

ANALYSIS, SYNTHESIS, AND DESIGN OF CHEMICAL PROCESSES

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Contents

Preface xxv

About the Authors xxix

List of Nomenclature xxxi

Chapter 0 Outcomes Assessment 1

0.1 Student Self-Assessment 2

0.2 Assessment by Faculty 4

0.3 Summary 6

References 6

SECTION I Conceptualization and Analysis of Chemical Processes 7

Chapter 1 Diagrams for Understanding Chemical Processes 9

1.1 Block Flow Diagram (BFD) 11

1.1.1 Block Flow Process Diagram 11

1.1.2 Block Flow Plant Diagram 12

1.2 Process Flow Diagram (PFD) 14

1.2.1 Process Topology 14

1.2.2 Stream Information 18

1.2.3 Equipment Information 21

1.2.4 Combining Topology, Stream Data, and Control Strategy to Give a PFD 21

1.3 Piping and Instrumentation Diagram (P&ID) 27

1.4 Additional Diagrams 32

1.5 Three-Dimensional Representation of a Process 34

1.6 The 3-D Plant Model 41

1.7 Operator and 3-D Immersive Training Simulators 43

1.7.1 Operator Training Simulators (OTS) 43

1.7.2 3-D Immersive Training Simulators (ITS) 45

1.7.3 Linking the ITS with an OTS 46

1.8 Summary 48

References 49

Short Answer Questions 49

Problems 50

Chapter 2 The Structure and Synthesis of Process Flow Diagrams 55

2.1 Hierarchy of Process Design 55

2.2 Step 1—Batch versus Continuous Process 56

2.3	Step 2—The Input/Output Structure of the Process	60
2.3.1	Process Concept Diagram	60
2.3.2	The Input/Output Structure of the Process Flow Diagram	61
2.3.3	The Input/Output Structure and Other Features of the Generic Block Flow Process Diagram	63
2.3.4	Other Considerations for the Input/Output Structure of the Process Flowsheet	65
2.3.5	What Information Can Be Determined Using the Input/Output Diagram for a Process?	68
2.4	Step 3—The Recycle Structure of the Process	70
2.4.1	Efficiency of Raw Material Usage	70
2.4.2	Identification and Definition of the Recycle Structure of the Process	71
2.4.3	Other Issues Affecting the Recycle Structure That Lead to Process Alternatives	75
2.5	Step 4—General Structure of the Separation System	83
2.6	Step 5—Heat-Exchanger Network or Process Energy Recovery System	83
2.7	Information Required and Sources	83
2.8	Summary	83
	References	85
	Short Answer Questions	86
	Problems	86
Chapter 3	Batch Processing	91
3.1	Design Calculations for Batch Processes	91
3.2	Gantt Charts and Scheduling	97
3.3	Nonoverlapping Operations, Overlapping Operations, and Cycle Times	98
3.4	Flowshop and Jobshop Plants	101
3.4.1	Flowshop Plants	101
3.4.2	Jobshop Plants	103
3.5	Product and Intermediate Storage and Parallel Process Units	106
3.5.1	Product Storage for Single-Product Campaigns	106
3.5.2	Intermediate Storage	108
3.5.3	Parallel Process Units	110
3.6	Design of Equipment for Multiproduct Batch Processes	111
3.7	Summary	113
	References	114
	Short Answer Questions	114
	Problems	114
Chapter 4	Chemical Product Design	123
4.1	Strategies for Chemical Product Design	124
4.2	Needs	125
4.3	Ideas	127
4.4	Selection	128
4.5	Manufacture	130
4.6	Batch Processing	131
4.7	Economic Considerations	131
4.8	Summary	132
	References	132

Chapter 5	Tracing Chemicals through the Process Flow Diagram	135
5.1	Guidelines and Tactics for Tracing Chemicals	135
5.2	Tracing Primary Paths Taken by Chemicals in a Chemical Process	136
5.3	Recycle and Bypass Streams	142
5.4	Tracing Nonreacting Chemicals	145
5.5	Limitations	145
5.6	Written Process Description	146
5.7	Summary	147
	Problems	147
Chapter 6	Understanding Process Conditions	149
6.1	Conditions of Special Concern for the Operation of Separation and Reactor Systems	150
6.1.1	Pressure	150
6.1.2	Temperature	150
6.2	Reasons for Operating at Conditions of Special Concern	152
6.3	Conditions of Special Concern for the Operation of Other Equipment	155
6.4	Analysis of Important Process Conditions	158
6.4.1	Evaluation of Reactor R-101	158
6.4.2	Evaluation of High-Pressure Phase Separator V-102	164
6.4.3	Evaluation of Large Temperature Driving Force in Exchanger E-101	164
6.4.4	Evaluation of Exchanger E-102	164
6.4.5	Pressure Control Valve on Stream 8	164
6.4.6	Pressure Control Valve on Stream from V-102 to V-103	164
6.5	Summary	165
	References	165
	Short Answer Questions	165
	Problems	166
SECTION II	Engineering Economic Analysis of Chemical Processes	169
Chapter 7	Estimation of Capital Costs	171
7.1	Classifications of Capital Cost Estimates	172
7.2	Estimation of Purchased Equipment Costs	175
7.2.1	Effect of Capacity on Purchased Equipment Cost	175
7.2.2	Effect of Time on Purchased Equipment Cost	179
7.3	Estimating the Total Capital Cost of a Plant	182
7.3.1	Lang Factor Technique	184
7.3.2	Module Costing Technique	185
7.3.3	Bare Module Cost for Equipment at Base Conditions	186
7.3.4	Bare Module Cost for Non-Base-Case Conditions	189
7.3.5	Combination of Pressure and MOC Information to Give the Bare Module Factor, F_{BM} and Bare Module Cost, C_{BM}	199
7.3.6	Algorithm for Calculating Bare Module Costs	200
7.3.7	Grassroots (Green Field) and Total Module Costs	201
7.3.8	A Computer Program (CAPCOST) for Capital Cost Estimation Using the Equipment Module Approach	204
7.4	Estimation of Plant Costs Based on Capacity Information	206

7.5	Summary	208
	References	208
	Short Answer Questions	209
	Problems	210
Chapter 8	Estimation of Manufacturing Costs	213
8.1	Factors Affecting the Cost of Manufacturing a Chemical Product	213
8.2	Cost of Operating Labor	218
8.3	Utility Costs	219
	8.3.1 Background Information on Utilities	219
	8.3.2 Calculation of Utility Costs	221
8.4	Raw Material Costs	234
8.5	Yearly Costs and Stream Factors	237
8.6	Estimating Utility Costs from the PFD	238
8.7	Cost of Treating Liquid and Solid Waste Streams	240
8.8	Evaluation of Cost of Manufacture for the Production of Benzene via the Hydrodealkylation of Toluene	241
8.9	Summary	242
	References	243
	Short Answer Questions	243
	Problems	244
Chapter 9	Engineering Economic Analysis	247
9.1	Investments and the Time Value of Money	248
9.2	Different Types of Interest	251
	9.2.1 Simple Interest	252
	9.2.2 Compound Interest	252
	9.2.3 Interest Rates Changing with Time	253
9.3	Time Basis for Compound Interest Calculations	254
	9.3.1 Effective Annual Interest Rate	254
	9.3.2 Continuously Compounded Interest	255
9.4	Cash Flow Diagrams	255
	9.4.1 Discrete Cash Flow Diagram	256
	9.4.2 Cumulative Cash Flow Diagram	258
9.5	Calculations from Cash Flow Diagrams	259
	9.5.1 Annuities—A Uniform Series of Cash Transactions	260
	9.5.2 Discount Factors	261
9.6	Inflation	266
9.7	Depreciation of Capital Investment	268
	9.7.1 Fixed Capital, Working Capital, and Land	269
	9.7.2 Different Types of Depreciation	269
	9.7.3 Current Depreciation Method (2017): Modified Accelerated Cost Recovery System (MACRS)	273
9.8	Taxation, Cash Flow, and Profit	274
9.9	Summary	277
	References	277
	Short Answer Questions	278
	Problems	278

Chapter 10	Profitability Analysis	285
10.1	A Typical Cash Flow Diagram for a New Project	285
10.2	Profitability Criteria for Project Evaluation	287
10.2.1	<i>Nondiscounted Profitability Criteria</i>	287
10.2.2	<i>Discounted Profitability Criteria</i>	291
10.3	Comparing Several Large Projects: Incremental Economic Analysis	295
10.4	Establishing Acceptable Returns from Investments: The Concept of Risk	298
10.5	Evaluation of Equipment Alternatives	299
10.5.1	<i>Equipment with the Same Expected Operating Lives</i>	299
10.5.2	<i>Equipment with Different Expected Operating Lives</i>	300
10.6	Incremental Analysis for Retrofitting Facilities	305
10.6.1	<i>Nondiscounted Methods for Incremental Analysis</i>	305
10.6.2	<i>Discounted Methods for Incremental Analysis</i>	308
10.7	Evaluation of Risk in Evaluating Profitability	309
10.7.1	<i>Forecasting Uncertainty in Chemical Processes</i>	310
10.7.2	<i>Quantifying Risk</i>	314
10.8	Profit Margin Analysis	325
10.9	Summary	326
	References	327
	Short Answer Questions	327
	Problems	328
SECTION III	Synthesis and Optimization of Chemical Processes	343
Chapter 11	Utilizing Experience-Based Principles to Confirm the Suitability of a Process Design	347
11.1	The Role of Experience in the Design Process	348
11.1.1	<i>Introduction to Technical Heuristics and Shortcut Methods</i>	348
11.1.2	<i>Maximizing the Benefits Obtained from Experience</i>	349
11.2	Presentation of Tables of Technical Heuristics and Guidelines	351
11.3	Summary	354
	List of Informational Tables	354
	References	368
	Problems	368
Chapter 12	Synthesis of the PFD from the Generic BFD	369
12.1	Information Needs and Sources	370
12.1.1	<i>Interactions with Other Engineers and Scientists</i>	370
12.1.2	<i>Reaction Kinetics Data</i>	370
12.1.3	<i>Physical Property Data</i>	371
12.2	Reactor Section	372
12.3	Separator Section	373
12.3.1	<i>General Guidelines for Choosing Separation Operations</i>	374
12.3.2	<i>Sequencing of Distillation Columns for Simple Distillation</i>	376
12.3.3	<i>Azeotropic Distillation</i>	378

12.4	Reactor Feed Preparation and Separator Feed Preparation Sections	388
12.5	Recycle Section	389
12.6	Environmental Control Section	389
12.7	Major Process Control Loops	390
12.8	Flow Summary Table	390
12.9	Major Equipment Summary Table	390
12.10	Summary	391
	References	391
	General Reference	392
	Problems	392

Chapter 13 Synthesis of a Process Using a Simulator and Simulator Troubleshooting 397

