

# **Pumps and Compressors**

Marc Borremans







Pumps and Compressors

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# **Pumps and Compressors**

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# Contents

Preface xv Acknowledgment xvii Used Symbols xix About the Companion Website xxv v

Part I Pumps 1

# **1 General Concepts** *3*

- 1.1 Hydrostatics 3
- 1.2 Flow 4
- 1.3 Law of Bernoulli 5
- 1.4 Static and Dynamic Pressure 5
- 1.5 Viscosity 6
- 1.6 Extension of Bernoulli's Law 11
- 1.7 Laminar and Turbulent Flow 12
- 1.8 Laminar Flow 13
- 1.8.1 Hydraulic Resistance 13
- 1.8.2 Hydraulic Diameter 14
- 1.9 Turbulent Flow 17
- 1.10 Moody's Diagram 20
- 1.11 Feed Pressure 24
- 1.11.1 Geodetic Feed Pressure 24
- 1.11.2 Static Feed Pressure 24
- 1.11.3 Manometric Feed Pressure 26
- 1.11.4 Theoretic Feed Pressure 27
- 1.12 Law of Bernoulli in Moving Reference Frames 27
- 1.13 Water Hammer (Hydraulic Shock) 28
- 1.14 Flow Mechanics 30
- 1.14.1 Hydrofoils 30
- 1.14.2 Applications 33

vi Contents

**Positive Displacement Pumps** 35 2 Reciprocating Pumps 35 2.1 2.1.1 Operation 35 2.1.2 Flow 35 213 Valves 37 2.1.4 Piston Sealing 37 2.1.5 Plunger Pumps 37 2.1.6 Hand Pump 40 2.1.7Double Acting Pump 41 2.1.8 Membrane Pumps 42 2.1.9 Triplex Pumps 45 2.1.10Hydrophore 45 2.2 Maximum Suction Head 48 2.2.1Theoretical 48 2.2.2 Vapor Pressure 49 2.2.3 Velocity 50 2.2.4 Barometer 51 2.2.5Friction 51 Acceleration 52 2.2.6 2.2.6.1 Kinematics 52 2.2.6.2 Dynamics 53 2.2.7 Air Chambers 56 2.2.7.1 Suction Side 56 2.2.7.2 Press Side 57 2.3 Characteristic Values 59 2.3.1Manometric Feed Pressure 59 2.3.2 Theoretical Pressure 59 2.3.3 Power and Efficiency 60 2.3.4 Example 62 2.3.5Characteristic Curve of the Pump 64 2.3.5.1 Characteristic of the Pipe Line 64 Characteristic of the Pump 64 2.3.5.2 2.3.5.3 Regulation 65 2.3.6 Conclusion 67 2.4 Hydraulic Pumps 69 2.4.1Introduction 69 2.4.2Sliding Vane Pump 69 2.4.3 Gear Pumps 71 2.4.3.1External Toothing 71 2.4.3.2 73 Internal Toothing 2.4.4Screw Pumps 76 2.4.5Radial Plunger Pumps 78 2.4.6 Axial Plunger Pumps 81 2.5 Other Displacement Pumps 85 2.5.1Lobe Pump 85 2.5.2 Peristaltic Pump 88 2.5.2.1Properties 89

Contents vii

- 2.5.2.2 Applications 91
- 2.5.3 Mono Pump 91
- 2.5.4 Flex Impeller Pump 95
- 2.5.5 Side Channel Pump 97
- **3 Dynamic Pumps** *103*
- 3.1 Radial Turbopumps (Centrifugal Pumps) 103
- 3.1.1 General 103
- 3.1.2 Impeller Forms 103
- 3.1.2.1 Closed Impeller 103
- 3.1.2.2 Half-Open Impeller 104
- 3.1.2.3 Open Impeller 108
- 3.1.3 Velocity Triangles 108
- 3.1.4 Flow 109
- 3.1.4.1 Definition 109
- 3.1.4.2 Flow Determining Component of the Velocity 110
- 3.1.4.3 The Relative Flow 110
- 3.1.5 Static Pressure in a Closed Pump 112
- 3.1.6 Theoretical Feed Pressure 114
- 3.1.6.1 Law of Bernoulli in Rotating Frame 114
- 3.1.6.2 Discussion 114
- 3.1.6.3 Theoretical Feed Pressure 115
- 3.1.7 Diffusor 119
- 3.1.8 Influence of Vane Angle 121
- 3.1.8.1 Graphically 121
- 3.1.8.2 Analytically 122
- 3.1.9 Pump Curve 122
- 3.1.9.1 System Curve 122
- 3.1.9.2 Build Up Pump Curve 123
- 3.1.9.3 Operating Point 124
- 3.1.10 Pump Efficiency 124
- 3.1.11 Influence RPM 125
- 3.1.12 First Set of Affinity Laws 125
- 3.1.13 Second Set of Affinity Laws 127
- 3.1.14 Surge 127
- 3.1.15 Application Field 128
- 3.1.16 Flow Regulation 130
- 3.1.16.1 Throttle Regulation 130
- 3.1.16.2 Bypass Regulation 133
- 3.1.16.3 Speed Regulation 134
- 3.1.16.4 Comparison 134
- 3.1.17 Start Up of the Pump 135
- 3.1.18 High Pressure Pumps 139
- 3.1.19 Roto-jet Pump 139
- 3.1.20 Vortex Pumps 142
- 3.2 Axial Turbopumps 144
- 3.2.1 Operation 144

