Pumps and Compressors

Marc Borremans

ASME PRESS

with website

WILEY

Pumps and Compressors

Wiley-ASME Press Series

Corrosion and Materials in Hydrocarbon Production: A Compendium of Operational and Engineering Aspects
Dr. Bijan Kermani, Don Harrop

Design and Analysis of Centrifugal Compressors
Rene Van den Braembussche

Case Studies in Fluid Mechanics with Sensitivities to Governing Variables
M. Kemal Atesmen

The Monte Carlo Ray-Trace Method in Radiation Heat Transfer and Applied Optics
J. Robert Mahan

Dynamics of Particles and Rigid Bodies: A Self-Learning Approach
Mohammed F. Daqaq

Primer on Engineering Standards, Expanded Textbook Edition
Maan H. Jawad, Owen R. Greulich

Engineering Optimization: Applications, Methods and Analysis
R. Russell Rhinehart

Compact Heat Exchangers: Analysis, Design and Optimization using FEM and CFD Approach
C. Ranganayakulu, Kankanhalli N. Seetharamu

Robust Adaptive Control for Fractional-Order Systems with Disturbance and Saturation
Mou Chen, Shuyi Shao, Peng Shi

Robot Manipulator Redundancy Resolution
Yunong Zhang, Long Jin

Stress in ASME Pressure Vessels, Boilers, and Nuclear Components
Maan H. Jawad

Combined Cooling, Heating, and Power Systems: Modeling, Optimization, and Operation
Yang Shi, Mingxi Liu, Fang Fang

Applications of Mathematical Heat Transfer and Fluid Flow Models in Engineering and Medicine
Abram S. Dorfman

Bioprocessing Piping and Equipment Design: A Companion Guide for the ASME BPE Standard
William M. (Bill) Huitt

Nonlinear Regression Modeling for Engineering Applications: Modeling, Model Validation, and Enabling Design of Experiments
R. Russell Rhinehart

Geothermal Heat Pump and Heat Engine Systems: Theory And Practice
Andrew D. Chiasson

Fundamentals of Mechanical Vibrations
Liang-Wu Cai

Introduction to Dynamics and Control in Mechanical Engineering Systems
Cho W. S. To

Pumps and Compressors

Marc Borremans
Erasmus University College Brussels
Anderlecht, Belgium

This Work is a co-publication between ASME Press and John Wiley & Sons Ltd.

This edition first published 2019

This Work is a co-publication between John Wiley & Sons Ltd and ASME Press

Registered Offices
John Wiley & Sons, Inc., 111 River Street, Hoboken, NJ 07030, USA
John Wiley & Sons Ltd, The Atrium, Southern Gate, Chichester, West Sussex, PO19 8SQ, UK

Editorial Office
The Atrium, Southern Gate, Chichester, West Sussex, PO19 8SQ, UK

For details of our global editorial offices, customer services, and more information about Wiley products visit us at www.wiley.com.

Wiley also publishes its books in a variety of electronic formats and by print-on-demand. Some content that appears in standard print versions of this book may not be available in other formats.

Library of Congress Cataloging-in-Publication Data

Names: Borremans, Marc, 1951- author.
Title: Pumps and Compressors / Marc Borremans, Erasmus University College Brussels, Anderlecht, Belgium.
Description: First edition. | Chichester, West Sussex : John Wiley & Sons Ltd, [2019] | Series: Wiley-ASME press series | Includes index. | Identifiers: LCCN 2019009784 (print) | LCCN 2019012441 (ebook) | ISBN 9781119534204 (Adobe PDF) | ISBN 9781119534082 (ePub) | ISBN 9781119534143 (hardcover)
Subjects: LCSH: Pumping machinery. | Compressors.
Classification: LCC TJ900 (ebook) | LCC TJ900 .B67 2019 (print) | DDC 621.6/9–dc23
LC record available at https://lccn.loc.gov/2019009784

Cover design: Wiley
Cover image: ©Franco Nadalin/EyeEm/Getty Images, ©esemelwe/Getty Images

Set in 10/12pt WarnockPro by SPi Global, Chennai, India

Printed and bound by CPI Group (UK) Ltd, Croydon, CR0 4YY

10 9 8 7 6 5 4 3 2 1

Contents

Preface

When I studied electro-mechanical engineering at the University of Brussels, my professor of applied mechanics explained how turbopumps and turbines work. He proved the equation of Euler: he drew two curved lines on the blackboard (yeah, that was 1970) and proved Euler's law, a proof of one page. Then he said, "This law applies to all pumps and turbines." That was it. I wondered, "What do I know about a pump or turbine?" I had no answer. It was only when I became a professor and had to teach courses like "Pumps and compressors" and "Steam and gas turbines" and started to read magazines, brochures, and books on the subject that I saw what those devices looked like without a casing and how they worked. I also think that a beautiful or detailed picture can explain much more than text alone or dry formulas. That opinion informs this book. There are more than 700 drawings and pictures in this book. I hope you like them!

I worked a lot as a professor. I started in 1973 giving courses in electricity, electrotechnics, electronics, and high-frequency techniques for four years. Then I became a professor in mechanics, giving courses in thermodynamics, applied thermodynamics (pumps and compressors, combustion engines, steam and gas turbines, refrigeration techniques, heat techniques), materials science, fluid mechanics, strength of materials, pneumatics and hydraulics, CAD2D, CAD3D, CNC, CAM, and so. For these subjects, I designed detailed courses, first on the typewriter (do you know what that is?) and then, from 1983, on the computer. My first computer had an 8-bit processor with 16 kB of RAM and the printer was a matrix printer. In total I offered 45 subjects. When I ended my career, my courses comprised 2857 pages per year.

When I retired in 2007, I started collecting information for this book, beginning with my own course, and spent a year constantly writing on and researching the subject. I wrote in my mother tongue: Dutch.

In 2007, a search for "pumps" on Google elicited no fewer than 94 million references. The word "compressor" had a hit rate of 18 million. This is because both devices account for a significant part of the infrastructure of buildings, houses, and factories. It is reckoned that in a petrochemical plant there is one pump installed for every employee.

With this knowledge and 35 years' teaching experience, I started to draw up my own sort of encyclopedia. I collected as much information as possible from the literature, company brochures, and the Internet in order to catalogue as many of the pumps and compressors on the market as possible, writing a description of them, including their properties. It is left to the reader to find out which pump or compressor is the best for a certain job. That choice will not be distilled immediately from the book. The choice of a pump or compressor is not an exact science; it is an assessment of the pros and cons before a definitive choice can be made. Reasons for choosing a pump for a specific job are based on price, maintenance, lifecycle, regulation, type of fluid, etc. A reference list at www.wiley.com/go/borremans/pumps provides a lot of information including videos and animations that can be found on the internet for most types of pumps and compressors.

Much later, in 2018, I translated it into English, not without doing more research over several months. What you hold in your hands is the result. Maybe you could do the same work. But by buying this book you spare yourself a lot of time and money. If you now search Google using the word "pump", you will get a lot of hits, but a lot of these will concern "shoe pumps." For the word "compressor" you'll get 21 million hits.

Beyond this, most pictures in the book are not available anymore on the Internet. Nowadays, nearly all companies just show the casings of their pumps and compressors. Somebody told me it is because they don't want their ideas to be stolen. But to me that is pointless: rival companies can just buy a pump, dismantle it, and reverse engineer it. Just like the Japanese did after World War II, but they added the concept of constant quality and so made many improvements.

This book also uses concepts of fluid mechanics and thermodynamics, two subjects I taught. In fact, pumps and compressors apply the concepts of these two basic branches of engineering science. Don't worry if this isn't your area of expertise: the information you will need to understand these branches is given in this book. When I started my career the subject I taught was called "Applied mechanics and thermodynamics" and later it was separated into "Pumps and compressors," "Combustion engines and turbines," and "Refrigeration techniques."

This book is intended for technical high school students, college students, plant engineers, process engineers, and pump and compressor sales reps. Of course, in high schools, one has to make abstract on the mathematical framework. The book is also a kind of encyclopedia of the greater part of pumps and compressors on the market. It is impossible for a teacher or professor to go through the whole book in one course.

