

Basics of Fluid Mechanics

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'We are like dwarfs sitting on the shoulders of giants'

from The Metalogicon by John in 1159

CONTENTS

Nomenclature	xiii
GNU Free Documentation License	xix
1. APPLICABILITY AND DEFINITIONS	xx
2. VERBATIM COPYING	xxi
3. COPYING IN QUANTITY	xxi
4. MODIFICATIONS	xxii
5. COMBINING DOCUMENTS	xxiv
6. COLLECTIONS OF DOCUMENTS	xxiv
7. AGGREGATION WITH INDEPENDENT WORKS	xxv
8. TRANSLATION	xxv
9. TERMINATION	xxv
10. FUTURE REVISIONS OF THIS LICENSE	xxv
ADDENDUM: How to use this License for your documents	xxvi
How to contribute to this book	xxvii
Credits	xxvii
Steven from artofproblemsolving.com	xxvii
Dan H. Olson	xxviii
Richard Hackbarth	xxviii
John Herbolenes	xxviii
Eliezer Bar-Meir	xxviii
Henry Schoumertate	xxviii
Your name here	xxviii
Typo corrections and other "minor" contributions	xxix
Version 0.1.8 August 6, 2008	xxxix
pages 189 size 2.6M	xxxix
Version 0.1 April 22, 2008	xxxix

pages 151 size 1.3M	xxxix
Properties	xl
Open Channel Flow	xl
1 Introduction	1
1.1 What is Fluid Mechanics?	1
1.2 Brief History	3
1.3 Kinds of Fluids	5
1.4 Shear Stress	6
1.5 Viscosity	9
1.5.1 General	9
1.5.2 Non-Newtonian Fluids	10
1.5.3 Kinematic Viscosity	11
1.5.4 Estimation of The Viscosity	12
1.5.5 Bulk Modulus	20
1.6 Surface Tension	22
1.6.1 Wetting of Surfaces	24
2 Review of Thermodynamics	33
2.1 Basic Definitions	33
3 Review of Mechanics	41
3.1 Center of Mass	41
3.1.1 Center of the Mass	41
3.1.2 Center of Area	42
3.2 Moment of Inertia	43
3.2.1 Moment of Inertia for Mass	43
3.2.2 Moment of Inertia for Area	44
3.2.3 Examples of Moment of Inertia	46
3.2.4 Product of Inertia	49
3.2.5 Principal Axes of Inertia	50
3.3 Newton's Laws of Motion	51
3.4 Angular Momentum and Torque	51
3.4.1 Tables of geometries	52
4 Fluids Statics	55
4.1 Introduction	55
4.2 The Hydrostatic Equation	55
4.3 Pressure and Density in a Gravitational Field	57
4.3.1 Constant Density in Gravitational Field	57
4.3.2 Pressure Measurement	58
4.3.3 Varying Density in a Gravity Field	62
4.3.4 The Pressure Effects Because Temperature Variations	65
4.3.5 Gravity Variations Effects on Pressure and Density	69
4.3.6 Liquid Phase	71

CONTENTS

v

4.4	Fluid in a Accelerated System	72
4.4.1	Fluid in a Linearly Accelerated System	72
4.4.2	Angular Acceleration Systems: Constant Density	74
4.5	Fluid Forces on Surfaces	76
4.5.1	Fluid Forces on Straight Surfaces	76
4.5.2	Force on Curved Surfaces	86
4.6	Buoyancy and Stability	92
4.6.1	Stability	100
4.6.2	Surface Tension	110
4.7	Rayleigh–Taylor Instability	110
I	Integral Analysis	117
5	Mass Conservation	119
5.1	Introduction	119
5.2	Control Volume	120
5.3	Continuity Equation	121
5.3.1	Non Deformable Control Volume	123
5.3.2	Constant Density Fluids	123
5.4	Reynolds Transport Theorem	130
5.5	Examples For Mass Conservation	132
5.6	The Details Picture – Velocity Area Relationship	138
5.7	More Examples for Mass Conservation	141
6	Momentum Conservation	143
6.1	Momentum Governing Equation	143
6.1.1	Introduction to Continuous	143
6.1.2	External Forces	144
6.1.3	The Momentum Governing Equation	145
6.1.4	Momentum Equation in Acceleration System	145
6.1.5	Momentum For Steady State and Uniform Flow	146
6.2	Momentum Equation Application	150
6.2.1	Momentum for Unsteady State and Uniform Flow	153
6.2.2	Momentum Application to Unsteady State	153
6.3	Conservation Moment Of Momentum	160
6.4	More Examples on Momentum Conservation	162
6.4.1	Qualitative Questions	164
7	Energy Conservation	167
7.1	The First Law of Thermodynamics	167
7.2	Limitation of Integral Approach	179
7.3	Approximation of Energy Equation	180
7.3.1	Energy Equation in Steady State	181
7.3.2	Energy Equation in Frictionless Flow and Steady State	181

7.4	Energy Equation in Accelerated System	183
7.4.1	Energy in Linear Acceleration Coordinate	183
7.4.2	Linear Accelerated System	184
7.4.3	Energy Equation in Rotating Coordinate System	184
7.4.4	Energy Equation in Accelerated Coordinate with Uniform Flow	186
II	Differential Analysis	187
8	Multi-Phase Flow	189
8.1	Introduction	189
8.2	History	189
8.3	What to Expect From This Chapter	190
8.4	Kind of Multi-Phase Flow	191
8.5	Classification of Liquid-Liquid Flow Regimes	192
8.5.1	Co-Current Flow	193
8.6	Multi-Phase Flow Variables Definitions	197
8.6.1	Multi-Phase Averaged Variables Definitions	198
8.7	Homogeneous Models	201
8.7.1	Pressure Loss Components	202
8.7.2	Lockhart Martinelli Model	204
8.8	Solid-Liquid Flow	205
8.8.1	Solid Particles with Heavier Density $\rho_S > \rho_L$	206
8.8.2	Solid With Lighter Density $\rho_S < \rho$ and With Gravity	208
8.9	Counter-Current Flow	209
8.9.1	Horizontal Counter-Current Flow	211
8.9.2	Flooding and Reversal Flow	212
8.10	Multi-Phase Conclusion	219
A	Mathematics For Fluid Mechanics	221
A.1	Ordinary Differential Equations	221
A.1.1	First Order Differential Equations	221
A.1.2	Variable Separation	221
A.1.3	The Integral Factor	222
A.1.4	Non-Linear Equation	223
A.1.5	Second Order Differential equations	225
A.1.6	Non-Homogeneous Equation	225
A.1.7	Non-Linear Second Order Equation	226
A.2	Vectors	228
A.2.1	Vector Algebra	228
A.3	Trigonometry	230
	Index	233
	Subjects Index	233
	Authors Index	236

LIST OF FIGURES

1.1	Diagram to explain part of relationships of fluid mechanics branches.	2
1.2	Density as a function of the size of sample.	6
1.3	Schematics to describe the shear stress in fluid mechanics.	6
1.4	The deformation of fluid due to shear stress.	7
1.5	The difference of power fluids.	9
1.6	Nitrogen and Argon viscosity.	10
1.7	The shear stress as a function of the shear rate.	10
1.8	Air viscosity as a function of the temperature.	11
1.9	Water viscosity as a function temperature.	12
1.10	Liquid metals viscosity as a function of the temperature.	15
1.11	Reduced viscosity as function of the reduced temperature.	17
1.12	Reduced viscosity as function of the reduced temperature.	18
1.13	Surface Tension control volume analysis.	21
1.14	Forces in Contact angle.	24
1.15	Description of wetting and non-wetting fluids.	24
1.16	Description of liquid surface.	26
1.17	The raising height as a function of the radii.	29
1.18	The raising height as a function of the radius.	29
3.1	Description of how the center of mass is calculated.	42
3.2	Thin body center of mass/area schematic.	42
3.3	The schematic that explains the summation of moment of inertia.	44
3.4	The schematic to explain the summation of moment of inertia.	45
3.5	Cylinder with the element for calculation moment of inertia.	45
3.6	Description of rectangular in x-y plane.	46
3.7	A square element for the calculations of inertia.	47
3.8	The ratio of the moment of inertia 2D to 3D.	47

3.9	Description of parabola - moment of inertia and center of area.	48
3.10	Product of inertia for triangle	50
4.1	Description of a fluid element in accelerated system.	55
4.2	Pressure lines a static fluid with a constant density.	58
4.3	A schematic to explain the measure of the atmospheric pressure.	58
4.4	Schematic of gas measurement utilizing the "U" tube.	59
4.5	Schematic of sensitive measurement device.	60
4.6	Hydrostatic pressure when there is compressibility in the liquid phase.	64
4.7	Two adjoin layers for stability analysis.	67
4.8	The varying gravity effects on density and pressure.	69
4.9	The effective gravity is for accelerated cart.	73
4.10	A cart slide on inclined plane.	73
4.11	Forces diagram of cart sliding on inclined plane.	74
4.12	Schematic to explain the angular angle.	74
4.13	Schematic angular angle to explain example 4.5	75
4.14	Rectangular area under pressure.	76
4.15	Schematic of submerged area.	78
4.16	The general forces acting on submerged area.	78
4.17	The general forces acting on non symmetrical straight area.	80
4.18	The general forces acting on non symmetrical straight area.	81
4.19	The effects of multi layers density on static forces.	84
4.20	The forces on curved area.	86
4.21	Schematic of Net Force on floating body.	86
4.22	Dam is a part of a circular shape.	88
4.23	Area above the dam arc subtract triangle.	88
4.24	Area above the dam arc calculation for the center.	89
4.25	Moment on arc element around Point "O."	90
4.26	Polynomial shape dam description.	91
4.27	The difference between the slop and the direction angle.	91
4.28	Schematic of Immersed Cylinder.	93
4.29	The floating forces on Immersed Cylinder.	93
4.30	Schematic of a thin wall floating body.	94
4.31	Schematic of floating bodies.	100
4.32	Schematic of floating cubic.	100
4.33	Stability analysis of floating body.	100
4.34	Cubic body dimensions for stability analysis.	102
4.35	Stability of cubic body infinity long.	103
4.36	The maximum height reverse as a function of density ratio.	103
4.37	Stability of two triangles put tougher.	104
4.38	The effects of liquid movement on the \overline{GM}	106
4.39	Measurement of \overline{GM} of floating body.	107
4.40	Calculations of \overline{GM} for abrupt shape body.	108
4.41	A heavy needle is floating on a liquid.	110

4.42	Description of depression to explain the Rayleigh–Taylor instability. . . .	111
4.43	Description of depression to explain the instability.	112
4.44	The cross section of the interface for max liquid.	114
4.45	Three liquids layers under rotation	115
5.1	Control volume and system in motion	119
5.2	Piston control volume	120
5.3	Schematics of velocities at the interface	121
5.4	Schematics of flow in a pipe with varying density	122
5.5	Filling of the bucket and choices of the control volumes	125
5.6	Height of the liquid for example 5.4	128
5.7	Boundary Layer control mass	133
5.8	Control volume usage to calculate local averaged velocity	138
5.9	Control volume and system in motion	139
5.10	Circular cross section for Finding U_x	140
5.11	Velocity for a circular shape	141
5.12	Boat for example 5.14	142
6.1	The explain for the direction relative to surface	144
6.2	Schematics of area impinged by a jet	147
6.3	Nozzle schematic for forces calculations	148
6.4	Propeller schematic to explain the change of momentum	150
6.5	Toy Sled pushed by the liquid jet	151
6.6	A rocket a with moving control volume	153
6.7	Schematic of a tank seating on wheels	155
6.8	A new control volume to find the velocity in discharge tank	156
6.9	The impeller of the centrifugal pump and the velocities diagram	161
6.10	Nozzle schematics water rocket	162
6.11	Flow out of un symmetrical tank	165
7.1	The work on the control volume	168
7.2	Discharge from a Large Container	170
7.3	Kinetic Energy and Averaged Velocity	171
7.4	The work on the control volume	180
8.1	Different fields of multi phase flow.	191
8.2	Stratified flow in horizontal tubes when the liquids flow is very slow. . .	193
8.3	Kind of Stratified flow in horizontal tubes.	194
8.4	Plug flow in horizontal tubes with the liquids flow is faster.	194
8.5	Modified Mandhane map for flow regime in horizontal tubes.	195
8.6	Gas and liquid in Flow in vertical tube against the gravity.	196
8.7	A dimensional vertical flow map low gravity against gravity.	197
8.8	The terminal velocity that left the solid particles.	207
8.9	The flow patterns in solid-liquid flow.	208
8.10	Counter–flow in vertical tubes map.	209

8.11 Counter-current flow in a can.	210
8.12 Image of counter-current flow in liquid-gas/solid-gas configurations. . .	210
8.13 Flood in vertical pipe.	211
8.14 A flow map to explain the horizontal counter-current flow.	212
8.15 A diagram to explain the flood in a two dimension geometry.	212
8.16 General forces diagram to calculated the in a two dimension geometry. .	218
A.1 The right hand rule	229

LIST OF TABLES

1	Books Under Potto Project	xxxvi
1.1	Sutherland's equation coefficients	13
1.2	Viscosity of selected gases	13
1.3	Viscosity of selected liquids	14
1.4	Properties at the critical stage	15
1.5	Bulk modulus for selected materials	20
1.6	The contact angle for air/water with selected materials.	25
1.7	The surface tension for selected materials.	31
1.7	continue	32
2.1	Properties of Various Ideal Gases [300K]	38
3.1	Moments of Inertia full shape.	53
3.2	Moment of inertia for various plane surfaces	54

NOMENCLATURE

- \bar{R} Universal gas constant, see equation (2.26), page 37
- τ The shear stress Tensor, see equation (6.7), page 144
- ℓ Units length., see equation (2.1), page 33
- \mathfrak{M} Angular Momentum, see equation (6.38), page 160
- μ viscosity at input temperature T, see equation (1.17), page 12
- μ_0 reference viscosity at reference temperature, T_{i0} , see equation (1.17), page 12
- F_{ext} External forces by non-fluids means, see equation (6.11), page 145
- U The velocity taken with the direction, see equation (6.1), page 143
- Ξ Martinelli parameter, see equation (8.43), page 205
- A The area of surface, see equation (4.117), page 86
- a The acceleration of object or system, see equation (4.0), page 55
- B_f Body force, see equation (2.9), page 35
- c.v.* subscribe for control volume, see equation (5.0), page 120
- C_p Specific pressure heat, see equation (2.23), page 37
- C_v Specific volume heat, see equation (2.22), page 37
- E_U Internal energy, see equation (2.3), page 34
- E_u Internal Energy per unit mass, see equation (2.6), page 34

E_i	System energy at state i , see equation (2.2), page 34
G	The gravitation constant, see equation (4.62), page 70
g_G	general Body force, see equation (4.0), page 55
H	Enthalpy, see equation (2.18), page 36
h	Specific enthalpy, see equation (2.18), page 36
k	the ratio of the specific heats, see equation (2.24), page 37
k_T	Fluid thermal conductivity, see equation (7.3), page 168
L	Angular momentum, see equation (3.38), page 51
P_{atmos}	Atmospheric Pressure, see equation (4.85), page 78
q	Energy per unit mass, see equation (2.6), page 34
Q_{12}	The energy transferred to the system between state 1 and state 2, see equation (2.2), page 34
R	Specific gas constant, see equation (2.27), page 38
S	Entropy of the system, see equation (2.13), page 36
S_{uth}	S_{uth} is Sutherland's constant and it is presented in the Table 1.1, see equation (1.17), page 12
T_τ	Torque, see equation (3.40), page 52
T_{i0}	reference temperature in degrees Kelvin, see equation (1.17), page 12
T_{in}	input temperature in degrees Kelvin, see equation (1.17), page 12
U	velocity, see equation (2.4), page 34
w	Work per unit mass, see equation (2.6), page 34
W_{12}	The work done by the system between state 1 and state 2, see equation (2.2), page 34
z	the coordinate in z direction, see equation (4.14), page 57
says	Subscribe says, see equation (5.0), page 120

The Book Change Log

Version 0.2.4

March 01, 2010 (2.9 M 280 pages)

- The energy conservation chapter was released.
- Some additions to mass conservation chapter on averaged velocity.
- Some additions to momentum conservation chapter.
- Additions to the mathematical appendix on vector algebra.
- Additions to the mathematical appendix on variables separation in second order ode equations.
- Add the macro `picText` to insert figure in lower right corner thanks to Steven from www.artofproblemsolving.com.
- Add the macro to improve emphases equation thanks to Steven from www.artofproblemsolving.com.
- Add example about the the third component of the velocity.
- English corrections, Thanks to Eliezer Bar-Meir

Version 0.2.3

Jan 01, 2010 (2.8 M 241 pages)

- The momentum conservation chapter was released.
- Corrections to Static Chapter.

- Add the macro ekes to equations in examples thanks to Steven from www.artofproblemsolving.com.
- English corrections, Thanks to Eliezer Bar-Meir

Version 0.1.9

Dec 01, 2009 (2.6 M 219 pages)

- The mass conservation chapter was released.
- Add Reynold's Transform explanation.
- Add example on angular rotation to statics chapter.
- Add the open question concept. Two open questions were released.
- English corrections, Thanks to Eliezer Bar-Meir

Version 0.1.8.5

Nov 01, 2009 (2.5 M 203 pages)

- First true draft for the mass conservation.
- Improve the dwarfing macro to allow flexibility with sub title.
- Add the first draft of the temperature-velocity diagram to the Therm's chapter.

Version 0.1.8.1

Sep 17, 2009 (2.5 M 197 pages)

- Continue fixing the long titles issues.
- Add some examples to static chapter.
- Add an example to mechanics chapter.

Version 0.1.8a

July 5, 2009 (2.6 M 183 pages)

- Fixing some long titles issues.
- Correcting the gas properties tables (thanks to Heru and Micheal)
- Move the gas tables to common area to all the books.

Version 0.1.8

Aug 6, 2008 (2.4 M 189 pages)

- Add the chapter on introduction to multi-phase flow
- Again additional improvement to the index (thanks to Irene).
- Add the Rayleigh–Taylor instability.
- Improve the doChap scrip to break up the book to chapters.

Version 0.1.6

Jun 30, 2008 (1.3 M 151 pages)

- Fix the English in the introduction chapter, (thanks to Touser).
- Improve the Index (thanks to Irene).
- Remove the multiphase chapter (it is not for public consumption yet).

Version 0.1.5a

Jun 11, 2008 (1.4 M 155 pages)

- Add the constant table list for the introduction chapter.
- Fix minor issues (English) in the introduction chapter.

Version 0.1.5

Jun 5, 2008 (1.4 M 149 pages)

- Add the introduction, viscosity and other properties of fluid.
- Fix very minor issues (English) in the static chapter.

Version 0.1.1

May 8, 2008 (1.1 M 111 pages)

- Major English corrections for the three chapters.
- Add the product of inertia to mechanics chapter.
- Minor corrections for all three chapters.

Version 0.1a

April 23, 2008

- The Thermodynamics chapter was released.
- The mechanics chapter was released.
- The static chapter was released (the most extensive and detailed chapter).

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- **Date(s) of contribution(s):** 1999 to present
- **Nature of contribution:** Original author.
- **Contact at:** barmeir at gmail.com

Steven from artofproblemsolving.com

- **Date(s) of contribution(s):** June 2005, Dec, 2009

- **Nature of contribution:** LaTeX formatting, help on building the useful equation and important equation macros.
- **Nature of contribution:** In 2009 creating the exEq macro to have different counter for example.

Dan H. Olson

- **Date(s) of contribution(s):** April 2008
- **Nature of contribution:** Some discussions about chapter on mechanics and correction of English.

Richard Hackbarth

- **Date(s) of contribution(s):** April 2008
- **Nature of contribution:** Some discussions about chapter on mechanics and correction of English.

John Herbolenes

- **Date(s) of contribution(s):** August 2009
- **Nature of contribution:** Provide some example for the static chapter.

Eliezer Bar-Meir

- **Date(s) of contribution(s):** Nov 2009, Dec 2009
- **Nature of contribution:** Correct many English mistakes Mass.
- **Nature of contribution:** Correct many English mistakes Momentum.

Henry Schoumertate

- **Date(s) of contribution(s):** Nov 2009
- **Nature of contribution:** Discussion on the mathematics of Reynolds Transforms.

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- **Date(s) of contribution(s):** Month and year of contribution
- **Nature of contribution:** Insert text here, describing how you contributed to the book.
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Typo corrections and other "minor" contributions

- **R. Gupta**, January 2008, help with the original **img** macro and other (LaTeX issues).
- **Tousher Yang** April 2008, review of statics and thermo chapters.

About This Author

Genick Bar-Meir holds a Ph.D. in Mechanical Engineering from University of Minnesota and a Master in Fluid Mechanics from Tel Aviv University. Dr. Bar-Meir was the last student of the late Dr. R.G.E. Eckert. Much of his time has been spend doing research in the field of heat and mass transfer (related to renewal energy issues) and this includes fluid mechanics related to manufacturing processes and design. Currently, he spends time writing books (there are already three very popular books) and softwares for the POTTO project (see Potto Prologue). The author enjoys to encourage his students to understand the material beyond the basic requirements of exams.

In his early part of his professional life, Bar-Meir was mainly interested in elegant models whether they have or not a practical applicability. Now, this author's views had changed and the virtue of the practical part of any model becomes the essential part of his ideas, books and software.

He developed models for Mass Transfer in high concentration that became a building blocks for many other models. These models are based on analytical solution to a family of equations¹. As the change in the view occurred, Bar-Meir developed models that explained several manufacturing processes such the rapid evacuation of gas from containers, the critical piston velocity in a partially filled chamber (related to hydraulic jump), application of supply and demand to rapid change power system and etc. All the models have practical applicability. These models have been extended by several research groups (needless to say with large research grants). For example, the Spanish Comision Interministerial provides grants TAP97-0489 and PB98-0007, and the CICYT and the European Commission provides 1FD97-2333 grants for minor aspects of that models. Moreover, the author's models were used in numerical works, in GM, British industry, Spain, and Canada.

In the area of compressible flow, it was commonly believed and taught that there is only weak and strong shock and it is continue by Prandtl–Meyer function. Bar–

¹Where the mathematicians were able only to prove that the solution exists.

Meir discovered the analytical solution for oblique shock and showed that there is a quiet buffer between the oblique shock and Prandtl–Meyer. He also build analytical solution to several moving shock cases. He described and categorized the filling and evacuating of chamber by compressible fluid in which he also found analytical solutions to cases where the working fluid was ideal gas. The common explanation to Prandtl–Meyer function shows that flow can turn in a sharp corner. Engineers have constructed design that based on this conclusion. Bar-Meir demonstrated that common Prandtl–Meyer explanation violates the conservation of mass and therefor the turn must be around a finite radius. The author's explanations on missing diameter and other issues in fanno flow and “naughty professor's question” are used in the industry.

In his book “Basics of Fluid Mechanics”, Bar-Meir demonstrated that fluids must have wavy surface when the materials flow together. All the previous models for the flooding phenomenon did not have a physical explanation to the dryness. He built a model to explain the flooding problem (two phase flow) based on the physics. He also constructed and explained many new categories for two flow regimes.

The author lives with his wife and three children. A past project of his was building a four stories house, practically from scratch. While he writes his programs and does other computer chores, he often feels clueless about computers and programing. While he is known to look like he knows about many things, the author just know to learn quickly. The author spent years working on the sea (ships) as a engine sea officer but now the author prefers to remain on solid ground.

Prologue For The POTTO Project

This books series was born out of frustrations in two respects. The first issue is the enormous price of college textbooks. It is unacceptable that the price of the college books will be over \$150 per book (over 10 hours of work for an average student in The United States).

The second issue that prompted the writing of this book is the fact that we as the public have to deal with a corrupted judicial system. As individuals we have to obey the law, particularly the copyright law with the “infinite²” time with the copyright holders. However, when applied to “small” individuals who are not able to hire a large legal firm, judges simply manufacture facts to make the little guy lose and pay for the defense of his work. On one hand, the corrupted court system defends the “big” guys and on the other hand, punishes the small “entrepreneur” who tries to defend his or her work. It has become very clear to the author and founder of the POTTO Project that this situation must be stopped. Hence, the creation of the POTTO Project. As R. Kook, one of this author’s sages, said instead of whining about arrogance and incorrectness, one should increase wisdom. This project is to increase wisdom and humility.

The Potto Project has far greater goals than simply correcting an abusive Judicial system or simply exposing abusive judges. It is apparent that writing textbooks especially for college students as a cooperation, like an open source, is a new idea³. Writing a book in the technical field is not the same as writing a novel. The writing of a technical book is really a collection of information and practice. There is always someone who can add to the book. The study of technical material isn’t only done by having to memorize the material, but also by coming to understand and be able to solve

²After the last decision of the Supreme Court in the case of Eldred v. Ashcroft (see <http://cyber.law.harvard.edu/openlaw/eldredvashcroft> for more information) copyrights practically remain indefinitely with the holder (not the creator).

³In some sense one can view the encyclopedia Wikipedia as an open content project (see http://en.wikipedia.org/wiki/Main_Page). The wikipedia is an excellent collection of articles which are written by various individuals.

related problems. The author has not found any technique that is more useful for this purpose than practicing the solving of problems and exercises. One can be successful when one solves as many problems as possible. To reach this possibility the collective book idea was created/adapted. While one can be as creative as possible, there are always others who can see new aspects of or add to the material. The collective material is much richer than any single person can create by himself.

The following example explains this point: The army ant is a kind of carnivorous ant that lives and hunts in the tropics, hunting animals that are even up to a hundred kilograms in weight. The secret of the ants' power lies in their collective intelligence. While a single ant is not intelligent enough to attack and hunt large prey, the collective power of their networking creates an extremely powerful intelligence to carry out this attack⁴. When an insect which is blind can be so powerful by networking, so can we in creating textbooks by this powerful tool.

Why would someone volunteer to be an author or organizer of such a book? This is the first question the undersigned was asked. The answer varies from individual to individual. It is hoped that because of the open nature of these books, they will become the most popular books and the most read books in their respected field. For example, the books on compressible flow and die casting became the most popular books in their respective area. In a way, the popularity of the books should be one of the incentives for potential contributors. The desire to be an author of a well-known book (at least in his/her profession) will convince some to put forth the effort. For some authors, the reason is the pure fun of writing and organizing educational material. Experience has shown that in explaining to others any given subject, one also begins to better understand the material. Thus, contributing to these books will help one to understand the material better. For others, the writing of or contributing to this kind of books will serve as a social function. The social function can have at least two components. One component is to come to know and socialize with many in the profession. For others the social part is as simple as a desire to reduce the price of college textbooks, especially for family members or relatives and those students lacking funds. For some contributors/authors, in the course of their teaching they have found that the textbook they were using contains sections that can be improved or that are not as good as their own notes. In these cases, they now have an opportunity to put their notes to use for others. Whatever the reasons, the undersigned believes that personal intentions are appropriate and are the author's/organizer's private affair.

If a contributor of a section in such a book can be easily identified, then that contributor will be the copyright holder of that specific section (even within question/answer sections). The book's contributor's names could be written by their sections. It is not just for experts to contribute, but also students who happened to be doing their homework. The student's contributions can be done by adding a question and perhaps the solution. Thus, this method is expected to accelerate the creation of these high quality books.

These books are written in a similar manner to the open source software

⁴see also in Franks, Nigel R.; "Army Ants: A Collective Intelligence," *American Scientist*, 77:139, 1989 (see for information <http://www.ex.ac.uk/bugclub/raiders.html>)

process. Someone has to write the skeleton and hopefully others will add “flesh and skin.” In this process, chapters or sections can be added after the skeleton has been written. It is also hoped that others will contribute to the question and answer sections in the book. But more than that, other books contain data⁵ which can be typeset in L^AT_EX. These data (tables, graphs and etc.) can be redone by anyone who has the time to do it. Thus, the contributions to books can be done by many who are not experts. Additionally, contributions can be made from any part of the world by those who wish to translate the book.

It is hoped that the books will be error-free. Nevertheless, some errors are possible and expected. Even if not complete, better discussions or better explanations are all welcome to these books. These books are intended to be “continuous” in the sense that there will be someone who will maintain and improve the books with time (the organizer(s)).

These books should be considered more as a project than to fit the traditional definition of “plain” books. Thus, the traditional role of author will be replaced by an organizer who will be the one to compile the book. The organizer of the book in some instances will be the main author of the work, while in other cases only the gate keeper. This may merely be the person who decides what will go into the book and what will not (gate keeper). Unlike a regular book, these works will have a version number because they are alive and continuously evolving.

In the last 5 years three textbooks have been constructed which are available for download. These books contain innovative ideas which make some chapters the best in the world. For example, the chapters on Fanno flow and Oblique shock contain many original ideas such as the full analytical solution to the oblique shock, many algorithms for calculating Fanno flow parameters which are not found in any other book. In addition, Potto has auxiliary materials such as the gas dynamics tables (the largest compressible flow tables collection in the world), Gas Dynamics Calculator (Potto-GDC), etc.

The combined number downloads of these books is over half a million (December 2009) or in a rate of 20,000 copies a month. Potto books on compressible flow and fluid mechanics are used as the main textbook or as a reference book in several universities around the world. The books are used in more than 165 different countries around the world. Every month people from about 110 different countries download these books. The book on compressible flow is also used by “young engineers and scientists” in NASA according to Dr. Farassat, NASA Langley Research Center.

The undersigned of this document intends to be the organizer/author/coordinator of the projects in the following areas:

⁵ Data are not copyrighted.

Table -1. Books under development in Potto project.

Project Name	Progress	Remarks	Version	Availability for Public Download
Compressible Flow	beta		0.4.8.2	✓
Die Casting	alpha		0.0.3	✓
Dynamics	NSY		0.0.0	✗
Fluid Mechanics	alpha		0.1.1	✓
Heat Transfer	NSY	Based on Eckert	0.0.0	✗
Mechanics	NSY		0.0.0	✗
Open Channel Flow	NSY		0.0.0	✗
Statics	early alpha	first chapter	0.0.1	✗
Strength of Material	NSY		0.0.0	✗
Thermodynamics	early alpha		0.0.01	✗
Two/Multi phases flow	NSY	Tel-Aviv' notes	0.0.0	✗

NSY = Not Started Yet

The meaning of the progress is as:

- The Alpha Stage is when some of the chapters are already in a rough draft;
- in Beta Stage is when all or almost all of the chapters have been written and are at least in a draft stage;
- in Gamma Stage is when all the chapters are written and some of the chapters are in a mature form; and
- the Advanced Stage is when all of the basic material is written and all that is left are aspects that are active, advanced topics, and special cases.

The mature stage of a chapter is when all or nearly all the sections are in a mature stage and have a mature bibliography as well as numerous examples for every section. The mature stage of a section is when all of the topics in the section are written, and all of the examples and data (tables, figures, etc.) are already presented. While some terms are defined in a relatively clear fashion, other definitions give merely a hint on the status. But such a thing is hard to define and should be enough for this stage.

The idea that a book can be created as a project has mushroomed from the open source software concept, but it has roots in the way science progresses. However, traditionally books have been improved by the same author(s), a process in which books

have a new version every a few years. There are book(s) that have continued after their author passed away, i.e., the *Boundary Layer Theory* originated⁶ by Hermann Schlichting but continues to this day. However, projects such as the Linux Documentation project demonstrated that books can be written as the cooperative effort of many individuals, many of whom volunteered to help.

Writing a textbook is comprised of many aspects, which include the actual writing of the text, writing examples, creating diagrams and figures, and writing the \LaTeX macros⁷ which will put the text into an attractive format. These chores can be done independently from each other and by more than one individual. Again, because of the open nature of this project, pieces of material and data can be used by different books.

⁶Originally authored by Dr. Schlichting, who passed way some years ago. A new version is created every several years.

⁷One can only expect that open source and readable format will be used for this project. But more than that, only \LaTeX , and perhaps troff, have the ability to produce the quality that one expects for these writings. The text processes, especially \LaTeX , are the only ones which have a cross platform ability to produce macros and a uniform feel and quality. Word processors, such as OpenOffice, Abiword, and Microsoft Word software, are not appropriate for these projects. Further, any text that is produced by Microsoft and kept in "Microsoft" format are against the spirit of this project In that they force spending money on Microsoft software.

Prologue For This Book

Version 0.1.8 August 6, 2008

pages 189 size 2.6M

When this author was an undergraduate student, he spend time to study the wave phenomenon at the interface of open channel flow. This issue is related to renewal energy of extracting energy from brine solution (think about the Dead Sea, so much energy). The common explanation to the wave existence was that there is always a disturbance which causes instability. This author was bothered by this explanation. Now, in this version, it was proven that this wavy interface is created due to the need to satisfy the continuous velocity and shear stress at the interface and not a disturbance.

Potto project books are characterized by high quality which marked by presentation of the new developments and clear explanations. This explanation (on the wavy interface) demonstrates this characteristic of Potto project books. The introduction to multi-phase is another example to this quality. While it is a hard work to discover and develop and bring this information to the students, it is very satisfying for the author. The number of downloads of this book results from this quality. Even in this early development stage, number of downloads per month is about 5000 copies.

Version 0.1 April 22, 2008

pages 151 size 1.3M

The topic of fluid mechanics is common to several disciplines: mechanical engineering, aerospace engineering, chemical engineering, and civil engineering. In fact, it is also related to disciplines like industrial engineering, and electrical engineering. While the emphasis is somewhat different in this book, the common material is presented and hopefully can be used by all. One can only admire the wonderful advances done by the

previous geniuses who work in this field. In this book it is hoped to insert, what and when a certain model is suitable than other models.

One of the difference in this book is the insertion of the introduction to multiphase flow. Clearly, multiphase is an advance topic. However, some minimal familiarity can be helpful for many engineers who have to deal with non pure single phase fluid.

This book is the third book in the series of POTTO project books. POTTO project books are open content textbooks so everyone are welcome to joint in. The topic of fluid mechanics was chosen just to fill the introduction chapter to compressible flow. During the writing it became apparent that it should be a book in its own right. In writing the chapter on fluid statics, there was a realization that it is the best chapter written on this topic. It is hoped that the other chapters will be as good this one.

This book is written in the spirit of my adviser and mentor E.R.G. Eckert. Eckert, aside from his research activity, wrote the book that brought a revolution in the education of the heat transfer. Up to Egret's book, the study of heat transfer was without any dimensional analysis. He wrote his book because he realized that the dimensional analysis utilized by him and his adviser (for the post doc), Ernst Schmidt, and their colleagues, must be taught in engineering classes. His book met strong criticism in which some called to "burn" his book. Today, however, there is no known place in world that does not teach according to Eckert's doctrine. It is assumed that the same kind of individual(s) who criticized Eckert's work will criticize this work. Indeed, the previous book, on compressible flow, met its opposition. For example, anonymous Wikipedia user name EMBaero claimed that the material in the book is plagiarizing, he just doesn't know from where and what. Maybe that was the reason that he felt that is okay to plagiarize the book on Wikipedia. These criticisms will not change the future or the success of the ideas in this work. As a wise person says "don't tell me that it is wrong, show me what is wrong"; this is the only reply. With all the above, it must be emphasized that this book is not expected to revolutionize the field but change some of the way things are taught.

The book is organized into several chapters which, as a traditional textbook, deals with a basic introduction to the fluid properties and concepts (under construction). The second chapter deals with Thermodynamics. The third book chapter is a review of mechanics. The next topic is statics. When the Static Chapter was written, this author did not realize that so many new ideas will be inserted into this topic. As traditional texts in this field, ideal flow will be presented with the issues of added mass and added forces (under construction). The classic issue of turbulence (and stability) will be presented. An introduction to multi-phase flow, not a traditional topic, will be presented next (again under construction). The next two chapters will deals with open channel flow and gas dynamics. At this stage, dimensional analysis will be present (again under construction).

How This Book Was Written

This book started because I needed an introduction to the compressible flow book. After a while it seems that is easier to write a whole book than the two original planned chapters. In writing this book, it was assumed that introductory book on fluid mechanics should not contained many new ideas but should be modern in the material presentation. There are numerous books on fluid mechanics but none of which is open content. The approach adapted in this book is practical, and more hands-on approach. This statement really meant that the book is intent to be used by students to solve their exams and also used by practitioners when they search for solutions for practical problems. So, issue of proofs so and so are here only either to explain a point or have a solution of exams. Otherwise, this book avoids this kind of issues.

The structure of Hansen, Streeter and Wylie, and Shames books were adapted and used as a scaffolding for this book. This author was influenced by Streeter and Wylie book which was his undergrad textbooks. The chapters are not written in order. The first 4 chapters were written first because they were supposed to be modified and used as fluid mechanics introduction in "Fundamentals of Compressible Flow." Later, multi-phase flow chapter was written.

The presentation of some of the chapters is slightly different from other books because the usability of the computers. The book does not provide the old style graphical solution methods yet provides the graphical explanation of things.

Of course, this book was written on Linux (Micro\$oftLess book). This book was written using the vim editor for editing (sorry never was able to be comfortable with emacs). The graphics were done by TGIF, the best graphic program that this author experienced so far. The figures were done by gle. The spell checking was done by ispell, and hope to find a way to use gaspell, a program that currently cannot be used on new Linux systems. The figure in cover page was created by Genick Bar-Meir, and is copyleft by him.

Preface

*"In the beginning, the POTTO project was without form, and void; and emptiness was upon the face of the bits and files. And the Fingers of the Author moved upon the face of the keyboard. And the Author said, Let there be words, and there were words."*⁸.

This book, Basics of Fluid Mechanics, describes the fundamentals of fluid mechanics phenomena for engineers and others. This book is designed to replace all introductory textbook(s) or instructor's notes for the fluid mechanics in undergraduate classes for engineering/science students but also for technical peoples. It is hoped that the book could be used as a reference book for people who have at least some basics knowledge of science areas such as calculus, physics, etc.

The structure of this book is such that many of the chapters could be usable independently. For example, if you need information about, say, statics' equations, you can read just chapter (4). I hope this makes the book easier to use as a reference manual. However, this manuscript is first and foremost a textbook, and secondly a reference manual only as a lucky coincidence.

I have tried to describe why the theories are the way they are, rather than just listing "seven easy steps" for each task. This means that a lot of information is presented which is not necessary for everyone. These explanations have been marked as such and can be skipped.⁹ Reading everything will, naturally, increase your understanding of the many aspects of fluid mechanics.

This book is written and maintained on a volunteer basis. Like all volunteer work, there is a limit on how much effort I was able to put into the book and its organization. Moreover, due to the fact that English is my third language and time limitations, the explanations are not as good as if I had a few years to perfect them. Nevertheless, I believe professionals working in many engineering fields will benefit from this information. This book contains many worked examples, which can be very useful for many.

I have left some issues which have unsatisfactory explanations in the book, marked with a Mata mark. I hope to improve or to add to these areas in the near future.

⁸To the power and glory of the mighty God. This book is only to explain his power.

⁹At the present, the book is not well organized. You have to remember that this book is a work in progress.

Furthermore, I hope that many others will participate of this project and will contribute to this book (even small contributions such as providing examples or editing mistakes are needed).

I have tried to make this text of the highest quality possible and am interested in your comments and ideas on how to make it better. Incorrect language, errors, ideas for new areas to cover, rewritten sections, more fundamental material, more mathematics (or less mathematics); I am interested in it all. I am particularly interested in the best arrangement of the book. If you want to be involved in the editing, graphic design, or proofreading, please drop me a line. You may contact me via Email at "barmeir@gmail.com".

Naturally, this book contains material that never was published before (sorry cannot avoid it). This material never went through a close content review. While close content peer review and publication in a professional publication is excellent idea in theory. In practice, this process leaves a large room to blockage of novel ideas and plagiarism. If you would like be "peer reviews" or critic to my new ideas please send me your comment(s). Even reaction/comments from individuals like David Marshall¹⁰.

Several people have helped me with this book, directly or indirectly. I would like to especially thank to my adviser, Dr. E. R. G. Eckert, whose work was the inspiration for this book. I also would like to thank to Jannie McRotien (Open Channel Flow chapter) and Touser Yang for their advices, ideas, and assistance.

The symbol META was added to provide typographical conventions to blurb as needed. This is mostly for the author's purposes and also for your amusement. There are also notes in the margin, but those are solely for the author's purposes, ignore them please. They will be removed gradually as the version number advances.

I encourage anyone with a penchant for writing, editing, graphic ability, \LaTeX knowledge, and material knowledge and a desire to provide open content textbooks and to improve them to join me in this project. If you have Internet e-mail access, you can contact me at "barmeir@gmail.com".

¹⁰Dr. Marshall wrote to this author that the author should review other people work before he write any thing new (well, literature review is always good, isn't it?). Over ten individuals wrote me about this letter. I am asking from everyone to assume that his reaction was innocent one. While his comment looks like unpleasant reaction, it brought or cause the expansion of the explanation for the oblique shock. However, other email that imply that someone will take care of this author aren't appreciated.

To Do List and Road Map

This book isn't complete and probably never will be completed. There will always new problems to add or to polish the explanations or include more new materials. Also issues that associated with the book like the software has to be improved. It is hoped the changes in T_EX and L^AT_EX related to this book in future will be minimal and minor. It is hoped that the style file will be converged to the final form rapidly. Nevertheless, there are specific issues which are on the "table" and they are described herein.

At this stage, many chapters are missing. Specific missing parts from every chapters are discussed below. These omissions, mistakes, approach problems are sometime appears in the book under the Meta simple like this

Meta

sample this part.

Meta End

You are always welcome to add a new material: problem, question, illustration or photo of experiment. Material can be further illuminate. Additional material can be provided to give a different angle on the issue at hand.

Properties

The chapter isn't in development stage yet.

Open Channel Flow

The chapter isn't in the development stage yet. Some parts were taken from Fundamentals of Die Casting Design book and are in a process of improvement.

CHAPTER 1

Introduction

1.1 What is Fluid Mechanics?

Fluid mechanics deals with the study of all fluids under static and dynamic situations. Fluid mechanics is a branch of continuous mechanics which deals with a relationship between forces, motions, and statical conditions in continuous material. This study area deals with many and diversified problems such as surface tension, fluid statics, flow in enclosed bodies, or flow round bodies (solid or otherwise), flow stability, etc. In fact, almost any action a person is doing involves some kind of a fluid mechanics problem. Furthermore, the boundary between the solid mechanics and fluid mechanics is some kind of gray shed and not a sharp distinction (see Figure 1.1 for the complex relationships between the different branches which only part of it should be drawn in the same time.). For example, glass appears as a solid material, but a closer look reveals that the glass is a liquid with a large viscosity. A proof of the glass “liquidity” is the change of the glass thickness in high windows in European Churches after hundred years. The bottom part of the glass is thicker than the top part. Materials like sand (some call it quick sand) and grains should be treated as liquids. It is known that these materials have the ability to drown people. Even material such as aluminum just below the mushy zone also behaves as a liquid similarly to butter. After it was established that the boundaries of fluid mechanics aren’t sharp, the discussion in this book is limited to simple and (mostly) Newtonian (sometimes power fluids) fluids which will be defined later.

The fluid mechanics study involve many fields that have no clear boundary between them. Researchers distinguish between orderly flow and chaotic flow as the laminar flow and the turbulent flow. The fluid mechanics can also be distinguish between a single phase flow and multiphase flow (flow made more than one phase or single distinguishable material). The last boundary (as all the boundaries in fluid mechanics)

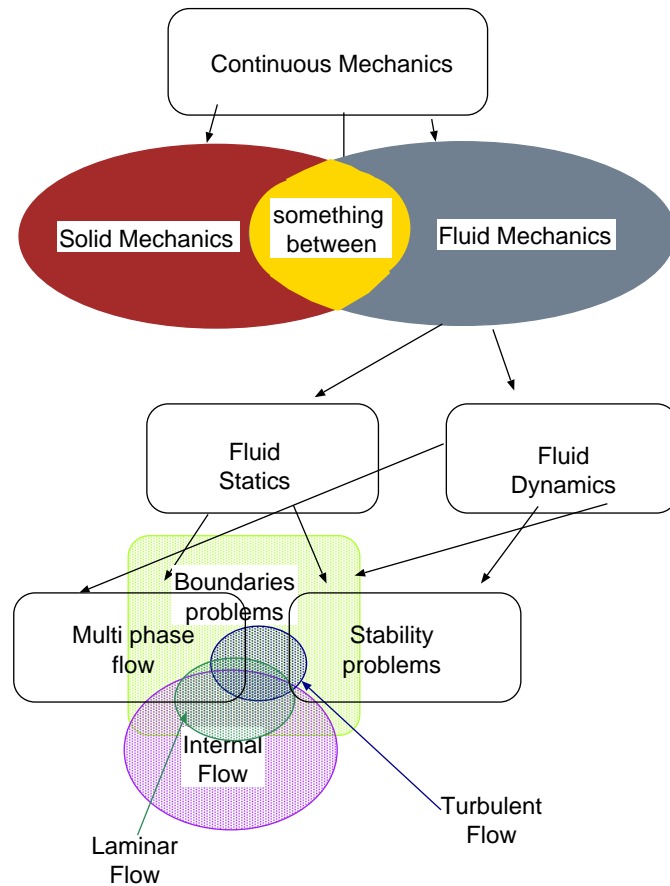


Fig. -1.1. Diagram to explain part of relationships of fluid mechanics branches.

isn't sharp because fluid can go through a phase change (condensation or evaporation) in the middle or during the flow and switch from a single phase flow to a multi phase flow. Moreover, flow with two phases (or materials) can be treated as a single phase (for example, air with dust particle).

After it was made clear that the boundaries of fluid mechanics aren't sharp, the study must make arbitrary boundaries between fields. Then the dimensional analysis will be used explain why in certain cases one distinguish area/principle is more relevant than the other and some effects can be neglected. Or, when a general model is need because more parameters are effecting the situation. It is this author's personal experience that the knowledge and ability to know in what area the situation lay is one of the main problems. For example, engineers in software company (EKK Inc, <http://ekkin.com/HTML>) analyzed a flow of a complete still liquid assuming a

complex turbulent flow model. Such absurd analysis are common among engineers who do not know which model can be applied. Thus, one of the main goals of this book is to explain what model should be applied. Before dealing with the boundaries, the simplified private cases must be explained.

There are two main approaches of presenting an introduction of fluid mechanics materials. The first approach introduces the fluid kinematic and then the basic governing equations, to be followed by stability, turbulence, boundary layer and internal and external flow. The second approach deals with the Integral Analysis to be followed with Differential Analysis, and continue with Empirical Analysis. These two approaches pose a dilemma to anyone who writes an introductory book for the fluid mechanics. These two approaches have justifications and positive points. Reviewing many books on fluid mechanics made it clear, there isn't a clear winner. This book attempts to find a hybrid approach in which the kinematic is presented first (aside to standard initial four chapters) follow by Integral analysis and continued by Differential analysis. The ideal flow (frictionless flow) should be expanded compared to the regular treatment. This book is unique in providing chapter on multiphase flow. Naturally, chapters on open channel flow (as a sub class of the multiphase flow) and compressible flow (with the latest developments) are provided.

1.2 *Brief History*

The need to have some understanding of fluid mechanics started with the need to obtain water supply. For example, people realized that wells have to be dug and crude pumping devices need to be constructed. Later, a large population created a need to solve waste (sewage) and some basic understanding was created. At some point, people realized that water can be used to move things and provide power. When cities increased to a larger size, aqueducts were constructed. These aqueducts reached their greatest size and grandeur in those of the City of Rome and China.

Yet, almost all knowledge of the ancients can be summarized as application of instincts, with the exception Archimedes (250 B.C.) on the principles of buoyancy. For example, larger tunnels built for a larger water supply, etc. There were no calculations even with the great need for water supply and transportation. The first progress in fluid mechanics was made by Leonardo Da Vinci (1452-1519) who built the first chambered canal lock near Milan. He also made several attempts to study the flight (birds) and developed some concepts on the origin of the forces. After his initial work, the knowledge of fluid mechanics (hydraulic) increasingly gained speed by the contributions of Galileo, Torricelli, Euler, Newton, Bernoulli family, and D'Alembert. At that stage theory and experiments had some discrepancy. This fact was acknowledged by D'Alembert who stated that, "The theory of fluids must necessarily be based upon experiment." For example the concept of ideal liquid that leads to motion with no resistance, conflicts with the reality.

This discrepancy between theory and practice is called the "D'Alembert paradox" and serves to demonstrate the limitations of theory alone in solving fluid problems. As in thermodynamics, two different of school of thoughts were created: the first be-

believed that the solution will come from theoretical aspect alone, and the second believed that solution is the pure practical (experimental) aspect of fluid mechanics. On the theoretical side, considerable contribution were made by Euler, La Grange, Helmholtz, Kirchhoff, Rayleigh, Rankine, and Kelvin. On the "experimental" side, mainly in pipes and open channels area, were Brahm, Bossut, Chezy, Dubuat, Fabre, Coulomb, Dupuit, d'Aubisson, Hagen, and Poiseuille.

In the middle of the nineteenth century, first Navier in the molecular level and later Stokes from continuous point of view succeeded in creating governing equations for real fluid motion. Thus, creating a matching between the two school of thoughts: experimental and theoretical. But, as in thermodynamics, people cannot relinquish control. As results it created today "strange" names: Hydrodynamics, Hydraulics, Gas Dynamics, and Aeronautics.

The Navier-Stokes equations, which describes the flow (or even Euler equations), were considered unsolvable during the mid nineteenth century because of the high complexity. This problem led to two consequences. Theoreticians tried to simplify the equations and arrive at approximated solutions representing specific cases. Examples of such work are Hermann von Helmholtz's concept of vortexes (1858), Lanchester's concept of circulatory flow (1894), and the Kutta-Joukowski circulation theory of lift (1906). The experimentalists, at the same time proposed many correlations to many fluid mechanics problems, for example, resistance by Darcy, Weisbach, Fanning, Ganguillet, and Manning. The obvious happened without theoretical guidance, the empirical formulas generated by fitting curves to experimental data (even sometime merely presenting the results in tabular form) resulting in formulas that the relationship between the physics and properties made very little sense.

At the end of the twenty century, the demand for vigorous scientific knowledge that can be applied to various liquids as opposed to formula for every fluid was created by the expansion of many industries. This demand coupled with new several novel concepts like the theoretical and experimental researches of Reynolds, the development of dimensional analysis by Rayleigh, and Froude's idea of the use of models change the science of the fluid mechanics. Perhaps the most radical concept that effects the fluid mechanics is of Prandtl's idea of boundary layer which is a combination of the modeling and dimensional analysis that leads to modern fluid mechanics. Therefore, many call Prandtl as the father of modern fluid mechanics. This concept leads to mathematical basis for many approximations. Thus, Prandtl and his students Blasius, von Karman, Meyer, and Blasius and several other individuals as Nikuradse, Rose, Taylor, Bhuckingham, Stanton, and many others, transformed the fluid mechanics to modern science that we have known today.

While the understanding of the fundamentals did not change much, after World War Two, the way how it was calculated changed. The introduction of the computers during the 60s and much more powerful personal computer has changed the field. There are many open source programs that can analyze many fluid mechanics situations. Today many problems can be analyzed by using the numerical tools and provide reasonable results. These programs in many cases can capture all the appropriate parameters and adequately provide a reasonable description of the physics. However, there are many

other cases that numerical analysis cannot provide any meaningful result (trends). For example, no weather prediction program can produce good engineering quality results (where the snow will fall within 50 kilometers accuracy. Building a car with this accuracy is a disaster). In the best scenario, these programs are as good as the input provided. Thus, assuming turbulent flow for still flow simply provides erroneous results (see for example, EKK, Inc).

1.3 Kinds of Fluids

Some differentiate fluid from solid by the reaction to shear stress. It is a known fact said that the fluid continuously and permanently deformed under shear stress while solid exhibits a finite deformation which does not change with time. It is also said that liquid cannot return to their original state after the deformation. This differentiation leads to three groups of materials: solids and liquids. This test creates a new material group that shows dual behaviors; under certain limits; it behaves like solid and under others it behaves like liquid (see Figure 1.1). The study of this kind of material called rheology and it will (almost) not be discussed in this book. It is evident from this discussion that when a liquid is at rest, no shear stress is applied.

The fluid is mainly divided into two categories: liquids and gases. The main difference between the liquids and gases state is that gas will occupy the whole volume while liquids has an almost fix volume. This difference can be, for most practical purposes considered, sharp even though in reality this difference isn't sharp. The difference between a gas phase to a liquid phase above the critical point are practically minor. But below the critical point, the change of water pressure by 1000% only change the volume by less than 1 percent. For example, a change in the volume by more 5% will required tens of thousands percent change of the pressure. So, if the change of pressure is significantly less than that, then the change of volume is at best 5%. Hence, the pressure will not affect the volume. In gaseous phase, any change in pressure directly affects the volume. The gas fills the volume and liquid cannot. Gas has no free interface/surface (since it does fill the entire volume).

There are several quantities that have to be addressed in this discussion. The first is **force** which was reviewed in physics. The unit used to measure is [N]. It must be remember that force is a vector, e.g it has a direction. The second quantity discussed here is the area. This quantity was discussed in physics class but here it has an additional meaning, and it is referred to the direction of the area. The direction of area is perpendicular to the area. The area is measured in [m^2]. Area of three-dimensional object has no single direction. Thus, these kinds of areas should be addressed infinitesimally and locally.

The traditional quantity, which is force per area has a new meaning. This is a result of division of a vector by a vector and it is referred to as tensor. In this book, the emphasis is on the physics, so at this stage the tensor will have to be broken into its components. Later, the discussion on the mathematic meaning will be presented (later version). For the discussion here, the pressure has three components, one in the area direction and two perpendicular to the area. The pressure component in the area

direction is called pressure (great way to confuse, isn't it?). The other two components are referred as the shear stresses. The units used for the pressure components is $[N/m^2]$.

The density is a property which requires that liquid to be continuous. The density can be changed and it is a function of time and space (location) but must have a continuous property. It doesn't mean that a sharp and abrupt change in the density cannot occur. It referred to density that is independent of the sampling size. Figure 1.2 shows the density as a function of the sample size. After certain sample size, the density remains constant. Thus, the density is defined as

$$\rho = \lim_{\Delta V \rightarrow \varepsilon} \frac{\Delta m}{\Delta V} \quad (1.1)$$

It must be noted that ε is chosen so that the continuous assumption is not broken, that is, it did not reach/reduced to the size where the atoms or molecular statistical calculations are significant (see Figure 1.2 for point where the green lines converge to constant density). When this assumption is broken, then, the principles of statistical mechanics must be utilized.

1.4 Shear Stress

The shear stress is part of the pressure tensor. However, here it will be treated as a separate issue. In solid mechanics, the shear stress is considered as the ratio of the force acting on area in the direction of the forces perpendicular to area. Different from solid, fluid cannot pull directly but through a solid surface. Consider liquid that undergoes a shear stress between a short distance of two plates as shown in Figure (1.3).

The upper plate velocity generally will be

$$U = f(A, F, h) \quad (1.2)$$

Where A is the area, the F denotes the force, h is the distance between the plates. From solid mechanics study, it was shown that when the force per area increases, the velocity of the plate increases also. Experiments show that the increase of height will increase the velocity up to a certain range. Consider moving the plate with a zero lubricant ($h \sim 0$) (results in large force) or a large amount of lubricant (smaller force). In this discussion, the aim is to develop differential equation, thus the small distance analysis is applicable.

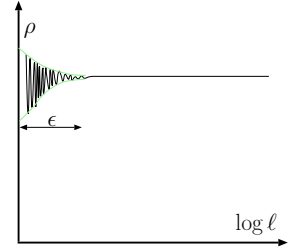


Fig. -1.2. Density as a function of the size of sample.

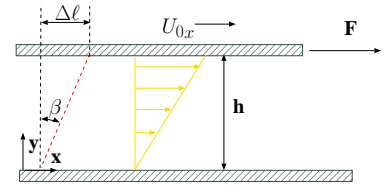


Fig. -1.3. Schematics to describe the shear stress in fluid mechanics.

For cases where the dependency is linear, the following can be written

$$U \propto \frac{hF}{A} \quad (1.3)$$

Equations (1.3) can be rearranged to be

$$\frac{U}{h} \propto \frac{F}{A} \quad (1.4)$$

Shear stress was defined as

$$\tau_{xy} = \frac{F}{A} \quad (1.5)$$

From equations (1.4) and (1.5) it follows that ratio of the velocity to height is proportional to shear stress. Hence, applying the coefficient to obtain a new equality as

$$\tau_{xy} = \mu \frac{U}{h} \quad (1.6)$$

Where μ is called the absolute viscosity or dynamic viscosity which will be discussed later in this chapter in great length.

In steady state, the distance the upper plate moves after small amount of time, δt is

$$d\ell = U \delta t \quad (1.7)$$

From Figure 1.4 it can be noticed that for a small angle, the regular approximation provides

$$d\ell = U \delta t = \overbrace{h \delta\beta}^{\text{geometry}} \quad (1.8)$$

From equation (1.8) it follows that

$$U = h \frac{\delta\beta}{\delta t} \quad (1.9)$$

Combining equation (1.9) with equation (1.6) yields

$$\tau_{xy} = \mu \frac{\delta\beta}{\delta t} \quad (1.10)$$

If the velocity profile is linear between the plate (it will be shown later that it is consistent with derivations of velocity), then it can be written for small a angel that

$$\frac{\delta\beta}{\delta t} = \frac{dU}{dy} \quad (1.11)$$

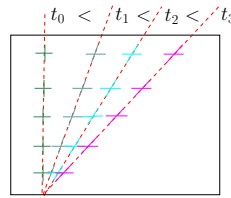


Fig. -1.4. The deformation of fluid due to shear stress as progression of time.

Materials which obey equation (1.10) referred to as Newtonian fluid. For this kind of substance

$$\tau_{xy} = \mu \frac{dU}{dy} \quad (1.12)$$

Newtonian fluids are fluids which the ratio is constant. Many fluids fall into this category such as air, water etc. This approximation is appropriate for many other fluids but only within some ranges.

Equation (1.9) can be interpreted as momentum in the x direction transferred into the y direction. Thus, the viscosity is the resistance to the flow (flux) or the movement. The property of viscosity, which is exhibited by all fluids, is due to the existence of cohesion and interaction between fluid molecules. These cohesion and interactions hamper the flux in y -direction. Some referred to shear stress as viscous flux of x -momentum in the y -direction. The units of shear stress are the same as flux per time as following

$$\frac{F}{A} \left[\frac{kg \ m}{sec^2 \ m^2} \right] = \frac{\dot{m}U}{A} \left[\frac{kg \ m}{sec \ sec \ m^2} \right]$$

Thus, the notation of τ_{xy} is easier to understand and visualize. In fact, this interpretation is more suitable to explain the molecular mechanism of the viscosity. The units of absolute viscosity are $[N \ sec/m^2]$.

Example 1.1:

A space of 1 [cm] width between two large plane surfaces is filled with glycerine. Calculate the force that is required to drag a very thin plate of 1 [m²] at a speed of 0.5 m/sec. It can be assumed that the plates remains in equidistance from each other and steady state is achieved instantly.

SOLUTION

Assuming Newtonian flow, the following can be written (see equation (1.6))

$$F = \frac{A \mu U}{h} \sim \frac{1 \times 1.069 \times 0.5}{0.01} = 53.45[N]$$

End Solution

Example 1.2:

Castor oil at 25°C fills the space between two concentric cylinders of 0.2[m] and 0.1[m] diameters with height of 0.1 [m]. Calculate the torque required to rotate the inner cylinder at 12 rpm, when the outer cylinder remains stationary. Assume steady state conditions.

SOLUTION

The velocity is

$$U = r \dot{\theta} = 2 \pi r_i \text{ rps} = 2 \times \pi \times 0.1 \times \overbrace{12/60}^{\text{rps}} = 0.4 \pi r_i$$

Where *rps* is revolution per second.

The same way as in example (1.1), the moment can be calculated as the force times the distance as

$$M = F \ell = \frac{\overbrace{\ell}^{r_i} \overbrace{2 \pi r_i h}^A \mu U}{r_o - r_i}$$

In this case $r_o - r_i = h$ thus,

$$M = \frac{2 \pi^2 \overbrace{0.1^3}^{r_i} \overbrace{0.986}^{\mu} 0.4}{h} \sim .0078 [N m]$$

End Solution

1.5 Viscosity

1.5.1 General

Viscosity varies widely with temperature. However, temperature variation has an opposite effect on the viscosities of liquids and gases. The difference is due to their fundamentally different viscosity characteristics. In gases, molecules are sparse and cohesion is negligible, while in the liquids, the molecules are more compact and cohesion is more dominant. Thus, in gases, the exchange of momentum between layers brought as a result of

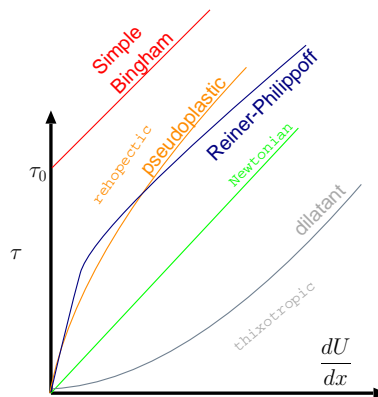


Fig. -1.5. The different of power fluids familys.

molecular movement normal to the general direction of flow, and it resists the flow. This molecular activity is known to increase with temperature, thus, the viscosity of gases will increase with temperature. This reasoning is a result of the considerations of the kinetic theory. This theory indicates that gas viscosities vary directly with the square root of temperature. In liquids, the momentum exchange due to molecular movement is small compared to the cohesive forces between the molecules. Thus, the viscosity is primarily dependent on the magnitude of these cohesive forces. Since these forces decrease rapidly with increases of temperature, liquid viscosities decrease as temperature increases.

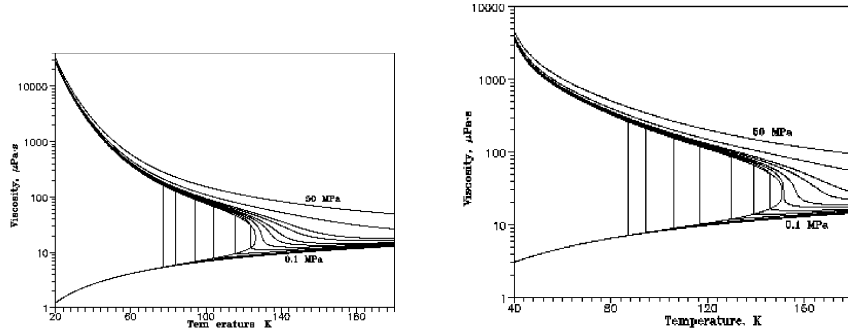


Fig. -1.6. Nitrogen (left) and Argon (right) viscosity as a function of the temperature and pressure after Lemmon and Jacobsen.