13.1	The Structure of a Process Simulator	398
13.2	Information Required to Complete a Process Simulation: Input Data	401
	13.2.1 Selection of Chemical Components	401
	13.2.2 Selection of Physical Property Models	401
	13.2.3 Selection and Input of Flowsheet Topology	404
	13.2.4 Selection of Feed Stream Properties	404
	13.2.5 Selection of Equipment Parameters	405
	13.2.6 Selection of Output Display Options	411
	13.2.7 Selection of Convergence Criteria and Running a Simulation	411
	13.2.8 Common Errors in Using Simulators	412
13.3	Handling Recycle Streams	413
13.4	Choosing Thermodynamic Models	415
	13.4.1 Pure-Component Properties	416
	13.4.2 Enthalpy	416
	13.4.3 Phase Equilibria	416
	13.4.4 Using Thermodynamic Models	424
13.5	Case Study: Toluene Hydrodealkylation Process	426
13.6	Electrolyte Systems Modeling	428
	13.6.1 Fundamentals of Modeling Electrolyte Systems	429
	13.6.2 Steps Needed to Build the Model of an Aqueous Electrolyte System and the Estimation of Parameters	435
13.7	Solids Modeling	440
	13.7.1 Physical Properties	440
	13.7.2 Parameter Requirements for Solids Model	442
Appendix 13.1	445	
	Calculation of Excess Gibbs Energy for Electrolyte Systems	445
Appendix 13.2	447	
	Steps to Build a Model of a Distillation Column for an Electrolyte System Using a Rate-Based Simulation with a Film Model for Mass Transfer, the Parameters Required at Each Stage, and Possible Sources of These Parameters	447
13.8	Summary	450
	References	451
	Short Answer Questions	454
	Problems	455

Chapter 14	Process Optimization	463
14.1	Background Information on Optimization	463
14.1.1	Common Misconceptions	465
14.1.2	Estimating Problem Difficulty	467
14.1.3	Top-Down and Bottom-Up Strategies	468
14.1.4	Communication of Optimization Results	468
14.2	Strategies	469
14.2.1	Base Case	469
14.2.2	Objective Functions	470
14.2.3	Analysis of the Base Costs	471
14.2.4	Identifying and Prioritizing Key Decision Variables	471
14.3	Topological Optimization	473
14.3.1	Introduction	473
14.3.2	Elimination of Unwanted Nonhazardous By-Products or Hazardous Waste Streams	473
14.3.3	Elimination and Rearrangement of Equipment	475
14.3.4	Alternative Separation Schemes and Reactor Configurations	477
14.4	Parametric Optimization	479
14.4.1	Single-Variable Optimization: A Case Study on T-201, the DME Separation Column	480
14.4.2	Two-Variable Optimization: The Effect of Pressure and Reflux Ratio on T-201, the DME Separation Column	481
14.4.3	Flowsheet Optimization Using Key Decision Variables	484
14.5	Lattice Search, Response Surface, and Mathematical Optimization Techniques	489
14.6	Process Flexibility and the Sensitivity of the Optimum	489
14.7	Optimization in Batch Systems	490
14.7.1	Problem of Scheduling Equipment	490
14.7.2	Problem of Optimum Cycle Time	495
14.8	Summary	497
	References	498
	Short Answer Questions	498
	Problems	498
Chapter 15	Pinch Technology	509
15.1	Introduction	509
15.2	Heat Integration and Network Design	510
15.3	Composite Temperature-Enthalpy Diagram	523
15.4	Composite Enthalpy Curves for Systems without a Pinch	524
15.5	Using the Composite Enthalpy Curve to Estimate Heat-Exchanger Surface Area	525
15.6	Effectiveness Factor (F) and the Number of Shells	529
15.7	Combining Costs to Give the EAO for the Network	534
15.8	Other Considerations	536
15.8.1	Materials of Construction and Operating Pressure Issues	536
15.8.2	Problems with Multiple Utilities	539
15.8.3	Handling Streams with Phase Changes	539

15.9	Heat-Exchanger Network Synthesis Analysis and Design (HENSAD) Program	540
15.10	Mass-Exchange Networks	541
15.11	Summary	550
	References	550
	Short Answer Questions	551
	Problems	552
Chapter 16	Advanced Topics Using Steady-State Simulators	561
16.1	Why the Need for Advanced Topics in Steady-State Simulation?	562
16.2	User-Added Models	562
16.2.1	Unit Operation Models	563
16.2.2	User Thermodynamic and Transport Models	564
16.2.3	User Kinetic Models	568
16.3	Solution Strategy for Steady-State Simulations	571
16.3.1	Sequential Modular (SM)	572
16.3.2	Equation-Oriented (EO)	585
16.3.3	Simultaneous Modular (SMod)	586
16.4	Studies with the Steady-State Simulation	589
16.4.1	Sensitivity Studies	589
16.4.2	Optimization Studies	589
16.5	Estimation of Physical Property Parameters	601
16.6	Summary	605
	References	605
	Short Answer Questions	607
	Problems	607
Chapter 17	Using Dynamic Simulators in Process Design	617
17.1	Why Is There a Need for Dynamic Simulation?	618
17.2	Setting Up a Dynamic Simulation	619
17.2.1	Step 1: Topological Change in the Steady-State Simulation	619
17.2.2	Step 2: Equipment Geometry and Size	622
17.2.3	Step 3: Additional Dynamic Data/Dynamic Specification	624
17.3	Dynamic Simulation Solution Methods	633
17.3.1	Initialization	634
17.3.2	Solution of the DAE System	634
17.4	Process Control	639
17.5	Summary	647
	References	647
	Short Answer Questions	648
	Problems	649
Chapter 18	Regulation and Control of Chemical Processes with Applications Using Commercial Software	655
18.1	A Simple Regulation Problem	656
18.2	The Characteristics of Regulating Valves	657
18.3	Regulating Flowrates and Pressures	660
18.4	The Measurement of Process Variables	662

- 18.5 Common Control Strategies Used in Chemical Processes 663
 - 18.5.1 Feedback Control and Regulation 663
 - 18.5.2 Feed-Forward Control and Regulation 665
 - 18.5.3 Combination Feedback and Feed-Forward Control 667
 - 18.5.4 Cascade Regulation 668
 - 18.5.5 Ratio Control 669
 - 18.5.6 Split-Range Control 671
- 18.6 Exchanging Heat and Work between Process and Utility Streams 674
 - 18.6.1 Increasing the Pressure of a Process Stream and Regulating Its Flowrate 674
 - 18.6.2 Exchanging Heat between Process Streams and Utilities 676
 - 18.6.3 Exchanging Heat between Process Streams 679
- 18.7 Logic Control 680
- 18.8 Advanced Process Control 682
 - 18.8.1 Statistical Process Control (SPC) 682
 - 18.8.2 Model-Based Control 683
- 18.9 Case Studies 683
 - 18.9.1 The Cumene Reactor, R-801 683
 - 18.9.2 A Basic Control System for a Binary Distillation Column 685
 - 18.9.3 A More Sophisticated Control System for a Binary Distillation Column 687
- 18.10 Putting It All Together: The Operator Training Simulator (OTS) 688
- 18.11 Summary 689
 - References 690
 - Problems 690

SECTION IV Chemical Equipment Design and Performance Process Equipment Design and Performance 695

Chapter 19 Process Fluid Mechanics 697

- 19.1 Basic Relationships in Fluid Mechanics 697
 - 19.1.1 Mass Balance 698
 - 19.1.2 Mechanical Energy Balance 700
 - 19.1.3 Force Balance 703
- 19.2 Fluid Flow Equipment 703
 - 19.2.1 Pipes 703
 - 19.2.2 Valves 705
 - 19.2.3 Pumps 706
 - 19.2.4 Compressors 707
- 19.3 Frictional Pipe Flow 709
 - 19.3.1 Calculating Frictional Losses 709
 - 19.3.2 Incompressible Flow 712
 - 19.3.3 Compressible Flow 719
 - 19.3.4 Choked Flow 720
- 19.4 Other Flow Situations 723
 - 19.4.1 Flow Past Submerged Objects 723
 - 19.4.2 Fluidized Beds 728
 - 19.4.3 Flowrate Measurement 730

19.5	Performance of Fluid Flow Equipment	736
19.5.1	Base-Case Ratios	736
19.5.2	Net Positive Suction Head	739
19.5.3	Pump and System Curves	743
19.5.4	Compressors	749
19.5.5	Performance of the Feed Section to a Process	752
	References	755
	Short Answer Questions	756
	Problems	757
Chapter 20	Process Heat Transfer	771
20.1	Basic Heat-Exchanger Relationships	771
20.1.1	Countercurrent Flow	771
20.1.2	Cocurrent Flow	773
20.1.3	Streams with Phase Changes	775
20.1.4	Nonlinear Q versus T Curves	776
20.1.5	Overall Heat Transfer Coefficient, U, Varies along the Exchanger	777
20.2	Heat-Exchange Equipment Design and Characteristics	779
20.2.1	Shell-and-Tube Heat Exchangers	779
20.3	LMTD Correction Factor for Multiple Shell and Tube Passes	789
20.3.1	Background	789
20.3.2	Basic Configuration of a Single-Shell-Pass, Double-Tube-Pass (1–2) Exchanger	790
20.3.3	Multiple Shell-and-Tube-Pass Exchangers	793
20.3.4	Cross-Flow Exchangers	797
20.3.5	LMTD Correction and Phase Change	797
20.4	Overall Heat Transfer Coefficients—Resistances in Series	798
20.5	Estimation of Individual Heat Transfer Coefficients and Fouling Resistances	800
20.5.1	Heat Transfer Resistances Due to Fouling	800
20.5.2	Thermal Conductivities of Common Metals and Tube Properties	800
20.5.3	Correlations for Film Heat Transfer Coefficients	803
20.6	Extended Surfaces	828
20.6.1	Rectangular Fin with Constant Thickness	829
20.6.2	Fin Efficiency for Other Fin Geometries	830
20.6.3	Total Heat Transfer Surface Effectiveness	831
20.7	Algorithm and Worked Examples for the Design of Heat Exchangers	837
20.7.1	Pressure Drop Considerations	837
20.7.2	Design Algorithm	838
20.8	Performance Problems	846
20.8.1	What Variables to Specify in Performance Problems	847
20.8.2	Using Ratios to Determine Heat-Exchanger Performance	847
20.8.3	Worked Examples for Performance Problems	850
	References	859
Appendix 20.A	Heat-Exchanger Effectiveness Charts	861
Appendix 20.B	Derivation of Fin Effectiveness for a Rectangular Fin	864
	Short Answer Questions	866
	Problems	866

Chapter 21	Separation Equipment	875
21.1	Basic Relationships in Separations	876
21.1.1	Mass Balances	876
21.1.2	Energy Balances	877
21.1.3	Equilibrium Relationships	877
21.1.4	Mass Transfer Relationships	878
21.1.5	Rate Expressions	882
21.2	Illustrative Diagrams	883
21.2.1	TP-xy Diagrams	883
21.2.2	McCabe-Thiele Diagram	888
21.2.3	Dilute Solutions—The Kremser and Colburn Methods	905
21.3	Equipment	911
21.3.1	Drums	911
21.3.2	Tray Towers	912
21.3.3	Packed Towers	926
21.3.4	Tray Tower or Packed Tower?	933
21.3.5	Performance of Packed and Tray Towers	933
	Case Study	934
21.4	Extraction Equipment	942
21.4.1	Mixer-Settlers	943
21.4.2	Static and Pulsed Columns	943
21.4.3	Agitated Columns	943
21.4.4	Centrifugal Extractors	943
21.5	Gas Permeation Membrane Separations	947
21.5.1	Equipment	947
21.5.2	Models for Gas Permeation Membranes	949
21.5.3	Practical Issues	950
	References	951
	Short Answer Questions	952
	Problems	954
Chapter 22	Reactors	961
22.1	Basic Relationships	962
22.1.1	Kinetics	962
22.1.2	Equilibrium	964
22.1.3	Additional Mass Transfer Effects	965
22.1.4	Mass Balances	970
22.1.5	Energy Balances	971
22.1.6	Reactor Models	972
22.2	Equipment Design for Nonisothermal Conditions	980
22.2.1	Nonisothermal Continuous Stirred Tank Reactor	980
22.2.2	Nonisothermal Plug Flow Reactor	984
22.2.3	Fluidized Bed Reactor	999
22.3	Performance Problems	1003
22.3.1	Ratios for Simple Cases	1003
22.3.2	More Complex Examples	1004
	References	1007
	Short Answer Questions	1007
	Problems	1008

