Taibah University - Yanbu Branch
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Mechanical Engineering Department

Heating, Ventilating, and Air Conditioning

HVAC ME472
Part 6

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Fluid Flow Basics

- The distribution of fluids by pipes, ducts. and conduits is essential to all heating and cooling systems.

- The fluids encountered are gases, vapors, liquids, and mixtures of liquid and vapor (two-phase flow).

- The adiabatic, steady flow of a fluid in a pipe or conduit is governed by the first law of thermodynamics, which leads to the equation:

$$\boxed{\frac{P_1}{\rho_1} + \frac{\overline{V}_1^2}{2g_c} + \frac{gz_1}{g_c} = \frac{P_2}{\rho_2} + \frac{\overline{V}_2^2}{2g_c} + \frac{gz_2}{g_c} + w + \frac{g}{g_c} l_f} \qquad (6\text{-}1)$$

where

P = static pressure, lbf/ft^2 or N/m^2
ρ = mass density at a cross section, lbm/ft^3 or kg/m^3
$\overline{V}$ = average velocity at a cross section, ft/sec or m/s
g = local acceleration of gravity, ft/sec^2 or m/s^2
g_c = constant = 32.17 (lbm-ft)/(lbf-sec^2) = 1.0 (kg-m)/(N-s^2)
z = elevation, ft or m
w = work, (ft-lbf)/lbm or J/kg
l_f = lost head, ft or m

Fluid Flow Basics

- For one-dimensional flow along a single conduit the mass rate of flow at any two cross sections 1 and 2 is given by: $\dot{m} = \rho_1 \overline{V}_1 A_1 = \rho_2 \overline{V}_2 A_2$ (6-2)

where:

$\dot{m}$ = mass flow rate, lbm/sec or kg/s
A = cross-sectional area normal to the flow, ft^2 or m^2

- When the fluid is incompressible Equation 6-2 becomes

$$\dot{Q} = \overline{V}_1 A_1 = \overline{V}_2 A_2$$)

where:

$\dot{Q}$ = volume flow rate, ft^3/sec or m^3/s

- Combing Eqs 6-1 and 6 -2 and solve for $\dot{W} = \dot{m}w$ and assume ρ = constant

$$\dot{W} = \dot{m}\left[\frac{P_1 - P_2}{\rho} + \frac{\overline{V}_1^2 - \overline{V}_2^2}{2g_c} + \frac{g(z_1 - z_2)}{g_c} - \frac{g}{g_c} l_f\right] \qquad (6\text{-}4a)$$

Where: $\dot{W}$ = power (work per unit time), $\frac{\text{ft-lbf}}{\text{sec}}$ or W

Fluid Flow Basics

- Some of the terms in Eqs. 6-1 and 6-4, be zero or negligibly small.
- When the fluid flowing is a liquid, the **velocity** terms are usually rather small and can be neglected.
- 1n the case of flowing gases, the **potential terms** are usually very small and can be neglected; however, the kinetic energy terms may be quite important.
- Obviously the work term will be zero when no pump, turbine, or fan is present.

- The total pressure is the sum of the static pressure and the velocity pressure

$$P_0 = P + \frac{\rho \bar{V}^2}{2g_c} \qquad \text{(6-4 b)}$$

Lost Head

- For incompressible flow in pipes and ducts the lost head is expressed as

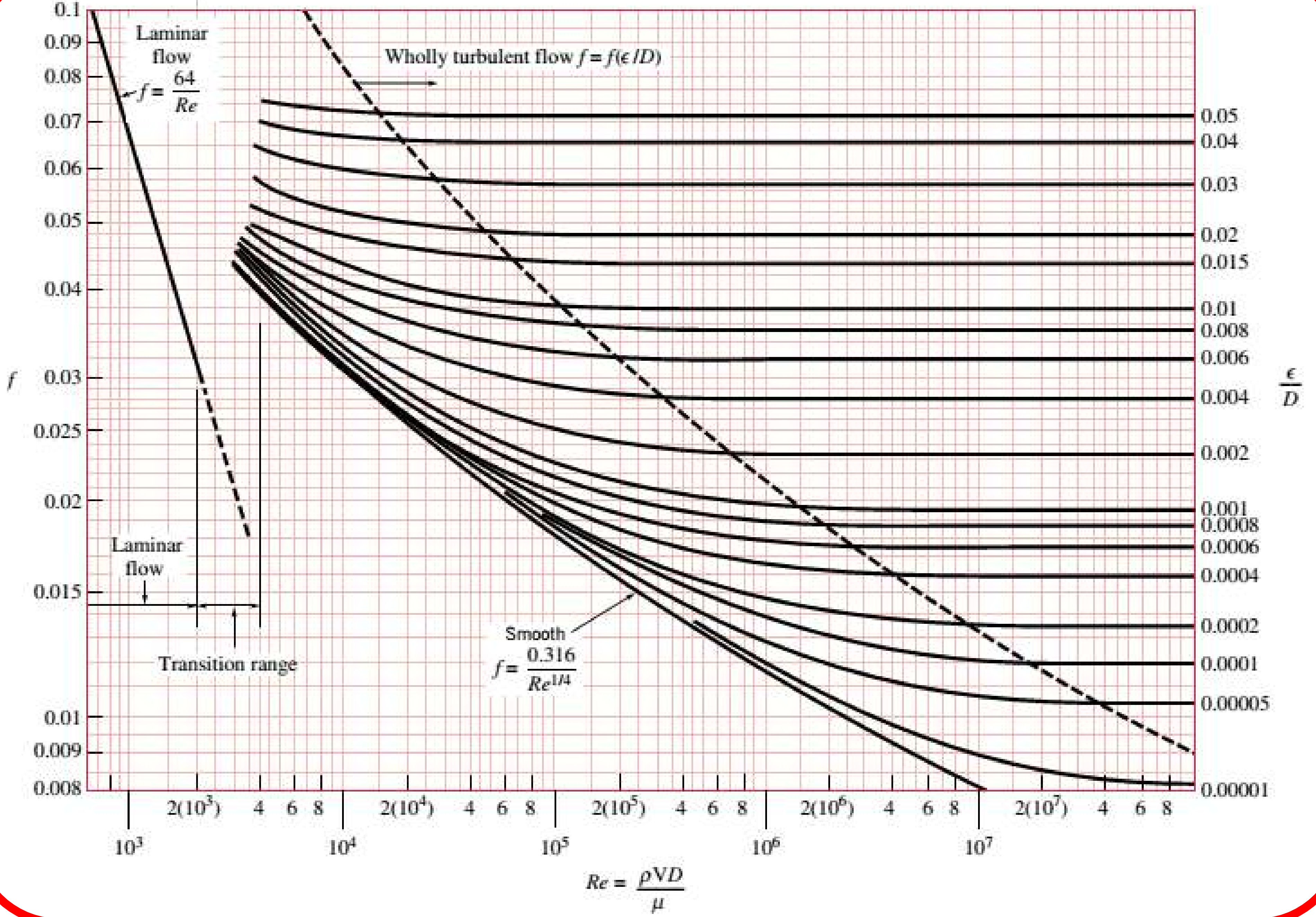
$$l_f = f \frac{L}{D} \frac{\overline{V}^2}{2g} \qquad (6\text{-}5)$$

where:

f = Moody friction factor
L = length of the pipe or duct, ft or m
D = diameter of the pipe or duct, ft or m
$\overline{V}$ = average velocity in the conduit, ft/sec or m/s
g = acceleration due to gravity, ft/sec^2 or m/s^2

- For conduits of noncircular cross section, the hydraulic diameter D_h is:

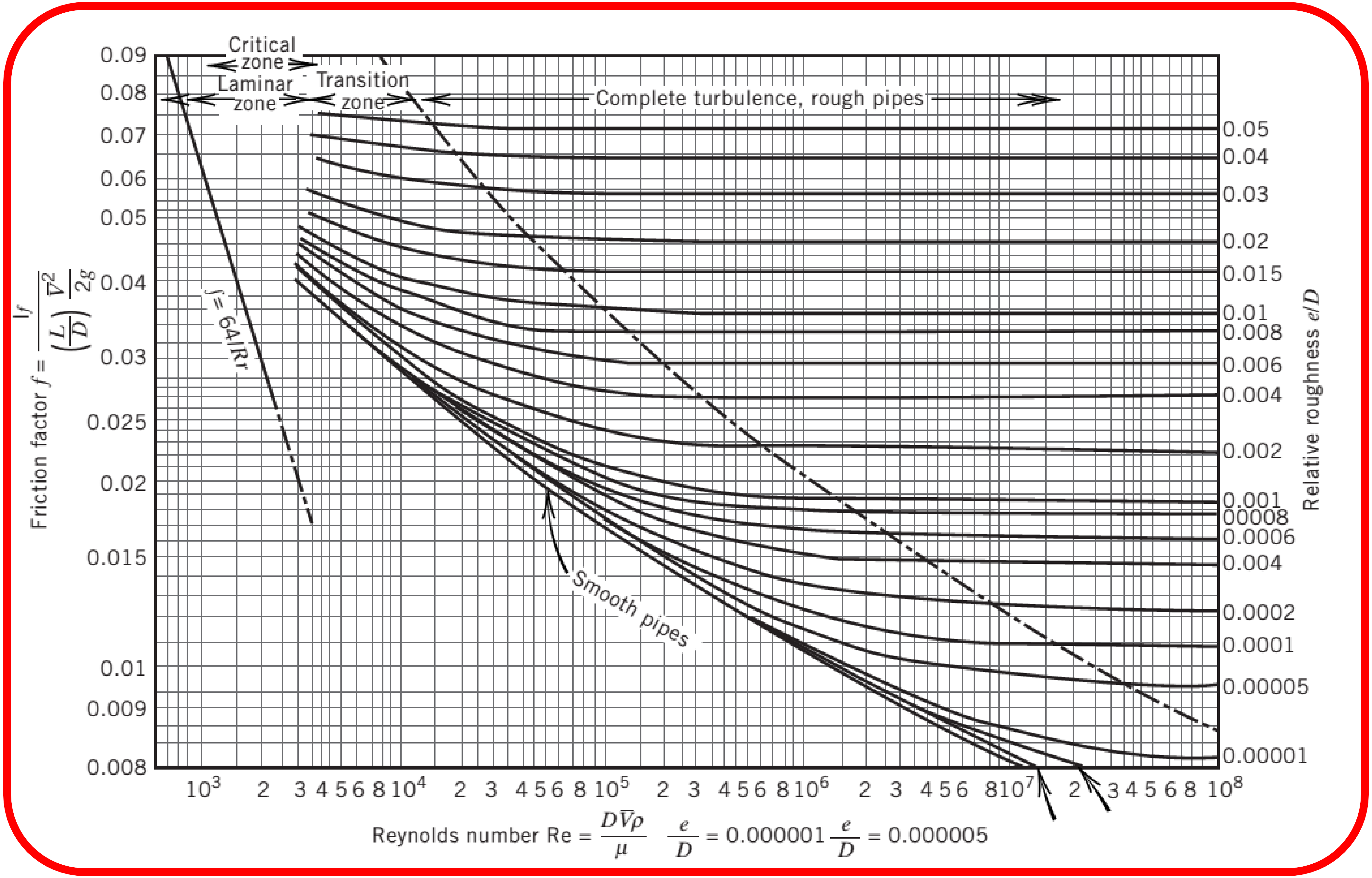
$$D_h = \frac{4(\text{cross-sectional area})}{\text{wetted perimeter}} \qquad (6\text{-}6)$$



Laminar flow
f = 64/Re
Wholly turbulent flow f = f(ε/D)
Laminar flow
Transition range
Smooth
f = 0.316/Re^1/4
f
ε/D
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.025
0.02
0.015
0.01
0.009
0.008
0.05
0.04
0.03
0.02
0.015
0.01
0.008
0.006
0.004
0.002
0.001
0.0008
0.0006
0.0004
0.0002
0.0001
0.00005
0.00001
10³
2(10³)
10⁴
2(10⁴)
10⁵
2(10⁵)
10⁶
2(10⁶)
10⁷
2(10⁷)
4
6
8
Re = ρVD/μ

Lost Head

Figure 6-1, Friction factors for pipe flow correlated by Moody, which is referred to as Moody diagram



Lost Head

- The friction factor is a function of:
  - ✓ Re number and the relative roughness *e/D* of the conduit in the transition zone.
  - ✓ Re number only for laminar flow.
  - ✓ Relative roughness *e/D* of the conduit in the complete turbulence zone.