Figure 1.6 demonstrates that viscosity increases slightly with pressure, but this variation is negligible for most engineering problems. Well above the critical point, both materials are only a function of the temperature. On the liquid side below the critical point, the pressure has minor effect on the viscosity. It must be stress that the viscosity in the dome is meaningless. There is no such a thing of viscosity at 30% liquid. It simply depends on the structure of the flow as will be discussed in the chapter on multi phase flow. The lines in the above diagrams are only to show constant pressure lines. Oils have the greatest increase of viscosity with pressure which is a good thing for many engineering purposes.

1.5.2 Non-Newtonian Fluids

In equation (1.5), the relationship between the velocity and the shear stress was assumed to be linear. Not all the materials obey this relationship. There is a large class of materials which shows a non-linear relationship with velocity for any shear stress. This class of materials can be approximated by a single polynomial term that is $a = bx^n$. From the physical point of view, the coefficient depends on the velocity gradient. This relationship is referred to as power relationship and it can be written as

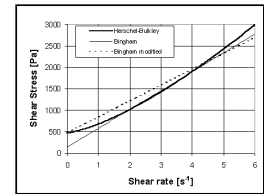


Fig. -1.7. The shear stress as a function of the shear rate.

$$\tau = K \overbrace{\left(\frac{dU}{dx}\right)^{n-1}}^{\text{viscosity}} \left(\frac{dU}{dx}\right) \tag{1.13}$$

The new coefficients (n, K) in equation (1.13) are constant. When $n = 1$ equation represent Newtonian fluid and K becomes the familiar μ . The viscosity coefficient is

always positive. When n , is above one, the liquid is dilatant. When n is below one, the fluid is pseudoplastic. The liquids which satisfy equation (1.13) are referred to as purely viscous fluids. Many fluids satisfy the above equation. Fluids that show increase in the viscosity (with increase of the shear) referred to as thixotropic and those that show decrease are called reoplectic fluids (see Figure 1.5).

Materials which behave up to a certain shear stress as a solid and above it as a liquid are referred to as Bingham liquids. In the simple case, the “liquid side” is like Newtonian fluid for large shear stress. The general relationship for simple Bingham flow is

$$\tau_{xy} = -\mu \pm \tau_0 \quad \text{if } |\tau_{yx}| > \tau_0 \quad (1.14)$$

$$\frac{dU_x}{dy} = 0 \quad \text{if } |\tau_{yx}| < \tau_0 \quad (1.15)$$

There are materials that simple Bingham model does not provide adequate explanation and a more sophisticated model is required. The Newtonian part of the model has to be replaced by power liquid. For example, according to Ferraris et al¹ concrete behaves as shown in Figure 1.7. However, for most practical purposes, this kind of figures isn't used in regular engineering practice.

1.5.3 Kinematic Viscosity

The kinematic viscosity is another way to look at the viscosity. The reason for this new definition is that some experimental data are given in this form. These results also explained better using the new definition. The kinematic viscosity embraces both the viscosity and density properties of a fluid. The above equation shows that the dimensions of ν to be square meter per second, $[m^2/sec]$, which are acceleration units (a combination of kinematic terms). This fact explains the name “kinematic” viscosity. The kinematic viscosity is defined as

$$\nu = \frac{\mu}{\rho} \quad (1.16)$$

The gas density increases with the temperature. The increase of the absolute viscosity with the temperature is enough to overcome the increase of density and thus, the kinematic viscosity also increase with the temperature for many materials.

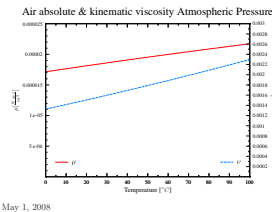
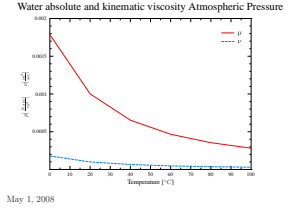


Fig. -1.8. Air viscosity as a function of the temperature.

¹C. Ferraris, F. de Larrard and N. Martys, Materials Science of Concrete VI, S. Mindess and J. Skalny, eds., 215-241 (2001)

1.5.4 Estimation of The Viscosity

The absolute viscosity of many fluids relatively doesn't change with the pressure but very sensitive to temperature. For isothermal flow, the viscosity can be considered constant in many cases. The variations of air and water as a function of the temperature at atmospheric pressure are plotted in Figures 1.8 and 1.9.



May 1, 2008

Fig. -1.9. Water viscosity as a function temperature.

Some common materials (pure and mixture) have expressions that provide an estimate. For many gases, Sutherland's equation is used and according to the literature, provides reasonable results² for the range of $-40^{\circ}C$ to $1600^{\circ}C$

$$\mu = \mu_0 \frac{0.555 T_{i0} + Suth}{0.555 T_{in} + Suth} \left(\frac{T}{T_0} \right)^{\frac{3}{2}} \quad (1.17)$$

Where

μ	viscosity at input temperature T
μ_0	reference viscosity at reference temperature, T_{i0}
T_{in}	input temperature in degrees Kelvin
T_{i0}	reference temperature in degrees Kelvin
$Suth$	Suth is Sutherland's constant and it is presented in the Table 1.1

Example 1.3:

Calculate the viscosity of air at 800K based on Sutherland's equation. Use the data provide in Table 1.1.

SOLUTION

Applying the constants from Sutherland's table provides

$$\mu = 0.00001827 \times \frac{0.555 \times 524.07 + 120}{0.555 \times 800 + 120} \times \left(\frac{800}{524.07} \right)^{\frac{3}{2}} \sim 2.51 \cdot 10^{-5} \left[\frac{N \cdot sec}{m^2} \right]$$

The viscosity increases almost by 40%. The observed viscosity is about $\sim 3.710^{-5}$.

End Solution

Liquid Metals

²This author is ambivalent about statement.

Material	coefficients	chemical formula	Sutherland	$T_{iO}[K]$	$\mu_0(N\ sec/m^2)$
ammonia		NH_3	370	527.67	0.00000982
standard air			120	524.07	0.00001827
carbon dioxide		CO_2	240	527.67	0.00001480
carbon monoxide		CO	118	518.67	0.00001720
hydrogen		H_2	72	528.93	0.0000876
nitrogen		N_2	111	540.99	0.0001781
oxygen		O_2	127	526.05	0.0002018
sulfur dioxide		SO_2	416	528.57	0.0001254

Table -1.1. The list for Sutherland's equation coefficients for selected materials.

Substance	formula	Temperature $T [^{\circ}C]$	Viscosity [$\frac{N\ sec}{m^2}$]
	$i - C_4 H_{10}$	23	0.0000076
	CH_4	20	0.0000109
	CO_2	20	0.0000146
oxygen	O_2	20	0.0000203
mercury vapor	Hg	380	0.0000654

Table -1.2. Viscosity of selected gases.

Substance	formula	Temperature $T [^{\circ}C]$	Viscosity [$\frac{N \text{ sec}}{m^2}$]
	$(C_2H_5)O$	20	0.000245
	C_6H_6	20	0.000647
	Br_2	26	0.000946
	C_2H_5OH	20	0.001194
	Hg	25	0.001547
	H_2SO_4	25	0.01915
Olive Oil		25	0.084
Castor Oil		25	0.986
Clucose		25	5-20
Corn Oil		20	0.072
SAE 30		-	0.15-0.200
SAE 50		$\sim 25^{\circ}C$	0.54
SAE 70		$\sim 25^{\circ}C$	1.6
Ketchup		$\sim 20^{\circ}C$	0,05
Ketchup		$\sim 25^{\circ}C$	0,098
Benzene		$\sim 20^{\circ}C$	0.000652
Firm glass		-	$\sim 1 \times 10^7$
Glycerol		20	1.069

Table -1.3. Viscosity of selected liquids.

chemical component	Molecular Weight	T_c [K]	P_c [Bar]	μ_c [$\frac{N \text{ sec}}{m^2}$]
H_2	2.016	33.3	12.9696	3.47
He	4.003	5.26	2.289945	2.54
Ne	20.183	44.5	27.256425	15.6
Ar	39.944	151	48.636	26.4
Xe	131.3	289.8	58.7685	49.
Air "mix"	28.97	132	36.8823	19.3
CO_2	44.01	304.2	73.865925	19.0
O_2	32.00	154.4	50.358525	18.0
C_2H_6	30.07	305.4	48.83865	21.0
CH_4	16.04	190.7	46.40685	15.9
Water		647.096 K	22.064 [MPa]	

Table -1.4. The properties at the critical stage and their values of selected materials.

Liquid metal can be considered as a Newtonian fluid for many applications. Furthermore, many aluminum alloys are behaving as a Newtonian liquid until the first solidification appears (assuming steady state thermodynamics properties). Even when there is a solidification (mushy zone), the metal behavior can be estimated as a Newtonian material (further reading can be done in this author's book "Fundamentals of Die Casting Design"). Figure 1.10 exhibits several liquid metals (from The Reactor Handbook, Vol. Atomic Energy Commission AEC-3646 U.S. Government Printing Office, Washington D.C. May 1995 p. 258.)

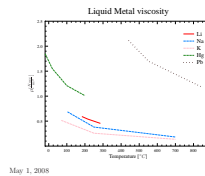


Fig. -1.10. Liquid metals viscosity as a function of the temperature.

The General Viscosity Graphs

In case "ordinary" fluids where information is limit, Hougen et al suggested to use graph similar to compressibility chart. In this graph, if one point is well documented, other points can be estimated. Furthermore, this graph also shows the trends. In Figure 1.11 the relative viscosity $\mu_r = \mu/\mu_c$ is plotted as a function of relative temperature, T_r . μ_c is the viscosity at critical condition and μ is the viscosity at any given condition. The lines of constant relative pressure, $P_r = P/P_c$ are drawn. The lower pressure is, for practical purpose, ~ 1 [bar].

The critical pressure can be evaluated in the following three ways. The simplest way is by obtaining the data from Table 1.4 or similar information. The second way, if the information is available and is close enough to the critical point, then the critical

viscosity is obtained as

$$\mu_c = \frac{\overbrace{\mu}^{\text{given}}}{\underbrace{\mu_r}_{\text{figure 1.11}}} \quad (1.18)$$

The third way, when none is available, is by utilizing the following approximation

$$\mu_c = \sqrt{M T_c \tilde{v}_c^{2/3}} \quad (1.19)$$

Where \tilde{v}_c is the critical molecular volume and M is molecular weight. Or

$$\mu_c = \sqrt{M P_c^{2/3} T_c^{-1/6}} \quad (1.20)$$

Calculate the reduced pressure and the reduced temperature and from the Figure 1.11 obtain the reduced viscosity.

Example 1.4:

Estimate the viscosity of oxygen, O_2 at $100^\circ C$ and $20[Bar]$.

SOLUTION

The critical condition of oxygen are $P_c = 50.35[Bar]$ $T_c = 154.4$ $\mu_c = 18 \left[\frac{N \cdot sec}{m^2} \right]$ The value of the reduced temperature is

$$T_r \sim \frac{373.15}{154.4} \sim 2.41$$

The value of the reduced pressure is

$$P_r \sim \frac{20}{50.35} \sim 0.4$$

From Figure 1.11 it can be obtained $\mu_r \sim 1.2$ and the predicted viscosity is

$$\mu = \mu_c \left(\overbrace{\frac{\mu}{\mu_c}}^{\text{Table}} \right) = 18 \times 1.2 = 21.6 [N \cdot sec / m^2]$$

The observed value is $24 [N \cdot sec / m^2]^3$.

End Solution

Viscosity of Mixtures

In general the viscosity of liquid mixture has to be evaluated experimentally. Even for homogeneous mixture, there isn't silver bullet to estimate the viscosity. In this book, only the mixture of low density gases is discussed for analytical expression. For most

³Kyama, Makita, Rev. Physical Chemistry Japan Vol. 26 No. 2 1956.

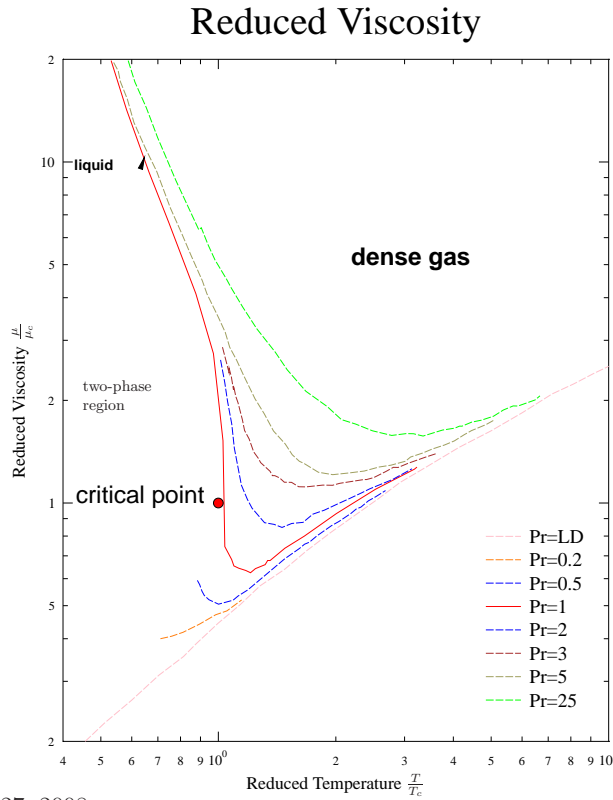


Fig. -1.11. Reduced viscosity as function of the reduced temperature.

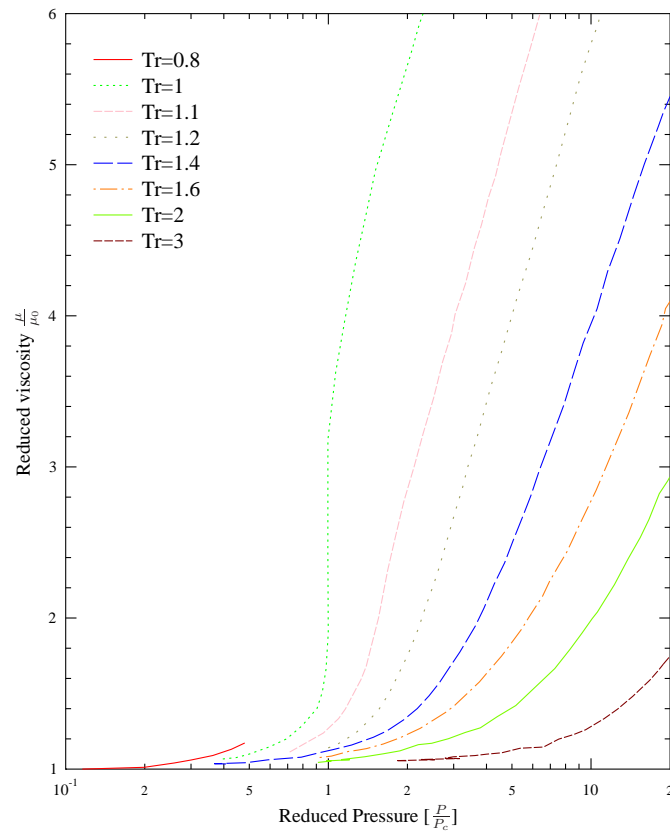
cases, the following Wilke's correlation for gas at low density provides a result in a reasonable range.

$$\mu_{mix} = \frac{\sum_{i=1}^n x_i \mu_i}{\sum_{j=1}^n x_j \Phi_{ij}} \quad (1.21)$$

where Φ_{ij} is defined as

$$\Phi_{ij} = \frac{1}{\sqrt{8}} \sqrt{1 + \frac{M_i}{M_j}} \left(1 + \sqrt{\frac{\mu_i}{\mu_j}} \sqrt{\frac{M_j}{M_i}} \right)^2 \quad (1.22)$$

Here, n is the number of the chemical components in the mixture. x_i is the mole fraction of component i , and μ_i is the viscosity of component i . The subscript i should be used for the j index. The dimensionless parameter Φ_{ij} is equal to one when $i = j$. The mixture viscosity is highly nonlinear function of the fractions of the components.



June 2, 2008

Fig. -1.12. Reduced viscosity as function of the reduced temperature.

Example 1.5:

Calculate the viscosity of a mixture (air) made of 20% oxygen, O_2 and 80% nitrogen N_2 for the temperature of $20^\circ C$.

SOLUTION

The following table summarizes the known details

i	Component	Molecular Weight, M	Mole Fraction, x	Viscosity, μ
1	O_2	32.	0.2	0.0000203
2	N_2	28.	0.8	0.00001754

i	j	M_i/M_j	μ_i/μ_j	Φ_{ij}
1	1	1.0	1.0	1.0
	2	1.143	1.157	1.0024
2	1	0.875	.86	0.996
	2	1.0	1.0	1.0

$$\mu_{mix} \sim \frac{0.2 \times 0.0000203}{0.2 \times 1.0 + 0.8 \times 1.0024} + \frac{0.8 \times 0.00001754}{0.2 \times 0.996 + 0.8 \times 1.0} \sim 0.0000181 \left[\frac{N \text{ sec}}{m^2} \right]$$

The observed value is $\sim 0.0000182 \left[\frac{N \text{ sec}}{m^2} \right]$.

End Solution

In very low pressure, in theory, the viscosity is only a function of the temperature with a "simple" molecular structure. For gases with very long molecular structure or complexity structure these formulas cannot be applied. For some mixtures of two liquids it was observed that at a low shear stress, the viscosity is dominated by a liquid with high viscosity and at high shear stress to be dominated by a liquid with the low viscosity liquid. The higher viscosity is more dominate at low shear stress. Reiner and Phillippoff suggested the following formula

$$\frac{dU_x}{dy} = \left(\frac{1}{\mu_\infty + \frac{\mu_0 - \mu_\infty}{1 + \left(\frac{\tau_{xy}}{\tau_s} \right)^2}} \right) \tau_{xy} \quad (1.23)$$

Where the term μ_∞ is the experimental value at high shear stress. The term μ_0 is the experimental viscosity at shear stress approaching zero. The term τ_s is the characteristic shear stress of the mixture. An example for values for this formula, for Molten Sulfur at temperature $120^\circ C$ are $\mu_\infty = 0.0215 \left(\frac{N \text{ sec}}{m^2} \right)$, $\mu_0 = 0.00105 \left(\frac{N \text{ sec}}{m^2} \right)$, and $\tau_s = 0.0000073 \left(\frac{kN}{m^2} \right)$. This equation (1.23) provides reasonable value only up to $\tau = 0.001 \left(\frac{kN}{m^2} \right)$.

Figure 1.12 can be used for a crude estimate of dense gases mixture. To estimate the viscosity of the mixture with n component Hougen and Watson's method for pseudocritical properties is adapted. In this method the following is defined as

$$P_{c \text{ mix}} = \sum_{i=1}^n x_i P_{c_i} \quad (1.24)$$

$$T_{c \text{ mix}} = \sum_{i=1}^n x_i T_{c_i} \quad (1.25)$$

and

$$\mu_{c_{mix}} = \sum_{i=1}^n x_i \mu_{c_i} \quad (1.26)$$

1.5.5 Bulk Modulus

Similar to solids (hook's law), liquids have a property that describes the volume change as results of pressure change for constant temperature. It can be noted that this property is not the result of the equation of state but related to it. The bulk modulus is defined as

$$B_T = -v \left(\frac{\partial P}{\partial v} \right)_T \quad (1.27)$$

Using the identity of $v = 1/\rho$ transfers equation (1.27) into

$$B_T = \rho \left(\frac{\partial P}{\partial \rho} \right)_T \quad (1.28)$$

The bulk modulus for several liquids is presented in Table 1.5.

Table -1.5. The bulk modulus for selected material with the critical temperature and pressure *na* → not available and *nf* → not found (exist but was not found in the literature).

chemical component	Bulk Modulus $10^9 \frac{N}{m}$	T_c	P_c
Acetic Acid	2.49	593K	57.8 [Bar]
Acetone	0.80	508 K	48 [Bar]
Benzene	1.10	562 K	4.74 [MPa]
Carbon Tetrachloride	1.32	556.4 K	4.49 [MPa]
Ethyl Alcohol	1.06	514 K	6.3 [Mpa]
Gasoline	1.3	nf	nf
Glycerol	4.03-4.52	850 K	7.5 [Bar]
Mercury	26.2-28.5	1750 K	172.00 [MPa]
Methyl Alcohol	0.97	Est 513	Est 78.5 [Bar]
Nitrobenzene	2.20	nf	nf
Olive Oil	1.60	nf	nf
Paraffin Oil	1.62	nf	nf
SAE 30 Oil	1.5	na	na
Seawater	2.34	na	na
Toluene	1.09	591.79 K	4.109 [MPa]
Turpentine	1.28	na	na
Water	2.15-2.174	647.096 K	22.064 [MPa]

In the literature, additional expansions for similar parameters are defined. The thermal expansion is defined as

$$\beta_P = \frac{1}{v} \left(\frac{\partial v}{\partial T} \right)_P \quad (1.29)$$

This parameter indicates the change of volume due to temperature change when the pressure is constant. Another definition is referred as coefficient of tension and it is defined as

$$\beta_v = \frac{1}{P} \left(\frac{\partial P}{\partial T} \right)_v \quad (1.30)$$

This parameter indicates the change of the pressure due to the change of temperature (where $v = \text{constant}$). These definitions are related to each other. This relationship is obtained by the observation that the pressure as a function of the temperature and specific volume as

$$P = f(T, v) \quad (1.31)$$

The full pressure derivative is

$$dP = \left(\frac{\partial P}{\partial T} \right)_v dT + \left(\frac{\partial P}{\partial v} \right)_T dv \quad (1.32)$$

On constant pressure lines, $dP = 0$, and therefore equation (1.32) is

$$0 = \left(\frac{\partial P}{\partial T} \right)_v dT + \left(\frac{\partial P}{\partial v} \right)_T dv \quad (1.33)$$

From equation (1.33) follows that

$$\left. \frac{dv}{dT} \right|_{P=\text{const}} = - \frac{\left(\frac{\partial P}{\partial T} \right)_v}{\left(\frac{\partial P}{\partial v} \right)_T} \quad (1.34)$$

Equation (1.34) indicates that relationship for these three coefficients is

$$\beta_T = - \frac{\beta_v}{\beta_P} \quad (1.35)$$

The last equation (1.35) sometimes is used in measurement of the bulk modulus.

The increase of the pressure increases the bulk modulus due to the molecules increase of the rejecting forces between each other when they are closer. In contrast,

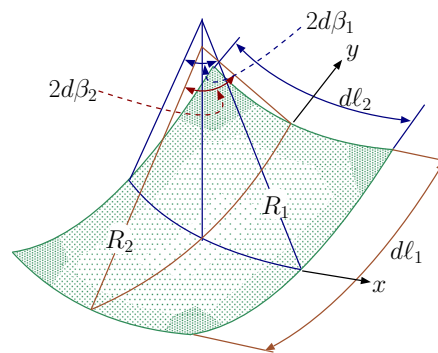


Fig. -1.13. Surface Tension control volume analysis.

the temperature increase results in reduction of the bulk modulus because the molecules are further away.

Example 1.6:

Calculate the modulus of liquid elasticity that reduced 0.035 per cent of its volume by applying a pressure of 5[Bar].

SOLUTION

Using the definition for the bulk modulus

$$\beta_T = -v \frac{\partial P}{\partial v} \sim \frac{v}{\Delta v} \Delta P = \frac{5}{0.00035} \sim 14285.714[\text{Bar}]$$

End Solution

Example 1.7:

Calculate the pressure needed to apply on water to reduce its volume by 1 per cent. Assume the temperature to be 20°C.

SOLUTION

Using the definition for the bulk modulus

$$\Delta P \sim \beta_T \frac{\Delta v}{v} \sim 2.15 \cdot 10^9 \cdot 0.01 = 2.15 \cdot 10^7 [\text{N/m}^2] = 215[\text{Bar}]$$

End Solution

1.6 Surface Tension

The surface tension manifested itself by a rise or depression of the liquid at the free surface edge. Surface tension is also responsible for the creation of the drops and bubbles. It is also responsible for the breakage of a liquid jet into other medium/phase to many drops (atomization). The surface tension is force per length and is measured by [N/m] and is acting to stretch the surface.

Surface tension results from a sharp change in the density between two adjacent phases or materials. There is a common misconception for the source of the surface tension. In many (physics, surface tension, and fluid mechanics) books explained that the surface tension is a result from unbalanced molecular cohesive forces. This explanation is wrong since it is in conflict with Newton's second law (see example ?). This erroneous explanation can be traced to Adam's book but earlier source may be found.

The relationship between the surface tension and the pressure on the two sides of the surface is based on geometry. Consider a small element of surface. The pressure

on one side is P_i and the pressure on the other side is P_o . When the surface tension is constant, the horizontal forces cancel each other because symmetry. In the vertical direction, the surface tension forces are pulling the surface upward. Thus, the pressure difference has to balance the surface tension. The forces in the vertical direction reads

$$(P_i - P_o) dl_1 dl_2 = \Delta P dl_1 dl_2 = 2 \sigma dl_1 \sin \beta_1 + 2 \sigma dl_2 \sin \beta_2 \quad (1.36)$$

For a very small area, the angles are very small and thus $(\sin \beta \sim \beta)$. Furthermore, it can be noticed that $dl_i \sim 2 R_i d\beta_i$. Thus, the equation (1.36) can be simplified as

$$\Delta P = \sigma \left(\frac{1}{R_1} + \frac{1}{R_2} \right) \quad (1.37)$$

Equation (1.37) predicts that pressure difference increase with inverse of the radius. There are two extreme cases: one) radius of infinite and radius of finite size. The second with two equal radii. The first case is for an infinite long cylinder for which the equation (1.37) is reduced to

$$\Delta P = \sigma \left(\frac{1}{R} \right) \quad (1.38)$$

Other extreme is for a sphere for which the main radii are the same and equation (1.37) is reduced to

$$\Delta P = \frac{2 \sigma}{R} \quad (1.39)$$

Where R is the radius of the sphere. A soap bubble is made of two layers, inner and outer, thus the pressure inside the bubble is

$$\Delta P = \frac{4 \sigma}{R} \quad (1.40)$$

Example 1.8:

A Tank filled with liquid, which contains n bubbles with equal radii, r . Calculate the minimum work required to increase the pressure in tank by ΔP . Assume that the liquid bulk modulus is infinity.

SOLUTION

The work is due to the change of the bubbles volume. The work is

$$w = \int_{r_0}^{r_f} \Delta P(v) dv \quad (1.41)$$

The minimum work will be for a reversible process. The reversible process requires very slow compression. It is worth noting that for very slow process, the temperature must

remain constant due to heat transfer. The relationship between pressure difference and the radius is described by equation (1.39) for reversible process. Hence the work is

$$w = \int_{r_0}^{r_f} \underbrace{\frac{\Delta P}{r}}_{\frac{2\sigma}{r}} \underbrace{4\pi r^2}_{dv} dr = 8\pi\sigma \int_{r_0}^{r_f} r dr = 4\pi\sigma (r_f^2 - r_0^2) \quad (1.42)$$

Where, r_0 is the radius at the initial stage and r_f is the radius at the final stage.

The work for n bubbles is then $4\pi\sigma n (r_f^2 - r_0^2)$. It can be noticed that the work is negative, that is the work is done on the system.

End Solution

1.6.1 Wetting of Surfaces

To explain the source of the contact angle, consider the point where three phases became in contact. This contact point occurs due to free surface reaching a solid boundary. The surface tension occurs between gas phase (G) to liquid phase (L) and also occurs between the solid (S) and the liquid phases as well as between the gas phase and the solid phase. In Figure 1.14, forces diagram is shown when control volume is chosen so that the masses of the solid, liquid, and gas can be ignored. Regardless to the magnitude of the surface tensions (except to zero) the forces cannot be balanced for the description of straight lines. For example, forces balanced along the line of solid boundary is

$$\sigma_{gs} - \sigma_{ls} - \sigma_{lg} \cos \beta = 0 \quad (1.43)$$

and in the tangent direction to the solid line the forces balance is

$$F_{solid} = \sigma_{lg} \sin \beta \quad (1.44)$$

substituting equation (1.44) into equation (1.43) yields

$$\sigma_{gs} - \sigma_{ls} = \frac{F_{solid}}{\tan \beta} \quad (1.45)$$

For $\beta = \pi/2 \implies \tan \beta = \infty$. Thus, the solid reaction force must be zero. The gas solid surface tension is different from the liquid solid surface tension and hence violating equation (1.43).

The surface tension forces must be balanced, thus, a contact angle is created to balance it. The contact angle is determined by whether the surface tension between the gas

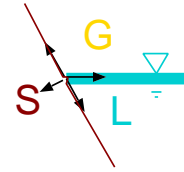


Fig. -1.14. Forces in Contact angle.

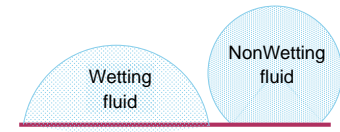


Fig. -1.15. Description of wetting and non-wetting fluids.

solid (gs) is larger or smaller than the surface tension of liquid solid (ls) and the local geometry. It must be noted that the solid boundary isn't straight. The surface tension is a molecular phenomenon, thus depends on the local structure of the surface and it provides the balance for these local structures.

The connection of the three phases—materials—mediums creates two situations which are categorized as wetting or non-wetting. There is a common definition of wetting the surface. If the angle of the contact between three materials is larger than 90° then it is non-wetting. On the other hand, if the angle is below 90° the material is wetting the surface (see Figure 1.15). The angle is determined by properties of the liquid, gas medium and the solid surface. And a small change on the solid surface can change the wetting condition to non-wetting. In fact there are commercial sprays that are intent to change the surface from wetting to non wetting. This fact is the reason that no reliable data can be provided with the exception to pure substances and perfect geometries. For example, water is described in many books as a wetting fluid. This statement is correct in most cases, however, when solid surface is made or coated with certain materials, the water is changed to be non-wetting (for example 3M selling product to "change" water to non-wetting). So, the wetness of fluids is a function of the solid as well.

Table -1.6. The contact angle for air, distilled water with selected materials to demonstrate the inconsistency.

chemical component	Contact Angle	Source
Steel	$\pi/3.7$	[1]
Steel,Nickel	$\pi/4.74$	[2]
Nickel	$\pi/4.74$ to $\pi/3.83$	[1]
Nickel	$\pi/4.76$ to $\pi/3.83$	[3]
Chrome-Nickel Steel	$\pi/3.7$	[4]
Silver	$\pi/6$ to $\pi/4.5$	[5]
Zink	$\pi/3.4$	[4]
Bronze	$\pi/3.2$	[4]
Copper	$\pi/4$	[4]
Copper	$\pi/3$	[7]
Copper	$\pi/2$	[8]

- 1 R. Siegel, E. G. Keshock (1975) "Effects of reduced gravity on nucleate boiling bubble dynamics in saturated water," AIChE Journal Volume 10 Issue 4, Pages 509 - 517. 1975
- 2 Bergles A. E. and Rohsenow W. M. "The determination of forced convection surface-

- boiling heat transfer, ASME, J. Heat Transfer, vol 1 pp 365 - 372.
- 3 Tolubinsky, V.I. and Ostrovsky, Y.N. (1966) "On the mechanism of boiling heat transfer",. International Journal of Heat and Mass Transfer, Vol. 9, No 12, pages 1465-1470.
 - 4 Arefeva E.I., Aladev O, I.T., (1958) "wlijanii smatchivaemosti na teploobmen pri kipenii," Injenerno Fizitcheskij Jurnal, 11-17 1(7) In Russian.
 - 5 Labuntsov D. A. (1939) "Approximate theory of heat transfer by developed nucleate boiling" In Sussian Izvestiya An SSSR , Energetika I transport, No 1.
 - 6 Basu, N., Warriar, G. R., and Dhir, V. K., (2002) "Onset of Nucleate Boiling and Active Nucleation Site Density during Subcooled Flow Boiling," ASME Journal of Heat Transfer, Vol. 124, pages 717 -728.
 - 7 Gaetner, R. F., and Westwater, J. W., (1960) "Population of Active Sites in Nucleate Boiling Heat Transfer," Chem. Eng. Prog. Symp., Ser. 56.
 - 8 Wang, C. H., and Dhir, V. K., (1993), "Effect of Surface Wettability on Active Nucleation Site Density During Pool Boiling of Water on a Vertical Surface," J. Heat Transfer 115, pp. 659-669

To explain the contour of the surface, and the contact angle consider simple "wetting" liquid contacting a solid material in two-dimensional shape as depicted in Figure 1.16. To solve the shape of the liquid surface, the pressure difference between the two sides of free surface has to be balanced by the surface tension. In Figure 1.16 describes the raising of the liquid as results of the surface tension. The surface tension reduces the pressure in the liquid above the liquid line (the dotted line in the Figure 1.16). The pressure just below the surface is $-g h(x) \rho$ (this pressure difference will be explained in more details in Chapter 4). The pressure, on the gas side, is the atmospheric pressure. This problem is a two dimensional problem and equation (1.38) is applicable to it. Appalling equation (1.38) and using the pressure difference yields

$$g h(x), \rho = \frac{\sigma}{R(x)} \quad (1.46)$$

The radius of any continuous function, $h = h(x)$, is

$$R(x) = \frac{\left(1 + [\dot{h}(x)]^2\right)^{3/2}}{\ddot{h}(x)} \quad (1.47)$$

Where \dot{h} is the derivative of h with respect to x .

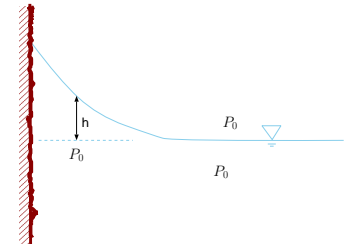


Fig. -1.16. Description of liquid surface.

Equation (1.47) can be derived either by forcing a circle at three points at $(x, x+dx, \text{ and } x+2dx)$ and thus finding the the diameter or by geometrical analysis of triangles build on points x and $x+dx$ (perpendicular to the tangent at these points). Substituting equation (1.47) into equation (1.46) yields

$$g h(x) \rho = \frac{\sigma}{\frac{\left(1 + [\dot{h}(x)]^2\right)^{3/2}}{\ddot{h}(x)}} \quad (1.48)$$

Equation (1.48) is non-linear differential equation for height and can be written as

$$\frac{g h \rho}{\sigma} \left(1 + \left[\frac{dh}{dx}\right]^2\right)^{3/2} - \frac{d^2h}{dx^2} = 0 \quad (1.49)$$

With the boundary conditions that specify either the derivative $\dot{h}(x = r) = 0$ (symmetry) and the derivative at $hx = \beta$ or heights in two points or other combinations. An alternative presentation of equation (1.48) is

$$g h \rho = \frac{\sigma \ddot{h}}{\left(1 + \dot{h}^2\right)^{3/2}} \quad (1.50)$$

Integrating equation (1.50) transforms into

$$\int \frac{g \rho}{\sigma} h dh = \int \frac{\ddot{h}}{\left(1 + \dot{h}^2\right)^{3/2}} dh \quad (1.51)$$

The constant $Lp \sigma / \rho g$ is referred to as Laplace's capillarity constant. The units of this constant are meter squared. The differential dh is \dot{h} . Using dummy variable and the identys $\dot{h} = \xi$ and hence, $\ddot{h} = \dot{\xi} = d\xi$ transforms equation (1.51) into

$$\int \frac{1}{Lp} h dh = \int \frac{\xi d\xi}{\left(1 + \xi^2\right)^{3/2}} \quad (1.52)$$

After the integration equation (1.52) becomes

$$\frac{h^2}{2 Lp} + constant = -\frac{1}{\left(1 + \dot{h}^2\right)^{1/2}} \quad (1.53)$$

At infinity, the height and the derivative of the height must be zero so $constant + 0 = -1/1$ and hence, $constant = -1$.

$$1 - \frac{h^2}{2Lp} = \frac{1}{(1 + \dot{h}^2)^{1/2}} \quad (1.54)$$

Equation (1.54) is a first order differential equation that can be solved by variables separation⁴. Equation (1.54) can be rearranged to be

$$(1 + \dot{h}^2)^{1/2} = \frac{1}{1 - \frac{h^2}{2Lp}} \quad (1.55)$$

Squaring both sides and moving the one to the right side yields

$$\dot{h}^2 = \left(\frac{1}{1 - \frac{h^2}{2Lp}} \right)^2 - 1 \quad (1.56)$$

The last stage of the separation is taking the square root of both sides to be

$$\dot{h} = \frac{dh}{dx} = \sqrt{\left(\frac{1}{1 - \frac{h^2}{2Lp}} \right)^2 - 1} \quad (1.57)$$

or

$$\frac{dh}{\sqrt{\left(\frac{1}{1 - \frac{h^2}{2Lp}} \right)^2 - 1}} = dx \quad (1.58)$$

Equation (1.58) can be integrated to yield

$$\int \frac{dh}{\sqrt{\left(\frac{1}{1 - \frac{h^2}{2Lp}} \right)^2 - 1}} = x + constant \quad (1.59)$$

The constant is determined by the boundary condition at $x = 0$. For example if $h(x = 0) = h_0$ then $constant = h_0$. This equation is studied extensively in classes on surface tension. Furthermore, this equation describes the dimensionless parameter that affects this phenomenon and this parameter will be studied in Chapter ?. This book is introductory, therefore this discussion on surface tension equation will be limited.

1.6.1.1 Capillarity

⁴This equation has an analytical solution which is $x = Lp\sqrt{4 - (h/Lp)^2} - Lp \operatorname{acosh}(2Lp/h) + constant$ where Lp is the Laplace constant. Shamefully, this author doesn't know how to show it in a two lines derivations.

1.6. SURFACE TENSION

The capillary forces referred to the fact that surface tension causes liquid to rise or penetrate into area (volume), otherwise it will not be there. It can be shown that the height that the liquid raised in a tube due to the surface tension is

$$h = \frac{2\sigma \cos \beta}{g \Delta \rho r} \quad (1.60)$$

Where $\Delta \rho$ is the difference of liquid density to the gas density and r is the radius of tube.

But this simplistic equation is unusable and useless unless the contact angle (assuming that the contact angle is constant or a repressive average can be found or provided or can be measured) is given. However, in reality there is no readily information for contact angle⁵ and therefore this equation is useful to show the trends. The maximum that the contact angle can be obtained in equation (1.60) when $\beta = 0$ and thus $\cos \beta = 1$. This angle is obtained when a perfect half a sphere shape exist of the liquid surface. In that case equation (1.60) becomes

$$h_{max} = \frac{2\sigma}{g \Delta \rho r} \quad (1.61)$$

Figure 1.18 exhibits the height as a function of the radius of the tube. The height based on equation (1.61) is shown in Figure 1.17 as blue line. The actual height is shown in the red line. Equation (1.61) provides reasonable results only in a certain range. For a small tube radius, equation (1.49) proved better results because the curve approaches hemispherical shaper (small gravity effect). For large radii equation (1.49) approaches the strait line (the liquid line) strong gravity effect. On the other hand, for extremely small radii equation (1.61) indicates that the high height which indicates a negative pressure. The liquid at a certain pressure will be vaporized and will breakdown the model upon this equation was constructed. Furthermore, the small scale indicates that the simplistic and continuous approach is not appropriate and a different model is needed. The conclusion of this discussion are shown in Figure 1.17. The actual dimension for many liquids (even water) is about 1-5 [mm].

The discussion above was referred to “wetting” contact angle. The depression of the liquid occurs in a “negative” contact angle similarly to “wetting.” The depression height, h is similar to equation (1.61) with a minus sign. However, the gravity is working against the surface tension and reducing the range and quality of the predictions of equation (1.61). The measurements of the height of distilled water and mercury are

⁵Actually, there are information about the contact angle. However, that information conflict each other and no real information is available see Table 1.6.

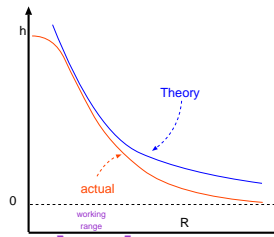
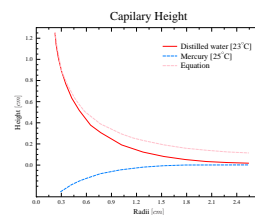


Fig. -1.17. The raising height as a function of the radii.



May 29, 2008

Fig. -1.18. The raising height as a function of the radius.

presented in Figure 1.18. The experimental results of these materials are with agreement with the discussion above.

The surface tension of a selected material is given in Table 1.7.

In conclusion, the surface tension issue is important only in case where the radius is very small and gravity is negligible. The surface tension depends on the two materials or mediums that it separates.

Example 1.9:

Calculate the diameter of a water droplet to attain pressure difference of $1000[N/m^2]$. You can assume that temperature is $20^\circ C$.

SOLUTION

The pressure inside the droplet is given by equation (1.39).

$$D = 2R = \frac{2\sigma}{\Delta P} = \frac{4 \times 0.0728}{1000} \sim 2.912 \cdot 10^{-4} [m]$$

End Solution

Example 1.10:

Calculate the pressure difference between a droplet of water at $20^\circ C$ when the droplet has a diameter of 0.02 cm.

SOLUTION

using equation

$$\Delta P = \frac{2\sigma}{r} \sim \frac{2 \times 0.0728}{0.0002} \sim 728.0 [N/m^2]$$

End Solution

Example 1.11:

Calculate the maximum force necessary to lift a thin wire ring of $0.04[m]$ diameter from a water surface at $20^\circ C$. Neglect the weight of the ring.

SOLUTION

$$F = 2(2\pi r \sigma) \cos \beta$$

The actual force is unknown since the contact angle is unknown. However, the maximum Force is obtained when $\beta = 0$ and thus $\cos \beta = 1$. Therefore,

$$F = 4\pi r \sigma = 4 \times \pi \times 0.04 \times 0.0728 \sim .0366 [N]$$

In this value the gravity is not accounted for.

End Solution

Table -1.7. The surface tension for selected materials at temperature 20°C when not mentioned.

chemical component	Surface Tension $\frac{mN}{m}$	T	correction $\frac{mN}{mK}$
Acetic Acid	27.6	20°C	n/a
Acetone	25.20	-	-0.1120
Aniline	43.4	22°C	-0.1085
Benzene	28.88	-	-0.1291
Benzylalcohol	39.00	-	-0.0920
Benzylbenzoate	45.95	-	-0.1066
Bromobenzene	36.50	-	-0.1160
Bromobenzene	36.50	-	-0.1160
Bromoform	41.50	-	-0.1308
Butyronitrile	28.10	-	-0.1037
Carbon disulfid	32.30	-	-0.1484
Quinoline	43.12	-	-0.1063
Chloro benzene	33.60	-	-0.1191
Chloroform	27.50	-	-0.1295
Cyclohexane	24.95	-	-0.1211
Cyclohexanol	34.40	25°C	-0.0966
Cyclopentanol	32.70	-	-0.1011
Carbon Tetrachloride	26.8	-	n/a
Carbon disulfid	32.30	-	-0.1484
Chlorobutane	23.10	-	-0.1117
Ethyl Alcohol	22.3	-	n/a
Ethanol	22.10	-	-0.0832
Ethylbenzene	29.20	-	-0.1094
Ethylbromide	24.20	-	-0.1159
Ethylene glycol	47.70	-	-0.0890
Formamide	58.20	-	-0.0842
Gasoline	~ 21	-	n/a
Glycerol	64.0	-	-0.0598
Helium	0.12	-269°C	n/a
Mercury	425-465.0	-	-0.2049
Methanol	22.70	-	-0.0773
Methyl naphthalene	38.60	-	-0.1118
Methyl Alcohol	22.6	-	n/a
Neon	5.15	-247°C	n/a
Nitrobenzene	43.90	-	-0.1177
Olive Oil	43.0-48.0	-	-0.067
Perfluoroheptane	12.85	-	-0.0972
Perfluorohexane	11.91	-	-0.0935

Table -1.7. The surface tension for selected materials (continue)

chemical component	Surface Tension $\frac{mN}{m}$	T	correction $\frac{mN}{m K}$
Perfluorooctane	14.00	-	-0.0902
Phenylisothiocyanate	41.50	-	-0.1172
Propanol	23.70	25°C	-0.0777
Pyridine	38.00	-	-0.1372
Pyrrol	36.60	-	-0.1100
SAE 30 Oil	n/a	-	n/a
Seawater	54-69	-	n/a
Toluene	28.4	-	-0.1189
Turpentine	27	-	n/a
Water	72.80	-	-0.1514
o-Xylene	30.10	-	-0.1101
m-Xylene	28.90	-	-0.1104

CHAPTER 2

Review of Thermodynamics

In this chapter, a review of several definitions of common thermodynamics terms is presented. This introduction is provided to bring the student back to current place with the material.

2.1 *Basic Definitions*

The following basic definitions are common to thermodynamics and will be used in this book.

Work

In mechanics, the work was defined as

$$\text{mechanical work} = \int \mathbf{F} \cdot d\mathbf{l} = \int PdV \quad (2.1)$$

This definition can be expanded to include two issues. The first issue that must be addressed, that work done on the surroundings by the system boundaries similarly is positive. Two, there is a transfer of energy so that its effect can cause work. It must be noted that electrical current is a work while heat transfer isn't.

System

This term will be used in this book and it is defined as a continuous (at least partially) fixed quantity of matter. The dimensions of this material can be changed. In this definition, it is assumed that the system speed is significantly lower than that of the speed of light. So, the mass can be assumed constant even though the true conservation law applied to the combination of mass energy (see Einstein's law). In fact for almost all engineering purpose this law is reduced to two separate laws of mass conservation and energy conservation.

Our system can receive energy, work, etc as long the mass remain constant the definition is not broken.

Thermodynamics First Law

This law refers to conservation of energy in a non accelerating system. Since all the systems can be calculated in a non accelerating systems, the conservation is applied to all systems. The statement describing the law is the following.

$$Q_{12} - W_{12} = E_2 - E_1 \quad (2.2)$$

The system energy is a state property. From the first law it directly implies that for process without heat transfer (adiabatic process) the following is true

$$W_{12} = E_1 - E_2 \quad (2.3)$$

Interesting results of equation (2.3) is that the way the work is done and/or intermediate states are irrelevant to final results. There are several definitions/separations of the kind of works and they include kinetic energy, potential energy (gravity), chemical potential, and electrical energy, etc. The internal energy is the energy that depends on the other properties of the system. For example for pure/homogeneous and simple gases it depends on two properties like temperature and pressure. The internal energy is denoted in this book as E_U and it will be treated as a state property.

The potential energy of the system is depended on the body force. A common body force is the gravity. For such body force, the potential energy is mgz where g is the gravity force (acceleration), m is the mass and the z is the vertical height from a datum. The kinetic energy is

$$K.E. = \frac{mU^2}{2} \quad (2.4)$$

Thus the energy equation can be written as

$$\frac{mU_1^2}{2} + mgz_1 + E_{U1} + Q = \frac{mU_2^2}{2} + mgz_2 + E_{U2} + W \quad (2.5)$$

For the unit mass of the system equation (2.5) is transformed into

$$\frac{U_1^2}{2} + gz_1 + E_{u1} + q = \frac{U_2^2}{2} + gz_2 + E_{u2} + w \quad (2.6)$$

where q is the energy per unit mass and w is the work per unit mass. The "new" internal energy, E_u , is the internal energy per unit mass.

Since the above equations are true between arbitrary points, choosing any point in time will make it correct. Thus differentiating the energy equation with respect to time yields the rate of change energy equation. The rate of change of the energy transfer is

$$\frac{DQ}{Dt} = \dot{Q} \quad (2.7)$$

In the same manner, the work change rate transferred through the boundaries of the system is

$$\frac{DW}{Dt} = \dot{W} \quad (2.8)$$

Since the system is with a fixed mass, the rate energy equation is

$$\dot{Q} - \dot{W} = \frac{DE_U}{Dt} + mU \frac{DU}{Dt} + m \frac{DB_f z}{Dt} \quad (2.9)$$

For the case were the body force, B_f , is constant with time like in the case of gravity equation (2.9) reduced to

$$\dot{Q} - \dot{W} = \frac{DE_U}{Dt} + mU \frac{DU}{Dt} + mg \frac{Dz}{Dt} \quad (2.10)$$

The time derivative operator, D/Dt is used instead of the common notation because it referred to system property derivative.

Thermodynamics Second Law

There are several definitions of the second law. No matter which definition is used to describe the second law it will end in a mathematical form. The most common mathematical form is Clausius inequality which state that

$$\oint \frac{\delta Q}{T} \geq 0 \quad (2.11)$$

The integration symbol with the circle represent integral of cycle (therefor circle) in with system return to the same condition. If there is no lost, it is referred as a reversible process and the inequality change to equality.

$$\oint \frac{\delta Q}{T} = 0 \quad (2.12)$$

The last integral can go though several states. These states are independent of the path the system goes through. Hence, the integral is independent of the path. This observation leads to the definition of entropy and designated as S and the derivative of entropy is

$$ds \equiv \left(\frac{\delta Q}{T} \right)_{\text{rev}} \quad (2.13)$$

Performing integration between two states results in

$$S_2 - S_1 = \int_1^2 \left(\frac{\delta Q}{T} \right)_{\text{rev}} = \int_1^2 dS \quad (2.14)$$

One of the conclusions that can be drawn from this analysis is for reversible and adiabatic process $dS = 0$. Thus, the process in which it is reversible and adiabatic, the entropy remains constant and referred to as isentropic process. It can be noted that there is a possibility that a process can be irreversible and the right amount of heat transfer to have zero change entropy change. Thus, the reverse conclusion that zero change of entropy leads to reversible process, isn't correct.

For reversible process equation (2.12) can be written as

$$\delta Q = TdS \quad (2.15)$$

and the work that the system is doing on the surroundings is

$$\delta W = PdV \quad (2.16)$$

Substituting equations (2.15) (2.16) into (2.10) results in

$$TdS = dE_U + PdV \quad (2.17)$$

Even though the derivation of the above equations were done assuming that there is no change of kinetic or potential energy, it still remain valid for all situations. Furthermore, it can be shown that it is valid for reversible and irreversible processes.

Enthalpy

It is a common practice to define a new property, which is the combination of already defined properties, the enthalpy of the system.

$$H = E_U + PV \quad (2.18)$$

The specific enthalpy is enthalpy per unit mass and denoted as, h .

Or in a differential form as

$$dH = dE_U + dPV + PdV \quad (2.19)$$

Combining equations (2.18) the (2.17) yields

$$TdS = dH - VdP \quad (2.20)$$

For isentropic process, equation (2.17) is reduced to $dH = VdP$. The equation (2.17) in mass unit is

$$Tds = du + Pdv = dh - \frac{dP}{\rho} \quad (2.21)$$

when the density enters through the relationship of $\rho = 1/v$.

Specific Heats

The change of internal energy and enthalpy requires new definitions. The first change of the internal energy and it is defined as the following

$$C_v \equiv \left(\frac{\partial E_u}{\partial T} \right) \quad (2.22)$$

And since the change of the enthalpy involve some kind of work is defined as

$$C_p \equiv \left(\frac{\partial h}{\partial T} \right) \quad (2.23)$$

The ratio between the specific pressure heat and the specific volume heat is called the ratio of the specific heat and it is denoted as, k .

$$k \equiv \frac{C_p}{C_v} \quad (2.24)$$

For solid, the ratio of the specific heats is almost 1 and therefore the difference between them is almost zero. Commonly the difference for solid is ignored and both are assumed to be the same and therefore referred as C . This approximation less strong for liquid but not by that much and in most cases it applied to the calculations. The ratio the specific heat of gases is larger than one.

Equation of state

Equation of state is a relation between state variables. Normally the relationship of temperature, pressure, and specific volume define the equation of state for gases. The simplest equation of state referred to as ideal gas. and it is defined as

$$P = \rho RT \quad (2.25)$$

Application of Avogadro's law, that "all gases at the same pressures and temperatures have the same number of molecules per unit of volume," allows the calculation of a "universal gas constant." This constant to match the standard units results in

$$\bar{R} = 8.3145 \frac{kJ}{kmol K} \quad (2.26)$$

Thus, the specific gas can be calculate as

$$R = \frac{\bar{R}}{M} \quad (2.27)$$

Table -2.1. Properties of Various Ideal Gases [300K]

Gas	Chemical Formula	Molecular Weight	$R \left[\frac{kJ}{KgK} \right]$	$C_P \left[\frac{kJ}{KgK} \right]$	$C_v \left[\frac{kJ}{KgK} \right]$	k
Air	-	28.970	0.28700	1.0035	0.7165	1.400
Argon	Ar	39.948	0.20813	0.5203	0.3122	1.667
Butane	C_4H_{10}	58.124	0.14304	1.7164	1.5734	1.091
Carbon Dioxide	CO_2	44.01	0.18892	0.8418	0.6529	1.289
Carbon Monoxide	CO	28.01	0.29683	1.0413	0.7445	1.400
Ethane	C_2H_6	30.07	0.27650	1.7662	1.4897	1.186
Ethylene	C_2H_4	28.054	0.29637	1.5482	1.2518	1.237
Helium	He	4.003	2.07703	5.1926	3.1156	1.667
Hydrogen	H_2	2.016	4.12418	14.2091	10.0849	1.409
Methane	CH_4	16.04	0.51835	2.2537	1.7354	1.299
Neon	Ne	20.183	0.41195	1.0299	0.6179	1.667
Nitrogen	N_2	28.013	0.29680	1.0416	0.7448	1.400
Octane	C_8H_{18}	114.230	0.07279	1.7113	1.6385	1.044
Oxygen	O_2	31.999	0.25983	0.9216	0.6618	1.393
Propane	C_3H_8	44.097	0.18855	1.6794	1.4909	1.126
Steam	H_2O	18.015	0.48152	1.8723	1.4108	1.327

The specific constants for select gas at 300K is provided in table 2.1.
From equation (2.25) of state for perfect gas it follows

$$d(Pv) = RdT \quad (2.28)$$

For perfect gas

$$dh = dE_u + d(Pv) = dE_u + d(RT) = f(T) \text{ (only)} \quad (2.29)$$

From the definition of enthalpy it follows that

$$d(Pv) = dh - dE_u \quad (2.30)$$

Utilizing equation (2.28) and subsisting into equation (2.30) and dividing by dT yields

$$C_p - C_v = R \quad (2.31)$$

This relationship is valid only for ideal/perfect gases.

The ratio of the specific heats can be expressed in several forms as

$$C_v = \frac{R}{k-1} \quad (2.32)$$

$$C_p = \frac{kR}{k-1} \quad (2.33)$$

The specific heat ratio, k value ranges from unity to about 1.667. These values depend on the molecular degrees of freedom (more explanation can be obtained in Van Wylen "F. of Classical thermodynamics." The values of several gases can be approximated as ideal gas and are provided in Table (2.1).

The entropy for ideal gas can be simplified as the following

$$s_2 - s_1 = \int_1^2 \left(\frac{dh}{T} - \frac{dP}{\rho T} \right) \quad (2.34)$$

Using the identities developed so far one can find that

$$s_2 - s_1 = \int_1^2 C_p \frac{dT}{T} - \int_1^2 \frac{R dP}{P} = C_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \quad (2.35)$$

Or using specific heat ratio equation (2.35) transformed into

$$\frac{s_2 - s_1}{R} = \frac{k}{k-1} \ln \frac{T_2}{T_1} - \ln \frac{P_2}{P_1} \quad (2.36)$$

For isentropic process, $\Delta s = 0$, the following is obtained

$$\ln \frac{T_2}{T_1} = \ln \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} \quad (2.37)$$

There are several famous identities that results from equation (2.37) as

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} = \left(\frac{P_2}{P_1} \right)^{k-1} \quad (2.38)$$

The ideal gas model is a simplified version of the real behavior of real gas. The real gas has a correction factor to account for the deviations from the ideal gas model. This correction factor referred as the compressibility factor and defined as

$$Z = \frac{PV}{RT} \quad (2.39)$$

CHAPTER 3

Review of Mechanics

This author would like to express his gratitude to Dan Olsen (former Minneapolis city Engineer) and his friend Richard Hackbarth.

This chapter provides a review of important definitions and concepts from Mechanics (statics and dynamics). These concepts and definitions will be used in this book and a review is needed.

3.1 Center of Mass

The center of mass is divided into two sections, first, center of the mass and two, center of area (two-dimensional body with equal distribution mass).

3.1.1 Center of the Mass

In many engineering problems, the center of mass is required to make the calculations. This concept is derived from the fact that a body has a center of mass/gravity which interacts with other bodies and that this force acts on the center (equivalent force). It turns out that this concept is very useful in calculating rotations, moment of inertia, etc. The center of mass doesn't depend on the coordinate system and on the way it is calculated. The physical meaning of the center of mass is that if a straight line force acts on the body in away through the center of gravity, the body will not rotate. In other words, if a body will be held by one point it will be enough to hold the body in the direction of the center of mass. Note, if the body isn't be held through the center of mass, then a moment in addition to force is required (to prevent the body for rotating). It is convenient to use the Cartesian system to explain this concept. Suppose that the body has a distribution of the mass (density, ρ) as a function of the location. The density "normally" defined as mass per volume. Here, the the line density is referred to density mass per unit length in the x direction.

In x coordinate, the center will be defined as

$$\bar{x} = \frac{1}{m} \int_V x \overbrace{\rho(x)dV}^{dm} \quad (3.1)$$

Here, the dV element has finite dimensions in y - z plane and infinitesimal dimension in x direction see Figure 3.1. Also, the mass, m is the total mass of the object. It can be noticed that center of mass in the x -direction isn't affected by the distribution in the y nor by z directions. In same fashion the center of mass can be defined in the other directions as following

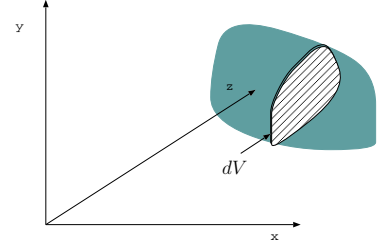


Fig. -3.1. Description of how the center of mass is calculated.

$$\bar{x}_i = \frac{1}{m} \int_V x_i \rho(x_i) dV \quad (3.2)$$

where x_i is the direction of either, x , y or z . The density, $\rho(x_i)$ is the line density as function of x_i . Thus, even for solid and uniform density the line density is a function of the geometry.

3.1.2 Center of Area

In the previous case, the body was a three dimensional shape. There are cases where the body can be approximated as a two-dimensional shape because the body is with a thin with uniform density. Consider a uniform thin body with constant thickness shown in Figure 3.2 which has density, ρ . Thus, equation (3.1) can be transferred into

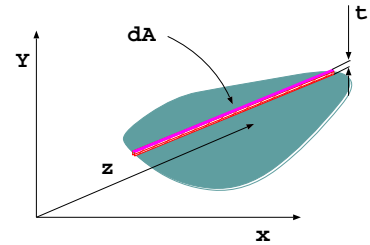


Fig. -3.2. Thin body center of mass/area schematic.

$$\bar{x} = \frac{1}{\underbrace{tA}_V \rho} \int_V x \overbrace{\rho t dA}^{dm} \quad (3.3)$$

The density, ρ and the thickness, t , are constant and can be canceled. Thus equation (3.3) can be transferred into

$$\bar{x}_i = \frac{1}{A} \int_A x_i dA \quad (3.4)$$

when the integral now over only the area as oppose over the volume.

Finding the centroid location should be done in the most convenient coordinate system since the location is coordinate independent.

3.2 Moment of Inertia

As it was divided for the body center of mass, the moment of inertia is divided into moment of inertia of mass and area.

3.2.1 Moment of Inertia for Mass

The moment of inertia turns out to be an essential part for the calculations of rotating bodies. Furthermore, it turns out that the moment of inertia has much wider applicability. Moment of inertia of mass is defined as

$$I_{rrm} = \int_m \rho r^2 dm \quad (3.5)$$

If the density is constant then equation (3.5) can be transformed into

$$I_{rrm} = \rho \int_V r^2 dV \quad (3.6)$$

The moment of inertia is independent of the coordinate system used for the calculation, but dependent on the location of axis of rotation relative to the body. Some people define the radius of gyration as an equivalent concepts for the center of mass concept and which means if all the mass were to locate in the one point/distance and to obtain the same of moment of inertia.

$$r_k = \sqrt{\frac{I_m}{m}} \quad (3.7)$$

The body has a different moment of inertia for every coordinate/axis and they are

$$\begin{aligned} I_{xx} &= \int_V r_x^2 dm = \int_V (y^2 + z^2) dm \\ I_{yy} &= \int_V r_y^2 dm = \int_V (x^2 + z^2) dm \\ I_{zz} &= \int_V r_z^2 dm = \int_V (x^2 + y^2) dm \end{aligned} \quad (3.8)$$

3.2.2 Moment of Inertia for Area

3.2.2.1 General Discussion

For body with thickness, t and uniform density the following can be written

$$I_{xxm} = \int_m r^2 dm = \rho t \overbrace{\int_A r^2 dA}^{\text{moment of inertia for area}} \quad (3.9)$$

The moment of inertia about axis is x can be defined as

$$I_{xx} = \int_A r^2 dA = \frac{I_{xxm}}{\rho t} \quad (3.10)$$

where r is distance of dA from the axis x and t is the thickness. Any point distance can be calculated from axis x as

$$x = \sqrt{y^2 + z^2} \quad (3.11)$$

Thus, equation (3.10) can be written as

$$I_{xx} = \int_A (y^2 + z^2) dA \quad (3.12)$$

In the same fashion for other two coordinates as

$$I_{yy} = \int_A (x^2 + z^2) dA \quad (3.13)$$

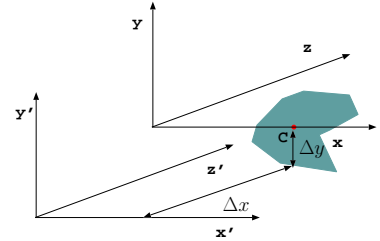


Fig. -3.3. The schematic that explains the summation of moment of inertia.

$$I_{zz} = \int_A (x^2 + y^2) dA \quad (3.14)$$

3.2.2.2 The Parallel Axis Theorem

The moment of inertia can be calculated for any axis. The knowledge about one axis can help calculating the moment of inertia for a parallel axis. Let I_{xx} the moment of inertia about axis xx which is at the center of mass/area.