Chapter 23 Other Equipment 1015

- 23.1 Pressure Vessels 1016
 - 23.1.1 Material Properties 1016
 - 23.1.2 Basic Design Equations 1016
- 23.2 Knockout Drums or Simple Phase Separators 1024
 - 23.2.1 Vapor-Liquid (V-L) Separation 1025
 - 23.2.2 Design of Vertical V-L Separators 1029
 - 23.2.3 Design of Horizontal V-L Separators 1032
 - 23.2.4 Mist Eliminators and Other Internals 1036
 - 23.2.5 Liquid-Liquid (L-L) Separation 1044
- 23.3 Steam Ejectors 1049
 - 23.3.1 Estimating Air Leaks into Vacuum Systems and the Load for Steam Ejectors 1050
 - 23.3.2 Single-Stage Steam Ejectors 1051
 - 23.3.3 Multistage Steam Ejectors 1054
 - 23.3.4 Performance of Steam Ejectors 1057
 - References 1058
 - Short Answer Questions 1059
 - Problems 1060

Chapter 24 Process Troubleshooting and Debottlenecking 1065

- 24.1 Recommended Methodology 1067
 - 24.1.1 Elements of Problem-Solving Strategies 1067
 - 24.1.2 Application to Troubleshooting Problems 1069
- 24.2 Troubleshooting Individual Units 1071
 - 24.2.1 Troubleshooting a Packed-Bed Absorber 1071
 - 24.2.2 Troubleshooting the Cumene Process Feed Section 1074
- 24.3 Troubleshooting Multiple Units 1076
 - 24.3.1 Troubleshooting Off-Specification Acrylic Acid Product 1076
 - 24.3.2 Troubleshooting Steam Release in Cumene Reactor 1078
- 24.4 A Process Troubleshooting Problem 1081
- 24.5 Debottlenecking Problems 1085
- 24.6 Summary 1091
 - References 1091
 - Problems 1091

SECTION V The Impact of Chemical Engineering Design on Society 1101**Chapter 25 Ethics and Professionalism 1103**

- 25.1 Ethics 1104
 - 25.1.1 Moral Autonomy 1105
 - 25.1.2 Rehearsal 1105
 - 25.1.3 Reflection in Action 1106
 - 25.1.4 Mobile Truth 1107
 - 25.1.5 Nonprofessional Responsibilities 1108
 - 25.1.6 Duties and Obligations 1110
 - 25.1.7 Codes of Ethics 1110

- 25.1.8 Whistle-Blowing [12] 1115
- 25.1.9 Ethical Dilemmas 1117
- 25.1.10 Additional Ethics Heuristics 1118
- 25.1.11 Other Resources 1118
- 25.2 Professional Registration 1121
 - 25.2.1 Engineer-in-Training 1122
 - 25.2.2 Registered Professional Engineer 1124
- 25.3 Legal Liability [13] 1125
- 25.4 Business Codes of Conduct [14, 15] 1126
- 25.5 Summary 1127
 - References 1128
 - Problems 1129

Chapter 26 Health, Safety, and the Environment 1131

- 26.1 Risk Assessment 1131
 - 26.1.1 Accident Statistics 1132
 - 26.1.2 Worst-Case Scenarios 1133
 - 26.1.3 The Role of the Chemical Engineer 1134
- 26.2 Regulations and Agencies 1134
 - 26.2.1 OSHA and NIOSH 1135
 - 26.2.2 Environmental Protection Agency (EPA) 1140
 - 26.2.3 Nongovernmental Organizations 1143
- 26.3 Fires and Explosions 1143
 - 26.3.1 Terminology 1143
 - 26.3.2 Pressure-Relief Systems 1145
- 26.4 Process Hazard Analysis 1145
 - 26.4.1 HAZOP (Hazard and Operability Study) 1146
 - 26.4.2 Dow Fire & Explosion Index and Chemical Exposure Index 1147
- 26.5 Chemical Safety and Hazard Investigation Board 1153
- 26.6 Inherently Safe Design 1153
- 26.7 Summary 1154
- 26.8 Glossary 1154
 - References 1156
 - Problems 1157

Chapter 27 Green Engineering 1159

- 27.1 Environmental Regulations 1159
- 27.2 Environmental Fate of Chemicals 1160
- 27.3 Green Chemistry 1163
- 27.4 Pollution Prevention during Process Design 1164
- 27.5 Analysis of a PFD for Pollution Performance and Environmental Performance 1166
- 27.6 An Example of the Economics of Pollution Prevention 1167
- 27.7 Life Cycle Analysis 1168
- 27.8 Summary 1169
 - References 1170
 - Problems 1171

SECTION VI Interpersonal and Communication Skills 1173**Chapter 28 Teamwork 1175**

- 28.1 **Groups 1175**
 - 28.1.1 *Characteristics of Effective Groups* 1176
 - 28.1.2 *Assessing and Improving the Effectiveness of a Group* 1178
 - 28.1.3 *Organizational Behaviors and Strategies* 1180
- 28.2 **Group Evolution 1184**
 - 28.2.1 *Forming* 1184
 - 28.2.2 *Storming* 1184
 - 28.2.3 *Norming* 1185
 - 28.2.4 *Performing* 1186
- 28.3 **Teams and Teamwork 1186**
 - 28.3.1 *When Groups Become Teams* 1186
 - 28.3.2 *Unique Characteristics of Teams* 1187
- 28.4 **Misconceptions 1189**
 - 28.4.1 *Team Exams* 1189
 - 28.4.2 *Overreliance on Team Members* 1189
- 28.5 **Learning in Teams 1189**
- 28.6 **Other Reading 1190**
- 28.7 **Summary 1191**
 - References 1192
 - Problems 1192

Chapter 29 Written and Oral Communication 1195

- 29.1 **Audience Analysis 1196**
- 29.2 **Written Communication 1196**
 - 29.2.1 *Design Reports* 1197
 - 29.2.2 *Transmittal Letters or Memos* 1198
 - 29.2.3 *Executive Summaries and Abstracts* 1198
 - 29.2.4 *Other Types of Written Communication* 1199
 - 29.2.5 *Exhibits (Figures and Tables)* 1200
 - 29.2.6 *References* 1200
 - 29.2.7 *Strategies for Writing* 1201
 - 29.2.8 *WVU and Auburn University Guidelines for Written Design Reports* 1202
- 29.3 **Oral Communication 1209**
 - 29.3.1 *Formal Oral Presentations* 1210
 - 29.3.2 *Briefings* 1211
 - 29.3.3 *Visual Aids* 1211
 - 29.3.4 *WVU and Auburn University Oral Presentation Guidelines* 1212
- 29.4 **Software and Author Responsibility 1215**
 - 29.4.1 *Spell Checkers* 1215
 - 29.4.2 *Thesaurus* 1215
 - 29.4.3 *Grammar Checkers* 1215
 - 29.4.4 *Graphs* 1216
 - 29.4.5 *Tables* 1217
 - 29.4.6 *Colors and Exotic Features* 1217
 - 29.4.7 *Raw Output from Process Simulators* 1217

- 29.5 Summary 1218
- References 1218
- Problems 1219

Chapter 30 A Report-Writing Case Study 1221

- 30.1 The Assignment Memorandum 1221
- 30.2 Response Memorandum 1222
- 30.3 Visual Aids 1224
- 30.4 Example Reports 1230
 - 30.4.1 *An Example of a Portion of a Student Written Report* 1231
 - 30.4.2 *An Example of an Improved Student Written Report* 1233
- 30.5 Checklist of Common Mistakes and Errors 1244
 - 30.5.1 *Common Mistakes for Visual Aids* 1243
 - 30.5.2 *Common Mistakes for Written Text* 1244

Appendix A Cost Equations and Curves for the CAPCOST Program 1247

- A.1 Purchased Equipment Costs 1247
- A.2 Pressure Factors 1264
 - A.2.1 *Pressure Factors for Process Vessels* 1264
 - A.2.2 *Pressure Factors for Other Process Equipment* 1264
- A.3 Material Factors and Bare Module Factors 1267
 - A.3.1 *Bare Module and Material Factors for Heat Exchangers, Process Vessels, and Pumps* 1267
 - A.3.2 *Bare Module and Material Factors for the Remaining Process Equipment* 1271
- References 1275

Appendix B Information for the Preliminary Design of Fifteen Chemical Processes 1277

- B.1 Dimethyl Ether (DME) Production, Unit 200 1278
 - B.1.1 *Process Description* 1278
 - B.1.2 *Reaction Kinetics* 1282
 - B.1.3 *Simulation (CHEMCAD) Hints* 1283
 - B.1.4 *References* 1283
- B.2 Ethylbenzene Production, Unit 300 1283
 - B.2.1 *Process Description [1, 2]* 1284
 - B.2.2 *Reaction Kinetics* 1284
 - B.2.3 *Simulation (CHEMCAD) Hints* 1291
 - B.2.4 *References* 1291
- B.3 Styrene Production, Unit 400 1291
 - B.3.1 *Process Description [1, 2]* 1291
 - B.3.2 *Reaction Kinetics* 1292
 - B.3.3 *Simulation (CHEMCAD) Hints* 1299
 - B.3.4 *References* 1299
- B.4 Drying Oil Production, Unit 500 1299
 - B.4.1 *Process Description* 1300
 - B.4.2 *Reaction Kinetics* 1300
 - B.4.3 *Simulation (CHEMCAD) Hints* 1300
 - B.4.4 *Reference* 1305

