ANALYSIS, SYNTHESIS, AND DESIGN OF CHEMICAL PROCESSES

FIFTH EDITION

RICHARD TURTON | JOSEPH A. SHAEIWITZ
DEBANGSU BHATTACHARYYA | WALLACE B. WHITING

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Contents

CHAPTER

19

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.

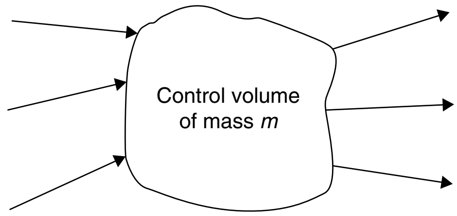

Figure 19.1 General Control Volume

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.

$$\sum_{i=1}^{m} \dot{m}_{i,in} = \sum_{i=1}^{m} \dot{m}_{i,out} \tag{19.1}$$

$$\dot{m}_1 = \dot{m}_2 \tag{19.2}$$

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

$$\dot{m} = \rho \dot{v} = \rho A u \tag{19.3}$$

where ρ is the density (mass/volume) and A is the cross-sectional area for flow (length^2). 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

$$G = \frac{\dot{m}}{A} = \rho u \tag{19.4}$$

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,

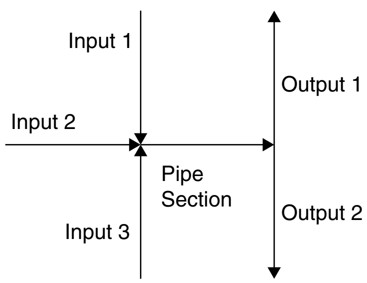

Figure 19.2 System with Multiple Inputs and Outputs

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 m^3/s, and its pipe diameter is 0.078 m. The volumetric flowrate of Stream 2 is 0.009 m^3/s, and its pipe diameter is 0.10 m.

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

Solution

a. 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 m^3/s. To obtain the mass flowrate, $\dot{m} = \rho \dot{v}_3$, so $\dot{m}_3 = \left(887\,\text{kg/m}^3\right)\left(0.015\,\text{m}^3/\text{s}\right) = 13.3\,\text{kg/s}$. Alternatively, the mass flowrate of Streams 1 and 2 could be calculated and added to get the same result.

b. From Equation (19.3), at constant density $u = \dot{v}/A$. Therefore,

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

$$u_2 = \frac{\dot{v}_2}{A_2} = \frac{4\dot{v}_2}{\pi D_2^2} = \frac{4\left(0.009\,\text{m}^3/\text{s}\right)}{\pi\left(0.01\,\text{m}\right)^2} = 1.15\,\text{m/s} \qquad \text{(E19.lb)}$$

c. The diameter at which $u_3 = 1$ m/s can be calculated from Equation (19.3) at constant density.

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

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.

d. From Equation (19.4), using the rounded values,

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

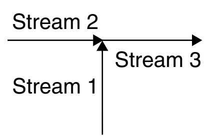

Figure E19.1 Physical Situation in Example 19.1

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

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

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 length^2/time^2. In SI units, since 1 J =1 kg m^2/s^2, it is clear that 1 J/kg = 1 m^2/s^2. In American Engineering units, since 1 lb_f = 32.2 ft lb_m/sec^2, this conversion factor (often called g_c) must be used to reconcile the units. The notation < > 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

$$\int_1^2 \frac{dP}{\rho} = \frac{\Delta P}{\rho} \tag{19.6}$$

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

$$\frac{<u^3>}{<u>} \approx <u>^2 \tag{19.7}$$

For laminar flow,

$$\frac{<u^3>}{<u>} \approx 2<u>^2 \tag{19.8}$$

For simplicity, $<u^2>$ is hereafter represented as $<u>^2$, 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):

$$\dot{W}_s = \dot{m} W_s \tag{19.9}$$

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

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

$$\frac{W_s}{\eta_t} = \frac{\dot{W}_s}{\eta_t \dot{m}} \text{ turbine} \tag{19.11}$$

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.