Table 10-1 Absolute Roughness Values for Some Pipe Materials

Type	Absolute Roughness *e* Feet	Absolute Roughness *e* mm
Commercial Steel	0.00015	0.457
Drawn Tubing or Plastic	0.000005	0.0015
Cast Iron	0.00085	0.2591
Galvanized Iron	0.0005	0.1524
Concrete	0.001	0.3048

❖ Table 6-1, Absolute roughness values for some pipe materials

Lost Head

- At high Reynolds number, the friction factor can be expressed by:

$$\frac{1}{\sqrt{f}} = 1.14 + 2\log(D/e) \quad (6\text{-}7a)$$

Values of the friction factor in the region between smooth pipes and complete turbulence, rough pipes can be expressed by Colebrook's natural roughness function

$$\boxed{\frac{1}{\sqrt{f}} = 1.14 + 2\log(D/e) - 2\log\left[1 + \frac{9.3}{\mathrm{Re}(e/D)\sqrt{f}}\right]}$$

- The Reynolds number is defined as:

$$\mathrm{Re} = \frac{\rho \bar{V} D}{\mu} = \frac{\bar{V} D}{\nu}$$

where:

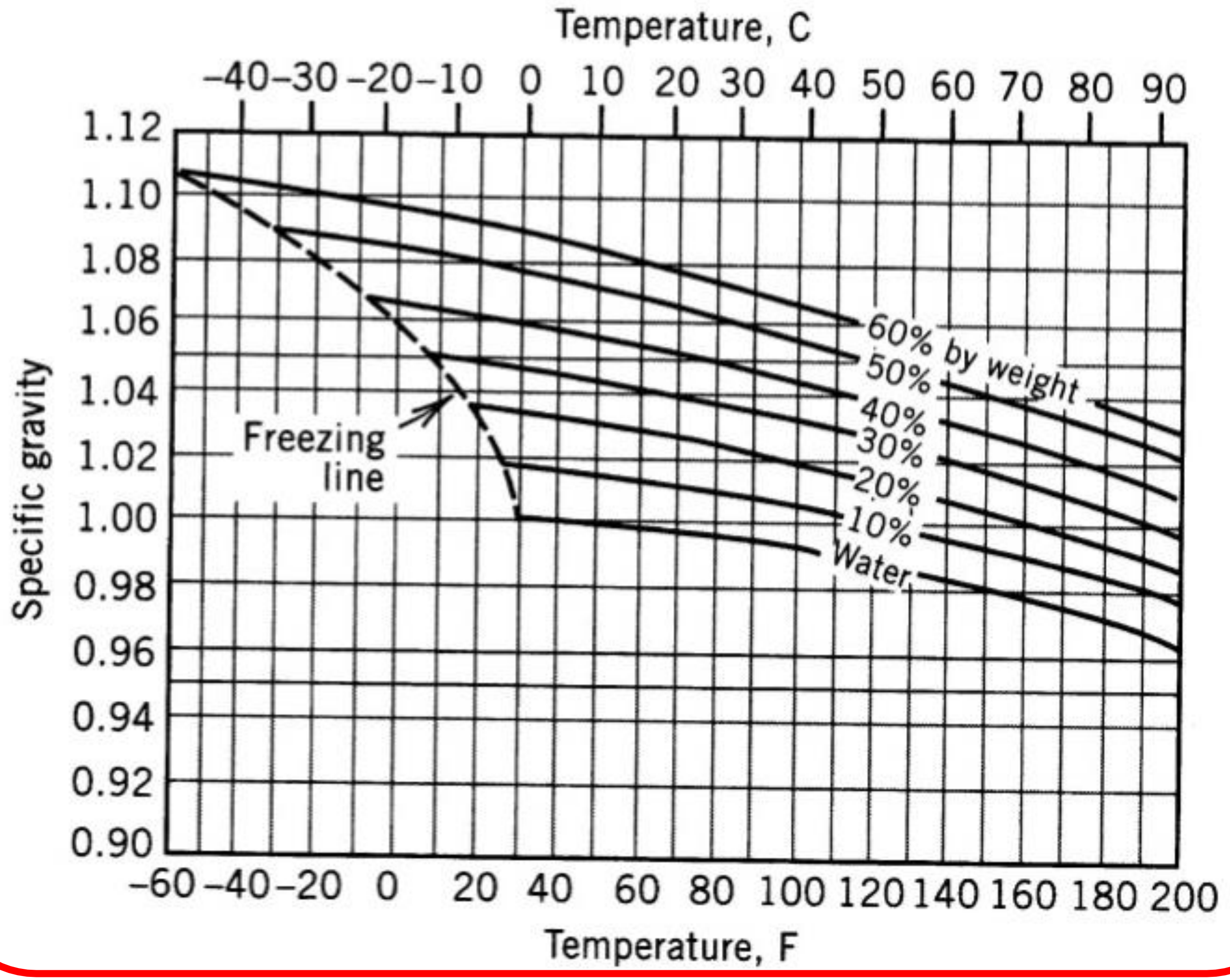
ρ = mass density of the flowing fluid, lbm/ft^3 or kg/m^3
μ = dynamic viscosity, lbm/(ft-sec) or (N-s)/m^2
ν = kinematic viscosity, ft^2/sec or m^2/s

- The hydraulic diameter is used to calculate Re when the conduit is noncircular

Lost Head

To prevent freezing it is often necessary to use a secondary coolant (brine solution), possibly a mixture of ethylene glycol and water. Figure 6-2 gives specific gravity and viscosity data for water and various solutions of ethylene glycol and water.

Note that the viscosity is given in centipoise[1lbm/(ft-sec)=1490 centipoise and 10^3 centipoise=1 (N-s)/m^2].



- **Figure 6-2.** Specific gravity of aqueous ethylene glycol solutions:

Lost Head

- ❑ Figure 6-3,

Viscosity of aqueous ethylene glycol solutions

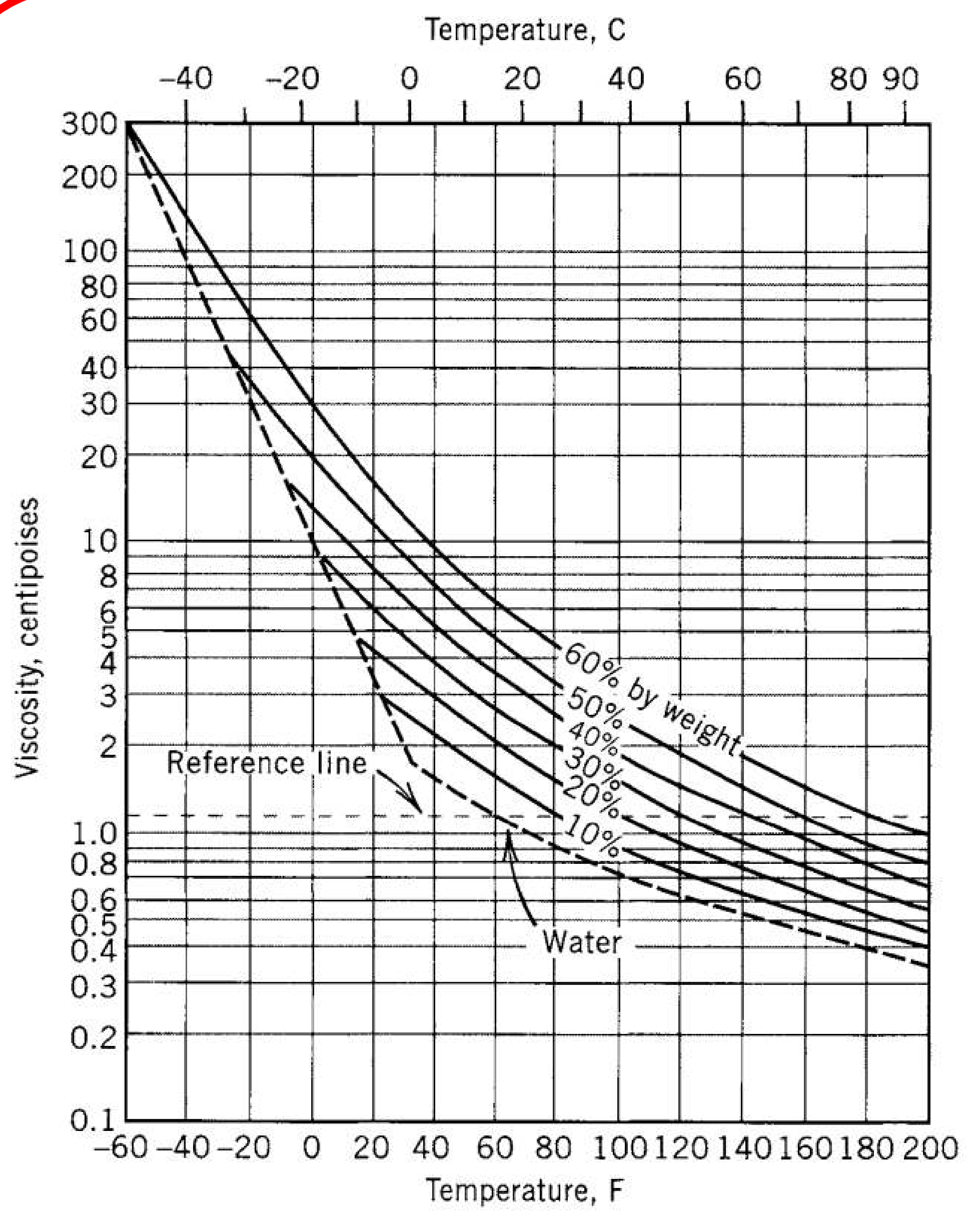


Figure 10-2*b* Viscosity of aqueous ethylene glycol solutions. (Adapted by permission from *ASHRAE Handbook, Fundamentals Volume,* 1989.)