- B.5 Production of Maleic Anhydride from Benzene, Unit 600 1305
 - B.5.1 Process Description 1305
 - B.5.2 Reaction Kinetics 1306
 - B.5.3 Simulation (CHEMCAD) Hints 1311
 - B.5.4 References 1311
- B.6 Ethylene Oxide Production, Unit 700 1311
 - B.6.1 Process Description [1, 2] 1311
 - B.6.2 Reaction Kinetics 1313
 - B.6.3 Simulation (CHEMCAD) Hints 1316
 - B.6.4 References 1317
- B.7 Formalin Production, Unit 800 1317
 - B.7.1 Process Description [1, 2] 1317
 - B.7.2 Reaction Kinetics 1319
 - B.7.3 Simulation (CHEMCAD) Hints 1319
 - B.7.4 References 1319
- B.8 Batch Production of L-Phenylalanine and L-Aspartic Acid, Unit 900 1323
 - B.8.1 Process Description 1323
 - B.8.2 Reaction Kinetics 1325
 - B.8.3 References 1329
- B.9 Acrylic Acid Production via The Catalytic Partial Oxidation of Propylene [1–5], Unit 1000 1329
 - B.9.1 Process Description 1330
 - B.9.2 Reaction Kinetics and Reactor Configuration 1331
 - B.9.3 Simulation (CHEMCAD) Hints 1337
 - B.9.4 References 1337
- B.10 Production of Acetone via the Dehydrogenation of Isopropyl Alcohol (IPA) [1–4], Unit 1100 1338
 - B.10.1 Process Description 1338
 - B.10.2 Reaction Kinetics 1338
 - B.10.3 Simulation (CHEMCAD) Hints 1344
 - B.10.4 References 1344
- B.11 Production of Heptenes from Propylene and Butenes [1], Unit 1200 1344
 - B.11.1 Process Description 1351
 - B.11.2 Reaction Kinetics 1351
 - B.11.3 Simulation (CHEMCAD) Hints 1352
 - B.11.4 Reference 1352
- B.12 Design of a Shift Reactor Unit to Convert CO to CO₂, Unit 1300 1352
 - B.12.1 Process Description 1352
 - B.12.2 Reaction Kinetics 1352
 - B.12.3 Simulation (Aspen Plus) Hints 1356
 - B.12.4 Reference 1356
- B.13 Design of a Dual-Stage Selexol Unit to Remove CO₂ and H₂S From Coal-Derived Synthesis Gas, Unit 1400 1356
 - B.13.1 Process Description 1356
 - B.13.2 Simulation (Aspen Plus) Hints 1358
 - B.13.3 References 1362

B.14	Design of a Claus Unit for the Conversion of H ₂ S to Elemental Sulfur, Unit 1500	1363
B.14.1	Process Description	1363
B.14.2	Reaction Kinetics	1369
B.14.3	Simulation (Aspen Plus) Hints	1370
B.14.4	References	1370
B.15	Modeling a Downward-Flow, Oxygen-Blown, Entrained-Flow Gasifier, Unit 1600	1371
B.15.1	Process Description	1371
B.15.2	Reaction Kinetics	1373
B.15.3	Simulation (Aspen Plus) Hints	1375
B.15.4	References	1377
Appendix C	Design Projects	1379
Project 1	Increasing the Production of 3-Chloro-1-Propene (Allyl Chloride) in Unit 600	1381
C.1.1	Background	1381
C.1.2	Process Description of the Beaumont Allyl Chloride Facility	1382
C.1.3	Specific Objectives of Assignment	1385
C.1.4	Additional Background Information	1386
C.1.5	Process Design Calculations	1388
	Fluidized-Bed Reactor, R-601	1388
	Reference	1393
Project 2	Design and Optimization of a New 20,000-Metric-Tons-per-Year Facility to Produce Allyl Chloride at La Nueva Cantina, Mexico	1394
C.2.1	Background	1394
C.2.2	Assignment	1394
C.2.3	Problem-Solving Methodology	1395
C.2.4	Process Information	1395
Project 3	Scale-Down of Phthalic Anhydride Production at TBWS Unit 700	1401
C.3.1	Background	1401
C.3.2	Phthalic Anhydride Production	1402
C.3.3	Other Information	1403
C.3.4	Assignment	1411
C.3.5	Report Format	1411
Project 4	The Design of a New 100,000-Metric-Tons-per-Year Phthalic Anhydride Production Facility	1412
C.4.1	Background	1412
C.4.2	Other Information	1412
C.4.3	Assignment	1416
C.4.4	Report Format	1416

Project 5	Problems at the Cumene Production Facility, Unit 800	1417
	C.5.1 Background	1417
	C.5.2 Cumene Production Reactions	1417
	C.5.3 Process Description	1417
	C.5.4 Recent Problems in Unit 800	1418
	C.5.5 Other Information	1420
	C.5.6 Assignment	1420
	C.5.7 Report Format	1420
	C.5.8 Process Calculations	1426
	<i>Calculations for Fuel Gas Exit Line for V-802</i>	1426
	<i>Calculations for P-801</i>	1427
	<i>Vapor Pressure of Stream 3</i>	1428
	<i>Calculations for P-802</i>	1429
Project 6	Design of a New, 100,000-Metric-Tons-per-Year Cumene Production Facility	1430
	C.6.1 Background	1430
	C.6.2 Assignment	1430
	C.6.3 Report Format	1432
<i>Index</i>		1433

Sample pages

Process Fluid Mechanics

WHAT YOU WILL LEARN

- The basic relationships for fluid flow—mass, energy, and force balances
- The primary types of fluid flow equipment—pipes, pumps, compressors, valves
- How to design a system for incompressible and compressible frictional flow of fluid in pipes
- How to design a system for frictional flow of fluid with submerged objects
- Methods for flow measurement
- How to analyze existing fluid flow equipment
- How to use the concept of net positive suction head (NPSH) to ensure safe and appropriate pump operation
- The analysis of pump and system curves
- How to use compressor curves and when to use compressor staging

The purpose of this chapter is to introduce the concepts needed to design piping systems, including pumps, compressors, turbines, valves, and other components, and to evaluate the performance of these systems once designed and implemented. The scope is limited to steady-state situations. Derivations are minimized, and the emphasis is on providing a set of useful, working equations that can be used to design and evaluate the performance of piping systems.

19.1 BASIC RELATIONSHIPS IN FLUID MECHANICS

In expressing the basic relationships for fluid flow, a general control volume is used, as illustrated in Figure 19.1. This control volume can be the fluid inside the pipes and equipment connected by the pipes, with the possibility of multiple inputs and multiple outputs. For the simple case of one input and one output, the subscript 1 refers to the upstream side and the subscript 2 refers to the downstream side.

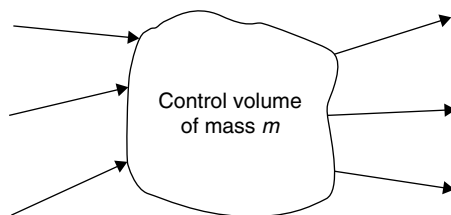


Figure 19.1 General Control Volume

19.1.1 Mass Balance

At steady state, mass is conserved, so the total mass flowrate (\dot{m} , mass/time) in must equal the total mass flowrate out. For a device with m inputs and n outputs, the appropriate relationship is given by Equation (19.1). For a single input and single output, Equation (19.2) is used.

$$\sum_{i=1}^m \dot{m}_{i,in} = \sum_{i=1}^n \dot{m}_{i,out} \quad (19.1)$$

$$\dot{m}_1 = \dot{m}_2 \quad (19.2)$$

In describing fluid flow, it is necessary to write the mass flowrate in terms of both volumetric flowrate (\dot{v} , volume/time) and velocity (u , length/time). These relationships are

$$\dot{m} = \rho \dot{v} = \rho A u \quad (19.3)$$

where ρ is the density (mass/volume) and A is the cross-sectional area for flow (length²). From Equation (19.3), for an incompressible fluid (constant density) at steady state, the volumetric flowrate is constant, and the velocity is constant for a constant cross-sectional area for flow. However, for a compressible fluid flowing with constant cross-sectional area, if the density changes, the volumetric flowrate and velocity both change in the opposite direction, since the mass flowrate is constant. Accordingly, if the density decreases, the volumetric flowrate and velocity both increase. For problems involving compressible flow, it is useful to define the superficial mass velocity, G (mass/area/time), as

$$G = \frac{\dot{m}}{A} = \rho u \quad (19.4)$$

The advantage of defining a superficial mass velocity is that it is constant for steady-state flow in a constant cross-sectional area, unlike density and velocity, and it shows that the product of density and velocity remains constant.

For a system with multiple inputs and/or multiple outputs at steady state, as is illustrated in Figure 19.2, the total mass flowrate into the system must equal the total mass flowrate out,

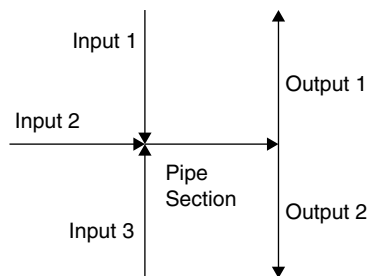


Figure 19.2 System with Multiple Inputs and Outputs

Equation (19.1). However, the output mass flowrate in each section differs depending on the size, length, and elevation of the piping involved. These problems are discussed later.

Example 19.1

Two streams of crude oil (specific gravity of 0.887) mix as shown in Figure E19.1. The volumetric flowrate of Stream 1 is $0.006 \text{ m}^3/\text{s}$, and its pipe diameter is 0.078 m . The volumetric flowrate of Stream 2 is $0.009 \text{ m}^3/\text{s}$, and its pipe diameter is 0.10 m .

- Determine the volumetric and mass flowrates of Stream 3.
- Determine the velocities in Streams 1 and 2.
- If the velocity is not to exceed 1 m/s in Stream 3, determine the minimum possible pipe diameter.
- Determine the superficial mass velocity Stream 3 using the pipe diameter calculated in Part (c).

Solution

- Since the density is constant, the volumetric flowrate of Stream 3 is the sum of the volumetric flowrates of Streams 1 and 2, $0.015 \text{ m}^3/\text{s}$. To obtain the mass flowrate, $\dot{m} = \rho \dot{v}$, so $\dot{m}_3 = (887 \text{ kg/m}^3)(0.015 \text{ m}^3/\text{s}) = 13.3 \text{ kg/s}$. Alternatively, the mass flowrate of Streams 1 and 2 could be calculated and added to get the same result.
- From Equation (19.3), at constant density $u = \dot{v}/A$. Therefore,

$$u_1 = \frac{\dot{v}_1}{A_1} = \frac{4\dot{v}_1}{\pi D_1^2} = \frac{4(0.006 \text{ m}^3/\text{s})}{\pi(0.078 \text{ m})^2} = 1.26 \text{ m/s} \quad (\text{E19.1a})$$

$$u_2 = \frac{\dot{v}_2}{A_2} = \frac{4\dot{v}_2}{\pi D_2^2} = \frac{4(0.009 \text{ m}^3/\text{s})}{\pi(0.10 \text{ m})^2} = 1.15 \text{ m/s} \quad (\text{E19.1b})$$

- The diameter at which $u_3 = 1 \text{ m/s}$ can be calculated from Equation (19.3) at constant density.

$$\dot{v}_3 = u_3 A_3 \Rightarrow 0.015 \text{ m}^3/\text{s} = (1 \text{ m/s}) \left(\frac{\pi D^2}{4} \right) \therefore D = 0.138 \text{ m} \quad (\text{E19.1c})$$

If the diameter were smaller, the cross-sectional area would be smaller, and from Equation (19.3), the velocity would be larger. Hence, the result in Equation (E19.1c) is the minimum possible diameter. As shown later, actual pipes are available only in discrete sizes, so it is necessary to use the next higher pipe diameter.