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

Solution

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

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

Figure E19.2 is an illustration of the system.

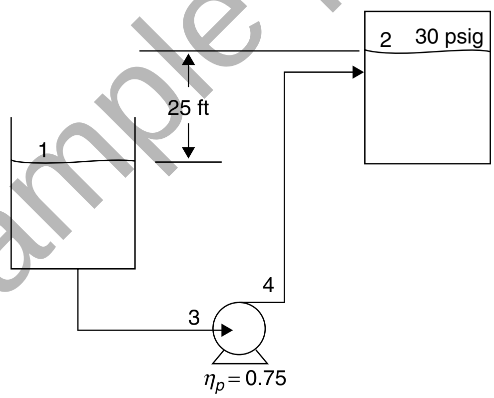

Figure E19.2 Physical Situation for Example 19.2

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 atm = 0 psig. The friction term is assumed to be zero in this problem, as stated. So, Equation (E19.2a) reduces to

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

and

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

so, $\dot{W}_s =\ 628.2\ \mathrm{ft\ lb_f/sec}$.

Converting to horsepower yields

$$\dot{W}_s=\frac{628.2\ \mathrm{ft\ lb_f/sec}}{550\ \mathrm{ft\ lb_f/hp/sec}}=1.14\ \mathrm{hp} \tag{E19.2d}$$

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

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

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

$$\frac{P\left(\mathrm{lb_f/in^2}\right)\left(12\,\mathrm{in/ft}\right)^2}{62.4\ \mathrm{lb/ft^3}}-\frac{0.75(628.2\ \mathrm{ft\ lb_f/sec})}{5\ \mathrm{lb/sec}}=0 \tag{E19.2f}$$

gives $\Delta P = 40.8\ \mathrm{lb_f/in^2}$.

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

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

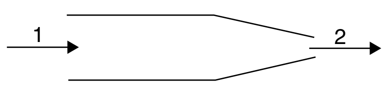

Figure E19.3 Illustration of Nozzle for Example 19.3

which yields

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

so $u_2 = 72.4$ ft/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

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

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

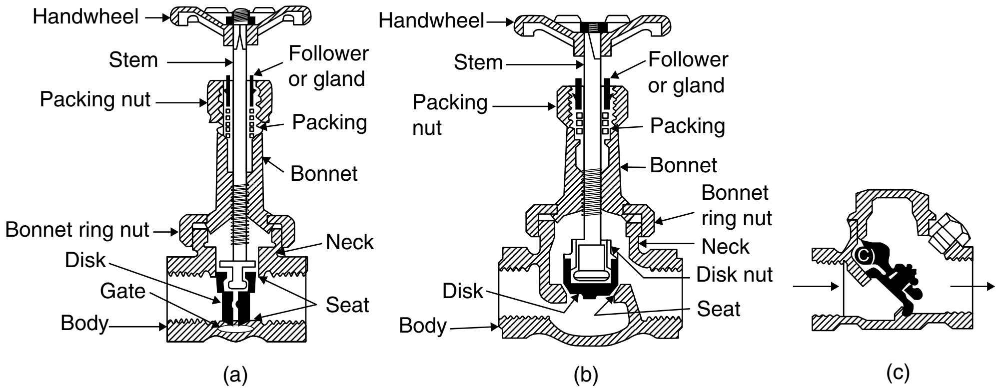

Figure 19.3 Common Types of Valves: (a) Gate, (b) Globe, (c) Swing Check (Reproduced by Permission from Couper, J. R. et al. *Chemical Process Equipment: Selection and Design,* 3rd ed. [New York, Elsevier, 2012] [3])