I use simple language to explain everything and in the hope that it will be easy for the reader to follow the reasoning. I oppose writing that forces the reader first has to make a grammatical analysis of every sentence.

Marc Borremans
borremans.m@telenet.be
https://www.borremansengineering.com

Acknowledgment

This colorful book wouldn't have been possible without the contribution of 67 enthusiastic companies, all over the world. They allowed me to use their pictures. They are, in alphabetical order:

Aerzen	Egger
Agilent	Eriks
Alfa Laval	Flowserve
Alup	Friotherm
Allweiler	Johnson
Andritz	Gardner Denver
Ateliers François	Gea
Atlas Copco	Glynwed/Reinhütte pumpen
Begeman	Grasso
Bitzer	Hibon
Boge	Hoerlinger Valve
Burhardt	Hermetic pumps GmbH
Börger	Hus (Verder)
Bornemann	Ingersoll Rand
Bosch-Rexroth GmbH	iLLmVARC
Cameron Compressor Systems	ITT Goldpumps
Clasal	SPXFlow
Cornell	Kaeser
Dab	Klima
Daikin,	Knf
Daurex	Koerting
Dewekom Engineering	KSB
Dura	LewVac
Ensival Moret	Leybold Vacuum
Ebbm-Papst	PDC Machines

Pfeiffer
Psg Dover
Ritz
Rotojet
Seepex
Sepco
Sihi-Sterling (Flowserve)
Speck Pumpe
Scam-Torino
Shamai
SKF
Systemair
Sulzer
Stork
Verderair
Viking Pump
Warman
Watts
Weir
Wemco

I would like to thank some people from Wiley, as well: Eric Willner for believing in the project, Steve Fassioms, Tim Bettsworth (the man with the sharp eye) for the editing, and in advance the typesetters, for their excellent work.

Used Symbols

Symbol	Meaning	Unit
a	Acceleration	m/s^2
a'	Acceleration in suction line	m/s^2
A	Section piston	m^2
A	Cross section channel	m^2
A	surface	m^2
A'	Section suction line	M^2
A'	Perpendicular surface	m^2
$c_\perp$	Velocity perpendicular on surface	m/s
$c_\parallel$	Velocity along surface	m/s
c	Absolute velocity at impeller	m/s
c	Absolute at inlet rotor	m/s
c_l	Velocity lower surface hydrofoil	m/s
c_p	Specific heat at constant pressure	J/kg.K
c_v	Specific heat at constant volume	J/kg.K
c_u	Velocity upper surface hyrdofoil	m/s
c_{1r}	Radial component	m/s
C_L	Lift factor	-
C_D	Drag factor	-
D	Diameter	m
D	Drag force	N
D	Diameter impeller	m
D_H	Hydraulic diameter	m
F	Force	N
F_C	Centrifugal force	N
g	Gravity acceleration	m/s^2
H	Height	m
h	Depth impeller	m

Symbol	Meaning	Unit
h	Specific enthalpy	J/kg
H_{geo}	Geodetic height	m
H_{man}	Manometric height (head)	m
H_p	Geodetic press height	m
H_s	Geodetic suction height	m
$H_{s,max}$	Maximum suction head	m
H_f	Friction loss head	m
k	Absolute roughness	m
l_1	Distance covered during suction stroke	m
l_2	Distance covered during press stroke	m
L	Length piston rod	m
L	Lift force	N
L	Length pipe line	m
L'	Fictive length suction line	m
m	Mass	kg
m^*	Displaced mass per cylinder	kg
M	Molar mass	kg/kmol
n	Polytropic exponent	-
N	Speed	rmp
N_s	Specific speed	$m^{3/4} \cdot s^{-3/2}$
N_{ss}	Suction specific speed	$m^{3/4} \cdot s^{-3/2}$
N_q	Dimionless specific speed	-
N_ω	Dimensionless specific speed	-
$NPSH_a$	Available net positive suction head	m
$NPSH_{ss}$	Suction net positive suction head	-
$NPSH_r$	Required net positive suction head	m
O_{cd}	Surface under curve cd	J
O_{ab}	Surface under curve ab	J
p_v	Vapor pressure	Pa
p	Static pressure	Pa
p_a	Atmospheric pressure	Pa
p_{abs}	Absolute pressure	Pa
p_{eff}	Effective pressure	Pa
p_{dyn}	Dynamic pressure	Pa
p_{man}	Manometric (feed) pressure	Pa
p_{geo}	Geodetic (feed) pressure	Pa
$p_{v,p}$	Vapor pressure press vessel	Pa
$p_{v,s}$	Vapor pressure suction vessel	Pa

Symbol	Meaning	Unit
$p_{r,p}$	Pressure press chamber	Pa
$p_{r,s}$	Pressure suction chamber	Pa
$p_{f,s}$	Friction pressure suction pipe	Pa
$p_{f,p}$	Friction pressure press pipe	
$p_{l,p}$	Friction loss in pump	
p_s	Static feed pressure	Pa
$P_{tot,s}$	Total pressure suction side	Pa
$P_{tot,p}$	Total pressure press side	Pa
P_t	Technical power on drive shaft	W
q	Specific heat	J/kg
Q_M	Mass flow	Kg/s
Q_V	Volumetric flow	m^3/s
$Q_{v,a}$	Average volumetric flow	m^3/s
Q_{Veff}	Effective volumetric flow	m^3/s
$Q_{n,nom}$	Nominal effective flow	m^3/s
u	Peripheral speed	m/s
U	Voltage	V
R	Universal gas constant	J/kmol.K
R, r	Crankstroke, radius	m
s	Stroke length	m
T	Absolute temperature	K
u	Circumpheral velocity impeller	m/s
v	Specific volume	m3/kg
V	Total volume	m3
w	Relative velocity	m/s
w	Specific work	J/kg
w_c	Specific compression work	J/kg
w_t	Specific technical work	J/kg
W_c	Total compression work	J
W_t	Total technical work	J
x	Position	m
z	Number of vanes (channels)	-

Greek Symbols

α	Absolute angle impeller	° or rad
β	Relative angle impeller	° or rad
γ	Isentropic exponent	-

n	Polytropic exponent	-
δ	Correction factor laminar flow	-
ε	Factor dead volume	-
ρ	Specific mass	kg/m^3
θ	Angle of incidence	° or rad
Δ	Difference	-
Δpot	Specific change of potential energy	J/kg
Δkin	Specific change of kinetic energy	J/kg
ΔH	Geodetical height	m
λ	Hydraulic resistance factor	-
λ	Volumetric efficiency	-
λ	Mean free length	m
τ	Formation time	s
$\varphi(..)$	Function of	-
ω	Angular velocity	rad/s
ζ	Correction factor laminar flow	-
ν	Kinematic viscosity	m/s^2
ν	Axial velocity	m/s
η	Dynamic viscosity	Pa·s
ξ	Hydraulic resistance factor	-
Γ	Reaction degree	-
Σ	Sum	-
Π	Product	-
Π_1	1st number of Rateau	-
Π_2	2nd number of Rateau	$m^{-1}\cdot s^2$

Indexes

1	State 1, inlet impeller
2	State 2, outlet impeller
3	State 3
a	State a
b	State b
c	State c
d	State d
d	Dead
s	Stroke
a	allowable
c	compression

t	technical
ab	State change ab
ac	State change ac
cd	State change cd
12	State change 12
13	State change 13
23	State change 23

Upper indexes

‘	1st stage
“	2nd stage

About the Companion Website

This book is accompanied by a companion website:

www.wiley.com/go/borremans/pumps

The website includes:

- References
- Videos

Scan this QR code to visit the companion website.

Part I

Pumps

1

General Concepts

CHAPTER MENU

1.1 Hydrostatics

Consider an incompressible liquid at rest. The law of Pascal applies (Figure 1.1):

$$p = p_a + \rho \cdot g \cdot H$$

where

- p: the static pressure at the considered point [Pa = N/m^2]
- p_a: atmospheric pressure (ca. 1 [bar] = 10^5 [Pa])
- ρ: specific mass of the liquid [kg/m^3]
- g: gravitational acceleration (9.81 [m^2/s])
- H: height beneath the liquid surface [m]).

The standard pressure at sea level amounts to 1.013 [bar]. This pressure is the *absolute pressure*.

In practice the *relative*, or *effective*, pressure p_{eff} is of importance. This is the difference between the absolute pressure p_{abs} and the atmospheric pressure p_a:

$$p_{eff} = p_{abs} - p_a$$

Pumps and Compressors, First Edition. Marc Borremans.

Companion website: www.wiley.com/go/borremans/pumps

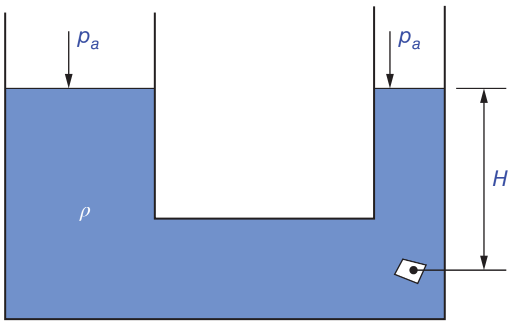


Figure 1.1 Law of Pascal.

One can distinguish the following pressure measurement apparatus:

- A *manometer* usually measures the effective pressure; this is an overpressure (mostly with respect to the atmospheric pressure).
- A *vacuum meter* measures an underpressure (with respect to the atmospheric pressure).
- A *barometer* measures the absolute pressure of the atmosphere.

Concerning the unities nowadays a consistent notation is held. One notes [bar(g)] of [bar_g] for effective pressure ("g" stands for gauge) and [bar(a)] or [bar_a] for absolute pressure.

1.2 Flow

Consider a pipe with variable section (Figure 1.2). At section 1 a fluid (gas or liquid) possesses a velocity c and a specific mass ρ. The cross-section there is A_1. Use an analogous notation for section 2.

The *mass flow*, i.e. the amount of mass that per unit of time flows through the section, also flows through section 2 (conservation of mass).

The mass flow Q_M is given by:

$$Q_M = \rho_1 \cdot c_1 \cdot A_1 = \rho_2 \cdot c_2 \cdot A_2 \; [\text{kg/s}]$$

In the case of an incompressible fluid (liquids) ρ is constant. One can use the *volumetric flow* Q_V instead, i.e. the amount of volume that flows per unit of time through a section:

$$Q_V = c_1 \cdot A_1 = c_2 \cdot A_2 \; \left[\frac{m^3}{s}\right]$$

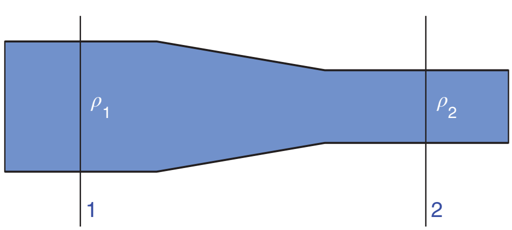


Figure 1.2 Flow.

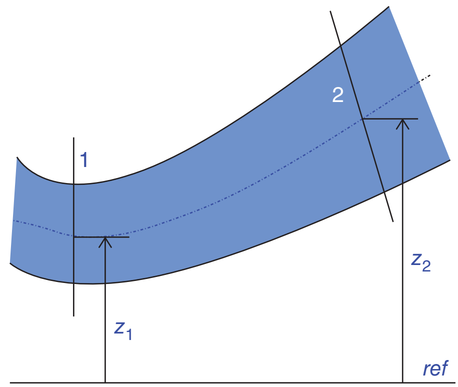


Figure 1.3 Law of Bernoulli.

1.3 Law of Bernoulli

Bernoulli expresses the conservation of energy for liquids. A fluid possesses pressure energy (dynamic pressure) and potential energy (Figure 1.3):

$$p_1 + \rho \cdot g \cdot z_1 + \rho \cdot \frac{c_1^2}{2} = p_2 + \rho \cdot g \cdot z_2 + \rho \cdot \frac{c_2^2}{2}$$

where z is the height coordinate.

1.4 Static and Dynamic Pressure

Consider a horizontal pipe. The law of Bernoulli applied to points 1 and 2 (Figure 1.4):

$$p_1 + \rho \cdot \frac{c_1^2}{2} = p_2 + \rho \cdot \frac{c_2^2}{2}$$

Point 2 is a *stagnation point*:

$$c_2 = 0$$

So:

$$p_2 = p_1 + \rho \cdot \frac{c_1^2}{2}$$

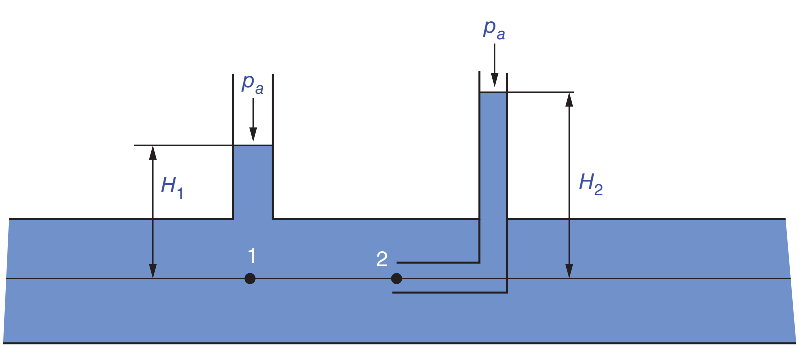


Figure 1.4 Static and dynamic pressure.

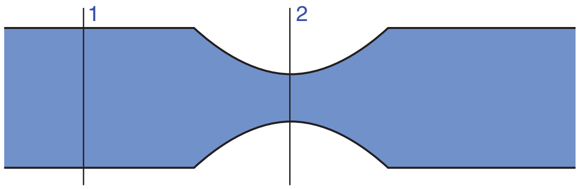


Figure 1.5 Venturi.

From which: $p_2 > p_1$
In vertical sense there is no movement and one can apply hydrostatics:

$$p_1 = p_a + \rho \cdot g \cdot H_1$$

$$p_2 = p_a + \rho \cdot g \cdot H_2$$

From which

$$p_2 = p_1 + \rho \cdot g \cdot (H_2 - H_1)$$

Look now at a tube with a narrowed passage (a venturi) (Figure 1.5):

$$Q_v = c_1 \cdot A_1 = c_2 \cdot A_2 \text{ with: } A_1 > A_2$$

$$c_1 < c_2$$

In the throat the liquid velocity will be greater than elsewhere.
The dynamic pressure in the throat is greater than elsewhere.
Application of Bernoulli leads to:

$$p_1 + \rho \cdot \frac{c_1^2}{2} = p_2 + \rho \cdot \frac{c_2^2}{2}$$

So that

$$p_1 > p_2$$

One finds that static pressure can be converted into dynamic pressure, and vice versa. That's why in many considerations in applied mechanics one often speaks of the *total pressure* of a fluid being the sum of static and dynamic pressure. The *total pressure* then expresses the "total energy content" of a liquid.

1.5 Viscosity

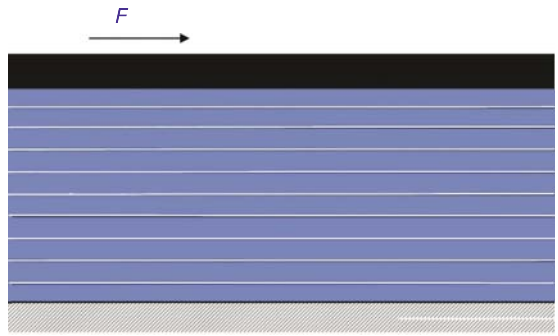
This paragraph is valid for liquids as well as for gasses, so we use the generic word "fluid."

Consider a fluid in an open channel (Figure 1.6). With the help of a plate a horizontal force F is applied in order to move the fluid. Let's imagine that the fluid consists of horizontal layers. At the top of the fluid the layer "adheres" by cohesion forces to the plate.

This layer thus moves with the velocity of the plate. At the bottom, however, the layer does not move at all because there it is bounded by cohesion forces to the bottom of the channel. It then is clear that every layer will possess its own velocity c, evolving from the highest velocity at the top to velocity zero at the bottom.