- From Equation (19.4), using the rounded values,

$$G_1 = \frac{\dot{m}_3}{A_3} = \frac{4\dot{m}_3}{\pi D_1^2} = \frac{4(13.3 \text{ kg/s})}{\pi(0.138 \text{ m})^2} = 889.2 \text{ kg/m}^2/\text{s} \quad (\text{E19.1d})$$

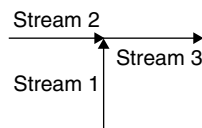


Figure E19.1 Physical Situation in Example 19.1

19.1.2 Mechanical Energy Balance

The mechanical energy balance represents the conversion between different forms of energy in piping systems. With the exception of temperature changes for a gas undergoing compression or expansion with no phase change, temperature is assumed to be constant. The mechanical energy balance is

$$\int_1^2 \frac{dP}{\rho} + \frac{1}{2} \Delta \left(\frac{\langle u^3 \rangle}{\langle u \rangle} \right) + g\Delta z + e_f - W_s = 0 \quad (19.5)$$

In Equation (19.5) and throughout this chapter, the difference, Δ , represents the value at Point 2 minus the value at Point 1, that is, out – in. The units in Equation (19.5) are energy/mass or length²/time². In SI units, since 1 J = 1 kg m²/s², it is clear that 1 J/kg = 1 m²/s². In American Engineering units, since 1 lb_f = 32.2 ft lb_m/sec², this conversion factor (often called g_c) must be used to reconcile the units. The notation $\langle \rangle$ represents the appropriate average quantity.

The first term in Equation (19.5) is the enthalpy of the system. On the basis of the constant temperature assumption, only pressure is involved. For incompressible fluids, such as liquids, density is constant, and the term reduces to

$$\int_1^2 \frac{dP}{\rho} = \frac{\Delta P}{\rho} \quad (19.6)$$

For compressible fluids, the integral must be evaluated using an equation of state.

The second term in Equation (19.5) is the kinetic energy term. For turbulent flow, a reasonable assumption is that

$$\frac{\langle u^3 \rangle}{\langle u \rangle} \approx \langle u \rangle^2 \quad (19.7)$$

For laminar flow,

$$\frac{\langle u^3 \rangle}{\langle u \rangle} \approx 2 \langle u \rangle^2 \quad (19.8)$$

For simplicity, $\langle u^2 \rangle$ is hereafter represented as $\langle u \rangle^2$, which is shortened to u^2 .

The third term in Equation (19.5) is the potential energy term. Based on the general control volume, Δz is positive if Point 2 is at a higher elevation than Point 1.

The fourth term in Equation (19.5) is often called the energy “loss” due to friction. Of course, energy is not lost—it is just expended to overcome friction, and it manifests as a change in temperature. The procedures for calculating frictional losses are discussed later.

The last term in Equation (19.5) represents the shaft work, that is, the work done on the system (fluid) by a pump or compressor or the work done by the system on a turbine. These devices are not 100% efficient. For example, more work must be applied to the pump than is transferred to the fluid, and less work is generated by the turbine than is expended by the fluid. In this book, work is defined as positive if done on the system (pump, compressor) and negative if done by the fluid (turbine). This convention is consistent with the flow of energy in or out of the system; however, many textbooks use the reverse sign convention. Equipment such as pumps, compressors, and turbines are described in terms of their power, where power is the rate of doing work. Therefore, a device power (\dot{W}_s , energy/time) is defined as the product of the mass flowrate (mass/time) and the shaft work (energy/mass):

$$\dot{W}_s = \dot{m}W_s \quad (19.9)$$

When efficiencies are included, the last term in Equation (19.5) becomes

$$\eta_p W_s = \frac{\eta_p \dot{W}_s}{\dot{m}} \text{ pump/compressor} \quad (19.10)$$

$$\frac{W_s}{\eta_t} = \frac{\dot{W}_s}{\eta_t \dot{m}} \text{ turbine} \quad (19.11)$$

Example 19.2

Water in an open (source or supply) tank is pumped to a second (destination) tank at a rate of 5 lb/sec with the water level in the destination tank 25 ft above the water level in the source tank, and it is assumed that the water level does not change with the flow of water. The destination tank is under a constant 30 psig pressure. The pump efficiency is 75%. Neglect friction.

- Determine the required horsepower of the pump.
- Determine the pressure increase provided by the pump assuming the suction and discharge lines have the same diameter.

Solution

- Turbulent flow in the pipes is assumed. The mechanical energy balance is

$$\frac{\Delta P}{\rho} + \frac{1}{2} \Delta u^2 + g \Delta z + e_f - \frac{\eta_p \dot{W}_s}{\dot{m}} = 0 \quad (\text{E19.2a})$$

Figure E19.2 is an illustration of the system.

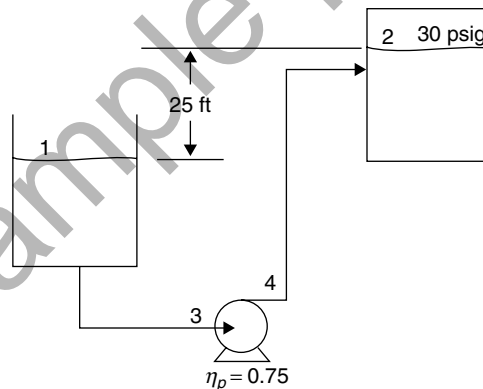


Figure E19.2 Physical Situation for Example 19.2

The control volume is the water in the tanks, the pipes, and the pump, and the locations of Points 1 and 2 are illustrated. The integral in the first term of Equation (19.2) is simplified to the first term in Equation (E19.2a), since the density of water is a constant. In general, the fluid velocity in tanks is assumed to be zero because tank diameters are much larger than pipe diameters, so the kinetic energy term for the liquid surface in the tank is essentially zero. Any fluid in contact with the atmosphere is at atmospheric pressure, so $P_1 = 1 \text{ atm} = 0 \text{ psig}$. The friction term is assumed to be zero in this problem, as stated. So, Equation (E19.2a) reduces to

$$\frac{P_2 - P_1}{\rho} + g(z_2 - z_1) - \frac{\eta_p \dot{W}_s}{\dot{m}} = 0 \quad (\text{E19.2b})$$

and

$$\frac{(30-0) \text{ lb}_f/\text{in}^2 (12 \text{ in}/\text{ft})^2}{62.4 \text{ lb}/\text{ft}^3} + \frac{32.2 \text{ ft}/\text{sec}^2}{32.2 \text{ ft lb}/\text{lb}_f/\text{sec}^2} (25-0) \text{ ft} - \frac{0.75 \dot{W}_s}{5 \text{ lb}/\text{sec}} = 0 \quad (\text{E19.2c})$$

so, $\dot{W}_s = 628.2 \text{ ft lb}_f/\text{sec}$.

Converting to horsepower yields

$$\dot{W}_s = \frac{628.2 \text{ ft lb}_f/\text{sec}}{550 \text{ ft lb}_f/\text{hp}/\text{sec}} = 1.14 \text{ hp} \quad (\text{E19.2d})$$

- b. To determine the pressure rise in the pump, the control volume is now taken as the fluid in the pump. So, the mechanical energy balance is written between Points 3 and 4. The mechanical energy balance reduces to

$$\frac{\Delta P}{\rho} - \frac{\eta_p \dot{W}_s}{\dot{m}} = 0 \quad (\text{E19.2e})$$

The kinetic energy term is zero because the suction and discharge pipes have the same diameter. Frictional losses are assumed to be zero in this example. The potential energy term is also assumed to be zero across the pump; however, since the discharge line of a pump may be higher than the suction line, in a more detailed analysis, that potential energy difference might be included. Solving

$$\frac{P(\text{lb}_f/\text{in}^2)(12 \text{ in}/\text{ft})^2}{62.4 \text{ lb}/\text{ft}^3} - \frac{0.75(628.2 \text{ ft lb}_f/\text{sec})}{5 \text{ lb}/\text{sec}} = 0 \quad (\text{E19.2f})$$

gives $\Delta P = 40.8 \text{ lb}_f/\text{in}^2$.

Example 19.3

A nozzle is a device that converts pressure into kinetic energy by forcing a fluid through a small-diameter opening. Turbines work in this way because the fluid (usually a gas) with a high kinetic energy impinges on turbine blades, causing spinning, and allowing the energy to be converted to electric power.

Consider a nozzle that forces 2 gal/min of water at 50 psia in a tube of 1-in inside diameter through a 0.1-in nozzle from which it discharges to atmosphere. Calculate the discharge velocity.

Solution

The system is illustrated in Figure E19.3. It is assumed that the velocity at a small distance from the end of the nozzle is identical to the velocity in the nozzle, but the contact with the atmosphere makes the pressure atmospheric.

For the case when frictional losses may be neglected, the mechanical energy balance reduces to

$$\frac{P_2 - P_1}{\rho} + \frac{u_2^2 - u_1^2}{2} = 0 \quad (\text{E19.3a})$$



Figure E19.3 Illustration of Nozzle for Example 19.3

which yields

$$\frac{(14.7 - 50) \text{ lb}_f/\text{in}^2 (12 \text{ in}/\text{ft})^2}{62.4 \text{ lb}/\text{ft}^3} + \frac{u_2^2 - \left[\frac{4(2 \text{ gal}/\text{min})(\text{ft}^3/7.48 \text{ gal})(\text{min}/60 \text{ sec})}{\pi(1/12 \text{ ft})^2} \right]^2}{2(32.2 \text{ ft}/\text{lb}_f/\text{sec}^2)} = 0 \quad (\text{E19.3b})$$

so $u_2 = 72.4 \text{ ft}/\text{sec}$. For a real system, there would be some frictional losses and the actual discharge velocity would be lower than calculated here.

This problem was solved under the assumption of turbulent flow. The criterion for turbulent flow is introduced later; however, for this system, the Reynolds number is about 2×10^5 , which is well into the turbulent flow region.

19.1.3 Force Balance

The force balance is essentially a statement of Newton's law. A common form for flow in pipes is

$$\Delta(\underline{\dot{m}u}) = \sum \underline{F} \quad (19.12)$$

where F are the forces on the system. The underlined parameters indicate vectors, since there are three spatial components of a force balance. For steady-state flow and the typical forces involved in fluid flow, Equation (19.12) reduces to

$$\dot{m}\Delta(\underline{u}) = \underline{F}_p + \underline{F}_d + \underline{F}_g + \underline{R} \quad (19.13)$$

where \underline{F}_p is the pressure force on the system, \underline{F}_d is the drag force on the system, \underline{F}_g is the gravitational force on the system, and \underline{R} is the restoring force on the system, that is, the force necessary to keep the system stationary. The term on the left side of Equation (19.12) is acceleration, confirming that Equation (19.12) is a statement of Newton's law. The most common application of Equation (19.13) is to determine the restoring forces on an elbow. These problems are not discussed here.

19.2 FLUID FLOW EQUIPMENT

The basic characteristics of fluid flow equipment are introduced in this section. The performance of pumps and compressors is dictated by their characteristic curves and, for pumps, the net positive suction head curve. The performance of these pieces of equipment is discussed in Section 19.5.

19.2.1 Pipes

Pipes and their associated fittings that are used to transport fluid through a chemical plant are usually made of metal. For noncorrosive fluids under conditions that are not of special concern, carbon steel is typical. For more extreme conditions, such as higher pressure, higher temperature, or corrosive fluids, stainless steel or other alloy steels may be needed. It may even be necessary, for very-high-temperature service such as for the flow of molten metals, to use refractory-lined pipes.

Pipes are sized using a nominal diameter and a schedule number. The higher the schedule number, the thicker the pipe walls, making pipes with a higher schedule number more suitable for higher-pressure operations. The nominal diameter is a number such as 2 in; however, there is no dimension of the pipe that is actually 2 in until the diameter reaches 14 in. For pipes with

a diameter of 14 in or larger, the nominal diameter is the outside diameter. Pipes typically have integer nominal diameters; however, for smaller diameters, they can be in increments of 0.25 in. At larger diameters, the nominal diameters may only be even integer values. Table 19.1 shows the dimensions of some schedules of standard pipe.