viii Contents

Volumetric Flow 144 3.2.2 3.2.2.1 Axial Velocity v 144 3.2.2.2 Perpendicular Surface A'148 3.2.3 Theoretical Feed Pressure 149 3.2.4 Diffusors 152 3.2.5 Vane Profile 152 3.2.6 Half-Axial Turbopumps 155 3.2.6.1 Motivation 155 3.2.6.2 Francis Vane Pump 155 3.2.6.3 Mixed Flow Pump 156 3.2.6.4 Characteristics of Turbopumps 158 3.2.7 Archimedes Screw 159 3.3 Turbopumps Advanced 161 3.3.1 1st Number of Rateau 161 3.3.2 2nd Number of Rateau 163 Homologous Series 164 3.3.3 3.3.4 Optimal Homologous Series 167 3.3.5 Rateau Numbers with Axial Pumps 168 3.3.6 The Specific Speed 168 Cavitation 172 3.3.7 3.3.8 NPSH 173 3.3.9 NPSH Characteristics 175 3.3.10 Counteracting Cavitation 175 3.3.11 Inducers 176 Double Sided Entry 180 3.3.12 3.3.13 Characteristics of Pumps 180 Suction Specific Speed 181 3.3.14 3.3.15 Series Connection 183 Parallel Connection 184 3.3.16 3.3.16.1 Simple Case 184 3.3.16.2 Case with Increasing Pump Curve 184 3.3.17 Influence Viscosity 187 3.3.18 Special Turbopumps 190 3.3.18.1 Submersible Pumps 190 3.3.18.2 Electropumps 192 3.3.19 Contaminated Liquids 193 3.3.20 Cutter Pumps 194 3.3.21 Mounting 195 4 Flow-Driven Pumps 205 4.1 General 205 4.2 Liquid Jet Liquid Pump 206 4.3 Liquid Jet Solid Pump 208 4.4Liquid Jet Mixers 209 4.5 Steam Jet Liquid Pump 209 4.6 The Feedback Pump 209

4.7 Air Pressure Pump 211

- **5 Sealing** *213*
- 5.1 Labyrinth Sealing 213
- 5.2 Lip Seals 217
- 5.3 V-Ring Seals 220
- 5.4 Gland Packing 222
- 5.5 Lantern Rings 226
- 5.6 Mechanical Seals 228
- 5.6.1 Fundamentals 228
- 5.6.2 Unbalanced Seals 231
- 5.6.3 Balanced Seals 233
- 5.6.4 The Configurations 235
- 5.6.4.1 Single Internal Seal 235
- 5.6.4.2 Single External Seal 236
- 5.6.4.3 Back-to-back Double Seal 236
- 5.6.4.4 Tandem Double Seal (Face-to-back Seal) 237
- 5.6.4.5 Dual Seal 239
- 5.6.4.6 Face-to-face Seal 239
- 5.6.5 Calculation of Liquid Flow 240
- 5.7 Hydrodynamic Seal 241
- 5.7.1 Hydrodynamic Seal with Back Vanes 241
- 5.7.2 Journal Bearing 242
- 5.7.3 Hydrodynamic Effect Converging Gap 243
- 5.7.4 Journal Bearing Lift Force 247
- 5.7.5 Hydrodynamic Mechanical Seals 248
- 5.8 Floating Ring Seals 250
- 5.9 Hermetic Pumps 252
- 5.9.1 Magnetic Coupling 252
- 5.9.2 Canned Motor Pump 255