The moment of inertia for axis x' is

$$I_{x'x'} = \int_A r'^2 dA = \int_A (y'^2 + z'^2) dA = \int_A [(y + \Delta y)^2 + (z + \Delta z)^2] dA \quad (3.15)$$

equation (3.15) can be expended as

$$I_{x'x'} = \overbrace{\int_A (y^2 + z^2) dA}^{I_{xx}} + \overbrace{2 \int_A (y \Delta y + z \Delta z) dA}^{=0} + \int_A ((\Delta y)^2 + (\Delta z)^2) dA \quad (3.16)$$

The first term in equation (3.16) on the right hand side is the moment of inertia about axis x and the second term is zero. The second term is zero because it is an integral of center about center thus is zero. The third term is a new term and can be written as

$$\int_A \overbrace{((\Delta y)^2 + (\Delta z)^2)}^{\text{constant}} dA = \overbrace{((\Delta y)^2 + (\Delta z)^2)}^{r^2} \overbrace{\int_A dA}^A = r^2 A \quad (3.17)$$

Hence, the relationship between the moment of inertia at xx and parallel axis $x'x'$ is

$$I_{x'x'} = I_{xx} + r^2 A \quad (3.18)$$

The moment of inertia of several areas is the sum of moment of inertia of each area see Figure 3.4 and therefore,

$$I_{xx} = \sum_{i=1}^n I_{xxi} \quad (3.19)$$

If the same areas are similar thus

$$I_{xx} = \sum_{i=1}^n I_{xxi} = n I_{xxi} \quad (3.20)$$

Equation (3.20) is very useful in the calculation of the moment of inertia utilizing the moment of inertia of known bodies. For example, the moment of inertia of half a circle is half of whole circle for axis at the center of circle. The moment of inertia can then move the center of area. of the

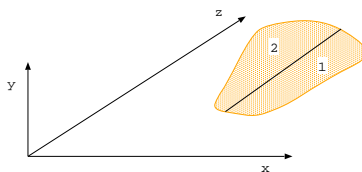


Fig. -3.4. The schematic to explain the summation of moment of inertia.

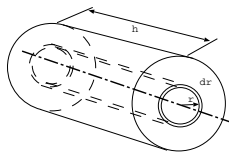


Fig. -3.5. Cylinder with the element for calculation moment of inertia.

3.2.3 Examples of Moment of Inertia

Example 3.1:

Calculate the moment of inertia for the mass of the cylinder about center axis which height of h and radius, r_0 , as shown in Figure 3.5. The material is with an uniform density and homogeneous.

SOLUTION

The element can be calculated using cylindrical coordinate. Here the convenient element is a shell of thickness dr which shown in Figure 3.5 as

$$I_{rr} = \rho \int_V r^2 dm = \rho \int_0^{r_0} r^2 \overbrace{h 2\pi r dr}^{dV} = \rho h 2\pi \frac{r_0^4}{4} = \frac{1}{2} \rho h \pi r_0^4 = \frac{1}{2} m r_0^2$$

The radius of gyration is

$$r_k = \sqrt{\frac{\frac{1}{2} m r_0^2}{m}} = \frac{r_0}{\sqrt{2}}$$

End Solution

Example 3.2:

Calculate the moment of inertia of the rectangular shape shown in Figure 3.6 around x coordinate.

SOLUTION

The moment of inertia is calculated utilizing equation (3.12) as following

$$I_{xx} = \int_A \left(\overbrace{y^2}^0 + z^2 \right) dA = \int_0^a z^2 \overbrace{bdz}^{dA} = \frac{a^3 b}{3}$$

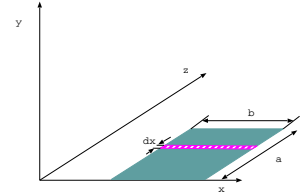


Fig. -3.6. Description of rectangular in x - y plane for calculation of moment of inertia.

This value will be used in later examples.

End Solution

Example 3.3:

To study the assumption of zero thickness, consider a simple shape to see the effects of this assumption. Calculate the moment of inertia about the center of mass of a square shape with a thickness, t compare the results to a square shape with zero thickness.

SOLUTION

The moment of inertia of transverse slice about y' (see Figure mech:fig:squareEll) is

$$dI_{xxm} = \rho \overbrace{dy}^t \overbrace{\frac{I_{xx}}{12}}^{ba^3} \quad (3.21)$$

The transformation into from local axis x to center axis, x' can be done as following

$$dI_{x'x'm} = \rho dy \left(\overbrace{\frac{I_{xx}}{12}}^{ba^3} + \underbrace{z^2}_{r^2} \underbrace{ba}_{A} \right) \quad (3.22)$$

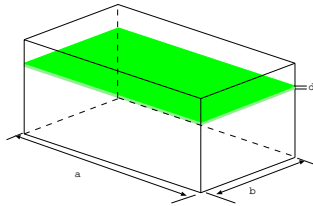


Fig. -3.7. A square element for the calculations of inertia of two-dimensional to three-dimensional deviations.

The total moment of inertia can be obtained by integration of equation (3.22) to write as

$$I_{xxm} = \rho \int_{-t/2}^{t/2} \left(\frac{ba^3}{12} + z^2 ba \right) dz = \rho t \frac{abt^2 + a^3b}{12} \quad (3.23)$$

Comparison with the thin body results in

$$\frac{I_{xx} \rho t}{I_{xxm}} = \frac{ba^3}{t^2 ba + ba^3} = \frac{1}{1 + \frac{t^2}{a^2}} \quad (3.24)$$

It can be noticed right away that equation (3.24) indicates that ratio approaches one when thickness ratio is approaches zero, $I_{xxm}(t \rightarrow 0) \rightarrow 1$. Additionally it can be noticed that the ratio a^2/t^2 is the only contributor to the error¹. The results are present in Figure 3.8. It can be noticed that the error is significant very fast even for small values of t/a while the with of the box, b has no effect on the error.

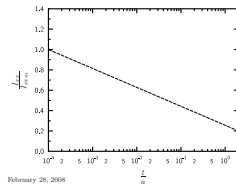


Fig. -3.8. The ratio of the moment of inertia of two-dimensional to three-dimensional.

End Solution

Example 3.4:

Calculate the center of area and moment of inertia of the parabola, $y = \alpha x^2$, shown in Figure 3.9.

¹This ratio is a dimensionless number that commonly has no special name. This author suggests to call this ratio as the B number.

SOLUTION

For $y = b$ the value of $x = \sqrt{b/\alpha}$. First the area inside the parabola calculated as

$$A = 2 \int_0^{\sqrt{b/\alpha}} \underbrace{\frac{dA}{2}}_{(b - \alpha\xi^2)} d\xi = \frac{2(3\alpha - 1)}{3} \left(\frac{b}{\alpha}\right)^{\frac{3}{2}}$$

The center of area can be calculated utilizing equation (3.4). The center of every element is at, $\left(\alpha\xi^2 + \frac{b - \alpha\xi^2}{2}\right)$ the element area is used before and therefore

$$x_c = \frac{1}{A} \int_0^{\sqrt{b/\alpha}} \underbrace{\left(\alpha\xi^2 + \frac{b - \alpha\xi^2}{2}\right)}_{x_c} \underbrace{(b - \alpha\xi^2)}_{dA} d\xi = \frac{3\alpha b}{15\alpha - 5} \tag{3.25}$$

The moment of inertia of the area about the center can be found using in equation (3.25) can be done in two steps first calculate the moment of inertia in this coordinate system and then move the coordinate system to center. Utilizing equation (3.12) and doing the integration from 0 to maximum y provides

$$I_{x'x'} = 4 \int_0^b \xi^2 \sqrt{\frac{\xi}{\alpha}} d\xi = \frac{2b^{7/2}}{7\sqrt{\alpha}}$$

Utilizing equation (3.18)

$$I_{xx} = I_{x'x'} - A \Delta x^2 = \frac{4b^{7/2}}{7\sqrt{\alpha}} - \frac{3\alpha - 1}{3} \left(\frac{b}{\alpha}\right)^{\frac{3}{2}} \left(\frac{3\alpha b}{15\alpha - 5}\right)^2$$

or after working the details results in

$$I_{xx} = \frac{\sqrt{b} (20b^3 - 14b^2)}{35\sqrt{\alpha}}$$

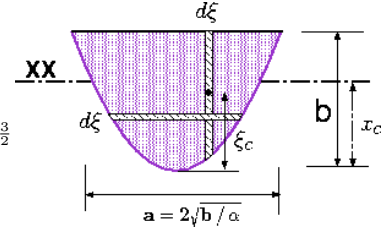
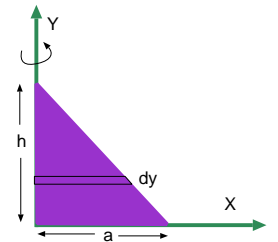


Fig. -3.9. Description of parabola for calculation of moment of inertia and center of area.

End Solution

Example 3.5:

Calculate the momenet of inertia of strait angle traingle about its y axis as shown in the Figure on the right. Assume that base is a and the hieght is h. What is the moment when a simetrical traingle is attached on left. What is the moment when a simetrical traingle is attached on bottom. What is the moment inertia when $a \rightarrow 0$. What is the moment inertia when $h \rightarrow 0$.



SOLUTION

The right edge line equation can be calculated as

$$\frac{y}{h} = \left(1 - \frac{x}{a}\right)$$

or

$$\frac{x}{a} = \left(1 - \frac{y}{h}\right)$$

Now using the moment of inertia of rectangle on the side (y) coordinate (see example (3.2))

$$\int_0^h \frac{a \left(1 - \frac{y}{h}\right)^3 dy}{3} = \frac{a^3 h}{4}$$

For two triangles attached to each other the moment of inertia will be sum as $\frac{a^3 h}{2}$
The rest is under construction.

End Solution

3.2.4 Product of Inertia

In addition to the moment of inertia, the product of inertia is commonly used. Here only the product of the area is defined and discussed. The product of inertia defined as

$$I_{x_i x_j} = \int_A x_i x_j dA \quad (3.26)$$

For example, the product of inertia for x and y axes is

$$I_{xy} = \int_A x y dA \quad (3.27)$$

Product of inertia can be positive or negative value as oppose the moment of inertia. The calculation of the product of inertia isn't different much for the calculation of the moment of inertia. The units of the product of inertia are the same as for moment of inertia.

Transfer of Axis Theorem

Same as for moment of inertia there is also similar theorem.

$$I_{x' y'} = \int_A x' y' dA = \int_A (x + \Delta x)(y + \Delta y) dA \quad (3.28)$$

expanding equation (3.28) results in

$$I_{x' y'} = \underbrace{\int_A x y dA}_{I_{xy}} + \overbrace{\int_A x dA}^{\Delta y} \Delta y + \overbrace{\int_A y dA}^{\Delta x} \Delta x + \underbrace{\int_A \Delta x \Delta y dA}_{\Delta x \Delta y A} \quad (3.29)$$

The final form is

$$I_{x'y'} = I_{xy} + \Delta x \Delta y A \quad (3.30)$$

There are several relationships should be mentioned

$$I_{xy} = I_{yx} \quad (3.31)$$

Symmetrical area has zero product of inertia.

Example 3.6:

Calculate the product of inertia of straight edge triangle.

SOLUTION

The equation of the line is

$$y = \frac{a}{b}x + a$$

The product of inertia at the center is zero. The total product of inertia is

$$I_{x'y'} = 0 + \underbrace{\frac{\Delta x}{3}}_a \underbrace{\frac{\Delta y}{3}}_b \underbrace{\left(\frac{A}{2}\right)}_{ab} = \frac{a^2 b^2}{18}$$

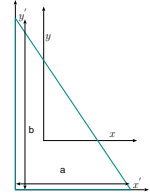


Fig. -3.10. Product of inertia for triangle.

End Solution

3.2.5 Principal Axes of Inertia

The inertia matrix or inertia tensor is

$$\begin{vmatrix} I_{xx} & -I_{xy} & -I_{xz} \\ -I_{yx} & I_{yy} & -I_{yz} \\ -I_{zx} & -I_{zy} & I_{zz} \end{vmatrix} \quad (3.32)$$

In linear algebra it was shown that for some angle equation (3.32) can be transform into

$$\begin{vmatrix} I_{x'x'} & 0 & 0 \\ 0 & I_{y'y'} & 0 \\ 0 & 0 & I_{z'z'} \end{vmatrix} \quad (3.33)$$

System which creates equation (3.33) referred as principle system.

3.3 Newton's Laws of Motion

These laws can be summarized in two statements one, for every action by body **A** on Body **B** there is opposite reaction by body **B** on body **A**. Two, which can expressed in mathematical form as

$$\sum \mathbf{F} = \frac{D(mU)}{Dt} \quad (3.34)$$

It can be noted that D replaces the traditional d since the additional meaning which be added. Yet, it can be treated as the regular derivative. This law apply to any body and any body can "broken" into many small bodies which connected to each other. These small "bodies" when became small enough equation (3.34) can be transformed to a continuous form as

$$\sum \mathbf{F} = \int_V \frac{D(\rho U)}{Dt} dV \quad (3.35)$$

The external forces are equal to internal forces the forces between the "small" bodies are cancel each other. Yet this examination provides a tool to study what happened in the fluid during operation of the forces.

Since the derivative with respect to time is independent of the volume, the derivative can be taken out of the integral and the alternative form can be written as

$$\sum \mathbf{F} = \frac{D}{Dt} \int_V \rho U dV \quad (3.36)$$

The velocity, U is a derivative of the location with respect to time, thus,

$$\sum \mathbf{F} = \frac{D^2}{Dt^2} \int_V \rho r dV \quad (3.37)$$

where r is the location of the particles from the origin.

The external forces are typically divided into two categories: body forces and surface forces. The body forces are forces that act from a distance like magnetic field or gravity. The surface forces are forces that act on the surface of the body (pressure, stresses). The same as in the dynamic class, the system acceleration called the internal forces. The acceleration is divided into three categories: Centrifugal, $\boldsymbol{\omega} \times (\mathbf{r} \times \boldsymbol{\omega})$, Angular, $\mathbf{r} \times \dot{\boldsymbol{\omega}}$, Coriolis, $2(\mathbf{U}_r \times \boldsymbol{\omega})$. The radial velocity is denoted as U_r .

3.4 Angular Momentum and Torque

The angular momentum of body, dm , is defined as

$$L = \mathbf{r} \times \mathbf{U} dm \quad (3.38)$$

The angular momentum of the entire system is calculated by integration (summation) of all the particles in the system as

$$L_s = \int_m \mathbf{r} \times \mathbf{U} dm \quad (3.39)$$

The change with time of angular momentum is called torque, in analogous to the momentum change of time which is the force.

$$T_\tau = \frac{DL}{Dt} = \frac{D}{Dt} (\mathbf{r} \times \mathbf{U} dm) \quad (3.40)$$

where T_τ is the torque. The torque of entire system is

$$T_{\tau_s} = \int_m \frac{DL}{Dt} = \frac{D}{Dt} \int_m (\mathbf{r} \times \mathbf{U} dm) \quad (3.41)$$

It can be noticed (well, it can be proved utilizing vector mechanics) that

$$T_\tau = \frac{D}{Dt} (\mathbf{r} \times \mathbf{U}) = \frac{D}{Dt} (\mathbf{r} \times \frac{D\mathbf{r}}{Dt}) = \frac{D^2\mathbf{r}}{Dt^2} \quad (3.42)$$

To understand these equations a bit better, consider a particle moving in x-y plane. A force is acting on the particle in the same plane (x-y) plane. The velocity can be written as $U = u\hat{i} + v\hat{j}$ and the location from the origin can be written as $\mathbf{r} = x\hat{i} + y\hat{j}$. The force can be written, in the same fashion, as $\mathbf{F} = F_x\hat{i} + F_y\hat{j}$. Utilizing equation (3.38) provides

$$\mathbf{L} = \mathbf{r} \times \mathbf{U} = \begin{pmatrix} \hat{i} & \hat{j} & \hat{k} \\ x & y & 0 \\ u & v & 0 \end{pmatrix} = (xv - yu)\hat{k} \quad (3.43)$$

Utilizing equation (3.40) to calculate the torque as

$$T_\tau = \mathbf{r} \times \mathbf{F} = \begin{pmatrix} \hat{i} & \hat{j} & \hat{k} \\ x & y & 0 \\ F_x & F_y & 0 \end{pmatrix} = (xF_x - yF_y)\hat{k} \quad (3.44)$$

Since the torque is a derivative with respect to the time of the angular momentum it is also can be written as

$$xF_x - yF_y = \frac{D}{Dt} [(xv - yu) dm] \quad (3.45)$$

The torque is a vector and the various components can be represented as

$$T_{\tau_x} = \hat{i} \bullet \frac{D}{Dt} \int_m \mathbf{r} \times \mathbf{U} dm \quad (3.46)$$

In the same way the component in y and z can be obtained.

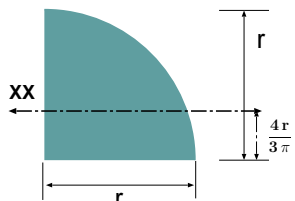
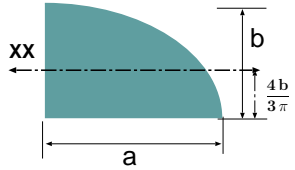
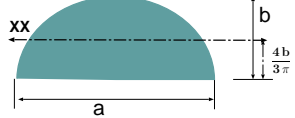
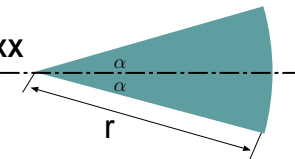
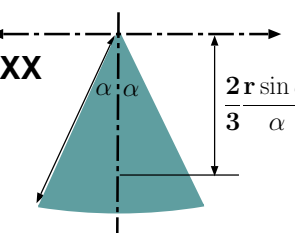
3.4.1 Tables of geometries

The following tables present several moment of inertias of commonly used geometries.

Table -3.1. Moments of Inertia for various plane surfaces about their center of gravity (full shapes)

Shape Name	Picture description	x_c, y_c	A	I_{xx}
Rectangle		$\frac{b}{2}; \frac{a}{2}$	ab	$\frac{ab^3}{12}$
Triangle		$\frac{a}{3}$	$\frac{ab}{3}$	$\frac{ab^3}{36}$
Circle		$\frac{b}{2}$	$\frac{\pi b^2}{4}$	$\frac{\pi b^4}{64}$
Ellipse		$\frac{b}{2} \frac{b}{2}$	$\frac{\pi ab}{4}$	$\frac{Ab^2}{64}$
$y = \alpha x^2$ Parabola		$\frac{3 \alpha b}{15 \alpha - 5}$	$\frac{6\alpha - 2}{3} \times \left(\frac{b}{\alpha}\right)^{\frac{3}{2}}$	$\frac{\sqrt{b} (20 b^3 - 14 b^2)}{35 \sqrt{\alpha}}$

Table -3.2. Moment of inertia for various plane surfaces about their center of gravity

Shape Name	Picture description	x_c, y_c	A	I_{xx}
Quadrant of Circle		$\frac{4r}{3\pi}$	$\frac{\pi r^2}{4}$	$r^4 \left(\frac{\pi}{16} - \frac{4}{9\pi} \right)$
Ellipsoidal Quadrant		$\frac{4b}{3\pi}$	$\frac{\pi ab}{4}$	$ab^3 \left(\frac{\pi}{16} - \frac{4}{9\pi} \right)$
Half of Elliptic		$\frac{4b}{3\pi}$	$\frac{\pi ab}{4}$	$ab^3 \left(\frac{\pi}{16} - \frac{4}{9\pi} \right)$
Circular Sector		0	$2\alpha r^2$	$\frac{r^4}{4} \left(\alpha - \frac{1}{2} \sin 2\alpha \right)$
Circular Sector		$\frac{2r \sin \alpha}{3}$	$2\alpha r^2$	$I_{x'x'} = \frac{r^4}{4} \left(\alpha + \frac{1}{2} \sin 2\alpha \right)$

CHAPTER 4

Fluids Statics

4.1 Introduction

The simplest situation that can occur in the study of fluid is when the fluid is at rest or quasi rest. This topic was introduced to most students in previous study of rigid body. However, here this topic will be more vigorously examined. Furthermore, the student will be exposed to stability analysis probably for the first time. Later, the methods discussed here will be expanded to more complicated dynamics situations.

4.2 The Hydrostatic Equation

A fluid element with dimensions of dx , dy , and dz is motionless in the accelerated system, with acceleration, a as shown in Figure 4.1. The system is in a body force field, $g_G(x, y, z)$. The combination of an acceleration and the body force results in effective body force which is

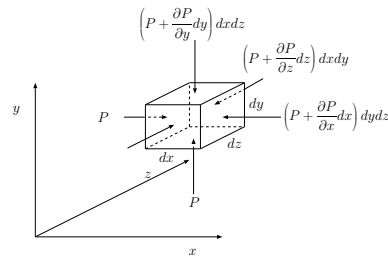


Fig. -4.1. Description of a fluid element in accelerated system under body forces.

$$\mathbf{g}_G - a = g_{\text{eff}} \quad (4.1)$$

Equation (4.1) can be reduced and simplified for the case of no acceleration, $a = 0$.

In these derivations, several assumptions must be made. The first assumption is that the change in the pressure is a continuous function. There is no requirement that the pressure has to be a monotonous function e.g. that pressure can increase and later decrease. The changes of the second derivative pressure are not significant

compared to the first derivative ($\partial P/\partial n \times dl \gg \partial^2 P/\partial n^2$). where n is the steepest direction of the pressure derivative and dl is the infinitesimal length. This mathematical statement simply requires that the pressure can deviate in such a way that the average on infinitesimal area can be found and expressed as only one direction. The net pressure force on the faces in the x direction results in

$$d\mathbf{F} = - \left(\frac{\partial P}{\partial x} \right) dydz \hat{i} \quad (4.2)$$

In the same fashion, the calculations of the three directions result in the total net pressure force as

$$\sum_{\text{surface}} F = - \left(\frac{\partial P}{\partial x} \hat{i} + \frac{\partial P}{\partial y} \hat{j} + \frac{\partial P}{\partial z} \hat{k} \right) \quad (4.3)$$

The term in the parentheses in equation (4.3) referred to in the literature as the pressure gradient. This mathematical operation has a geometrical interpretation. If the pressure, P , was a two-dimensional height (that is only a function of x and y) then the gradient is the steepest ascent of the height (to the valley). The second point is that the gradient is a vector (that is, it has a direction). Even though, the pressure is treated, now, as a scalar function (there no reference to the shear stress in part of the pressure) the gradient is a vector. For example, the dot product of the following is

$$\hat{i} \bullet \text{grad}P = \frac{\partial P}{\partial x} \quad (4.4)$$

In general, if the coordinates were to “rotate/transform” to a new system which has a different orientation the dot product results in

$$\hat{i}_n \bullet \text{grad}P = \frac{\partial P}{\partial n} \quad (4.5)$$

where i_n is the unit vector in the n direction and $\partial/\partial n$ is a derivative in that direction.

As before, the effective gravity force is utilized in case where the gravity is the only body force and in an accelerated system. The body (element) is in rest and therefore the net force is zero

$$\sum_{\text{total}} \mathbf{F} = \sum_{\text{surface}} \mathbf{F} + \sum_{\text{body}} \mathbf{F} \quad (4.6)$$

Hence, the utilizing the above derivations one can obtain

$$-\text{grad}P dx dy dz + \rho g_{\text{eff}} dx dy dz = 0 \quad (4.7)$$

or

$$\text{grad}P = \rho g_{\text{eff}} \quad (4.8)$$

Some refer to equation (4.8) as the Fluid Static Equation. This equation can be integrated and therefore solved. However, there are several physical implications to this equation which should be discussed and are presented here. First, a discussion on a simple condition and will continue in more challenging situations.

4.3 Pressure and Density in a Gravitational Field

In this section, a discussion on the pressure and the density in various conditions is presented.

4.3.1 Constant Density in Gravitational Field

The simplest case is when the density, ρ , pressure, P , and temperature, T (in a way no function of the location) are constant. Traditionally, the z coordinate is used as the (negative) direction of the gravity¹. The effective body force is

$$g_{\text{eff}} = -\mathbf{g} \hat{k} \quad (4.9)$$

Utilizing equation (4.9) and substituting it into equation (4.8) results into three simple partial differential equations. These equations are

$$\frac{\partial P}{\partial x} = \frac{\partial P}{\partial y} = 0 \quad (4.10)$$

and

$$\frac{\partial P}{\partial z} = -\rho \mathbf{g} \quad (4.11)$$

Equations (4.10) can be integrated to yield

$$P(x, y) = \text{constant} \quad (4.12)$$

and constant in equation (4.12) can be absorbed by the integration of equation (4.11) and therefore

$$P(x, y, z) = -\rho g z + \text{constant} \quad (4.13)$$

The integration constant is determined from the initial conditions or another point. For example, if at point z_0 the pressure is P_0 then the equation (4.13) becomes

$$P(z) - P_0 = -\rho g(z - z_0) \quad (4.14)$$

¹This situation were the tradition is appropriated, it will be used. There are fields where x or y are designed to the direction of the gravity and opposite direction. For this reason sometime there will be a deviation from the above statement.

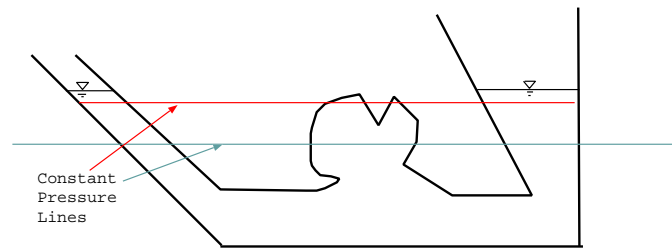


Fig. -4.2. Pressure lines a static fluid with a constant density.

It is evident from equation (4.13) that the pressure depends only on z and/or the constant pressure lines are in the plane of x and y . Figure 4.2 describes the constant pressure lines in the container under the gravity body force. The pressure lines are continuous even in area where there is a discontinuous fluid. The reason that a solid boundary doesn't break the continuity of the pressure lines is because there is always a path to some of the planes.

It is convenient to reverse the direction of z to get rid of the negative sign and to define h as the dependent of the fluid that is $h \equiv -(z - z_0)$ so equation (4.14) becomes

$$P(h) - P_0 = \rho gh \quad (4.15)$$

In the literature, the right hand side of the equation (4.15) is defined as piezometric pressure.

4.3.2 Pressure Measurement

4.3.2.1 Measuring the Atmospheric Pressure

One of the application of this concept is the idea of measuring the atmospheric pressure. Consider a situation described in Figure 4.3. The liquid is filling the tube and is brought into a steady state. The pressure above the liquid on the right side is the vapor pressure. Using liquid with a very low vapor pressure like mercury, will result in a device that can measure the pressure without additional information (the temperature).

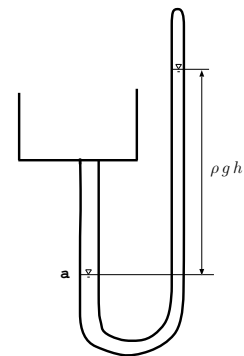


Fig. -4.3. A schematic to explain the measure of the atmospheric pressure.

Example 4.1:

Calculate the atmospheric pressure at 20°C . The high of the Mercury is 0.76 [m] and the gravity acceleration is 9.82 [m/sec] . Assume that the mercury vapor pressure is 0.000179264 [kPa] . The description of the height is given in Figure 4.3. The mercury density is $13545.85\text{ [kg/m}^3\text{]}$.

SOLUTION

The pressure is uniform or constant plane perpendicular to the gravity. Hence, knowing any point on this plane provides the pressure anywhere on the plane. The atmospheric pressure at point **a** is the same as the pressure on the right hand side of the tube. Equation (4.15) can be utilized and it can be noticed that pressure at point **a** is

$$P_a = \rho g h + P_{\text{vapor}} \quad (4.16)$$

The density of the mercury is given along with the gravity and therefore,

$$P_a = 13545.85 \times 9.82 \times 0.76 \sim 101095.39\text{ [Pa]} \sim 1.01\text{ [Bar]}$$

The vapor pressure is about 1×10^{-4} percent of the total results.

End Solution

The main reason the mercury is used because its of large density and the fact that it is in a liquid phase in most of the measurement range. The third reason is the low vapor (partial) pressure of the mercury. The partial pressure of mercury is in the range of the 0.000001793 [Bar] which is insignificant compared to the total measurement as can be observed from the above example.

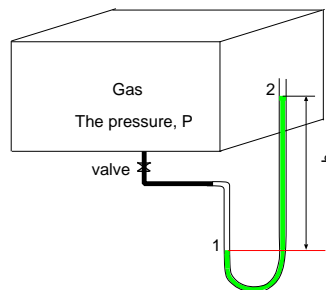


Fig. -4.4. Schematic of gas measurement utilizing the "U" tube.

Example 4.2:

A liquid² **a** in amount H_a and a liquid **b** in amount H_b in to an U tube. The ratio of the liquid densities is $\alpha = \rho_1/\rho_2$. The width of the U tube is L . Locate the liquids surfaces.

SOLUTION

The question is to find the equilibrium point where two liquids balance each other. If the width of the U tube is equal or larger than total length of the two liquids then the whole liquid will be in bottom part. For smaller width, L , the ratio between two sides will be as

$$\rho_1 h_1 = \rho_2 h_2 \rightarrow h_2 = \alpha h_1$$

²This example was requested by several students who found their instructor solution unsatisfactory.

The mass conservation results in

$$H_a + H_b = L + h_1 + h_2$$

Thus two equations and two unknowns provide the solution which is

$$h_1 = \frac{H_a + H_b - L}{1 + \alpha}$$

When $H_a > L$ and $\rho_a (H_a - L) \geq \rho_b$ (or the opposite) the liquid a will be on the two sides of the U tube. Thus, the balance is

$$h_1 \rho_b + h_2 \rho_a = h_3 \rho_a$$

where h_1 is the height of liquid b where h_2 is the height of “extra” liquid a and same side as liquid b and where h_3 is the height of liquid b on the other side. When in this case h_1 is equal to H_b . The additional equation is the mass conservation as

$$H_a = h_2 + L + h_3$$

The solution is

$$h_2 = \frac{(H_a - L) \rho_a - H_b \rho_b}{2 \rho_a}$$

End Solution

4.3.2.2 Pressure Measurement

The idea describes the atmospheric measurement can be extended to measure the pressure gas chambers. Consider a chamber filled with gas needed to be measured (see Figure 4.4). One technique is to attached “U” tube to the chamber and measure the pressure. This way, the gas is prevented from escaping and its pressure can be measured with a minimal interference to the gas (some gas enters to the tube).

The gas density is significantly lower than the liquid density and therefore can be neglected. The pressure at point “1” is

$$P_1 = P_{atmos} + \rho g h \quad (4.17)$$

Since the atmospheric pressure was measured previously (the technique was shown in the previous section) the pressure of the chamber can be measured.

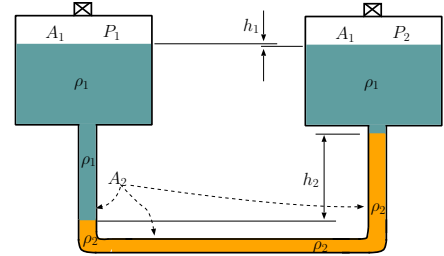


Fig. -4.5. Schematic of sensitive measurement device.

4.3.2.3 Magnified Pressure Measurement

For situations where the pressure difference is very small, engineers invented more sensitive measuring device. This device is build around the fact that the height is a function of the densities difference. In the previous technique, the density of one side was neglected (the gas side) compared to other side (liquid). This technique utilizes the opposite range. The two sides of the densities are very close to each other so the height became large. Figure 4.5 shows a typical and simple schematic of such an instrument. If the pressure differences between P_1 and P_2 is small this instrument can “magnified” height, h_1 and provide “better” accuracy reading. This device is based on the following mathematical explanation.

In steady state, the pressure balance (only differences) is

$$P_1 + g \rho_1 (h_1 + h_2) = P_2 + g h_2 \rho_2 \quad (4.18)$$

It can be noticed that the “missing height” is canceled between the two sides. It can be noticed that h_1 can be positive or negative or zero and it depends on the ratio that two containers filled with the light density liquid. Additionally, it can be observed that h_1 is relatively small because $A_1 \gg A_2$. The densities of the liquid are chosen so that they are close to each other but not equal. The densities of the liquids are chosen to be much heavier than the measured gas density. Thus, in writing equation (4.18) the gas density was neglected. The pressure difference can be expressed as

$$P_1 - P_2 = g [\rho_2 h_2 - \rho_1 (h_1 + h_2)] \quad (4.19)$$

If the light liquid volume in the two containers is known, it provides the relationship between h_1 and h_2 . For example, if the volumes in two containers is equal then

$$-h_1 A_1 = h_2 A_2 \longrightarrow h_1 = -\frac{h_2 A_2}{A_1} \quad (4.20)$$

Liquid volumes do not necessarily have to be equal. Additional parameter, the volume ratio, will be introduced when the volumes ratio isn't equal. The calculations as results of this additional parameter does not cause significant complications. Here, this ratio equals to one and it simplify the equation (4.20). But this ratio easily can be inserted into the derivations. With the equation for height (4.20) equation (4.18) becomes

$$P_1 - P_2 = g h_2 \left(\rho_2 - \rho_1 \left(1 - \frac{A_2}{A_1} \right) \right) \quad (4.21)$$

or the height is

$$h_2 = \frac{P_1 - P_2}{g \left[(\rho_2 - \rho_1) + \rho_1 \frac{A_2}{A_1} \right]} \quad (4.22)$$

For the small value of the area ratio, $A_2/A_1 \ll 1$, then equation (4.22) becomes

$$h_2 = \frac{P_1 - P_2}{g (\rho_2 - \rho_1)} \quad (4.23)$$

Some refer to the density difference shown in equation (4.23) as “magnification factor” since it replace the regular density, ρ_2 .

4.3.3 Varying Density in a Gravity Field

There are several cases that will be discussed here which are categorized as gases, liquids and other. In the gas phase, the equation of state is simply the ideal gas model or the ideal gas with the compressibility factor (sometime referred to as real gas). The equation of state for liquid can be approximated or replaced by utilizing the bulk modulus. These relationships will be used to find the functionality between pressure, density and location.

4.3.3.1 Gas Phase under Hydrostatic Pressure

Ideal Gas under Hydrostatic Pressure

The gas density vary gradually with the pressure. As first approximation, the ideal gas model can be employed to describe the density. Thus equation (4.11) becomes

$$\frac{\partial P}{\partial z} = -\frac{gP}{RT} \quad (4.24)$$

Separating the variables and changing the partial derivatives to full derivative (just a notation for this case) results in

$$\frac{dP}{P} = -\frac{g dz}{RT} \quad (4.25)$$

Equation (4.25) can be integrated from point "0" to any point to yield

$$\ln \frac{P}{P_0} = -\frac{g}{RT} (z - z_0) \quad (4.26)$$

It is convenient to rearrange equation (4.26) to the following

$$\frac{P}{P_0} = e^{-\left(\frac{g(z-z_0)}{RT}\right)} \quad (4.27)$$

Here the pressure ratio is related to the height exponentially. Equation (4.27) can be expanded to show the difference to standard assumption of constant pressure as

$$\frac{P}{P_0} = 1 - \overbrace{\frac{h \rho_0 g}{RT}} + \frac{(z - z_0)^2 g}{6 RT} + \dots \quad (4.28)$$

Or in a simplified form where the transformation of $h = (z - z_0)$ to be

$$\frac{P}{P_0} = 1 + \frac{\rho_0 g}{P_0} \left(h - \overbrace{\frac{h^2}{6} + \dots}^{\text{correction factor}} \right) \quad (4.29)$$

Equation (4.29) is useful in mathematical derivations but should be ignored for practical use³.

Real Gas under Hydrostatic Pressure

The mathematical derivations for ideal gas can be reused as a foundation for the real gas model ($P = Z\rho RT$). For a large range of P/P_c and T/T_c , the value of the compressibility factor, Z , can be assumed constant and therefore can be swallowed into equations (4.27) and (4.28). The compressibility is defined in equation (2.39). The modified equation is

$$\frac{P}{P_0} = e^{-\left(\frac{g(z-z_0)}{ZRT}\right)} \quad (4.30)$$

Or in a series form which is

$$\frac{P}{P_0} = 1 - \frac{(z-z_0)g}{ZRT} + \frac{(z-z_0)^2 g}{6ZRT} + \dots \quad (4.31)$$

Without going through the mathematics, the first approximation should be noticed that the compressibility factor, Z enter the equation as h/Z and not just h . Another point that is worth discussing is the relationship of Z to other gas properties. In general, the relationship is very complicated and in some ranges Z cannot be assumed constant. In these cases, a numerical integration must be carried out.

4.3.3.2 Liquid Phase Under Hydrostatic Pressure

The bulk modulus was defined in equation (1.28). The simplest approach is to assume that the bulk modulus is constant (or has some representative average). For these cases, there are two differential equations that needed to be solved. Fortunately, here, only one hydrostatic equation depends on density equation. So, the differential equation for density should be solved first. The governing differential density equation (see equation (1.28)) is

$$\rho = B_T \frac{\partial \rho}{\partial P} \quad (4.32)$$

The variables for equation (4.32) should be separated and then the integration can be carried out as

$$\int_{P_0}^P dP = \int_{\rho_0}^{\rho} B_T \frac{d\rho}{\rho} \quad (4.33)$$

The integration of equation (4.33) yields

$$P - P_0 = B_T \ln \frac{\rho}{\rho_0} \quad (4.34)$$

³These derivations are left for a mathematical mind person. These deviations have a limited practical purpose. However, they are presented here for students who need to answer questions on this issue.

Equation (4.34) can be represented in a more convenient form as

$$\rho = \rho_0 e^{\frac{P-P_0}{B_T}} \quad (4.35)$$

Equation (4.35) is the counterpart for the equation of state of ideal gas for the liquid phase. Utilizing equation (4.35) in equation (4.11) transformed into

$$\frac{\partial P}{\partial z} = -g\rho_0 e^{\frac{P-P_0}{B_T}} \quad (4.36)$$

Equation (4.36) can be integrated to yield

$$\frac{B_T}{g\rho_0} e^{\frac{P-P_0}{B_T}} = z + Constant \quad (4.37)$$

It can be noted that B_T has units of pressure and therefore the ratio in front of the exponent in equation (4.37) has units of length. The integration constant, with units of length, can be evaluated at any specific point. If at $z = 0$ the pressure is P_0 and the density is ρ_0 then the constant is

$$Constant = \frac{B_T}{g\rho_0} \quad (4.38)$$

This constant, $B_T/g\rho_0$, is a typical length of the problem. Additional discussion will be presented in the dimensionless issues chapter (currently under construction). The solution becomes

$$\frac{B_T}{g\rho_0} \left(e^{\frac{P-P_0}{B_T}} - 1 \right) = z \quad (4.39)$$

Or in a dimensionless form

$$\left(e^{\frac{P-P_0}{B_T}} - 1 \right) = \frac{z g \rho_0}{B_T} \quad (4.40)$$

The solution is presented in equation (4.39) and is plotted in Figure 4.6. The solution is a reverse function (that is not $P = f(z)$ but $z = f(P)$) it is a monotonous function which is easy to solve for any numerical value (that is only one z corresponds to any Pressure). Sometimes, the solution is presented as

$$\frac{P}{P_0} = \frac{B_T}{P_0} \ln \left(\frac{g\rho_0 z}{B_T} + 1 \right) + 1 \quad (4.41)$$

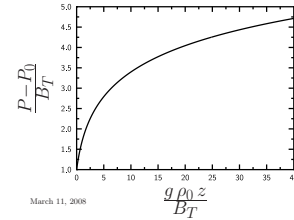


Fig. -4.6. Hydrostatic pressure when there is compressibility in the liquid phase.

An approximation of equation (4.40) is presented for historical reasons and in order to compare the constant density assumption. The exponent can be expanded as

$$\left(\underbrace{(P - P_0)}_{\text{piezometric pressure}} + \overbrace{\frac{B_T}{2} \left(\frac{P - P_0}{B_T} \right)^2 + \frac{B_T}{6} \left(\frac{P - P_0}{B_T} \right)^3 + \dots}_{\text{corrections}} \right) = z g \rho_0 \quad (4.42)$$

It can be noticed that equation (4.42) is reduced to the standard equation when the normalized pressure ratio, P/B_T is small ($\ll 1$). Additionally, it can be observed that the correction is on the left hand side and not as the "traditional" correction on the piezometric pressure side.

4.3.4 The Pressure Effects Because Temperature Variations

4.3.4.1 The Basic Analysis

There are situations when the main change of the density results from other effects. For example, when the temperature field is not uniform, the density is affected and thus the pressure is a location function (for example, in the atmosphere the temperature is assumed to be a linear with the height under certain conditions.). A bit more complicate case is when the gas is a function of the pressure and another parameter. Air can be a function of the temperature field and the pressure. For the atmosphere, it is commonly assumed that the temperature is a linear function of the height.

Here, a simple case is examined for which the temperature is a linear function of the height as

$$\frac{dT}{dh} = -C_x \quad (4.43)$$

where h here referred to height or distance. Hence, the temperature–distance function can be written as

$$T = \text{Constant} - C_x h \quad (4.44)$$

where the *Constant* is the integration constant which can be obtained by utilizing the initial condition. For $h = 0$, the temperature is T_0 and using it leads to

$$T = T_0 - C_x h \quad (4.45)$$

Combining equation (4.45) with (4.11) results in

$$\frac{\partial P}{\partial h} = -\frac{g P}{R(T_0 - C_x h)} \quad (4.46)$$

Separating the variables in equation (4.46) and changing the formal ∂ to the informal d to obtain

$$\frac{dP}{P} = -\frac{g dh}{R(T_0 - C_x h)} \quad (4.47)$$

Defining a new variable⁴ as $\xi = (T_0 - C_x h)$ for which $\xi_0 = T_0 - C_x h_0$ and $d/d\xi = -C_x d/dh$. Using these definitions results in

$$\frac{dP}{P} = \frac{g}{RC_x} \frac{d\xi}{\xi} \quad (4.48)$$

After integration of equation (4.47) and reusing (the reverse definitions) the variables transformed the result into

$$\ln \frac{P}{P_0} = \frac{g}{RC_x} \ln \frac{T_0 - C_x h}{T_0} \quad (4.49)$$

Or in a more convenient form as

$$\frac{P}{P_0} = \left(\frac{T_0 - C_x h}{T_0} \right)^{\left(\frac{g}{RC_x} \right)} \quad (4.50)$$

It can be noticed that equation (4.50) is a monotonous function which decreases with height because the term in the brackets is less than one. This situation is roughly representing the pressure in the atmosphere and results in a temperature decrease. It can be observed that C_x has a “double role” which can change the pressure ratio. Equation (4.50) can be approximated by two approaches/ideas. The first approximation for a small distance, h , and the second approximation for a small temperature gradient. It can be recalled that the following expansions are

$$\frac{P}{P_0} = \lim_{h \rightarrow 0} \left(1 - \frac{C_x}{T_0} h \right)^{\frac{g}{RC_x}} = 1 - \frac{\frac{g h \rho_0}{P_0}}{T_0 R} - \frac{\text{correction factor}}{2 T_0^2 R^2} h^2 - \dots \quad (4.51)$$

Equation (4.51) shows that the first two terms are the standard terms (negative sign is as expected i.e. negative direction). The correction factor occurs only at the third term which is important for larger heights. It is worth to point out that the above statement has a qualitative meaning when additional parameter is added. However, this kind of analysis will be presented in the dimensional analysis chapter⁵.

⁴A colleague asked this author to insert this explanation for his students. If you feel that it is too simple, please, just ignore it.

⁵These concepts are very essential in all the thermo–fluid science. I am grateful to my adviser E.R.G. Eckert who was the pioneer of the dimensional analysis in heat transfer and was kind to show me some of his ideas.

The second approximation for small C_x is

$$\frac{P}{P_0} = \lim_{C_x \rightarrow 0} \left(1 - \frac{C_x}{T_0} h\right)^{\frac{g}{R C_x}} = e^{-\frac{g h}{R T_0}} - \frac{g h^2 C_x}{2 T_0^2 R} e^{-\frac{g h}{R T_0}} - \dots \quad (4.52)$$

Equation (4.52) shows that the correction factor (lapse coefficient), C_x , influences at only large values of height. It has to be noted that these equations (4.51) and (4.52) are not properly represented without the characteristic height. It has to be inserted to make the physical significance clearer.

Equation (4.50) represents only the pressure ratio. For engineering purposes, it is sometimes important to obtain the density ratio. This relationship can be obtained from combining equations (4.50) and (4.45). The simplest assumption to combine these equations is by assuming the ideal gas model, equation (2.25), to yield

$$\frac{\rho}{\rho_0} = \frac{P T_0}{P_0 T} = \underbrace{\left(1 - \frac{C_x h}{T_0}\right)^{\left(\frac{g}{R C_x}\right)}}_{\frac{P}{P_0}} \underbrace{\left(1 + \frac{C_x h}{T}\right)}_{\frac{T_0}{T}} \quad (4.53)$$

***** Advance material can be skipped *****

4.3.4.2 The Stability Analysis

It is interesting to study whether this solution (4.50) is stable and if so under what conditions. Suppose that for some reason, a small slab of material moves from a layer at height, h , to layer at height $h + dh$ (see Figure 4.7)

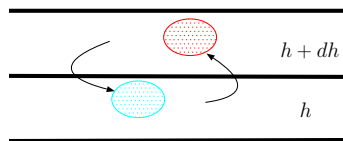


Fig. -4.7. Two adjoin layers for stability analysis.

What could happen? There are two main possibilities one: the slab could return to the original layer or two: stay at the new layer (or even move further, higher heights). The first case is referred to as the stable condition and the second case referred to as the unstable condition. The whole system falls apart and does not stay if the analysis predicts unstable conditions. A weak wind or other disturbances can make the unstable system to move to a new condition.

This question is determined by the net forces acting on the slab. Whether these forces are toward the original layer or not. The two forces that act on the slab are the gravity force and the surroundings pressure (buoyant forces). Clearly, the slab is in equilibrium with its surroundings before the movement (not necessarily stable). Under equilibrium, the body forces that acting on the slab are equal to zero. That is, the surroundings "pressure" forces (buoyancy forces) are equal to gravity forces. The buoyancy forces are proportional to the ratio of the density of the slab to surrounding layer density. Thus, the stability question is whether the slab density from layer h , $\rho'(h)$ undergoing a free expansion is higher or lower than the density of the layer $h + dh$. If

$\rho'(h) > \rho(h + dh)$ then the situation is stable. The term $\rho'(h)$ is slab from layer h that had undergone the free expansion.

The reason that the free expansion is chosen to explain the process that the slab undergoes when it moves from layer h to layer $h + dh$ is because it is the simplest. In reality, the free expansion is not far way from the actual process. The two processes that occurred here are thermal and the change of pressure (at the speed of sound). The thermal process is in the range of [cm/sec] while the speed of sound is about 300 [m/sec]. That is, the pressure process is about thousands times faster than the thermal process. The second issue that occurs during the "expansion" is the shock (in the reverse case $[h + dh] \rightarrow h$). However, this shock is insignificant (check book on Fundamentals of Compressible Flow Mechanics by this author on the French problem).

The slab density at layer $h + dh$ can be obtained using equation (4.53) as following

$$\frac{\rho(h + dh)}{\rho(h)} = \frac{P T_0}{P_0 T} = \left(1 - \frac{C_x dh}{T_0}\right)^{\left(\frac{g}{R C_x}\right)} \left(1 + \frac{C_x dh}{T}\right) \quad (4.54)$$

The pressure and temperature change when the slab moves from layer at h to layer $h + dh$. The process, under the above discussion and simplifications, can be assumed to be adiabatic (that is, no significant heat transfer occurs in the short period of time). The little slab undergoes isentropic expansion as following for which (see equation (2.25))

$$\frac{\rho'(h + dh)}{\rho(h)} = \left(\frac{P'(h + dh)}{P(h)}\right)^{1/k} \quad (4.55)$$

When the symbol $'$ denotes the slab that moves from layer h to layer $h + dh$. The pressure ratio is given by equation (4.50) but can be approximated by equation (4.51) and thus

$$\frac{\rho'(h + dh)}{\rho(h)} = \left(1 - \frac{g dh}{T(h) R}\right)^{1/k} \quad (4.56)$$

Again using the ideal gas model for equation (4.57) transformed into

$$\frac{\rho'(h + dh)}{\rho(h)} = \left(1 - \frac{\rho g dh}{P}\right)^{1/k} \quad (4.57)$$

Expanding equation (4.57) in Taylor series results in

$$\left(1 - \frac{\rho g dh}{P}\right)^{1/k} = 1 - \frac{g \rho dh}{P k} - \frac{(g^2 \rho^2 k - g^2 \rho^2) dh^2}{2 P^2 k^2} - \dots \quad (4.58)$$

The density at layer $h + dh$ can be obtained from (4.54) and then it is expanded in Taylor series as

$$\frac{\rho(h + dh)}{\rho(h)} = \left(1 - \frac{C_x dh}{T_0}\right)^{\left(\frac{g}{R C_x}\right)} \left(1 + \frac{C_x dh}{T}\right) \sim 1 - \left(\frac{g \rho}{P} - \frac{C_x}{T}\right) dh + \dots \quad (4.59)$$

The comparison of the right hand terms of equations (4.59) and (4.58) provides the conditions to determine the stability.

From a mathematical point of view, to keep the inequality for a small dh only the first term need to be compared as

$$\frac{g \rho}{P k} > \frac{g \rho}{P} - \frac{C_x}{T} \quad (4.60)$$

After rearrangement of the inequality (4.60) and using the ideal gas identity, it transformed to

$$\begin{aligned} \frac{C_x}{T} &> \frac{(k-1) g \rho}{k P} \\ C_x &< \frac{k-1}{k} \frac{g}{R} \end{aligned} \quad (4.61)$$

The analysis shows that the maximum amount depends on the gravity and gas properties. It should be noted that this value should be changed a bit since the k should be replaced by polytropic expansion n . When lapse rate C_x is equal to the right hand side of the inequality, it is said that situation is neutral. However, one has to bear in mind that this analysis only provides a range and isn't exact. Thus, around this value additional analysis is needed ⁶.

One of the common question this author has been asked is about the forces of continuation. What is the source of the force(s) that make this situation when unstable continue to be unstable? Supposed that the situation became unstable and the layers have been exchanged, would the situation become stable now? One has to remember that temperature gradient forces continuous heat transfer which the source temperature change after the movement to the new layer. Thus, the unstable situation is continuously unstable.

4.3.5 Gravity Variations Effects on Pressure and Density

Until now the study focus on the change of density and pressure of the fluid. Equation (4.11) has two terms on the right hand side, the density, ρ and the body force, g . The body force was assumed until now to be constant. This assumption must be deviated when the distance from the body source is significantly change. At first glance, the body force is independent of the fluid. The source of the gravity force in gas is another body, while the gravity force source in liquid can be the liquid itself.

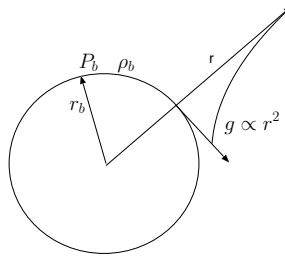


Fig. -4.8. The varying gravity effects on density and pressure.

⁶The same issue of the floating ice. See example for the floating ice in cup.

Thus, the discussion is separated into two different issues. The issues of magnetohydrodynamics are too advance for undergraduate student and therefore, will not be introduced here.

4.3.5.1 Ideal Gas in Varying Gravity

In physics, it was explained that the gravity is a function of the distance from the center of the plant/body. Assuming that the pressure is affected by this gravity/body force. The gravity force is reversely proportional to r^2 . The gravity force can be assumed that for infinity, $r \rightarrow \infty$ the pressure is about zero. Again, equation (4.11) can be used (semi one directional situation) when r is used as direction and thus

$$\frac{\partial P}{\partial r} = -\rho \frac{G}{r^2} \quad (4.62)$$

where here G denotes the general gravity constant. The regular method of separation is employed to obtain

$$\int_{P_b}^P \frac{dP}{P} = -\frac{G}{RT} \int_{r_b}^r \frac{dr}{r^2} \quad (4.63)$$

where the subscript b denotes the conditions at the body surface. The integration of equation (4.63) results in

$$\ln \frac{P}{P_b} = -\frac{G}{RT} \left(\frac{1}{r_b} - \frac{1}{r} \right) \quad (4.64)$$

Or in a simplified form as

$$\frac{\rho}{\rho_b} = \frac{P}{P_b} = e^{-\frac{G}{RT} \frac{r-r_b}{r r_b}} \quad (4.65)$$

Equation (4.65) demonstrates that the pressure is reduced with the distance. It can be noticed that for $r \rightarrow r_b$ the pressure is approaching $P \rightarrow P_b$. This equation confirms that the density in outer space is zero $\rho(\infty) = 0$. As before, equation (4.65) can be expanded in taylor series as

$$\frac{\rho}{\rho_b} = \frac{P}{P_b} = 1 - \overbrace{\frac{G(r-r_b)}{RT}}^{\text{standard}} - \overbrace{\frac{(2GRT + G^2 r_b)(r-r_b)^2}{2r_b(RT)^2}}^{\text{correction factor}} + \dots \quad (4.66)$$

Notice that G isn't our beloved and familiar g and also that $G r_b / RT$ is a dimensionless number (later in dimensionless chapter about it and its meaning).

4.3.5.2 Real Gas in Varying Gravity

The regular assumption of constant compressibility, Z , is employed. It has to remember when this assumption isn't accurate enough, numerical integration is a possible solution.

Thus, equation (4.63) is transformed into

$$\int_{P_b}^P \frac{dP}{P} = -\frac{G}{ZRT} \int_{r_b}^r \frac{dr}{r^2} \quad (4.67)$$

With the same process as before for ideal gas case, one can obtain

$$\frac{\rho}{\rho_b} = \frac{P}{P_b} = \mathbf{e}^{-\frac{G}{ZRT} \frac{r-r_b}{r r_b}} \quad (4.68)$$

Equation (4.65) demonstrates that the pressure is reduced with the distance. It can be observed that for $r \rightarrow r_b$ the pressure is approaching $P \rightarrow P_b$. This equation confirms that the density in outer space is zero $\rho(\infty) = 0$. As before Taylor series for equation (4.65) is

$$\frac{\rho}{\rho_b} = \frac{P}{P_b} = 1 - \overbrace{\frac{G(r-r_b)}{ZRT}}^{\text{standard}} - \overbrace{\frac{(2GZRT + G^2 r_b)(r-r_b)^2}{2r_b(ZRT)^2}}^{\text{correction factor}} + \dots \quad (4.69)$$

It can be noted that compressibility factor can act as increase or decrease of the ideal gas model depending on whether it is above one or below one.

4.3.5.3 Liquid in Under Varying Gravity

For comparison reason consider the deepest location in the ocean which is about 11,000 [m]. If the liquid "equation of state" (4.35) is used with the hydrostatic fluid equation results in

$$\frac{\partial P}{\partial r} = -\rho_0 \mathbf{e}^{\frac{P-P_0}{B_T}} \frac{G}{r^2} \quad (4.70)$$

which the solution of equation (4.70) is

$$\mathbf{e}^{\frac{P_0-P}{B_T}} = \text{Constant} - \frac{B_T g \rho_0}{r} \quad (4.71)$$

Since this author is not aware to which practical situation this solution should be applied, it is left for the reader to apply according to problem, if applicable.

4.3.6 Liquid Phase

While for most practical purposes, the Cartesian coordinates provides sufficient treatment to the problem, there are situations where the spherical coordinates must be considered and used.

Derivations of the fluid static in spherical coordinates are

$$\frac{1}{r^2} \frac{d}{dr} \left(\frac{r^2 dP}{\rho dr} \right) + 4\pi G\rho = 0 \quad (4.72)$$

Or in a vector form as

$$\nabla \cdot \left(\frac{1}{\rho} \nabla P \right) + 4\pi G\rho = 0 \quad (4.73)$$

4.4 Fluid in a Accelerated System

Up to this stage, body forces were considered as one-dimensional. The general, the linear acceleration have three components as opposed to the previous case of only one. However, the previous derivations can be easily extended. Equation (4.8) can be transformed into a different coordinate system where the main coordinate is in the direction of the effective gravity. Thus, the previous method can be used and there is no need to solve new three (or two) different equations. As before, the constant pressure plane is perpendicular to the direction of the effective gravity. Generally the acceleration is divided into two categories: linear and angular and they will be discussed in this order.

4.4.1 Fluid in a Linearly Accelerated System

For example, in a two dimensional system, for the effective gravity

$$g_{eff} = a\hat{i} + g\hat{k} \quad (4.74)$$

where the magnitude of the effective gravity is

$$|g_{eff}| = \sqrt{g^2 + a^2} \quad (4.75)$$

and the angle/direction can be obtained from

$$\tan\beta = \frac{a}{g} \quad (4.76)$$

Perhaps the best way to explain the linear acceleration is by examples. Consider the following example to illustrate the situation.

Example 4.3:

What is the angle of the liquid surface for a container in an accelerated system of $a = 5[m/sec]$?

SOLUTION

This question is one of the traditional question the fluid static and is straight forward.

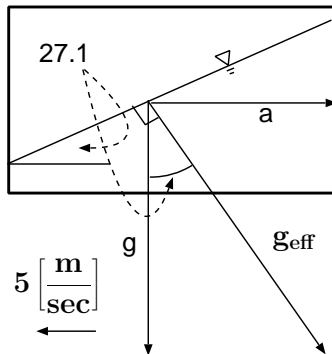


Fig. -4.9. The effective gravity is for accelerated cart.

This requires to find the angle of the effective body force. This angle/direction can be found using the following

$$\tan^{-1} \beta = \tan^{-1} \frac{a}{g} = \frac{5}{9.81} \sim 27.01^\circ$$

The magnitude of the effective acceleration is

$$|g_{eff}| = \sqrt{5^2 + 9.81^2} = 11.015[m/sec^2]$$

End Solution

Example 4.4:

Cart partially is filled with liquid and is sliding on an inclined plane as shown in Figure 4.10. Calculate the shape of the surface. If there is a resistance what will be the angle? What happen when the slope angle is straight (the cart is dropping straight down)?

SOLUTION

(a)

The angle can be found when the acceleration of the cart is found. If there is no resistance, the acceleration in the cart direction is determined from

$$a = g \sin \beta \tag{4.77}$$

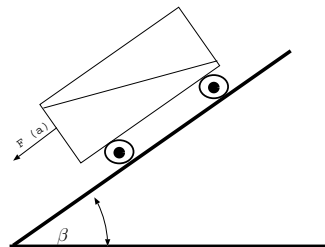


Fig. -4.10. A cart slide on inclined plane.

The effective body force is acting perpendicular to the slope. Thus, the liquid surface is parallel to the surface of the inclination surface.

(b)

In case of resistance force (either of friction due to the air or resistance in the wheels) reduces the acceleration of the cart. In that case the effective body moves closer to the gravity forces. The net body force depends on the mass of the liquid and the net acceleration is

$$a = \mathbf{g} - \frac{F_{net}}{m} \quad (4.78)$$

The angle of the surface, $\alpha < \beta$, is now

$$\tan \alpha = \frac{\mathbf{g} - \frac{F_{net}}{m}}{\mathbf{g} \cos \beta} \quad (4.79)$$

(c)

In the case when the angle of the inclination turned to be straight (direct falling) the effective body force is zero. The pressure is uniform in the tank and no pressure difference can be found. So, the pressure at any point in the liquid is the same and equal to the atmospheric pressure.

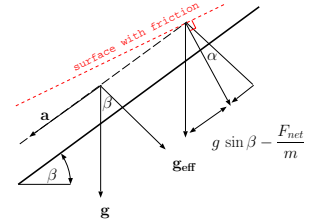


Fig. -4.11. Forces diagram of cart sliding on inclined plane.

4.4.2 Angular Acceleration Systems: Constant Density

For simplification reasons, the first case to deal with a rotation in a perpendicular to the gravity. That is the effective body force can be written as

$$\mathbf{g}_{eff} = -g \hat{k} + \omega^2 r \hat{r} \quad (4.80)$$

The lines of constant pressure are not straight lines but lines of parabolic shape. The angle of the line depends on the radius as

$$\frac{dz}{dr} = -\frac{g}{\omega^2 r} \quad (4.81)$$

Equation (4.81) can be integrated as

$$z - z_0 = \frac{\omega^2 r^2}{2g} \quad (4.82)$$

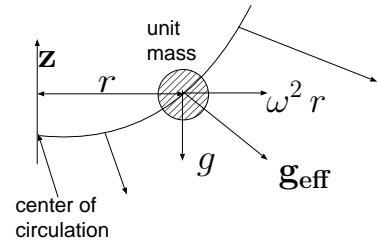


Fig. -4.12. Schematic to explain the angular angle.

Notice that the integration constant was substituted by z_0 . The constant pressure will be along

$$P - P_0 = \rho g \left[(z_0 - z) + \frac{\omega^2 r^2}{2g} \right] \tag{4.83}$$

To illustrate this point, example 4.5 is provided.

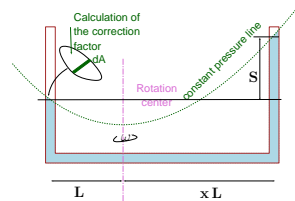


Fig. -4.13. Schematic angular angle to explain example 4.5.

Example 4.5:

A “U” tube with a length of $(1+x)L$ is rotating at angular velocity of ω . The center of rotation is a distance, L from the “left” hand side. Because the asymmetrical nature of the problem there is difference in the heights in the U tube arms of S as shown in Figure 4.13. Expresses the relationship between the different parameters of the problem.