Table 19.1 Dimensions of Standard Steel Pipe

Nominal Size (in)	Outside Diameter		Schedule Number	Wall Thickness		Inside Diameter		Inside Cross-Sectional Area	
	in	mm		in	mm	in	mm	10 ² ft ²	10 ⁴ m ²
1/8	0.405	10.29	40	0.068	1.73	0.269	6.83	0.040	0.3664
			80	0.095	2.41	0.215	5.46	0.025	0.2341
1/4	0.540	13.72	40	0.088	2.24	0.364	9.25	0.072	0.6720
			80	0.119	3.02	0.302	7.67	0.050	0.4620
3/8	0.675	17.15	40	0.091	2.31	0.493	12.52	0.133	1.231
			80	0.126	3.20	0.423	10.74	0.098	0.9059
1/2	0.840	21.34	40	0.109	2.77	0.622	15.80	0.211	1.961
			80	0.147	3.73	0.546	13.87	0.163	1.511
3/4	1.050	26.67	40	0.113	2.87	0.824	20.93	0.371	3.441
			80	0.154	3.91	0.742	18.85	0.300	2.791
1	1.315	33.40	40	0.133	3.38	1.049	26.64	0.600	5.574
			80	0.179	4.45	0.957	24.31	0.499	4.641
1 1/4	1.660	42.16	40	0.140	3.56	1.380	35.05	1.040	9.648
			80	0.191	4.85	1.278	32.46	0.891	8.275
1 1/2	1.900	48.26	40	0.145	3.68	1.610	40.89	1.414	13.13
			80	0.200	5.08	1.500	38.10	1.225	11.40
2	2.375	60.33	40	0.154	3.91	2.067	52.50	2.330	21.65
			80	0.218	5.54	1.939	49.25	2.050	19.05
2 1/2	2.875	73.03	40	0.203	5.16	2.469	62.71	3.322	30.89
			80	0.276	7.01	2.323	59.00	2.942	27.30
3	3.500	88.90	40	0.216	5.59	3.068	77.92	5.130	47.69
			80	0.300	7.62	2.900	73.66	4.587	42.61
3 1/2	4.000	101.6	40	0.226	5.74	3.548	90.12	6.870	63.79
			80	0.318	8.08	3.364	85.45	6.170	57.35
4	4.500	114.3	40	0.237	6.02	4.026	102.3	8.840	82.19
			80	0.337	8.56	3.826	97.18	7.986	74.17
5	5.563	141.3	40	0.258	6.55	5.047	128.2	13.90	129.1
			80	0.375	9.53	4.813	122.3	12.63	117.5
6	6.625	168.3	40	0.280	7.11	6.065	154.1	20.06	186.5
			80	0.432	10.97	5.761	146.3	18.10	168.1

Nominal Size (in)	Outside Diameter		Schedule Number	Wall Thickness		Inside Diameter		Inside Cross-Sectional Area	
	in	mm		in	mm	in	mm	10 ² ft ²	10 ⁴ m ²
8	8.625	219.1	40	0.322	8.18	7.981	202.7	34.74	322.7
			80	0.500	12.70	7.625	193.7	31.71	294.7
10	10.75	273.1	40	0.365	9.27	10.02	254.5	54.75	508.6
			80	0.594	15.09	9.562	242.8	49.87	463.3
12	12.75	304.8	40	0.406	10.31	11.94	303.3	77.73	722.1
			80	0.688	17.48	11.37	288.8	70.56	655.5
14	14	355.6	40	0.438	11.13	13.12	333.2	93.97	873.0
			80	0.750	19.05	12.50	317.5	85.22	791.7
16	16	406.4	40	0.500	12.70	15.00	381.0	122.7	1140
			80	0.844	21.44	14.31	363.5	111.7	1038
18	18	457.2	40	0.562	14.27	16.88	428.8	155.3	1443
			80	0.938	23.83	16.12	409.4	141.8	1317
20	20	508.0	40	0.597	15.16	18.81	477.8	193.0	1793
			80	1.031	26.19	17.94	455.7	175.5	1630
24	24	635.0	40	0.688	17.48	22.62	574.5	279.2	2594
			80	1.219	30.96	21.56	547.6	253.6	2356

Source: Adapted from Geankoplis, C., *Transport Processes and Separation Process Principles*, 4th ed., Prentice Hall, Upper Saddle River, 2003 [1]; Perry, R. H., and D. Green, *Perry's Chemical Engineers' Handbook*, 6th ed., McGraw-Hill, New York, 1984, Section 5 [2].

Tubing is commonly used in heat exchangers. The dimensions and use of tubing are discussed in Chapter 20.

Pipes are typically connected by screw threads, flanges, or welds. Welds and flanges are more suitable for larger diameters and higher-pressure operation. Proper welds are stronger and do not leak, whereas screwed or flanged connections can leak, especially at higher pressures. Changes in direction are usually accomplished by elbows or tees, and those changes in direction are usually 90°.

19.2.2 Valves

Valves are found in piping systems. Valves are about the only way to regulate anything in a chemical process. Valves serve several functions. They are used to regulate flowrate, reduce pressure by adding resistance, or isolate (turn flow on/off) equipment.

Two common types of valves are gate valves and globe valves. Figure 19.3 shows illustrations of several common types of valves.

Gate valves are used for on/off control of fluid flow. The flow path through a gate valve is roughly straight, so when the valve is fully open, the pressure drop is very small. However, gate valves are not suitable for flowrate regulation because the flowrate does not change much until the “gate” is almost closed. There are also ball valves, in which a quarter turn opens a flow channel, and they can also be used for on/off regulation.

Globe valves are more suitable than gate valves for flowrate and pressure regulation. Because the flow path is not straight, globe valves have a higher pressure drop even when wide open. Globe valves are well suited for flowrate regulation because the flowrate is responsive to valve position. In a control system, the valve stem is raised or lowered pneumatically (by instrument air) or via

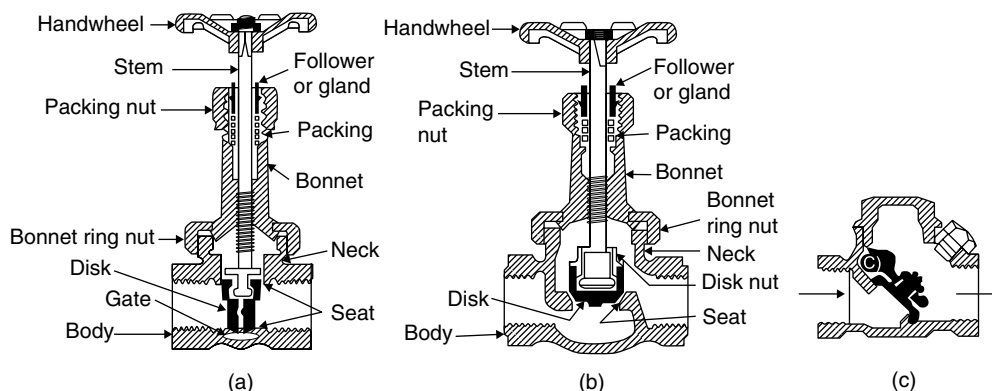


Figure 19.3 Common Types of Valves: (a) Gate, (b) Globe, (c) Swing Check (Reproduced by Permission from Couper, J. R. et al. *Chemical Process Equipment: Selection and Design*, 3rd ed. [New York, Elsevier, 2012] [3])

an electric motor in response to a measured parameter, such as a flowrate. Pneumatic systems can be designed for the valve to fail open or closed, the choice depending on the service. Failure is defined as loss of instrument air pressure. For example, for a valve controlling the flowrate of a fluid removing heat from a reactor with a highly exothermic reaction, the valve would be designed to fail open so that the reactor cooling is not lost.

Check valves, such as the swing check valve, are used to ensure unidirectional flow. In Figure 19.3(c), if the flow is left to right, the swing is opened and flow proceeds. If the flow is right to left, the swing closes, and there is no flow in that direction. Such valves are often placed on the discharge side of pumps to ensure that there is no flow reversal through the pump.

19.2.3 Pumps

Pumps are used to transport liquids, and pumps can be damaged by the presence of vapor, a phenomenon discussed in Section 19.5.2. The two major classifications for pumps are *positive displacement* and *centrifugal*. For a more detailed summary of all types of pumps, see Couper et al. [3] or Green and Perry [4].

Positive-displacement pumps are often called **constant-volume pumps** because a fixed amount of liquid is taken into a chamber at a low pressure and pushed out of the chamber at a high pressure. The chamber has a fixed volume, hence the name. An example of a positive-displacement pump is a reciprocating pump, illustrated in Figure 19.4(a). Specifically, this is an example of a piston pump in which the piston moves in one direction to pull liquid into the chamber and then moves in the opposite direction to discharge liquid out of the chamber at a higher pressure. There are other variations of positive-displacement pumps, such as rotary pumps in which the chamber moves between the inlet and discharge points. In general, positive-displacement pumps can increase pressure more than centrifugal pumps and run at higher pressures overall. These characteristics define their applicability. Efficiencies tend to be between 50% and 80%. Positive-displacement pumps are preferred for higher pressures, higher viscosities, and anticipated viscosity variations.

In centrifugal pumps, which are a common workhorse in the chemical industry, the pressure is increased by the centrifugal action of an impeller. An impeller is a rotating shaft with blades, and it might be tempting to call it a propeller because an impeller resembles a propeller. (While there might be a resemblance, the term **propeller** is reserved for rotating shafts with blades that move an object, such as a boat or airplane.) The blades of an impeller have small openings, known as

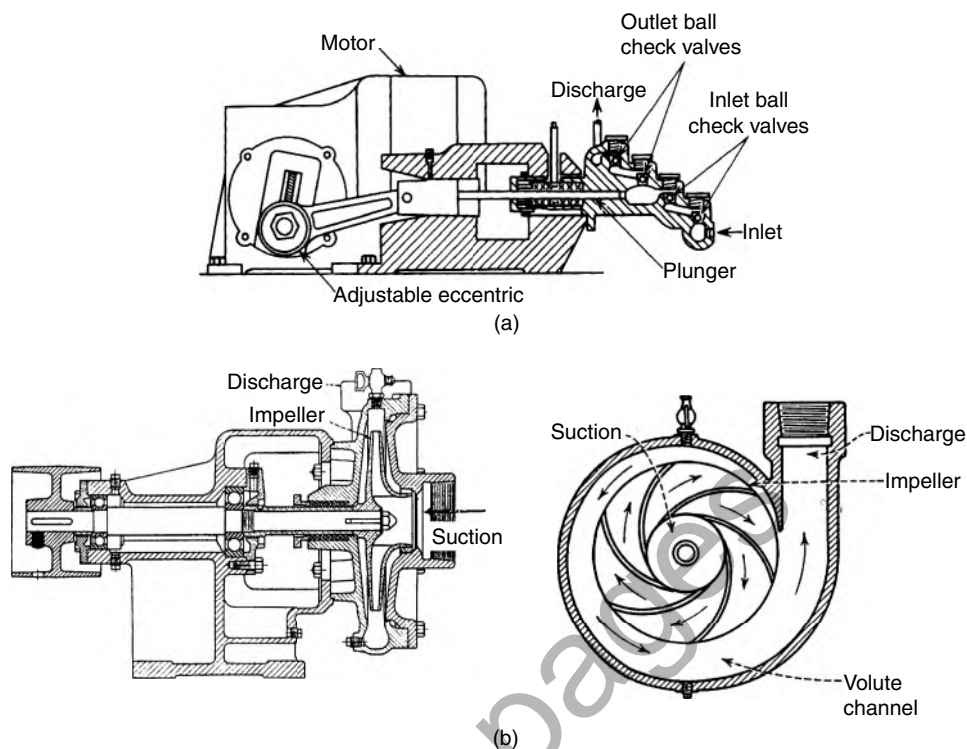


Figure 19.4 (a) Inner Workings of Positive-Displacement Pump, (b) Inner Workings of Centrifugal Pump ([a] Reproduced by Permission from McCabe, W. L. et al., *Unit Operations of Chemical Engineering*, 5th ed. [New York, McGraw-Hill, 1993] [5]; [b] Reproduced by Permission from Couper, J. R. et al., *Chemical Process Equipment: Selection and Design*, 3rd ed. [New York, Elsevier, 2012] [3])

vanes, that increase the kinetic energy of the liquid. The liquid is then discharged through a **volute** in which the kinetic energy is converted into pressure. Figure 19.4(b) shows a centrifugal pump. Centrifugal pumps often come with impellers of different diameters, which enable pumps to be used for different services (different pressure increases). Of course, shutdown is required to change the impeller. Although standard centrifugal pump impellers only spin at a constant rate, variable-speed centrifugal pumps also are available.