Part II Compressors 257

**6 General** 259

- 6.1 Terminology 259
- 6.2 Normal Volume 259
- 6.3 Ideal Gasses 260
- 6.4 Work and Power 261
- 6.4.1 Compression Work 261
- 6.4.2 Technical Work 262
- 6.4.3 Technical Power 264
- 6.5 Nozzles 264
- 6.6 Flow 266
- 6.7 Choice and Selection 269
- 6.8 Psychrometrics 270
- 6.8.1 Partial Pressure 270
- 6.8.2 Equivalent Molar Mass 272
- 6.8.3 Moist Air 272

**x** Contents

- 6.8.4 Water Content 273
- 6.8.5 Saturated and Unsaturated Air (with Water) 273
- 6.8.6 Relation Between x and  $p_W$  274

### 7 Piston Compressors 275

- 7.1 Indicator Diagram 275
- 7.2 Parts 276
- 7.2.1 Cylinders 276
- 7.2.2 Sealing 276
- 7.2.3 Valves 277
- 7.3 Volumetric Efficiency 280
- 7.4 Membrane Compressor 286
- 7.5 Work and Power 286
- 7.5.1 Technical Work 286
- 7.5.2 Isothermal Compression 289
- 7.5.3 Polytropic Compression 290
- 7.5.4 Conclusions 291
- 7.5.5 Efficiency of a Piston Compressor 292
- 7.6 Two-stage Compressor 294
- 7.6.1 Motivation 294
- 7.6.2 Two Stages 295
- 7.6.2.1 General 295
- 7.6.2.2 Indicator Diagram 297
- 7.6.2.3 Intermediate Pressure 297
- 7.6.2.4 Work Per Stage 299
- 7.6.2.5 Compression Temperatures 299
- 7.6.2.6 Volumetric Efficiency 300
- 7.6.2.7 Cylinder Dimensions 300
- 7.6.2.8 Mounting 301
- 7.7 Three or More Stages 301
- 7.8 Problems with Water Condensation 301
- 7.9 Flow Regulation 305
- 7.9.1 Continuous Speed Regulation 305
- 7.9.2 Throttling Suction Line 305
- 7.9.3 Keeping Suction Valve Open 306
- 7.9.4 Dead Volume 306
- 7.10 Star Triangle Connection 308
- 7.10.1 Speed Regulation with VFD 311
- 7.11 Refrigeration Piston Compressor *314*
- 8 Other Displacement Compressors 317
- 8.1 Roots Compressor 317
- 8.1.1 Operation *317*
- 8.1.2 Technical Work 317
- 8.1.3 Properties 319
- 8.2 Vane Compressor 321
- 8.2.1 Operation 321

Contents xi

8.2.2 Properties 325 8.3 Screw Compressor 326 8.3.1 Operation 326 8.3.2 Properties 330 8.3.3 Regulation 332 Refrigerant Compressors 8.3.4 334 8.4 Mono-screw Compressor 334 8.4.1 Operation 334 8.4.2 Properties 335 Regulation 338 8.4.3 8.5 Scroll Compressor 341 8.6 Tooth Rotor Compressor 342 8.7 Rolling Piston 342 8.7.1 Operation "Rotary" 342 Swing Compressor 344 8.7.2 Liquid Ring Compressor 348 8.8 8.8.1 Operation 348 8.8.2 Properties 349 8.9 Regulation Displacement Compressors 351 8.9.1 Blow Off 351 8.9.2 Bypass Regulation 352 Throttling the Suction Line 352 8.9.3 8.9.4 Start–Stop Regulation 352 8.9.5 Full Load–No Load Regulation 353 8.9.6 Speed Control with a Frequency Regulator 353 8.10 Refrigerant Compressors 353 9 Turbocompressors 355 9.1 Centrifugal Fans 355 9.1.1 General 355 9.1.2 Static and Dynamic Pressure 357 Types of Vanes 359 9.1.3 9.1.3.1 Forward-curved Vanes 359 Aerodynamical Vanes 359 9.1.3.2 9.1.3.3 Backward-curved Vanes 359 9.1.3.4 Radial Vanes 361 Radial Tip Vanes 361 9.1.3.5 9.1.4 Behavior of the Different Impeller Types 362 9.1.4.1 Backward-curved Vanes 362 9.1.4.2 Forward-curved Vanes 362 9.1.5 Study of the Characteristics 363 9.1.6 Selection of a Fan 365 9.2 Cross-stream Fans 370 9.3 Side Channel Fans 370 9.4 Turbo Fan 372 9.5 Centrifugal Compressor 374 9.6 Refrigerant Turbocompressor 375

- xii Contents
  - 9.7 Axial Fans 375
  - 9.7.1 General 375
  - 9.7.2 Reaction Degree Axial Fan 378
  - 9.7.2.1 Definition 378
  - 9.7.3 Contrarotating Axial Fans 386
  - 9.7.4 Variable Pitch Axial Fan 388
  - 9.8 Axial Compressor 390
  - 9.9 Calculation Example 393
  - 9.10 Surge Limit 396
  - 9.11 Choke Limit (Stonewall Point) 397
  - 9.11.1 Introducing Nozzles 397
  - 9.11.1.1 Calculation of the Discharge Speed 397
  - 9.11.1.2 Calculation of the Flow 398
  - 9.11.1.3 The Flow Function  $\psi$  399
  - 9.11.1.4 The Critical Pressure 400
  - 9.11.2 Behavior at Changing Counter Pressure 402
  - 9.12 Comparison Axial/Radial Compressor 404
  - 9.13 Regulation of Turbocompressors 406
  - 9.13.1 Rotation Speed 406
  - 9.13.2 Throttling 407
  - 9.13.3 Variable Guide Vanes 407
  - 9.13.3.1 Axial Compressor 407
  - 9.13.3.2 Centrifugal Compressor 409
  - 9.14 Efficiency of Turbocompressors 410
  - **10** Jet Ejectors 415
  - 10.1 Steam Ejector Compressor 415
  - 10.1.1 General 415
  - 10.1.2 Jet Pumps with Mixing Heat Exchangers 417
  - 10.1.3 Jet Pump with Three Surface Heat Exchangers 417
  - 10.2 Gas Jet Ejector 421
  - 10.3 Applications 422
  - 10.3.1 Application 1 422
  - 10.3.2 Application 2 423

### **11 Vacuum Pumps** *425*

- 11.1 Vacuum Areas 425
- 11.1.1 Kinetic Gas Theory 425
- 11.1.2 Formation Time 427
- 11.2 Measuring Devices 428
- 11.2.1 Introduction 428
- 11.2.2 Bourdon Measuring Devices 428
- 11.2.3 Pirani Devices 430
- 11.2.4 Thermocouple gauges 430
- 11.2.5 Capacity Membrane Gauge 431
- 11.2.6 Ionization Gauges 432
- 11.2.7 Cathode Gauges 432

Contents xiii

- 11.3 Types of Flow *433*
- 11.4 Rough Vacuum (1000–1 [mbar]) 435
- 11.4.1 Membrane Pumps 435
- 11.4.2 Steam Jet Vacuum Pumps 436
- 11.4.3 Liquid Vacuum Ejector Pump 439
- 11.4.4 Gas Jet Vacuum Pump 439
- 11.4.5 Centrifugal Vacuum Pumps 441
- 11.4.6 Liquid Ring Pumps 443
- 11.5 Medium Vacuum (1–10<sup>-3</sup> [mbar]) 444
- 11.5.1 Vane Pump 444
- 11.5.2 The Gas Ballast 445
- 11.5.3 Screw Vacuum Pumps 448
- 11.5.4 Scroll Vacuum Pump 449
- 11.5.5 Rolling Piston 449
- 11.5.6 Claw Pump 451
- 11.5.7 Roots Vacuum Pumps 451
- 11.6 High Vacuum  $(10^{-3}-10^{-7} \text{ [mbar]})$  455
- 11.6.1 Diffusion Pumps 456
- 11.6.2 Diffusion Ejector Pumps (Booster Pumps) 459
- 11.6.3 Turbomolecular Pump 459
- 11.7 Ultrahigh Vacuum (10<sup>-7</sup>–10<sup>-14</sup> [mbar]) 462
- 11.7.1 Sorption Pumps 462
- 11.7.2 Adsorption Pumps 462
- 11.7.3 Sublimation Pump 465
- 11.7.4 Ion Getter Pump 466
- A The Velocity Profile and Mean Velocity for a Laminar Flow 469
- **B** Calculation of  $\lambda$  for a Laminar Flow 473

Index 475

# 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!

xv

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 16kB 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.

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# xvi Preface

# 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
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Atlas Copco	Glynwed/Reinhütte
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xvii

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# **Used Symbols**

Symbol	Meaning	Unit
a	Acceleration	m/s <sup>2</sup>
a'	Acceleration in suction line	$m/s^2$
Α	Section piston	m <sup>2</sup>
Α	Cross section channel	m <sup>2</sup>
Α	surface	m <sup>2</sup>
$A^{'}$	Section suction line	$M^2$
$A^{'}$	Perpendicular surface	m <sup>2</sup>
$c_{\perp}$	Velocity perpendicular on surface	m/s
c <sub>ll</sub>	Velocity along surface	m/s
С	Absolute velocity at impeller	m/s
С	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 <sub>v</sub>	Specific heat at constant volume	J/kg.K
C <sub>u</sub>	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	Ν
D	Diameter impeller	m
$D_H$	Hydraulic diameter	m
F	Force	Ν
$F_{C}$	Centrifugal force	Ν
g	Gravity acceleration	m/s <sup>2</sup>
Н	Height	m
h	Depth impeller	m

xix

xx Used Symbols
-----------------

Symbol	Meaning	Unit
h	Specific enthalpy	J/kg
$H_{geo}$	Geodetic height	m
H <sub>man</sub>	Manometric height (head)	m
$H_{p}$	Geodetic press height	m
H <sub>s</sub>	Geodetic suction height	m
H <sub>s,max</sub>	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	Ν
L	Length pipe line	m
$L^{'}$	Fictive length suction line	m
т	Mass	kg
m <sup>*</sup>	Displaced mass per cylinder	kg
М	Molar mass	kg/kmol
п	Polytropic exponent	-
Ν	Speed	rmp
$N_s$	Specific speed	${ m m}^{3/4}  \cdot  { m s}^{-3/2}$
N <sub>ss</sub>	Suction specific speed	${ m m}^{3/4}  \cdot  { m s}^{-3/2}$
$N_q$	Dimionless specific speed	-
$N_{\omega}$	Dimensionless specific speed	-
NPSH <sub>a</sub>	Available net positive suction head	m
NPSH <sub>ss</sub>	Suction net positive suction head	-
NPSH <sub>r</sub>	Required net positive suction head	m
$O_{cd}$	Surface under curve cd	J
$O_{ab}$	Surface under curve ab	J
$p_{\nu}$	Vapor pressure	Pa
p	Static pressure	Pa
$p_a$	Atmospheric pressure	Pa
$p_{abs}$	Absolute pressure	Pa
$p_{e\!f\!f}$	Effective pressure	Ра
$p_{dyn}$	Dynamic pressure	Ра
<i>p</i> <sub>man</sub>	Manometric (feed) pressure	Pa
$p_{geo}$	Geodetic (feed) pressure	Pa
$p_{\nu,p}$	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 <sub>tot,s</sub>	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 <sup>3</sup> /s
$Q_{\nu,a}$	Average volumetric flow	m <sup>3</sup> /s
$Q_{Veff}$	Effective volumetric flow	m <sup>3</sup> /s
Q <sub>n,nom</sub>	Nominal effective flow	m <sup>3</sup> /s
и	Peripheral speed	m/s
U	Voltage	V
R	Universal gas constant	J/kmol.K
<i>R</i> , <i>r</i>	Crankstroke, radius	m
S	Stroke length	m
Т	Absolute temperature	Κ
и	Circumpheral velocity impeller	m/s
ν	Specific volume	m3/kg
V	Total volume	m3
w	Relative velocity	m/s
w	Specific work	J/kg
w <sub>c</sub>	Specific compression work	J/kg
w <sub>t</sub>	Specific technical work	J/kg
W <sub>c</sub>	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	-

xxii Used Symbols

п	Polytropic exponent	-
δ	Correction factor laminar flow	-
ε	Factor dead volume	-
ρ	Specific mass	kg/m <sup>3</sup>
heta	Angle of incidence	° or rad
Δ	Difference	-
$\Delta pot$	Specific change of potential energy	J/kg
$\Delta kin$	Specific change of kinetic energy	J/kg
$\Delta H$	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	-
П	Product	-
$\Pi_1$	1 <sup>st</sup> number of Rateau	-
$\Pi_2$	2 <sup>nd</sup> 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
с	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