Every layer will exercise a resistance on the adjacent layer (a shearing stress): the layer at the top will have a braking action on the second layer, the third layer will be braked by the second one, and so on… One speaks of "viscous friction" between the layers.

Figure 1.6 Viscosity.



If one wants to maintain a velocity of a moving fluid then a force F is necessary. Newton states that in order to keep an object at constant speed no force is necessary (of course this holds true only if there are no disturbing forces, like here). This force overcomes the internal friction of the fluid, i.e. the friction that the layers exercise on each other.

For so-called *Newtonian* liquids and gasses, the next law applies:

$$F = \eta \cdot A \cdot \frac{dc}{dz}$$

where:
A [m^2]: the section of which the force F is applied
c [m/s]: the local velocity of the fluid
z [m]: the position on the vertical axis
η [Pa · s]: the dynamic viscosity (old unit, non-SI: 1 Poise = 0.1 [Pa · s])
The unit Poiseuille is a shortcut for 1 [Pa · s].

The preceding case of an open channel can easily be extended to the case of a tube wherein a fluid moves: it suffices to mirror the case of Figure 1.6 around the plate (Figure 1.7).

Then, too, a force F is necessary to guide a fluid through a pipe, or in other words a pressure difference over the pipe is needed to compensate for the internal friction of the fluid.

The order of magnitude of the dynamic viscosity for gasses at room temperature is $10 \cdot 10^{-6}$[Pa·s].

In the range of low pressures, this is from 0.1 to 10 [bar], for ideal and real gasses η is independent of the pressure.

Figure 1.7 Channel.

The dynamic viscosity for liquids varies between large ranges, so typical values cannot be given. With increasing pressure, the dynamic viscosity of most liquids increases nearly proportionally with the pressure.

For gasses, the viscous effects come about by exchanges of impulse (mass multiplied by velocity) of the molecules: when a layer of gas is brought into movement the molecules in that layer will lose kinetic energy because of collisions with other molecules. For liquids, viscosity is caused by the intramolecular cohesion forces that brake the shift of the layers. That's why the viscosity of liquids decreases with increasing temperature there where for gasses the increasing motion of the molecules promotes the exchange of impulse so that the viscosity of gasses increases with temperature.

The dynamic viscosity of liquids at moderate pressures decreases exponentially with temperature. But, for gasses, by approximation:

$$\eta \div \sqrt{T}$$

The force F varies in every layer. So, we should write:

$$F = \eta \cdot A \cdot \frac{\partial c}{\partial z}$$

where F is the local force and $\frac{\partial c}{\partial z}$ is the local velocity gradient, or *rate of shear deformation*.

Dividing F by A leads to the local shear stress τ (Figure 1.8 for *a laminar flow* – see later).

$$\tau = \frac{F}{A} = \eta \cdot \frac{\partial c}{\partial z}$$

See Appendix A for a calculation of the velocity profile and the mean velocity c_m.

Not all liquids behave as Newtonian fluids (like water); in the pump industry the following liquids are common:

- *Pseudo plastic liquids*: the viscosity will decrease with increasing shear stress, e.g. paint and shampoos.
- The inverse includes *dilatant liquids*: the viscosity will increase as the shear stress increases, e.g. honey and quicksand.

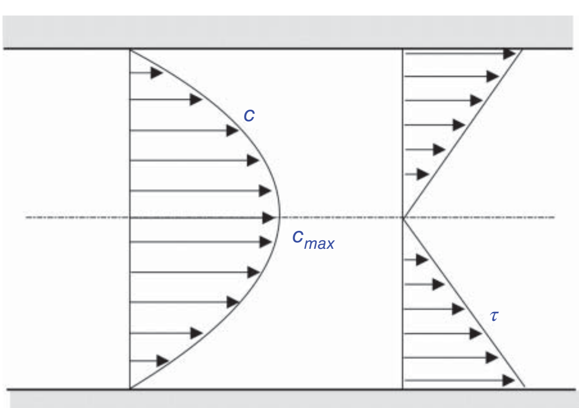


Figure 1.8 Velocity and shear stress profile in laminar flow.

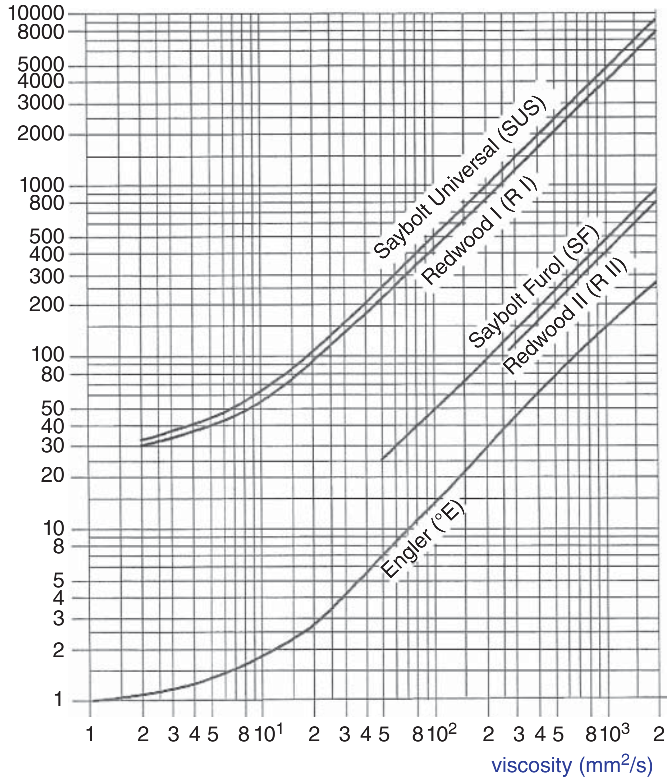
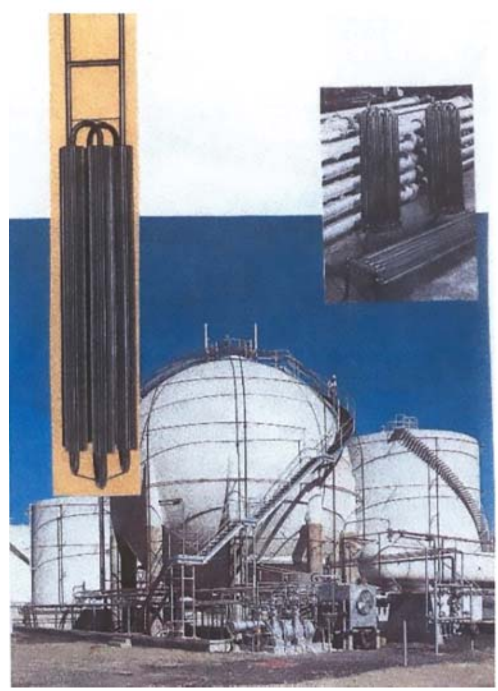


Figure 1.9 Conversion between different viscosity units.

Figure 1.10 Heating of heavy oils.



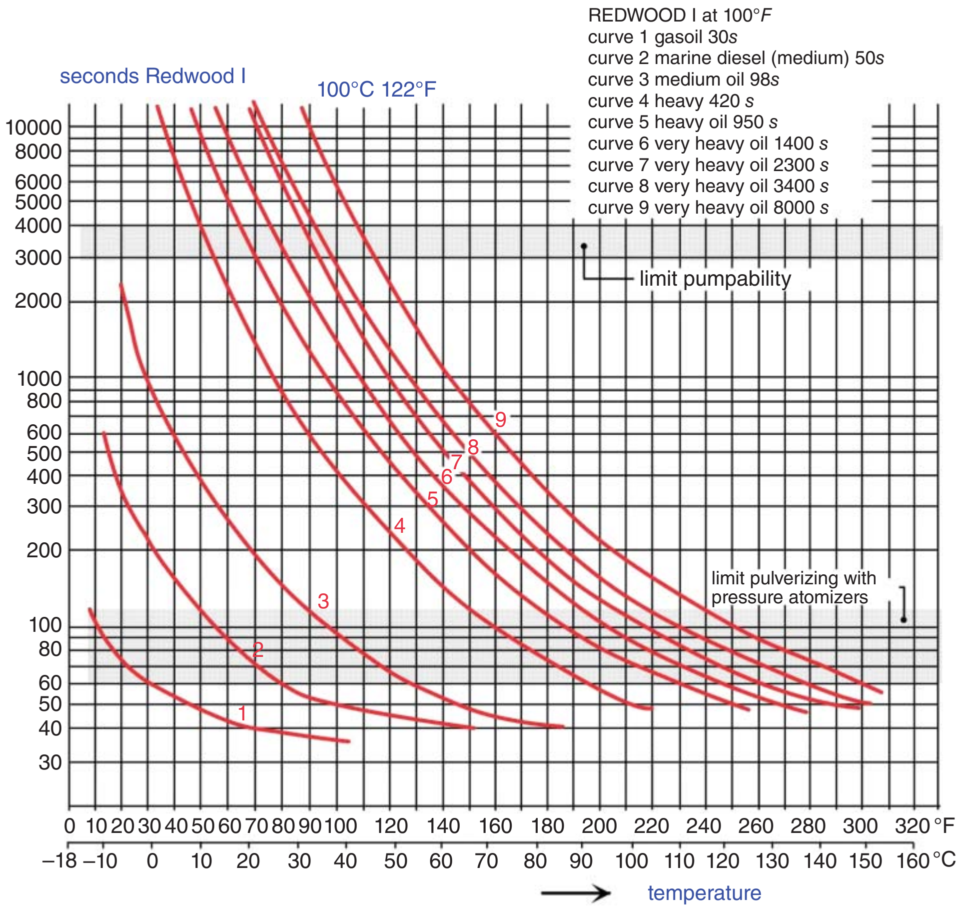


Figure 1.11 Heavy oils and their viscosity.

- A *plastic liquid* behaves very strangely: applying a certain shear stress leads to a Newtonian behavior, but with increasing shear stress it becomes pseudoplastic and then dilatant, e.g. ketchup and lubricating grease.
- *Thixotropic liquids* will, when the shear stress disappears, show an increasing viscosity with evolving time, e.g. yogurt, print ink, and sludge. When they are in a pump and the pump works they become thin. But if the pump stops, they become thick. After a while the pump is not able to pump them anymore.

In practice it turns out that the expression $\frac{\eta}{\rho}$ frequently appears in equations. That's why one defines the kinematic viscosity ν:

$$\nu \equiv \frac{\eta}{\rho}$$

The units are: [m²/s]

Sometimes one uses an old non-SI unit, the Stokes. 1 [St] = 10^{-4} [m^2/s]. Other so-called technical units are Redwood I, Redwood II, Saybolt universal, Saybolt FUROL and Engler degrees.

Table 1.1 Guide values for fluids.

Medium		m/s
Oil	In pipelines	1–3
Water	In long pipelines	0.5–1
	Behind piston pumps	1–2
	Behind turbopumps	1.5–3
	For turbines	2–7
Gas	Low pressure	5–30
	Middle pressure	5–20
	High pressure	3–6
Compressed air	In pipelines	2–4
Vapor	1–10 [bar]	15–20
	10–40 [bar]	20–40
	40–125 [bar]	30–60

An oil with a viscosity of 3.6 [mm^2/s] at 20 °C will need twice as much efflux time than water with a viscosity of 1.8 [mm^2/s] at 20 °C.

The conversion between the different viscosity units is represented in Figure 1.9.

As pointed out earlier, the viscosity of a liquid decreases with temperature. That may imply that some liquids have to be heated before transport and to be able to form drops in an atomizing burner (Figures 1.10 and 1.11).

Guide values for fluids are given in Table 1.1.

1.6 Extension of Bernoulli's Law

Consider a pipeline with friction loss (Figure 1.12).

The classic law of Bernoulli:

$$p_1 + \rho \cdot \frac{c_1^2}{2} = p_2 + \rho \cdot \frac{c_2^2}{2}$$

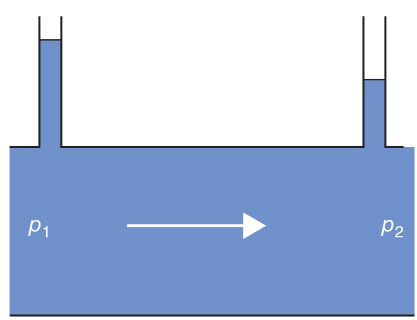


Figure 1.12 Friction loss.

is not valid anymore because:

$$p_1 + \rho \cdot \frac{c_1^2}{2} > p_2 + \rho \cdot \frac{c_2^2}{2}$$

Note the pressure loss by friction p_f, then:

$$p_1 + \rho \cdot \frac{c_1^2}{2} = p_2 + \rho \cdot \frac{c_2^2}{2} + p_f$$

In general this becomes:

$$p_1 + \rho \cdot \frac{c_1^2}{2} + \rho \cdot g \cdot z_1 = p_2 + \rho \cdot \frac{c_2^2}{2} + \rho \cdot g \cdot z_2 + p_f$$

Dividing by $\rho \cdot g$ leads to the law of Bernoulli being seen in the unit meter:

$$\frac{p_1}{\rho \cdot g} + \frac{c_1^2}{2 \cdot g} + z_1 = \frac{p_2}{\rho \cdot g} + \frac{c_2^2}{2 \cdot g} + z_2 + H_f$$

With the friction loss in meters:

$$H_f = \frac{p_f}{\rho \cdot g} \text{ [m]}$$

1.7 Laminar and Turbulent Flow

Consider a vessel where a pipeline is connected to (Figure 1.13). On the pipeline a valve is connected so that the flow can be regulated and thus the velocity of the liquid also. Color is then added via an opening. When the velocity is not too high the stream is very regular: the colored parts run in parallel layers. This is a *laminar* flow. At higher velocities the color particles form an irregular path and begin to mix with the liquid; the layers exchange energy with each other. This is a *turbulent* flow.

The difference between laminar and turbulent flow is made by the number of Reynolds (*Re*), defined by:

$$\text{Re} \equiv \frac{c_m \cdot D}{\nu}$$

where:

c_m: the mean velocity of the liquid [m/s]
D: the diameter of the pipeline [m]
ν: the kinematic viscosity [m^2/s]

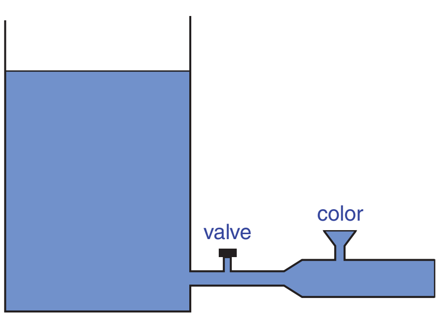


Figure 1.13 Types of flow.

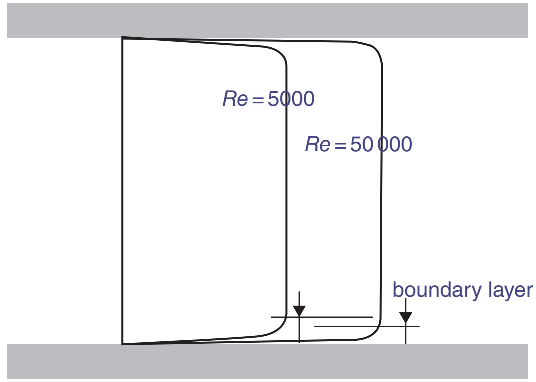


Figure 1.14 Turbulent flow: velocity profile.

It is easy to check that the number of Reynolds is, indeed, dimensionless.

When for a certain flow in a round tube Reynolds is lower than 2300 the flow is laminar; otherwise, it is turbulent.

In a laminar flow the friction losses p_f are proportional to the liquid velocity c, but in a turbulent flow they are quadratically proportional to this velocity.

The question arises if in practice one deals with laminar or turbulent flows. Take the case of water at 10 °C, flowing through a pipe with a diameter of 0.1 [m]. In order for the flow to be laminar the velocity may not be higher than:

$$c_m = \frac{2300 \cdot 1,3 \cdot 10^{-6}}{0,1} = 0.03\ [\mathrm{m/s}]$$

This velocity is very low. Practically one deals mostly with a turbulent flow. Only with very viscous oils or substances is the flow laminar. But, in a turbulent flow, the friction losses increase very quickly with the velocity. This last one will be limited as much as possible. For water in pipelines: ca. 1.5 [m/s].

In a turbulent flow the velocity profile is quite different from that of a laminar flow (Figure 1.14). Apart from a boundary layer, where the flow is laminar because of shear stresses, the velocity is constant. This is because the particles of the liquid are mixed and in this way exchange impulse with each other: so no difference in velocity can exist; if it did, *it would be destroyed by the interaction of the particles.*

Sometimes the liquid is so viscous that it has to be heated before transportation or pulverization in a nozzle from a burner.

1.8 Laminar Flow

1.8.1 Hydraulic Resistance

For a laminar or turbulent flow the following expression is valid:

$$p_f = \lambda \cdot \frac{L}{D} \cdot \frac{\rho \cdot c_m^2}{2}$$

It is possible to prove that for a laminar flow (see Appendix A):

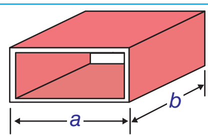
$$\lambda = \frac{64}{Re}$$

For a noncircular section, still for a laminar flow, one introduces a correction factor:

$$\lambda = \zeta \cdot \frac{64}{Re}$$

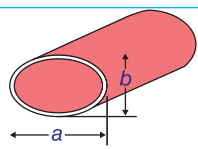
Example 1.1 ***Concentric eccentric profile***
The value of ζ can be found in tables or diagrams (Figure 1.15).

Example 1.2 ***Rectangular profile***



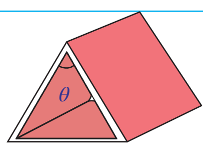
a/b	1	2	3	4	6	8	∞
ζ	0.98	0.97	1.07	1.14	1.23	1.29	1.5

Example 1.3 ***Ellipse***



a/b	1	2	4	8	16
ζ	1	1.05	1.14	1.20	1.22

Example 1.4 ***Isosceles triangle***



θ	10°	30°	60°	90°	120°
ζ	1.79	0.82	0.83	0.82	0.80

1.8.2 Hydraulic Diameter

Consider a noncylindrical pipeline (Figure 1.16) of length L. In general that can also be an *open* channel. The cross-section where the fluid flows through is labelled A_w, the wet perimeter P_w. The aim is now to find an equivalent cylindrical tube for such cases. The diameter of that pipeline is called the *hydraulic diameter* D_H.

In order to let the noncylindrical pipeline behave like a circular line of diameter D and length L the viscous shear stresses in the fluid should lead to the same pressure drop p_f.

Isolate in both cases a piece of liquid mass between section 1 and 2.

On this mass the following forces are acting:

- The liquid on the left exercises a force $p \cdot A$, with:

$$A = \frac{\pi \cdot D_H^2}{4}$$

- The liquid on the right exercises a force $(p - p_f) \cdot A$

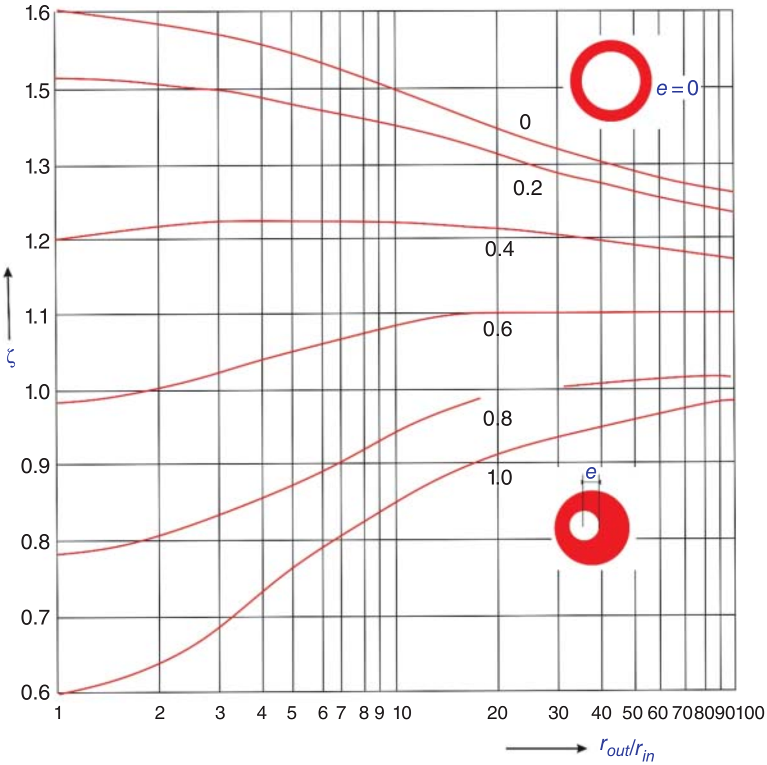
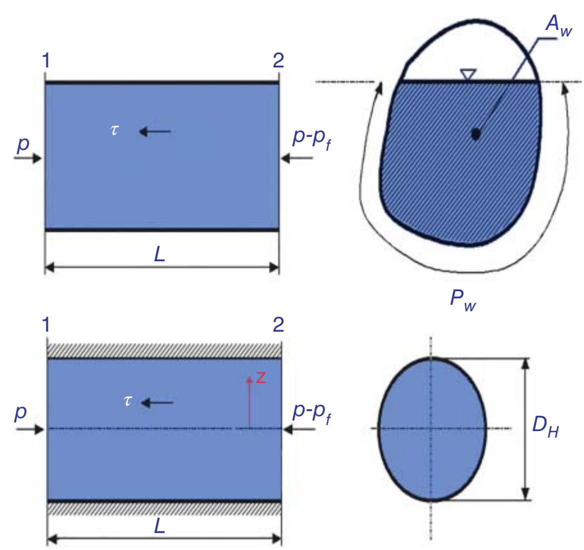


Figure 1.15 Concentric eccentric profile: (a) rectangular profile; (b) ellipse; (c) isosceles triangle.

Figure 1.16 Hydraulic diameter.



- At the cylinder wall shear stresses are exercised according to:

$$\tau = \eta \cdot \left(\frac{dc}{dz}\right)_{\frac{D_H}{2}}$$

The total shear force is then:

$$\tau \cdot (\pi \cdot D_H) \cdot L$$

where $\pi \cdot D_H$ is the surface on which the shear stress is active.

If the liquid mass in the pipeline or open channel moves with a constant mean velocity, the sum of all forces is zero, according to Newton's law:

$$\frac{\pi \cdot D_H^2}{4} \cdot p_f - \tau \cdot (\pi \cdot D_H) \cdot L$$

From where:

$$p_f = \frac{4 \cdot \tau \cdot L}{D_H}$$

Isolate now the liquid mass that flows between sections 1 and 2 of the noncylindrical channel.

On this mass the following forces are acting:

- The force $p \cdot A_w$
- The force $(p - p_f) \cdot A_w$
- The shear force $\tau \cdot P_w \cdot L$

where she shear stresses are active on the *wet* surface.

If the liquid mass is moving at a constant mean speed then the sum of all these forces is zero:

$$A_w \cdot p_f - \tau \cdot P_w \cdot L = 0$$

From where the pressure drop:

$$p_f = \frac{\tau \cdot P_w \cdot L}{A_w}$$

Identification of the two expressions for p_f:

$$D_H = \frac{4 \cdot A_w}{P_w}$$

where the number of Reynolds is calculated with the hydraulic diameter as character dimension:

$$Re = \frac{c \cdot D_H}{\nu}$$

Example 1.5 ***Square tube***

$$D_H = \frac{4 \cdot A_w}{P_w} = \frac{4 \cdot a^2}{4 \cdot a} = a$$

Example 1.6 ***Rectangular tube***

$$D_H = \frac{4 \cdot A_w}{P_w} = \frac{4 \cdot a \cdot b}{2 \cdot (a + b)} = \frac{2 \cdot a \cdot b}{a + b}$$

Example 1.7 ***Concentric passage***

$$D_H = \frac{4 \cdot A_w}{P_w} = \frac{4 \cdot \pi \cdot (r_{out}^2 - r_{in}^2)}{2 \cdot \pi \cdot (r_{our} + r_{in})} = 2 \cdot (r_{out} - r_{in})$$

1.9 Turbulent Flow

For straight pipes the pressure (friction) loss p_f is given by the formulae of Darcy–Weisbach:

$$p_f = \lambda \cdot \frac{L}{D} \cdot \left(\rho \cdot \frac{{c_m}^2}{2} \right)$$

Herein:

- c_m: mean velocity in pipe [m/s]
- D: diameter pipe [m]
- L: length of pipe [m]
- λ: hydraulic resistance factor (dependent on many factors, such as fluid, dimensions, etc.), dimensionless

To determine the value of λ one dispenses with tables, graphs, and empirical formulas.

Example 1.8 ***For smooth pipes and $3 \cdot 10^3 < Re < 10^6$***
Formula of Blasius: $\lambda = 0.316 \cdot \mathrm{Re}^{-0.25}$
In case the pipe is rough inside, apart from the internal friction in the fluid, friction may occur against the wall of the pipe. This leads to additional losses (see later literature).

Example 1.9 ***For water***
Formula of Lang: $\lambda = 0.02 + \frac{0{,}0018}{\sqrt{c \cdot D}}$
(often the approximation: $\lambda = 0.03$ is made)
On the other hand, pressure losses occur in all sorts of "obstacles" like bends, elbows, widenings or restrictions.
For these cases one uses the general formula:

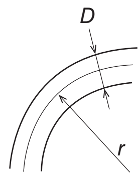
$$p_f = \xi \cdot \left(\rho \cdot \frac{{c_m}^2}{2} \right)$$

The values of ξ also are extracted from tables, graphs, and empirical formulas. Turbulent flows are characterized by the independence of *Re*.

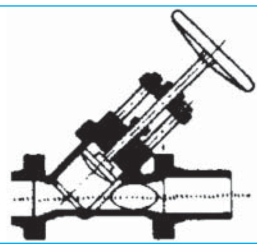
Table 1.2 Perpendicular bend.

D/r	0.4	0.6	0.8	1	1.2	1.4	1.6	1.8
φ	0.13	0.18	0.25	0.4	0.64	1	1.55	2.17

Example 1.10 *Perpendicular bend (Table 1.2)*

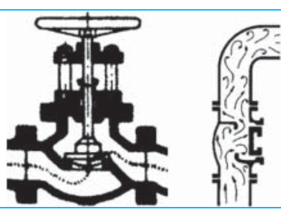


Example 1.11 *Slide shut-off valve*



$\xi = 0.8$

Example 1.12 *Plug valve*

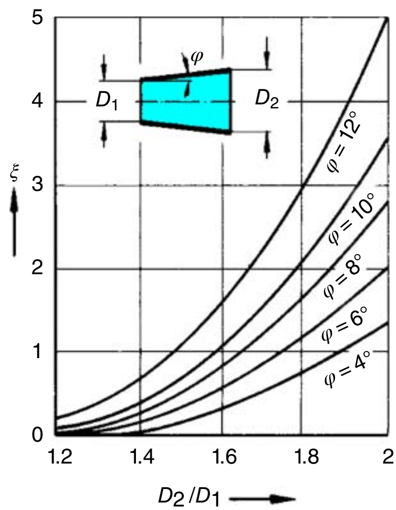


$\xi = 2$

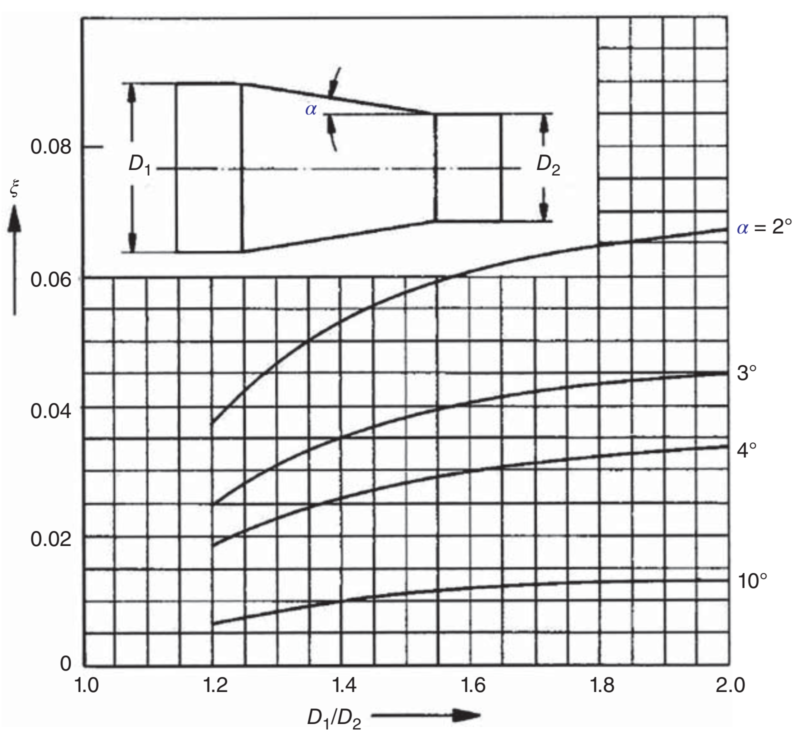
Example 1.13 *Suction strainer*

$\xi = 2$

Example 1.14 ***Diffusor***
When one looks at the value of ξ in the case of widenings (diffusors), it is clear that it increases very fast with the top angle of the cone $(2 \cdot \varphi)$. For that reason, with regard to minimizing the friction losses, one will limit this top angle to 10°.



Example 1.15 ***Converging passage***



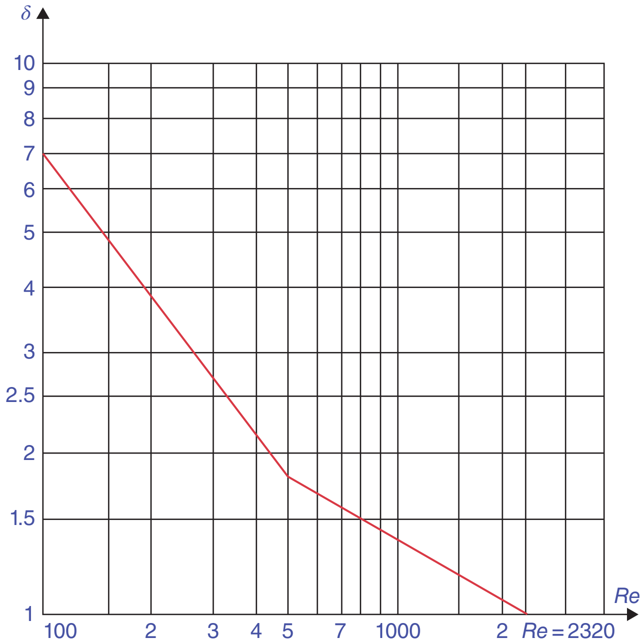


Figure 1.17 Conversion factor δ.

For a laminar flow the resistance factor ξ can be found by first calculating ξ for a turbulent flow and correcting with a conversion factor δ:

$$\xi_{lam} = \delta \cdot \xi_{turb}$$

where δ can be found from the diagram in Figure 1.17.

1.10 Moody's Diagram

Moody's diagram is valid for circular tubes in laminar flow and for all tubes with hydraulic diameter D_H in turbulent flow (Figure 1.18).

The surface condition of the inner side of the pipe is given by its relative roughness $\epsilon = \frac{k}{D}$, where D is the diameter of the tube and ϵ the roughness of the inner surface. The absolute roughness is not identical to the technical or natural roughness but is an equivalent, artificial roughness that is defined as *sand roughness*. It is given by sand grains with diameter k on smooth tubes that reproduce the natural roughness of sand (Figure 1.19).

In general: $\lambda = \varphi(Re, \varepsilon)$ (Table 1.3).

The friction factor λ increases in old pipes because of corrosion and sediments. According to the engineer who did the experiments the resistance must be multiplied with an aging factor(see Table 1.4).

Example 1.16 Consider a pumping installation with a water flow $Q_V = 360$ [m^3/h]. The geodetic suction head is 2.8 [m] and the geodetic pressure head 22 [m]. The suction pipe is 6 [m] long and is provided with a suction strainer with check valve ($\xi = 4$)

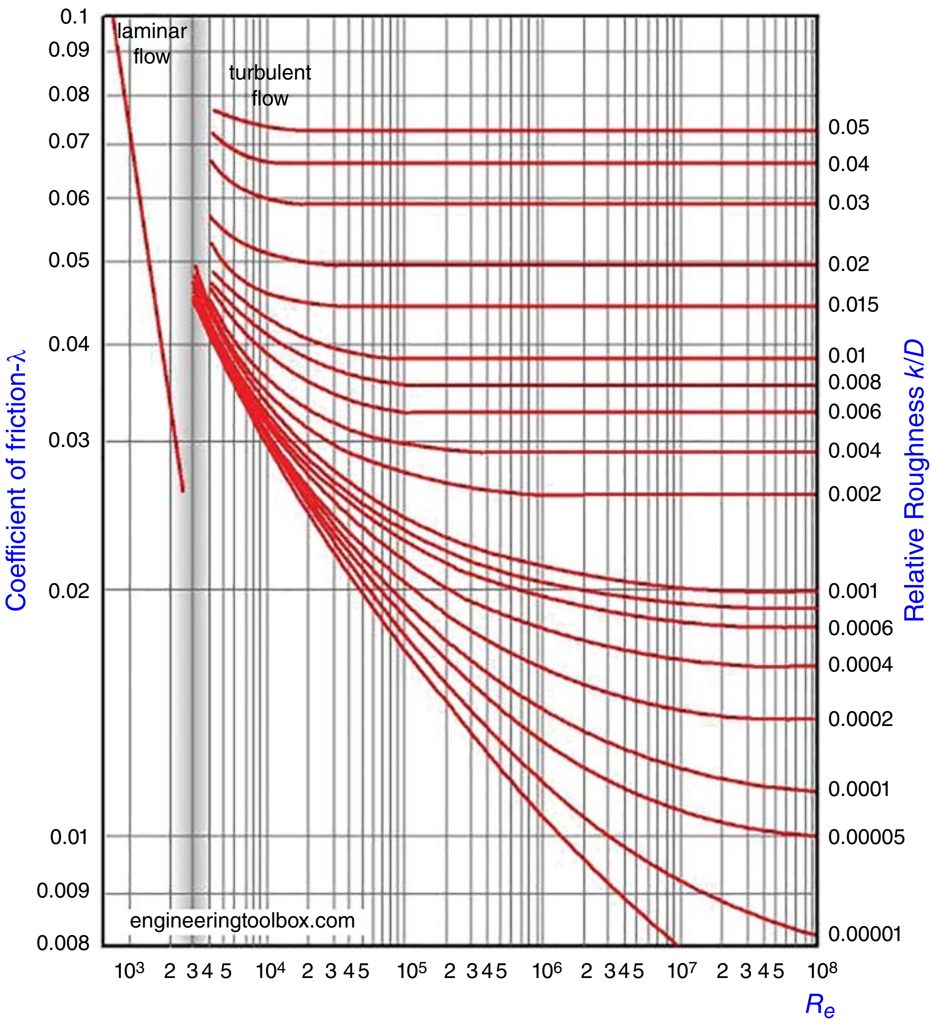


Figure 1.18 Moody diagram. (Source: Courtesy of Engineering Toolbox).

and a bend with radius 0.5 [m]. The press pipe is 150 [m] long and made up of five perpendicular bends (radius 0.5 [m]), five check valves ($\xi = 1$), and two sliding shut-off valves.

Determine the pressure that the pump should deliver (Figure 1.20).

At first, we determine the diameter D of the pipelines. We start by assuming a maximum water velocity of 1.5 [m/s]:

$$Q_V = \frac{\pi \cdot D^2}{4} \text{ from what: } D = 0.29 \text{ [m]}$$

We choose $D = 30$ [cm]. Backwards calculation of the diameter with this velocity leads to $c = 1.415$ [m/s].

From the various losses the ϕ values are calculated separately:

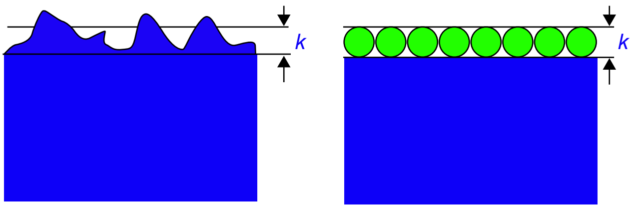


Figure 1.19 Absolute roughness k.

Table 1.3 Absolute roughness.

Sort	State	k (mm)
Drawn tubes in glass, lead, copper, brass		0.0015
PVC, polyethylene		0.05
Drawn steel tube	New	0.05
	Moderate rusted	0.4
	Heavy rusted	3
Welded steel tubes	New	0.05
	Moderate rusted	0.3
	Heavy rusted	4
Galvanized tubes		0.15
Cast iron	New	0.25
	Moderate rusted	1
	Heavy rusted	4
Concrete tubes	Smooth	0.3
	Rough	3
Wood	Scraped	0.18
	Not scraped	0.9

Table 1.4 Aging factor.

Year	2	5	10	20	30	40	50
Factor	1.1	1.2	1.35	1.75	2.10	2.60	3

The straight pipeline:

$$\xi_1 = \lambda \cdot \frac{L}{D} \text{ with } \lambda \cong 0.03$$

$$\xi_1 = 0.03 \cdot \frac{156}{0.3} = 1.6$$

The "obstacles"

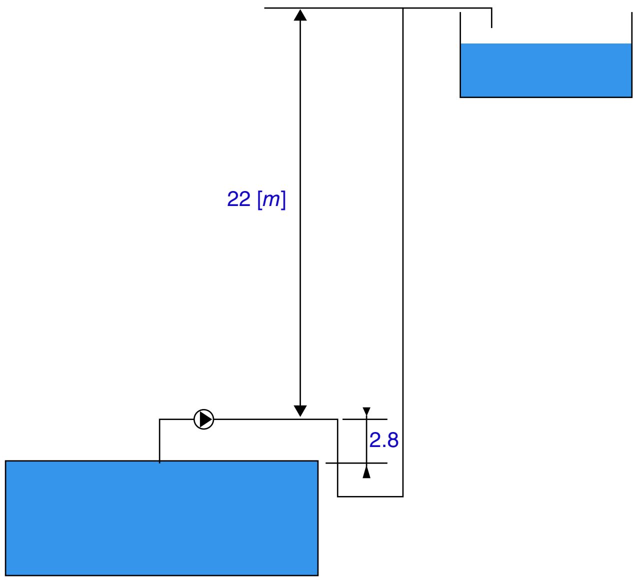


Figure 1.20 Example.

- Six bends with radius $r = 0.5$ [m], from which:

  $$\frac{D}{r} = \frac{0.3}{0.5} = 0.6$$

  According to the table: $\xi_2 = 6 \cdot 0.18 = 1.08$
- Two sliding vane valves:

  $$\xi_3 = 2 \cdot 0.8 = 1.6$$

- Suction strainer with check valve:

  $$\xi_4 = 4$$

- Five check valves:

  $$\xi_5 = 5$$

The total pressure loss p_f caused by friction amounts to:

$$p_f = (15.6 + 1.08 + 1.6 + 4 + 5) \cdot \frac{10^3 \cdot 1.415^2}{2} = 27\,310 \text{ [Pa]}$$

Furthermore, a pressure p_{geo} is needed to overcome the geodetic height:

$$p_{geo} = \rho \cdot g \cdot H_{geo} = 10\,00 \cdot 9.81 \cdot 24.8 = 243\,288 \text{ [Pa]}$$

Finally, a dynamic pressure p_{dyn} is needed to give the water kinetic energy:

$$p_{dyn} = \rho \cdot \frac{c^2}{2} = 1001 \text{ [Pa]}$$

So, the pump must deliver a pressure p of:

$$p = p_f + p_{geo} + p_{dyn} = 271\,599 \text{ [Pa]} = 2.72 \text{ [bar]}$$

We find out that the dynamic pressure is so small it can be neglected.