Centrifugal pumps can handle a wide range of capacities and pressures, and depending on the exact type of pump, the efficiencies can range from 20% to 90%.

19.2.4 Compressors

Devices that increase the pressure of gases fall into three categories: fans, blowers, and compressors. Figure 19.5 illustrates some of this equipment. For a more detailed summary of all types of pumps, see Couper et al. [3] or Green and Perry [4].

Fans provide very low-pressure increases (<1 psi [7 kPa]) for low volumes and are typically used to move air. Blowers are essentially mini-compressors, providing a maximum pressure of about 30 psi (200 kPa). Blowers can be either positive displacement or centrifugal, and while their general construction is similar to pumps, there are many internal differences. Compressors, which can also be either positive displacement or centrifugal, can provide outlet pressures of 1500 psi (10 MPa) and sometimes even 10 times that much.

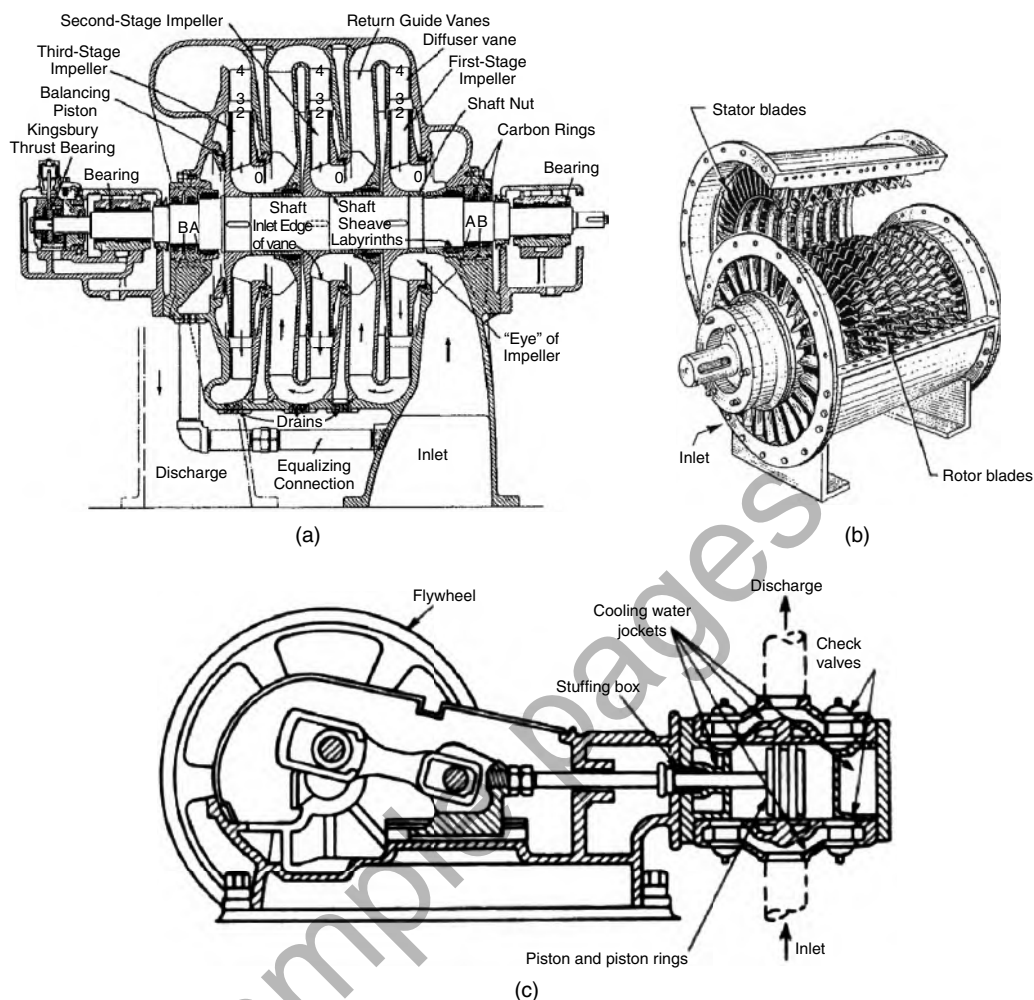


Figure 19.5 Inner Working of Compressors: (a) Centrifugal, (b) Axial, (c) Positive Displacement ([a] and [b] Reproduced by Permission from Couper, J. R. et al., *Chemical Process Equipment: Selection and Design*, 3rd ed. [New York: Elsevier, 2012]; [c] Reproduced by Permission from McCabe, W. L. et al., *Unit Operations of Chemical Engineering*, 5th ed. [New York: McGraw-Hill, 1993])

In a centrifugal compressor, the impeller may spin at tens of thousands of revolutions per minute. If liquid droplets or solid particles are present in the gas, they hit the impeller blades at such high relative velocity that the impeller blades will erode rapidly and may cause bearings to become damaged, leading to mechanical failure. The compressor casing also may crack. Therefore, it is important to ensure that the gas in a centrifugal compressor does not contain solids and liquids. A filter can be used to keep particles out of a compressor, and a packed-bed adsorbent can also be used, for example, to remove water vapor from inlet air. Knockout drums are often provided between compressor stages with intercooling to allow the disengagement of any condensed drops of liquid and are covered in more detail in Chapter 23, Section 23.2. The seals on compressors are temperature sensitive, so a maximum temperature in one stage of a compressor is generally not exceeded, which is another reason for staged, intercooled compressor systems. It should also be noted that compressors are often large and expensive pieces of equipment that often have a large number of auxiliary systems associated with them. The coverage given in this text is very simplified but allows the estimate of the power required.

Positive-displacement compressors typically handle lower flowrates but can produce higher pressures compared to centrifugal compressors. Efficiencies for both types of compressor tend to be high, above 75%.

19.3 FRICTIONAL PIPE FLOW

19.3.1 Calculating Frictional Losses

The fourth term in Equation (19.5) must be evaluated to include friction in the mechanical energy balance. There are different expressions for this term depending on the type of flow and the system involved. In general, the friction term is

$$e_f = \frac{2fLu^2}{D} = \frac{32fL\dot{V}^2}{\pi^2 D^5} \quad (19.14)$$

where L is the pipe length, D is the pipe diameter, and f is the Fanning friction factor. (The Fanning friction factor is typically used by chemical engineers. There is also the Moody friction factor, which is four times the Fanning friction factor. Care must be used when obtaining friction factor values from different sources. It is even more confusing, since the plot of friction factor versus Reynolds number is called a *Moody plot* for both friction factors.) The friction factor is a function of the Reynolds number ($Re = Du\rho/\mu$, where μ is the fluid viscosity), and its form depends on the flow regime (laminar or turbulent), and for turbulent flow, f is also a function of the pipe roughness factor (e , a length that represents small asperities on the pipe wall; values are given at the top of Figure 19.6), which is a tabulated value.

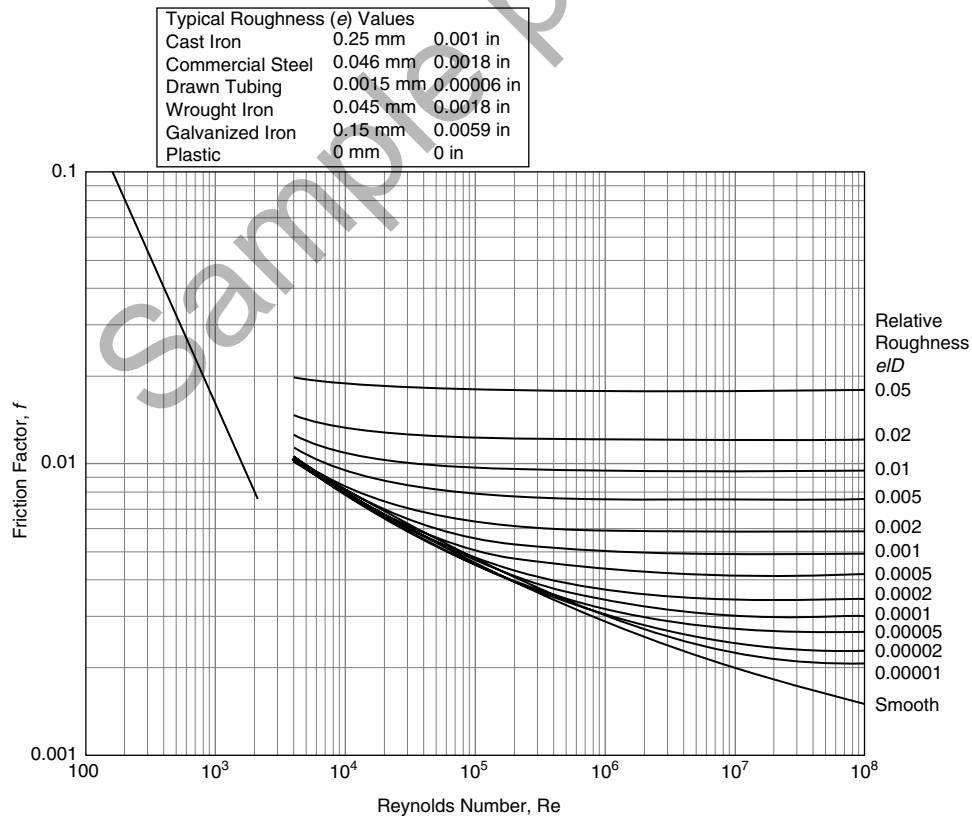


Figure 19.6 Moody Plot for the Fanning Friction Factor in Pipes

Historically, the friction factor was measured and the data were plotted in graphical form. Figure 19.6 is such a plot. A key observation from Figure 19.6 is that, with the exception of smooth pipes, the friction factor asymptotically approaches a constant value above a Reynolds number of approximately 10^5 . This is called fully developed turbulent flow, and the friction factor becomes constant and can be used to simplify certain calculations, examples of which are presented later. Typical values for the pipe roughness for some common materials are shown at the top of Figure 19.6.