" 2nd stage		1st stage
	<i>и</i>	2nd stage

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The website includes:

- References
- Videos

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Part I

Pumps

# 1

# **General Concepts**

### **CHAPTER MENU**

Hydrostatics, 3 Flow, 4 Law of Bernoulli, 5 Static and Dynamic Pressure, 5 Viscosity, 6 Extension of Bernoulli's Law, 11 Laminar and Turbulent Flow, 11 Laminar Flow, 13 Turbulent Flow, 13 Turbulent Flow, 17 Moody's Diagram, 20 Feed Pressure, 24 Law of Bernoulli in Moving Reference Frames, 27 Water Hammer (Hydraulic Shock), 28 Flow Mechanics, 30

# 1.1 Hydrostatics

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

3

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

where

- *p*: the static pressure at the considered point [Pa = N/m<sup>2</sup>]
- $p_a$ : atmospheric pressure (ca. 1 [bar] = 10<sup>5</sup> [Pa])
- ρ: specific mass of the liquid [kg/m<sup>3</sup>]
- *g*: gravitational acceleration (9.81 [m<sup>2</sup>/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  $p_a$ :

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

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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<sub>a</sub>] for effective pressure ("g" stands for gauge) and [bar(a)] or [bar<sub>a</sub>] 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  $\rho$ . 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)  $\rho$  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.



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):

$$Q_{\nu} = c_1 \cdot A_1 = c_2 \cdot A_2$$
 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

 $c \ \rm [m/s]:$  the local velocity of the fluid

*z* [m]: the position on the vertical axis

 $\eta$  [Pa · s]: the dynamic viscosity (old unit, non-SI: 1 Poise = 0.1 [Pa · s])

The unit Poiseuille is a shortcut for 1 [Pa  $\cdot$  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  $\eta$  is independent of the pressure.



Figure 1.7 Channel.

### 8 1 General Concepts

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  $\tau$  (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.







- 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  $\nu$ :

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

The units are:  $[m^2/s]$ 

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

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

Table 1.1 Guide values for fluids.

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.



### **12** 1 General Concepts

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:

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

# 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:

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

where:

 $c_m$ : the mean velocity of the liquid [m/s] D: the diameter of the pipeline [m]  $\nu$ : the kinematic viscosity [m<sup>2</sup>/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:

$$c_m = \frac{2300 \cdot 1, 3 \cdot 10^{-6}}{0.1} = 0.03 \, [\text{m/s}]$$

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):

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

# 14 1 General Concepts

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  $\zeta$  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* 

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

Example 1.4 Isosceles triangle

0	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.



# **16** *1 General Concepts*

• 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:

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

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:

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

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

$$D_{H} = \frac{4 \cdot A_{w}}{P_{w}} = \frac{4 \cdot \pi \cdot (r_{out}^{2} - r_{in}^{2})}{2 \cdot \pi \cdot (r_{our} + r_{in})} = 2 \cdot (r_{out} - r_{in})$$

#### **Turbulent Flow** 1.9

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<sub>m</sub>*: 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  $\lambda$  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:  $\lambda = 0.316 \cdot \text{Re}^{-0.25}$ 

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:  $\lambda = 0.02 + \frac{0.0018}{\sqrt{c.D}}$ (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  $\xi$  also are extracted from tables, graphs, and empirical formulas. Turbulent flows are characterized by the independence of Re.

# **18** 1 General Concepts

 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



Example 1.12 *Plug valve* 



**Example 1.13** Suction strainer



# Example 1.14 Diffusor

When one looks at the value of  $\xi$  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  $\delta$ .

For a laminar flow the resistance factor  $\xi$  can be found by first calculating  $\xi$  for a turbulent flow and correcting with a conversion factor  $\delta$ :

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

where  $\delta$  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  $\epsilon$  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:  $\lambda = \varphi(Re, \epsilon)$  (Table 1.3).

The friction factor  $\lambda$  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  $Q_V = 360 \text{ [m}^3/\text{h]}$ . 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  $\phi$  values are calculated separately:



Figure 1.19 Absolute roughness k.

# Table 1.3 Absolute roughness.

**22** 1 General Concepts

Sort	State	<i>k</i> (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  $p_{\it geo}$  is needed to overcome the geodetic height:

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

Finally, a dynamic pressure  $p_{\mathit{dyn}}$  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:

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

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