Compare the lost head for water and a 30 percent ethylene glycol solution flowing at the rate of 110 gallons per minute (gpm) in a 3 in. standard (Schedule 40) commercial steel pipe 200 ft in length. Temperature of the water is 50 F.

Equation 6-5 will be used. From Table C-1,inside diameter of 3 in. nominal diameter Schedule 40 pipe D= 3.068 in. and the inside cross-sectional area for flow is A= $0.0513\ ft^2$. The Reynolds number is given by Eq. 6-5, and average velocity in the pipe is

$$\overline{V} = \frac{\dot{Q}}{A} = \frac{110\ \text{gal/min}}{(7.48\ \text{gal/ft}^3)(0.0513\ \text{ft}^2)} = 287\ \text{ft/min} = 4.78\ \text{ft/sec}$$

The absolute viscosity of pure water at 50 F is 1.4 centipoise, or 9.4×10^{-4} lbm/(ft-sec), from Fig. 10-2*b*. Then

$$\text{Re} = \frac{62.4(4.78)\ (3.068/12)}{9.4 \times 10^{-4}} = 8.1 \times 10^4$$

From Fig. 10-1 the absolute roughness *e* is 0.00015 for commercial steel pipe. The relative roughness is then

$$e/D = 12(0.00015/3.068) = 0.00058$$

The flow is in the transition zone, and the friction factor *f* is 0.021 from Fig. 10-1. The lost head for pure water is then computed using Eq. 10-6:

$$l_{fw} = 0.021 \times \frac{200}{3.068/12} \times \frac{(4.78)^2}{2(32.2)} = 5.83\ \text{ft of water}$$

The absolute viscosity of the 30 percent ethylene glycol solution is 3.1 centipoise from Fig. 10-2*b*, and its specific gravity is 1.042 from Fig. 10-2*a*. The Reynolds number for this case is

$$\text{Re} = \frac{1.042(62.4)\ (4.78)\ (3.068/12)}{3.1/1490} = 3.8 \times 10^4$$

and the friction factor is 0.024 from Fig. 10-1. Then

$$l_{fe} = 0.024 \times \frac{200}{3.068/12} \times \frac{(4.78)^2}{2(32.2)} = 6.66\ \text{ft of E.G.S.}$$
$$= 6.94\ \text{ft of water}$$

The increase in lost head with the brine solution is

$$\text{Percent increase} = \frac{100(6.94 - 5.83)}{5.83} = 19\ \text{percent}$$

Table C-1 Steel Pipe Dimensions—English and SI Units

Nominal Pipe Size, in.	Schedule Number	Diameter O.D. in.	Diameter O.D. mm	Diameter I.D. in.	Diameter I.D. mm	Wall Thickness in	Wall Thickness mm	Inside Cross-Sectional Area ft^2	Inside Cross-Sectional Area $10^{-3}m^2$
$\frac{1}{4}$	40	0.540	13.7	0.364	9.25	0.088	2.23	0.00072	0.067
	80			0.302	7.67	0.119	3.02	0.00050	0.046
$\frac{3}{8}$	40	0.675	17.1	0.493	12.5	0.091	2.31	0.00133	0.124
	80			0.423	10.7	0.126	3.20	0.00098	0.091
$\frac{1}{2}$	40	0.840	21.3	0.622	15.8	0.109	2.77	0.00211	0.196
	80			0.546	13.9	0.147	3.73	0.00163	0.151
$\frac{3}{4}$	40	1.050	26.7	0.824	20.9	0.113	2.87	0.00371	0.345
	80			0.742	18.8	0.154	3.91	0.00300	0.279
1	40	1.315	33.4	1.049	26.6	0.133	3.38	0.00600	0.557
	80			0.957	24.3	0.179	4.55	0.00499	0.464
$1\frac{1}{2}$	40	1.900	48.3	1.610	40.9	0.145	3.68	0.01414	1.314
	80			1.500	38.1	0.200	5.08	0.01225	1.138
2	40	2.375	60.3	2.067	52.5	0.154	3.91	0.02330	2.165
	80			1.939	49.3	0.218	5.54	0.02050	1.905
$2\frac{1}{2}$	40	2.875	73.0	2.469	62.7	0.203	5.16	0.03322	3.086
	80			2.323	59.0	0.276	7.01	0.02942	2.733
3	40	3.500	88.9	3.068	77.9	0.216	5.49	0.05130	4.766
	80			2.900	73.7	0.300	7.62	0.04587	4.262
4	40	4.500	114.3	4.026	102.3	0.237	6.02	0.08840	8.213
	80			3.836	97.2	0.337	8.56	0.07986	7.419
5	40	5.563	141.3	5.047	128.1	0.258	6.55	0.1390	12.91
	80			4.813	122.3	0.375	9.53	0.1263	11.73
6	40	6.625	168.3	6.065	154.1	0.280	7.11	0.2006	18.64
	80			5.761	146.3	0.432	11.0	0.1810	16.82
8	40	8.625	219.1	7.981	202.7	0.322	8.18	0.3474	32.28
	80			7.625	193.7	0.500	12.7	0.3171	29.46
10	40	10.75	273.1	10.020	254.5	0.365	9.27	0.5475	50.86
	80			9.750	247.7	0.500	12.7	0.5185	48.17

Source: Adapted from A.S.A. Standards B36.10.

System Characteristics

- The behavior of a piping system may be conveniently represented by plotting **total head versus volume flow rate** as shown in Figure 6-4.

$$H_p = \frac{g_c(P_{01} - P_{02})}{g\rho} + (z_1 - z_2) - l_f \quad \text{6-8)}$$

where H_p represents the total head required to produce the change in static, velocity, and elevation head and to offset the lost head.

- The elevation head is the same regardless of the flow rate.
- **System characteristics** are useful in analyzing complex circuits such as the parallel arrangement of Fig. 6-5.

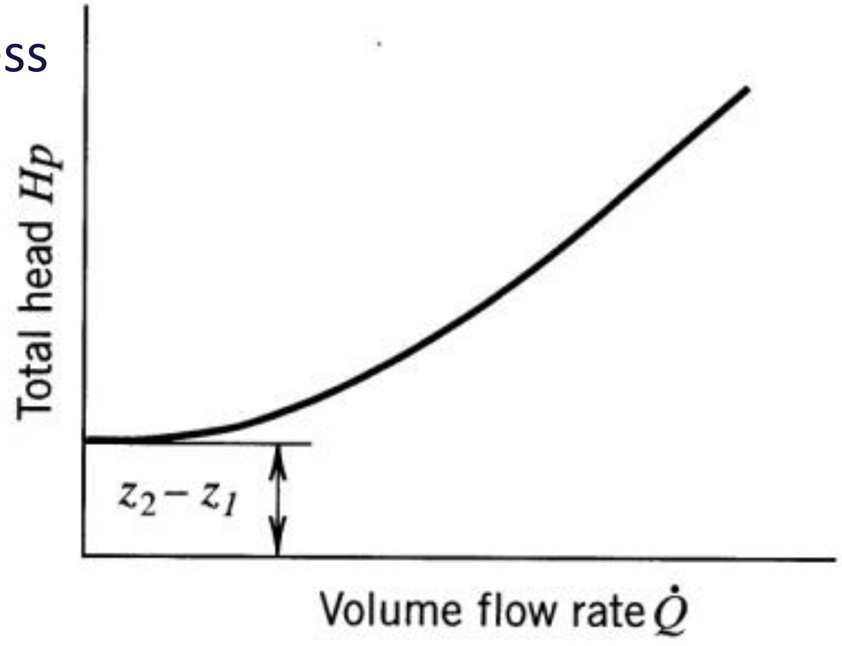
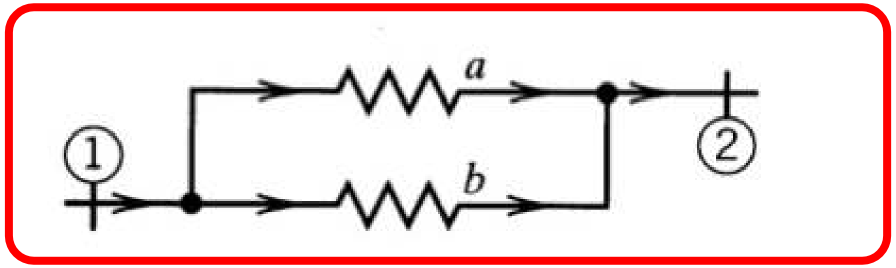


Figure 6-4

System Characteristics

- **Series circuits** as shown in Figure 6-6, have a common flow rate and the total heads are additive
- The total flow rate for the **parallel arrangement (6-7)** is equal to the sum of $\dot{Q}_a$, and $\dot{Q}_b$ where the total head is the same for both circuits.



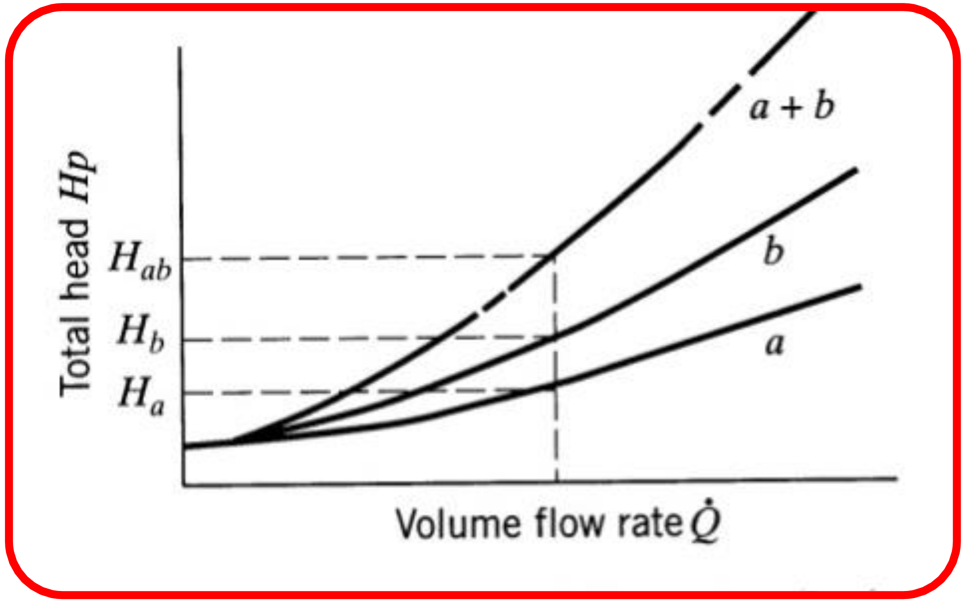


Figure 6-6

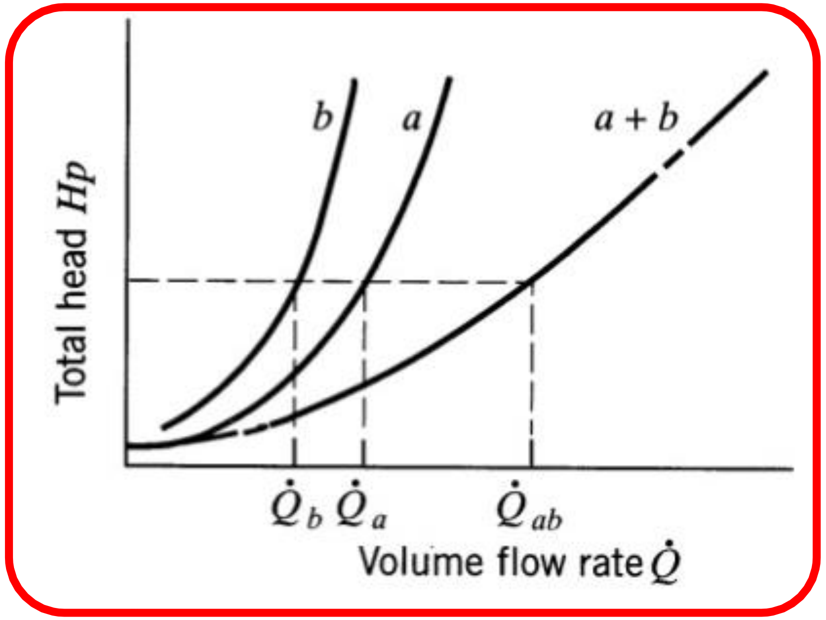


Figure 6-7

Flow Measurements

- **Measurement** of flow rate in piping and duct systems are usually required or indications of **flow rate or velocity may be needed for control purposes**.
- Common devices for making these measurements are the **pitot tube** and the **orifice**, or **venturi meter**.
- The pitot tube which shown in Fig. 6-8 senses both total and static pressure.
- The difference, the velocity pressure, is measured with a manometer or sensed electronically.
- When Eq. 6 -1 is applied to a streamline between the tip of the pitot tube and a point a short distance upstream,