The friction factor for laminar flow is a theoretical result derivable from the Hagen-Poiseuille equation [6] and is valid for $Re < 2100$.

$$f = \frac{16}{Re} = \frac{16\mu}{Du\rho} \quad (19.15)$$

For turbulent flow, the data have been fit to equations. One such equation is the Pavlov equation ([7] [cited in Levenspiel [8]]):

$$\frac{1}{f^{0.5}} = -4 \log_{10} \left[\frac{e}{3.7D} + \left(\frac{6.81}{Re} \right)^{0.9} \right] \quad (19.16)$$

The Pavlov equation provides results within a few percent of the measured data. There are more accurate equations; however, they are not explicit in the friction factor. Any of these curve fits provides significantly more accuracy than reading a graph.

For flow in pipes containing valves, elbows, and other pipe fittings, there are two common methods for including the additional frictional losses created by this equipment. One is the **equivalent length** method, whereby additional pipe length is added to the value of L in Equation (19.14). The other method is the **velocity head** method, in which a value (K_i) is assigned to each valve, fitting, and so on, and an additional frictional loss term is added to the frictional loss term in Equation (19.14). These terms are of the form

$$\sum_i \frac{K_i u_i^2}{2} \quad (19.17)$$

where the index i indicates a sum over all valves, elbows, and similar components in the system. If there are different pipe diameters within the system, the velocity in Equation (19.17) is specific to each section of pipe, and a term for each section of pipe must be included. It should be noted that the equivalent K_i value for straight pipe (K_{pipe}) is given by

$$K_{pipe} = \frac{4fL}{D} \quad (19.18)$$

Tables 19.2 and 19.3 show equivalent lengths and K_i values for some common items found in pipe networks, for turbulent flow and for laminar flow, respectively. The values are different for laminar and turbulent flow. Darby [9] presents analytical expressions for the K values that can be used for more exact calculations.

Another common situation involves frictional loss in a packed bed, that is, a vessel packed with solids. One application is if the solids are catalysts, making the packed bed a reactor. The frictional loss term for packed beds is obtained from the Ergun equation, which yields a friction term for a packed bed as

$$e_f = \frac{Lu_s^2(1-\varepsilon)}{\varepsilon^3 D_p} \left[\frac{150(1-\varepsilon)\mu}{D_p u_s \rho} + 1.75 \right] \quad (19.19)$$

where u_s is the superficial velocity (based on pipe diameter, not particle diameter), D_p is the particle diameter (assumed spherical here; corrections are available for nonspherical shape), and ε is the packing void fraction, which is the volume fraction in the packed bed not occupied by solids.

Table 19.2 Frictional Losses for Turbulent Flow

Type of Fitting or Valve	Frictional Loss, Number of Velocity Heads, K_f	Frictional Loss, Equivalent Length of Straight Pipe, in Pipe Diameters, L_{eq}/D
45° elbow	0.35	17
90° elbow	0.75	35
Tee	1	50
Return bend	1.5	75
Coupling	0.04	2
Union	0.04	2
Gate valve, wide open	0.17	9
Gate valve, half open	4.5	225
Globe valve, wide open	6.0	300
Globe valve, half open	9.5	475
Angle valve, wide open	2.0	100
Check valve, ball	70.0	3500
Check valve, swing	2.0	100
Contraction	$0.55(1 - A_2/A_1)$	$27.5(1 - A_2/A_1)$
Contraction $A_2 \ll A_1$	0.55	27.5
Expansion	$(1 - A_1/A_2)^2$	$50(1 - A_1/A_2)^2$
Expansion $A_1 \ll A_2$	1	50

Source: From Geankoplis, C., *Transport Processes and Separation Process Principles*, 4th ed., (Upper Saddle River, NJ: Prentice Hall, 2003); Perry, R. H., and D. Green, *Perry's Chemical Engineers' Handbook*, 6th ed. (New York: McGraw-Hill, 1984), Section 5.

Table 19.3 Frictional Loss for Laminar Flow

Reynolds number	Frictional Loss, Number of Velocity Heads, K_f					
	50	100	200	400	1000	Turbulent
90° elbow	17	7	2.5	1.2	0.85	0.75
Tee	9	4.8	3.0	2.0	1.4	1.0
Globe valve	28	22	17	14	10	6.0
Check valve, swing	55	17	9	5.8	3.2	2.0

Source: From Geankoplis, C., *Transport Processes and Separation Process Principles*, 4th ed. (Upper Saddle River, NJ: Prentice Hall, 2003), 99–100, citing Kittredge, C. P., and D. S. Rowley, "Resistance Coefficients for Laminar and Turbulent Flow Through One-Half-Inch Valves and Fittings," *Trans. ASME*, 79 (1957): 1759–1766.

When Equation (19.19) is used in the mechanical energy balance, one unknown parameter, such as velocity, pressure drop, or particle diameter, can be obtained.

For incompressible flow in packed beds, the Ergun equation, Equation (19.19), is used for the friction term in the mechanical energy balance.

For the expansion and contraction losses, A_1 is the cross-sectional area of the pipe, subscript 1 is the upstream area, and subscript 2 is the downstream area.

19.3.2 Incompressible Flow

19.3.2.1 Single-Pipe Systems

Incompressible flow problems fall into three categories:

1. Any parameter unknown in the mechanical energy balance other than velocity (flowrate) or diameter
2. Unknown velocity (flowrate)
3. Unknown diameter

For turbulent flow problems with any unknown other than velocity (or flowrate) or diameter, in the mechanical energy balance, Equation (19.5), there is a second unknown: the friction factor. The friction factor can be calculated from Equation (19.15). The solution method can use a sequential calculation, solving Equation (19.5) for the unknown once the friction factor is calculated. If there are valves, elbows, and so on, the length term in Equation (19.15) can be adjusted appropriately or Equation (19.17) can be used. Alternatively, Equations (19.14) and (19.16) can be solved simultaneously to yield all the unknowns. Example 19.5 shows both of these calculation methods. For laminar flow problems, Equation (19.15) can be combined with Equation (19.14) in the mechanical energy balance to solve any problem analytically.

For turbulent flow, if the velocity is unknown, Equations (19.5) and (19.15) must be solved simultaneously for the velocity or flowrate and the friction factor. When solving for a velocity directly, if the pump work term must be included, it is necessary to express the mass flowrate in terms of velocity. If solving for the volumetric flowrate, the second equality in Equation (19.13) must be used, and if a kinetic energy term is required in the mechanical energy balance, the velocities must be expressed in terms of volumetric flowrate. In the friction factor equation, the Reynolds number also needs to be expressed in terms of the volumetric flowrate as follows:

$$\text{Re} = \frac{Dv\rho}{\mu} = \frac{D\rho}{\mu} \frac{\dot{v}}{A} = \frac{D\rho}{\mu} \frac{4\dot{v}}{\pi D^2} = \frac{4\dot{v}\rho}{\pi D\mu} \quad (19.20)$$

For laminar flow, an analytical solution is possible simply by using Equation (19.14) for the friction factor in the mechanical energy balance.

For turbulent flow, if the diameter is unknown, Equations (19.5) and (19.13) (second equality involving flowrate and diameter to the fifth power) must be solved simultaneously, using Equation (19.20) for the Reynolds number. For laminar flow, an analytical solution may once again be possible by using Equation (19.12) for the friction factor in the mechanical energy balance. If kinetic energy terms are involved, an unknown diameter will appear when expressing velocity in terms of flowrate. If minor losses are involved, the equivalent length will include a diameter term, and the K-value method will include a diameter in the conversion between flowrate and velocity.

Examples 19.4 and 19.5 illustrate the methods for solving these types of problems.

Example 19.4

Consider a physical situation similar to that in Example 19.2. The flowrate between tanks is 10 lb/sec. The source-tank level is 10 ft off of the ground, and the discharge-tank level is 50 ft off of the ground. For this example, both tanks are open to the atmosphere. The suction-side pipe is 2-in. schedule-40, commercial steel, and the discharge-side pipe is 1.5-in. schedule-40, commercial steel. The length of the suction line is 25 ft, and the length of the discharge line is 60 ft. The pump efficiency is 75%. Losses due to fittings, expansions, and contractions may be assumed negligible for this problem.

- Determine the required horsepower of the pump.
- Determine the pressures before and after the pump.

Solution

- The physical situation is depicted in Figure E19.4.

For the control volume of the fluid in both tanks, the pipes, and the pump, the mechanical energy balance reduces to

$$g\Delta z + e_{f,\text{suct}} + e_{f,\text{disch}} - \frac{\eta_p \dot{W}_s}{\dot{m}} = 0 \quad (\text{E19.4a})$$

The pressure term is zero, because both tanks are open to the atmosphere ($P_1 = P_2 = 1 \text{ atm}$). The kinetic energy term is zero, because the velocities of the fluid at the surfaces of both tanks are assumed to be zero. There are two friction terms, one for the suction side of the pump and one for the discharge side of the pump, because the friction factors are different due to the different pipe diameters.

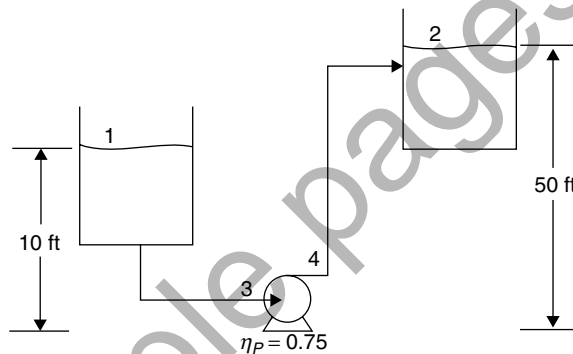


Figure E19.4

To calculate the friction terms, the Reynolds numbers must be calculated first for each section to determine whether the flow is laminar or turbulent. Since a temperature is not provided, the density is assumed to be 62.4 lb/ft^3 , and the viscosity is assumed to be $1 \text{ cP} = 6.72 \times 10^{-4} \text{ lb/ft}\cdot\text{sec}$. Using Table 19.1 for the schedule pipe diameter and cross-sectional area, the Reynolds number for the suction side is

$$\text{Re} = \frac{D u \rho}{\mu} = \frac{(2.067/12 \text{ ft}) \left(\frac{10 \text{ lb/sec}}{(0.0233 \text{ ft}^2)(62.4 \text{ lb/ft}^3)} \right) (62.4 \text{ lb/ft}^3)}{6.72 \times 10^{-4} \text{ lb/ft}\cdot\text{sec}} = 110,000 \quad (\text{E19.4b})$$

Similarly, the Reynolds number for the discharge side is 141,200. Therefore, the flow is turbulent in both sections of pipe. The friction factor is now calculated for each section of pipe. For the suction side, with commercial-steel pipe ($e = 0.0018 \text{ in}$ from the top of Figure 19.6),

$$\frac{1}{f^{0.5}} = -4 \log_{10} \left[\frac{0.0018 \text{ in}}{3.7(2.067 \text{ in})} + \left(\frac{6.81}{110,010} \right)^{0.9} \right] \quad (\text{E19.4c})$$

so $f_{\text{suct}} = 0.0054$. Similarly, $f_{\text{disch}} = 0.0055$. Now, the mechanical energy balance on the entire system is used to solve for the pump power: