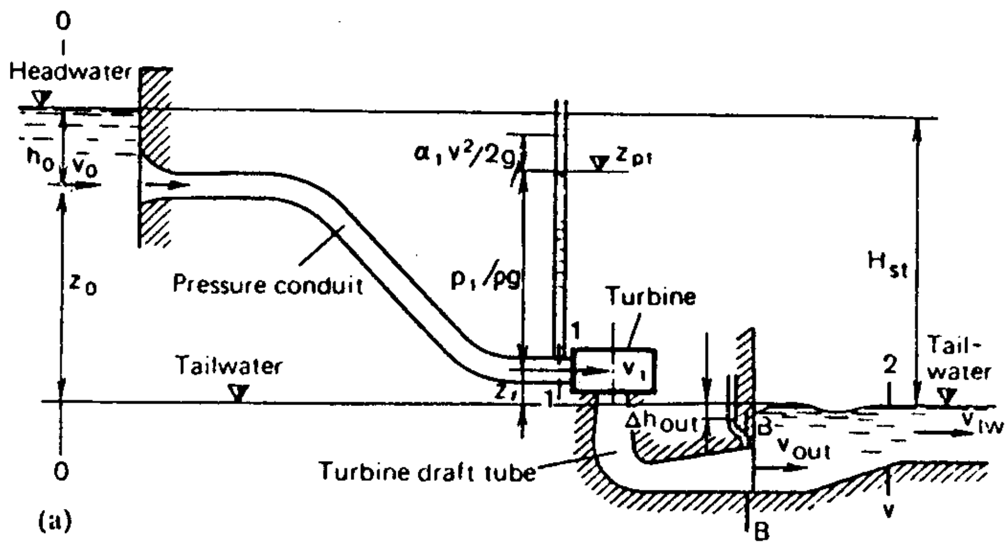
Low head 2-15 m

Medium head 15- 50m

High head >50m

The total power driven from a turbine is given by the equation

$$P = \eta\gamma QH$$



(a)

5 Introduction to water hammer analysis

The results of steady flow analysis discussed so far fail to deliver useful results when flow in a pipe is unsteady. Such flow conditions may occur during closure and opening of flow control valves. For instance, a gate or valve at the end of the penstock pipe (the high pressure pipe bringing water to turbines) controls the discharge of the turbine. As soon as this automatically operated gate opening is altered, the pipe flow has to be adjusted to the new magnitude of flow. In doing so, there are rapid pressure oscillations in the pipe, often accompanied by hammering like sound. Hence the phenomenon is called water hammer although it also occurs in other fluids.

Water hammer can be defined as the change in pressure above or below the normal working pressure caused by the change in the velocity of flow. Depending on the rate and magnitude of the velocity change, this can lead to high pressure fluctuations in such pipe lines as supply lines for drinking water, discharge line for sewerage water, oil-transmission line, penstock pipeline, etc.

The phenomenon is described taking an example of a simple pipe fitted with a valve at its end, as shown in the figure below:

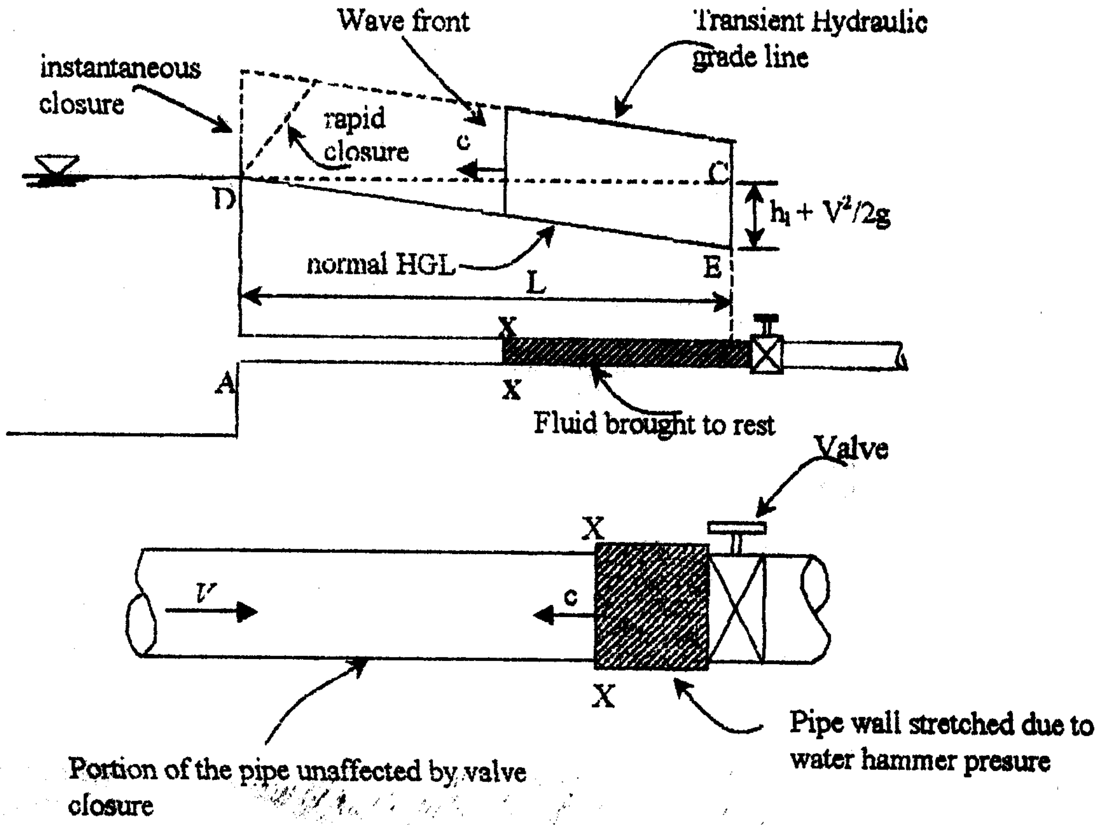


Fig 5.1 Conditions for instantaneous pipe closure

If the valve is suddenly closed, the fluid immediately upstream of the valve is stopped but the remaining fluid continues to move towards the valve as if nothing has happened. As the fluid in the immediate neighborhood of the valve has been brought to rest, it is slightly compressed by the fluid still flowing with its original velocity towards the valve. Not only will the fluid be compressed but also the wall of the pipe surrounding this will get stretched by the excess pressure produced. The next element of the fluid now finds an increased pressure with static condition in front of it, therefore, it to come to rest, is itself compressed and expands the pipe wall slightly. Each element of the fluid thus stops the element following it until all the fluid in the pipe has been brought to rest, figure above. Thus the pressure gets propagate in the upstream direction at the sonic wave speed. This speed is given by the following equation:

$$c = \sqrt{\frac{K}{\rho}}$$

Where K= the bulk modulus and ρ is the mass density of the fluid, if elasticity of the material is negligible.

If the elasticity of the pipe and that of the fluid are taken in to consideration a more accurate expression for the pressure wave velocity (also known as water hammer wave celerity) is

Where K = bulk modulus of fluid (2100 N/mm^2 for water)

E = elastic modulus of pipe material (210000 N/mm^2 for steel)

α = a coefficient which takes the values:

5/4 – ν for pipe supported at one end only

$1 - \nu^2$ for pipe with both ends fixed

1 for pipe with expansion joints, where ν is Poisson ratio

t and d are pipe thick ness and diameter respectively.

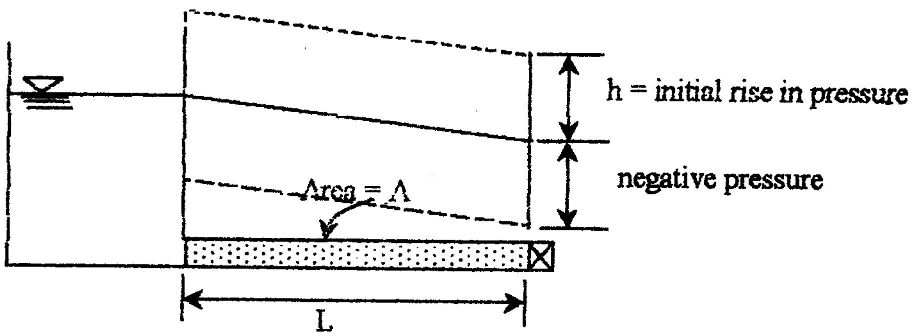
Hence at any instant after the closure of the valve before all the fluid has stopped there is a discontinuity of conditions (such as pressures, velocity, etc) represented by line XX in the figure above.

There are two approaches to the water hammer problem, namely the rigid eater column theory and the elastic water column theory. In the rigid water column theory the compressibility effects of water are neglected where as the latter one the same is taken in to consideration. The rigid water column theory is easier top apply but is useful only for specific cases where the compressibility of water can safely be neglected.

Thus the pressure wave takes a time of 2L/c to make a round trip between the valve and the inlet, if the length of the pipe is L. In practice if the closure time is longer than this time, the closure is said to be gradual and the rigid water column theory gives a fairly accurate result. This is because for a gradual closure the resulting pressure is relatively low and hence the associated change in volume of the flowing fluid is also small. If the time of closure is less than 2L/c, however, compressibility effects cannot be neglected hence the rigid water column theory gives an approximate result. In practice if the closure is rapid, i.e. time of closure is less than 2L/c, a more accurate method is adopted which takes into consideration the compressibility of the fluid and elastic deformation of the pipe.

When the pressure wave reaches the inlet at A, the entire fluid mass in pipe length L comes to rest and subjected to an excess pressure throughout. There will be a transient hydraulic grade line parallel to the original hydraulic grade line but at a distance Δp/γ, where Δp is the excess water hammer pressure. When the pressure wave reaches the tank it gets relieved and drops to that corresponding to the water surface in the tank. Hence an unbalanced pressure head is created, since the water in the pipe is under excess pressure. This causes the water to flow back into the tank. This flow will not last long, however, as it creates vacuum immediately upstream of the valve and the flow direction is reversed. Hence the pressure just upstream of the valve swings from positive to negative pressure, which is then propagated towards the inlet.

Rigid water column theory: It is assumed in this method of analysis that any increase in pressure change is transmitted instantaneously throughout the fluid in the conduit, disregarding the compressibility of the fluid and the elasticity of the conduit.
To derive the basic equation using the rigid water column theory, consider the following simple case where pipe fitted with valve at one of its ends conveys liquid from a tank.



Considering the entire water mass in the pipe, which is the approach used in rigid water column theory, the unbalanced head from the right is h. Here frictional effects are neglected, again one of the limitations of this approach. Under the action of the unbalanced force due to the excess pressure the motion of the liquid mass is retarded, hence one can write
From Newton's second law,

$$h\gamma A = -\rho AL\frac{dV}{dt}$$

From which

$$h = -\frac{L}{g}\frac{dV}{dt}$$

The minus sign is used because the velocity is decreasing with time.
This equation can be used for calculating the head rise associated wit slow deceleration of water column. It may be used for calculating the water level variation in a surge shaft (tank) following load rejection, starting up in pumping line, etc.

When the compressibility of the fluid and the pipe material is taken in to account (Semi-elastic approach), if the pipe closure is uniform, and the entire mass of fluid between the inlet and the valve reduces its velocity in time Δt, then

$$h = \frac{L}{g}\frac{\Delta V}{\Delta t} \qquad \text{But} \qquad \Delta t = \frac{L}{c}$$

And $h = \frac{c}{g}\Delta V$ and if the valve closure is complete, $h = \frac{c}{g}V$

Valve closure cannot occur in reality with in zero time. Hence classification of closure time in to rapid and slow is adopted. If the closure time is less than 2L/c, the closure is classified as rapid and the above equation for maximum water hammer pressure can be used. If, however, the closure time is greater than 2L/c then due to the reflected negative pressure wave the maximum pressure is reduced. Here it is proposed to reduce the maximum pressure according to time proportion, i.e.

$$\frac{h}{h_{max}} = \frac{2L}{c.t_c}$$

This equation gives the values of pressure at the closing point that is required. Instead the pressure wave propagation and change in velocity are both needed. In addition practical flow conditions are much more complex than that used in the derivation of the simple formula given above. In such situations more accurate derivation of equations of motion is needed, which is beyond the scope of the course.