Solving for V_1, gives:

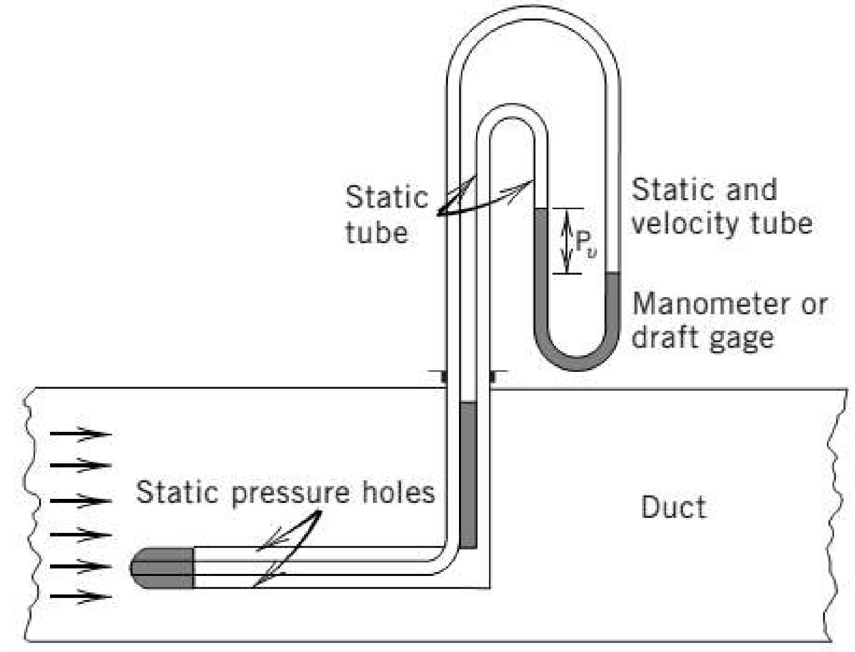


Figure 10-7 Pitot tube in a duct.

Figure(6-8)

$$\frac{P_1}{\rho} + \frac{V_1^2}{2g_c} = \frac{P_2}{\rho} = \frac{P_{02}}{\rho}$$

(6-9)

$$\frac{P_{02} - P_1}{\rho} = \frac{V_1^2}{2g_c} = P_v$$

$$V_1 = \left(2g_c \frac{P_{02} - P_1}{\rho}\right)^{1/2} \quad (6\text{-}10)$$

Flow Measurements (Example 6-2)

A pitot tube is installed in an air duct on the center line. The velocity pressure as indicated by an inclined gage is 0.32 in. of water, the air temperature is 60 F, and barometric pressure is 29.92 in. of Hg. Assuming that fully developed turbulent flow exists where the average velocity is approximately 82 % of the center-line value, compute the volume and mass flow rates for a 10 in. diameter duct.

The mass and volume flow rates are obtained from the average velocity, using Eqs. 6-2 and 6-3. The average velocity is fixed by the center-line velocity in this case, which is computed by using Eq. 10-12. Since the fluid flowing is air, the density term in Eq. 6-10 is that for air, ρ_a. The pressure difference $P_{02} - P_1$ is the measured pressure indicated by the inclined gage as 0.32 in. of water (y). The pressure equivalent of this column of water is given by

$$P_{02} - P_1 = y \frac{g}{g_c} \rho w$$

$$P_{02} - P_1 = \left(\frac{0.32}{12}\right) \text{ft} \left(\frac{32.2}{32.2}\right) \frac{\text{lbf}}{\text{lbmw}} (62.4) \frac{\text{lbmw}}{\text{ft}^3}$$

$$= 1.664 \frac{\text{lbf}}{\text{ft}^2}$$

To get the density of the air we assume an ideal gas:

$$\rho_a = \frac{P_a}{R_a T_a} = \frac{(29.92)\ (0.491)\ (144)}{(53.35)\ (60 + 460)} = 0.076 \frac{\text{lbma}}{\text{ft}^3}$$

which neglects the slight pressurization of the air in the duct. The center-line velocity is given by Eq. 6-10

$$V_{cl} = \left[\frac{(2)\ (32.2)\ (1.644)}{0.076}\right]^{1/2} = 37.6\ \text{ft/sec}$$

and the average velocity is

$$V = 0.82\, V_{cl} = (0.82)\ (37.6) = 30.8\ \text{ft/sec}$$

The mass flow rate is given by Eq. 6 -2 with the area given by

$$A = \frac{\pi}{4}\left(\frac{10}{12}\right)^2 = 0.545\ \text{ft}^2$$

$$\dot{m} = \rho_a \overline{V} A = 0.076\,(30.8)\,0.545 = 1.28\ \text{lbm/sec}$$

The volume flow rate is

$$\dot{Q} = \overline{V} A = 30.8\,(0.545)\,60 = 1007\ \text{ft}^3/\text{min}$$

Centrifugal Pump

- The centrifugal pump is the most commonly used type of pump in HVAC systems.
- **The essential parts** of a centrifugal pump are the **rotating member**, or **impeller**, and the surrounding case.
- The impeller is usually driven by an electric motor, which may be close-coupled (on the same shaft as the impeller) or flexible coupled.
- The fluid enters the center of the rotating impeller, is thrown into the **volute**, and flows outward through the diffuser (Fig. 6-9).
- The fluid **leaving** the impeller has **high kinetic energy that is converted to static pressure in the volute and diffuser**.
- Although there are **various types** of impellers and casings, the principle of operation is the same for all pumps.

Centrifugal Pump

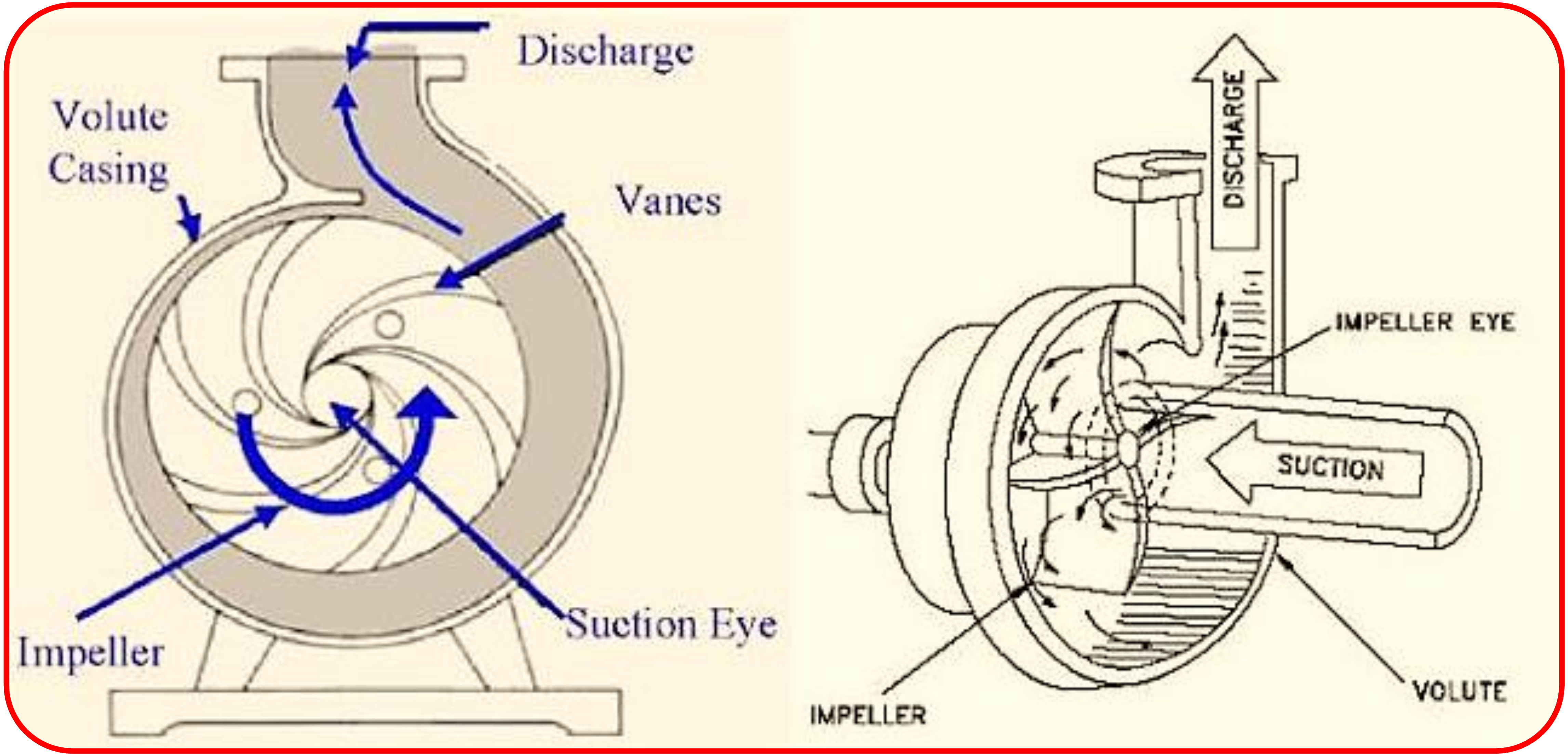
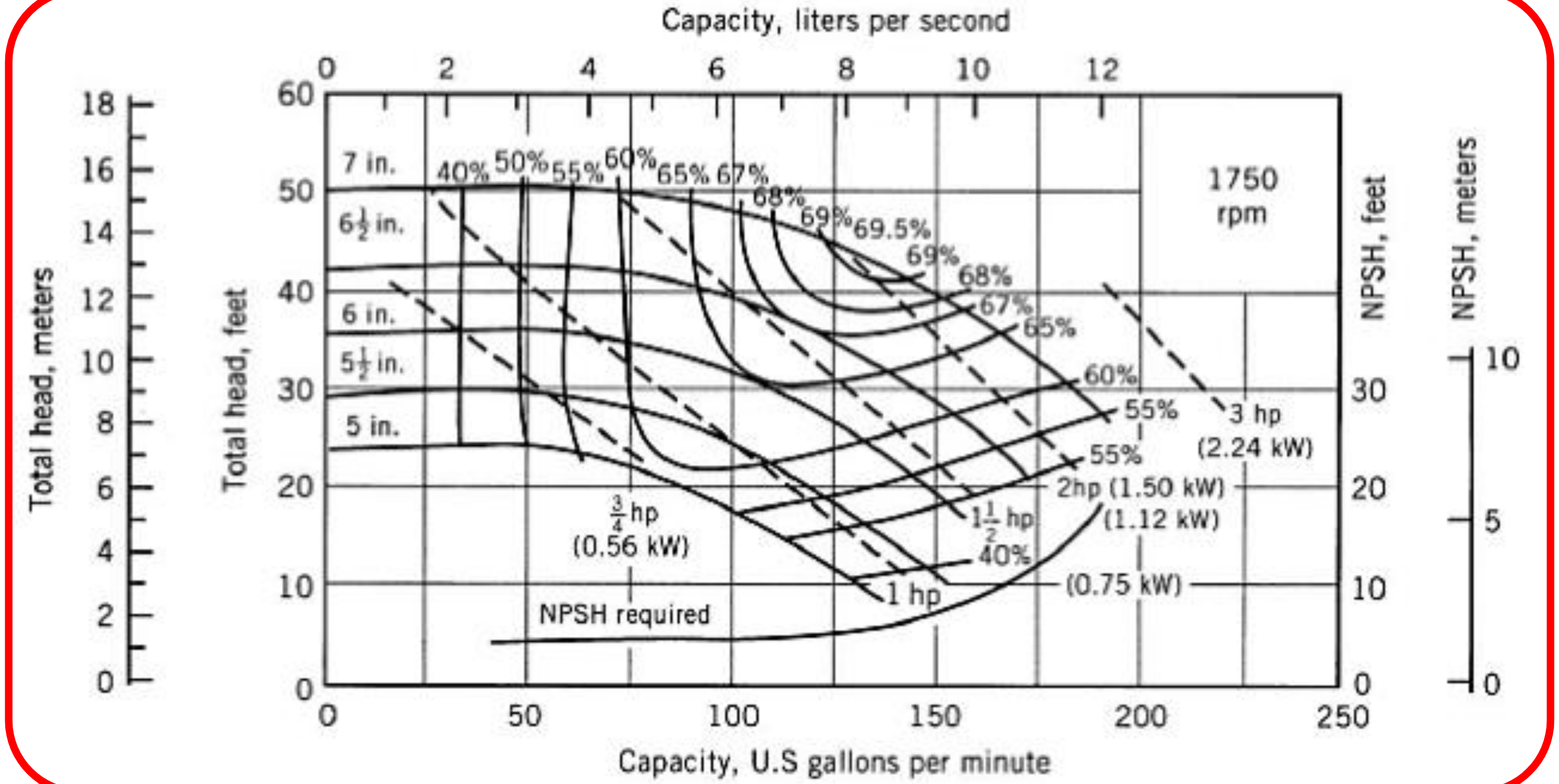


Figure 6-9

Centrifugal Pump

- The pump characteristics curves give the **total dynamic head, efficiency, shaft power, and the net positive suction head as a function of capacity, speed**.
- There are different characteristics curves for different speeds of the impeller



net positive suction head

Centrifugal Pump

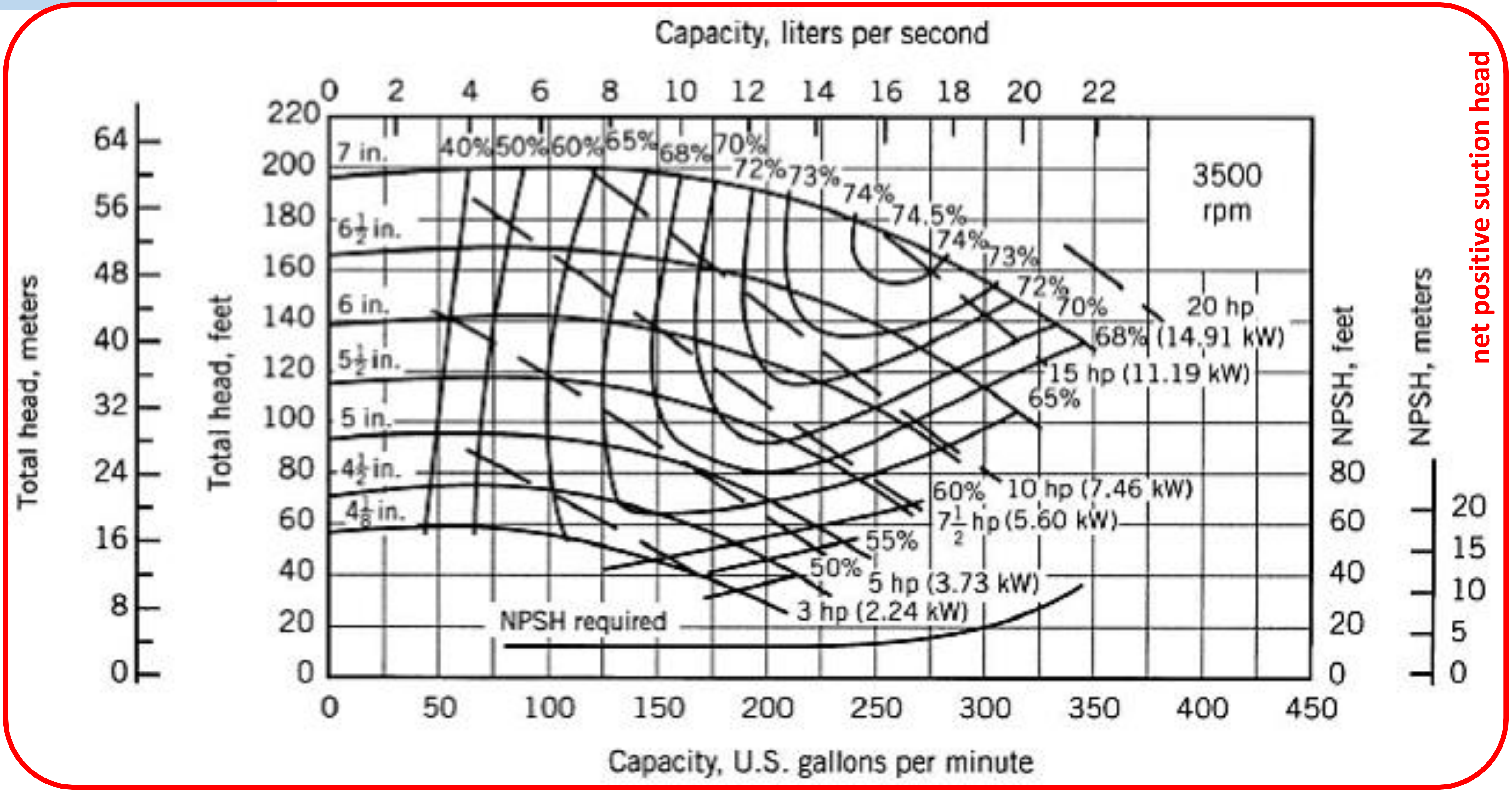


Figure 6-11, Centrifugal pump performance data at 3500 rpm

Centrifugal Pump

- The total dynamic head furnished by a pump can be obtained by:

$$H_p = \frac{wg_c}{g} = \frac{g_c(P_1 - P_2)}{g\rho} + \frac{\overline{V}_1^2 - \overline{V}_2^2}{2g} + (z_1 - z_2) \quad 6\text{-}11)$$

- The elevation head is **zero** or negligible.
- The lost head is unavailable as useful energy and is omitted from the equation.
- Losses are typically accounted for by the efficiency; the ratio of the useful power actually imparted to the fluid to the shaft power input

$$\eta_p = \frac{\dot{W}}{\dot{W}_s} = \frac{\dot{m}w}{\dot{W}_s} = \frac{\rho\dot{Q}w}{\dot{W}_s} \qquad (6\text{-}12)$$

- The Shaft Power then could be obtained as:

$$\dot{W}_s = \frac{\dot{m}w}{\eta_p} = \frac{\rho\dot{Q}w}{\eta_p} = \frac{\rho\dot{Q}H}{\eta_p g_c} \quad .3)$$

Centrifugal Pump

- If the static pressure of the fluid entering a pump approaches the vapor pressure of the liquid, vapor bubbles will form in the impeller passages.
- This condition is detrimental to pump performance, and the collapse of the bubbles is noisy and may damage the pump, this is known as cavitation.
- **The amount of pressure in excess of the vapor pressure required to prevent cavitation (head) is known as the required net positive suction head (NPSHR).**
- Whereas each pump has its own NPSHR, each system has its own **available** net positive suction head (NPSH**A**):

$$\text{NPSHA} = \frac{P_s g_c}{\rho g} + \frac{\overline{V}_s^2}{2g} - \frac{P_v g_c}{\rho g} \qquad (6\text{-}14)$$

$P_s g_c/\rho g$ = static head at the pump inlet, ft or m, absolute

$\overline{V}_s^2/2g$ = velocity head at the pump inlet, ft or m

$P_v g_c/\rho g$ = static vapor pressure head of the liquid at the pumping temperature, ft or m, absolute

- **The net positive suction head available must always be greater than the NPSHR or noise and cavitation will result.**

Centrifugal Pump - Example 6-3

Suppose a pump is installed in a system as shown. The pump is operating at 3500 rpm with the 6 in. impeller and delivering 200 gpm. The suction line is standard 4 in. pipe that has an inside diameter of 4.026 in. Compute the NPSHA, and compare it with the NPSHR.

The water temperature is 60 F. Losses l_f= 5 ft

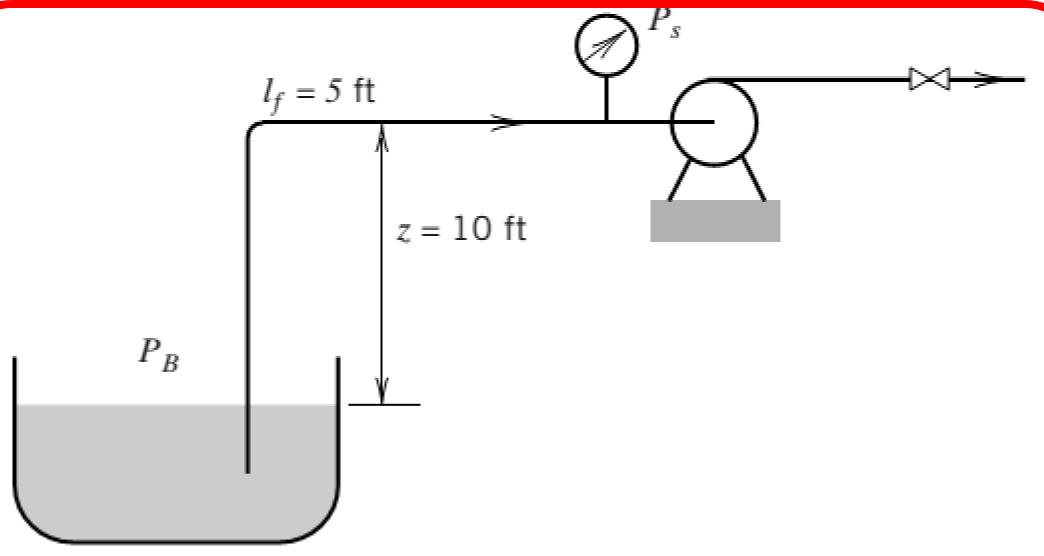


Figure : An open system with suction lift.

NPSHA, available net positive suction head

SOLUTION

From Fig. 6-11 the NPSHR is 10 ft of head. The available net positive suction head is computed from Eq. 6-14 however, the form will be changed slightly through the application of Eq. 6 -1 between the water surface and the pump inlet:

$$\frac{P_B g_c}{\rho g} = \frac{P_s g_c}{\rho g} + \frac{\overline{V}_s^2}{2g} + z_s + l_f$$

or

$$\frac{P_s g_c}{\rho g} + \frac{\overline{V}_s^2}{2g} = \frac{P_B g_c}{\rho g} - z_s - l_f$$

Then Eq. 6-14 becomes

$$\text{NPSHA} = \frac{P_B g_c}{\rho g} - z_s - l_f - \frac{P_v}{\rho}\frac{g_c}{g} \qquad 1$$

Assuming standard barometric pressure,

$$\frac{P_B g_c}{\rho g} = \frac{29.92(13.55)}{12} = 33.78 \text{ ft of water}$$

$$\frac{P_v g_c}{\rho g} = \frac{0.2562(144)}{62.4} = 0.59 \text{ ft of water}$$

where P_v is read from Table A-1*a* at 60 F. Then from Eq. 1

$$\text{NPSHA} = 33.78 - 10 - 5 - 0.59 = 18.19 \text{ ft of water}$$

which is almost twice as large as the NPSHR. However, if the water temperature is increased to 160 F and other factors remain constant, the NPSHA becomes

$$\text{NPSHA} = 33.78 - 10 - 5 - \left(\frac{4.74 \times 144}{61}\right) = 7.6 \text{ ft}$$

and is less than the NSPHR of 10 ft. Cavitation will undoubtedly result.

Piping System Fundamentals

- There are many different types of piping systems used with HVAC components, and there are many specialty items and refinements that make up these systems.

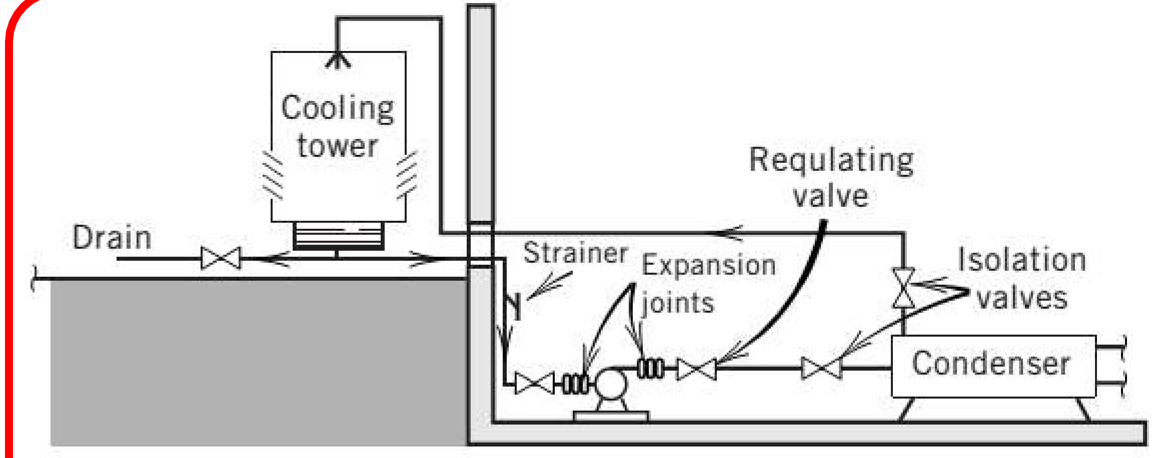


Figure 6-16 A simple open-loop condenser water system.

➢ Basic Open-Loop System

- The open-loop system has at least two points of interface between the water and the atmosphere.
- The cooling tower shows the usual valves, filters, and fittings installed in this type of circuit.
- The isolation valves provide for maintenance without complete drainage of the system, whereas a ball or plug valve should be provided at the pump outlet for adjustment of the flow rate.
- Expansion joints and a rigid base support, to isolate the pump.

Piping System Fundamentals

➢ <u>Basic Closed-Loop System</u>

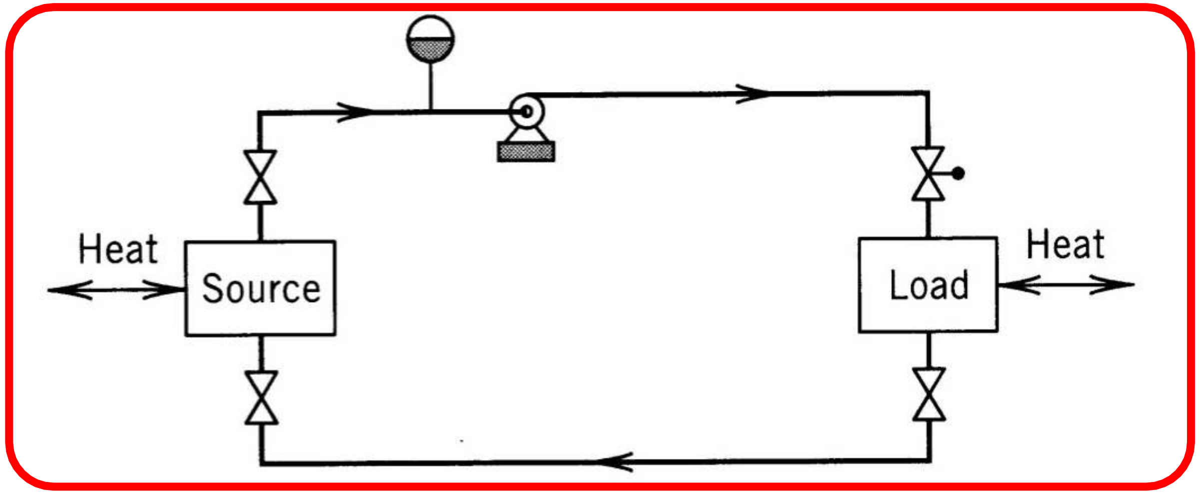


Figure 6-17

- A closed-loop system, shown, has no more than one interface with a compressible gas or flexible surface such as an open or closed expansion tank.
- There is <u>no motivation</u> of flow by static head in a closed system and the entire system is filled with liquid.
- There are two main groups of components<u>: thermal and hydraulic</u>.
- The thermal components are the source, chiller or boiler, the load, cooling or heating coils, and the expansion tank.
- The hydraulic components are the <u>distribution system,pump,and the expansion tank</u>.
- Actual systems will have additional components such as isolation and control valves, flow meters, expansion joints, pump and pipe supports, etc.

Piping System Fundamentals

➢ Pipe Sizing Criteria

- ❑ Piping systems often pass through or near occupied spaces where noise generated by the flowing fluid may be objectionable.
- ❑ A common recommendation sets a velocity limit of 4 ft/sec or 1.2 m/s for pipes 2 in. and smaller.
- ❑ For larger sizes a limit on the head loss of 4 ft per 100 ft of pipe is imposed. This corresponds to about 0.4 kPa/m in SI units.
- ❑ These criteria should not be treated as **hard rules but rather as guides**.
- ❑ Noise is caused by entrained air, locations where abrupt pressure drops occur, and turbulence in general.
- ❑ A reasonable effort to design a balanced system will prevent drastic valve adjustments and will contribute to a quieter system.

Piping System Fundamentals

➢ Pipe Sizing

- The problem of sizing the pipe consists mostly of applying the design criteria discussed earlier.
- Where possible the pipes should be sized so that drastic valve adjustments are not required.
- To facilitate the actual pipe sizing and computation of head loss, charts as those shown in Figs. 6-18 and 19 for pipe and copper tubing were developed.
- These figures are based on **60 F (16 C)** water and give head losses that are about *10 percent high for hot water*.
- Theses figures show that head loss may be obtained directly from the ***flow rate and nominal pipe size or from flow rate and water velocity***.
- When the head loss and flow rate are specified, a pipe size and velocity may be obtained.

Piping System Fundamentals

➢ Pipe Sizing — Figure 6-18

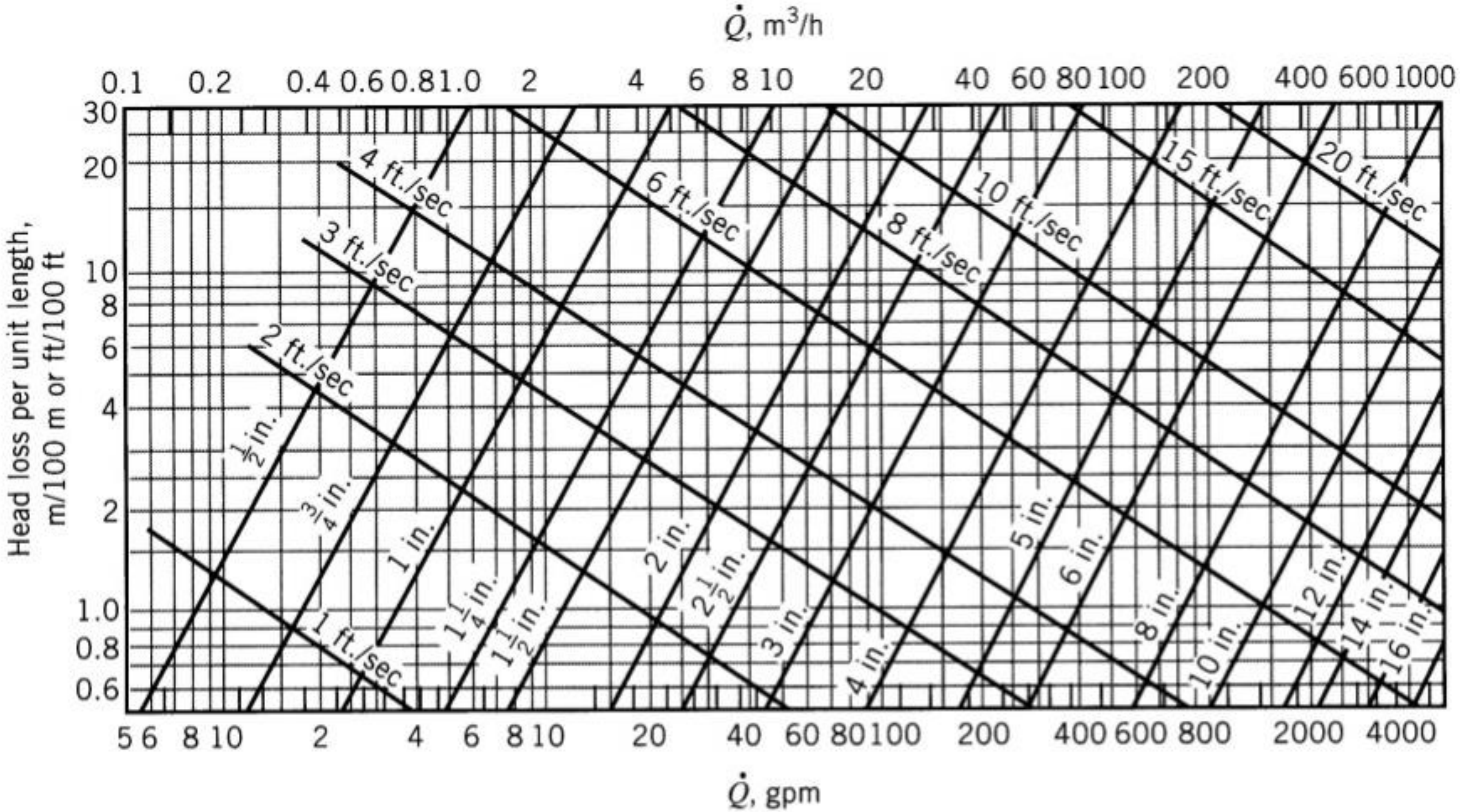


Figure 10-20 Friction loss due to flow of water in commercial steel pipe (schedule 40). (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1989.)