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

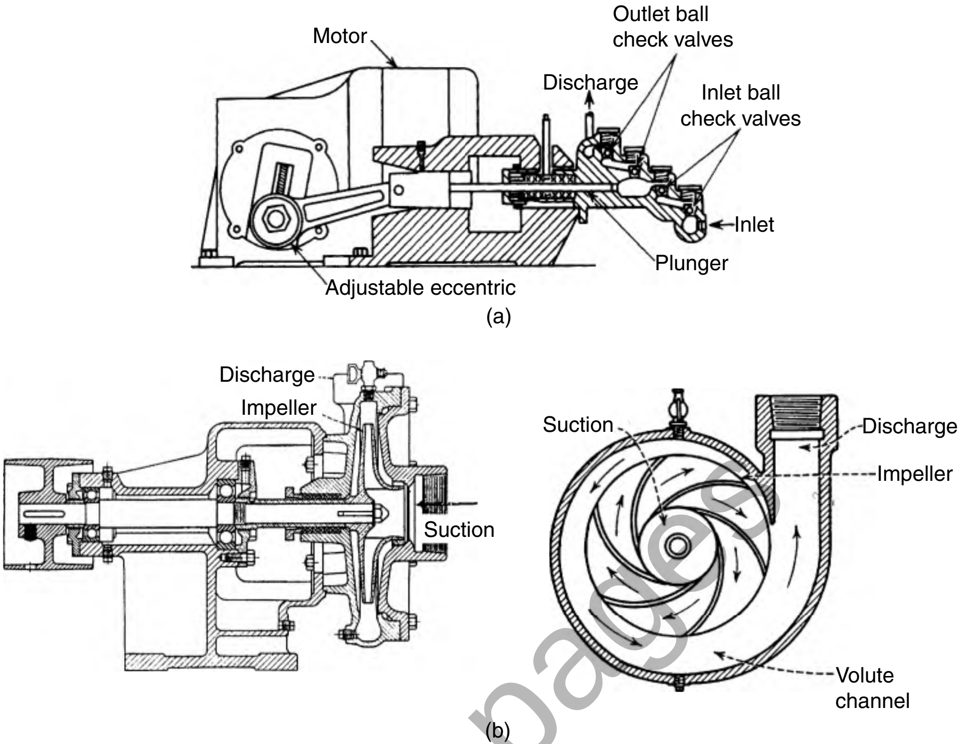

Figure 19.4 (a) Inner Workings of Positive-Displacement Pump, (b) Inner Workings of Centrifugal Pump ([a] Reproduced by Permission from McCabe, W. L. et al., *Unit Operations of Chemical Engineering,* 5th ed. [New York, McGraw-Hill, 1993] [5]; [b] Reproduced by Permission from Couper, J. R. et al., *Chemical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

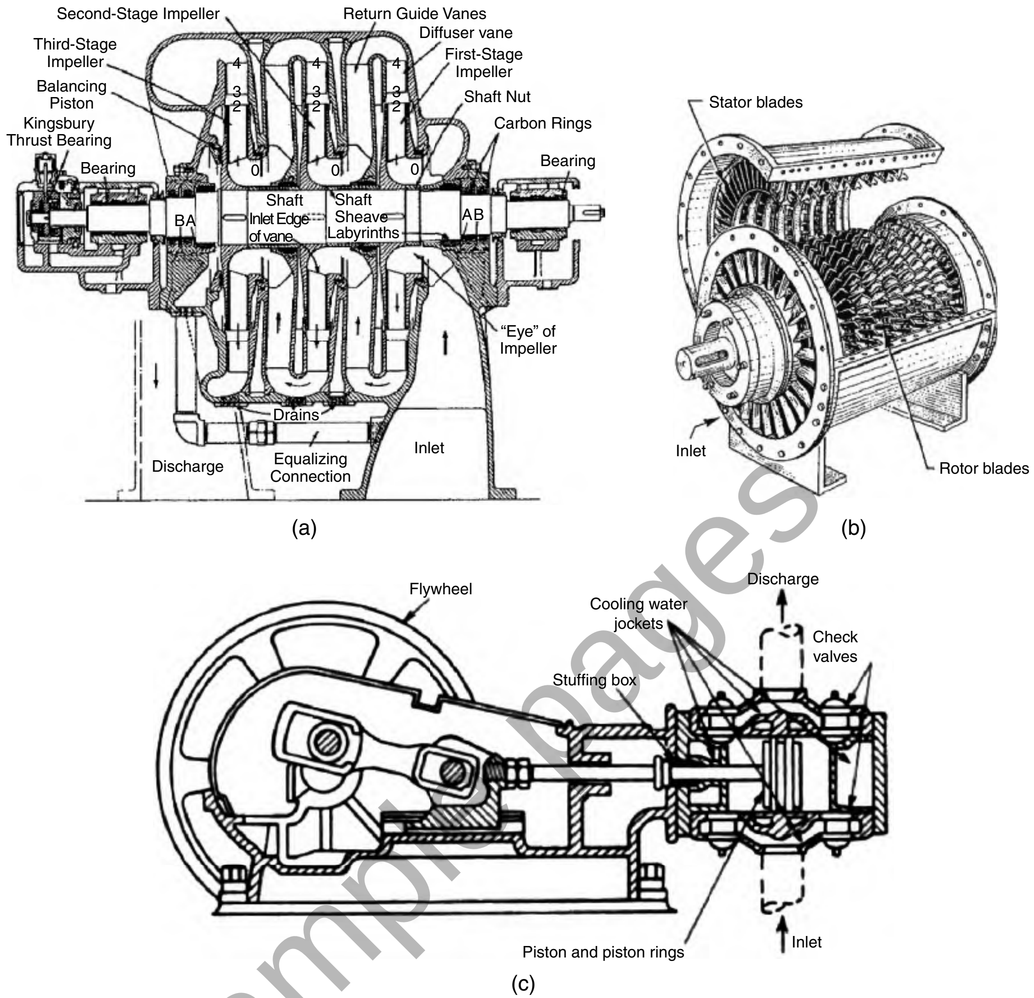

Figure 19.5 Inner Working of Compressors: (a) Centrifugal, (b) Axial, (c) Positive Displacement ([a] and [b] Reproduced by Permission from Couper, J. R. et al., *Chemical Process Equipment: Selection and Design,* 3rd ed. [New York: Elsevier, 2012]; [c] Reproduced by Permission from McCabe, W. L. et al.,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

$$e_f = \frac{2fLu^2}{D} = \frac{32fL\dot{v}^2}{\pi^2 D^5} \tag{19.14}$$

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 ($\mathrm{Re} = Du\rho/\mu$, 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.

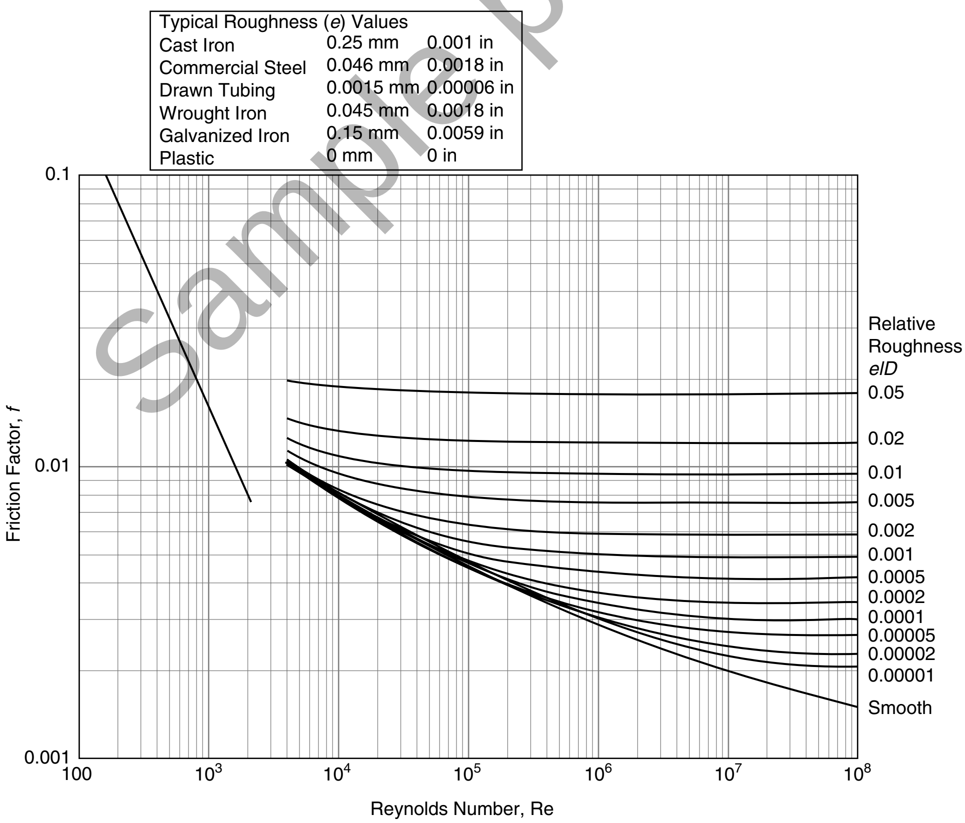

Figure 19.6 Moody Plot for the Fanning Friction Factor in Pipes

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.

$$f = \frac{16}{\text{Re}} = \frac{16\mu}{Du\rho} \tag{19.15}$$

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

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

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

$$\sum_i \frac{K_i u_i^2}{2} \tag{19.17}$$

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

$$K_{pipe} = \frac{4fL}{D} \tag{19.18}$$

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

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

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 $A_2 << A_1$	0.55	27.5
Expansion	$(1 - A_1/A_2)^2$	$50(1 - A_1/A_2)^2$
Expansion $A_1 << A_2$	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

	Frictional Loss, Number of Velocity Heads, K_f					
Reynolds number	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_i 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:

$$Re = \frac{Du\rho}{\mu} = \frac{D\rho}{\mu}\frac{\dot{v}}{A} = \frac{D\rho}{\mu}\frac{4\dot{v}}{\pi D^2} = \frac{4\dot{v}\rho}{\pi D\mu} \tag{19.20}$$

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.

a. Determine the required horsepower of the pump.
b. Determine the pressures before and after the pump.

Solution

a. 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

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

The pressure term is zero, because both tanks are open to the atmosphere ($P_1 = P_2 = 1$ 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.

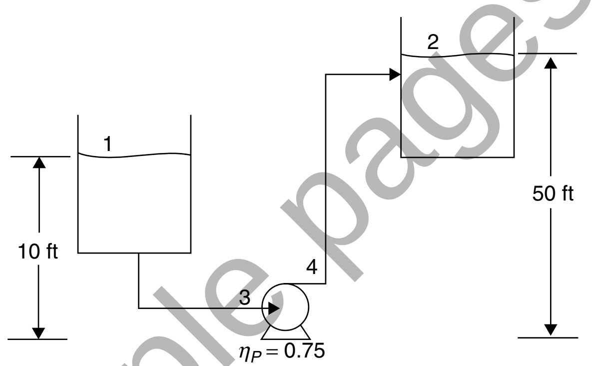

Figure E19.4

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 cP $= 6.72\times10^{-4}$ lb/ft/sec. Using Table 19.1 for the schedule pipe diameter and cross-sectional area, the Reynolds number for the suction side is

$$\mathrm{Re} = \frac{Du\rho}{\mu} = \frac{(2.067/12\ \mathrm{ft})\left(\dfrac{10\ \mathrm{lb/sec}}{(0.0233\ \mathrm{ft}^2)(62.4\ \mathrm{lb/ft}^3)}\right)(62.4\ \mathrm{lb/ft}^3)}{6.72\times10^{-4}\ \mathrm{lb/ft/sec}} = 110{,}000 \qquad \text{(E19.4b)}$$

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 ($e = 0.0018$ in from the top of Figure 19.6),

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

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