SOLUTION

It first assumed the height is uniform at the tube (see for the open question on this assumption). The pressure at the interface at the two sides of the tube is same. Thus, equation (4.82) represent the pressure line. Taking the “left” wing of U tube

$$\underbrace{\text{change in } z \text{ direction}}_{z_l - z_0} = \underbrace{\text{change in } r \text{ direction}}_{\frac{\omega^2 L^2}{2g}}$$

The same can be said for the other side

$$z_r - z_0 = \frac{\omega^2 x^2 L^2}{2g}$$

Thus subtracting the two equations above from each other results in

$$z_r - z_l = \frac{L\omega^2 (1 - x^2)}{2g}$$

It can be noticed that this kind equipment can be used to find the gravity.

Example 4.6:

Assume the diameter of the U tube is R_t . What will be the correction factor if the curvature in the liquid in the tube is taken in to account. How would you suggest to define the height in the tube?

SOLUTION

In Figure 4.13 shown the infinitesimal area used in these calculations. The distance of the infinitesimal area from the rotation center is r . The height of the infinitesimal area is h . Notice that the curvature in the two sides are different from each other. The volume above the lower point is V which is only a function of the geometry.

End Solution

Example 4.7:

In the U tube in example 4.5 is rotating with upper part height of ℓ . At what rotating velocity liquid start to exit the U tube? If the rotation of U tube is exactly at the center, what happen the rotation approach very large value?

4.5 Fluid Forces on Surfaces

The forces that fluids (at static conditions) extracts on surfaces are very important for engineering purposes. This section deals with these calculations. These calculations are divided into two categories, straight surfaces and curved surfaces.

4.5.1 Fluid Forces on Straight Surfaces

A motivation is needed before going through the routine of derivations. Initially, a simple case will be examined. Later, how the calculations can be simplified will be shown.

Example 4.8:

Consider a rectangular shape gate as shown in Figure 4.14. Calculate the minimum forces, F_1 and F_2 to maintain the gate in position. Assuming that the atmospheric pressure can be ignored.

SOLUTION

The forces can be calculated by looking at the moment around point "O." The element of moment is $a d\xi$ for the width of the gate and is

$$dM = \underbrace{P a d\xi}_{dA} (\ell + \xi)$$

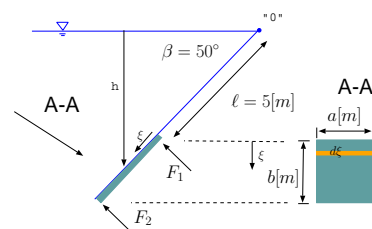


Fig. -4.14. Rectangular area under pressure.

The pressure, P can be expressed as a function ξ as the following

$$P = g \rho (\ell + \xi) \sin \beta$$

The liquid total moment on the gate is

$$M = \int_0^b g \rho (\ell + \xi) \sin \beta a d\xi (\ell + \xi)$$

The integral can be simplified as

$$M = g a \rho \sin \beta \int_0^b (\ell + \xi)^2 d\xi \quad (4.84)$$

The solution of the above integral is

$$M = g \rho a \sin \beta \left(\frac{3 b \ell^2 + 3 b^2 \ell + b^3}{3} \right)$$

This value provides the moment that F_1 and F_2 should extract. Additional equation is needed. It is the total force, which is

$$F_{total} = \int_0^b g \rho (\ell + \xi) \sin \beta a d\xi$$

The total force integration provides

$$F_{total} = g \rho a \sin \beta \int_0^b (\ell + \xi) d\xi = g \rho a \sin \beta \left(\frac{2 b \ell + b^2}{2} \right)$$

The forces on the gate have to provide

$$F_1 + F_2 = g \rho a \sin \beta \left(\frac{2 b \ell + b^2}{2} \right)$$

Additionally, the moment of forces around point "O" is

$$F_1 \ell + F_2 (\ell + b) = g \rho a \sin \beta \left(\frac{3 b \ell^2 + 3 b^2 \ell + b^3}{3} \right)$$

The solution of these equations is

$$F_1 = \frac{(3 \ell + b) a b g \rho \sin \beta}{6}$$

$$F_2 = \frac{(3 \ell + 2 b) a b g \rho \sin \beta}{6}$$

The above calculations are time consuming and engineers always try to make life simpler. Looking at the above calculations, it can be observed that there is a moment of area in equation (4.84) and also a center of area. These concepts have been introduced in Chapter 3. Several represented areas for which moment of inertia and center of area have been tabulated in Chapter 3. These tabulated values can be used to solve this kind of problems.

Symmetrical Shapes

Consider the two-dimensional symmetrical area that are under pressure as shown in Figure 4.15. The symmetry is around any axes parallel to axis x . The total force and moment that the liquid extracting on the area need to be calculated. First, the force is

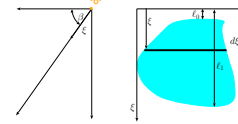


Fig. -4.15. Schematic of submerged area to explain the center forces and moments.

$$F = \int_A P dA = \int (P_{atmos} + \rho g h) dA = A P_{atmos} + \rho g \int_{\ell_0}^{\ell_1} \overbrace{(\xi + \ell_0) \sin \beta}^{h(\xi)} dA \tag{4.85}$$

In this case, the atmospheric pressure can include any additional liquid layer above layer “touching” area. The “atmospheric” pressure can be set to zero.

The boundaries of the integral of equation (4.85) refer to starting point and ending points not to the start area and end area. The integral in equation (4.85) can be further developed as

$$F_{total} = A P_{atmos} + \rho g \sin \beta \left(\ell_0 A + \overbrace{\int_{\ell_0}^{\ell_1} \xi dA}^{x_c A} \right) \tag{4.86}$$

In a final form as

$$F_{total} = A [P_{atmos} + \rho g \sin \beta (\ell_0 + x_c)] \tag{4.87}$$

The moment of the liquid on the area around point “O” is

$$M_y = \int_{\xi_0}^{\xi_1} P(\xi) \xi dA \tag{4.88}$$

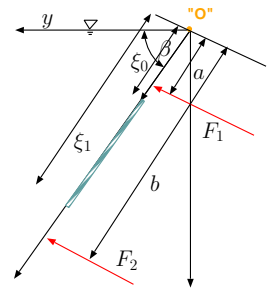


Fig. -4.16. The general forces acting on submerged area.

$$M_y = \int_{\xi_0}^{\xi_1} (P_{atmos} + g \rho \overbrace{h(\xi)}^{\xi \sin \beta}) \xi dA \quad (4.89)$$

Or separating the parts as

$$M_y = P_{atmos} \overbrace{\int_{\xi_0}^{\xi_1} \xi dA}^{x_c A} + g \rho \sin \beta \overbrace{\int_{\xi_0}^{\xi_1} \xi^2 dA}^{I_{x'x'}} \quad (4.90)$$

The moment of inertia, $I_{x'x'}$, is about the axis through point “O” into the page. Equation (4.90) can be written in more compact form as

$$M_y = P_{atmos} x_c A + g \rho \sin \beta I_{x'x'} \quad (4.91)$$

Example 4.8 can be generalized to solve any two forces needed to balance the area/gate. Consider the general symmetrical body shown in figure 4.16 which has two forces that balance the body. Equations (4.87) and (4.91) can be combined the moment and force acting on the general area. If the “atmospheric pressure” can be zero or include additional layer of liquid. The forces balance reads

$$F_1 + F_2 = A [P_{atmos} + \rho g \sin \beta (\ell_0 + x_c)] \quad (4.92)$$

and moments balance reads

$$F_1 a + F_2 b = P_{atmos} x_c A + g \rho \sin \beta I_{x'x'} \quad (4.93)$$

The solution of these equations is

$$F_1 = \frac{\left[\left(\rho \sin \beta - \frac{P_{atmos}}{g b} \right) x_c + \ell_0 \rho \sin \beta + \frac{P_{atmos}}{g} \right] b A - I_{x'x'} \rho \sin \beta}{g (b - a)} \quad (4.94)$$

and

$$F_2 = \frac{I_{x'x'} \rho \sin \beta - \left[\left(\rho \sin \beta - \frac{P_{atmos}}{g a} \right) x_c + \ell_0 \rho \sin \beta + \frac{P_{atmos}}{g} \right] a A}{g (b - a)} \quad (4.95)$$

In the solution, the forces can be negative or positive, and the distance a or b can be positive or negative. Additionally, the atmospheric pressure can contain either an additional liquid layer above the “touching” area or even atmospheric pressure simply can be set up to zero. In symmetrical area only two forces are required since the moment is one dimensional. However, in non-symmetrical area there two different moments and

therefor three forces are required. Thus, additional equation is required. This equation is for the additional moment around the x axis (see for explanation in Figure 4.17). The moment around the y axis is given by equation (4.91) and the total force is given by (4.87). The moment around the x axis (which was arbitrary chosen) should be

$$M_x = \int_A y P dA \quad (4.96)$$

Substituting the components for the pressure transforms equation (4.96) into

$$M_x = \int_A y (P_{atmos} + \rho g \xi \sin \beta) dA \quad (4.97)$$

The integral in equation (4.96) can be written as

$$M_x = P_{atmos} \overbrace{\int_A y dA}^{Ay_c} + \rho g \sin \beta \overbrace{\int_A \xi y dA}^{I_{x'y'}} \quad (4.98)$$

The compact form can be written as

$$M_x = P_{atmos} A y_c + \rho g \sin \beta I_{x'y'} \quad (4.99)$$

The product of inertia was presented in Chapter 3. These equations (4.87), (4.91) and (4.99) provide the base for solving any problem for straight area under pressure with uniform density. There are many combinations of problems (e.g. two forces and moment) but no general solution is provided. Example to illustrate the use of these equations is provided.

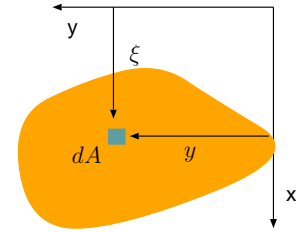


Fig. -4.17. The general forces acting on non symmetrical straight area.

Example 4.9:

Calculate the forces that are required to balance the triangular shape shown in the Figure 4.18.

SOLUTION

The three equations that needs to be solved are

$$F_1 + F_2 + F_3 = F_{total} \quad (4.100)$$

The moment around x axis is

$$F_1 b = M_y \quad (4.101)$$

The moment around y axis is

$$F_1 \ell_1 + F_2 (a + \ell_0) + F_3 \ell_0 = M_x \tag{4.102}$$

The right hand side of these equations are given before in equations (4.87), (4.91) and (4.99).

The moment of inertia of the triangle around x is made of two triangles (as shown in the Figure (4.18) for triangle 1 and 2). Triangle 1 can be calculated as the moment of inertia around its center which is $\ell_0 + 2 * (\ell_1 - \ell_0) / 3$. The height of triangle 1 is $(\ell_1 - \ell_0)$ and its width b and thus, moment of inertia about its center is $I_{xx} = b(\ell_1 - \ell_0)^3 / 36$. The moment of inertia for triangle 1 about y is

$$I_{xx1} = \frac{b(\ell_1 - \ell_0)^3}{36} + \overbrace{\frac{b(\ell_1 - \ell_0)}{3}}^{A_1} \overbrace{\left(\ell_0 + \frac{2(\ell_1 - \ell_0)}{3}\right)^2}^{\Delta x_1^2}$$

The height of the triangle 2 is $a - (\ell_1 - \ell_0)$ and its width b and thus, the moment of inertia about its center is

$$I_{xx2} = \frac{b[a - (\ell_1 - \ell_0)]^3}{36} + \overbrace{\frac{b[a - (\ell_1 - \ell_0)]}{3}}^{A_2} \overbrace{\left(\ell_1 + \frac{[a - (\ell_1 - \ell_0)]}{3}\right)^2}^{\Delta x_2^2}$$

and the total moment of inertia

$$I_{xx} = I_{xx1} + I_{xx2}$$

The product of inertia of the triangle can be obtain by integration. It can be noticed that upper line of the triangle is $y = \frac{(\ell_1 - \ell_0)x}{b} + \ell_0$. The lower line of the triangle is $y = \frac{(\ell_1 - \ell_0 - a)x}{b} + \ell_0 + a$.

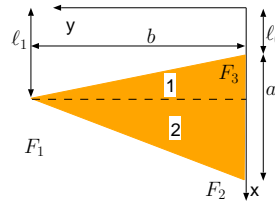


Fig. -4.18. The general forces acting on non symmetrical straight area.

$$I_{xy} = \int_0^b \left[\int_{\frac{(\ell_1 - \ell_0)x}{b} + \ell_0}^{\frac{(\ell_1 - \ell_0 - a)x}{b} + \ell_0 + a} xy dx \right] dy = \frac{2ab^2 \ell_1 + 2ab^2 \ell_0 + a^2 b^2}{24}$$

The solution of this set equations is

$$F_1 = \left[\frac{ab}{3} \right]^A \frac{(g(6\ell_1 + 3a) + 6g\ell_0) \rho \sin \beta + 8P_{atmos}}{24},$$

$$\frac{F_2}{\left[\frac{ab}{3} \right]} = - \frac{\left((3\ell_1 - 14a) - \ell_0 \left(\frac{12\ell_1}{a} - 27 \right) + \frac{12\ell_0^2}{a} \right) g \rho \sin \beta}{\left(\left(\frac{24\ell_1}{a} - 24 \right) + \frac{48\ell_0}{a} \right) P_{atmos}},$$

$$\frac{F_3}{\left[\frac{ab}{3}\right]} = \frac{\left(\left(a - \frac{15\ell_1}{a}\right) + \ell_0 \left(27 - \frac{12\ell_1}{a}\right) + \frac{12\ell_0^2}{a}\right) g \rho \sin \beta}{72} + \frac{\left(\left(\frac{24\ell_1}{a} + 24\right) + \frac{48\ell_0}{a}\right) P_{atmos}}{72}$$

End Solution

4.5.1.1 Pressure Center

In the literature, pressure centers are commonly defined. These definitions are mathematical in nature and has physical meaning of equivalent force that will act through this center. The definition is derived or obtained from equation (4.91) and equation (4.99). The pressure center is the distance that will create the moment with the hydrostatic force on point "O." Thus, the pressure center in the x direction is

$$x_p = \frac{1}{F} \int_A x P dA \quad (4.103)$$

In the same way, the pressure center in the y direction is defined as

$$y_p = \frac{1}{F} \int_A y P dA \quad (4.104)$$

To show relationship between the pressure center and the other properties, it can be found by setting the atmospheric pressure and ℓ_0 to zero as following

$$x_p = \frac{g \rho \sin \beta I_{x'x'}}{A \rho g \sin \beta x_c} \quad (4.105)$$

Expanding $I_{x'x'}$ according to equation (3.15) results in

$$x_p = \frac{I_{xx}}{x_c A} + x_c \quad (4.106)$$

and in the same fashion in y direction

$$y_p = \frac{I_{yy}}{y_c A} + y_c \quad (4.107)$$

It has to emphasis that these definitions are useful only for case where the atmospheric pressure can be neglected or canceled and where ℓ_0 is zero. Thus, these limitations diminish the usefulness of pressure center definitions. In fact, the reader can find that direct calculations can sometimes simplify the problem.

4.5.1.2 Multiply Layers

In the previous sections, the density was assumed to be constant. For non constant density the derivations aren't "clean" but are similar. Consider straight/flat body that

is under liquid with a varying density⁷. If density can be represented by average density, the force that is acting on the body is

$$F_{total} = \int_A g \rho h dA \sim \bar{\rho} \int_A g h dA \quad (4.108)$$

In cases where average density cannot be represented reasonably⁸, the integral has to be carried out. In cases where density is non-continuous, but constant in segments, the following can be said

$$F_{total} = \int_A g \rho h dA = \int_{A_1} g \rho_1 h dA + \int_{A_2} g \rho_2 h dA + \dots + \int_{A_n} g \rho_n h dA \quad (4.109)$$

As before for single density, the following can be written

$$F_{total} = g \sin \beta \left[\rho_1 \int_{A_1}^{\overbrace{x_{c1} A_1}} \xi dA + \rho_2 \int_{A_2}^{\overbrace{x_{c2} A_2}} \xi dA + \dots + \rho_n \int_{A_n}^{\overbrace{x_{cn} A_n}} \xi dA \right] \quad (4.110)$$

Or in a compact form and in addition considering the “atmospheric” pressure can be written as

$$F_{total} = P_{atmos} A_{total} + g \sin \beta \sum_{i=1}^n \rho_i x_{c_i} A_i \quad (4.111)$$

where the density, ρ_i is the density of the layer i and A_i and x_{c_i} are geometrical properties of the area which is in contact with that layer. The atmospheric pressure can be entered into the calculation in the same way as before. Moreover, the atmospheric pressure can include all the layer(s) that do(es) not with the “contact” area.

The moment around axis y , M_y under the same considerations as before is

$$M_y = \int_A g \rho \xi^2 \sin \beta dA \quad (4.112)$$

After similar separation of the total integral, one can find that

$$M_y = g \sin \beta \sum_{i=1}^n \rho_i I_{x'x'_i} \quad (4.113)$$

⁷This statement also means that density is a monotonous function. Why? Because of the buoyancy issue. It also means that the density can be a non-continuous function.

⁸A qualitative discussion on what is reasonable is not presented here. However, if the variation of the density is within 10% and/or the accuracy of the calculation is minimal, the reasonable average can be used.

If the atmospheric pressure also enters into the calculations one can find that

$$M_y = P_{atmos} x_c A_{total} + g \sin \beta \sum_{i=1}^n \rho_i I_{x'x'_i} \quad (4.114)$$

In the same fashion one can obtain the moment for x axis as

$$M_x = P_{atmos} y_c A_{total} + g \sin \beta \sum_{i=1}^n \rho_i I_{x'y'_i} \quad (4.115)$$

To illustrate how to work with these equations the following example is provided.

Example 4.10:

Consider the hypothetical Figure 4.19. The last layer is made of water with density of $1000[kg/m^3]$. The densities are $\rho_1 = 500[kg/m^3]$, $\rho_2 = 800[kg/m^3]$, $\rho_3 = 850[kg/m^3]$, and $\rho_4 = 1000[kg/m^3]$. Calculate the forces at points a_1 and b_1 . Assume that the layers are stables without any movement between the liquids. Also neglect all mass transfer phenomena that may occur. The heights are: $h_1 = 1[m]$, $h_2 = 2[m]$, $h_3 = 3[m]$, and $h_4 = 4[m]$. The forces distances are $a_1 = 1.5[m]$, $a_2 = 1.75[m]$, and $b_1 = 4.5[m]$. The angle of inclination is $\beta = 45^\circ$.

SOLUTION

Since there are only two unknowns, only two equations are needed, which are (4.114) and (4.111). The solution method of this example is applied for cases with less layers (for example by setting the specific height difference to be zero). Equation (4.114) can be used by modifying it, as can be noticed that instead of using the regular atmospheric pressure the new "atmospheric" pressure can be used as

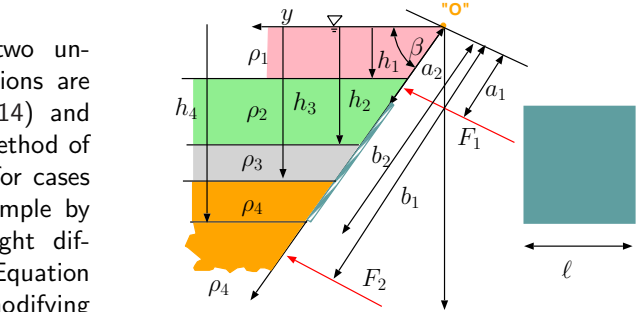


Fig. -4.19. The effects of multi layers density on static forces.

$$P_{atmos}' = P_{atmos} + \rho_1 g h_1$$

The distance for the center for each area is at the middle of each of the "small"

rectangular. The geometries of each areas are

$$\begin{aligned} x_{c1} &= \frac{a_2 + \frac{h_2}{\sin \beta}}{2} & A_1 &= \ell \left(\frac{h_2}{\sin \beta} - a_2 \right) & I_{x'x'_1} &= \frac{\ell \left(\frac{h_2}{\sin \beta} - a_2 \right)^3}{36} + (x_{c1})^2 A_1 \\ x_{c2} &= \frac{h_2 + h_3}{2 \sin \beta} & A_2 &= \frac{\ell}{\sin \beta} (h_3 - h_2) & I_{x'x'_2} &= \frac{\ell (h_3 - h_2)^3}{36 \sin \beta} + (x_{c2})^2 A_2 \\ x_{c3} &= \frac{h_3 + h_4}{2 \sin \beta} & A_3 &= \frac{\ell}{\sin \beta} (h_4 - h_3) & I_{x'x'_3} &= \frac{\ell (h_4 - h_3)^3}{36 \sin \beta} + (x_{c3})^2 A_3 \end{aligned}$$

After inserting the values, the following equations are obtained

Thus, the first equation is

$$F_1 + F_2 = P_{atmos} \overbrace{\ell (b_2 - a_2)}^{A_{total}} + g \sin \beta \sum_{i=1}^3 \rho_{i+1} x_{c_i} A_i$$

The second equation is (4.114) to be written for the moment around the point "O" as

$$F_1 a_1 + F_2 b_1 = P_{atmos} \overbrace{\frac{(b_2 + a_2)}{2} \ell (b_2 - a_2)}^{x_c A_{total}} + g \sin \beta \sum_{i=1}^3 \rho_{i+1} I_{x'x'_i}$$

The solution for the above equation is

$$\begin{aligned} F_1 &= \frac{2 b_1 g \sin \beta \sum_{i=1}^3 \rho_{i+1} x_{c_i} A_i - 2 g \sin \beta \sum_{i=1}^3 \rho_{i+1} I_{x'x'_i}}{2 b_1 - 2 a_1} - \frac{(b_2^2 - 2 b_1 b_2 + 2 a_2 b_1 - a_2^2) \ell P_{atmos}}{2 b_1 - 2 a_1} \\ F_2 &= \frac{2 g \sin \beta \sum_{i=1}^3 \rho_{i+1} I_{x'x'_i} - 2 a_1 g \sin \beta \sum_{i=1}^3 \rho_{i+1} x_{c_i} A_i}{2 b_1 - 2 a_1} + \frac{(b_2^2 + 2 a_1 b_2 + a_2^2 - 2 a_1 a_2) \ell P_{atmos}}{2 b_1 - 2 a_1} \end{aligned}$$

The solution isn't provided in the complete long form since it makes things messy. It is simpler to compute the terms separately. A mini source code for the calculations is provided in the the text source. The intermediate results in SI units ([m], [m²], [m⁴]) are:

$$\begin{aligned} x_{c1} &= 2.2892 & x_{c2} &= 3.5355 & x_{c3} &= 4.9497 \\ A_1 &= 2.696 & A_2 &= 3.535 & A_3 &= 3.535 \\ I_{x'x'_1} &= 14.215 & I_{x'x'_2} &= 44.292 & I_{x'x'_3} &= 86.718 \end{aligned}$$

The final answer is

$$F_1 = 304809.79[N]$$

and

$$F_2 = 958923.92[N]$$

4.5.2 Force on Curved Surfaces

The pressure is acting on surfaces perpendicular to the direction of the surface (no shear forces assumption). At this stage, the pressure is treated as a scalar function. The element force is

$$d\mathbf{F} = -P \hat{n} d\mathbf{A} \tag{4.116}$$

Here, the conventional notation is used which is to denote the area, dA , outward as positive. The total force on the area will be the integral of the unit force

$$\mathbf{F} = - \int_A P \hat{n} d\mathbf{A} \tag{4.117}$$

The result of the integral is a vector. So, if the y component of the force is needed, only a dot product is needed as

$$dF_y = d\mathbf{F} \cdot \hat{j} \tag{4.118}$$

From this analysis (equation (4.118)) it can be observed that the force in the direction of y , for example, is simply the integral of the area perpendicular to y as

$$F_y = \int_A P dA_y \tag{4.119}$$

The same can be said for the x direction.

The force in the z direction is

$$F_z = \int_A h g \rho dA_z \tag{4.120}$$

The force which acting on the z direction is the weight of the liquid above the projected area plus the atmospheric pressure. This force component can be combined with the other components in the other directions to be

$$F_{total} = \sqrt{F_z^2 + F_x^2 + F_y^2} \tag{4.121}$$

And the angle in “ xz ” plane is

$$\tan \theta_{xz} = \frac{F_z}{F_x} \tag{4.122}$$

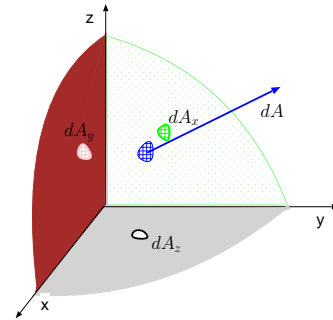


Fig. -4.20. The forces on curved area.

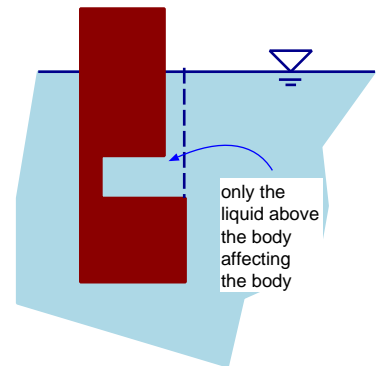


Fig. -4.21. Schematic of Net Force on floating body.

and the angle in the other plane, “ yz ” is

$$\tan \theta_{zy} = \frac{F_z}{F_y} \quad (4.123)$$

The moment due to the curved surface require integration to obtain the value. There are no readily made expressions for these 3–dimensional geometries. However, for some geometries there are readily calculated center of mass and when combined with two other components provide the moment (force with direction line).

Cut–Out Shapes Effects

There are bodies with a shape that the vertical direction (z direction) is “cut–out” aren’t continuous. Equation (4.120) implicitly means that the net force on the body in z direction is only the actual liquid above it. For example, Figure 4.21 shows a floating body with cut–out slot into it. The atmospheric pressure acts on the area with continuous lines. Inside the slot, the atmospheric pressure with its piezometric pressure is canceled by the upper part of the slot. Thus, only the net force is the actual liquid in the slot which is acting on the body. Additional point that is worth mentioning is that the depth where the cut–out occur is insignificant (neglecting the change in the density).

Example 4.11:

Calculate the force and the moment around point “ O ” that is acting on the dam (see Figure (4.22)). The dam is made of an arc with the angle of $\theta_0 = 45^\circ$ and radius of $r = 2[m]$. You can assume that the liquid density is constant and equal to $1000 [kg/m^3]$. The gravity is $9.8[m/sec^2]$ and width of the dam is $b = 4[m]$. Compare the different methods of computations, direct and indirect.

SOLUTION

The force in the x direction is

$$F_x = \int_A P \overbrace{r \cos \theta}^{dA_x} d\theta \quad (4.124)$$

Note that the direction of the area is taken into account (sign). The differential area that will be used is, $b r d\theta$ where b is the width of the dam (into the page). The pressure is only a function of θ and it is

$$P = P_{atmos} + \rho g r \sin \theta$$

The force that is acting on the x direction of the dam is $A_x \times P$. When the area A_x is $b r d\theta \cos \theta$. The atmospheric pressure does cancel itself (at least if the atmospheric pressure on both sides of the dam is the same.). The net force will be

$$F_x = \int_0^{\theta_0} \overbrace{\rho g r \sin \theta}^P \overbrace{b r \cos \theta}^{dA_x} d\theta$$

The integration results in

$$F_x = \frac{\rho g b r^2}{2} (1 - \cos^2(\theta_0))$$

Another way to do this calculation is by calculating the pressure at mid point and then multiply it by the projected area, A_x (see Figure 4.23) as

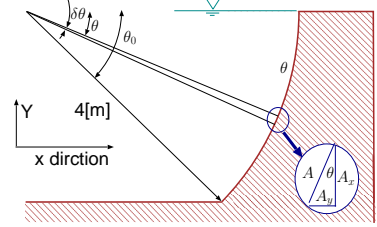


Fig. -4.22. Dam is a part of a circular shape.

$$F_x = \rho g \underbrace{b r \sin \theta_0}_{A_x} \underbrace{\frac{r \sin \theta_0}{2}}_{x_c} = \frac{\rho g b r}{2} \sin^2 \theta$$

Notice that $dA_x(\cos \theta)$ and $A_x(\sin \theta)$ are different, why?

The values to evaluate the last equation are provided in the question and simplify sub-size into it as

$$F_x = \frac{1000 \times 9.8 \times 4 \times 2}{2} \sin(45^\circ) = 19600.0[N]$$

Since the last two equations are identical (use the sinuous theorem to prove it $\sin^2 \theta + \cos^2 = 1$), clearly the discussion earlier was right (not a good proof LOL⁹). The force in the y direction is the area times width.

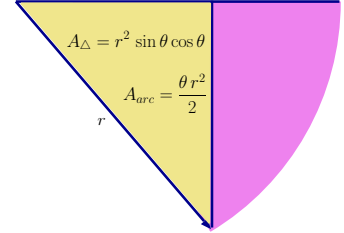


Fig. -4.23. Area above the dam arc subtract triangle.

$$F_y = - \left(\frac{\theta_0 r^2}{2} - \frac{r^2 \sin \theta_0 \cos \theta_0}{2} \right) b g \rho \sim 22375.216[N]$$

The center area (purple area in Figure 4.23) should be calculated as

$$y_c = \frac{y_c A_{arc} - y_c A_{triangle}}{A}$$

The center area above the dam requires to know the center area of the arc and triangle shapes. Some mathematics are required because the shift in the arc orientation. The arc center (see Figure 4.24) is at

$$y_{c_{arc}} = \frac{4 r \sin^2 \left(\frac{\theta}{2} \right)}{3 \theta}$$

⁹Well, just a demonstration!

All the other geometrical values are obtained from Tables 3.1 and 3.2. and substituting the proper values results in

$$y_{c_r} = \frac{\frac{A_{arc}}{\theta r^2} \overbrace{4 r \sin\left(\frac{\theta}{2}\right) \cos\left(\frac{\theta}{2}\right)}^{y_c} - \frac{\overbrace{2 r \cos \theta}^{y_c} \overbrace{\sin \theta r^2}^{A_{triangle}}}{\frac{\theta r^2}{2} - \frac{r^2 \sin \theta \cos \theta}{2}}}{\underbrace{\frac{2}{A_{arc}}}_{\frac{2}{A_{arc}}} - \underbrace{\frac{2}{A_{triangle}}}_{\frac{2}{A_{triangle}}}}$$

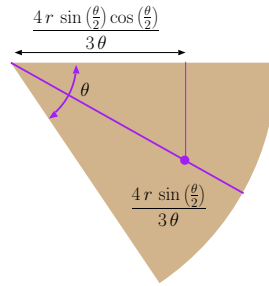


Fig. -4.24. Area above the dam arc calculation for the center.

This value is the reverse value and it is

$$y_{c_r} = 1.65174[m]$$

The result of the arc center from point "O" (above calculation area) is

$$y_c = r - y_{c_r} = 2 - 1.65174 \sim 0.348[m]$$

The moment is

$$M_v = y_c F_y \sim 0.348 \times 22375.2 \sim 7792.31759[N \times m]$$

The center pressure for x area is

$$x_p = x_c + \frac{I_{xx}}{x_c A} = \frac{r \cos \theta_0}{2} + \frac{\frac{\overbrace{b^3 (r \cos \theta_0)^3}^{I_{xx}}}{36}}{\frac{r \cos \theta_0}{2} b (r \cos \theta_0)} = \frac{5 r \cos \theta_0}{9}$$

The moment due to hydrostatic pressure is

$$M_h = x_p F_x = \frac{5 r \cos \theta_0}{9} F_x \sim 15399.21[N \times m]$$

The total moment is the combination of the two and it is

$$M_{total} = 23191.5[N \times m]$$

For direct integration of the moment is done as following

$$dF = P dA = \int_0^{\theta_0} \rho g \sin \theta b r d\theta$$

and element moment is

$$dM = dF \times \ell = dF \overbrace{2r \sin\left(\frac{\theta}{2}\right)}^{\ell} \cos\left(\frac{\theta}{2}\right)$$

and the total moment is

$$M = \int_0^{\theta_0} dM$$

or

$$M = \int_0^{\theta_0} \rho g \sin\theta b r 2r \sin\left(\frac{\theta}{2}\right) \cos\left(\frac{\theta}{2}\right) d\theta$$

The solution of the last equation is

$$M = \frac{g r \rho (2\theta_0 - \sin(2\theta_0))}{4}$$

The vertical force can be obtained by

$$F_v = \int_0^{\theta_0} P dA_v$$

or

$$F_v = \int_0^{\theta_0} \overbrace{\rho g r \sin\theta}^P \overbrace{r d\theta \cos\theta}^{dA_v}$$

$$F_v = \frac{g r^2 \rho}{2} \left(1 - \cos(2\theta_0)\right)$$

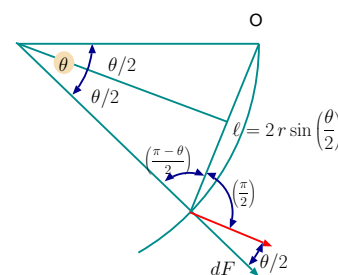


Fig. -4.25. Moment on arc element around

Here, the traditional approach was presented first, and the direct approach was presented second. It is much simpler now to use the second method. In fact, there are many programs or hand held devices that can carry numerical integration by inserting the function and the boundaries.

To demonstrate this point further, consider a more general case of a polynomial function. The reason that a polynomial function was chosen is that almost all the continuous functions can be represented by a Taylor series, and thus, this example provides for practical purposes of the general solution for curved surfaces.

Example 4.12:

For the liquid shown in Figure 4.26, calculate the moment around point "O" and the force created by the liquid per unit depth. The function of the dam shape is $y = \sum_{i=1}^n a_i x^i$ and it is a monotonous function (this restriction can be relaxed somewhat). Also calculate the horizontal and vertical forces.

SOLUTION

4.5. FLUID FORCES ON SURFACES

The calculations are done per unit depth (into the page) and do not require the actual depth of the dam.

The element force (see Figure 4.26) in this case is

$$dF = \overbrace{(b-y)}^P g \rho \overbrace{\sqrt{dx^2 + dy^2}}^{dA}$$

The size of the differential area is the square root of the dx^2 and dy^2 (see Figure 4.26). It can be noticed that the differential area that is used here should be multiplied by the depth. From mathematics, it can be shown that

$$\sqrt{dx^2 + dy^2} = dx \sqrt{1 + \left(\frac{dy}{dx}\right)^2}$$

The right side can be evaluated for any given function. For example, in this case describing the dam function is

$$\sqrt{1 + \left(\frac{dy}{dx}\right)^2} = \sqrt{1 + \left(\sum_{i=1}^n i a_i x^{i-1}\right)^2}$$

The value of x_b is where $y = b$ and can be obtained by finding the first and positive root of the equation of

$$0 = \sum_{i=1}^n a_i x^i - b$$

To evaluate the moment, expression of the distance and angle to point "O" are needed (see Figure 4.27). The distance between the point on the dam at x to the point "O" is

$$\ell(x) = \sqrt{(b-y)^2 + (x_b-x)^2}$$

The angle between the force and the distance to point "O" is

$$\theta(x) = \tan^{-1}\left(\frac{dy}{dx}\right) - \tan^{-1}\left(\frac{b-y}{x_b-x}\right)$$

The element moment in this case is

$$dM = \ell(x) \overbrace{(b-y) g \rho \sqrt{1 + \left(\frac{dy}{dx}\right)^2}}^{dF} \cos \theta(x) dx$$

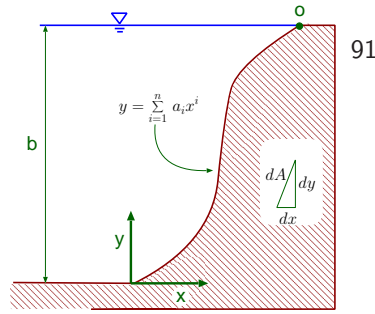


Fig. -4.26. Polynomial shape dam description for the moment around point "O" and force calculations.

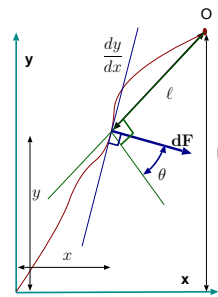


Fig. -4.27. The difference between the slope and the direction angle.

To make this example less abstract, consider the specific case of $y = 2x^6$. In this case, only one term is provided and x_b can be calculated as following

$$x_b = \sqrt[6]{\frac{b}{2}}$$

Notice that $\sqrt[6]{\frac{b}{2}}$ is measured in meters. The number “2” is a dimensional number with units of $[1/m^5]$. The derivative at x is

$$\frac{dy}{dx} = 12x^5$$

and the derivative is dimensionless (a dimensionless number). The distance is

$$\ell = \sqrt{(b - 2x^6)^2 + \left(\sqrt[6]{\frac{b}{2}} - x\right)^2}$$

The angle can be expressed as

$$\theta = \tan^{-1}(12x^5) - \tan^{-1}\left(\frac{b - 2x^6}{\sqrt[6]{\frac{b}{2}} - x}\right)$$

The total moment is

$$M = \int_0^{\sqrt[6]{b}} \ell(x) \cos \theta(x) (b - 2x^6) g \rho \sqrt{1 + 12x^5} dx$$

This integral doesn't have analytical solution. However, for a given value b this integral can be evaluate. The horizontal force is

$$F_h = b \rho g \frac{b}{2} = \frac{\rho g b^2}{2}$$

The vertical force per unit depth is the volume above the dam as

$$F_v = \int_0^{\sqrt[6]{b}} (b - 2x^6) \rho g dx = \rho g \frac{5b^{\frac{7}{6}}}{7}$$

In going over these calculations, the calculations of the center of the area were not carried out. This omission saves considerable time. In fact, trying to find the center of the area will double the work. This author find this method to be simpler for complicated geometries while the indirect method has advantage for very simple geometries.

End Solution

4.6 Buoyancy and Stability

4.6. BUOYANCY AND STABILITY

One of the oldest known scientific research on fluid mechanics relates to buoyancy due to question of money was carried by Archimedes. Archimedes principle is related to question of density and volume. While Archimedes did not know much about integrals, he was able to capture the essence. Here, because this material is presented in a different era, more advance mathematics will be used. While the question of the stability was not scientifically examined in the past, the floating vessels structure (more than 150 years ago) show some understanding¹⁰.

The total forces the liquid exerts on a body are considered as a buoyancy issue. To understand this issue, consider a cubical and a cylindrical body that is immersed in liquid and center in a depth of, h_0 as shown in Figure 4.28. The force to hold the cylinder at the place must be made of integration of the pressure around the surface of the square and cylinder bodies. The forces on square geometry body are made only of vertical forces because the two sides cancel each other. However, on the vertical direction, the pressure on the two surfaces are different. On the upper surface the pressure is $\rho g(h_0 - a/2)$. On the lower surface the pressure is $\rho g(h_0 + a/2)$. The force due to the liquid pressure per unit depth (into the page) is

$$F = \rho g ((h_0 - a/2) - (h_0 + a/2)) \ell b = -\rho g a b \ell = -\rho g V \quad (4.125)$$

In this case the ℓ represents a depth (into the page). Rearranging equation (4.125) to be

$$\frac{F}{V} = \rho g \quad (4.126)$$

The force on the immersed body is equal to the weight of the displaced liquid. This analysis can be generalized by noticing two things. All the horizontal forces are canceled. Any body that has a projected area that has two sides, those will cancel each other. Another way to look at this point is by approximation. For any two rectangle bodies, the horizontal forces are canceling each other. Thus even these bodies are in contact with each other, the imaginary pressure make it so that they cancel each other.

On the other hand, any shape is made of many small rectangles. The force on every rectangular shape is made of its weight of the volume. Thus, the total force is made of the sum of all the small rectangles which is the weight of the sum of all volume.

In illustration of this concept, consider the cylindrical shape in Figure 4.28. The force per area (see Figure 4.29) is

$$dF = \overbrace{\rho g (h_0 - r \sin \theta)}^P \overbrace{\sin \theta r d\theta}^{dA_{vertical}} \quad (4.127)$$

¹⁰This topic was the author's class name in high school. It was taught by people like these, 150 years ago and more, ship builders who know how to calculate GM but weren't aware of scientific principles behind it. If the reader wonders why such a class is taught in a high school, perhaps the name can explain it: Sea Officers High School.

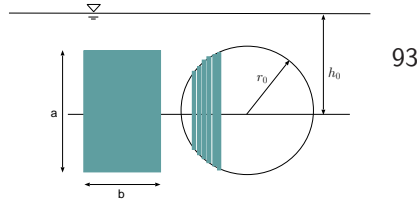


Fig. -4.28. Schematic of Immersed Cylinder.

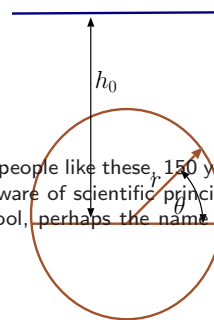


Fig. -4.29. The floating forces on Immersed Cylinder.

The total force will be the integral of the equation (4.127)

$$F = \int_0^{2\pi} \rho g (h_0 - r \sin \theta) r d\theta \sin \theta \quad (4.128)$$

Rearranging equation (4.127) transforms it to

$$F = r g \rho \int_0^{2\pi} (h_0 - r \sin \theta) \sin \theta d\theta \quad (4.129)$$

The solution of equation (4.129) is

$$F = -\pi r^2 \rho g \quad (4.130)$$

The negative sign indicate that the force acting upwards. While the horizontal force is

$$F_v = \int_0^{2\pi} (h_0 - r \sin \theta) \cos \theta d\theta = 0 \quad (4.131)$$

Typical examples to explain the buoyancy are of the vessel with thin walls put upside down into liquid. The second example of the speed of the floating bodies. Since there are no better examples, these examples are a must.

Example 4.13:

A cylindrical body, shown in Figure 4.30, is floating in liquid with density, ρ_l . The body was inserted into liquid in a such a way that the air had remained in it. Express the maximum wall thickness, t , as a function of the density of the wall, ρ_s liquid density, ρ_l and the surroundings air temperature, T_1 for the body to float. In the case where thickness is half the maximum, calculate the pressure inside the container. The container diameter is w . Assume that the wall thickness is small compared with the other dimensions ($t \ll w$ and $t \ll h$).

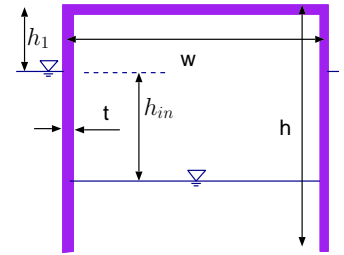


Fig. -4.30. Schematic of a thin wall floating body.

SOLUTION

The air mass in the container is

$$m_{air} = \underbrace{\pi w^2 h}_V \underbrace{\frac{\rho_{air} P_{atmos}}{RT}}_{\rho_{air}}$$

The mass of the container is

$$m_{container} = \left(\underbrace{\pi w^2 + 2\pi w h}_A \right) t \rho_s$$

The liquid amount enters into the cavity is such that the air pressure in the cavity equals to the pressure at the interface (in the cavity). Note that for the maximum thickness, the height, h_1 has to be zero. Thus, the pressure at the interface can be written as

$$P_{in} = \rho_l g h_{in}$$

On the other hand, the pressure at the interface from the air point of view (ideal gas model) should be

$$P_{in} = \frac{m_{air} R T_1}{\underbrace{h_{in} \pi w^2}_V}$$

Since the air mass didn't change and it is known, it can be inserted into the above equation.

$$\rho_l g h_{in} + P_{atmos} = P_{in} = \frac{(\pi w^2 h) \overbrace{P_{atmos}}^{\rho} R T_1}{h_{in} \pi w^2}$$

The last equation can be simplified into

$$\rho_l g h_{in} + P_{atmos} = \frac{h P_{atmos}}{h_{in}}$$

And the solution for h_{in} is

$$h_{in} = -\frac{P_{atmos} + \sqrt{4 g h P_{atmos} \rho_l + P_{atmos}^2}}{2 g \rho_l}$$

and

$$h_{in} = \frac{\sqrt{4 g h P_{atmos} \rho_l + P_{atmos}^2} - P_{atmos}}{2 g \rho_l}$$

The solution must be positive, so that the last solution is the only physical solution.

Example 4.14:

Calculate the minimum density an infinitely long equilateral triangle (three equal sides) has to be so that the sharp end is in the water.

***** Advance material can be skipped *****

Extreme Cases

The solution demonstrates that when $h \rightarrow 0$ then $h_{in} \rightarrow 0$. When the gravity approaches zero (macro gravity) then

$$h_{in} = \frac{P_{atmos}}{\rho_l g} + h - \frac{h^2 \rho_l g}{P_{atmos}} + \frac{2 h^3 \rho_l^2 g^2}{P_{atmos}^2} - \frac{5 h^4 \rho_l^3 g^3}{P_{atmos}^3} + \dots$$

This "strange" result shows that bodies don't float in the normal sense. When the floating is under vacuum condition, the following height can be expanded into

$$h_{in} = \sqrt{\frac{h P_{atmos}}{g \rho_l}} + \frac{P_{atmos}}{2 g \rho_l} + \dots$$

which shows that the large quantity of liquid enters into the container as it is expected.

End advance material

Archimedes theorem states that the force balance is at displaced weight liquid (of the same volume) should be the same as the container, the air. Thus,

$$\underbrace{\pi w^2 (h - h_{in}) g}_{\text{net displayed water}} = \underbrace{(\pi w^2 + 2 \pi w h) t \rho_s g}_{\text{container}} + \underbrace{\pi w^2 h \left(\frac{P_{atmos}}{R T_1} \right) g}_{\text{air}}$$

If air mass is neglected the maximum thickness is

$$t_{max} = \frac{2 g h w \rho_l + P_{atmos} w - w \sqrt{4 g h P_{atmos} \rho_l + P_{atmos}^2}}{(2 g w + 4 g h) \rho_l \rho_s}$$

The condition to have physical value for the maximum thickness is

$$2 g h \rho_l + P_{atmos} \geq \sqrt{4 g h P_{atmos} \rho_l + P_{atmos}^2}$$

The full solution is

$$t_{max} = \frac{(w R \sqrt{4 g h P_{atmos} \rho_l + P_{atmos}^2} - 2 g h w R \rho_l - P_{atmos} w R) T_1 + 2 g h P_{atmos} w \rho_l}{(2 g w + 4 g h) R \rho_l \rho_s T_1}$$

In this analysis the air temperature in the container immediately after insertion in the liquid has different value from the final temperature. It is reasonable as the first approximation to assume that the process is adiabatic and isentropic. Thus, the temperature in the cavity immediately after the insertion is

$$\frac{T_i}{T_f} = \left(\frac{P_i}{P_f} \right)$$

The final temperature and pressure were calculated previously. The equation of state is

$$P_i = \frac{m_{air} R T_i}{V_i}$$

The new unknown must provide additional equation which is

$$V_i = \pi w^2 h_i$$

Thickness Below The Maximum

For the half thickness $t = \frac{t_{max}}{2}$ the general solution for any given thickness below maximum is presented. The thickness is known, but the liquid displacement is still unknown. The pressure at the interface (after long time) is

$$\rho_l g h_{in} + P_{atmos} = \frac{\pi w^2 h \frac{P_{atmos}}{RT_1} RT_1}{(h_{in} + h_1) \pi w^2}$$

which can be simplified to

$$\rho_l g h_{in} + P_{atmos} = \frac{h P_{atmos}}{h_{in} + h_1}$$

The second equation is Archimedes' equation, which is

$$\pi w^2 (h - h_{in} - h_1) = (\pi w^2 + 2\pi w h) t \rho_s g + \pi w^2 h \left(\frac{P_{atmos}}{RT_1} \right) g$$

End Solution

Example 4.15:

A body is pushed into the liquid to a distance, h_0 and left at rest. Calculate acceleration and time for a body to reach the surface. The body's density is $\alpha \rho_l$, where α is ratio between the body density to the liquid density and ($0 < \alpha < 1$). Is the body volume important?

SOLUTION

The net force is

$$F = \underbrace{V g \rho_l}_{\text{liquid weight}} - \underbrace{V g \alpha \rho_l}_{\text{body weight}} = V g \rho_l (1 - \alpha)$$

But on the other side the internal force is

$$F = m a = \overbrace{V \alpha \rho_l}^m a$$

Thus, the acceleration is

$$a = g \left(\frac{1 - \alpha}{\alpha} \right)$$

If the object is left at rest (no movement) thus time will be ($h = 1/2 a t^2$)

$$t = \sqrt{\frac{2 h \alpha}{g(1 - \alpha)}}$$

If the object is very light ($\alpha \rightarrow 0$) then

$$t_{min} = \sqrt{\frac{2h\alpha}{g}} + \frac{\sqrt{2gh}\alpha^{\frac{3}{2}}}{2g} + \frac{3\sqrt{2gh}\alpha^{\frac{5}{2}}}{8g} + \frac{5\sqrt{2gh}\alpha^{\frac{7}{2}}}{16g} + \dots$$

From the above equation, it can be observed that only the density ratio is important. This idea can lead to experiment in “large gravity” because the acceleration can be magnified and it is much more than the reverse of free falling.

End Solution

Example 4.16:

In some situations, it is desired to find equivalent of force of a certain shape to be replaced by another force of a “standard” shape. Consider the force that acts on a half sphere. Find equivalent cylinder that has the same diameter that

SOLUTION

The force act on the half sphere can be found by integrating the forces around the sphere. The element force is

$$dF = (\rho_L - \rho_S) g \overbrace{r \cos \phi}^h \overbrace{\cos \theta}^{\overbrace{dA_x}^{dA}} \overbrace{\cos \phi}^{\overbrace{dA}^{r^2 d\theta d\phi}} r^2 d\theta d\phi$$

The total force is then

$$F_x = \int_0^\pi \int_0^\pi (\rho_L - \rho_S) g \cos^2 \phi \cos^2 \theta r^3 d\theta d\phi$$

The result of the integration the force on sphere is

$$F_s = \frac{\pi^2 (\rho_L - \rho_S) r^3}{4}$$

The force on equivalent cylinder is

$$F_c = \pi r^2 (\rho_L - \rho_S) h$$

These forces have to be equivalent and thus

$$\frac{\pi^2 (\rho_L - \rho_S) r^3}{4} = \pi r^2 (\rho_L - \rho_S) h$$

Thus the height is

$$\frac{h}{r} = \frac{\pi}{4}$$

End Solution

Example 4.17:

In the introduction to this section, it was assumed that above liquid is a gas with inconsequential density. Suppose that the above layer is another liquid which has a bit lighter density. Body with density between the two liquids, $\rho_l < \rho_s < \rho_h$ is floating between the two liquids. Develop the relationship between the densities of liquids and solid and the location of the solid cubical. There situations where density is a function of the depth. What will be location of solid body if the liquid density varied parabolically.

SOLUTION

In discussion to this section it was shown that net force is the body volume times the the density of the liquid. In the same vein, the body can be separated into two which one in first liquid and second in the second liquid. In this case there are two different liquid densities. The net force down is the weight of the body $\rho_c h A$. Where h is the height of the body and A is its cross section. This force is balance according to above explanation by the two liquid as

$$\rho_c h A = A h (\alpha \rho_l + (1 - \alpha) \rho_h)$$

Where α is the fraction that is in low liquid. After rearrangement it became

$$\alpha = \frac{\rho_c - \rho_h}{\rho_l - \rho_h}$$

the second part deals with the case where the density varied parabolically. The density as a function of x coordinate along h starting at point ρ_h is

$$\rho(x) = \rho_h - \left(\frac{x}{h}\right)^2 (\rho_h - \rho_l)$$

Thus the equilibration will be achieved, A is canceled on both sides, when

$$\rho_c h = \int_{x_1}^{x_1+h} \left[\rho_h - \left(\frac{x}{h}\right)^2 (\rho_h - \rho_l) \right] dx$$

After the integration the equation transferred into

$$\rho_c h = \frac{(3 \rho_l - 3 \rho_h) x_1^2 + (3 h \rho_l - 3 h \rho_h) x_1 + h^2 \rho_l + 2 h^2 \rho_h}{3 h}$$

And the location where the lower point of the body (the physical), x_1 , will be at

$$X_1 = \frac{\sqrt{3} \sqrt{3 h^2 \rho_l^2 + (4 \rho_c - 6 h^2 \rho_h) \rho_l + 3 h^2 \rho_h^2 - 12 \rho_c \rho_h + 3 h \rho_l - 3 h \rho_h}}{6 \rho_h - 2 \rho_l}$$

For linear relationship the the following results can be obtained.

$$x_1 = \frac{h \rho_l + h \rho_h - 6 \rho_c}{2 \rho_l - 2 \rho_h}$$

In many cases in reality the variations occur in small zone compare to the size of the body. Thus, the calculations can be carried out under assumption sharp change. However, if body is small compare the zone of variation, they have to accounted for.

4.6.1 Stability

Figure 4.31 shows a body made of hollow balloon and a heavy sphere connected by a thin and light rod. This arrangement has mass centroid close to the middle of the sphere. The buoyant center is below the middle of the balloon. If this arrangement is inserted into liquid and it will be floating, the balloon will be on the top and sphere on the bottom. Tilting the body with a small angle from its resting position creates a shift in the forces direction (examine Figure 4.31b). These forces create a moment which wants to return the body to the resting (original) position. When the body is at the position shown in Figure 4.31c, the body is unstable and any tilt from the original position creates moment that will further continue to move the body from its original position. This analysis doesn't violate the second law of thermodynamics. Moving bodies from an unstable position is in essence like a potential.

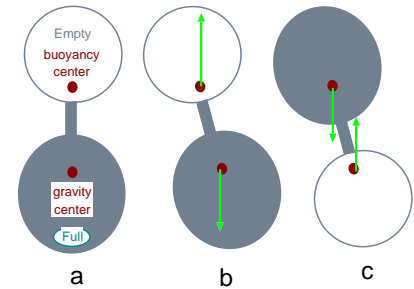


Fig. -4.31. Schematic of floating bodies.

A wooden cubic (made of pine, for example) is inserted into water. Part of the block floats above water line. The cubic mass (gravity) centroid is in the middle of the cubic. However the buoyant center is the middle of the volume under the water (see Figure 4.32). This situation is similar to Figure 4.31c. However, any experiment of this cubic wood shows that it is stable locally. Small amount of tilting of the cubic results in returning to the original position. When tilting a larger amount than $\pi/4$, it results in a flipping into the next stable position. The cubic is stable in six positions (every cubic has six faces). In fact, in any of these six positions, the body is in situation like in 4.31c. The reason for this local stability of the cubic is that other positions are less stable. If one draws the stability (later about this criterion) as a function of the rotation angle will show a sinusoidal function with four peaks in a whole rotation.

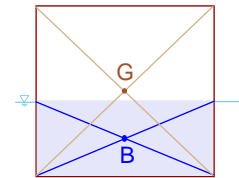


Fig. -4.32. Schematic of floating cubic.

So, the body stability must be based on the difference between the body's local positions rather than the "absolute" stability. That is, the body is "stable" in some points more than others in their vicinity. These points are raised from the buoyant force analysis. When the body is tilted at a small angle, β , the immersed part of the body center changes to a new location, B'

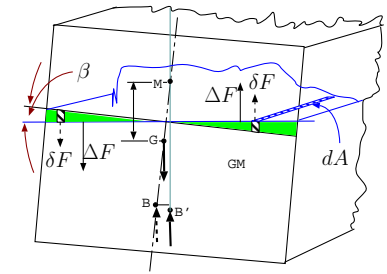


Fig. -4.33. Stability analysis of floating body.

as shown in Figure 4.33. The center of the mass (gravity) is still in the old location since the body did not change. The stability of the body is divided into three categories. If the new immersed volume creates a new center in such way that the couple forces (gravity and buoyancy) try to return the body, the original state is referred as the stable body and vice versa. The third state is when the couple forces do have zero moment, it is referred to as the neutral stable.

The body, shown in Figure 4.33, when given a tilted position, move to a new buoyant center, \mathbf{B}' . This deviation of the buoyant center from the old buoyant center location, \mathbf{B} , should to be calculated. This analysis is based on the difference of the displaced liquid. The right green area (volume) in Figure 4.33 is displaced by the same area (really the volume) on left since the weight of the body didn't change¹¹ so the total immersed section is constant. For small angle, β , the moment is calculated as the integration of the small force shown in the Figure 4.33 as ΔF . The displacement of the buoyant center can be calculated by examining the moment these forces are creating. The body weight creates opposite moment to balance the moment of the displaced liquid volume.

$$\overline{BB'}W = \mathbf{M} \quad (4.132)$$

Where \mathbf{M} is the moment created by the displaced areas (volumes), $\overline{BB'}$ is the distance between points \mathbf{B} and point \mathbf{B}' , and, W referred to the weight of the body. It can be noticed that the distance $\overline{BB'}$ is an approximation for small angles (neglecting the vertical component.). So the perpendicular distance, $\overline{BB'}$, should be

$$\overline{BB'} = \frac{\mathbf{M}}{W} \quad (4.133)$$

The moment \mathbf{M} can be calculated as

$$\mathbf{M} = \int_A \underbrace{g \rho_l x \beta dA}_{dV} x = g \rho_l \beta \int_A x^2 dA \quad (4.134)$$

The integral in the right side of equation (4.134) is referred to as the area moment of inertia and was discussed in Chapter 3. The distance, $\overline{BB'}$ can be written from equation (4.134) as

$$\overline{BB'} = \frac{g \rho_l I_{xx}}{\rho_s V_{body}} \quad (4.135)$$

¹¹It is correct to state: area only when the body is symmetrical. However, when the body is not symmetrical, the analysis is still correct because the volume and not the area is used.

The point where the gravity force direction is intersecting with the center line of the cross section is referred as metacentric point, \mathbf{M} . The location of the metacentric point can be obtained from the geometry as

$$\overline{BM} = \frac{\overline{BB'}}{\sin \beta} \quad (4.136)$$

And combining equations (4.135) with (4.136) yields

$$\overline{BM} = \frac{\oint \rho_l \beta I_{xx}}{\oint \rho_s \sin \beta V_{body}} = \frac{\rho_l I_{xx}}{\rho_s V_{body}} \quad (4.137)$$

For small angle ($\beta \sim 0$)

$$\lim_{\beta \rightarrow 0} \frac{\sin \beta}{\beta} \sim 1 \quad (4.138)$$

It is remarkable that the results is independent of the angle. Looking at Figure 4.33, the geometrical quantities can be related as

$$\overline{GM} = \frac{\overline{BM}}{\rho_s V_{body}} - \overline{BG} \quad (4.139)$$

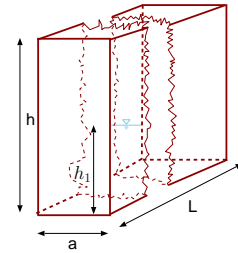


Fig. -4.34. Cubic body dimensions for stability analysis.

To understand these principles consider the following examples.

Example 4.18:

A solid block of wood of uniform density, $\rho_s = \alpha \rho_l$ where ($0 \leq \alpha \leq 1$) is floating in a liquid. Construct a graph that shows the relationship of the \overline{GM} as a function of ratio height to width. Show that the block's length, L , is insignificant for this analysis.

SOLUTION

Equation (4.139) requires that several quantities should be expressed. The moment of inertia for a block is given in Table 3.1 and is $I_{xx} = \frac{La^3}{12}$. Where L is the length into the page. The distance \overline{BG} obtained from Archimedes' theorem and can be expressed as

$$W = \rho_s \overbrace{a h L}^V = \rho_l \overbrace{a h_1 L}^{\text{immersed volume}} \implies h_1 = \frac{\rho_s}{\rho_l} h$$

Thus, the distance \overline{BG} is (see Figure 4.32)

$$\overline{BG} = \frac{h}{2} - \frac{\overbrace{\rho_s}^{h_1}}{\rho_l} h \frac{1}{2} = \frac{h}{2} \left(1 - \frac{\rho_s}{\rho_l} \right) \tag{4.140}$$

$$GM = \frac{\cancel{g} \rho_l \overbrace{L^3}^{I_{xx}}}{\cancel{g} \rho_s \underbrace{a h L}_V} - \frac{h}{2} \left(1 - \frac{\rho_s}{\rho_l} \right)$$

Simplifying the above equation provides

$$\frac{\overline{GM}}{h} = \frac{1}{12\alpha} \left(\frac{a}{h} \right)^2 - \frac{1}{2} (1 - \alpha) \tag{4.141}$$

where α is the density ratio. Notice that \overline{GM}/h isn't a function of the depth, L . This equation leads to the condition where the maximum height above which the body is not stable any more as

$$\frac{a}{h} \geq \sqrt{6(1 - \alpha)\alpha} \tag{4.142}$$

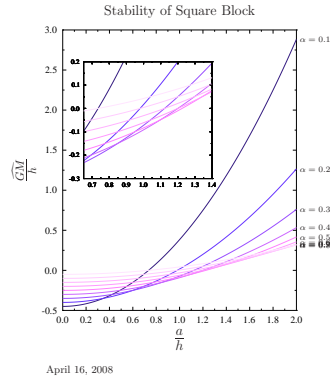


Fig. -4.35. Stability of cubic body infinity long.

End Solution

One of the interesting point that the above analysis was that there is a point above which the ratio of the height to the body width is not stable any more. In cylindrical shape equivalent equation to equation (4.142) can be expressed. For cylinder (circle) the moment of inertia is $I_{xx} = \pi b^4/64$. The distance \overline{BG} is the same as for the square shape (cubic) (see above (4.140)). Thus, the equation is

$$\frac{\overline{GM}}{h} = \frac{g}{64\alpha} \left(\frac{b}{h} \right)^2 - \frac{1}{2} (1 - \alpha)$$

And the condition for maximum height for stability is

$$\frac{b}{h} \geq \sqrt{32(1 - \alpha)\alpha}$$

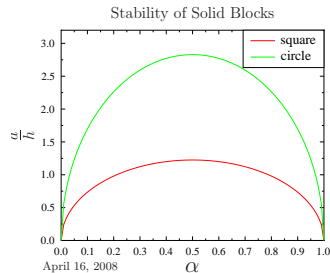


Fig. -4.36. The maximum height reverse as a function of density ratio.

This kind of analysis can be carried for different shapes and the results are shown for these two shapes in Figure 4.36. It can be noticed that the square body is more stable than the circular shape body.

Principle Main Axes

Any body has infinite number of different axes around which moment of inertia can be calculated. For each of these axes there is a different moment of inertia. With the exception of the circular shape, every geometrical shape has an axis in which the moment of inertia is without the product of inertia. This axis is where the main rotation of the body will occur. Some analysis of floating bodies are done by breaking the rotation of arbitrary axis to rotate around the two main axes. For stability analysis, it is enough to find if the body is stable around the smallest moment of inertia. For example, a square shape body has larger moment of inertia around diagonal. The difference between the previous calculation and the moment of inertia around the diagonal is

$$\Delta I_{xx} = \frac{\overbrace{\sqrt{2} a \left(\frac{\sqrt{3} a}{2}\right)^3}^{I \text{ diagonal axis}}}{6} - \frac{\underbrace{a^4}_{\text{"normal" axis}}}{12} \sim 0.07 a^4$$

Which show that if the body is stable at main axes, it must be stable at the “diagonal” axis. Thus, this problem is reduced to find the stability for principle axis.

Unstable Bodies

What happen when one increases the height ratio above the maximum height ratio? The body will flip into the side and turn to the next stable point (angle). This is not a hypothetical question, but rather practical. This happens when a ship is overloaded with containers above the maximum height. In commercial ships, the fuel is stored at the bottom of the ship and thus the mass center (point *G*) is changing during the voyage. So, the ship that was stable (positive *GM*) leaving the initial port might became unstable (negative *GM*) before reaching the destination port.

Example 4.19:

One way to make a ship to be a hydrodynamic is by making the body as narrow as possible. Suppose that two opposite sides triangle (prism) attached to each other to create a long “ship” see Figure 4.37. Supposed that $a/h \rightarrow 0$ the body will be unstable. On the other side if the $a/h \rightarrow \infty$ the body is very stable. What is the minimum ratio of a/h that keep the body stable at half of the volume in liquid (water). Assume that density ratio is $\rho_l/\rho_s = \bar{\rho}$.

SOLUTION

The answer to the question is that the limiting case where $\overline{GM} = 0$. To find this ratio equation terms in (4.139) have to found. The Volume of the body is

$$V = 2 \left(\frac{a^2 h}{2} \right) = a^2 h$$

The moment of inertia is triage (see for

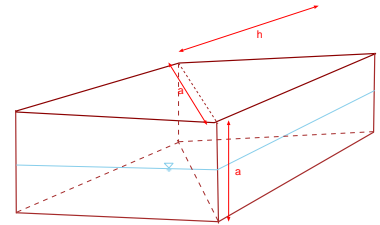


Fig. -4.37. Stability of two triangles put together.

explanation in example (3.5) is

$$I_{xx} = \frac{a h^3}{2}$$

And the volume is

$$V_{body} = a^2 \sqrt{h^2 - \frac{a^2}{4}} = a^2 h \sqrt{1 - \frac{1}{4} \frac{a^2}{h^2}}$$

The point **B** is a function of the density ratio of the solid and liquid. Denote the liquid density as ρ_l and solid density as ρ_s . The point **B** can be expressed as

$$B = \frac{a \rho_s}{2 \rho_l}$$

And thus the distance \overline{BG} is

$$\overline{BG} = \frac{a}{2} \left(1 - \frac{\rho_s}{\rho_l} \right)$$

The limiting condition requires that $\overline{GM} = 0$ so that

$$\frac{\rho_l I_{xx}}{\rho_s V_{body}} = \overline{BG}$$

Or explicitly

$$\frac{\rho_l \frac{a h^3}{2}}{\rho_s a^2 h \sqrt{1 - \frac{1}{4} \frac{a^2}{h^2}}} = \frac{a}{2} \left(1 - \frac{\rho_s}{\rho_l} \right)$$

After rearrangement and using the definitions of $\xi = h/a$ $\bar{\rho} = \rho_l/\rho_s$ results in

$$\frac{\bar{\rho} \xi^2}{\sqrt{1 - \frac{\xi^2}{4}}} = \left(1 - \frac{1}{\bar{\rho}} \right)$$

The solution of the above solution is obtained by squaring both sides and defining a new variable such as $x = \xi^2$. After the about manipulation and selecting the positive value and to keep stability as

$$x < \frac{\sqrt{\frac{64 \bar{\rho}^4 - 64 \bar{\rho}^3 + \bar{\rho}^2 - 2 \bar{\rho} + 1}{\bar{\rho}} + \frac{1}{\bar{\rho}} - 1}}{2 \sqrt{2} \bar{\rho}}$$

End Solution

4.6.1.1 Stability of Body with Shifting Mass Centroid

A new point can be defined as G_c . This point is the intersection of the center line with the vertical line from G' .

$$\overline{G'G_c} = \frac{\overline{GG'}}{\sin \beta} \tag{4.147}$$

The distance that was used before \overline{GM} is replaced by the criterion for stability by $\overline{G_cM}$ and is expressed as

$$\overline{G_cM} = \frac{g \rho_A I_{xxA}}{\rho_s V_{body}} - \overline{BG} - \frac{1}{m_{total}} \frac{I_{xxb}}{V_b} \tag{4.148}$$

If there are more than one tank partially filled with liquid, the general formula is

$$\overline{G_cM} = \frac{g \rho_A I_{xxA}}{\rho_s V_{body}} - \overline{BG} - \frac{1}{m_{total}} \sum_{i=1}^n \frac{I_{xxbi}}{V_{bi}} \tag{4.149}$$

One way to reduce the effect of the moving mass center due to liquid is done by substituting a single tank with several tanks. The moment of inertia of the combine two tanks is smaller than the moment of inertia of a single tank. Increasing the number of tanks reduces the moment of inertia. The engineer could design the tanks in such a way that the moment of inertia is operationally changed. This control of the stability, \overline{GM} , can be achieved by having some tanks spanning across the entire body with tanks spanning on parts of the body. Movement of the liquid (mostly the fuel and water) provides way to control the stability, GM , of the ship.

4.6.1.2 Metacentric Height, \overline{GM} , Measurement

The metacentric height can be measured by finding the change in the angle when a weight is moved on the floating body.

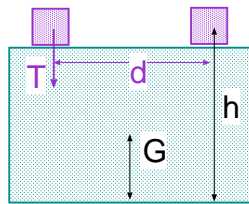
Moving the weight, T a distance, d then the moment created is

$$M_{weight} = T d \tag{4.150}$$

This moment is balanced by

$$M_{righting} = W_{total} \overline{GM}_{new} \theta \tag{4.151}$$

Fig. -4.39. Measurement of GM of floating body.



Where, W_{total} , is the total weight of the floating body including measuring weight. The angle, θ , is measured as the difference in the orientation of the floating body. The metacentric height is

$$\overline{GM}_{new} = \frac{T d}{W_{total} \theta} \tag{4.152}$$

If the change in the \overline{GM} can be neglected, equation (4.152) provides the solution. The calculation of \overline{GM} can be improved by taking into account the effect of the measuring weight. The change in height of G is

$$g m_{total} G_{new} = g m_{ship} G_{actual} + g T h \quad (4.153)$$

Combining equation (4.153) with equation (4.152) results in

$$\overline{GM}_{actual} = \overline{GM}_{new} \frac{m_{total}}{m_{ship}} - h \frac{T}{m_{ship}} \quad (4.154)$$

The weight of the ship is obtained from looking at the ship depth.

4.6.1.3 Stability of Submerged Bodies

The analysis of submerged bodies is different from the stability when the body lay between two fluid layers with different density. When the body is submerged in a single fluid layer, then none of the changes of buoyant centroid occurs. Thus, the mass centroid must be below the buoyant centroid in order to have stable condition.

However, all fluids have density varied in some degree. In cases where the density changes significantly, it must be taken into account. For an example of such a case is an object floating in a solar pond where the upper layer is made of water with lower salinity than the bottom layer (change up to 20% of the density). When the floating object is immersed in the two layers, the stability analysis must take into account the changes of the displaced liquids of the two liquid layers. The calculations for such cases are a bit more complicated but based on the similar principles. Generally, this density change helps to increase the stability of the floating bodies. This analysis is out of the scope of this book (for now).

4.6.1.4 Stability of None Systematical or “Strange” Bodies

While most floating bodies are symmetrical or semi-symmetrical, there are situations where the body has a “strange” and/or un-symmetrical body. Consider the first strange body that has an abrupt step change as shown in Figure 4.40. The body weight doesn't change during the rotation that the green area on the left and the green area on right are the same (see Figure 4.40). There are two situations that can occur. After the tilting, the upper part of the body is above the liquid or part of the body is submerged under the water. The mathematical condition for the border is when $b = 3a$. For the case of $b < 3a$

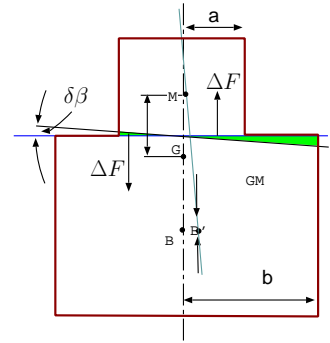


Fig. -4.40. Calculations of \overline{GM} for abrupt shape body.

the calculation of moment of inertia are similar to the previous case. The moment created by change in the displaced liquid (area) act in the same fashion as the before. The center of the moment is needed be found. This point is the intersection of the liquid line with the brown middle line. The moment of inertia should be calculated around this axis.

For the case where $b < 3a$ some part is under the liquid. The amount of area under the liquid section depends on the tilting angle. These calculations are done as if none of the body under the liquid. This point is intersection point liquid with lower body and it is needed to be calculated. The moment of inertia is calculated around this point (note the body is "ended" at end of the upper body). However, the moment to return the body is larger than actually was calculated and the bodies tend to be more stable (also for other reasons).

4.6.1.5 Neutral frequency of Floating Bodies

This case is similar to pendulum (or mass attached to spring). The governing equation for the pendulum is

$$\ell \ddot{\beta} - g \beta = 0 \quad (4.155)$$

Where here ℓ is length of the rode (or the line/wire) connecting the mass with the rotation point. Thus, the frequency of pendulum is $\frac{1}{2\pi} \sqrt{\frac{g}{\ell}}$ which measured in Hz . The period of the cycle is $2\pi \sqrt{\ell/g}$. Similar situation exists in the case of floating bodies. The basic differential equation is used to balance and is

$$\overbrace{I \ddot{\beta}}^{\text{rotation}} - \overbrace{V \rho_s \overline{GM} \beta}^{\text{rotating moment}} = 0 \quad (4.156)$$

In the same fashion the frequency of the floating body is

$$\frac{1}{2\pi} \sqrt{\frac{V \rho_s \overline{GM}}{I_{body}}} \quad (4.157)$$

and the period time is

$$2\pi \sqrt{\frac{I_{body}}{V \rho_s \overline{GM}}} \quad (4.158)$$

In general, the larger \overline{GM} the more stable the floating body is. Increase in \overline{GM} increases the frequency of the floating body. If the floating body is used for transport humans and/or other creatures or sensitive cargo it requires to reduce the \overline{GM} so that the traveling will be smother.

4.6.2 Surface Tension

The surface tension is one of the mathematically complex topic and related to many phenomena like boiling, coating, etc. In this section, only simplified topics like constant value will be discussed.

Example 4.20:

In interaction of the molecules shown in Figure ? describe the existence of surface tension. Explain why this description is erroneous?

SOLUTION

The upper layer of the molecules have unbalanced force towards the liquid phase. Newton's law states when there is unbalanced force, the body should be accelerate. However, in this case, the liquid is not in motion. Thus, the common explanation is wrong.

End Solution



Fig. -4.41. A heavy needle is floating on a liquid.

Example 4.21:

Needle is made of steel and is heavier than water and many other liquids. However, the surface tension between the needle and the liquid hold the needle above the liquid. After certain diameter, the needle cannot be held by the liquid. Calculate the maximum diameter needle that can be inserted into liquid without drowning.

SOLUTION

Under Construction

End Solution

4.7 Rayleigh–Taylor Instability

RayleighTaylor instability (or RT instability) is named after Lord Rayleigh and G. I. Taylor. There are situations where a heavy liquid layer is placed over a lighter fluid layer. This situation has engineering implications in several industries. For example in die casting, liquid metal is injected in a cavity filled with air. In poor designs or other situations, some air is not evacuated and stay in small cavity on the edges of the shape to be casted. Thus, it can create a situation where the liquid metal is above the air but cannot penetrate into the cavity because of instability.

This instability deals with a dense, heavy fluid that is being placed above a lighter fluid in a gravity field perpendicular to interface. Example for such systems are dense water over oil (liquid-liquid), or water over air(gas-liquid). The original Rayleigh's paper deals with the dynamics and density variations. For example, density variations according to the bulk modulus (see section 4.3.3.2) are always stable but unstable of the density is in the reversed order.

Supposed that a liquid density is arbitrary function of the height. This distortion can be as a result of heavy fluid above the lighter liquid. This analysis asks the question what happen when a small amount of liquid from above layer enter into the lower layer? Whether this liquid continue and will grow or will it return to its original conditions? The surface tension is the opposite mechanism that will returns the liquid to its original place. This analysis is referred to the case of infinite or very large surface. The simplified case is the two different uniform densities. For example a heavy fluid density, ρ_L , above lower fluid with lower density, ρ_G .

For perfectly straight interface, the heavy fluid will stay above the lighter fluid. If the surface will disturbed, some of heavy liquid moves down. This disturbance can grow or returned to its original situation. This condition is determined by competing forces, the surface density, and the buoyancy forces. The fluid above the depression is in equilibrium with the sounding pressure since the material is extending to infinity. Thus, the force that acting to get the above fluid down is the buoyancy force of the fluid in the depression.

The depression is returned to its original position if the surface forces are large enough. In that case, this situation is considered to be stable. On the other hand, if the surface forces (surface tension) are not sufficient, the situation is unstable and the heavy liquid enters into the liquid fluid zone and vice versa. As usual there is the neutral stable when the forces are equal. Any continues function can be expanded in serious of cosines. Thus, example of a cosine function will be examined. The conditions that required from this function will be required from all the other functions. The disturbance is of the following

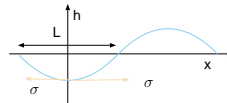


Fig. -4.42. Description of depression to explain the Rayleigh-Taylor instability.

$$h = -h_{max} \cos \frac{2\pi x}{L} \tag{4.159}$$

where h_{max} is the maximum depression and L is the characteristic length of the depression. The depression has different radius as a function of distance from the center of the depression, x . The weakest point is at $x = 0$ because symmetrical reasons the surface tension does not act against the gravity as shown in Figure (4.42). Thus, if the center point of the depression can "hold" the intrusive fluid then the whole system is stable.

The radius of any equation is expressed by equation (1.47). The first derivative of \cos around zero is \sin which is approaching zero or equal to zero. Thus, equation

(1.47) can be approximated as

$$\frac{1}{R} = \frac{d^2 h}{dx^2} \quad (4.160)$$

For equation (4.159) the radius is

$$\frac{1}{R} = -\frac{4\pi^2 h_{max}}{L^2} \quad (4.161)$$

According to equation (1.38) the pressure difference or the pressure jump is due to the surface tension at this point must be

$$P_H - P_L = \frac{4h_{max}\sigma\pi^2}{L^2} \quad (4.162)$$

The pressure difference due to the gravity at the edge of the disturbance is then

$$P_H - P_L = g(\rho_H - \rho_L)h_{max} \quad (4.163)$$

Comparing equations (4.162) and (4.163) show that if the relationship is

$$\frac{4\sigma\pi^2}{L^2} > g(\rho_H - \rho_L) \quad (4.164)$$

It should be noted that h_{max} is irrelevant for this analysis as it is canceled. The point where the situation is neutral stable

$$L_c = \sqrt{\frac{4\pi^2\sigma}{g(\rho_H - \rho_L)}} \quad (4.165)$$

An alternative approach to analyze this instability is suggested here. Consider the situation described in Figure 4.43. If all the heavy liquid “attempts” to move straight down, the lighter liquid will “prevent” it. The lighter liquid needs to move up at the same time but in a different place. The heavier liquid needs to move in one side and the lighter liquid in another location. In this process the heavier liquid “enter” the lighter liquid in one point and creates a depression as shown in Figure 4.43.

To analyze it, considered two control volumes bounded by the blue lines in Figure 4.43. The first control volume is made of a cylinder with a radius r and the second is the depression below it. The “extra” lines of the depression should be ignored, they are not part of the control volume. The horizontal forces around the control volume are canceling each other. At the top, the force is atmospheric pressure times the area. At the cylinder bottom, the force is $\rho g h \times A$. This acts

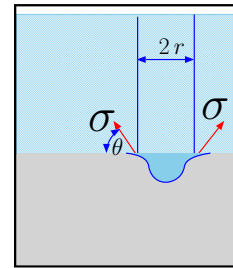


Fig. -4.43. Description of depression to explain the instability.

against the gravity force which make the cylinder to be in equilibrium with its surroundings if the pressure at bottom is indeed $\rho g h$.

For the depression, the force at the top is the same force at the bottom of the cylinder. At the bottom, the force is the integral around the depression. It can be approximated as a flat cylinder that has depth of $r \pi/4$ (read the explanation in the example 4.16) This value is exact if the shape is a perfect half sphere. In reality, the error is not significant. Additionally when the depression occurs, the liquid level is reduced a bit and the lighter liquid is filling the missing portion. Thus, the force at the bottom is

$$F_{bottom} \sim \pi r^2 \left[\left(\frac{\pi r}{4} + h \right) (\rho_L - \rho_G) g + P_{atmos} \right] \quad (4.166)$$

The net force is then

$$F_{bottom} \sim \pi r^2 \left(\frac{\pi r}{4} \right) (\rho_L - \rho_G) g \quad (4.167)$$

The force that hold this column is the surface tension. As shown in Figure 4.43, the total force is then

$$F_{\sigma} = 2 \pi r \sigma \cos \theta \quad (4.168)$$

The forces balance on the depression is then

$$2 \pi r \sigma \cos \theta \sim \pi r^2 \left(\frac{\pi r}{4} \right) (\rho_L - \rho_G) g \quad (4.169)$$

The radius is obtained by

$$r \sim \sqrt{\frac{2 \pi \sigma \cos \theta}{(\rho_L - \rho_G) g}} \quad (4.170)$$

The maximum surface tension is when the angle, $\theta = \pi/2$. At that case, the radius is

$$r \sim \sqrt{\frac{2 \pi \sigma}{(\rho_L - \rho_G) g}} \quad (4.171)$$

The maximum possible radius of the depression depends on the geometry of the container. For the cylindrical geometry, the maximum depression radius is about half for the container radius (see Figure 4.44). This radius is limited because the lighter liquid has to enter at the same time into the heavier liquid zone. Since the "exchange" volumes of these two process are the same, the specific radius is limited. Thus, it can be written that the minimum radius is

$$r_{mintube} = 2 \sqrt{\frac{2 \pi \sigma}{g (\rho_L - \rho_G)}} \quad (4.172)$$

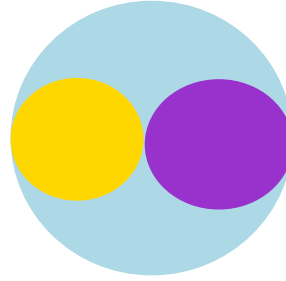


Fig. -4.44. The cross section of the interface. The purple color represents the maximum heavy liquid raising area. The yellow color represents the maximum lighter liquid that are “going down.”

The actual radius will be much larger. The heavier liquid can stay on top of the lighter liquid without being turned upside down when the radius is smaller than the equation 4.172. This analysis introduces a new dimensional number that will be discussed in a greater length in the Dimensionless chapter. In equation (4.172) the angle was assumed to be 90 degrees. However, this angle is never obtained. The actual value of this angle is about $\pi/4$ to $\pi/3$ and in only extreme cases the angle exceed this value (considering dynamics). In Figure 4.44, it was shown that the depression and the raised area are the same. The actual area of the depression is only a fraction of the interfacial cross section and is a function. For example, the depression is larger for square area. These two scenarios should be inserting into equation 4.168 by introducing experimental coefficient.

Example 4.22:

Estimate the minimum radius to insert liquid aluminum into represent tube at temperature of 600[K]. Assume that the surface tension is 400[mN/m]. The density of the aluminum is 2400kg/m³.

SOLUTION

The depression radius is assume to be significantly smaller and thus equation (4.171) can be used. The density of air is negligible as can be seen from the temperature compare to the aluminum density.

$$r \sim \sqrt{\frac{8\pi \overbrace{0.4}^{\sigma}}{2400 \times 9.81}}$$

The minimum radius is $r \sim 0.02[m]$ which demonstrates the assumption of $h \gg r$ was appropriate.

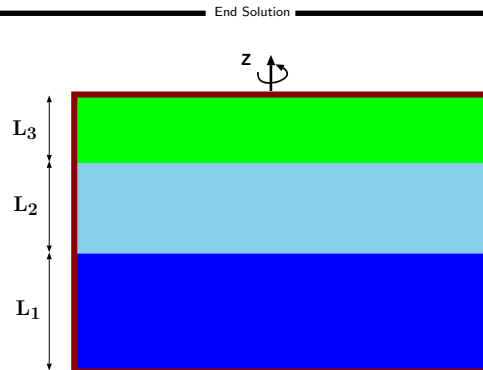


Fig. -4.45. Three liquids layers under rotation with various critical situations.

Open Question by April 15, 2010

The best solution of the following question will win 18 U.S. dollars and your name will be associated with the solution in this book.

Example 4.23:

A canister shown in Figure 4.45 has three layer of different fluids with different densities. Assume that the fluids do not mix. The canister is rotate with circular velocity, ω . Describe the interface of the fluids consider all the limiting cases. Is there a difference if the fluids are compressible? Where is the maximum pressure points? For the case that the fluids are compressible, the canister top center is connected to another tank with equal pressure to the canister before the rotation (the connection point). What happen after the canister start to be rotated? Calculated the volume that will enter or leave, for known geometries of the fluids. You the ideal gas model. You can assume that the process is isothermal. Is there difference if the process is isentropic? If so what is the difference?

Part I

Integral Analysis

CHAPTER 5

The Control Volume and Mass Conservation

5.1 Introduction

This chapter presents a discussion on the control volume and will be focused on the conservation of the mass. When the fluid system moves or changes, one wants to find or predict the velocities in the system. The main target of such analysis is to find the value of certain variables. This kind of analysis is reasonable and it referred to in the literature as the Lagrangian Analysis. This name is in honored J. L. Langrange (1736–1813) who formulated the equations of motion for the moving fluid particles.

Even though this system looks reasonable, the Lagrangian system turned out to be difficult to solve and to analyze. This method applied and used in very few cases. The main difficulty lies in the fact that every particle has to be traced to its original state. Leonard Euler (1707–1783) suggested an alternative approach. In Euler's approach the focus is on a defined point or a defined volume. This methods is referred as Eulerian method.

The Eulerian method focuses on a defined area or location to find the needed information. The use of the Eulerian methods leads to a set differentiation equations that is referred to as Navier–Stokes equations which are commonly used. These differential equations will be used in the later part of this book. Ad-

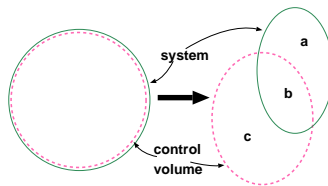


Fig. -5.1. Control volume and system before and after motion.

ditionally, the Eulerian system leads to integral equations which are the focus of this part of the book. The Eulerian method plays well with the physical intuition of most people. This method has its limitations and for some cases the Lagrangian is preferred (and sometimes the only possibility). Therefore a limited discussion on the Lagrangian system will be presented (later version).

Lagrangian equations are associated with the system while the Eulerian equation are associated with the control volume. The difference between the system and the control volume is shown in Figure ???. The green lines in Figure ??? represent the system. The red dotted lines are the control volume. At certain time the system and the control volume are identical location. After a certain time, some of the mass in the system exited the control volume which are marked "a" in Figure ???. The material that remained in the control volume is marked as "b". At the same time, the control volume gains some material which is marked as "c".

5.2 Control Volume

The Eulerian method requires to define a control volume (some time more than one). The control volume is a defined volume that was discussed earlier. The control volume is differentiated into two categories of control volumes, non-deformable and deformable.

Non-deformable control volume is a control volume which is fixed in space relative to an one coordinate system. This coordinate system may be in a relative motion to another (almost absolute) coordinate system.

Deformable control volume is a volume having part of all of its boundaries in motion during the process at hand.

In the case where no mass crosses the boundaries, the control volume is a system. Every control volume is the focus of the certain interest and will be dealt with the basic equations, mass, momentum, energy, entropy etc.

Two examples of control volume are presented to illustrate difference between a deformable control volume and non-deformable control volume. Flow in conduits can be analyzed by looking in a control volume between two locations. The coordinate system could be fixed to the conduit. The control volume chosen is non-deformable control volume. The control volume should be chosen so that the analysis should be simple and dealt with as less as possible issues which are not in question. When a piston pushing gases a good choice of control volume is a deformable control volume that is a head the piston inside the cylinder as shown in Figure 5.2.

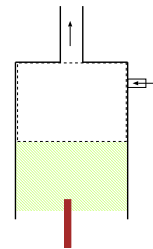


Fig. -5.2. Control volume of a moving piston with in and out flow.

5.3 Continuity Equation

In this chapter and the next three chapters, the conservation equations will be applied to the control volume. In this chapter, the mass conservation will be discussed. The system mass change is

$$\frac{D m_{sys}}{Dt} = \frac{D}{Dt} \int_{V_{sys}} \rho dV = 0 \quad (5.1)$$

The system mass after some time, according to Figure ??, is made of

$$m_{sys} = m_{c.v.} + m_a - m_c \quad (5.2)$$

The change of the system mass is by definition is zero. The change with time (time derivative of equation (5.2)) results in

$$0 = \frac{D m_{sys}}{Dt} = \frac{d m_{c.v.}}{dt} + \frac{d m_a}{dt} - \frac{d m_c}{dt} \quad (5.3)$$

The first term in equation (5.3) is the derivative of the mass in the control volume and at any given time is

$$\frac{d m_{c.v.}(t)}{dt} = \frac{d}{dt} \int_{V_{c.v.}} \rho dV \quad (5.4)$$

and is a function of the time.

The interface of the control volume can move. The actual velocity of the fluid leaving the control volume is the relative velocity (see Figure 5.3). The relative velocity is

$$\vec{U}_r = \vec{U}_f - \vec{U}_b \quad (5.5)$$

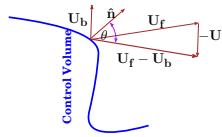


Fig. -5.3. Schematics of velocities at the interface.

Where U_f is the liquid velocity and U_b is the boundary velocity (see Figure 5.3). The velocity component that is perpendicular to the surface is

$$U_{rn} = -\hat{n} \cdot \vec{U}_r = U_r \cos \theta \quad (5.6)$$

Where \hat{n} is an unit vector perpendicular to the surface. The convention of direction is taken positive if flow out the control volume and negative if the flow is into the control volume. The mass flow out of the control volume is the system mass that is not included in the control volume. Thus, the flow out is

$$\frac{d m_a}{dt} = \int_{S_{cv}} \rho_s U_{rn} dA \quad (5.7)$$

It has to be emphasized that the density is taken at the surface thus the subscript s . In the same manner, the flow rate in is

$$\frac{dm_b}{dt} = \int_{S_{c.v.}} \rho_s U_{rn} dA \quad (5.8)$$

It can be noticed that the two equations (5.8) and (5.7) are similar and can be combined, taking the positive or negative value of U_{rn} with integration of the entire system as

$$\frac{dm_a}{dt} - \frac{dm_b}{dt} = \int_{S_{c.v.}} \rho_s U_{rn} dA \quad (5.9)$$

applying negative value to keep the convention. Substituting equation (5.9) into equation (5.3) results in

$$\frac{d}{dt} \int_{c.v.} \rho_s dV = - \int_{S_{c.v.}} \rho U_{rn} dA \quad (5.10)$$

Equation (5.10) is essentially accounting of the mass. Again notice the negative sign in surface integral. The negative sign is because flow out marked positive which reduces of the mass (negative derivative) in the control volume. The change of mass change inside the control volume is net flow in or out of the control system.

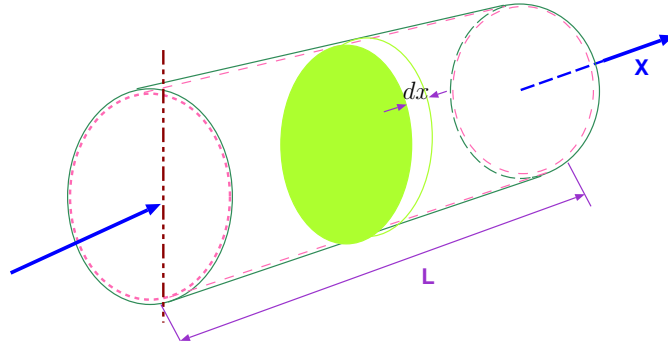


Fig. -5.4. Schematics of flow in pipe with varying density as a function time for example 5.1.

The next example is provided to illustrate this concept.

Example 5.1:

The density changes in a pipe, due to temperature variation and other reasons, can be approximated as

$$\frac{\rho(x, t)}{\rho_0} = \left(1 - \frac{x}{L}\right)^2 \cos \frac{t}{t_0}.$$

The conduit shown in Figure 5.4 length is L and its area is A . Express the mass flow in and/or out, and the mass in the conduit as function of time. Write the expression for the mass change in the pipe.

SOLUTION

Here it is very convenient to choose a non-deformable control volume that is inside the conduit dV is chosen as $\pi R^2 dx$. Using equation (5.10), the flow out (or in) is

$$\frac{d}{dt} \int_{c.v.} \rho dV = \frac{d}{dt} \int_{c.v.} \overbrace{\rho_0 \left(1 - \frac{x}{L}\right)^2 \cos\left(\frac{t}{t_0}\right)}^{\rho(t)} \overbrace{\pi R^2 dx}^{dV}$$

The density is not a function of radius, r and angle, θ and they can be taken out the integral as

$$\frac{d}{dt} \int_{c.v.} \rho dV = \pi R^2 \frac{d}{dt} \int_{c.v.} \rho_0 \left(1 - \frac{x}{L}\right)^2 \cos\left(\frac{t}{t_0}\right) dx$$

which results in

$$\text{Flow Out} = \overbrace{\pi R^2}^A \frac{d}{dt} \int_0^L \rho_0 \left(1 - \frac{x}{L}\right)^2 \cos\frac{t}{t_0} dx = -\frac{\pi R^2 L \rho_0}{3 t_0} \sin\left(\frac{t}{t_0}\right)$$

The flow out is a function of length, L , and time, t , and is the change of the mass in the control volume.

End Solution

5.3.1 Non Deformable Control Volume

When the control volume is fixed with time, the derivative in equation (5.10) can enter the integral since the boundaries are fixed in time and hence,

$$\int_{V_{c.v.}} \frac{d\rho}{dt} dV = - \int_{S_{c.v.}} \rho U_{rn} dA \quad (5.11)$$

Equation (5.11) is simpler than equation (5.10).

5.3.2 Constant Density Fluids

Further simplifications of equations (5.10) can be obtained by assuming constant density and the equation (5.10) become conservation of the volume.

5.3.2.1 Non Deformable Control Volume

For this case the volume is constant therefore the mass is constant, and hence the mass change of the control volume is zero. Hence, the net flow (in and out) is zero. This condition can be written mathematically as

$$\overbrace{\frac{d}{dt} \int}^{=0} \rightarrow \int_{S_{c.v.}} V_{rn} dA = 0 \quad (5.12)$$

or in a more explicit form as

$$\int_{S_{in}} V_{rn} dA = \int_{S_{out}} V_{rn} dA = 0 \quad (5.13)$$

Notice that the density does not play a role in this equation since it is canceled out. Physically, the meaning is that volume flow rate in and the volume flow rate out have to equal.

5.3.2.2 Deformable Control Volume

The left hand side of question (5.10) can be examined further to develop a simpler equation by using the extend Leibniz integral rule for a constant density and result in

$$\frac{d}{dt} \int_{c.v.} \rho dV = \overbrace{\int_{c.v.} \frac{d\rho}{dt} dV}^{\text{thus, } =0} + \rho \int_{S_{c.v.}} \hat{n} \cdot U_b dA = \rho \int_{S_{c.v.}} U_{bn} dA \quad (5.14)$$

where U_b is the boundary velocity and U_{bn} is the normal component of the boundary velocity.

$$\int_{S_{c.v.}} U_{bn} dA = \int_{S_{c.v.}} U_{rn} dA \quad (5.15)$$

The meaning of the equation (5.15) is the net growth (or decrease) of the Control volume is by net volume flow into it. Example 5.2 illustrates this point.

Example 5.2:

Liquid fills a bucket as shown in Figure 5.5. The average velocity of the liquid at the exit of the filling pipe is U_p and cross section of the pipe is A_p . The liquid fills a bucket with cross section area of \mathbf{A} and instantaneous height is h . Find the height as a function of the other parameters. Assume that the density is constant and at the boundary interface $A_j = 0.7 A_p$. And where A_j is the area of jet when touching the liquid boundary in bucket. The last assumption is result of the energy equation (with

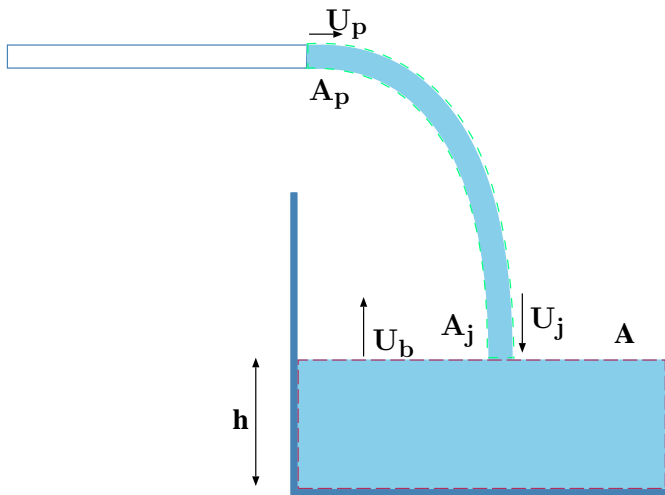


Fig. -5.5. Filling of the bucket and choices of the deformable control volumes for example 5.2.

some influence of momentum equation). The relationship is function of the distance of the pipe from the boundary of the liquid. However, this effect can be neglected for this range which this problem. In reality, the ratio is determined by height of the pipe from the liquid surface in the bucket. Calculate the bucket liquid interface velocity.

SOLUTION

This problem requires two deformable control volumes. The first control is around the jet and second is around the liquid in the bucket. In this analysis, several assumptions must be made. First, no liquid leaves the jet and enters the air. Second, the liquid in the bucket has a straight surface. This assumption is a strong assumption for certain conditions but it will be not discussed here since it is advance topic. Third, there are no evaporation or condensation processes. Fourth, the air effects are negligible. The control volume around the jet is deformable because the length of the jet shrinks with the time. The mass conservation of the liquid in the bucket is

$$\overbrace{\int_{c.v.} U_{bn} dA}^{\text{boundary change}} = \overbrace{\int_{c.v.} U_{rn} dA}^{\text{flow in}}$$

where U_{bn} is the perpendicular component of velocity of the boundary. Substituting the known values for U_{rn} results in

$$\int_{c.v.} U_b dA = \int_{c.v.} \overbrace{(U_j + U_b)}^{U_{rn}} dA$$

The integration can be carried when the area of jet is assumed to be known as

$$U_b A = A_j (U_j + U_b) \quad (5.11.a)$$

To find the jet velocity, U_j , the second control volume around the jet is used as the following

$$\underbrace{U_p A_p}_{\text{flow in}} - \underbrace{A_j (U_b + U_j)}_{\text{flow out}} = \underbrace{-A_j U_b}_{\text{boundary change}} \quad (5.11.b)$$

The above two equations (5.11.a) and (5.11.b) are enough to solve for the two unknowns. Substituting the first equation, (5.11.a) into (5.11.b) and using the ratio of $A_j = 0.7 A_p$ results

$$U_p A_p - U_b A = -0.7 A_p U_b \quad (5.11.c)$$

The solution of equation (5.11.c) is

$$U_b = \frac{A_p}{A - 0.7 A_p}$$

It is interesting that many individuals intuitively will suggest that the solution is $U_b A_p / A$. When examining solution there are two limits. The first limit is when $A_p = A/0.7$ which is

$$U_b = \frac{A_p}{0} = \infty$$

The physical meaning is that surface is filled instantly. The other limit is that and $A_p/A \rightarrow 0$ then

$$U_b = \frac{A_p}{A}$$

which is the result for the “intuitive” solution. It also interesting to point out that if the filling was from other surface (not the top surface), e.g. the side, the velocity will be $U_b = U_p$ in the limiting case and not infinity. The reason for this difference is that the liquid already fill the bucket and has not to move into bucket.

End Solution

Example 5.3:

Balloon is attached to a rigid supply in which is supplied by a constant the mass rate, m_i . Calculate the velocity of the balloon boundaries assuming constant density.

SOLUTION

The applicable equation is

$$\int_{c.v.} U_{bn} dA = \int_{c.v.} U_{rn} dA$$

The entrance is fixed, thus the relative velocity, U_{rn} is

$$U_{rn} = \begin{cases} -U_p & \text{@ the valve} \\ 0 & \text{every else} \end{cases}$$

Assume equal distribution of the velocity in balloon surface and that the center of the balloon is moving, thus the velocity has the following form

$$U_b = U_x \hat{x} + U_{br} \hat{r}$$

Where \hat{x} is unit coordinate in x direction and U_x is the velocity of the center and where \hat{r} is unit coordinate in radius from the center of the balloon and U_{br} is the velocity in that direction. The right side of equation (5.15) is the net change due to the boundary is

$$\int_{S_{c.v.}} (U_x \hat{x} + U_{br} \hat{r}) \cdot \hat{n} dA = \overbrace{\int_{S_{c.v.}} (U_x \hat{x}) \cdot \hat{n} dA}^{\text{center movement}} + \overbrace{\int_{S_{c.v.}} (U_{br} \hat{r}) \cdot \hat{n} dA}^{\text{net boundary change}}$$

The first integral is zero because it is like movement of solid body and also yield this value mathematically (excises for mathematical oriented student). The second integral (notice $\hat{n} = \hat{r}$) yields

$$\int_{S_{c.v.}} (U_{br} \hat{r}) \cdot \hat{n} dA = 4 \pi r^2 U_{br}$$

Substituting into the general equation yields

$$\rho \overbrace{4 \pi r^2}^A U_{br} = \rho U_p A_p = m_i$$

Hence,

$$U_{br} = \frac{m_i}{\rho 4 \pi r^2}$$

The center velocity is (also) exactly U_{br} . The total velocity of boundary is

$$U_t = \frac{m_i}{\rho 4 \pi r^2} (\hat{x} + \hat{r})$$

It can be noticed that the velocity at the opposite to the connection to the rigid pipe which is double of the center velocity.

End Solution

5.3.2.3 One-Dimensional Control Volume

Additional simplification of the continuity equation is of one dimensional flow. This simplification provides very useful description for many fluid flow phenomena. The main assumption made in this model is that the properties in the across section are only function of x coordinate . This assumptions leads

$$\int_{A_2} \rho_2 U_2 dA - \int_{A_1} \rho_1 U_1 dA = \frac{d}{dt} \int_{V(x)} \rho(x) \overbrace{A(x) dx}^{dV} \quad (5.16)$$

When the density can be considered constant equation (5.16) is reduced to

$$\int_{A_2} U_2 dA - \int_{A_1} U_1 dA = \frac{d}{dt} \int A(x) dx \quad (5.17)$$

For steady state but with variations of the velocity and variation of the density reduces equation (5.16) to become

$$\int_{A_2} \rho_2 U_2 dA = \int_{A_1} \rho_1 U_1 dA \quad (5.18)$$

For steady state and uniform density and velocity equation (5.18) reduces further to

$$\rho_1 A_1 U_1 = \rho_2 A_2 U_2 \quad (5.19)$$

For incompressible flow (constant density), continuity equation is at its minimum form of

$$U_1 A_1 = A_2 U_2 \quad (5.20)$$

The next example is of semi one-dimensional example to illustrate equation (5.16).

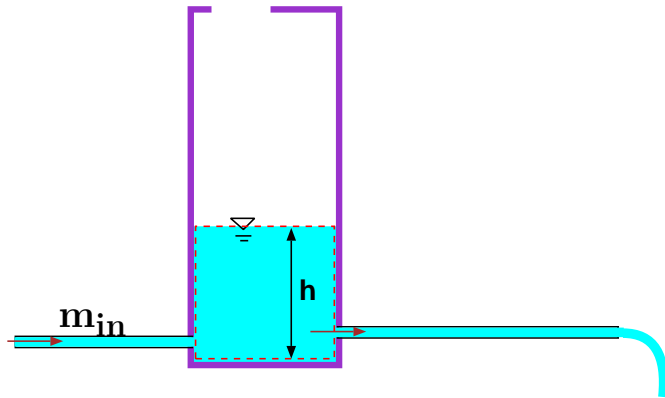


Fig. -5.6. Height of the liquid for example 5.4.

Example 5.4:

Liquid flows into tank in a constant mass flow rate of a . The mass flow rate out is function of the height. First assume that $q_{out} = b h$ second Assume as $q_{out} = b \sqrt{h}$. For the first case, determine the height, h as function of the time. Is there a critical value and then if exist find the critical value of the system parameters. Assume that the height at time zero is h_0 . What happen if the $h_0 = 0$?

SOLUTION

The control volume for both cases is the same and it is around the liquid in the tank. It can be noticed that control volume satisfy the demand of one dimensional since the flow is only function of x coordinate. For case one the right hand side term in equation (5.16) is

$$\rho \frac{d}{dt} \int_0^L h dx = \rho L \frac{dh}{dt}$$

Substituting into equation equation (5.16) is

$$\rho L \frac{dh}{dt} = \overbrace{b_1 h}^{\text{flow out}} - \overbrace{m_i}^{\text{flow in}}$$

solution is

$$h = \overbrace{\frac{m_i}{b_1} e^{-\frac{b_1 t}{\rho L}}}^{\text{homogeneous solution}} + \overbrace{c_1 e^{\frac{b_1 t}{\rho L}}}^{\text{private solution}}$$

The solution has the homogeneous solution (solution without the m_i) and the solution of the m_i part. The solution can rearranged to a new form (a discussion why this form is preferred will be provided in dimensional chapter).

$$\frac{h b_1}{m_1} = e^{-\frac{b_1 t}{\rho L}} + c e^{\frac{b_1 t}{\rho L}}$$

With the initial condition that at $h(t=0) = h_0$ the constant coefficient can be found as

$$\frac{h_0 b_1}{m_1} = 1 - c \implies c = 1 - \frac{h_0 b_1}{m_i}$$

which the solution is

$$\frac{h b_1}{m_1} = e^{-\frac{b_1 t}{\rho L}} + \left[1 - \frac{h_0 b_1}{m_i} \right] e^{\frac{b_1 t}{\rho L}}$$

It can be observed that if $1 = \frac{h_0 b_1}{m_i}$ is the critical point of this solution. If the term $\frac{h_0 b_1}{m_i}$ is larger than one then the solution reduced to a negative number. However, negative number for height is not possible and the height solution approach zero. If the reverse case appeared, the height will increase. Essentially, the critical ratio state if the flow in is larger or lower than the flow out determine the condition of the height.

For second case, the governing equation (5.16) is

$$\rho L \frac{dh}{dt} = \overbrace{b \sqrt{h}}^{\text{flow out}} - \overbrace{m_i}^{\text{flow in}}$$

with the general solution of

$$\ln \left[\left(\frac{\sqrt{h} b}{m_i} - 1 \right) \frac{m_i}{\rho L} \right] + \frac{\sqrt{h} b}{m_i} - 1 = (t + c) \frac{\sqrt{h} b}{2 \rho L}$$

The constant is obtained when the initial condition that at $h(t = 0) = h_0$ and it left as exercise for the reader.

End Solution

5.4 Reynolds Transport Theorem

It can be noticed that the same derivations carried for the density can be carried for other intensive properties such as specific entropy, specific enthalpy. Suppose that g is intensive property (which can be a scalar or a vector) undergoes change with time. The change of accumulative property will be then

$$\frac{D}{Dt} \int_{sys} f \rho dV = \frac{d}{dt} \int_{c.v.} f \rho dV + \int_{c.v.} f \rho U_{rn} dA \quad (5.21)$$

This theorem named after Reynolds, Osborne, (1842-1912) which is actually a three dimensional generalization of Leibniz integral rule¹. To make the previous derivation clearer, the Reynolds Transport Theorem will be reproofed and discussed. The ideas are the similar but extended some what.

Leibniz integral rule² is an one dimensional and it is defined as

$$\frac{d}{dy} \int_{x_1(y)}^{x_2(y)} f(x, y) dx = \int_{x_1(y)}^{x_2(y)} \frac{\partial f}{\partial y} dx + f(x_2, y) \frac{dx_2}{dy} - f(x_1, y) \frac{dx_1}{dy} \quad (5.22)$$

Initially, a proof will be provided and the physical meaning will be explained. Assume that there is a function that satisfy the following

$$G(x, y) = \int^x f(\alpha, y) d\alpha \quad (5.23)$$

Notice that lower boundary of the integral is missing and is only the upper limit of the function is present³. For its derivative of equation (5.23) is

$$f(x, y) = \frac{\partial G}{\partial x} \quad (5.24)$$

differentiating (chain rule $duv = u dv + v du$) by part of left hand side of the Leibniz integral rule (it can be shown which are identical) is

$$\frac{d [G(x_2, y) - G(x_1, y)]}{dy} = \overbrace{\frac{\partial G}{\partial x_2} \frac{dx_2}{dy}}^1 + \overbrace{\frac{\partial G}{\partial y}(x_2, y)}^2 - \overbrace{\frac{\partial G}{\partial x_1} \frac{dx_1}{dy}}^3 - \overbrace{\frac{\partial G}{\partial y}(x_1, y)}^4 \quad (5.25)$$

¹These papers can be read on-line at <http://www.archive.org/details/papersonmechanic01reynrich>.

²This material is not necessarily but is added her for completeness. This author find material just given so no questions will be asked.

³There was a suggestion to insert arbitrary constant which will be canceled and will a provide rigorous proof. This is engineering book and thus, the exact mathematical proof is not the concern here. Nevertheless, if there will be a demand for such, it will be provided.

The terms 2 and 4 in equation (5.25) are actually (the x_2 is treated as a different variable)

$$\int_{x_1(y)}^{x_2(y)} \frac{\partial f(x, y)}{\partial y} dx \quad (5.26)$$

The first term (1) in equation (5.25) is

$$\frac{\partial G}{\partial x_2} \frac{dx_2}{dy} = f(x_2, y) \frac{dx_2}{dy} \quad (5.27)$$

The same can be said for the third term (3). Thus this explanation is a proof the Leibniz rule.

The above "proof" is mathematical in nature and physical explanation is also provided. Suppose that a fluid is flowing in a conduit. The intensive property, f is investigated or the accumulative property, F . The interesting information that commonly needed is the change of the accumulative property, F , with time. The change with time is

$$\frac{DF}{Dt} = \frac{D}{Dt} \int_{sys} \rho f dV \quad (5.28)$$

For one dimensional situation the change with time is

$$\frac{DF}{Dt} = \frac{D}{Dt} \int_{sys} \rho f A(x) dx \quad (5.29)$$

If two limiting points (for the one dimensional) are moving with a different coordinate system, the mass will be different and it will not be a system. This limiting condition is the control volume for which some of the mass will leave or enter. Since the change is very short (differential), the flow in (or out) will be the velocity of fluid minus the boundary at x_1 , $U_{rn} = U_1 - U_b$. The same can be said for the other side. The accumulative flow of the property in, F , is then

$$F_{in} = \overbrace{f_1 \rho}^{F_1} \overbrace{U_{rn}}^{\frac{dx_1}{dt}} \quad (5.30)$$

The accumulative flow of the property out, F , is then

$$F_{out} = \overbrace{f_2 \rho}^{F_2} \overbrace{U_{rn}}^{\frac{dx_2}{dt}} \quad (5.31)$$

The change with time of the accumulative property, F , between the boundaries is

$$\frac{d}{dt} \int_{c.v.} \rho(x) f A(x) dA \quad (5.32)$$

When put together it brings back the Leibniz integral rule. Since the time variable, t , is arbitrary and it can be replaced by any letter. The above discussion is one of the physical meaning the Leibniz rule.

Reynolds Transport theorem is a generalization of the Leibniz rule and thus the same arguments are used. The only difference is that the velocity has three components and only the perpendicular component enters into the calculations.

$$\frac{D}{DT} \int_{sys} f \rho dV = \frac{d}{dt} \int_{c.v} f \rho dV + \int_{S_{c.v.}} f \rho U_{rn} dA \quad (5.33)$$

5.5 Examples For Mass Conservation

Several examples are provided to illustrate the topic.

Example 5.5:

Liquid enters a circular pipe with a linear velocity profile as a function of the radius with maximum velocity of U_{max} . After magical mixing, the velocity became uniform. Write the equation which describes the velocity at the entrance. What is the magical averaged velocity at the exit? Assume no-slip condition.

SOLUTION

The velocity profile is linear with radius. Additionally, later a discussion on relationship between velocity at interface to solid also referred as the (no) slip condition will be provided. This assumption is good for most cases with very few exceptions. It will be assumed that the velocity at the interface is zero. Thus, the boundary condition is $U(r = R) = 0$ and $U(r = 0) = U_{max}$. Therefore the velocity profile is

$$U(r) = U_{max} \left(1 - \frac{r}{R}\right)$$

Where R is radius and r is the working radius (for the integration). The magical averaged velocity is obtained using the equation (5.13). For which

$$\int_0^R U_{max} \left(1 - \frac{r}{R}\right) 2\pi r dr = U_{ave} \pi R^2 \quad (5.V.a)$$

The integration of the equation (5.V.a) is

$$U_{max} \pi \frac{R^2}{6} = U_{ave} \pi R^2 \quad (5.V.b)$$

The solution of equation (b) results in average velocity as

$$U_{ave} = \frac{U_{max}}{6} \quad (5.V.c)$$

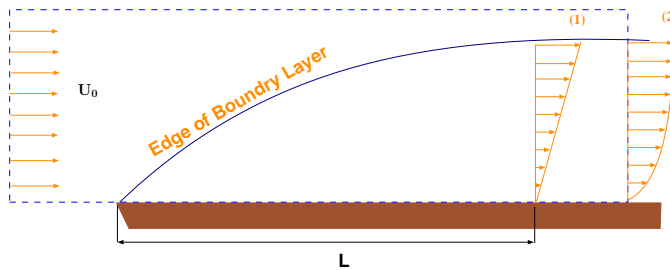


Fig. -5.7. Boundary Layer control mass.

Example 5.6:

Experiments have shown that a layer of liquid that attached itself to the surface and it is referred to as boundary layer. The assumption is that fluid attaches itself to surface. The slowed liquid is slowing the layer above it. The boundary layer is growing with \$x\$ because the boundary effect is penetrating further into fluid. A common boundary layer analysis uses the Reynolds transform theorem. In this case, calculate the relationship of the mass transfer across the control volume. For simplicity assume slowed fluid has a linear velocity profile. Then assume parabolic velocity profile as

$$U_x(y) = 2U_0 \left[\frac{y}{\delta} + \frac{1}{2} \left(\frac{y}{\delta} \right)^2 \right]$$

and calculate the mass transfer across the control volume. Compare the two different velocity profiles affecting on the mass transfer.

SOLUTION

Assuming the velocity profile is linear thus, (to satisfy the boundary condition) it will be

$$U_x(y) = \frac{U_0 y}{\delta}$$

The chosen control volume is rectangular of \$L \times \delta\$. Where \$\delta\$ is the height of the boundary layer at exit point of the flow as shown in Figure 5.7. The control volume has three surfaces that mass can cross, the left, right, and upper. No mass can cross the lower surface (solid boundary). The situation is steady state and thus using equation (5.13) results in

$$\overbrace{\int_0^\delta U_0 dy}^{\text{in}} - \overbrace{\int_0^\delta \frac{U_0 y}{\delta} dy}^{\text{out}} = \overbrace{\int_0^L U_x dx}^{\text{y direction}}$$

It can be noticed that the convention used in this chapter of “in” as negative is not “followed.” The integral simply multiply by negative one. The above integrals on the

right hand side can be combined as

$$\int_0^\delta U_0 \left(1 - \frac{y}{\delta}\right) dy = \int_0^L U x dx$$

the integration results in

$$\frac{U_0 \delta}{2} = \int_0^L U x dx$$

or for parabolic profile

$$\int_0^\delta U_0 dy - \int_0^\delta U_0 \left[\frac{y}{\delta} + \left(\frac{y}{\delta}\right)^2 \right] dy = \int_0^L U x dx$$

or

$$\int_0^\delta U_0 \left[1 - \frac{y}{\delta} - \left(\frac{y}{\delta}\right)^2 \right] dy = U_0$$

the integration results in

$$\frac{U_0 \delta}{2} = \int_0^L U x dx$$

End Solution

Example 5.7:

Air flows into a jet engine at 5 kg/sec while fuel flow into the jet is at 0.1 kg/sec. The burned gases leaves at the exhaust which has cross area 0.1 m² with velocity of 500 m/sec. What is the density of the gases at the exhaust?

SOLUTION

The mass conservation equation (5.13) is used. Thus, the flow out is (5 + 0.1) 5.1 kg/sec The density is

$$\rho = \frac{\dot{m}}{AU} = \frac{5.1 \text{ kg/sec}}{0.01 \text{ m}^2 \cdot 500 \text{ m/sec}} = 1.02 \text{ kg/m}^3$$

End Solution

The mass (volume) flow rate is given by direct quantity like $x \text{ kg/sec}$. However sometime, the mass (or the volume) is given by indirect quantity such as the effect of flow. The next example deal with such reversed mass flow rate.

Example 5.8:

The tank is filled by two valves which one filled tank in 3 hours and the second by 6 hours. The tank also has three emptying valves of 5 hours, 7 hours, and 8 hours. The tank is 3/4 fulls, calculate the time for tank reach empty or full state when all the valves are open. Is there a combination of valves that make the tank at steady state?

SOLUTION

Easier measurement of valve flow rate can be expressed as fraction of the tank per hour. For example valve of 3 hours can be converted to $1/3$ tank per hour. Thus, mass flow rate in is

$$\dot{m}_{in} = 1/3 + 1/6 = 1/2 \text{ tank/hour}$$

The mass flow rate out is

$$\dot{m}_{out} = 1/5 + 1/7 + 1/8 = \frac{131}{280}$$

Thus, if all the valves are open the tank will be filled. The time to completely filled the tank is

$$\frac{1}{\frac{1}{2} - \frac{131}{280}} = \frac{70}{159} \text{ hour}$$

The rest is under construction.

End Solution

Example 5.9:

Inflated cylinder is supplied in its center with constant mass flow. Assume that the gas mass is supplied in uniform way of m_i [kg/m/sec]. Assume that the cylinder inflated uniformly and pressure inside the cylinder is uniform. The gas inside the cylinder obeys the ideal gas law. The pressure inside the cylinder is linearly proportional to the volume. For simplicity, assume that the process is isothermal. Calculate the cylinder boundaries velocity.

SOLUTION

The applicable equation is

$$\overbrace{\int_{V_{c.v.}} \frac{d\rho}{dt} dV}^{\text{increase pressure}} + \overbrace{\int_{S_{c.v.}} \rho U_b dV}^{\text{boundary velocity}} = \overbrace{\int_{S_{c.v.}} \rho U_{rn} dA}^{\text{in or out flow rate}}$$

Every term in the above equation is analyzed but first the equation of state and volume to pressure relationship have to be provided.

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

and relationship between the volume and pressure is

$$P = f \pi R_c^2$$

Where R_c is the instantaneous cylinder radius. Combining the above two equations results in

$$\rho = \frac{f \pi R_c^2}{RT}$$

Where f is a coefficient with the right dimension. It also can be noticed that boundary velocity is related to the radius in the following form

$$U_b = \frac{dR_c}{dt}$$

The first term requires to find the derivative of density with respect to time which is

$$\frac{d\rho}{dt} = \frac{d}{dt} \left(\frac{f \pi R_c^2}{RT} \right) = \frac{2 f \pi R_c}{RT} \overbrace{\frac{dR_c}{dt}}^{U_b}$$

Thus the first term is

$$\int_{V_{c.v.}} \frac{d\rho}{dt} \overbrace{dV}^{2 \pi R_c} = \int_{V_{c.v.}} \frac{2 f \pi R_c}{RT} U_b \overbrace{dV}^{2 \pi R_c dR_c} = \frac{4 f \pi^2 R_c^3}{3 RT} U_b$$

The integral can be carried when U_b is independent of the R_c ⁴ The second term is

$$\int_{S_{c.v.}} \rho U_b dA = \overbrace{\frac{f \pi R_c^2}{RT}}^{\rho} U_b \overbrace{2 \pi R_c}^A = \left(\frac{f \pi^3 R_c^2}{RT} \right) U_b$$

substituting in the governing equation obtained the form of

$$\frac{f \pi^2 R_c^3}{RT} U_b + \frac{4 f \pi^2 R_c^3}{3 RT} U_b = m_i$$

The boundary velocity is then

$$U_b = \frac{m_i}{\frac{7 f \pi^2 R_c^3}{3 RT}} G = \frac{3 m_i RT}{7 f \pi^2 R_c^3}$$

End Solution

Example 5.10:

A balloon is attached to a rigid supply and is supplied by a constant mass rate, m_i . Assume that gas obeys the ideal gas law. Assume that balloon volume is a linear function of the pressure inside the balloon such as $P = f_v V$. Where f_v is a coefficient describing the balloon physical characters. Calculate the velocity of the balloon boundaries under the assumption of isothermal process.

⁴The proof of this idea is based on the chain differentiation similar to Leibniz rule. When the derivative of the second part is $dU_b/dR_c = 0$.

SOLUTION

The question is more complicated than Example 5.10. The ideal gas law is

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

The relationship between the pressure and volume is

$$P = f_v V = \frac{4 f_v \pi R_b^3}{3}$$

The combining of the ideal gas law with the relationship between the pressure and volume results

$$\rho = \frac{4 f_v \pi R_b^3}{3 R T}$$

The applicable equation is

$$\int_{V_{c.v.}} \frac{d\rho}{dt} dV + \int_{S_{c.v.}} \rho (U_c \hat{x} + U_b \hat{r}) dA = \int_{S_{c.v.}} \rho U_{rn} dA$$

The right hand side of the above equation is

$$\int_{S_{c.v.}} \rho U_{rn} dA = m_i$$

The density change is

$$\frac{d\rho}{dt} = \frac{12 f_v \pi R_b^2}{RT} \overbrace{\frac{dR_b}{dt}}^{U_b}$$

The first term is

$$\int_0^{R_b} \overbrace{\frac{12 f_v \pi R_b^2}{RT}}^{\neq f(r)} U_b \overbrace{4 \pi r^2 dr}^{dV} = \frac{16 f_v \pi^2 R_b^5}{3 R T} U_b$$

The second term is

$$\int_A \frac{4 f_v \pi R_b^3}{3 R T} U_b dA = \frac{4 f_v \pi R_b^3}{3 R T} U_b \overbrace{4 \pi R_b^2}^A = \frac{8 f_v \pi^2 R_b^5}{3 R T} U_b$$

Substituting the two equations of the applicable equation results

$$U_b = \frac{1}{8} \frac{m_i R T}{f_v \pi^2 R_b^5}$$

Notice that first term is used to increase the pressure and second the change of the boundary.

Open Question: Answer must be received by April 15, 2010

The best solution of the following question will win 18 U.S. dollars and your name will be associated with the solution in this book.

Example 5.11:

Solve example 5.10 under the assumption that the process is isentropic. Also assume that the relationship between the pressure and the volume is $P = f_v V^2$. What are the units of the coefficient f_v in this problem? What are the units of the coefficient in the previous problem?

5.6 The Details Picture – Velocity Area Relationship

The integral approach is intended to deal with the “big” picture. Indeed the method is used in this part of the book for this purpose. However, there is very little written about the usability of this approach to provide way to calculate the average quantities in the control system. Sometime is desirable to find the averaged velocity or velocity distribution inside a control volume. There is no general way to provide these quantities. Therefore an example will be provided to demonstrate the use of this approach.

Consider a container filled with liquid on which one exit opened and the liquid flows out as shown in Figure 5.8. The velocity has three components in each of the coordinates under the assume that flow is uniform and the surface is straight⁵. The integral approached is used to calculate the averaged velocity of each to the components. To relate the velocity in the z direction with the flow rate out or the exit the velocity mass balance is constructed. A similar control volume construction to find the velocity of the boundary velocity (height) can be carried out. The control volume is bounded by the container wall including the exit of the flow. The upper boundary is surface parallel to upper surface but at Z distance from the bottom. The mass balance reads

$$\int_V \frac{d\rho}{dt} dV + \int_A U_{bn} \rho dA + \int_A U_{rn} \rho dA = 0 \quad (5.34)$$

⁵The liquid surface is not straight for this kind of problem. However, under certain conditions it is reasonable to assume straight surface which have been done for this problem.

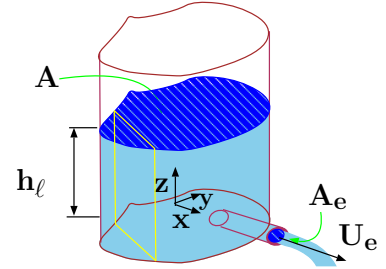


Fig. -5.8. Control volume usage to calculate local averaged velocity in Three coordinates.

For constant density (conservation of volume) equation⁶ and ($h > z$) reduces to

$$\int_A U_{rn} \rho dA = 0 \tag{5.35}$$

In the container case for uniform velocity equation 5.35 becomes

$$U_z A = U_e A_e \implies U_z = -\frac{A_e}{A} U_e \tag{5.36}$$

It can be noticed that the boundary is not moving and the mass inside does not change inside this control volume. The velocity U_z is the averaged velocity downward.

The x component of velocity is obtained by using a different control volume. The control volume is shown in Figure 5.9. The boundary are the container far from the flow exit with blue line projection into page (area) shown in the Figure 5.9. The mass conservation for constant density of this control volume is

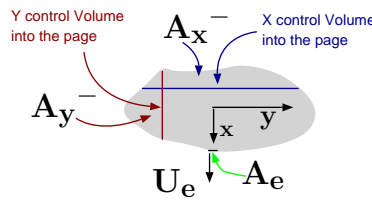


Fig. -5.9. Control volume and system before and after motion.

$$-\int_A U_{bn} \rho dA + \int_A U_{rn} \rho dA = 0 \tag{5.37}$$

Usage of control volume not included in the previous analysis provides the velocity at the upper boundary which is the same as the velocity at y direction. Substituting into (5.37) results in

$$\int_{A_x^-} \frac{A_e}{A} U_e \rho dA + \int_{A_{yz}} U_x \rho dA = 0 \tag{5.38}$$

Where A_x^- is the area shown the Figure under this label. The area A_{yz} referred to area into the page in Figure 5.9 under the blue line. Because averaged velocities and constant density are used transformed equation (5.38) into

$$\frac{A_e}{A} A_x^- U_e + U_x \overbrace{Y(x) h}^{A_{yz}} = 0 \tag{5.39}$$

Where $Y(x)$ is the length of the (blue) line of the boundary. It can be notice that the velocity, U_x is generally increasing with x because A_x^- increase with x .

The calculations for the y directions are similar to the one done for x direction. The only difference is that the velocity has two different directions. One zone is right to the exit with flow to the left and one zone to left with averaged velocity to right. If the volumes on the left and the right are symmetrical the averaged velocity will be zero.

⁶The point where ($z = h$) the boundary term is substituted the flow in term.

Example 5.12:

Calculate the velocity, U_x for a cross section of circular shape (cylinder).

SOLUTION

The relationship for this geometry needed to be expressed. The length of the line $Y(x)$ is

$$Y(x) = 2r \sqrt{1 - \left(1 - \frac{x}{r}\right)^2} \quad (5.XII.a)$$

This relationship also can be expressed in the term of α as

$$Y(x) = 2r \sin \alpha \quad (5.XII.b)$$

Since this expression is simpler it will be adapted. When the relationship between radius angle and x are

$$x = r(1 - \sin \alpha) \quad (5.XII.c)$$

The area A_x^- is expressed in term of α as

$$A_x^- = \left(\alpha - \frac{1}{2} \sin(2\alpha)\right) r^2 \quad (5.XII.d)$$

Thus the velocity, U_x is

$$\frac{A_e}{A} \left(\alpha - \frac{1}{2} \sin(2\alpha)\right) r^2 U_e + U_x 2r \sin \alpha h = 0 \quad (5.XII.e)$$

$$U_x = \frac{A_e r}{A h} \frac{\left(\alpha - \frac{1}{2} \sin(2\alpha)\right)}{\sin \alpha} U_e \quad (5.XII.f)$$

Averaged velocity is defined as

$$\bar{U}_x = \frac{1}{S} \int_S U dS \quad (5.XII.g)$$

Where here S represent some length. The same way it can be represented for angle calculations. The value dS is $r \cos \alpha$. Integrating the velocity for the entire container and dividing by the angle, α provides the averaged velocity.

$$\bar{U}_x = \frac{1}{2r} \int_0^\pi \frac{A_e r}{A h} \frac{\left(\alpha - \frac{1}{2} \sin(2\alpha)\right)}{\tan \alpha} U_e r d\alpha \quad (5.XII.h)$$

which results in

$$\bar{U}_x = \frac{(\pi - 1)}{4} \frac{A_e r}{A h} U_e \quad (5.XII.i)$$

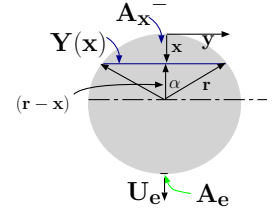


Fig. -5.10. Circular cross section for Finding U_x and various cross sections.

End Solution

Example 5.13:

Calculate the velocity, U_y for a cross section of circular shape (cylinder). What is the averaged velocity if only half section is used. State your assumptions and how it similar to the previous example.

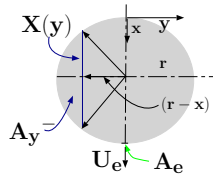


Fig. -5.11. y velocity for a circular shape

SOLUTION

The flow out in the x direction is zero because symmetrical reasons. That is the flow field is a mirror images. Thus, every point has different velocity with the same value in the opposite direction.

The flow in half of the cylinder either the right or the left has non zero averaged velocity. The calculations are similar to those in the previous to example 5.12. The main concept that must be recognized is the half of the flow must have come from one side and the other come from the other side. Thus, equation (5.39) modified to be

$$\frac{A_e}{A} A_x - U_e + U_x \overbrace{Y(x)}^{A_{yz}} h = 0 \tag{5.40}$$

The integral is the same as before but the upper limit is only to $\pi/2$

$$\bar{U}_x = \frac{1}{2r} \int_0^{\pi/2} \frac{A_e r}{A h} \frac{(\alpha - \frac{1}{2} \sin(2\alpha))}{\tan \alpha} U_e r d\alpha \tag{5.XIII.a}$$

which results in

$$\bar{U}_x = \frac{(\pi - 2)}{8} \frac{A_e r}{A h} U_e \tag{5.XIII.b}$$

End Solution

5.7 More Examples for Mass Conservation

Typical question about the relative velocity that appeared in many fluid mechanics exams is the following.

Example 5.14:

A boat travels at speed of 10m/sec upstream in a river that flows at a speed of 5m/s. The inboard engine uses a pump to suck

in water at the front $A_{in} = 0.2 \text{ m}^2$ and eject it through the back of the boat with exist area of $A_{out} = 0.05 \text{ m}^2$. The water absolute velocity leaving the back is 50 m/sec , what are the relative velocities entering and leaving the boat and the pumping rate?

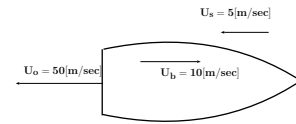


Fig. -5.12. Schematic of the boat for example 5.14

SOLUTION

The boat is assumed (implicitly is stated) to be steady state and the density is constant. However, the calculation have to be made in the frame of reference moving with the boat. The relative jet discharge velocity is

$$U_{r_{out}} = 50 - (10 + 5) = 35 \text{ [m/sec]}$$

The volume flow rate is then

$$Q_{out} = A_{out} U_{r_{out}} = 35 \times 0.05 = 1.75 \text{ m}^3 \text{ /sec}$$

The flow rate at entrance is the same as the exit thus,

$$U_{r_{in}} = \frac{A_{out}}{A_{in}} U_{r_{out}} = \frac{0.05}{0.2} 35 = 8.75 \text{ m/sec}$$

CHAPTER 6

Momentum Conservation for Control Volume

6.1 Momentum Governing Equation

6.1.1 Introduction to Continuous

In the previous chapter, the Reynolds Transport Theorem (RTT) was applied to mass conservation. Mass is a scalar (quantity without magnitude). This chapter deals with momentum conservation which is a vector. The Reynolds Transport Theorem (RTT) is applicable to any quantity and the discussion here will deal with forces that acting on the control volume. Newton's second law for single body is as the following

$$\mathbf{F} = \frac{d(m\mathbf{U})}{dt} \quad (6.1)$$

It can be noticed that bold notation for the velocity is \mathbf{U} (and not U) to represent that the velocity has a direction. For several bodies (n), Newton's law becomes

$$\sum_{i=1}^n \mathbf{F}_i = \sum_{i=1}^n \frac{d(m\mathbf{U})_i}{dt} \quad (6.2)$$

The fluid can be broken into infinitesimal elements which turn the above equation (6.2) into a continuous form of small bodies which results in

$$\sum_{i=1}^n \mathbf{F}_i = \frac{D}{Dt} \int_{sys} \mathbf{U} \overbrace{\rho dV}^{\text{element mass}} \quad (6.3)$$

Note that the notation D/Dt is used and not d/dt to signify that it referred to a derivative of the system. The Reynold's Transport Theorem (RTT) has to be used on the right hand side.

6.1.2 External Forces

First, the terms on the left hand side, or the forces, have to be discussed. The forces, excluding the external forces, are the body forces, and the surface forces as the following

$$\mathbf{F}_{total} = \mathbf{F}_b + \mathbf{F}_s \quad (6.4)$$

In this book (at least in this discussion), the main body force is the gravity. The gravity acts on all the system elements. The total gravity force is

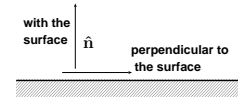
$$\sum \mathbf{F}_b = \int_{sys} \mathbf{g} \overbrace{\rho dV}^{element\ mass} \quad (6.5)$$

which acts through the mass center towards the center of Earth. After infinitesimal time the gravity force acting on the system is the same for control volume, hence,

$$\int_{sys} \mathbf{g} \rho dV = \int_{cv} \mathbf{g} \rho dV \quad (6.6)$$

The integral yields a force trough the center mass which has to be found separately.

In this chapter, the surface forces are divided into two categories: one perpendicular to the surface and one with the surface direction (in the surface plain see Figure 6.1.). Thus, it can be written as



$$\sum \mathbf{F}_s = \int_{c.v.} \mathbf{S}_n dA + \int_{c.v.} \boldsymbol{\tau} dA \quad (6.7)$$

Fig. -6.1. The explain for the direction relative to surface perpendicular and with the surface.

Where the surface "force", \mathbf{S}_n , is in the surface direction, and $\boldsymbol{\tau}$ are the shear stresses. The surface "force", \mathbf{S}_n , is made out of two components, one due to viscosity (solid body) and two consequence of the fluid pressure. Here for simplicity, only the pressure component is used which is reasonable for most situations. Thus,

$$\mathbf{S}_n = -P \hat{n} + \overbrace{\mathbf{S}_\nu}^{\sim 0} \quad (6.8)$$

Where \mathbf{S}_ν is perpendicular stress due to viscosity. Again, \hat{n} is an unit vector outward of element area and the negative sign is applied so that the resulting force acts on the body.

6.1.3 The Momentum Governing Equation

The right hand side, according Reynolds Transport Theorem (RTT), is

$$\frac{D}{Dt} \int_{sys} \rho \mathbf{U} dV = \frac{t}{dt} \int_{c.v.} \rho \mathbf{U} dV + \int_{c.v.} \rho \mathbf{U} \mathbf{U}_{rn} dA \quad (6.9)$$

The liquid velocity, \mathbf{U} , is measured in the frame of reference and \mathbf{U}_{rn} is the liquid relative velocity to boundary of the control volume measured in the same frame of reference.

Thus, the general form of the momentum equation without the external forces is

$$\int_{c.v.} \mathbf{g} \rho dV - \int_{c.v.} \mathbf{P} dA + \int_{c.v.} \boldsymbol{\tau} \cdot d\mathbf{A} = \frac{t}{dt} \int_{c.v.} \rho \mathbf{U} dV + \int_{c.v.} \rho \mathbf{U} \mathbf{U}_{rn} dV \quad (6.10)$$

With external forces equation (6.10) is transformed to

$$\sum \mathbf{F}_{ext} + \int_{c.v.} \mathbf{g} \rho dV - \int_{c.v.} \mathbf{P} \cdot d\mathbf{A} + \int_{c.v.} \boldsymbol{\tau} \cdot d\mathbf{A} = \frac{t}{dt} \int_{c.v.} \rho \mathbf{U} dV + \int_{c.v.} \rho \mathbf{U} \mathbf{U}_{rn} dV \quad (6.11)$$

The external forces, F_{ext} , are the forces resulting from support of the control volume by non-fluid elements. These external forces are commonly associated with pipe, ducts, supporting solid structures, friction (non-fluid), etc.

Equation (6.11) is a vector equation which can be broken into its three components. In Cartesian coordinate, for example in the x coordinate, the components are

$$\sum F_x + \int_{c.v.} (\mathbf{g} \cdot \hat{i}) \rho dV - \int_{c.v.} \mathbf{P} \cos \theta_x dA + \int_{c.v.} \boldsymbol{\tau}_x \cdot d\mathbf{A} = \frac{t}{dt} \int_{c.v.} \rho U_x dV + \int_{c.v.} \rho U_x \cdot \mathbf{U}_{rn} dA \quad (6.12)$$

where θ_x is the angle between \hat{n} and \hat{i} or $(\hat{n} \cdot \hat{i})$.

6.1.4 Momentum Equation in Acceleration System

For accelerate system, the right hand side has to include the following acceleration

$$\mathbf{a}_{acc} = \boldsymbol{\omega} \times (\mathbf{r} \times \boldsymbol{\omega}) + 2\mathbf{U} \times \boldsymbol{\omega} + \mathbf{r} \times \dot{\boldsymbol{\omega}} - \mathbf{a}_0 \quad (6.13)$$

Where \mathbf{r} is the distance from the center of the frame of reference and the add force is

$$\mathbf{F}_{add} = \int_{V_{c.v.}} \mathbf{a}_{acc} \rho dV \quad (6.14)$$

Integral of Uniform Pressure on Body

In this kind of calculations, it common to obtain a situation where one of the term will be an integral of the pressure over the body surface. This situation is a similar idea that was shown in Section 4.6. In this case the resulting force due to the pressure is zero to all directions.

6.1.5 Momentum For Steady State and Uniform Flow

The momentum equation can be simplified for the steady state condition as it was shown in example 6.3. The unsteady term (where the time derivative) is zero.

$$\sum \mathbf{F}_{ext} + \int_{c.v.} \mathbf{g} \rho dV - \int_{c.v.} \mathbf{P} dA + \int_{c.v.} \boldsymbol{\tau} dA = \int_{c.v.} \rho \mathbf{U} U_{rn} dA \quad (6.15)$$

6.1.5.1 Momentum for For Constant Pressure and Frictionless Flow

Another important sub category of simplification deals with flow under approximation of the frictionless flow and uniform pressure. This kind of situations arise when friction (forces) is small compared to kinetic momentum change. Additionally, in these situations, flow is exposed to the atmosphere and thus (almost) uniform pressure surrounding the control volume. In this situation, the mass flow rate in and out are equal. Thus, equation (6.15) is further reduced to

$$\mathbf{F} = \int_{out} \rho \mathbf{U} \overbrace{(\mathbf{U} \cdot \hat{n})}^{U_{rn}} dA - \int_{in} \rho \mathbf{U} \overbrace{(\mathbf{U} \cdot \hat{n})}^{U_{rn}} dA \quad (6.16)$$

In situations where the velocity is provided and known (remember that density is constant) the integral can be replaced by

$$\mathbf{F} = \dot{m} \overline{\mathbf{U}}_o - \dot{m} \overline{\mathbf{U}}_i \quad (6.17)$$

The average velocity is related to the velocity profile by the following integral

$$\overline{U}^2 = \frac{1}{A} \int_A [U(r)]^2 dA \quad (6.18)$$

Equation (6.18) is applicable to any velocity profile and any geometrical shape.

Example 6.1:

Calculate the average velocity for given the parabolic velocity profile for a circular pipe.

SOLUTION

The velocity profile is

$$U\left(\frac{r}{R}\right) = U_{max} \left[1 - \left(\frac{r}{R}\right)^2\right] \tag{6.1.a}$$

Substituting equation (6.1.a) into equation (6.18)

$$\bar{U}^2 = \frac{1}{2\pi R^2} \int_0^R [U(r)]^2 2\pi r dr \tag{6.1.b}$$

results in

$$\bar{U}^2 = (U_{max})^2 \int_0^1 (1 - \bar{r}^2)^2 \bar{r} d\bar{r} = \frac{1}{6} (U_{max})^2 \tag{6.1.c}$$

Thus,

$$\bar{U} = \frac{U_{max}}{\sqrt{6}}$$

End Solution

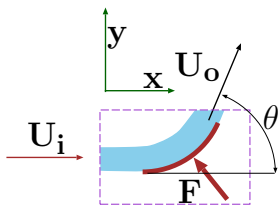


Fig a. Schematics of area impinged by a jet for example 6.2.

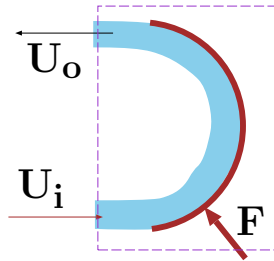


Fig b. Schematics of maximum angle for impinged by a jet.

Fig. -6.2. Schematics of area impinged by a jet and angle effects.

Example 6.2:

A jet is impinging on a stationary surface by changing only the jet direction (see Figure 6.2). Neglect the friction, calculate the force and the angle which the support has to apply to keep the system in equilibrium. What is the angle for which maximum force will be created?

SOLUTION

Equation (6.11) can be reduced, because it is a steady state, to

$$\mathbf{F} = \int_{out} \rho \mathbf{U} \overbrace{(\mathbf{U} \cdot \hat{n})}^{U_{rn}} dA - \int_{in} \rho \mathbf{U} \overbrace{(\mathbf{U} \cdot \hat{n})}^{U_{rn}} dA = \dot{m} \mathbf{U}_o - \dot{m} \mathbf{U}_i \tag{6.11.a}$$

It can be noticed that even though the velocity change direction, the mass flow rate remains constant. Equation (6.11.a) can be explicitly written for the two coordinates. The equation for the x coordinate is

$$F_x = \dot{m} (\cos \theta U_o - U_i)$$

or since $U_i = U_o$

$$F_x = \dot{m} U_i (\cos \theta - 1)$$

It can be observed that the maximum force, F_x occurs when $\cos \theta = \pi$. It can be proven by setting $dF_x/d\theta = 0$ which yields $\theta = 0$ a minimum and the previous solution. Hence

$$F_x|_{max} = -2 \dot{m} U_i$$

and the force in the y direction is

$$F_y = \dot{m} U_i \sin \theta$$

the combined forces are

$$F_{total} = \sqrt{F_x^2 + F_y^2} = \dot{m} U_i \sqrt{(\cos \theta - 1)^2 + \sin^2 \theta}$$

Which results in

$$F_{total} = \dot{m} U_i \sin(\theta/2)$$

with the force angle of

$$\tan \phi = \pi - \frac{F_y}{F_x} = \frac{\pi}{2} - \frac{\theta}{2}$$

For angle between $0 < \theta < \pi$ the maximum occur at $\theta = \pi$ and the minimum at $\theta \sim 0$. For small angle analysis is important in the calculations of flow around thin wings.

End Solution

Example 6.3:

Liquid flows through a symmetrical nozzle as shown in the Figure 6.3 with a mass flow rate of $0.01 [gk/sec]$. The entrance pressure is $3[Bar]$ and the entrance velocity is $5 [m/sec]$. The exit velocity is uniform but unknown. The exit pressure is $1[Bar]$. The entrance area is $0.0005[m^2]$ and the exit area is $0.0001[cm^2]$. What is the exit velocity? What is the force acting the nozzle? Assume that the density is constant $\rho = 1000[kg/m^3]$ and the volume in the nozzle is $0.0015 [m^3]$.

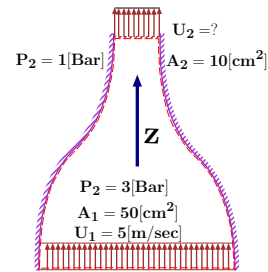


Fig. -6.3. Nozzle schematic for the discussion on the forces and for example 6.3.

SOLUTION

The chosen control volume is shown in Figure 6.3. First, the velocity has to be found. This situation is a steady state for constant density. Then

$$A_1 U_1 = A_2 U_2$$

and after rearrangement, the exit velocity is

$$U_2 = \frac{A_1}{A_2} U_1 = \frac{0.0005}{0.0001} \times 5 = 25[m/sec]$$

Equation (6.12) is applicable but should be transformed into the z direction which is

$$\begin{aligned} \sum F_z + \int_{c.v.} \mathbf{g} \cdot \hat{k} \rho dV + \int_{c.v.} \mathbf{P} \cos \theta_z dA + \int_{c.v.} \boldsymbol{\tau}_z dA = \\ \underbrace{=0}_{\frac{t}{dt} \int_{c.v.} \rho \mathbf{U}_z dV} + \int_{c.v.} \rho \mathbf{U}_z \cdot \mathbf{U}_{rn} dA \end{aligned} \quad (6.III.a)$$

The control volume does not cross any solid body (or surface) there is no external forces. Hence,

$$\begin{aligned} \underbrace{\sum F_z}_{=0} + \int_{c.v.} \mathbf{g} \cdot \hat{k} \rho dV + \underbrace{\int_{c.v.} \mathbf{P} \cos \theta_z dA}_{\text{liquid surface}} + \\ \underbrace{\int_{c.v.} \boldsymbol{\tau}_z dA}_{\text{forces on the nozzle } F_{nozzle}} = \int_{c.v.} \rho \mathbf{U}_z \cdot \mathbf{U}_{rn} dA \\ \underbrace{\int_{c.v.} \mathbf{P} \cos \theta_z dA}_{\text{solid surface}} + \int_{c.v.} \boldsymbol{\tau}_z dA = \int_{c.v.} \rho \mathbf{U}_z \cdot \mathbf{U}_{rn} dA \end{aligned} \quad (6.III.b)$$

All the forces that act on the nozzle are combined as

$$\sum F_{nozzle} + \int_{c.v.} \mathbf{g} \cdot \hat{k} \rho dV + \int_{c.v.} \mathbf{P} \cos \theta_z dA = \int_{c.v.} \rho \mathbf{U}_z \cdot \mathbf{U}_{rn} dA \quad (6.III.c)$$

The second term or the body force which acts through the center of the nozzle is

$$\mathbf{F}_b = - \int_{c.v.} \mathbf{g} \cdot \hat{n} \rho dV = -g \rho V_{nozzle}$$

Notice that in the results the gravity is not bold since only the magnitude is used. The part of the pressure which act on the nozzle in the z direction is

$$- \int_{c.v.} P dA = \int_1 P dA - \int_2 P dA = PA|_1 - PA|_2$$

The last term in equation (6.III.c) is

$$\int_{c.v.} \rho \mathbf{U}_z \cdot \mathbf{U}_{rn} dA = \int_{A_2} U_2 (U_2) dA - \int_{A_1} U_1 (U_1) dA$$

which results in

$$\int_{c.v.} \rho \mathbf{U}_z \cdot \mathbf{U}_{rn} dA = \rho (U_2^2 A_2 - U_1^2 A_1)$$

Combining all transform equation (6.III.c) into

$$F_z = -g \rho V_{nozzle} + PA|_2 - PA|_1 + \rho (U_2^2 A_2 - U_1^2 A_1) \tag{6.III.d}$$

$$F_z = 9.8 \times 1000 \times$$

End Solution

6.2 Momentum Equation Application

Momentum Equation Applied to Propellers

The propeller is a mechanical devise that is used to increase the fluid momentum. Many times it is used for propulsion purposes of airplanes, ships and there devices (thrust) as shown in Figure 6.4. The propeller can be stationary like in cooling tours, fan etc. The other common used of propeller is used mostly to move fluids as a pump.

The propeller analysis of unsteady is complicated due to the difficulty in understanding the velocity field. For a steady state the analysis is simpler and used here to provide an example of steady state. In the Figure 6.4 the fluid flows from the left to the right. Either it is assumed that some of the fluid enters into the container and fluid outside is not affected by the propeller. Or there is a line (or surface) in which the fluid outside changes only the flow direction. This surface is called slip surface. Of course it is only approximation but is provided a crude tool. Improvements can be made to this analysis. Here, this analysis is used for academic purposes.

As first approximation, the pressure around around control volume is the same. Thus, pressure drops from the calculation. The one dimensional momentum equation is reduced

$$F = \rho (U_2^2 - U_1^2) \tag{6.19}$$

Combining the control volume between points 1 and

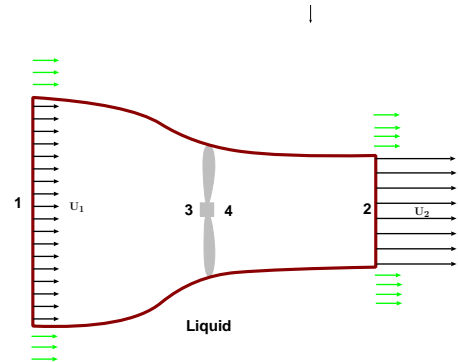


Fig. -6.4. Propeller schematic to explain the change of momentum due to velocity.

3 with (note that there are no external forces) with points 4 and 2 results in

$$\rho (U_2^2 - U_1^2) = P_4 - P_3 \quad (6.20)$$

This analysis provide way to calculate the work needed to move this propeller. Note that in this analysis it was assumed that the flow is horizontal that $z_1 = z_2$ and/or the change is insignificant.

Jet Propulsion

Jet propulsion is a mechanism in which the air planes and other devices are propelled. Essentially, the air is sucked into engine and with addition heating (burning fuel) the velocity is increased. Further increase of the exit area with the increased of the burned gases further increase the thrust. The analysis of such device is complicated and there is a whole class dedicated for such topic in many universities. Here, a very limited discussion related to the steady state is offered.

The difference between the jets propulsion and propellers is based on the energy supplied. The propellers are moved by a mechanical work which is converted to thrust. In Jet propulsion, the thermal energy is converted to thrust. Hence, this direct conversion can be, and is, in many case more efficient. Furthermore, as it will be shown in the Chapter on compressible flow it allows to achieve velocity above speed of sound, a major obstacle in the past.

The inlet area and exit area are different for most jets and if the mass of the fuel is neglected then

$$F = \rho (A_2 U_2^2 - A_1 U_1^2) \quad (6.21)$$

An academic example to demonstrate how a steady state calculations are done for a moving control volume. Notice that

Example 6.4:

A sled toy shown in Figure ?? is pushed by liquid jet. Calculate the friction force on the toy when the toy is at steady state with velocity, U_0 . Assume that the jet is horizontal and the reflecting jet is vertical. The velocity of the jet is uniform. Neglect the friction between the liquid (jet) and the toy and air and toy. Calculate the absolute velocity of the jet exit. Assume that the friction between the toy and surface (ground) is relative to the vertical force. The dynamics friction is μ_d .

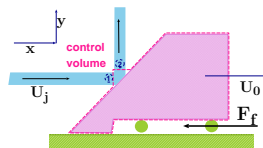


Fig. -6.5. Toy Sled pushed by the liquid jet in a steady state for example 6.4.

SOLUTION

The chosen control volume is attached to the toy and thus steady state is obtained. The frame of reference is moving with the toy velocity, \mathbf{U}_0 . The applicable mass conservation equation for steady state is

$$A_1 U_1 = A_2 U_2$$

The momentum equation in the x direction is

$$\mathbf{F}_f + \int_{c.v.} \mathbf{g} \rho dV - \int_{c.v.} \mathbf{P} dA + \int_{c.v.} \boldsymbol{\tau} dA = \int_{c.v.} \rho \mathbf{U} \mathbf{U}_{rn} dV \quad (6.IV.a)$$

The relative velocity into the control volume is

$$\mathbf{U}_{1j} = (U_j - U_0) \hat{x}$$

The relative velocity out the control volume is

$$\mathbf{U}_{2j} = (U_j - U_0) \hat{y}$$

The absolute exit velocity is

$$\mathbf{U}_2 = U_0 \hat{x} + (U_j - U_0) \hat{y}$$

For small volume, the gravity can be neglected also because this term is small compared to other terms, thus

$$\int_{c.v.} \mathbf{g} \rho dV \sim 0$$

The same can be said for air friction as

$$\int_{c.v.} \boldsymbol{\tau} dA \sim 0$$

The pressure is uniform around the control volume and thus the integral is

$$\int_{c.v.} \mathbf{P} dA = 0$$

The control volume was chosen so that the pressure calculation is minimized.

The momentum flux is

$$\int_{S_{c.v.}} \rho U_x U_i r_n dA = A \rho U_{1j}^2 \quad (6.IV.b)$$

The substituting (6.IV.b) into equation (6.IV.a) yields

$$F_f = A \rho U_{1j}^2 \quad (6.IV.c)$$

The friction can be obtained from the momentum equation in the y direction

$$m_{toy} g + A \rho U_{1j}^2 = F_{earth}$$

According to the statement of question the friction force is

$$F_f = \mu_d (m_{toy} g + A \rho U_{1j}^2)$$

The momentum in the x direction becomes

$$\mu_d (m_{toy} g + A \rho U_{1j}^2) = A \rho U_{1j}^2 = A \rho (U_j - U_0)^2$$

The toy velocity is then

$$U_0 = U_j - \sqrt{\frac{\mu_d m_{toy} g}{A \rho (1 - \mu_d)}}$$

Increase of the friction reduce the velocity. Additionally larger toy mass decrease the velocity.

End Solution

6.2.1 Momentum for Unsteady State and Uniform Flow

The main problem in solving the unsteady state situation is that the control volume is accelerating. A possible way to solve the problem is by expressing the terms in an equation (6.10). This method is cumbersome in many cases. Alternative method of solution is done by attaching the frame of reference to the accelerating body. One such example of such idea is associated with the Rocket Mechanics which is present here.

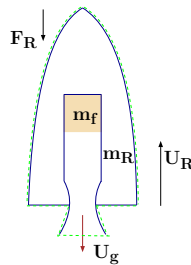


Fig. -6.6. A rocket a with moving control volume.

6.2.2 Momentum Application to Unsteady State

Rocket Mechanics

A rocket is a devise similar to jet propulsion. The difference is the fact that the oxidant is on board with the fuel. The two components are burned and the gases are ejected through a nozzle. This mechanism is useful for specific locations because it is independent of the medium though which it travels. In contrast to other mechanisms such as jet propulsion which obtain the oxygen from the medium which they travel the rockets carry the oxygen with it. The rocket is accelerating and thus the frame for reference is moving the with the rocket. The velocity of the rocket in the rocket frame of reference is zero. However, the derivative with respect to time, $dU/dt \neq 0$ is not zero. The

resistance of the medium is Denote as F_R . The momentum equation is

$$\begin{aligned} \overbrace{\int_{c.v.} \tau dA}^{F_R} + \int_{c.v.} \mathbf{g} \rho dV + \overbrace{\int_{c.v.} \mathbf{P} dA}^0 - \int_{c.v.} \rho a_0 dV = \\ \frac{d}{dt} \int_{V_{c.v.}} \rho U_y dV + \int_{c.v.} \rho U_y U_{rn} dA \end{aligned} \quad (6.22)$$

There are no external forces in this control volume. The first term is F_R The pressure term vanish because the pressure essentially is the same and the difference can be neglected. The gravity term is an instantaneous mass times the gravity constant the same can be said on the acceleration term. Yet, the acceleration is the derivative of the velocity and thus

$$\int \rho a_0 dV = \frac{dU}{dt} (m_R + m_f) \quad (6.23)$$

The first term on the right hand side is the change of the momentum in the rocket volume. This change is due the change in the volume of the oxidant and the fuel.

$$\frac{d}{dt} \int_{V_{c.v.}} \rho U_y dV = \frac{d}{dt} [(m_R + m_f) U] \quad (6.24)$$

Clearly, the change of the rocket mass can be considered minimal or even neglected. The oxidant and fuel flow outside. However, inside the rocket the change in the velocity is due to change in the reduction of the volume of the oxidant and fuel. This change is minimal and for this analysis, it can be neglected. The last term is

$$\int_{c.v.} \rho U_y U_{rn} dA = \dot{m} (U_g - U_R) \quad (6.25)$$

Combining all the above term results in

$$-F_R - (m_R + m_f) g + \frac{dU}{dt} (m_R + m_f) = \dot{m} (U_g - U_R) \quad (6.26)$$

Denoting $\mathcal{M}_T = m_R + m_f$ and thus $d\mathcal{M}/dt = \dot{m}$ and $U_e = U_g - U_R$. As first approximation, for constant fuel consumption (and almost oxidant), gas flow out is constant as well. Thus, for constant constant gas consumption equation (6.26) transformed to

$$-F_R - \mathcal{M}_T g + \frac{dU}{dt} \mathcal{M}_T = \dot{\mathcal{M}}_T U_e \quad (6.27)$$

Separating the variables equation (6.27) yields

$$dU = \left(\frac{-\dot{\mathcal{M}}_T U_e}{\mathcal{M}_T} - \frac{F_R}{\mathcal{M}_T} - g \right) dt \quad (6.28)$$

Before integrating equation (6.28), it can be noticed that the friction resistance F_R , is a function of the several parameters such the duration, the speed (the Reynolds number), material that surface made and the medium it flow in altitude. For simplicity here the part close to Earth (to the atmosphere) is assumed to be small compared to the distance in space. Thus it is assume that $F_R = 0$. Integrating equation (6.28) with limits of $U(t = 0) = 0$ provides

$$\int_0^U dU = -\dot{\mathcal{M}}_T U_e \int_0^t \frac{dt}{\mathcal{M}_T} - \int_0^t g dt \quad (6.29)$$

the results of the integration is (notice $\mathcal{M} = \mathcal{M}_0 - t\dot{\mathcal{M}}$)

$$U = U_e \ln \left(\frac{\mathcal{M}_0}{\mathcal{M}_0 - t\dot{\mathcal{M}}} \right) - g t \quad (6.30)$$

The following is an elaborated example which deals with an unsteady two dimensional problem. This problem demonstrates the used of control volume to find method of approximation for not given velocity profiles¹

Example 6.5:

A tank with wheels is filled with liquid is depicted in Figure 6.7. The tank upper part is opened to the atmosphere. At initial time the valve on the tank is opened and the liquid flows out with an uniform velocity profile. The tank mass with the wheels (the solid parts) is known, m_t . Calculate the tank velocity for two cases. One the wheels have a constant resistance with the ground and two the resistance linear function of the weight. Assume that the exit velocity is a linear function of the height.

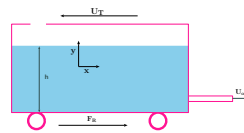


Fig. -6.7. Schematic of a tank seating on wheel for unsteady state discussion

SOLUTION

This problem is similar to the rocket mechanics with a twist, the source of the propulsion is the potential energy. Furthermore, the fluid has two velocity components verse one component in the rocket mechanics. The control volume is shown in Figure 6.7. The frame of reference is moving with the tank. This situation is unsteady state thus equation (6.12) for two dimensions is used. The mass conservation equation is

$$\frac{d}{dt} \int_{V_{c.v.}} \rho dV + \int_{S_{c.v.}} \rho dA = 0 \quad (6.V.a)$$

¹A variation of this problem has appeared in many books in the literature. However, in the past it was not noticed that a slight change in configuration leads to a constant x velocity. This problem was aroused in manufacturing industry. This author was called for consultation and to solve a related problem. For which he noticed this "constant velocity."

Equation (6.V.a) can be transferred to

$$\frac{dm_{c.v.}}{dt} = -\rho U_0 A_0 = -m_0 \quad (6.V.b)$$

Where m_0 is mass flow rate out. Equation (6.V.b) can be further reduced due to constant density to

$$\frac{d(Ah)}{dt} + U_0 A_0 = 0 \quad (6.V.c)$$

It can be noticed that the area of the tank is almost constant ($A = \text{constant}$) thus

$$A \frac{dh}{dt} + U_0 A_0 = 0 \implies \frac{dh}{dt} = -\frac{U_0 A_0}{A} \quad (6.31)$$

The relationship between the height and the flow now can be used.

$$U_0 = \mathcal{B} h \quad (6.V.d)$$

Where \mathcal{B} is the coefficient that has the right units to mach equation (6.V.d) that represent the resistance in the system and substitute the energy equation. Substituting equation (6.V.d) into equation (6.V.c) results in

$$\frac{dh}{dt} + \frac{\mathcal{B} h A_0}{A} = 0 \quad (6.V.e)$$

Equation (6.V.e) is a first order differential equation which can be solved with the initial condition $h(t=0) = h_0$. The solution (see for details in the Appendix A.1.1) is

$$h(t) = h_0 e^{-\frac{t A_0 \mathcal{B}}{A}} \quad (6.V.f)$$

To find the average velocity in the x direction a new control volume is used. The boundary of this control volume are the tank boundary on the left with the straight surface as depicted in Figure 6.8. The last boundary is variable surface in a distance x from the tank left part. The tank depth, is not relevant. The mass conservation for this control volume is

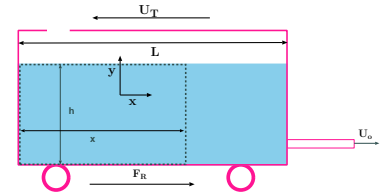


Fig. -6.8. A new control volume to find the velocity in discharge tank for example 6.5.

$$w x \frac{dh}{dt} = -w h \bar{U}_x \quad (6.V.g)$$

Where here w is the depth or width of the tank. Substituting (6.V.f) into (6.V.g) results

$$\bar{U}_x(x) = \frac{x A_0 \cancel{h_0 \mathcal{B}}}{A \cancel{h}} e^{-\frac{t A_0 \mathcal{B}}{A}} = \frac{x A_0 \mathcal{B}}{A} \quad (6.V.h)$$

The average x component of the velocity is a linear function of x . Perhaps surprising, it also can be noticed that $\overline{U_x}(x)$ is a not function of the time. Using this function, the average velocity in the tank is

$$\overline{U_x} = \frac{1}{L} \int_0^L \frac{x A_0 \mathcal{B}}{A} = \frac{L A_0 \mathcal{B}}{2 A} \quad (6.V.i)$$

It can be noticed that $\overline{U_x}$ is not function of height, h . In fact, it can be shown that average velocity is a function of cross section (what direction?).

Using a similar control volume², the average velocity in the y direction is

$$\overline{U_y} = \frac{dh}{dt} = -\frac{h_0 A_0 \mathcal{B}}{A} e^{-\frac{t A_0 \mathcal{B}}{A}} \quad (6.V.j)$$

It can be noticed that the velocity in the y is a function of time as oppose to the x direction.

The applicable momentum equation (in the tank frame of reference) is (6.11) which is reduced to

$$-\mathbf{F}_R - (m_t + m_f) \mathbf{g} - \overbrace{\mathbf{a}(m_t + m_f)}^{\text{acceleration}} = \frac{d}{dt} [(m_t + m_f) \mathbf{U}_r] + U_0 m_o \quad (6.V.k)$$

Where \mathbf{U}_r is the relative fluid velocity to the tank (if there was no tank movement). m_f and m_t are the mass of the fluid and the mass of tank respectively. The acceleration of the tank is $\mathbf{a} = -\hat{i}a_0$ or $\hat{i} \cdot \mathbf{a} = -a$. And the additional force for accelerated system is

$$-\hat{i} \cdot \int_{V_{c.v.}} \mathbf{a} \rho dV = m_{c.v.} a$$

The mass in the control volume include the mass of the liquid with mass of the solid part (including the wheels).

$$m_{c.v.} = m_f + m_T$$

because the density of the air is very small the change of the air mass is very small as well ($\rho_a \ll \rho$).

The pressure around the control volume is uniform thus

$$\int_{S_{c.v.}} P \cos \theta_x dA \sim 0$$

and the resistance due to air is negligible, hence

$$\int_{S_{c.v.}} \tau dA \sim 0$$

²The boundaries are the upper (free surface) and tank side with a y distance from the free surface. $\int U_{bn} dA = \int U_{rn} dA \implies U_{bn} = U_{rn}$.

The momentum flow rate out of the tank is

$$\int_{S_{c.v.}} \rho U_x U_{rn} dA = \rho U_o^2 A_o = m_o U_o \quad (6.32)$$

In the x coordinate the momentum equation is

$$-F_x + (m_t + m_f) a = \frac{d}{dt} [(m_t + m_f) U_x] + U_o \dot{m}_f \quad (6.V.l)$$

Where F_x is the x component of the reaction which is opposite to the movement direction. The momentum equation in the y coordinate it is

$$F_y - (m_t + m_f) g = \frac{d}{dt} [(m_t + m_f) U_y] \quad (6.V.m)$$

There is no mass flow in the y direction and U_y is component of the velocity in the y direction.

The tank movement cause movement of the air which cause momentum change. This momentum is function of the tank volume times the air density times tank velocity ($h_o \times A \times \rho_a$). This effect is known as the add mass/momentum and will be discussed in the Dimensional Analysis and Ideal Flow Chapters. Here this effect is neglected.

The main problem of integral analysis approach is that it does not provide a way to analysis the time derivative since the velocity profile is not given inside the control volume. This limitation can be partially overcome by assuming some kind of average. It can be noticed that the velocity in the tank has two components. The first component is downward (y) direction and the second in the exit direction (x). The velocity in the y direction does not contribute to the momentum in the x direction. The average velocity in the tank (because constant density and more about it later section) is

$$\bar{U}_x = \frac{1}{V_t} \int_{V_f} U_x dV$$

Because the integral is replaced by the average it is transferred to

$$\int_{V_f} \rho U_x dV \sim m_{c.v.} \bar{U}_x$$

Thus, if the difference between the actual and averaged momentum is neglected then

$$\frac{d}{dt} \int_{V_f} \rho U_x dV \sim \frac{d}{dt} (m_{c.v.} \bar{U}_x) = \frac{d m_{c.v.}}{dt} \bar{U}_x + \frac{d \bar{U}_x}{dt} m_{c.v.} \quad (6.V.n)$$

Noticing that the derivative with time of control volume mass is the flow out in equation (6.V.n) becomes

$$\frac{d m_{c.v.}}{dt} \bar{U}_x + \frac{d \bar{U}_x}{dt} m_{c.v.} = - \overbrace{\dot{m}_0}^{\text{mass rate out}} \bar{U}_x = -m_0 \frac{L A_0 B}{2 A} \quad (6.V.o)$$

Combining all the terms results in

$$-F_x + a (m_f + m_t) = -m_0 \frac{L A_0 \mathcal{B}}{2A} - U_0 m_0 \quad (6.V.p)$$

Rearranging and noticing that $a = dU_T/dt$ transformed equation (6.V.p) into

$$a = \frac{F_x}{m_f + m_t} - m_0 \left(\frac{L A_0 \mathcal{B} + 2 A U_0 (m_f + m_t)}{2 A (m_f + m_t)} \right) \quad (6.V.q)$$

If the $F_x \geq m_0 \left(\frac{L A_0 \mathcal{B}}{2A} + U_0 \right)$ the toy will not move. However, if it is the opposite the toy start to move. From equation (6.V.d) the mass flow out is

$$m_0(t) = \mathcal{B} h_0 \overbrace{e^{-\frac{U_0}{h} \frac{t A_0 \mathcal{B}}{A}}} \quad A_0 \rho \quad (6.V.r)$$

The mass in the control volume is

$$m_f = \rho A h_0 \overbrace{e^{-\frac{v}{t A_0 \mathcal{B}} \frac{t A_0 \mathcal{B}}{A}}} \quad (6.V.s)$$

The initial condition is that $U_T(t = 0) = 0$. Substituting equations (6.V.r) and (6.V.s) into equation (6.V.q) transforms it to a differential equation which is integrated if R_x is constant.

For the second case where R_x is a function of the R_y as

$$R_x = \mu R_y \quad (6.33)$$

The y component of the average velocity is function of the time. The change in the accumulative momentum is

$$\frac{d}{dt} [(m_f) \overline{U}_y] = \frac{dm_f}{dt} \overline{U}_y + \frac{d\overline{U}_y}{dt} m_f \quad (6.V.t)$$

The reason that m_f is used because the solid parts do not have velocity in the y direction. Rearranging the momentum equation in the y direction transformed

$$F_y = \left(m_t + \rho A h_0 \overbrace{e^{-\frac{m_f}{t A_0 \mathcal{B}} \frac{t A_0 \mathcal{B}}{A}}} \right) g + 2 \left(\frac{\rho h_0 A_0^2 \mathcal{B}^2}{A} \right)^2 e^{-\frac{t A_0 \mathcal{B}}{A}} \quad (6.V.u)$$

The actual results of the integrations are not provided since the main purpose of this exercise is to learn how to use the integral analysis.

End Solution

Average Velocity Estimates

In example 6.1 relationship between momentum of maximum velocity to average velocity was presented. Here, relationship between momentum for the average velocity to the actual velocity is presented. There are situations where actual velocity profile is not known but its function can be approximated. For example, the velocity profile can be estimated using the ideal fluid theory but the actual values are not known. For example, the flow profile in example 6.5 can be estimated even by hand sketching.

For these cases a correction factor can be used. This correction factor can be calculated by finding the relation between the two cases. The momentum for average velocity is

$$M_a = m_{c.v} \bar{U} = \rho V \int_{c.v} U dV \quad (6.34)$$

The actual momentum for control volume is

$$M_c = \int_{c.v.} \rho U_x dV \quad (6.35)$$

These two have to equal thus,

$$C \rho V \int_{c.v} U dV = \int_{c.v.} \rho U_x dV \quad (6.36)$$

If the density is constant then the coefficient $C \equiv 1$ is one. However, if the density is not constant its coefficient is not equal to one.

6.3 Conservation Moment Of Momentum

The angular momentum can be derived in the same manner as the momentum equation for control volume. The force

$$\mathbf{F} = \frac{D}{Dt} \int_{V_{sys}} \rho \mathbf{U} dV \quad (6.37)$$

The angular momentum that will be obtained by calculating the change of every element in the system as

$$\mathfrak{M} = \mathbf{r} \times \mathbf{F} = \frac{D}{Dt} \int_{V_{sys}} \rho \mathbf{r} \times \mathbf{U} dV \quad (6.38)$$

Now the left hand side has to be transformed into the control volume as

$$\mathfrak{M} = \frac{d}{dt} \int_{V_{c.v.}} \rho (\mathbf{r} \times \mathbf{U}) dV + \int_{S_{c.v}} \rho (\mathbf{r} \times \mathbf{U}) \mathbf{U}_{rn} dA \quad (6.39)$$

The angular momentum equation, applying equation (6.39) to uniform and steady state flow with neglected pressure gradient is reduced to

$$\mathfrak{M} = \dot{m} (r_2 \times U_2 + r_2 \times U_1) \tag{6.40}$$

Introduction to Turbo Machinery

The analysis of many turbomachinery such as centrifugal pump is fundamentally based on the angular momentum. To demonstrate this idea, the following discussion is provided. A pump impeller is shown in Figure 6.9 commonly used in industry. The impeller increases the velocity of the fluid by increasing the radius of the particles. The inside particle is obtained larger velocity and due to centrifugal forces is moving to outer radius for which additionally increase of velocity occur. The pressure on the outer side is uniform thus does not create a moment. The flow is assumed to enter the impeller radially with average velocity U_1 . Here it is assumed that fluid is incompressible ($\rho = constant$). The height of the impeller is h . The exit liquid velocity, U_2 has two components, one the tangential velocity, U_{t2} and radial component, U_{n2} . The relative exit velocity is U_{lr2} and the velocity of the impeller edge is U_{m2} . Notice that tangential liquid velocity, U_{t2} is not equal to the impeller outer edge velocity U_{m2} . It is assumed that required torque is function U_2 , r , and h .

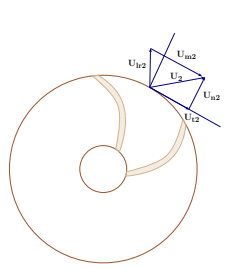


Fig. -6.9. The impeller of the centrifugal pump and the velocities diagram at the exit.

$$\mathfrak{M} = \dot{m} r_2 U_{t2} \tag{6.41}$$

Multiplying equation (6.41) results in

$$\mathfrak{M} \omega = \dot{m} \overbrace{r_2 \omega}^{U_{m2}} U_{t2} \tag{6.42}$$

The shaft work is given by the left side and hence,

$$\dot{W} = \dot{m} U_{m2} U_{t2} \tag{6.43}$$

The difference between U_{m2} to U_{t2} is related to the efficiency of the pump which will be discussed in the chapter on the turbomachinery.

Example 6.6:

A centrifugal pump is pumping $600 \text{ 2}[m^3/hour]$. The thickness of the impeller, h is $2[cm]$ and the exit diameter is $0.40[m]$. The angular velocity is 1200 r.p.m. Assume that angle velocity is leaving the impeller is 125° . Estimate what is the minimum energy required by the pump.

6.4 More Examples on Momentum Conservation

Example 6.7:

A design of a rocket is based on the idea that density increase of the leaving jet increases the acceleration of the rocket see Figure

6.10. Assume that this idea has a good engineering logic. Liquid fills the lower part of the rocket tank. The upper part of the rocket tank is filled with compressed gas. Select the control volume in such a way that provides the ability to find the rocket acceleration. What is the instantaneous velocity of the rocket at time zero? Develop the expression for the pressure (assuming no friction with the walls). Develop expression for rocket velocity. Assume that the gas is obeying the perfect gas model. What are the parameters that effect the problem.

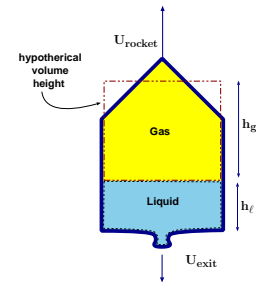


Fig. -6.10. Nozzle schematics water rocket for the discussion on the forces for example 6.7

SOLUTION

Under construction for time being only hints³

In the solution of this problem several assumptions must be made so that the integral system can be employed.

- The surface remained straight at the times and no liquid residue remains behind.
- The gas obeys the ideal gas law.
- The process is isothermal (can be isentropic process).
- No gas leaves the rocket.
- The mixing between the liquid and gas is negligible.
- The gas mass is negligible in comparison to the liquid mass and/or the rocket.
- No resistance to the rocket (can be added).
- The cross section of the liquid is constant.

In this problem the energy source is the pressure of the gas which propels the rocket. Once the gas pressure reduced to be equal or below the outside pressure the

³This problem appeared in the previous version (0.2.3) without a solution. Several people ask to provide a solution or some hints for the solution. The following is not the solution but rather the approach how to treat this problem.

rocket have no power for propulsion. Additionally, the initial take off is requires a larger pressure.

The mass conservation is similar to the rocket hence it is

$$\frac{dm}{dt} = -U_e A_e \quad (6.VII.a)$$

The mass conservation on the gas zone is a byproduct of the mass conservation of the liquid. Furthermore, it can be observed that the gas pressure is a direct function of the mass flow out.

The gas pressure at the initial point is

$$P_0 = \rho_0 R T \quad (6.VII.b)$$

Per the assumption the gas mass remain constant and is denoted as m_g . Using the above definition, equation (6.VII.b) becomes

$$P_0 = \frac{m_g R T}{V_{0g}} \quad (6.VII.c)$$

The relationship between the gas volume

$$V_g = \bar{h}_g A \quad (6.VII.d)$$

The gas geometry is replaced by a virtual constant cross section which cross section of the liquid (probably the same as the base of the gas phase). The change of the gas volume is

$$\frac{dV_g}{dt} = A \frac{dh_g}{dt} = -A \frac{dh_\ell}{dt} \quad (6.VII.e)$$

The last identify in the above equation is based on the idea what ever height concede by the liquid is taken by the gas. The minus sign is to account for change of "direction" of the liquid height. The total change of the gas volume can be obtained by integration as

$$V_g = A (h_{g0} - \Delta h_\ell) \quad (6.VII.f)$$

It must be point out that integral is not function of time since the height as function of time is known at this stage.

The initial pressure now can be expressed as

$$P_0 = \frac{m_g R T}{h_{g0} A} \quad (6.VII.g)$$

The pressure at any time is

$$P = \frac{m_g R T}{h_g A} \quad (6.VII.h)$$

Thus the pressure ratio is

$$\frac{P}{P_0} = \frac{h_{g0}}{h_g} = \frac{h_{g0}}{h_{g0} - \Delta h_\ell} = h_{g0} \frac{1}{1 - \frac{\Delta h_\ell}{h_{g0}}} \quad (6.VII.i)$$

Equation (6.VII.a) can be written as

$$m_\ell(t) = m_{\ell 0} - \int_0^t U_e A_e dt \quad (6.VII.j)$$

From equation (6.VII.a) it also can be written that

$$\frac{dh_\ell}{dt} = \frac{U_e A_e}{\rho_e A} \quad (6.VII.k)$$

According to the assumption the flow out is linear function of the pressure inside thus,

$$U_e = f(P) + g h_\ell \rho \simeq f(P) = \zeta P \quad (6.VII.l)$$

Where ζ here is a constant which the right units.

The liquid momentum balance is

$$-g(m_R + m_\ell) - a(m_R + m_\ell) = \overbrace{\frac{d}{dt}(m_R + m_\ell)U}^{=0} + bc + (U_R + U_\ell)m_\ell \quad (6.VII.m)$$

Where bc is the change of the liquid mass due the boundary movement.

End Solution

Example 6.8:

A rocket is filled with only compressed gas. At a specific moment the valve is opened and the rocket is allowed to fly. What is the minimum pressure which make the rocket fly. What are the parameters that effect the rocket velocity. Develop an expression for the rocket velocity.

Example 6.9:

In Example 6.5 it was mentioned that there are only two velocity components. What was the assumption that the third velocity component was neglected.

6.4.1 Qualitative Questions

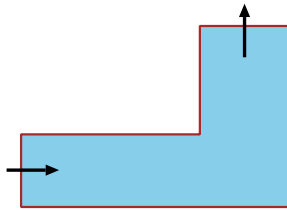
Example 6.10:

For each following figures discuss and state force direction and the momentum that act on the control volume due to .

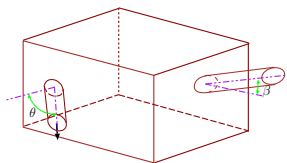
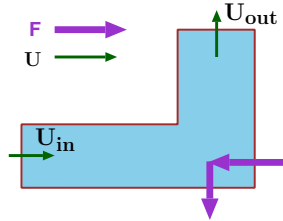
Example 6.11:

Situations

Explanations



Flow in and out of Angle



Flow in and out at angle from a tank

A similar tank as shown in Figure 6.11 is built with a exit located in uneven distance from the the right and the left and is filled with liquid. The exit is located on the left hand side at the front. What are the direction of the forces that keep the control volume in the same location? Hints, consider the unsteady effects. Look at the directions which the unsteady state momentum in the tank change its value.

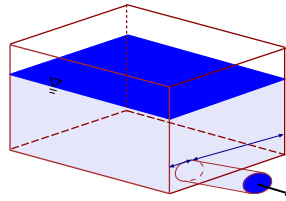


Fig. -6.11. Flow out of un symmetrical tank for example 6.11

CHAPTER 7

Energy Conservation

7.1 *The First Law of Thermodynamics*

This chapter focuses on the energy conservation which is the first law of thermodynamics¹. The fluid, as all phases and materials, obeys this law which creates strange and wonderful phenomena such as a shock and choked flow. Moreover, this law allows to solve problems, which were assumed in the previous chapters. For example, the relationship between height and flow rate was assumed previously, here it will be derived. Additionally a discussion on various energy approximation is presented.

It was shown in Chapter 2 that the energy rate equation (2.10) for a system is

$$\dot{Q} - \dot{W} = \frac{D E_U}{Dt} + mU \frac{DU}{Dt} + mg \frac{Dz}{Dt} \quad (7.1)$$

This equation can be rearranged to be

$$\dot{Q} - \dot{W} = \frac{D}{Dt} \left(E_U + m \frac{U^2}{2} + mgz \right) \quad (7.2)$$

Equation (7.2) is similar to equation (6.3) in which the right hand side has to be interpreted and the left hand side interpolated using the Reynold's Transport Theorem (RTT)². The right hand side is very complicated and only some of the effects will be discussed (It is only an introductory material).

¹Thermodynamics is the favorite topic of this author since it was his major in high school. Clearly this topic is very important and will be extensively discussed here. However, during time of the constructing this book only a simple skeleton by Potto standards will be build.

²Some view the right hand side as external effects while the left side of the equation represents the internal effects. This simplistic representation is correct only under extreme conditions. For example, the above view is wrong when the heat convection, which is external force, is included on the right hand side.

The energy transfer is carried (mostly³) by heat transfer to the system or the control volume. There are three modes of heat transfer, conduction, convection⁴ and radiation. In most problems, the radiation is minimal. Hence, the discussion here will be restricted to convection and conduction. Issues related to radiation are very complicated and considered advance material and hence will be left out. The issues of convection are mostly covered by the terms on the left hand side. The main heat transfer mode on the left hand side is conduction. Conduction for most simple cases is governed by Fourier's Law which is

$$d\dot{q} = k_T \frac{dT}{dn} dA \quad (7.3)$$

Where $d\dot{q}$ is heat transfer to an infinitesimal small area per time and k_T is the heat conduction coefficient. The heat derivative is normalized into area direction. The total heat transfer to the control volume is

$$\dot{Q} = \int_{A_{cv}} k \frac{dT}{dn} dA \quad (7.4)$$

The work done on the system is more complicated to express than the heat transfer. There are two kinds of works that the system does on the surroundings. The first kind work is by the friction or the shear stress and the second by normal force. As in the previous chapter, the surface forces are divided into two categories: one perpendicular to the surface and one with the surface direction. The work done by system on the surroundings (see Figure 7.4) is

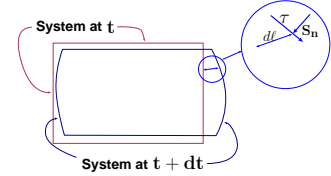


Fig. -7.1. The work on the control volume is done by two different mechanisms

$$dw = \underbrace{-\mathbf{S} dA}_{d\mathbf{F}} \cdot d\ell = -(\mathbf{S}_n + \boldsymbol{\tau}) \cdot \underbrace{d\ell dA}_{dV} \quad (7.5)$$

The change of the work for an infinitesimal time (excluding the shaft work) is

$$\frac{dw}{dt} = -(\mathbf{S}_n + \boldsymbol{\tau}) \cdot \underbrace{\frac{d\ell}{dt}}_U dA = -(\mathbf{S}_n + \boldsymbol{\tau}) \cdot \mathbf{U} dA \quad (7.6)$$

The total work for the system including the shaft work is

$$\dot{W} = - \int_{Ac.v.} (\mathbf{S}_n + \boldsymbol{\tau}) \mathbf{U} dA - W_{shaft} \quad (7.7)$$

³There are other methods such as magnetic fields (like microwave) which are not part of this book.

⁴When dealing with convection, actual mass transfer must occur and thus no convection is possible to a system by the definition of system.

The energy equation (7.2) for system is

$$\int_{A_{sys}} k_T \frac{dT}{dn} dA + \int_{A_{sys}} (\mathbf{S}_n + \boldsymbol{\tau}) \cdot d\mathbf{V} + \dot{W}_{shaft} = \frac{D}{Dt} \int_{V_{sys}} \rho \left(E_U + m \frac{U^2}{2} + g z \right) dV \quad (7.8)$$

Equation (7.8) does not apply any restrictions on the system. The system can contain solid parts as well several different kinds of fluids. Now Reynolds Transport Theorem can be used to transformed the left hand side of equation (7.8) and thus yields

$$\int_{A_{cv}} k_T \frac{dT}{dn} dA + \int_{A_{cv}} (\mathbf{S}_n + \boldsymbol{\tau}) \cdot d\mathbf{A} + \dot{W}_{shaft} = \frac{d}{dt} \int_{V_{cv}} \rho \left(E_u + m \frac{U^2}{2} + g z \right) dV + \int_{A_{cv}} \left(E_u + m \frac{U^2}{2} + g z \right) \rho U_{rn} dA \quad (7.9)$$

From now on the notation of the control volume and system will be dropped since all equations which deals with the control volume. In the last term in equation (7.9) the velocity appears twice. Note that U is the velocity in the frame of reference while U_{rn} is the velocity relative to the boundary. As it was discussed in the previous chapter the normal stress component is replaced by the pressure (see equation (6.8) for more details). The work rate (excluding the shaft work) is

$$\dot{W} \cong \overbrace{\int_S P \hat{n} \cdot \mathbf{U} dA}^{\text{flow work}} - \int_S \boldsymbol{\tau} \cdot \mathbf{U} \hat{n} dA \quad (7.10)$$

The first term on the right hand side is referred to in the literature as the flow work and is

$$\int_S P \hat{n} \cdot \mathbf{U} dA = \int_S P \overbrace{(U - U_b) \hat{n}}^{U_{rn}} dA + \int_S P U_{bn} dA \quad (7.11)$$

Equation (7.11) can be further manipulated to become

$$\int_S P \hat{n} \cdot \mathbf{U} dA = \overbrace{\int_S \frac{P}{\rho} \rho U_{rn} dA}^{\text{work due to the flow}} + \overbrace{\int_S P U_{bn} dA}^{\text{work due to boundaries movement}} \quad (7.12)$$

The second term is referred to as the shear work and is defined as

$$\dot{W}_{shear} = - \int_S \boldsymbol{\tau} \cdot \mathbf{U} dA \quad (7.13)$$

Substituting all these terms into the governing equation yields

$$\begin{aligned} \dot{Q} - \dot{W}_{shear} - \dot{W}_{shaft} = \frac{d}{dt} \int_V \left(E_u + \frac{U^2}{2} + gz \right) dV + \\ \int_S \left(E_u + \frac{P}{\rho} + \frac{U^2}{2} + gz \right) U_{rn} \rho dA + \int_S P U_{rn} dA \end{aligned} \quad (7.14)$$

The new term P/ρ combined with the internal energy, E_u is referred to as the enthalpy, h , which was discussed on page 36. With these definitions equation (7.14) transformed

$$\begin{aligned} \dot{Q} - \dot{W}_{shear} + \dot{W}_{shaft} = \frac{d}{dt} \int_V \left(E_u + \frac{U^2}{2} + gz \right) dV + \\ \int_S \left(h + \frac{U^2}{2} + gz \right) U_{rn} \rho dA + \int_S P U_{rn} dA \end{aligned} \quad (7.15)$$

Equation (7.15) describes the energy conservation for the control volume in stationary coordinates.

The example of flow from a tank or container is presented to demonstrate how to treat some of terms in equation (7.15).

Flow Out From A Container

In the previous chapters of this book, the flow rate out of a tank or container was assumed to be a linear function of the height. The flow out is related to the height but in a more complicate function and is the focus of this discussion. The energy equation with mass conservation will be utilized for this analysis. In this analysis several assumptions are made which includes the following: constant density, the gas density is very small compared to liquid density, and exit area is relatively small, so the velocity can be assumed uniform (not a function of the opening height)⁵, surface tension effects are negligible and the liquid surface is straight⁶. Additionally, the temperature is assumed to constant.

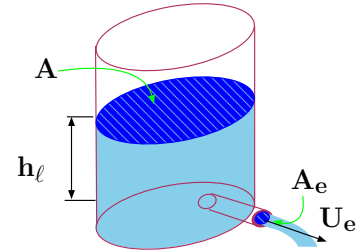


Fig. -7.2. Discharge from a Large Container with a small diameter.

⁵Later a discussion about the height opening effects will be discussed.

⁶This assumption is appropriated only under certain conditions which include the geometry of the tank or container and the liquid properties. A discussion about this issue will be presented in the Dimensional Chapter and is out of the scope of this chapter. Also note that the straight surface assumption is not the same surface tension effects zero.

The control volume is chosen so that all the liquid is included up to exit of the pipe. The conservation of the mass is

$$\frac{d}{dt} \int_V \rho dV + \int_A \rho U_{rn} dA = 0 \tag{7.16}$$

which also can be written (because $\frac{d\rho}{dt} = 0$) as

$$\int_A U_{bn} dA + \int_A U_{rn} dA = 0 \tag{7.17}$$

Equation (7.17) provides the relationship between boundary velocity to the exit velocity as

$$AU_b = A_e U_e \tag{7.18}$$

Note that the boundary velocity is not the averaged velocity but the actual velocity. The averaged velocity in z direction is same as the boundary velocity

$$U_b = U_z = \frac{dh}{dt} = \frac{A_e}{A} U_e \tag{7.19}$$

The x component of the averaged velocity is a function of the geometry and was calculated in Example 5.12 to be larger than

$$\overline{U}_x \gtrsim \frac{2r}{h} \frac{A_e}{A} U_e \implies \overline{U}_x \cong \frac{2r}{h} U_b = \frac{2r}{h} \frac{dh}{dt} \tag{7.20}$$

In this analysis, for simplicity, this quantity will be used.

The averaged velocity in the y direction is zero because the flow is symmetrical⁷. However, the change of the kinetic energy due to the change in the velocity field isn't zero. The kinetic energy of the tank or container is based on the half part as shown in Figure 7.3. Similar estimate that was done for x direction can be done to every side of the opening if they are not symmetrical. Since in this case the geometry is assumed to be symmetrical one side is sufficient as

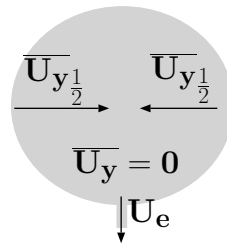


Fig. -7.3. How to compensate and estimate the kinetic energy when averaged Velocity is zero.

$$\overline{U}_y \cong \frac{(\pi - 2)r}{8h} \frac{dh}{dt} \tag{7.21}$$

Also notice that the surface velocity is not zero. The surface has three velocity components which non have them vanish. However, in this discussion it is assumed that surface has only one component in z direction. Hence it requires that velocity profile in $x y$ to be parabolic. Second reason for this exercise the surface velocity has only one component is to avoid dealing with Bar-Meir's instability.

⁷For the mass conservation analysis, the velocity is zero for symmetrical geometry and some other geometries. However, for the energy analysis the averaged velocity cannot be considered zero.

The energy balance can be expressed by equation (7.15) which is applicable to this case. The temperature is constant⁸. In this light, the following approximation can be written

$$\dot{Q} = \frac{E_u}{dt} = h_{in} - h_{out} = 0 \quad (7.22)$$

The boundary shear work is zero because the velocity at tank boundary or walls is zero. Furthermore, the shear stresses at the exit are normal to the flow direction hence the shear work is vanished. At the free surface the velocity has only normal component⁹ and thus shear work vanishes there as well. Additionally, the internal shear work is assumed negligible.

$$\dot{W}_{shear} = \dot{W}_{shaft} = 0 \quad (7.23)$$

Now the energy equation deals with no “external” effects. Note that the exit velocity on the upper surface is zero $U_{rn} = 0$.

Combining all these information results in

$$\underbrace{\frac{d}{dt} \int_V \left(\frac{U^2}{2} + gz \right) \rho dV}_{\text{internal energy change}} + \underbrace{\int_A \left(\frac{P_e}{\rho} + \frac{U_e^2}{2} \right) U_e \rho dA}_{\text{energy in and out}} - \underbrace{\int_A P_a U_b dA}_{\text{upper surface work}} = 0 \quad (7.24)$$

Where U_b is the upper boundary velocity, P_a is the external pressure and P_e is the exit pressure¹⁰.

The pressure terms in equation (7.24) are

$$\int_A \frac{P_e}{\rho} U_e \rho dA - \int_A P_a U_b dA = P_e \int_A U_e dA - P_a \int_A U_b dA \quad (7.25)$$

It can be noticed that $P_a = P_e$ hence

$$P_a \left(\overbrace{\int_A U_e dA - \int_A U_b dA}^{=0} \right) = 0 \quad (7.26)$$

The governing equation (7.24) is reduced to

$$\frac{d}{dt} \int_V \left(\frac{U^2}{2} + gz \right) \rho dV - \int_A \left(\frac{U_e^2}{2} \right) U_e \rho dA = 0 \quad (7.27)$$

⁸This approach is a common approximation. Yet, why this approach is correct in most cases is not explained here. Clearly, the dissipation creates a loss that has temperature component. In this case, this change is a function of Eckert number, Ec which is very small. The dissipation can be neglected for small Ec number. Ec number is named after this author’s adviser, E.R.G. Eckert. A discussion about this effect will be presented in the dimensional analysis chapter.

⁹It is only the same assumption discussed earlier.

¹⁰It is assumed that the pressure in exit across section is uniform and equal surroundings pressure.

The minus sign is because the flow is out of the control volume.

Similarly to the previous chapter which the integral momentum will be replaced by some kind of average. The terms under the time derivative can be divided into two terms as

$$\frac{d}{dt} \int_V \left(\frac{U^2}{2} + gz \right) \rho dV = \frac{d}{dt} \int_V \frac{U^2}{2} dV + \frac{d}{dt} \int_V gz \rho dV \quad (7.28)$$

The second integral (in the r.h.s) of equation (7.28) is

$$\frac{d}{dt} \int_V gz \rho dV = g \rho \frac{d}{dt} \int_A \int_0^h z \overbrace{dz}^{dV} dA \quad (7.29)$$

Where h is the height or the distance from the surface to exit. The inside integral can be evaluated as

$$\int_0^h z dz = \frac{h^2}{2} \quad (7.30)$$

Substituting the results of equation (7.30) into equation (7.29) yields

$$g \rho \frac{d}{dt} \int_A \frac{h^2}{2} dA = g \rho \frac{d}{dt} \left(\frac{h}{2} \overbrace{h A}^V \right) = g \rho A h \frac{d h}{dt} \quad (7.31)$$

The kinetic energy related to the averaged velocity with a correction factor which depends on the geometry and the velocity profile. Furthermore, Even the averaged velocity is zero the kinetic energy is not zero and another method should be used.

A discussion on the correction factor is presented to provide a better “averaged” velocity. A comparison between the actual kinetic energy and the kinetic energy due to the “averaged” velocity (to be called the averaged kinetic energy) provides a correction coefficient. The first integral can be estimated by examining the velocity profile effects. The averaged velocity is

$$U_{ave} = \frac{1}{V} \int_V U dV \quad (7.32)$$

The total kinetic energy for the averaged velocity is

$$\rho U_{ave}^2 V = \rho \left(\frac{1}{V} \int_V U dV \right)^2 V = \rho \left(\int_V U dV \right)^2 \quad (7.33)$$

The general correction factor is the ratio of the above value to the actual kinetic energy as

$$C_F = \frac{\left(\int_V \rho U dV\right)^2}{\int_V \rho U^2 dV} \neq \frac{\rho (U_{ave})^2 V}{\int_V \rho U^2 dV} \quad (7.34)$$

Here, C_F is the correction coefficient. Note, the inequality sign because the density distribution for compressible fluid. The correction factor for a constant density fluid is

$$C_F = \frac{\left(\int_V \rho U dV\right)^2}{\int_V \rho U^2 dV} = \frac{\left(\rho \int_V U dV\right)^2}{\rho \int_V U^2 dV} = \frac{U_{ave}^2 V}{\int_V U^2 dV} \quad (7.35)$$

This integral can be evaluated for any given velocity profile. A large family of velocity profiles is laminar or parabolic (for one directional flow)¹¹. For a pipe geometry, the velocity is

$$U\left(\frac{r}{R}\right) = U(\bar{r}) = U_{max} (1 - \bar{r}^2) = 2U_{ave} (1 - \bar{r}^2) \quad (7.36)$$

It can be noticed that the velocity is presented as a function of the reduced radius¹². The relationship between U_{max} to the averaged velocity, U_{ave} is obtained by using equation (7.32) which yields 1/2.

Substituting equation (7.36) into equation (7.35) results

$$\frac{U_{ave}^2 V}{\int_V U^2 dV} = \frac{U_{ave}^2 V}{\int_V (2U_{ave} (1 - \bar{r}^2))^2 dV} = \frac{U_{ave}^2 V}{\frac{4U_{ave}^2 \pi L R^2}{3}} = \frac{3}{4} \quad (7.37)$$

The correction factor for many other velocity profiles and other geometries can be smaller or larger than this value. For circular shape, a good guess number is about 1.1. In this case, for simplicity reason, it is assumed that the averaged velocity indeed represent the energy in the tank or container. Calculations according to this point can improve the accurately based on the above discussion.

The difference between the “averaged momentum” velocity and the “averaged kinetic” velocity is also due to the fact that energy is added for different directions while in the momentum case, different directions cancel each other out.

¹¹Laminar flow is not necessarily implies that the flow velocity profile is parabolic. The flow is parabolic only when the flow is driven by pressure or gravity. More about this issue in the Differential Analysis Chapter.

¹²The advantage is described in the Dimensional Analysis Chapter.

The unsteady state term then obtains the form

$$\frac{d}{dt} \int_V \rho \left(\frac{U^2}{2} + gy \right) dV \cong \rho \frac{d}{dt} \left(\left[\frac{\bar{U}^2}{2} + \frac{gh}{2} \right] \overbrace{hA}^V \right) \quad (7.38)$$

The relationship between the boundary velocity to the height (by definition) is

$$U_b = \frac{dh}{dt} \quad (7.39)$$

Therefore, the velocity in the z direction¹³ is

$$U_z = \frac{dh}{dt} \quad (7.40)$$

$$U_e = \frac{A}{A_e} \frac{dh}{dt} = -U_b \frac{dh}{dt} \quad (7.41)$$

Combining all the three components of the velocity (Pythagorean Theorem) as

$$\bar{U}^2 \cong \bar{U}_x^2 + \bar{U}_y^2 + \bar{U}_z^2 \quad (7.42)$$

$$\bar{U}^2 \cong \left(\frac{(\pi-2)r}{8h} \frac{dh}{dt} \right)^2 + \left(\frac{(\pi-1)r}{4h} \frac{dh}{dt} \right)^2 + \left(\frac{dh}{dt} \right)^2 \quad (7.43)$$

$$\bar{U} \cong \frac{dh}{dt} \sqrt{\overbrace{\left(\frac{(\pi-2)r}{8h} \right)^2 + \left(\frac{(\pi-1)r}{4h} \right)^2}^{f(G)} + 1^2} \quad (7.44)$$

It can be noticed that $f(G)$ is a weak function of the height inverse. Analytical solution of the governing equation is possible including this effect of the height. However, the mathematical complication are enormous¹⁴ and this effect is assumed neglected and the function to be constant.

¹³A similar point was provided in mass conservation Chapter 5. However, it easy can be proved by construction the same control volume. The reader is encouraged to do it to get acquainted with this concept.

¹⁴The solution not the derivation is about one page. It must be remembered that is effect extremely important in the later stages of the emptying of the tank. But in the same vain, some other effects have to be taken into account which were neglected in construction of this model such as upper surface shape.

The last term is

$$\int_A \frac{U_e^2}{2} U_e \rho dA = \frac{U_e^2}{2} U_e \rho A_e = \left(\frac{dh}{dt} \frac{A}{A_e} \right)^2 U_e \rho A_e \quad (7.45)$$

Combining all the terms into equation (7.27) results in

$$\cancel{\rho} \frac{d}{dt} \left(\left[\frac{\bar{U}^2}{2} + \frac{gh}{2} \right] \overbrace{hA}^V \right) - \frac{1}{2} \left(\frac{dh}{dt} \right)^2 \left(\frac{A}{A_e} \right)^2 U_e \cancel{\rho} A_e = 0 \quad (7.46)$$

taking the derivative of first term on l.h.s. results in

$$\frac{d}{dt} \left[\frac{\bar{U}^2}{2} + \frac{gh}{2} \right] hA + \left[\frac{\bar{U}^2}{2} + \frac{gh}{2} \right] A \frac{dh}{dt} - \frac{1}{2} \left(\frac{dh}{dt} \right)^2 \left(\frac{A}{A_e} \right)^2 U_e A_e = 0 \quad (7.47)$$

Equation (7.47) can be rearranged and simplified and combined with mass conservation ¹⁵.

***** Advance material can be skipped *****

Dividing equation (7.46) by $U_e A_e$ and utilizing equation (7.40)

$$\frac{d}{dt} \left[\frac{\bar{U}^2}{2} + \frac{gh}{2} \right] \frac{hA}{U_e A_e} + \left[\frac{\bar{U}^2}{2} + \frac{gh}{2} \right] \frac{\overbrace{A \frac{dh}{dt}}^{A \frac{A_e}{A} U_e}}{A \cancel{\rho} A_e} - \frac{1}{2} \left(\frac{dh}{dt} \right)^2 \left(\frac{A}{A_e} \right)^2 U_e \cancel{A_e} = 0 \quad (7.48)$$

Notice that $\bar{U} = U_b f(G)$ and thus

$$\overbrace{\frac{f(G) U_b}{\bar{U}}} \frac{d\bar{U}}{dt} \frac{hA}{U_e A_e} + \frac{gh}{2} \frac{dh}{dt} \frac{hA}{U_e A_e} + \left[\frac{\bar{U}^2}{2} + \frac{gh}{2} \right] - \frac{1}{2} \left(\frac{dh}{dt} \right)^2 \left(\frac{A}{A_e} \right)^2 = 0 \quad (7.49)$$

Further rearranging to eliminate the “flow rate” transforms to

$$f(G) h \frac{d\bar{U}}{dt} \left(\frac{U_b A}{U_e A_e} \right) + \frac{gh}{2} \frac{dh}{dt} \frac{A}{U_e A_e} + \left[\frac{f(G)^2}{2} \left(\frac{dh}{dt} \right)^2 + \frac{gh}{2} \right] - \frac{1}{2} \left(\frac{dh}{dt} \right)^2 \left(\frac{A}{A_e} \right)^2 = 0 \quad (7.50)$$

$$f(G)^2 h \frac{d^2 h}{dt^2} + \frac{gh}{2} + \left[\frac{f(G)^2}{2} \left(\frac{dh}{dt} \right)^2 + \frac{gh}{2} \right] - \frac{1}{2} \left(\frac{dh}{dt} \right)^2 \left(\frac{A}{A_e} \right)^2 = 0 \quad (7.51)$$

¹⁵This part can be skipped to end of “advanced material”.

***** End Advance material *****

Combining the gh terms into one yields

$$f(G)^2 h \frac{d^2 h}{dt^2} + g h + \frac{1}{2} \left(\frac{dh}{dt} \right)^2 \left[f(G)^2 - \left(\frac{A}{A_e} \right)^2 \right] = 0 \quad (7.52)$$

Defining a new tank emptying parameter, T_e , as

$$T_e = \left(\frac{A}{f(G) A_e} \right)^2 \quad (7.53)$$

This parameter represents the characteristics of the tank which controls the emptying process. Dividing equation (7.52) by $f(G)^2$ and using this parameter, equation (7.52) after minor rearrangement transformed to

$$h \left(\frac{d^2 h}{dt^2} + \frac{g A_e^2}{T_e A^2} \right) + \frac{1}{2} \left(\frac{dh}{dt} \right)^2 [1 - T_e] = 0 \quad (7.54)$$

The solution can either of these equations¹⁶

$$- \int \frac{dh}{\sqrt{\frac{(k_1 T_e - 2 k_1) e^{\ln(h) T_e} + 2 g h^2}{h (T_e - 2) f(G)}}} = t + k_2 \quad (7.55)$$

or

$$\int \frac{dh}{\sqrt{\frac{(k_1 T_e - 2 k_1) e^{\ln(h) T_e} + 2 g h^2}{h (T_e - 2) f(G)}}} = t + k_2 \quad (7.56)$$

The solution with the positive solution has no physical meaning because the height cannot increase with time. Thus define function of the height as

$$f(h) = - \int \frac{dh}{\sqrt{\frac{(k_1 T_e - 2 k_1) e^{\ln(h) T_e} + 2 g h^2}{h (T_e - 2) f(G)}}} \quad (7.57)$$

¹⁶A discussion about this equation appear in the mathematical appendix.

The initial condition for this case are: one the height initial is

$$h(0) = h_0 \quad (7.58)$$

The initial boundary velocity is

$$\frac{dh}{dt} = 0 \quad (7.59)$$

This condition pose a physical limitation¹⁷ which will be ignored. The first condition yields

$$k_2 = -f(h_0) \quad (7.60)$$

The second condition provides

$$\frac{dh}{dt} = 0 = \sqrt{\frac{(k_1 T_e - 2 k_1) e^{\ln(h_0) T_e} + 2 g h_0^2}{h_0 (T_e - 2) f(G)}} \quad (7.61)$$

The complication of the above solution suggest a simplification in which

$$\frac{d^2 h}{dt^2} \ll \frac{g A_e^2}{T_e A^2} \quad (7.62)$$

which reduces equation (7.54) into

$$h \left(\frac{g A_e^2}{T_e A^2} \right) + \frac{1}{2} \left(\frac{dh}{dt} \right)^2 [1 - T_e] = 0 \quad (7.63)$$

While equation (7.63) is still non linear equation, the non linear element can be removed by taking negative branch (height reduction) of the equation as

$$\left(\frac{dh}{dt} \right)^2 = \frac{2 g h}{-1 + \left(\frac{A}{A_e} \right)^2} \quad (7.64)$$

It can be noticed that T_e “disappeared” from the equation. And taking the “positive” branch

$$\frac{dh}{dt} = \frac{\sqrt{2 g h}}{\sqrt{1 - \left(\frac{A}{A_e} \right)^2}} \quad (7.65)$$

The nature of first order Ordinary Differential Equation that they allow only one initial condition. This initial condition is the initial height of the liquid. The initial velocity

¹⁷For the initial condition speed of sound has to be taken into account. Thus for a very short time, the information about opening of the valve did not reached to the surface. This information travel in characteristic sound speed which is over 1000 *m/sec*. However, if this phenomenon is ignored this solution is correct.

field was eliminated by the approximation (remove the acceleration term). Thus it is assumed that the initial velocity is not relevant at the core of the process at hand. It is correct only for large ratio of h/r and the error became very substantial for small value of h/r .

Equation (7.65) integrated to yield

$$\left(1 - \left(\frac{A}{A_e}\right)^2\right) \int_{h_0}^h \frac{dh}{\sqrt{2gh}} = \int_0^t dt \quad (7.66)$$

The initial condition has been inserted into the integral which its solution is

$$\left(1 - \left(\frac{A}{A_e}\right)^2\right) \frac{h - h_0}{\sqrt{2gh}} = t \quad (7.67)$$

$$U_e = \frac{dh}{dt} \frac{A}{A_e} = \frac{\sqrt{2gh}}{\sqrt{1 - \left(\frac{A}{A_e}\right)^2}} \frac{A}{A_e} = \frac{\sqrt{2gh}}{\sqrt{1 - \left(\frac{A_e}{A}\right)^2}} \quad (7.68)$$

If the area ratio $A_e/A \ll 1$ then

$$U \cong \sqrt{2gh} \quad (7.69)$$

Equation (7.69) is referred in the literature as Torricelli's equation¹⁸

This analysis has several drawbacks which limits the accuracy of the calculations. Yet, this analysis demonstrates the usefulness of the integral analysis to provide a reasonable solution. This analysis can be improved by experimental investigating the phenomenon. The experimental coefficient can be added to account for the dissipation and other effects such

$$\frac{dh}{dt} \cong C \sqrt{2gh} \quad (7.70)$$

7.2 Limitation of Integral Approach

Some of accuracy issues to enhance the quality and improvements of the integral method were suggested in the analysis of the emptying tank. There are problems that the integral methods even with these enhancements simply cannot tackle.

The improvements to the integral methods are the corrections to the estimates of the energy or other quantities in the conservation equations. In the calculations

¹⁸Evangelista Torricelli (October 15, 1608 – October 25, 1647) was an Italian physicist and mathematician. He derived this equation based on similar principle to Bernoulli equation (which later leads to Bernoulli's equation). Today the exact reference to his work is lost only "sketches" of his lecture elude work. He was student (not formal) and follower of Galileo Galilei. It seems that Torricelli was an honest man who gave to others and he died at young age of 39 while in his prime.

of the exit velocity of a tank, two such corrections were presented. The first type is the prediction of the velocities profile (or the concentration profile). The second type of corrections is the understanding that averaged of the total field is different from the averaged of different zooms. In the case of the tank, the averaged velocity in x direction is zero yet the averaged velocity in the two zooms (two halves) is not zero. In fact, the averaged energy in the x direction contributes or effects the energy equation. The accuracy issues that integral methods intrinsically suffers from no ability to exact flow field and thus lost the accuracy as was discussed in the example. The integral method does not handle the problems such as the free surface with reasonable accuracy. Furthermore, the knowledge of whether the flow is laminar or turbulent (later on this issue) has to come from different techniques. Hence the prediction can skew the actual predictions.

In the analysis of the tank it was assumed that the dissipation can be ignored. In cases that dissipation play major role, the integral does not provide a sufficient tool to analyze the issue at hand. For example, the analysis of the oscillating manometer cannot be carried by the integral methods. A liquid in manometer is disturbed from a rest by a distance of H_0 . The description $H(t)$ as a function of time requires exact knowledge of the velocity field. Additionally, the integral methods is too crude to handle issues of free interface. These problem were minor for the emptying the tank but for the oscillating manometer it is the core of the problem. Hence different techniques are required.

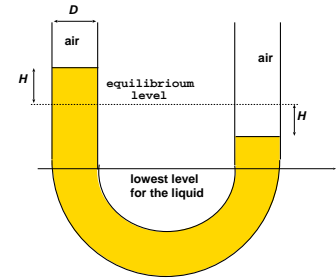


Fig. -7.4. The work on the control volume is done by two different mechanisms

The discussion on the limitations was not provided to discard usage of this method but rather to provide a guidance of use with caution. The integral method is a powerful and yet simple method but has to be used with the limitations of the method in mind.

7.3 Approximation of Energy Equation

The emptying the tank problem was complicated even with all the simplifications that were carried. Engineers in order to reduce the work further simplify the energy equation. It turn out that these simplifications can provide reasonable results and key understanding of the physical phenomena and yet with less work, the problems can be solved. The following sections provides further explanation.

7.3.1 Energy Equation in Steady State

The steady state situation provides several ways to reduce the complexity. The time derivative term can be eliminated since the time derivative is zero. The acceleration term must be eliminated for the obvious reason. Hence the energy equation is reduced to

$$\dot{Q} - \dot{W}_{shear} - \dot{W}_{shaft} = \int_S \left(h + \frac{U^2}{2} + gz \right) U_{rn} \rho dA + \int_S P U_{bn} dA \quad (7.71)$$

If the flow is uniform or can be estimated as uniform, equation (7.71) is reduced to

$$\dot{Q} - \dot{W}_{shear} - \dot{W}_{shaft} = \left(h + \frac{U^2}{2} + gz \right) U_{rn} \rho A_{out} - \left(h + \frac{U^2}{2} + gz \right) U_{rn} \rho A_{in} + P U_{bn} A_{out} - P U_{bn} A_{in} \quad (7.72)$$

It can be noticed that last term in equation (7.72) for non-deformable control volume does not vanished. The reason is that while the velocity is constant, the pressure is different. For a stationary fix control volume the energy equation, under this simplification transformed to

$$\dot{Q} - \dot{W}_{shear} - \dot{W}_{shaft} = \left(h + \frac{U^2}{2} + gz \right) U_{rn} \rho A_{out} - \left(h + \frac{U^2}{2} + gz \right) U_{rn} \rho A_{in} \quad (7.73)$$

Dividing equation the mass flow rate provides

$$\dot{q} - \dot{w}_{shear} - \dot{w}_{shaft} = \left(h + \frac{U^2}{2} + gz \right) \Big|_{out} - \left(h + \frac{U^2}{2} + gz \right) \Big|_{in} \quad (7.74)$$

7.3.2 Energy Equation in Frictionless Flow and Steady State

In cases where the flow can be estimated without friction or where a quick solution is needed the friction and other losses are illuminated from the calculations. This imaginary fluid reduces the amount of work in the calculations and Ideal Flow Chapter is dedicated in this book. The second law is the core of "no losses" and can be employed when calculations of this sort information is needed. Equation (2.21) which can be written as

$$dq_{rev} = T ds = dE_u + P dv \quad (7.75)$$

Using the multiplication rule change equation (7.75)

$$dq_{rev} = dE_u + d(Pv) - v dP = dE_u + d\left(\frac{P}{\rho}\right) - v dP \quad (7.76)$$

integrating equation (7.76) yields

$$\int dq_{rev} = \int dE_u + \int d\left(\frac{P}{\rho}\right) - \int v dP \quad (7.77)$$

$$q_{rev} = E_u + \left(\frac{P}{\rho}\right) - \int \frac{dP}{\rho} \quad (7.78)$$

Integration over the entire system results in

$$Q_{rev} = \int_V \overbrace{\left(E_u + \left(\frac{P}{\rho}\right)\right)}^h \rho dV - \int_V \left(\int \frac{dP}{\rho}\right) \rho dV \quad (7.79)$$

Taking time derivative of the equation (7.79) becomes

$$\dot{Q}_{rev} = \frac{D}{Dt} \int_V \overbrace{\left(E_u + \left(\frac{P}{\rho}\right)\right)}^h \rho dV - \frac{D}{Dt} \int_V \left(\int \frac{dP}{\rho}\right) \rho dV \quad (7.80)$$

Using the Reynolds Transport Theorem to transport equation to control volume results in

$$\dot{Q}_{rev} = \frac{d}{dt} \int_V h \rho dV + \int_A h U_{rn} \rho dA + \frac{D}{Dt} \int_V \left(\int \frac{dP}{\rho}\right) \rho dV \quad (7.81)$$

As before equation (7.80) can be simplified for uniform flow as

$$\dot{Q}_{rev} = \dot{m} \left[(h_{out} - h_{in}) - \left(\int \frac{dP}{\rho} \Big|_{out} - \int \frac{dP}{\rho} \Big|_{in} \right) \right] \quad (7.82)$$

or

$$\dot{q}_{rev} = (h_{out} - h_{in}) - \left(\int \frac{dP}{\rho} \Big|_{out} - \int \frac{dP}{\rho} \Big|_{in} \right) \quad (7.83)$$

Subtracting equation (7.83) from equation (7.74) results in

$$0 = w_{shaft} + \overbrace{\left(\int \frac{dP}{\rho} \Big|_2 - \int \frac{dP}{\rho} \Big|_1 \right)}^{\text{change in pressure energy}} + \overbrace{\frac{U_2^2 - U_1^2}{2}}^{\text{change in kinetic energy}} + \overbrace{g(z_2 - z_1)}^{\text{change in potential energy}} \quad (7.84)$$

Equation (7.84) for constant density is

$$0 = w_{shaft} + \frac{P_2 - P_1}{\rho} + \frac{U_2^2 - U_1^2}{2} + g(z_2 - z_1) \quad (7.85)$$

For no shaft work equation (7.85) reduced to

$$0 = \frac{P_2 - P_1}{\rho} + \frac{U_2^2 - U_1^2}{2} + g(z_2 - z_1) \quad (7.86)$$

7.4 Energy Equation in Accelerated System

In the discussion so far, it was assumed that the control volume is at rest. The only acceptance to the above statement, is the gravity that was compensated by the gravity potential. In building the gravity potential it was assumed that the gravity is a conservative force. It was pointed earlier in this book that accelerated forces can be translated to potential force. In many cases, the control volume is moving in accelerated coordinates. These accelerations will be translated to potential energy.

The accelerations are referring to two kinds of acceleration, linear and rotational. There is no conceptual difference between these two accelerations. However, the mathematical treatment is somewhat different which is the reason for the separation. General Acceleration can be broken into a linear acceleration and a rotating acceleration.

7.4.1 Energy in Linear Acceleration Coordinate

The potential is defined as

$$P.E. = - \int_{ref}^2 \mathbf{F} \cdot d\mathbf{l} \quad (7.87)$$

In Chapter 3 a discussion about gravitational energy potential was presented. For example, for the gravity force is

$$F = - \frac{G M m}{r^2} \quad (7.88)$$

Where G is the gravity coefficient and M is the mass of the Earth. r and m are the distance and mass respectively. The gravity potential is then

$$PE_{gravity} = - \int_{\infty}^r - \frac{G M m}{r^2} dr \quad (7.89)$$

The reference was set to infinity. The gravity force for fluid element in small distance then is $g dz dm$. The work this element moving from point 1 to point 2 is

$$\int_1^2 g dz dm = g (z_2 - z_1) dm \quad (7.90)$$

The total work or potential is the integral over the whole mass.

7.4.2 Linear Accelerated System

The acceleration can be employed in similar fashion as the gravity force. The linear acceleration “creates” a conservative force of constant force and direction. The “potential” of moving the mass in the field provides the energy. The Force due to the acceleration of the field can be broken into three coordinates. Thus, the element of the potential is

$$dPE_a = \mathbf{a} \cdot d\mathbf{l} dm \quad (7.91)$$

The total potential for element material

$$PE_a = \int_{(0)}^{(1)} \mathbf{a} \cdot d\mathbf{l} dm = (a_x (x_1 - x_0) a_y (y_1 - y_0) a_z (z_1 - z_0)) dm \quad (7.92)$$

At the origin (of the coordinates) $x = 0$, $y = 0$, and $z = 0$. Using this trick the notion of the $a_x (x_1 - x_0)$ can be replaced by $a_x x$. The same can be done for the other two coordinates. The potential of unit material is

$$PE_{a\text{total}} = \int_{sys} (a_x x + a_y y + a_z z) \rho dV \quad (7.93)$$

The change of the potential with time is

$$\frac{D}{Dt} PE_{a\text{total}} = \frac{D}{Dt} \int_{sys} (a_x x + a_y y + a_z z) dm \quad (7.94)$$

Equation can be added to the energy equation as

$$\dot{Q} - \dot{W} = \frac{D}{Dt} \int_{sys} \left[E_u + \frac{U^2}{2} + a_x x + a_y y + (a_z + g)z \right] \rho dV \quad (7.95)$$

The Reynolds Transport Theorem is used to transferred the calculations to control volume as

Energy Equation in Linear Accelerated Coordinate

$$\begin{aligned} \dot{Q} - \dot{W} = & \frac{d}{dt} \int_{cv} \left[E_u + \frac{U^2}{2} + a_x x + a_y y + (a_z + g)z \right] \rho dV \\ & + \int_{cv} \left(h + \frac{U^2}{2} + a_x x + a_y y + (a_z + g)z \right) U_{rn} \rho dA \\ & + \int_{cv} P U_{bn} dA \end{aligned} \quad (7.96)$$

7.4.3 Energy Equation in Rotating Coordinate System

The coordinate system rotating around fix axis creates similar conservative potential as the linear system. There are two kinds of acceleration due this rotation one is the

centrifugal and second the Coriolis force. To understand it better, consider a particle which moves with the our rotating system. The forces acting on particles are

$$\mathbf{F} = \left(\overbrace{\omega^2 r \hat{r}}^{\text{centrifugal}} + \overbrace{2\mathbf{U} \times \boldsymbol{\omega}}^{\text{Coriolis}} \right) dm \quad (7.97)$$

The work or the potential then is

$$PE = (\omega^2 r \hat{r} + 2\mathbf{U} \times \boldsymbol{\omega}) \cdot d\boldsymbol{\ell} dm \quad (7.98)$$

The cylindrical coordinate are

$$d\boldsymbol{\ell} = dr \hat{r} + r d\theta \hat{\theta} + dz \hat{k} \quad (7.99)$$

where \hat{r} , $\hat{\theta}$, and \hat{k} are units vector in the coordinates r , θ and z respectively. The potential is then

$$PE = (\omega^2 r \hat{r} + 2\mathbf{U} \times \boldsymbol{\omega}) \cdot (dr \hat{r} + r d\theta \hat{\theta} + dz \hat{k}) dm \quad (7.100)$$

The first term results in $\omega^2 r^2$ (see for explanation in the appendix 228 for vector explanation). The cross product is zero of

$$\mathbf{U} \times \boldsymbol{\omega} \times \mathbf{U} = \mathbf{U} \times \boldsymbol{\omega} \times \boldsymbol{\omega} = 0$$

because the first multiplication is perpendicular to the last multiplication. The second part is

$$(2\mathbf{U} \times \boldsymbol{\omega}) \cdot d\boldsymbol{\ell} dm \quad (7.101)$$

This multiplication does not vanish with the exception of the direction of \mathbf{U} . However, the most important direction is the direction of the velocity. This multiplication creates lines (surfaces) of constant values. From physical point of view, the flux of this property is important only in the direction of the velocity. Hence, this term canceled and does not contribute to the potential.

The net change of the potential energy due the centrifugal motion is

$$PE_{centrifugal} = - \int_1^2 \omega^2 r^2 dr dm = \frac{\omega^2 (r_1^2 - r_2^2)}{2} dm \quad (7.102)$$

Inserting the potential energy due to the centrifugal forces into the energy equation yields

Energy Equation in Accelerated Coordinate

$$\begin{aligned} \dot{Q} - \dot{W} = & \frac{d}{dt} \int_{cv} \left[E_u + \frac{U^2}{2} + a_x x + a_y y + (a_z + g)z - \frac{\omega^2 r^2}{2} \right] \rho dV \\ & + \int_{cv} \left(h + \frac{U^2}{2} + a_x x + a_y y + (a_z + g)z - \frac{\omega^2 r^2}{2} \right) U_{rn} \rho dA \\ & + \int_{cv} P U_{bn} dA \end{aligned} \quad (7.103)$$

7.4.4 Energy Equation in Accelerated Coordinate with Uniform Flow

One of the way to simplify the general equation (7.103) to assume uniform flow.

Part II

Differential Analysis

CHAPTER 8

Multi-Phase Flow

8.1 Introduction

Traditionally, the topic of multi-phase flow is ignored in an introductory class on fluid mechanics. For many engineers, this class will be the only opportunity to be exposed to this topic. The knowledge in this topic without any doubts, is required for many engineering problems. Calculations of many kinds of flow deals with more than one phase or material flow¹. The author believes that the trends and effects of multiphase flow could and should be introduced and considered by engineers. In the past, books on multiphase flow were written more as a literature review or heavy on the mathematics. It is recognized that multiphase flow is still evolving. In fact, there is not a consensus to the exact map of many flow regimes. This book attempts to describe these issues as a fundamentals of physical aspects and less as a literature review. This chapter provides information that is more or less in consensus². Additionally, the nature of multiphase flow requires solving many equations. Thus, in many books the representations is by writing the whole set governing equations. Here, it is believed that the interactions/calculations requires a full year class and hence, only the trends and simple calculations are described.

8.2 History

The study of multi-phase flow started for practical purposes after World War II. Initially the models were using simple assumptions. For simple models, there are two possibilities (1) the fluids/materials are flowing in well homogeneous mixed (where the main problem

¹An example, there was a Ph.D. working for the government who analyzed filing cavity with liquid metal (aluminum), who did not consider the flow as two-phase flow and ignoring the air. As result, his analysis is in the twilight zone not in the real world.

²Or when the scientific principles simply dictate.

to find the viscosity), (2) the fluids/materials are flowing separately where the actual total loss pressure can be correlated based on the separate pressure loss of each of the material. If the pressure loss was linear then the total loss will be the summation of the two pressure losses (of the lighter liquid (gas) and the heavy liquid). Under this assumption the total is not linear and experimental correlation was made. The flow patterns or regimes were not considered. This was suggested by Lockhart and Martinelli who use a model where the flow of the two fluids are independent of each other. They postulate that there is a relationship between the pressure loss of a single phase and combine phases pressure loss as a function of the pressure loss of the other phase. It turned out this idea provides a good crude results in some cases.

Researchers that followed Lockhart and Martinelli looked for a different map for different combination of phases. When it became apparent that specific models were needed for different situations, researchers started to look for different flow regimes and provided different models. Also the researchers looked at the situation when the different regimes are applicable. Which leads to the concept of flow regime maps. Taitle and Duckler suggested a map based on five non-dimensional groups which are considered as the most useful today. However, Taitle and Duckler's map is not universal and it is only applied to certain liquid-gas conditions. For example, Taitle-Duckler's map is not applicable for microgravity.

8.3 *What to Expect From This Chapter*

As oppose to the tradition of the other chapters in this book and all other Potto project books, a description of what to expect in this chapter is provided. It is an attempt to explain and convince all the readers that the multi-phase flow must be included in introductory class on fluid mechanics³. Hence, this chapter will explain the core concepts of the multiphase flow and their relationship, and importance to real world.

This chapter will provide: a category of combination of phases, the concept of flow regimes, multi-phase flow parameters definitions, flow parameters effects on the flow regimes, partial discussion on speed of sound of different regimes, double choking phenomenon (hopefully), and calculation of pressure drop of simple homogeneous model. This chapter will introduce these concepts so that the engineer not only be able to understand a conversation on multi-phase but also, and more importantly, will know and understand the trends. However, this chapter will not provide a discussion of transient problems, phase change or transfer processes during flow, and actual calculation of pressure of the different regimes.

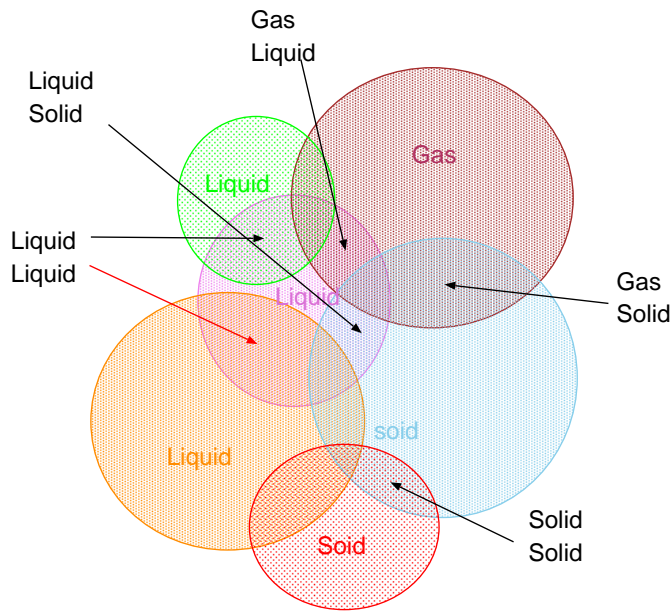


Fig. -8.1. Different fields of multi phase flow.

8.4 Kind of Multi-Phase Flow

All the flows are a form of multiphase flow. The discussion in the previous chapters is only as approximation when multiphase can be “reduced” into a single phase flow. For example, consider air flow that was discussed and presented earlier as a single phase flow. Air is not a pure material but a mixture of many gases. In fact, many properties of air are calculated as if the air is made of well mixed gases of Nitrogen and Oxygen. The results of the calculations of a mixture do not change much if it is assumed that the air flow as stratified flow⁴ of many concentration layers (thus, many layers (infinite) of different materials). Practically for many cases, the homogeneous assumption is enough and suitable. However, this assumption will not be appropriate when the air is stratified because of large body forces, or a large acceleration. Adopting this assumption might lead to a larger error. Hence, there are situations when air flow has to be considered as multiphase flow and this effect has to be taken into account.

In our calculation, it is assumed that air is made of only gases. The creation

³This author feels that he is in a unique position to influence many in the field of fluid mechanics. This fact is due to the sheer number of the downloaded Potta books. The number of the downloads of the book on Fundamental of compressible flow has exceed more than 100,000 in about two and half years. It also provides an opportunity to bring the latest advances in the fields since this author does not need to “sell” the book to a publisher or convince a “committee.”

⁴Different concentration of oxygen as a function of the height. While the difference of the concentration between the top to bottom is insignificant, nonetheless it exists.

of clean room is a proof that air contains small particles. In almost all situations, the cleanness of the air or the fact that air is a mixture is ignored. The engineering accuracy is enough to totally ignore it. Yet, there are situations where cleanness of the air can affect the flow. For example, the cleanness of air can reduce the speed of sound. In the past, the breaks in long trains were activated by reduction of the compressed line (a patent no. 360070 issued to George Westinghouse, Jr., March 29, 1887). In a four (4) miles long train, the breaks would started to work after about 20 seconds in the last wagon. Thus, a 10% change of the speed of sound due to dust particles in air could reduce the stopping time by 2 seconds (50 meter difference in stopping) and can cause an accident.

One way to categorize the multiphase is by the materials flows, For example, the flow of oil and water in one pipe is a multiphase flow. This flow is used by engineers to reduce the cost of moving crude oil through a long pipes system. The “average” viscosity is meaningless since in many cases the water follows around the oil. The water flow is the source of the friction. However, it is more common to categorize the flow by the distinct phases that flow in the tube. Since there are three phases, they can be solid–liquid, solid–gas, liquid–gas and solid–liquid–gas flow. This notion eliminates many other flow categories that can and should be included in multiphase flow. This category should include any distinction of phase/material. There are many more categories, for example, sand and grain (which are “solids”) flow with rocks and is referred to solid–solid flow. The category of liquid–gas should be really viewed as the extreme case of liquid-liquid where the density ratio is extremely large. The same can be said for gas–gas flow. For the gas, the density is a strong function of the temperature and pressure. Open Channel flow is, although important, is only an extreme case of liquid-gas flow and is a sub category of the multiphase flow.

The multiphase is an important part of many processes. The multiphase can be found in nature, living bodies (bio–fluids), and industries. Gas–solid can be found in sand storms, and avalanches. The body inhales solid particle with breathing air. Many industries are involved with this flow category such as dust collection, fluidized bed, solid propellant rocket, paint spray, spray casting, plasma and river flow with live creatures (small organisms to large fish) flow of ice berg, mud flow etc. The liquid–solid, in nature can be blood flow, and river flow. This flow also appears in any industrial process that are involved in solidification (for example die casting) and in moving solid particles. Liquid–liquid flow is probably the most common flow in the nature. Flow of air is actually the flow of several light liquids (gases). Many natural phenomenon are multiphase flow, for an example, rain. Many industrial process also include liquid-liquid such as painting, hydraulic with two or more kind of liquids.

8.5 Classification of Liquid-Liquid Flow Regimes

The general discussion on liquid–liquid will be provided and the gas–liquid flow will be discussed as a special case. Generally, there are two possibilities for two different materials to flow (it is also correct for solid–liquid and any other combination). The materials can flow in the same direction and it is referred as co–current flow. When the

materials flow in the opposite direction, it is referred as counter-current. In general, the co-current is the more common. Additionally, the counter-current flow must have special configurations of long length of flow. Generally, the counter-current flow has a limited length window of possibility in a vertical flow in conduits with the exception of magnetohydrodynamics. The flow regimes are referred to the arrangement of the fluids.

The main difference between the liquid-liquid flow to gas-liquid flow is that gas density is extremely lighter than the liquid density. For example, water and air flow as oppose to water and oil flow. The other characteristic that is different between the gas flow and the liquid flow is the variation of the density. For example, a reduction of the pressure by half will double the gas volumetric flow rate while the change in the liquid is negligible. Thus, the flow of gas-liquid can have several flow regimes in one situation while the flow of liquid-liquid will (probably) have only one flow regime.

8.5.1 Co-Current Flow

In Co-Current flow, two liquids can have three main categories: vertical, horizontal, and what ever between them. The vertical configuration has two cases, up or down. It is common to differentiate between the vertical (and near vertical) and horizontal (and near horizontal). There is no exact meaning to the word "near vertical" or "near horizontal" and there is no consensus on the limiting angles (not to mention to have limits as a function with any parameter that determine the limiting angle). The flow in inclined angle (that not covered by the word "near") exhibits flow regimes not much different from the other two. Yet, the limits between the flow regimes are considerably different. This issue of incline flow will not be covered in this chapter.

8.5.1.1 Horizontal Flow

The typical regimes for horizontal flow are stratified flow (open channel flow, and non open channel flow), dispersed bubble flow, plug flow, and annular flow. For low velocity (low flow rate) of the two liquids, the heavy liquid flows on the bottom and lighter liquid flows on the top⁵ as depicted in Figure 8.2. This kind of flow regime is referred to as horizontal flow. When the flow rate of the lighter liquid is almost zero, the flow is referred to as open channel flow. This definition (open channel flow) continues for small amount of lighter liquid as long as the heavier flow can be calculated as open channel flow (ignoring the lighter liquid). The geometries (even the boundaries) of open channel flow are very diverse. Open channel flow appears in many nature (river) as well in industrial process such as the die casting process where liquid metal is injected into a cylinder (tube) shape. The channel flow will be discussed in a greater detail in Open Channel Flow chapter.

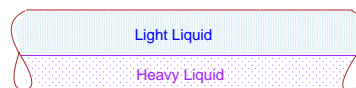


Fig. -8.2. Stratified flow in horizontal tubes when the liquids flow is very slow.

⁵With the exception of the extremely smaller diameter where Rayleigh-Taylor instability is an important issue.

As the lighter liquid (or the gas phase) flow rate increases (superficial velocity), the friction between the phases increase. The superficial velocity is referred to as the velocity that any phase will have if the other phase was not exist. This friction is one of the cause for the instability which manifested itself as waves and changing the surface from straight line to a different configuration (see Figure 8.3). The wave shape is created to keep the gas and the liquid velocity equal and at the same time to have shear stress to be balance by surface tension. The configuration of the cross section not only depend on the surface tension, and other physical properties of the fluids but also on the material of the conduit.

As the lighter liquid velocity increases two things can happen (1) wave size increase and (2) the shape of cross section continue to deform. Some referred to this regime as wavy stratified flow but this definition is not accepted by all as a category by itself. In fact, all the two phase flow are categorized by wavy flow which will proven later.

There are two paths that can occur on the heavier liquid flow rate. If the heavier flow rate is small, then the wave cannot reach to the crown and the shape is deformed to the point that all the heavier liquid is around the periphery. This kind of flow regime is referred to as annular flow. If the heavier liquid flow rate is larger⁶ than the distance, for the wave to reach the conduit crown is smaller. At some point, when the lighter liquid flow increases, the heavier liquid wave reaches to the crown of the pipe. At this stage, the flow pattern is referred to as slug flow or plug flow. Plug flow is characterized by regions of lighter liquid filled with drops of the heavier liquid with Plug (or Slug) of the heavier liquid (with bubble of the lighter liquid). These plugs are separated by large “chunks” that almost fill the entire tube. The plugs are flowing in a succession (see Figure 8.4). The pressure drop of this kind of regime is significantly larger than the stratified flow. The slug flow cannot be assumed to be as homogeneous flow nor it can exhibit some average viscosity. The “average” viscosity depends on the flow and thus making it as insignificant way to do the calculations. Further increase of the lighter liquid flow rate move the flow regime into annular flow. Thus, the possibility to go through slug flow regime depends on if there is enough liquid flow rate.

Choking occurs in compressible flow when the flow rate is above a certain point. All liquids are compressible to some degree. For liquid which the density is a strong and primary function of the pressure, choking occurs relatively closer/sooner. Thus, the flow that starts as a stratified flow will turned into a slug flow or stratified wavy⁷ flow after a certain distance depends on the heavy flow rate (if

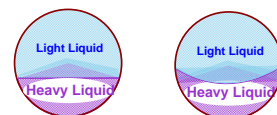


Fig. -8.3. Kind of Stratified flow in horizontal tubes.

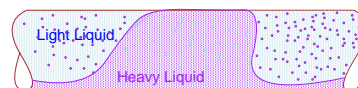


Fig. -8.4. Plug flow in horizontal tubes when the liquids flow is faster.

⁶The liquid level is higher.

⁷Well, all the flow is wavy, thus it is arbitrary definition.

this category is accepted). After a certain distance, the flow become annular or the flow will choke. The choking can occur before the annular flow regime is obtained depending on the velocity and compressibility of the lighter liquid. Hence, as in compressible flow, liquid-liquid flow has a maximum combined of the flow rate (both phases). This maximum is known as double choking phenomenon.

The reverse way is referred to the process where the starting point is high flow rate and the flow rate is decreasing. As in many fluid mechanics and magnetic fields, the return path is not move the exact same way. There is even a possibility to return on different flow regime. For example, flow that had slug flow in its path can be returned as stratified wavy flow. This phenomenon is refer to as hysteresis.

Flow that is under small angle from the horizontal will be similar to the horizontal flow. However, there is no consensus how far is the “near” means. Qualitatively, the “near” angle depends on the length of the pipe. The angle decreases with the length of the pipe. Besides the length, other parameters can affect the “near.”

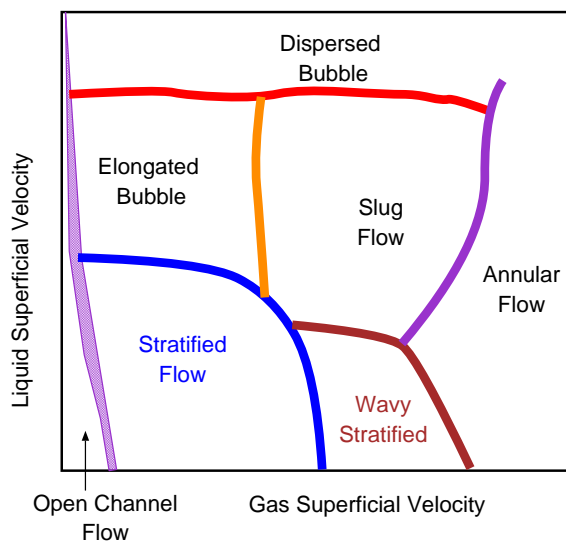


Fig. -8.5. Modified Mandhane map for flow regime in horizontal tubes.

The results of the above discussion are depicted in Figure 8.5. As many things in multiphase, this map is only characteristics of the “normal” conditions, e.g. in normal gravitation, weak to strong surface tension effects (air/water in “normal” gravity), etc.

8.5.1.2 Vertical Flow

The vertical flow has two possibilities, with the gravity or against it. In engineering application, the vertical flow against the gravity is more common used. There is a difference between flowing with the gravity and flowing against the gravity. The buoyancy

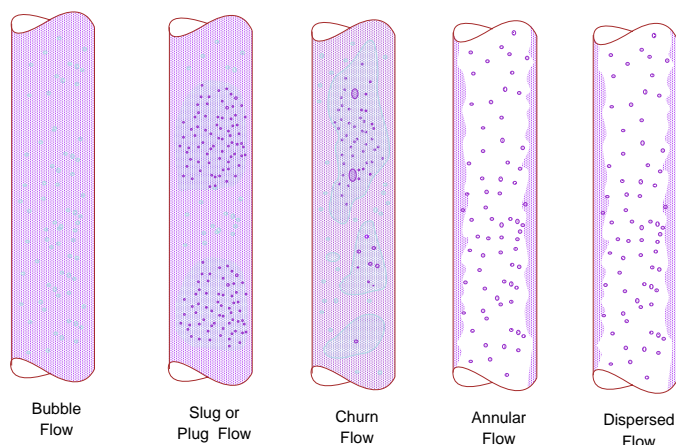


Fig. -8.6. Gas and liquid in Flow in vertical tube against the gravity.

is acting in two different directions for these two flow regimes. For the flow against gravity, the lighter liquid has a buoyancy that acts as an “extra force” to move it faster and this effect is opposite for the heavier liquid. The opposite is for the flow with gravity. Thus, there are different flow regimes for these two situations. The main reason that causes the difference is that the heavier liquid is more dominated by gravity (body forces) while the lighter liquid is dominated by the pressure driving forces.

Flow Against Gravity

For vertical flow against gravity, the flow cannot start as a stratified flow. The heavier liquid has to occupy almost the entire cross section before it can flow because of the gravity forces. Thus, the flow starts as a bubble flow. The increase of the lighter liquid flow rate will increase the number of bubbles until some bubbles start to collide. When many bubbles collide, they create a large bubble and the flow is referred to as slug flow or plug flow (see Figure 8.6). Notice, the different mechanism in creating the plug flow in horizontal flow compared to the vertical flow.

Further increase of lighter liquid flow rate will increase the slug size as more bubbles collide to create “super slug”; the flow regime is referred as elongated bubble flow. The flow is less stable as more turbulent flow and several “super slug” or churn flow appears in more chaotic way, see Figure 8.6. After additional increase of “super slug”, all these “elongated slug” unite to become an annular flow. Again, it can be noted the difference in the mechanism that create annular flow for vertical and horizontal flow. Any further increase transforms the outer liquid layer into bubbles in the inner liquid. Flow of near vertical against the gravity in two-phase does not deviate from vertical. The choking can occur at any point depends on the fluids and temperature and pressure.

8.5.1.3 Vertical Flow Under Micro Gravity

The above discussion mostly explained the flow in a vertical configuration when the surface tension can be neglected. In cases where the surface tension is very important. For example, out in space between gas and liquid (large density difference) the situation is different. The flow starts as dispersed bubble (some call it as “gas continuous”) because the gas phase occupies most of column. The liquid flows through a trickle or channeled flow that only partially wets part of the tube. The interaction between the phases is minimal and can be considered as the “open channel flow” of the vertical configuration. As the gas flow increases, the liquid becomes more turbulent and some parts enter into the gas phase as drops. When the flow rate of the gas increases further, all the gas phase change into tiny drops of liquid and this kind of regime referred to as mist flow. At a higher rate of liquid flow and a low flow rate of gas, the regime liquid fills the entire void and the gas is in small bubble and this flow referred to as bubbly flow. In the medium range of the flow rate of gas and liquid, there is pulse flow in which liquid is moving in frequent pulses. The common map is based on dimensionless parameters. Here, it is presented in a dimension form to explain the trends (see Figure 8.7). In the literature, Figure 8.7 presented in dimensionless coordinates. The abscissa is a function of combination of Froude, Reynolds, and Weber numbers. The ordinate is a combination of flow rate ratio and density ratio.

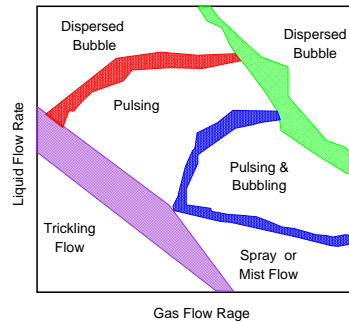


Fig. -8.7. A dimensional vertical flow map under very low gravity against the gravity.

Flow With The Gravity

As opposed to the flow against gravity, this flow can start with stratified flow. A good example for this flow regime is a water fall. The initial part for this flow is more significant. Since the heavy liquid can be supplied from the “wrong” point/side, the initial part has a larger section compared to the flow against the gravity flow. After the flow has settled, the flow continues in a stratified configuration. The transitions between the flow regimes is similar to stratified flow. However, the points where these transitions occur are different from the horizontal flow. While this author is not aware of an actual model, it must be possible to construct a model that connects this configuration with the stratified flow where the transitions will be dependent on the angle of inclinations.

8.6 Multi-Phase Flow Variables Definitions

Since the gas-liquid system is a specific case of the liquid-liquid system, both will be united in this discussion. However, for the convenience of the terms “gas and liquid” will be used to signify the lighter and heavier liquid, respectively. The liquid-liquid (also

gas–liquid) flow is an extremely complex three–dimensional transient problem since the flow conditions in a pipe may vary along its length, over its cross section, and with time. To simplify the descriptions of the problem and yet to retain the important features of the flow, some variables are defined so that the flow can be described as a one-dimensional flow. This method is the most common and important to analyze two-phase flow pressure drop and other parameters. Perhaps, the only serious missing point in this discussion is the change of the flow along the distance of the tube.

8.6.1 Multi–Phase Averaged Variables Definitions

The total mass flow rate through the tube is the sum of the mass flow rates of the two phases

$$\dot{m} = \dot{m}_G + \dot{m}_L \quad (8.1)$$

It is common to define the mass velocity instead of the regular velocity because the “regular” velocity changes along the length of the pipe. The gas mass velocity is

$$G_G = \frac{\dot{m}_G}{A} \quad (8.2)$$

Where A is the entire area of the tube. It has to be noted that this mass velocity does not exist in reality. The liquid mass velocity is

$$G_L = \frac{\dot{m}_L}{A} \quad (8.3)$$

The mass flow of the tube is then

$$G = \frac{\dot{m}}{A} \quad (8.4)$$

It has to be emphasized that this mass velocity is the actual velocity.

The volumetric flow rate is not constant (since the density is not constant) along the flow rate and it is defined as

$$Q_G = \frac{G_G}{\rho_G} = U_{sG} \quad (8.5)$$

and for the liquid

$$Q_L = \frac{G_L}{\rho_L} \quad (8.6)$$

For liquid with very high bulk modulus (almost constant density), the volumetric flow rate can be considered as constant. The total volumetric volume vary along the tube length and is

$$Q = Q_L + Q_G \quad (8.7)$$

Ratio of the gas flow rate to the total flow rate is called the 'quality' or the "dryness fraction" and is given by

$$X = \frac{\dot{m}_G}{\dot{m}} = \frac{G_G}{G} \quad (8.8)$$

In a similar fashion, the value of $(1 - X)$ is referred to as the "wetness fraction." The last two fractions remain constant along the tube length as long the gas and liquid masses remain constant. The ratio of the gas flow cross sectional area to the total cross sectional area is referred as the void fraction and defined as

$$\alpha = \frac{A_G}{A} \quad (8.9)$$

This fraction is vary along tube length since the gas density is not constant along the tube length. The liquid fraction or liquid holdup is

$$L_H = 1 - \alpha = \frac{A_L}{A} \quad (8.10)$$

It must be noted that Liquid holdup, L_H is not constant for the same reasons the void fraction is not constant.

The actual velocities depend on the other phase since the actual cross section the phase flows is dependent on the other phase. Thus, a superficial velocity is commonly defined in which if only one phase is using the entire tube. The gas superficial velocity is therefore defined as

$$U_{sG} = \frac{G_G}{\rho_G} = \frac{X \dot{m}}{\rho_G A} = Q_G \quad (8.11)$$

The liquid superficial velocity is

$$U_{sL} = \frac{G_L}{\rho_L} = \frac{(1 - X) \dot{m}}{\rho_L A} = Q_L \quad (8.12)$$

Since $U_{sL} = Q_L$ and similarly for the gas then

$$U_m = U_{sG} + U_{sL} \quad (8.13)$$

Where U_m is the averaged velocity. It can be noticed that U_m is not constant along the tube.

The average superficial velocity of the gas and liquid are different. Thus, the ratio of these velocities is referred to as the slip velocity and is defined as the following

$$SLP = \frac{U_G}{U_L} \quad (8.14)$$

Slip ratio is usually greater than unity. Also, it can be noted that the slip velocity is not constant along the tube.

For the same velocity of phases ($SLP = 1$), the mixture density is defined as

$$\rho_m = \alpha \rho_G + (1 - \alpha) \rho_L \quad (8.15)$$

This density represents the density taken at the “frozen” cross section (assume the volume is the cross section times infinitesimal thickness of dx).

The average density of the material flowing in the tube can be evaluated by looking at the definition of density. The density of any material is defined as $\rho = m/V$ and thus, for the flowing material it is

$$\rho = \frac{\dot{m}}{Q} \quad (8.16)$$

Where Q is the volumetric flow rate. Substituting equations (8.1) and (8.7) into equation (8.16) results in

$$\rho_{average} = \frac{\overbrace{X \dot{m}_G}^{\dot{m}_G} + \overbrace{(1-X) \dot{m}_L}^{\dot{m}_L}}{Q_G + Q_L} = \frac{X \dot{m}_G + (1-X) \dot{m}_L}{\underbrace{X \dot{m}_G}_{\frac{\rho_G}{Q_G}} + \underbrace{(1-X) \dot{m}_L}_{\frac{\rho_L}{Q_L}}} \quad (8.17)$$

Equation (8.17) can be simplified by canceling the \dot{m} and noticing the $(1-X) + X = 1$ to become

$$\rho_{average} = \frac{1}{\frac{X}{\rho_G} + \frac{(1-X)}{\rho_L}} \quad (8.18)$$

The average specific volume of the flow is then

$$v_{average} = \frac{1}{\rho_{average}} = \frac{X}{\rho_G} + \frac{(1-X)}{\rho_L} = X v_G + (1-X) v_L \quad (8.19)$$

The relationship between X and α is

$$X = \frac{\dot{m}_G}{\dot{m}_G + \dot{m}_L} = \frac{\rho_G U_G \overbrace{A_G}^{\alpha}}{\underbrace{\rho_L U_L A(1-\alpha)}_{A_L} + \rho_G U_G A \alpha} = \frac{\rho_G U_G \alpha}{\rho_L U_L (1-\alpha) + \rho_G U_G \alpha} \quad (8.20)$$

If the slip is one $SLP = 1$, thus equation (8.20) becomes

$$X = \frac{\rho_G \alpha}{\rho_L (1-\alpha) + \rho_G \alpha} \quad (8.21)$$

8.7 Homogeneous Models

Before discussing the homogeneous models, it is worthwhile to appreciate the complexity of the flow. For the construction of fluid basic equations, it was assumed that the flow is continuous. Now, this assumption has to be broken, and the flow is continuous only in many chunks (small segments). Furthermore, these segments are not defined but results of the conditions imposed on the flow. In fact, the different flow regimes are examples of typical configuration of segments of continuous flow. Initially, it was assumed that the different flow regimes can be neglected at least for the pressure loss (not correct for the heat transfer). The single phase was studied earlier in this book and there is a considerable amount of information about it. Thus, the simplest is to use it for approximation.

The average velocity (see also equation (8.13)) is

$$U_m = \frac{Q_L + Q_G}{A} = U_{sL} + U_{sG} = U_m \quad (8.22)$$

It can be noted that the continuity equation is satisfied as

$$\dot{m} = \rho_m U_m A \quad (8.23)$$

Example 8.1:

Under what conditions equation (8.23) is correct?

SOLUTION

Under construction

End Solution

The governing momentum equation can be approximated as

$$\dot{m} \frac{dU_m}{dx} = -A \frac{dP}{dx} - S \tau_w - A \rho_m g \sin \theta \quad (8.24)$$

or modifying equation (8.24) as

$$-\frac{dP}{dx} = -\frac{S}{A} \tau_w - \frac{\dot{m}}{A} \frac{dU_m}{dx} + \rho_m g \sin \theta \quad (8.25)$$

The energy equation can be approximated as

$$\frac{dq}{dx} - \frac{dw}{dx} = \dot{m} \frac{d}{dx} \left(h_m + \frac{U_m^2}{2} + g x \sin \theta \right) \quad (8.26)$$

8.7.1 Pressure Loss Components

In a tube flowing upward in incline angle θ , the pressure loss is affected by friction loss, acceleration, and body force (gravitation). These losses are non-linear and depend on each other. For example, the gravitation pressure loss reduce the pressure and thus the density must change and hence, acceleration must occur. However, for small distances (dx) and some situations, this dependency can be neglected. In that case, from equation (8.25), the total pressure loss can be written as

$$\frac{dP}{dx} = \overbrace{\left. \frac{dP}{dx} \right|_f}^{\text{friction}} + \overbrace{\left. \frac{dP}{dx} \right|_a}^{\text{acceleration}} + \overbrace{\left. \frac{dP}{dx} \right|_g}^{\text{gravity}} \quad (8.27)$$

Every part of the total pressure loss will be discussed in the following section.

8.7.1.1 Friction Pressure Loss

The frictional pressure loss for a conduit can be calculated as

$$-\left. \frac{dP}{dx} \right|_f = \frac{S}{A} \tau_w \quad (8.28)$$

Where S is the perimeter of the fluid. For calculating the frictional pressure loss in the pipe is

$$-\left. \frac{dP}{dx} \right|_f = \frac{4 \tau_w}{D} \quad (8.29)$$

The wall shear stress can be estimated by

$$\tau_w = f \frac{\rho_m U_m^2}{2} \quad (8.30)$$

The friction factor is measured for a single phase flow where the average velocity is directly related to the wall shear stress. There is not available experimental data for the relationship of the averaged velocity of the two (or more) phases and wall shear stress. In fact, this friction factor was not measured for the “averaged” viscosity of the two phase flow. Yet, since there isn’t anything better, the experimental data that was developed and measured for single flow is used.

The friction factor is obtained by using the correlation

$$f = C \left(\frac{\rho_m U_m D}{\mu_m} \right)^{-n} \quad (8.31)$$

Where C and n are constants which depend on the flow regimes (turbulent or laminar flow). For laminar flow $C = 16$ and $n = 1$. For turbulent flow $C = 0.079$ and $n = 0.25$.

There are several suggestions for the average viscosity. For example, Duckler suggest the following

$$\mu_m = \frac{\mu_G Q_G}{Q_G + Q_L} + \frac{\mu_L Q_L}{Q_G + Q_L} \quad (8.32)$$

Duckler linear formula does not provide always good approximation and Cichilli suggest similar to equation (8.18) average viscosity as

$$\mu_{average} = \frac{1}{\frac{X}{\mu_G} + \frac{(1-X)}{\mu_L}} \quad (8.33)$$

Or simply make the average viscosity depends on the mass fraction as

$$\mu_m = X \mu_G + (1 - X) \mu_L \quad (8.34)$$

Using this formula, the friction loss can be estimated.

8.7.1.2 Acceleration Pressure Loss

The acceleration pressure loss can be estimated by

$$-\left. \frac{dP}{dx} \right|_a = \dot{m} \frac{dU_m}{dx} \quad (8.35)$$

The acceleration pressure loss (can be positive or negative) results from change of density and the change of cross section. Equation (8.35) can be written as

$$-\left. \frac{dP}{dx} \right|_a = \dot{m} \frac{d}{dx} \left(\frac{\dot{m}}{A \rho_m} \right) \quad (8.36)$$

Or in an explicit way equation (8.36) becomes

$$-\left. \frac{dP}{dx} \right|_a = \dot{m}^2 \left[\underbrace{\frac{1}{A} \frac{d}{dx} \left(\frac{1}{\rho_m} \right)}_{\text{pressure loss due to density change}} + \underbrace{\frac{1}{\rho_m A^2} \frac{dA}{dx}}_{\text{pressure loss due to area change}} \right] \quad (8.37)$$

There are several special cases. The first case where the cross section is constant, $dA/dx = 0$. In second case is where the mass flow rates of gas and liquid is constant in which the derivative of X is zero, $dX/dx = 0$. The third special case is for constant density of one phase only, $d\rho_L/dx = 0$. For the last point, the private case is where densities are constant for both phases.

8.7.1.3 Gravity Pressure Loss

Gravity was discussed in Chapter 4 and is

$$\left. \frac{dP}{dx} \right|_g = g \rho_m \sin \theta \quad (8.38)$$

The density change during the flow can be represented as a function of density. The density in equation (8.38) is the density without the “movement” (the “static” density).

8.7.1.4 Total Pressure Loss

The total pressure between two points, (a and b) can be calculated with integration as

$$\Delta P_{ab} = \int_a^b \frac{dP}{dx} dx \quad (8.39)$$

and therefore

$$\Delta P_{ab} = \overbrace{\Delta P_{abf}}^{\text{friction}} + \overbrace{\Delta P_{aba}}^{\text{acceleration}} + \overbrace{\Delta P_{abg}}^{\text{gravity}} \quad (8.40)$$

8.7.2 Lockhart Martinelli Model

The second method is by assumption that every phase flow separately. One such popular model by Lockhart and Martinelli⁸. Lockhart and Martinelli built model based on the assumption that the separated pressure loss are independent from each other. Lockhart Martinelli parameters are defined as the ratio of the pressure loss of two phases and pressure of a single phase. Thus, there are two parameters as shown below.

$$\phi_G = \sqrt{\left. \frac{dP}{dx} \right|_{TP} / \left. \frac{dP}{dx} \right|_{SG}} \Big|_f \quad (8.41)$$

Where the TP denotes the two phases and SG denotes the pressure loss for the single gas phase. Equivalent definition for the liquid side is

$$\phi_L = \sqrt{\left. \frac{dP}{dx} \right|_{TP} / \left. \frac{dP}{dx} \right|_{SL}} \Big|_f \quad (8.42)$$

Where the SL denotes the pressure loss for the single liquid phase.

⁸This method was considered a military secret, private communication with Y., Taitle

The ratio of the pressure loss for a single liquid phase and the pressure loss for a single gas phase is

$$\Xi = \sqrt{\left. \frac{dP}{dx} \right|_{SL} / \left. \frac{dP}{dx} \right|_{SG}} \quad (8.43)$$

where Ξ is Martinelli parameter.

It is assumed that the pressure loss for both phases are equal.

$$\left. \frac{dP}{dx} \right|_{SG} = \left. \frac{dP}{dx} \right|_{SL} \quad (8.44)$$

The pressure loss for the liquid phase is

$$\left. \frac{dP}{dx} \right|_L = \frac{2 f_L U_L^2 \rho_l}{D_L} \quad (8.45)$$

For the gas phase, the pressure loss is

$$\left. \frac{dP}{dx} \right|_G = \frac{2 f_G U_G^2 \rho_l}{D_G} \quad (8.46)$$

Simplified model is when there is no interaction between the two phases.

To insert the Diagram.

8.8 Solid-Liquid Flow

Solid-liquid system is simpler to analyze than the liquid-liquid system. In solid-liquid, the effect of the surface tension are very minimal and can be ignored. Thus, in this discussion, it is assumed that the surface tension is insignificant compared to the gravity forces. The word "solid" is not really mean solid but a combination of many solid particles. Different combination of solid particle creates different "liquid." Therefore, there will be a discussion about different particle size and different geometry (round, cubic, etc). The uniformity is categorizing the particle sizes, distribution, and geometry. For example, analysis of small coal particles in water is different from large coal particles in water.

The density of the solid can be above or below the liquid. Consider the case where the solid is heavier than the liquid phase. It is also assumed that the "liquids" density does not change significantly and it is far from the choking point. In that case there are four possibilities for vertical flow:

1. The flow with the gravity and lighter density solid particles.
2. The flow with the gravity and heavier density solid particles.
3. The flow against the gravity and lighter density solid particles.

4. The flow against the gravity and heavier density solid particles.

All these possibilities are different. However, there are two sets of similar characteristics, possibility, 1 and 4 and the second set is 2 and 3. The first set is similar because the solid particles are moving faster than the liquid velocity and vice versa for the second set (slower than the liquid). The discussion here is about the last case (4) because very little is known about the other cases.

8.8.1 Solid Particles with Heavier Density $\rho_S > \rho_L$

Solid-liquid flow has several combination flow regimes.

When the liquid velocity is very small, the liquid cannot carry the solid particles because there is not enough resistance to lift up the solid particles. A particle in a middle of the vertical liquid flow experience several forces. The force balance of spherical particle in field viscous fluid (creeping flow) is

$$\underbrace{\frac{\pi D^3 g (\rho_S - \rho_L)}{6}}_{\text{gravity and buoyancy forces}} = \underbrace{\frac{C_{D\infty} \pi D^2 \rho_L U_L^2}{8}}_{\text{drag forces}} \quad (8.47)$$

Where $C_{D\infty}$ is the drag coefficient and is a function of Reynolds number, Re , and D is the equivalent radius of the particles. The Reynolds number defined as

$$Re = \frac{U_L D \rho_L}{\mu_L} \quad (8.48)$$

Inserting equating (8.48) into equation (8.47) become

$$\underbrace{C_{D\infty}(U_L)}_{f(Re)} U_L^2 = \frac{4 D g (\rho_S - \rho_L)}{3 \rho_L} \quad (8.49)$$

Equation (8.49) relates the liquid velocity that needed to maintain the particle "floating" to the liquid and particles properties. The drag coefficient, $C_{D\infty}$ is complicated function of the Reynolds number. However, it can be approximated for several regimes. The first regime is for $Re < 1$ where Stokes' Law can be approximated as

$$C_{D\infty} = \frac{24}{Re} \quad (8.50)$$

In transitional region $1 < Re < 1000$

$$C_{D\infty} = \frac{24}{Re} \left(1 + \frac{1}{6} Re^{2/3} \right) \quad (8.51)$$

For larger Reynolds numbers, the Newton's Law region, $C_{D\infty}$, is nearly constant as

$$C_{D\infty} = 0.44 \quad (8.52)$$

In most cases of solid-liquid system, the Reynolds number is in the second range⁹. For the first region, the velocity is small to lift the particle unless the density difference is very small (that very small force can lift the particles). In very large range (especially for gas) the choking might be approached. Thus, in many cases the middle region is applicable.

So far the discussion was about single particle. When there are more than one particle in the cross section, then the actual velocity that every particle experience depends on the void fraction. The simplest assumption that the change of the cross section of the fluid create a parameter that multiply the single particle as

$$C_{D\infty}|_{\alpha} = C_{D\infty} f(\alpha) \quad (8.53)$$

When the subscript α is indicating the void, the function $f(\alpha)$ is not a linear function. In the literature there are many functions for various conditions.

Minimum velocity is the velocity when the particle is “floating”. If the velocity is larger, the particle will drift with the liquid. When the velocity is lower, the particle will sink into the liquid. When the velocity of liquid is higher than the minimum velocity many particles will be floating. It has to remember that not all the particle are uniform in size or shape. Consequently, the minimum velocity is a range of velocity rather than a sharp transition point.

As the solid particles are not pushed by a pump but moved by the forces the fluid applies to them. Thus, the only velocity that can be applied is the fluid velocity. Yet, the solid particles can be supplied at different rate. Thus, the discussion will be focus on the fluid velocity. For small gas/liquid velocity, the particles are what some call fixed fluidized bed. Increasing the fluid velocity beyond a minimum will move the particles and it is referred to as mix fluidized bed. Additional increase of the fluid velocity will move all the particles and this is referred to as fully fluidized bed. For the case of liquid, further increase will

create a slug flow. This slug flow is when slug shape (domes) are almost empty of the solid particle. For the case of gas, additional increase create “tunnels” of empty almost from solid particles. Additional increase in the fluid velocity causes large turbulence and the ordinary domes are replaced by churn type flow or large bubbles that are almost empty of the solid particles. Further increase of the fluid flow increases the empty spots to the whole flow. In that case, the sparse solid particles are dispersed all over. This regimes is referred to as Pneumatic conveying (see Figure 8.9).

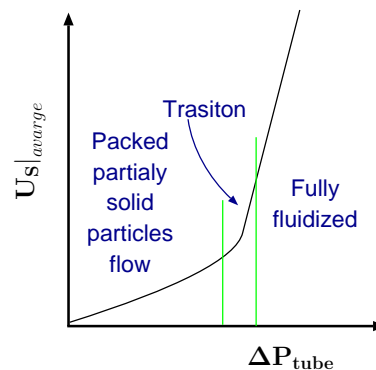


Fig. -8.8. The terminal velocity that left the solid particles.

⁹It be wonderful if flow was in the last range? The critical velocity could be found immediately.

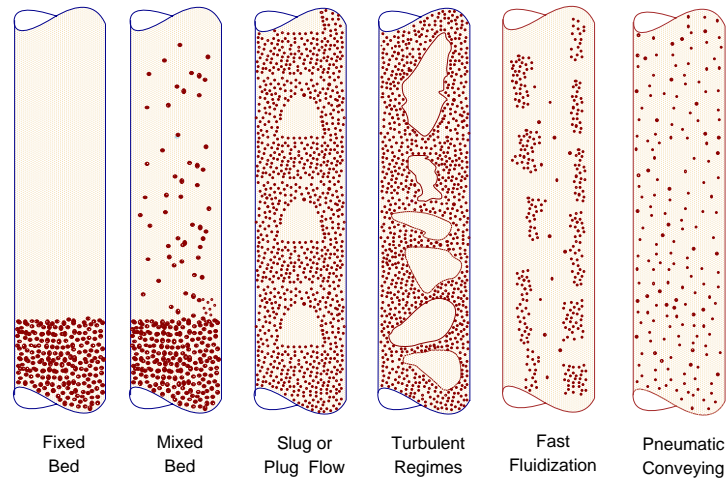


Fig. -8.9. The flow patterns in solid-liquid flow.

One of the main difference between the liquid and gas flow in this category is the speed of sound. In the gas phase, the speed of sound is reduced dramatically with increase of the solid particles concentration (further reading "Fundamentals of Compressible Flow" chapter on Fanno Flow by this author is recommended). Thus, the velocity of gas is limited when reaching the Mach somewhere between $1/\sqrt{k}$ and 1 since the gas will be choked (neglecting the double choking phenomenon). Hence, the length of conduit is very limited. The speed of sound of the liquid does not change much. Hence, this limitation does not (effectively) exist for most cases of solid-liquid flow.

8.8.2 Solid With Lighter Density $\rho_S < \rho$ and With Gravity

This situation is minimal and very few cases exist. However, it must be pointed out that even in solid-gas, the fluid density can be higher than the solid (especially with micro gravity). There was very little investigations and known about the solid-liquid flowing down (with the gravity). Furthermore, there is very little knowledge about the solid-liquid when the solid density is smaller than the liquid density. There is no known flow map for this kind of flow that this author is aware of.

Nevertheless, several conclusions and/or expectations can be drawn. The issue of minimum terminal velocity is not exist and therefor there is no fixed or mixed fluidized bed. The flow is fully fluidized for any liquid flow rate. The flow can have slug flow but more likely will be in fast Fluidization regime. The forces that act on the spherical particle are the buoyancy force and drag force. The buoyancy is accelerating the particle

and drag force are reducing the speed as

$$\frac{\pi D^3 g(\rho_S - \rho_L)}{6} = \frac{C_{D\infty} \pi D^2 \rho_L (U_S - U_L)^2}{8} \tag{8.54}$$

From equation 8.54, it can be observed that increase of the liquid velocity will increase the solid particle velocity at the same amount. Thus, for large velocity of the fluid it can be observed that $U_L/U_S \rightarrow 1$. However, for a small fluid velocity the velocity ratio is very large, $U_L/U_S \rightarrow 0$. The effective body force “seems” by the particles can be in some cases larger than the gravity. The flow regimes will be similar but the transition will be in different points.

The solid-liquid horizontal flow has some similarity to horizontal gas-liquid flow. Initially the solid particles will be carried by the liquid to the top. When the liquid velocity increase and became turbulent, some of the particles enter into the liquid core. Further increase of the liquid velocity appear as somewhat similar to slug flow. However, this author have not seen any evidence that show the annular flow does not appear in solid-liquid flow.

8.9 Counter-Current Flow

This discussion will be only on liquid-liquid systems (which also includes liquid-gas systems). This kind of flow is probably the most common to be realized by the masses. For example, opening a can of milk or juice. Typically if only one hole is opened on the top of the can, the liquid will flow in pulse regime. Most people know that two holes are needed to empty the can easily and continuously. Otherwise, the flow will be in a pulse regime.

In most cases, the possibility to have counter-current flow is limited to having short length of tubes. In only certain configurations of the infinite long pipes the counter-current flow can exist. In that case, the pressure difference and gravity (body forces) dominates the flow. The inertia components of the flow, for long tubes, cannot compensate for the pressure gradient. In short tube, the pressure difference in one phase can be positive while the pressure difference in the other phase can be negative. The pressure difference in the interface must be finite. Hence, the counter-current flow can have opposite pressure gradient for short conduit. But in most cases, the heavy phase (liquid) is pushed by the gravity and lighter phase (gas) is driven by the pressure difference.

The counter-current flow occurs, for example, when cavity is filled or emptied with a liquid. The two phase regimes “occurs” mainly in entrance to the cavity. For example,

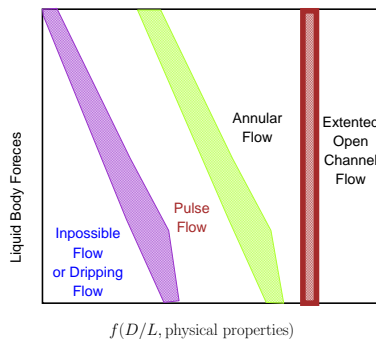


Fig. -8.10. Counter-flow in vertical tubes map.



Fig. -8.11. Counter-current flow in a can (the left figure) has only one hole thus pulse flow and a flow with two holes (right picture).

Figure 8.11 depicts emptying of can filled with liquid. The air is “attempting” to enter the cavity to fill the vacuum created thus forcing pulse flow. If there are two holes, in some cases, liquid flows through one hole and the air through the second hole and the flow will be continuous. It also can be noticed that if there is one hole (orifice) and a long and narrow tube, the liquid will stay in the cavity (neglecting other phenomena such as dripping flow.).

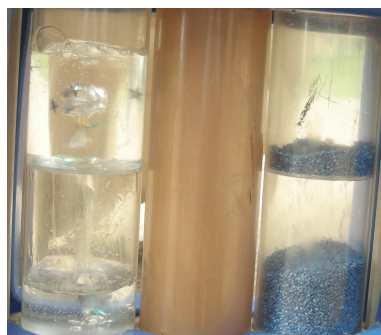


Fig. -8.12. Picture of Counter-current flow in liquid-gas and solid-gas configurations. The container is made of two compartments. The upper compartment is filled with the heavy phase (liquid, water solution, or small wood particles) by rotating the container. Even though the solid-gas ratio is smaller, it can be noticed that the solid-gas is faster than the liquid-gas flow.

There are three flow regimes¹⁰ that have been observed. The first flow pattern is pulse flow regime. In this flow regime, the phases flow turns into different direction (see Figure 8.12). The name pulse flow is used to signify that the flow is flowing in pulses that occurs in a certain frequency. This is opposed to counter-current solid-gas flow when almost no pulse was observed. Initially, due to the gravity, the heavy liquid is leaving the can. Then the pressure in the can is reduced compared to the outside and some lighter liquid (gas) entered into the can. Then, the pressure in the can increase,

¹⁰Caution! this statement should be considered as “so far found”. There must be other flow regimes that were not observed or defined. For example, elongated pulse flow was observed but measured. This field hasn’t been well explored. There are more things to be examined and to be studied.

and some heavy liquid will start to flow. This process continues until almost the liquid is evacuated (some liquid stays due to surface tension). In many situations, the volume flow rate of the two-phase is almost equal. The duration of the cycle depends on several factors. The cycle duration can be replaced by frequency. The analysis of the frequency is much more complex issue and will not be dealt here.

Annular Flow in Counter-current flow

The other flow regime is annular flow in which the heavier phase is on the periphery of the conduit (In the literature, there are some who claim that heavy liquid will be inside). The analysis is provided, but somehow it contradicts with the experimental evidence. Probably, one or more of the assumptions that the analysis is based on is erroneous). In very small diameters of tubes the counter-current flow is not possible because of the surface tension (see section 4.7). The ratio of the diameter to the length with some combinations of the physical properties (surface tension etc) determines the point where the counter flow can start. At this point, the pulsing flow will start and larger diameter will increase the flow and turn the flow into annular flow. Additional increase of the diameter will change the flow regime into extended open channel flow. Extended open channel flow retains the characteristic of open channel flow that the lighter liquid (almost) does not affect the heavier liquid flow. Example of such flow in nature is water falls in which water flows down and air (wind) flows up.

The driving force is the second parameter which affects the flow existence. When the driving (body) force is very small, no counter-current flow is possible. Consider the can in zero gravity field, no counter-current flow possible. However, if the can was on the sun (ignoring the heat transfer issue), the flow regime in the can moves from pulse to annular flow. Further increase of the body force will move the flow to be in the extended "open channel flow."

In the vertical co-current flow there are two possibilities, flow with gravity or against it. As opposed to the co-current flow, the counter-current flow has no possibility for these two cases. The heavy liquid will flow with the body forces (gravity). Thus it should be considered as non-existent flow.

8.9.1 Horizontal Counter-Current Flow

Up to this point, the discussion was focused on the vertical tubes. In horizontal tubes, there is an additional flow regime which is stratified. Horizontal flow is different from vertical flow from the stability issues. A heavier liquid layer can flow above a lighter liquid. This situation is unstable for large diameter but as in static (see section (4.7) page 110) it can be considered stable for small diameters. A flow in a very narrow tube with heavy fluid above the lighter fluid should be considered as a separate issue.

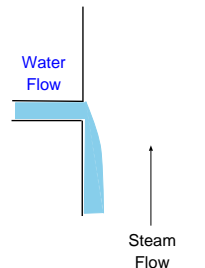


Fig. -8.13. Flood in vertical pipe.

When the flow rate of both fluids is very small, the flow will be stratified counter-current flow. The flow will change to pulse flow when the heavy liquid flow rate increases. Further increase of the flow will result in a single phase flow regime. Thus, closing the window of this kind of flow. Thus, this increase terminates the two phase flow possibility. The flow map of the horizontal flow is different from the vertical flow and is shown in Figure 8.14. A flow in an angle of inclination is closer to vertical flow unless the angle of inclination is very small. The stratified counter flow has a lower pressure loss (for the liquid side). The change to pulse flow increases the pressure loss dramatically.

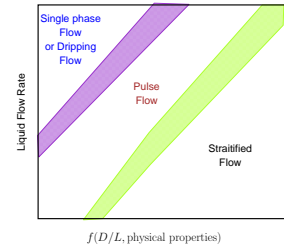


Fig. -8.14. A flow map to explain the horizontal counter-current flow.

8.9.2 Flooding and Reversal Flow

The limits of one kind the counter-current flow regimes, that is stratified flow are discussed here. This problem appears in nuclear engineering (or boiler engineering) where there is a need to make sure that liquid (water) inserted into the pipe reaching the heating zone. When there is no water (in liquid phase), the fire could melt or damage the boiler. In some situations, the fire can be too large or/and the water supply failed below a critical value the water turn into steam. The steam will flow in the opposite direction. To analyze this situation consider a two dimensional conduit with a liquid inserted in the left side as depicted in Figure 8.13. The liquid velocity at very low gas velocity is constant but not uniform. Further increase of the gas velocity will reduce the average liquid velocity. Additional increase of the gas velocity will bring it to a point where the liquid will flow in a reverse direction and/or disappear (dried out).

A simplified model for this situation is for a two dimensional configuration where the liquid is flowing down and the gas is flowing up as shown in Figure 8.15. It is assumed that both fluids are flowing in a laminar regime and steady state. Additionally, it is assumed that the entrance effects can be neglected. The liquid flow rate, Q_L , is unknown. However, the pressure difference in the (x direction) is known and equal to zero. The boundary conditions for the liquid is that velocity at the wall is zero and the velocity at the interface is the same for both phases $U_G = U_L$ or $\tau_i|_G = \tau_i|_L$. As it will be shown later, both conditions cannot coexist. The model can be improved by considering turbulence, mass transfer, wavy interface, etc¹¹.

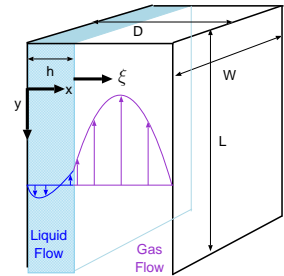


Fig. -8.15. A diagram to explain the flood in a two dimension geometry.

¹¹The circular configuration is under construction and will be appeared as a separated article momentarily.

This model is presented to exhibit the trends and the special features of counter-current flow. Assuming the pressure difference in the flow direction for the gas is constant and uniform. It is assumed that the last assumption does not contribute or change significantly the results. The underline rationale for this assumption is that gas density does not change significantly for short pipes (for more information look for the book "Fundamentals of Compressible Flow" in Pottot book series in the Fanno flow chapter.).

The liquid film thickness is unknown and can be expressed as a function of the above boundary conditions. Thus, the liquid flow rate is a function of the boundary conditions. On the liquid side, the gravitational force has to be balanced by the shear forces as

$$\frac{d\tau_{xy}}{dx} = \rho_L g \quad (8.55)$$

The integration of equation (8.55) results in

$$\tau_{xy} = \rho_L g x + C_1 \quad (8.56)$$

The integration constant, C_1 , can be found from the boundary condition where $\tau_{xy}(x = h) = \tau_i$. Hence,

$$\tau_i = \rho_L g h + C_1 \quad (8.57)$$

The integration constant is then $C_1 = \tau_i - \rho_L g h$ which leads to

$$\tau_{xy} = \rho_L g (x - h) + \tau_i \quad (8.58)$$

Substituting the newtonian fluid relationship into equation (8.58) to obtain

$$\mu_L \frac{dU_y}{dx} = \rho_L g (x - h) + \tau_i \quad (8.59)$$

or in a simplified form as

$$\frac{dU_y}{dx} = \frac{\rho_L g (x - h)}{\mu_L} + \frac{\tau_i}{\mu_L} \quad (8.60)$$

Equation (8.60) can be integrated to yield

$$U_y = \frac{\rho_L g}{\mu_L} \left(\frac{x^2}{2} - h x \right) + \frac{\tau_i x}{\mu_L} + C_2 \quad (8.61)$$

The liquid velocity at the wall, $[U(x = 0) = 0]$, is zero and the integration coefficient can be found to be

$$C_2 = 0 \quad (8.62)$$

The liquid velocity profile is then

$$U_y = \frac{\rho_L g}{\mu_L} \left(\frac{x^2}{2} - h x \right) + \frac{\tau_i x}{\mu_L} \quad (8.63)$$

The velocity at the liquid–gas interface is

$$U_y(x = h) = \frac{\tau_i h}{\mu_L} - \frac{\rho_L g h^2}{2 \mu_L} \quad (8.64)$$

The velocity can vanish (zero) inside the film in another point which can be obtained from

$$0 = \frac{\rho_L g}{\mu_L} \left(\frac{x^2}{2} - h x \right) + \frac{\tau_i x}{\mu_L} \quad (8.65)$$

The solution for equation (8.65) is

$$x|_{@U_L=0} = 2h - \frac{2\tau_i}{\mu_L g \rho_L} \quad (8.66)$$

The maximum x value is limited by the liquid film thickness, h . The minimum shear stress that start to create reversible velocity is obtained when $x = h$ which is

$$0 = \frac{\rho_L g}{\mu_L} \left(\frac{h^2}{2} - h h \right) + \frac{\tau_i h}{\mu_L} \quad (8.67)$$

$$\hookrightarrow \tau_{i0} = \frac{h g \rho_L}{2}$$

If the shear stress is below this critical shear stress τ_{i0} then no part of the liquid will have a reversed velocity. The notation of τ_{i0} denotes the special value at which a starting shear stress value is obtained to have reversed flow. The point where the liquid flow rate is zero is important and it is referred to as initial flashing point.

The flow rate can be calculated by integrating the velocity across the entire liquid thickness of the film.

$$\frac{Q}{w} = \int_0^h U_y dx = \int_0^h \left[\frac{\rho_L g}{\mu_L} \left(\frac{x^2}{2} - h x \right) + \frac{\tau_i x}{\mu_L} \right] dx \quad (8.68)$$

Where w is the thickness of the conduit (see Figure 8.15). Integration equation (8.68) results in

$$\frac{Q}{w} = \frac{h^2 (3\tau_i - 2g h \rho_L)}{6 \mu_L} \quad (8.69)$$

It is interesting to find the point where the liquid mass flow rate is zero. This point can be obtained when equation (8.69) is equated to zero. There are three solutions for equation (8.69). The first two solutions are identical in which the film height is $h = 0$ and the liquid flow rate is zero. But, also, the flow rate is zero when $3\tau_i = 2g h \rho_L$. This request is identical to the demand in which

$$\tau_{i \text{ critical}} = \frac{2g h \rho_L}{3} \quad (8.70)$$

This critical shear stress, for a given film thickness, reduces the flow rate to zero or effectively “drying” the liquid (which is different than equation (8.67)).

For this shear stress, the critical upward interface velocity is

$$U_{critical}|_{interface} = \frac{1}{6} \left(\frac{\rho_L g h^2}{\mu_L} \right) \quad (8.71)$$

The wall shear stress is the last thing that will be done on the liquid side. The wall shear stress is

$$\tau_L|_{@wall} = \mu_L \left. \frac{dU}{dx} \right|_{x=0} = \mu_L \left(\frac{\rho_L g}{\mu_L} \left(2x - h \right) + \frac{\overbrace{2gh\rho_L}^{\tau_i}}{3} \frac{1}{\mu_L} \right) \Big|_{x=0} \quad (8.72)$$

Simplifying equation (8.72)¹² becomes (notice the change of the sign accounting for the direction)

$$\tau_L|_{@wall} = \frac{gh\rho_L}{3} \quad (8.73)$$

Again, the gas is assumed to be in a laminar flow as well. The shear stress on gas side is balanced by the pressure gradient in the y direction. The momentum balance on element in the gas side is

$$\frac{d\tau_{xyG}}{dx} = \frac{dP}{dy} \quad (8.74)$$

The pressure gradient is a function of the gas compressibility. For simplicity, it is assumed that pressure gradient is linear. This assumption means or implies that the gas is incompressible flow. If the gas was compressible with an ideal gas equation of state then the pressure gradient is logarithmic. Here, for simplicity reasons, the linear equation is used. In reality the logarithmic equation should be used (a discussion can be found in “Fundamentals of Compressible Flow” a Potto project book). Thus, equation (8.74) can be rewritten as

$$\frac{d\tau_{xyG}}{dx} = \frac{\Delta P}{\Delta y} = \frac{\Delta P}{L} \quad (8.75)$$

Where $\Delta y = L$ is the entire length of the flow and ΔP is the pressure difference of the entire length. Utilizing the Newtonian relationship, the differential equation is

$$\frac{d^2U_G}{dx^2} = \frac{\Delta P}{\mu_G L} \quad (8.76)$$

¹²Also noticing that equation (8.70) has to be equal $gh\rho_L$ to support the weight of the liquid.

Equation (8.76) can be integrated twice to yield

$$U_G = \frac{\Delta P}{\mu_G L} x^2 + C_1 x + C_2 \quad (8.77)$$

This velocity profile must satisfy zero velocity at the right wall. The velocity at the interface is the same as the liquid phase velocity or the shear stress are equal. Mathematically these boundary conditions are

$$U_G(x = D) = 0 \quad (8.78)$$

and

$$\begin{aligned} U_G(x = h) &= U_L(x = h) & (a) \quad \text{or} & \quad (8.79) \\ \tau_G(x = h) &= \tau_L(x = h) & (b) & \end{aligned}$$

Applying B.C. (8.78) into equation (8.77) results in

$$\begin{aligned} U_G = 0 &= \frac{\Delta P}{\mu_G L} D^2 + C_1 D + C_2 & (8.80) \\ \hookrightarrow C_2 &= -\frac{\Delta P}{\mu_G L} D^2 + C_1 D \end{aligned}$$

Which leads to

$$U_G = \frac{\Delta P}{\mu_G L} (x^2 - D^2) + C_1 (x - D) \quad (8.81)$$

At the other boundary condition, equation (8.79)(a), becomes

$$\frac{\rho_L g h^2}{6 \mu_L} = \frac{\Delta P}{\mu_G L} (h^2 - D^2) + C_1 (h - D) \quad (8.82)$$

The last integration constant, C_1 can be evaluated as

$$C_1 = \frac{\rho_L g h^2}{6 \mu_L (h - D)} - \frac{\Delta P (h + D)}{\mu_G L} \quad (8.83)$$

With the integration constants evaluated, the gas velocity profile is

$$U_G = \frac{\Delta P}{\mu_G L} (x^2 - D^2) + \frac{\rho_L g h^2 (x - D)}{6 \mu_L (h - D)} - \frac{\Delta P (h + D) (x - D)}{\mu_G L} \quad (8.84)$$

The velocity in Equation (8.84) is equal to the velocity equation (8.64) when ($x = h$). However, in that case, it is easy to show that the gas shear stress is not equal to the liquid shear stress at the interface (when the velocities are assumed to be the equal). The difference in shear stresses at the interface due to this assumption, of the equal velocities, cause this assumption to be not physical.

The second choice is to use the equal shear stresses at the interface, condition (8.79)(b). This condition requires that

$$\mu_G \frac{dU_G}{dx} = \mu_L \frac{dU_L}{dx} \quad (8.85)$$

The expressions for the derivatives are

$$\overbrace{\frac{2h\Delta P}{L} + \mu_G C_1}^{\text{gas side}} = \overbrace{\frac{2gh\rho_L}{3}}^{\text{liquid side}} \quad (8.86)$$

As result, the integration constant is

$$C_1 = \frac{2gh\rho_L}{3\mu_G} - \frac{2h\Delta P}{\mu_G L} \quad (8.87)$$

The gas velocity profile is then

$$U_G = \frac{\Delta P}{\mu_G L} (x^2 - D^2) + \left(\frac{2gh\rho_L}{3\mu_G} - \frac{2h\Delta P}{\mu_G L} \right) (x - D) \quad (8.88)$$

The gas velocity at the interface is then

$$U_G|_{@x=h} = \frac{\Delta P}{\mu_G L} (h^2 - D^2) + \left(\frac{2gh\rho_L}{3\mu_G} - \frac{2h\Delta P}{\mu_G L} \right) (h - D) \quad (8.89)$$

This gas interface velocity is different than the velocity of the liquid side. The velocity at interface can have a "slip" in very low density and for short distances. The shear stress at the interface must be equal, if no special effects occurs. Since there no possibility to have both the shear stress and velocity on both sides of the interface, different thing(s) must happen. It was assumed that the interface is straight but is impossible. Then if the interface becomes wavy, the two conditions can co-exist.

The wall shear stress is

$$\tau_G|_{@wall} = \mu_G \left. \frac{dU_G}{dx} \right|_{x=D} = \mu_G \left(\frac{\Delta P 2x}{\mu_G L} + \left(\frac{2gh\rho_L}{3\mu_G} - \frac{2h\Delta P}{\mu_G L} \right) \right)_{x=D} \quad (8.90)$$

or in a simplified form as

$$\tau_G|_{@wall} = \frac{2\Delta P (D - h)}{L} + \frac{2gh\rho_L}{3} \quad (8.91)$$

The Required Pressure Difference

The pressure difference to create the flooding (drying) has to take into account the fact that the surface is wavy. However, as first estimate the waviness of the surface can be neglected. The estimation of the pressure difference under the assumption of equal shear stress can be applied. In the same fashion the pressure difference under the assumption the equal velocity can be calculated.

The actual pressure difference can be between these two assumptions but not must be between them. This model and its assumptions are too simplistic and the actual pressure difference is larger. However, this explanation is to show magnitudes and trends and hence it provided here.

To calculate the required pressure that cause the liquid to dry, the total balance is needed. The control volume include the gas and liquid volumes. Figure 8.16 describes the general forces that acts on the control volume. There are two forces that act against the gravity and two forces with the gravity. The gravity force on the gas can be neglected in most cases. The gravity force on the liquid is the liquid volume times the liquid volume as

$$F_{gL} = \rho g \overbrace{hL}^{\text{Volme/w}} \quad (8.92)$$

The total momentum balance is (see Figure 8.16)

$$F_{gL} + \overbrace{L}^{A/w} \tau_{w_G} = \overbrace{L}^{A/w} \tau_{w_L} + \overbrace{D \Delta P}^{\text{force due to pressure}} \quad (8.93)$$

Substituting the different terms into (8.93) result in

$$\rho g L h + L \left(\frac{2 \Delta P (D - h)}{L} + \frac{2 g h \rho_L}{3} \right) = L \frac{g h \rho_L}{3} + D \Delta P \quad (8.94)$$

Simplifying equation (8.94) results in

$$\frac{4 \rho g L h}{3} = (2 h - D) \Delta P \quad (8.95)$$

or

$$\Delta P = \frac{4 \rho g L h}{3 (2 h - D)} \quad (8.96)$$

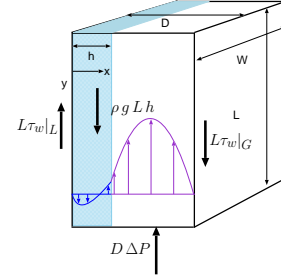


Fig. -8.16. General forces diagram to calculated the in a two dimension geometry.

This analysis shows far more reaching conclusion than initial anticipation expected. The interface between the two liquid flowing together is wavy. Unless the derivations or assumptions are wrong, this analysis equation (8.96) indicates that when $D > 2h$ is a special case (extend open channel flow).

8.10 Multi-Phase Conclusion

For the first time multi-phase is included in a standard introductory textbook on fluid mechanics. There are several points that should be noticed in this chapter. There are many flow regimes in multi-phase flow that "regular" fluid cannot be used to solve it such as flooding. In that case, the appropriate model for the flow regime should be employed. The homogeneous models or combined models like Lockhart–Martinelli can be employed in some cases. In other case where more accurate measurement are needed a specific model is required. Perhaps as a side conclusion but important, the assumption of straight line is not appropriate when two liquid with different viscosity are flowing.

APPENDIX A

The Mathematics Backgrounds for Fluid Mechanics

A.1 Ordinary Differential Equations

In this section a brief summary of ordinary differential equations is presented. It is not intent to be a replacement to a standard textbook but as a quick reference. It is suggested that the reader for more in depth information will read “Differential Equations and Boundary Value Problems” by Boyce de-Prima or any other book in this area. Ordinary differential equations are defined by their order which is defined by the order of the highest derivative. If the highest derivative is first order the equation is referred as first order differential equation etc. Ordinary separated to linear and non-linear equations. The linearly is determined by the multiplication of function or its derivatives.

A.1.1 First Order Differential Equations

A.1.2 Variable Separation

A first order derivative equation has several forms and there is no one solution fit all but families of solutions. The most general form is

$$f\left(y, \frac{dy}{dx}, x\right) = 0 \quad (\text{A.1})$$

Or it can be simplified to the first form as

$$\frac{dy}{dx} = F(x, y) \quad (\text{A.2})$$

In some case $F(x, y)$ can be written as $X(x)Y(y)$. In that case it is said that F is spreadable. Then equation (A.2) can be written as

$$\frac{dy}{Y(y)} = X(x)dx \quad (\text{A.3})$$

Equation can be integrated either analytically or numerically and the solution is

$$\int \frac{dy}{Y(y)} = \int X(x)dx \quad (\text{A.4})$$

The limits of the integral is (are) the initial condition. The initial condition is the value the function has at some point. The initial condition referred to the fact that the value are given commonly at time equal to zero

A.1.3 The Integral Factor

Another methods is referred to as integration factor which deals with a limited but very important class of equations. This family is referred as linear equation. The general form of the equation is

$$\frac{dy}{dx} + g(x)y = m(x) \quad (\text{A.5})$$

Multiplying equation (A.5) by unknown function $N(x)$ transformed it to

$$N(x) \frac{dy}{dx} + N(x)g(x)y = N(x)m(x) \quad (\text{A.6})$$

What is needed from $N(x)$ is to provide a full differential such as

$$N(x) \frac{dy}{dx} + N(x)g(x)y = \frac{d[N(x)g(x)y]}{dx} \quad (\text{A.7})$$

This condition requires that

$$\frac{dN(x)}{dx} = N(x)g(x) \implies \frac{dN(x)}{N(x)} = g(x)dx \quad (\text{A.8})$$

Equation (A.8) is integrated to be

$$\ln(N(x)) = \int g(x)dx \implies N(x) = e^{\int g(x)dx} \quad (\text{A.9})$$

Using the differentiation chain role provides

$$\frac{dN(x)}{dx} = e^{\int g(x)dx} \overbrace{\frac{d}{du}}^{\frac{du}{dx}} \underbrace{g(x)}_{g(x)} \quad (\text{A.10})$$

which indeed satisfy equation (A.7). Thus equation (A.6) becomes

$$\frac{d[N(x)g(x)y]}{dx} = N(x)m(x) \quad (\text{A.11})$$

Multiplying equation (A.11) by dx and integrating results in

$$N(x)g(x)y = \int N(x)m(x)dx \quad (\text{A.12})$$

The solution is then

$$y = \frac{\int N(x)m(x)dx}{g(x)e^{\int g(x)dx}} \quad (\text{A.13})$$

Example A.1:

Find the solution for a typical problem of

$$\frac{dy}{dx} + y = 1$$

SOLUTION

Substituting into equation (A.13) provides

$$y = e^{-x} (e^x + c) = 1 + ce^{-x}$$

End Solution

A.1.4 Non-Linear Equation

Non-Linear equations are equations that power of function or function derivative is not equal to one. Many non linear equation can be transformed into linear equation and use a linear equation technique to solve them. One such equation is called in the literature as the Bernoulli Equations¹. This equation is

$$\frac{du}{dt} + m(t)u = n(t) \overbrace{u^p}^{\text{non-linear part}} \quad (\text{A.14})$$

The solution method is to transfer it to linear by the $v = u^{1-p}$ which transform equation (A.14) into

$$\frac{dv}{dt} + (1-p)m(t)v = (1-n)n(t) \quad (\text{A.15})$$

¹Not to be confused with Bernoulli equation without the s that referred to the energy equation.

This equation can be linearized and can be solved using the linear methods. Then after the transferred equation is solved using the reversed transformation the actual solution is

$$u = v^{(n-1)} \quad (\text{A.16})$$

Example A.2:

Solve the following Bernoulli equation

$$\frac{du}{dt} + t^2 u = \sin(t) u^2$$

A.1.4.1 Homogeneous Equations

Homogeneous function is given as

$$\frac{du}{dt} = f(u, t) = f(au, at) \quad (\text{A.17})$$

for a that can be any real positive number. For this case, the transformation of

$$v = \frac{u}{t} \quad (\text{A.18})$$

which transformed equation (A.17) into

$$t \frac{dv}{dt} + v = f(1, v) \quad (\text{A.19})$$

and the solution

$$\ln |t| = \int \frac{dv}{f(1, v) - v} + c \quad (\text{A.20})$$

A.1.4.2 Variables Separable Equations

In fluid mechanics and many other field there are differential that referred to variables separable equation. In fact, this kind of class of equation appear all over this book. For this sort equation it can be written that

$$\frac{du}{dt} = f(t)g(u) \quad (\text{A.21})$$

The solution of this kind of equation is

$$\int \frac{du}{g(u)} = \int f(t) dt \quad (\text{A.22})$$

A.1.5 Second Order Differential equations

The simplest equations is constant coefficients such

$$a \frac{d^2u}{dt^2} + b \frac{du}{dt} + cu = 0 \quad (\text{A.23})$$

Substituting e^{st} results in the quadratic equation

$$as^2 + bs + c = 0 \quad (\text{A.24})$$

If $b^2 > 4ac$ then there are two unique solutions for the quadratic equation and the general solution is

$$u = c_1 e^{s_1 t} + c_2 e^{s_2 t} \quad (\text{A.25})$$

For the case of $b^2 = 4ac$ the general solution is

$$u = c_1 e^{s_1 t} + c_2 t e^{s_1 t} \quad (\text{A.26})$$

In the case of $b^2 < 4ac$ the solution of the quadratic equation is a complex number which means that the solution has exponential and trigonometric functions as

$$u = c_1 e^{\alpha t} \cos(\beta t) + c_2 e^{\alpha t} \sin(\beta t) \quad (\text{A.27})$$

Where the real part is

$$\alpha = \frac{-b}{2a} \quad (\text{A.28})$$

and the imaginary number is

$$\beta = \frac{\sqrt{4ac - b^2}}{2a} \quad (\text{A.29})$$

A.1.6 Non-Homogeneous Equation

Equation that not equal to zero in this form

$$a \frac{d^2u}{dt^2} + b \frac{du}{dt} + cu = l(x) \quad (\text{A.30})$$

The solution of the homogeneous equation is zero so additional solution of $l(x)$ though the operation the right hand side is the total solution as

$$u_{total} = u_h + u_p \quad (\text{A.31})$$

Where the solution u_h is the solution of the homogeneous solution and u_p is the solution of the particular function $l(x)$. If the function on the right hand side is polynomial than the solution is will

$$u_{total} = u_h + \sum_{i=1}^n u_{p_i} \quad (\text{A.32})$$

Example A.3:

Solve the non-homogeneous equation

$$\frac{d^2u}{dt^2} - 5 \frac{du}{dt} + 6u = t + t^2$$

SOLUTION

The homogeneous solution

$$u(t) = c_1 e^{2t} + c_1 e^{3t} \quad (1.111.a)$$

the particular solution or the t is

$$u(t) = \frac{6t + 5}{36} \quad (1.111.b)$$

and the solution of the t^2 is

$$u(t) = \frac{18t^2 + 30t + 19}{108} \quad (1.111.c)$$

The total solution

$$u(t) = c_1 e^{2t} + c_1 e^{3t} + \frac{9t^2 + 24t + 17}{54} \quad (1.111.d)$$

End Solution

A.1.7 Non-Linear Second Order Equation

Some of the techniques that were discussed in the previous can be used for the second order differential equation such variable separation. If the following equation

$$f(u, \dot{u}, \ddot{u}) = 0 \quad (A.33)$$

can be written in the form

$$f_1(u)\dot{u} = f_2(\dot{u})\ddot{u} \quad (A.34)$$

then the equation (A.34) it is referred that equation is separable. The derivative of \dot{u} can be treated as new function v and $\dot{v} = \ddot{u}$. Hence, equation (A.34) can be integrated

$$\int_{u_0}^u f_1(u)\dot{u} = \int_{\dot{u}_0}^{\dot{u}} f_2(\dot{u})\ddot{u} \quad (A.35)$$

The integration results in a first order differential equation which should be dealt with the previous methods. It can be noticed that initial condition of the function is used

twice. The physical reason is that the equation represents strong effect of the function at a certain point. This equation family is not well discussed in mathematical textbooks².

Example A.4:

Solve the equation

$$\sqrt{u} \frac{du}{dt} - \sin\left(\frac{du}{dt}\right) \frac{d^2u}{dt^2} = 0$$

What happen to dt ? With the initial condition of $u(0) = 0$ and $\dot{u}(0) = 0$

Special cases or families of second order differential equation

$$u - u \left(\frac{du}{dt}\right) \left(\frac{d^2u}{dt^2}\right) = 0 \quad (\text{A.36})$$

which one solution of $u = k_1$ and the “homogeneous” solution

$$1 = \left(\frac{du}{dt}\right) \left(\frac{d^2u}{dt^2}\right) \quad (\text{A.37})$$

which can be transformed into $v = \frac{dv}{dt}$

$$1 = v \frac{dv}{dt} \implies dt = v dv \quad (\text{A.38})$$

which can be solved with the previous methods.

This equation typically result in the derivations of the energy integral equation. It is non-linear because the second derivative is square and the function multiply the second derivative.

$$u \left(\frac{d^2u}{dt^2}\right) + \left(\frac{du}{dt}\right)^2 = 0 \quad (\text{A.39})$$

It can be noticed that

$$\frac{d}{dt} \left(u \frac{du}{dt}\right) = 0 \quad (\text{A.40})$$

This concept leads to the solution which is

$$\frac{u^2}{2k_1} = t + k_2 \quad (\text{A.41})$$

For non-homogeneous equation they can be integrated as well.

²This author work in this area during his master but to his shame he did not produce any paper. The paper are still his drawer and waiting some spare time.

Example A.5:

Show that the solution of

$$u \left(\frac{d^2 u}{dt^2} \right) + \left(\frac{du}{dt} \right)^2 + u = 0 \quad (1.V.a)$$

is

$$-\frac{\sqrt{3} \int \frac{u}{\sqrt{3k_1 - u^3}} du}{\sqrt{2}} = t + k_2 \quad (1.V.b)$$

$$\frac{\sqrt{3} \int \frac{u}{\sqrt{3k_1 - u^3}} du}{\sqrt{2}} = t + k_2 \quad (1.V.c)$$

A.2 Vectors

Vector is quantity with direction. The length of the vector is

$$\|\mathbf{U}\| = \sqrt{U_x^2 + U_y^2 + U_z^2} \quad (A.42)$$

Vector can be normalized

$$\hat{\mathbf{U}} = \frac{\mathbf{U}}{\|\mathbf{U}\|} = \frac{U_1}{\|\mathbf{U}\|} \mathbf{h}_1 + \frac{U_2}{\|\mathbf{U}\|} \mathbf{h}_2 + \frac{U_3}{\|\mathbf{U}\|} \mathbf{h}_3 \quad (A.43)$$

A.2.1 Vector Algebra

Vectors obey several standard mathematical operations and are applied to scalar. The following are vectors, \mathbf{U} , \mathbf{V} , and \mathbf{W} . And in this discussion a and b are scalars. Then the following can be said

1. $(\mathbf{U} + \mathbf{V}) + \mathbf{W} = (\mathbf{U} + \mathbf{V} + \mathbf{W})$
2. $\mathbf{U} + \mathbf{V} = \mathbf{V} + \mathbf{U}$
3. Zero vector is such that $\mathbf{U} + \mathbf{0} = \mathbf{U}$
4. Additive inverse $\mathbf{U} - \mathbf{U} = \mathbf{0}$
5. $a(\mathbf{U} + \mathbf{V}) = a\mathbf{U} + a\mathbf{V}$
6. $a(b\mathbf{U}) = ab\mathbf{U}$

The multiplication and the division have different meaning that is in scalar operation. There two kinds of multiplications for vectors. The first multiplication is the dot product which defined in equation (A.44).

Notice that this multiplication has no negative value as in scalar multiplication.

$$\mathbf{U} \cdot \mathbf{V} = \overbrace{|\mathbf{U}| \cdot |\mathbf{V}|}^{\text{regular scalar multiplication}} \cos(\overbrace{\angle(\mathbf{U}, \mathbf{V})}^{\text{angle}}) \quad (\text{A.44})$$

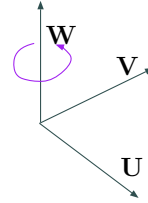


Fig. -A.1. The right hand rule, multiplication of \mathbf{U} by \mathbf{V} results in \mathbf{W} .

The second multiplication is the cross product. The result of the cross product is also vector as opposed to scalar in dot product. The cross product is defined in an orthogonal coordinate (\hat{h}_1 , \hat{h}_2 , and \hat{h}_3) as

$$\mathbf{U} \times \mathbf{V} = |\mathbf{U}| \cdot |\mathbf{V}| \sin(\overbrace{\angle(\mathbf{U}, \mathbf{V})}^{\text{angle}}) \hat{\mathbf{n}} \quad (\text{A.45})$$

where θ is the angle between \mathbf{U} and \mathbf{V} , and $\hat{\mathbf{n}}$ is a unit vector perpendicular to both \mathbf{U} and \mathbf{V} which obeys the right hand rule. The right hand rule is referred to the direction of resulting vector. Note that \mathbf{U} and \mathbf{V} are not necessarily orthogonal. Additionally note that order of multiplication has significance. This multiplication has no negative value but different direction.

One of the consequence of this definitions is

$$\hat{h}_1^2 = \hat{h}_2^2 = \hat{h}_3^2 = 0 \quad (\text{A.46})$$

In right hand orthogonal coordinate system

$$\hat{h}_1 \times \hat{h}_2 = \hat{h}_3 \quad (\text{A.47})$$

$$\hat{h}_2 \times \hat{h}_3 = \hat{h}_1 \quad (\text{A.48})$$

$$\hat{h}_3 \times \hat{h}_1 = \hat{h}_2 \quad (\text{A.49})$$

The cross product can be written as

$$\mathbf{U} \times \mathbf{V} = (U_2 V_3 - U_3 V_2) \hat{h}_1 + (U_3 V_1 - U_1 V_3) \hat{h}_2 + (U_1 V_2 - U_2 V_1) \hat{h}_3 \quad (\text{A.50})$$

Equation (A.50) in matrix form as

$$\mathbf{U} \times \mathbf{V} = \begin{pmatrix} \hat{h}_1 & \hat{h}_2 & \hat{h}_3 \\ U_2 & U_2 & U_3 \\ V_2 & V_2 & V_3 \end{pmatrix} \quad (\text{A.51})$$

The most complex all this algebraic operation is the division. In the multiplication in vector world have two kinds one that results in scalar and one result in a vector.

Just from looking that possible combinations for multiplication show that there are at least possibilities due the the angle and scalar and vector. The reason that these combinations, for example scalar and $\cos \theta$ and not scalar and scalar with $\sin \theta$, is that this combination has physical meaning. The same can be said for the combination of the vector with $\sin \theta$. It is that previous experience that help to define the definition of the division.

The possible combinations of the division is very large. For example, the results can be scalar with the angle (with \cos or \sin), vector with the angle, etc. These four above combinations are not the only possibilities (not including the left hand system). It turn out these combinations have very little³ physical meaning. Additional possibility is that every combination of one vector element is divided by the other vector element. Since every vector element has three possible elements the total combination is 3×3 . There at least two possibilities how to treat these elements. It turned out that combination of three vectors has a physical meaning. The three vectors have a need for additional notation of vector of vector which is referred as a tensor. The following combination is commonly suggested

$$\frac{\mathbf{U}}{\mathbf{V}} = \begin{pmatrix} \frac{U_1}{V_1} & \frac{U_2}{V_1} & \frac{U_3}{V_1} \\ \frac{U_1}{V_2} & \frac{U_2}{V_2} & \frac{U_3}{V_2} \\ \frac{U_1}{V_3} & \frac{U_2}{V_3} & \frac{U_3}{V_3} \end{pmatrix} \quad (\text{A.52})$$

One such example of this division is the pressure which the explanation is commonality avoided or eliminated from the fluid mechanics books including the direct approach in this book.

This tensor or the matrix can undergo regular linear algebra operation such as eigenvalue value and eigen "vectors." Also note the multiplying matrices and inverse matrix are also available operation to this results.

A.3 Trigonometry

These identities were set up by Keone Hon with slight modification

1. $\sin(\alpha + \beta) = \sin \alpha \cos \beta + \sin \beta \cos \alpha$
2. $\sin(\alpha - \beta) = \sin \alpha \cos \beta - \sin \beta \cos \alpha$
3. $\cos(\alpha + \beta) = \cos \alpha \cos \beta - \sin \alpha \sin \beta$

³This author did find any physical meaning these combinations but there could be and those the word "little" is used.

4. $\cos(\alpha - \beta) = \cos \alpha \cos \beta + \sin \alpha \sin \beta$

5. $\tan(\alpha + \beta) = \frac{\tan \alpha + \tan \beta}{1 - \tan \alpha \tan \beta}$

6. $\tan(\alpha - \beta) = \frac{\tan \alpha - \tan \beta}{1 + \tan \alpha \tan \beta}$

1. $\sin 2\alpha = 2 \sin \alpha \cos \alpha$

2. $\cos 2\alpha = \cos^2 \alpha - \sin^2 \alpha = 2 \cos^2 \alpha - 1 = 1 - 2 \sin^2 \alpha$

3. $\tan 2\alpha = \frac{2 \tan \alpha}{1 - \tan^2 \alpha}$

4. $\sin \frac{\alpha}{2} = \pm \sqrt{\frac{1 - \cos \alpha}{2}}$ (determine whether it is + or - by finding the quadrant that $\frac{\alpha}{2}$ lies in)

5. $\cos \frac{\alpha}{2} = \pm \sqrt{\frac{1 + \cos \alpha}{2}}$ (same as above)

6. $\tan \frac{\alpha}{2} = \frac{1 - \cos \alpha}{\sin \alpha} = \frac{\sin \alpha}{1 + \cos \alpha}$

for formulas 3-6, consider the triangle with sides of length a , b , and c , and opposite angles A , B , and C , respectively

1. $\sin^2 \alpha = \frac{1 - 2 \cos(2\alpha)}{2}$

2. $\cos^2 \alpha = \frac{1 + 2 \cos(2\alpha)}{2}$

3. $\frac{\sin A}{a} = \frac{\sin B}{b} = \frac{\sin C}{c}$ (Law of Sines)

4. $c^2 = a^2 + b^2 - 2ab \cos C$ (Law of Cosines)

5. Area of triangle = $\frac{1}{2}ab \sin C$

6. Area of triangle = $\sqrt{s(s-a)(s-b)(s-c)}$, where $s = \frac{a+b+c}{2}$ (Heron's Formula)

Subjects Index

A

absolute viscosity, 7, 8, 11, 12
Add mass, 158
Add momentum, 158
Arc shape, 88
Archimedes, 3
Atmospheric pressure, 58
Average Momentum, 147
Average Velocity Integral Analysis, 158, 160
Averaged kinetic energy, 173
Averaged momentum energy, 173

B

Bingham's model, 11
Body force, 55, 56, 58, 67, 69
 effective, 57
Boundary Layer, 133
bulk modulus, 20, 21
buoyancy, 3, 92–94
buoyant force, 67, 100

C

Co-current flow, 193
Compressibility factor, 62, 71
Conduction, 168
Conservative force, 183
Convection, 168
Correction factor, 66
Correction factor for averaged velocity, 173
Counter-current
 Pulse flow, 210
Counter-current flow, 193, 209
 Annular flow, 211
 Extended Open channel flow, 211
Cut-out shapes, 87

D

D'Alembert paradox, 3
Deformable control volume, 120
Density ratio, 67, 103

dilettante, 11
Double choking phenomenon, 195

E

Energy conservation, 167
Energy Equation
 Accelerated System, 183
 Linear accelerate System, 183
 Rotating Coordinate System, 184
Energy equation
 Frictionless Flow, 181
 Steady State, 181
External forces, 144

F

First Law of Thermodynamics, 167
Fixed fluidized bed, 207
Flow out tank, 170
Flow regime map, 190
Flow regimes in one pipe, 194
Free expansion, 67–69
Fully fluidized bed, 207

G

Gas-gas flow, 192

H

horizontal counter-current flow, 211
Horizontal flow, 193
Hydrostatic pressure, 55, 89

I

Ideal gas, 62
Integral analysis
 big picture, 138
 small picture, 138

K

kinematic viscosity, 11

L

Lapse rate, 69

Leibniz integral rule, 130
 Limitation of the integral approach, 179
 Linear acceleration, 72
 Liquid phase, 64
 Liquid–Liquid Regimes, 192
 Lockhart martinelli model, 204

M

“Magnification factor”, 61
 Mass velocity, 198
 Metacentric point, 102
 Minimum velocity solid–liquid flow, 206
 Mixed fluidized bed, 207
 Momentum Conservation, 143
 Momentum equation
 Accelerated system, 145
 Multi–phase flow, 189
 Multiphase flow against the gravity, 196

N

Neutral moment
 Zero moment, 101
 Neutral stable, 69, 101, 111, 112
 Newtonian fluids, 1, 8
 Non–deformable control volume, 120
 Non–Linear Equation, 223

O

Open channel flow, 193
 Oscillating manometer, 180

P

Pendulum action, 109
 Piezometric pressure, 58
 Pneumatic conveying, 207
 Polynomial function, 90
 Pressure center, 82
 pseudoplastic, 11
 Pulse flow, 210
 purely viscous fluids, 11

R

Radiation, 168
 Rayleigh–Taylor instability, 110, 193
 Real gas, 62

Return path for flow regimes, 195
 Reynolds Transport Theorem, 130
 Rocket mechanics, 153

S

Scalar function, 56, 86
 Second Law of Thermodynamics, 181
 shear stress, 6
 Solid–fluid flow
 Gas dynamics aspects, 208
 Solid–fluid flow, 205
 Solid–liquid flow, 205
 Solid–solid flow, 192
 Spherical coordinates, 71
 Stability analysis, 67
 stability analysis, 69, 93
 cubic, 100
 Stability in counter–current flow, 211
 Stable condition, 67, 108
 stratified flow, 193
 Superficial velocity, 194
 Sutherland’s equation, 12

T

Tank emptying parameters, 177
 Terminal velocity, 206
 thixotropic, 11
 Torricelli’s equation, 179
 Total moment, 77
 Transition to continuous, 143
 Triangle shape, 88, 95
 Turbomachinery, 161
 Two–Phase
 Gas superficial velocity, 199
 Liquid holdup, 199
 Quality of dryness, 199
 Reversal flow, 212
 Slip velocity, 199
 Void Fraction, 199
 Wetness fraction, 199

U

Unstable condition, 67
 Unsteady State Momentum, 153

V

Vapor pressure, 58

Vectors, 228

Vectors Algebra, 228

Vertical counter-current flow, 210

Vertical flow, 193

W

Watson's method, 19

Westinghouse patent, 192

Authors Index**B**

Bhuckingham, 4
Blasiu, 4
Blasius, 4

C

Cichilli, 203

D

Darcy, 4
Duckler, 190, 203

E

Evangelista Torricelli, 179

F

Fanning, 4
Froude, 4

G

Ganguillet, 4

H

Helmholtz, Hermann von, 4

K

Kutta-Joukowski, 4

L

Leibniz, 130
Lockhart, 190

M

Manning, 4
Martinelli, 190
Meye, 4

N

Nikuradse, 4

P

Prandtl, 4

R

Rayleigh, 4, 110
Reynolds, Osborne, 130
Rose, 4

S

Stanton, 4

T

Taitle, 190
Taylor, G.I., 110

V

von Karma, 4

W

Weisbach, 4
Westinghouse, 192