Piping System Fundamentals

➢ Pipe Sizing

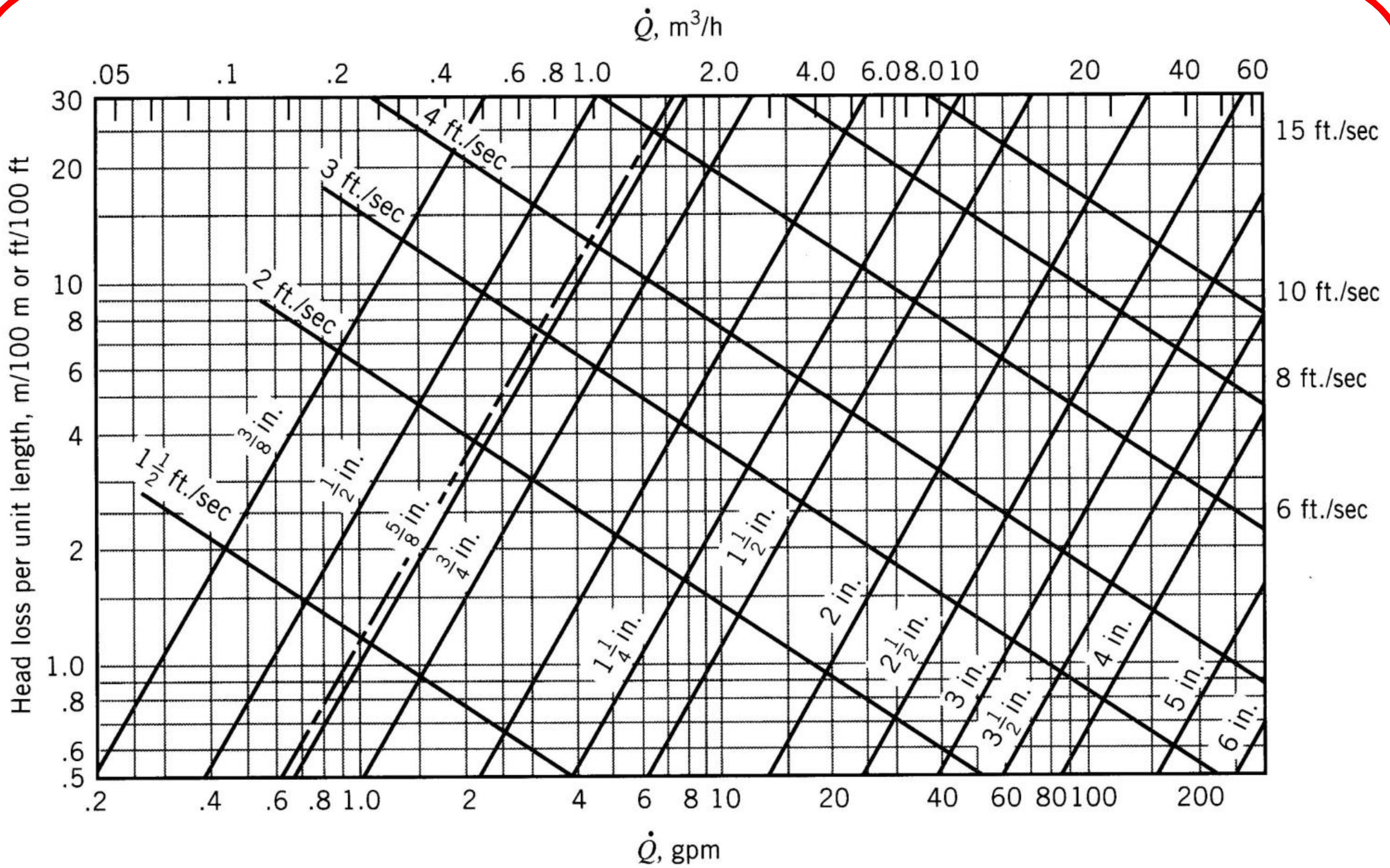


Figure 10-21 Friction loss due to flow of water in type L copper tubing. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1989.)

Figure 6-19

Piping System Fundamentals

➢ Pipe Sizing

- Pipe fittings and valves also introduce losses in **head.**
- These losses are usually allowed for by use of a resistance coefficient *K*, which is the number of velocity heads lost because of the valve or fitting.

$$l_f = K\frac{\overline{V}^2}{2g} \qquad (6\text{-}15)$$

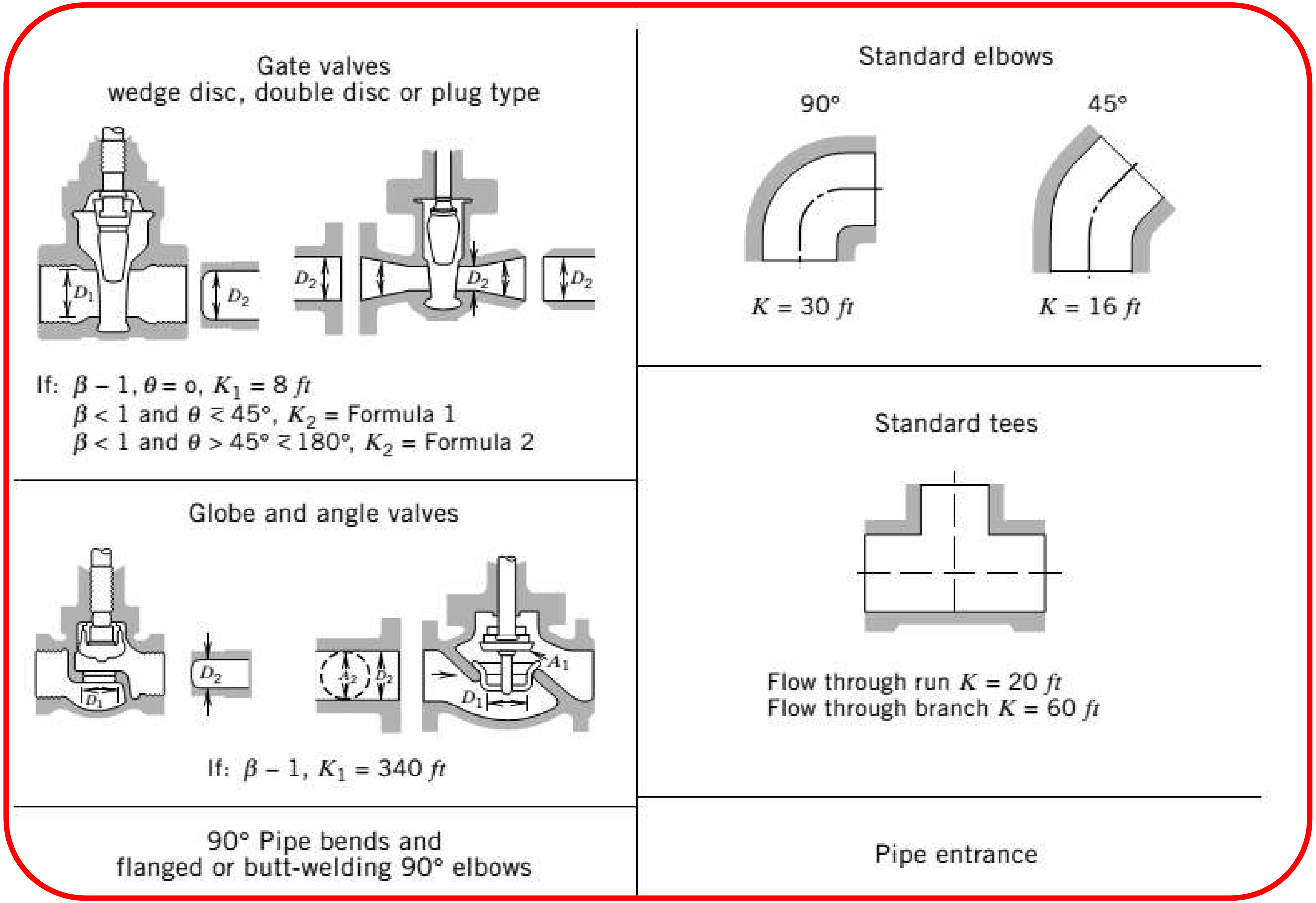
Comparing this definition with Eq. (6-4 b), we get that $K = f\frac{L}{D}$ (6-16)

- The ratio ***L/D is the equivalent length***, in pipe diameters, of straight pipe that will cause the same pressure loss as the valve or fitting under the same flow conditions.
- This is a convenient concept to use when one is computing head loss in a piping system.

Piping System Fundamentals

➢ Pipe Sizing

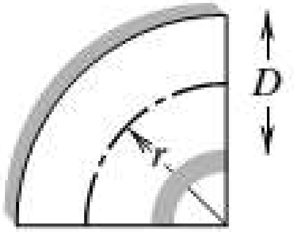
Figure 6-20, Resistance Coefficients *K* for various valves and fittings



Piping System Fundamentals

➢ Pipe Sizing

Figure 6-21, Resistance Coefficients K for various valves and fittings



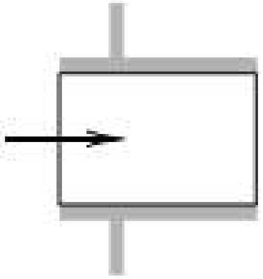
r/D	K	r/D	K
1	$20\ f_t$	10	$30\ f_t$
2	$12\ f_t$	12	$34\ f_t$
3	$12\ f_t$	14	$38\ f_t$
4	$14\ f_t$	16	$42\ f_t$
6	$17\ f_t$	18	$46\ f_t$
8	$24\ f_t$	20	$50\ f_t$

The resistance coefficient K_B for pipe bends other than 90° may be determined as follows:

$$K_B = (n-1)\left(0.25\,\pi\, f_T \frac{r}{D} + 0.5\,K\right) + K$$

n = number of 90° bends
K = resistance coefficient for one 90° bend (per table)

Inward projecting

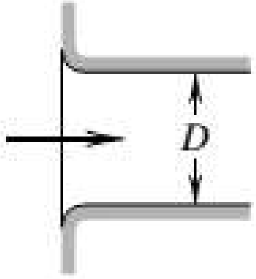


$K = 0.78$

r/D	K
0.00*	0.5
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
0.15 & up	0.04

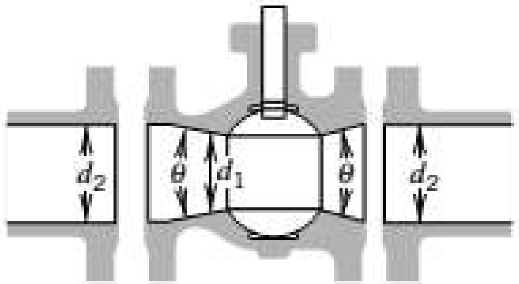
*Sharp-edged

Flush



For K, see table

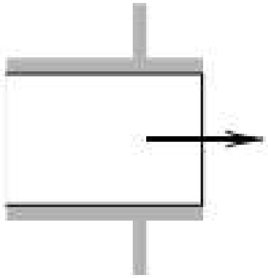
Ball valves



If: $\beta = 1, \theta = o,\ K_1 = 3\, f_T$
$\beta < 1$ and $\theta \lesssim 45°$, K_2 = Formula 1
$\beta < 1$ and $\theta > 45° \lesssim 180°$, K_2 = Formula 2

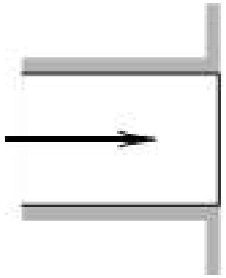
Pipe enxit

Projecting



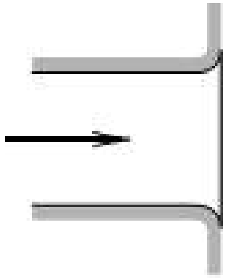
$K = 1.0$

Sharp-edged



$K = 1.0$

Rounded

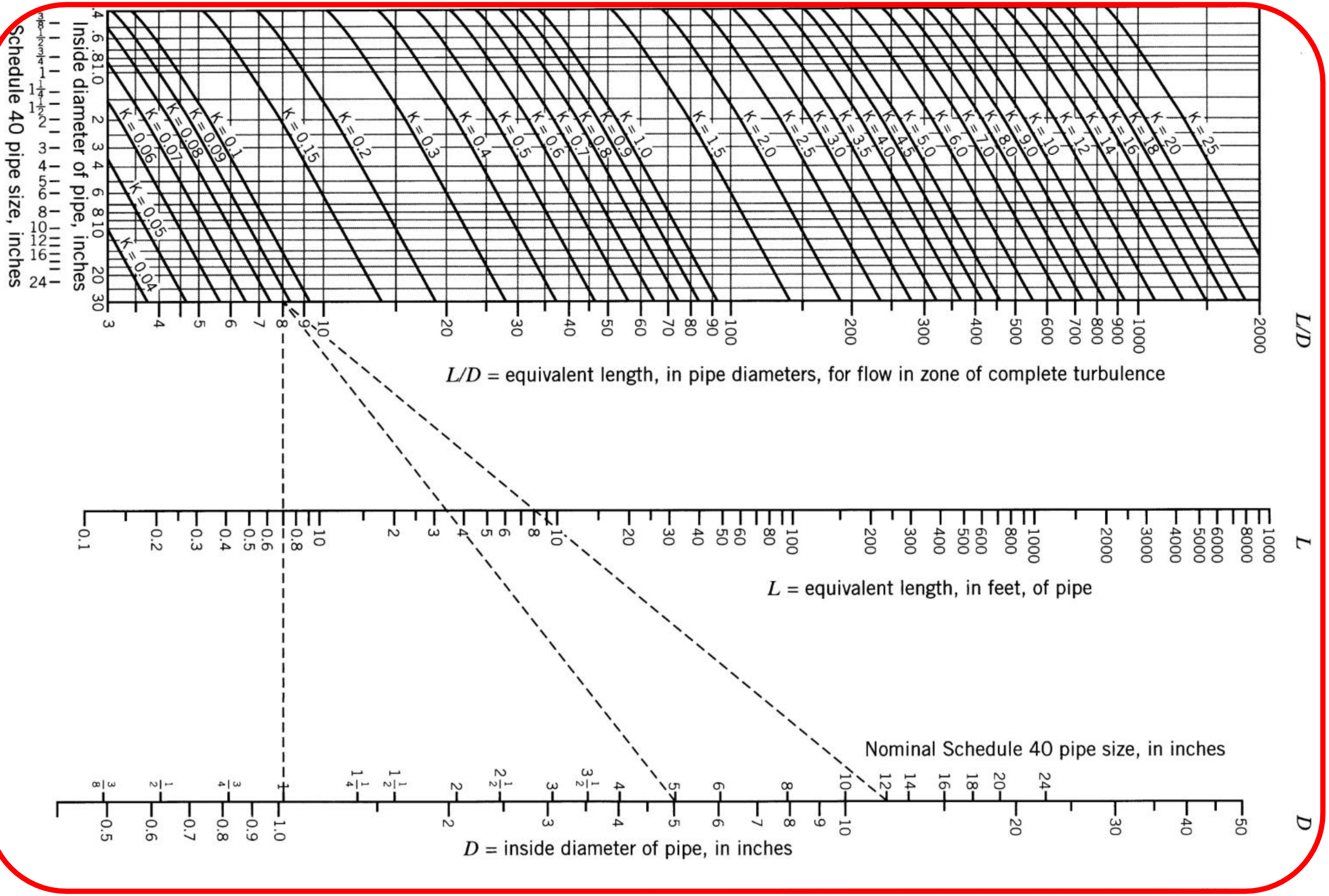


$K = 1.0$

Piping System Fundamentals

➢ **Pipe Sizing**

Figure 6-22, Equivalent *L* and *L/D* and resistance coefficient *K*

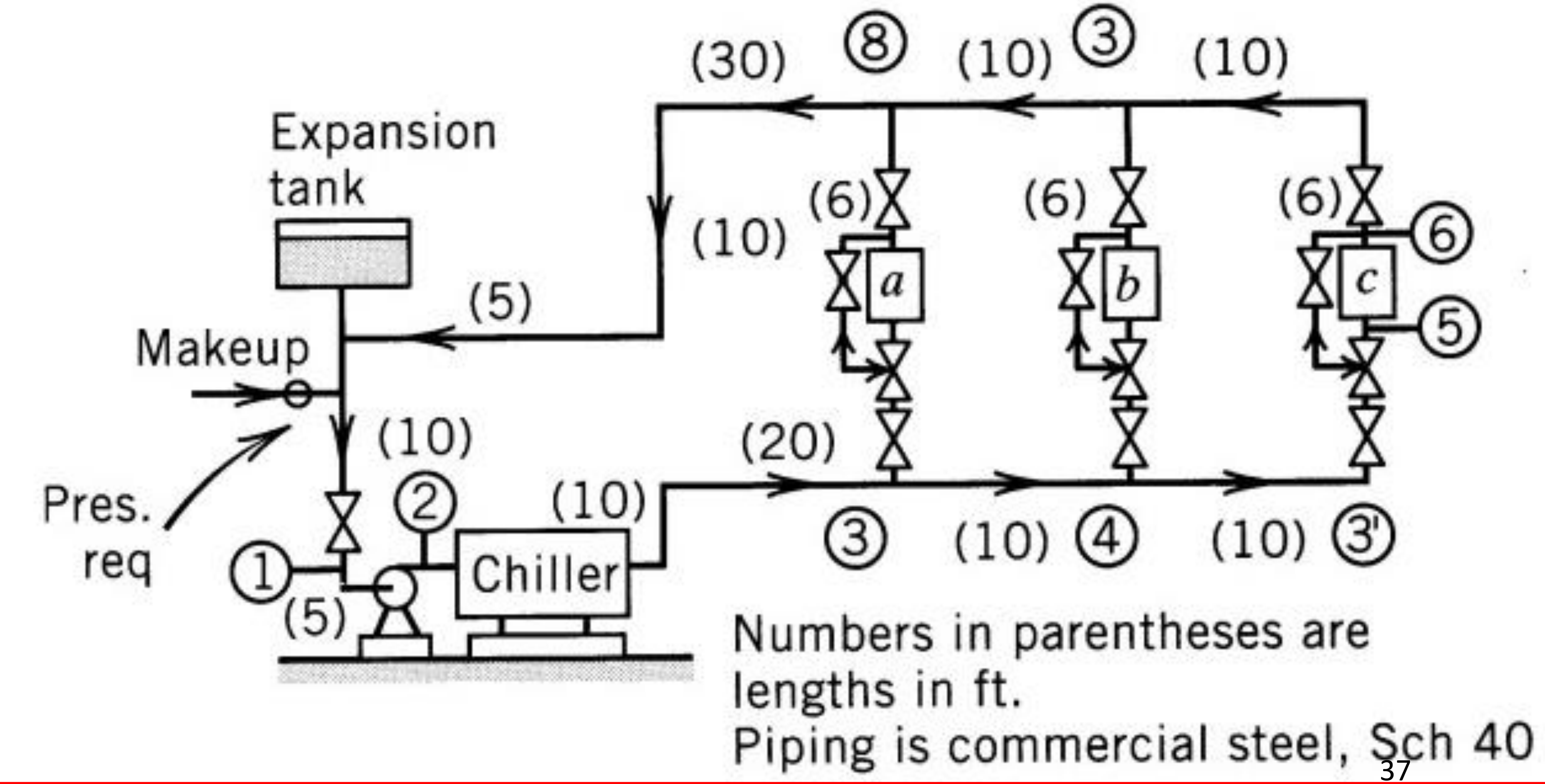


Piping System Fundamentals – Example 10-8

The figure below shows a closed, constant flow <u>two-pipe</u> water system such as might be found in an equipment room. The terminal units *a, b,* and *c* are air-handling units that contain air-to-water finned tube heat exchangers. An actual system could contain a <u>hot water</u> generator or a <u>chiller</u>; a chiller is to be considered here.

- **Size the piping and specify the <u>pumping requirements</u>.**

Unit	$\dot{Q}$ gpm	Lost head ft	C_v, 3-Way Valves
Chiller	60	14	—
a	30	15	25
b	20	10	18
c	10	10	8



Pump Control

The method most frequently used to control pumps is to sense a critical pressure differential some place in the circuit. For example, the path to and from one particular coil in a tertiary circuit will require the greatest pressure differential of all the coils in that circuit. Therefore, the differential pressure sensor for pump speed control should be located across that coil and control valve and set so that the pump will always produce enough head for that coil. Frequently the critical coil is the one located farthest from the pump. The secondary pump system will be controlled in the same general way. In this case, the critical tertiary circuit must be identified and the pressure sensor located accordingly. It may also be necessary to sense flow rate to control pump cycling where two or more pumps operate in parallel.