CHAPTER 33

PIPE SIZING

THIS chapter includes tables and charts to size piping for various fluid flow systems. Further details on specific piping systems can be found in appropriate chapters of the ASHRAE Handbook series.

There are two related but distinct concerns when designing a fluid flow system: sizing the pipe and determining the flow-pressure relationship. The two are often confused because they can use the same equations and design tools. Nevertheless, they should be determined separately.

The emphasis in this chapter is on the problem of sizing the pipe, and to this end design charts and tables for specific fluids are presented in addition to the equations that describe the flow of fluids in pipes. Once a system has been sized, it should be analyzed with more detailed methods of calculation to determine the pump head required to achieve the desired flow. Computerized methods are well suited to handling the details of calculating losses around an extensive system.

PRESSURE DROP EQUATIONS

Darcy-Weisbach Equation

Pressure drop caused by fluid friction in fully developed flows of all "well-behaved" (Newtonian) fluids is described by the Darcy-Weisbach equation:

$$\Delta p = f\left(\frac{L}{D}\right)\left(\frac{\rho}{g_c}\right)\left(\frac{V^2}{2}\right) \tag{1}$$

where

Δp = pressure drop, lb_f/ft^2
f = friction factor, dimensionless (from Moody chart, Figure 13 in Chapter 2)
L = length of pipe, ft
D = internal diameter of pipe, ft
ρ = fluid density at mean temperature, lb_m/ft^3
V = average velocity, fps
g_c = units conversion factor, 32.2 $ft \cdot lb_m/lb_f \cdot s^2$

This equation is often presented in head or specific energy form as

$$\Delta h = \left(\frac{\Delta p}{\rho}\right)\left(\frac{g_c}{g}\right) = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right) \tag{2}$$

where

Δh = head loss, ft
g = acceleration of gravity, ft/s^2

In this form, the density of the fluid does not appear explicitly (although it is in the Reynolds number, which influences f).

The friction factor f is a function of pipe roughness ε, inside diameter D, and parameter Re, the Reynolds number:

$$\mathrm{Re} = DV\rho/\mu \tag{3}$$

where

Re = Reynolds number, dimensionless
ε = absolute roughness of pipe wall, ft
μ = dynamic viscosity of fluid, $lb_m/ft \cdot s$

The friction factor is frequently presented on a Moody chart (Figure 13 in Chapter 2) giving f as a function of Re with ε/D as a parameter.

A useful fit of smooth and rough pipe data for the usual turbulent flow regime is the **Colebrook equation**:

$$\frac{1}{\sqrt{f}} = 1.74 - 2\log\left(\frac{2\varepsilon}{D} + \frac{18.7}{\mathrm{Re}\sqrt{f}}\right) \tag{4}$$

Another form of Equation (4) appears in Chapter 2, but the two are equivalent. Equation (4) is more useful in showing behavior at limiting cases—as ε/D approaches 0 (smooth limit), the $18.7/\mathrm{Re}\sqrt{f}$ term dominates; at high ε/D and Re (fully rough limit), the $2\varepsilon/D$ term dominates.

Equation (4) is implicit in f; that is, f appears on both sides, so a value for f is usually obtained iteratively.

Hazen-Williams Equation

A less widely used alternative to the Darcy-Weisbach formulation for calculating pressure drop is the Hazen-Williams equation, which is expressed as

$$\Delta p = 3.022L\left(\frac{V}{C}\right)^{1.852}\left(\frac{1}{D}\right)^{1.167}\left(\frac{\rho g}{g_c}\right) \tag{5}$$

or

$$\Delta h = 3.022L\left(\frac{V}{C}\right)^{1.852}\left(\frac{1}{D}\right)^{1.167} \tag{6}$$

where C = roughness factor.

Typical values of C are 150 for plastic pipe and copper tubing, 140 for new steel pipe, down to 100 and below for badly corroded or very rough pipe.

Valve and Fitting Losses

Valves and fittings cause pressure losses greater than those caused by the pipe alone. One formulation expresses losses as

$$\Delta p = K\left(\frac{\rho}{g_c}\right)\left(\frac{V^2}{2}\right) \quad \text{or} \quad \Delta h = K\left(\frac{V^2}{2g}\right) \tag{7}$$

where K = geometry- and size-dependent loss coefficient (Tables 1, 2, and 3).

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

Table 1 K Factors—Screwed Pipe Fittings

Nominal Pipe Dia., in.	90° Ell Reg.	90° Ell Long	45° Ell	Return Bend	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve	Bell Mouth Inlet	Square Inlet	Projected Inlet
3/8	2.5	—	0.38	2.5	0.90	2.7	20	0.40	—	8.0	0.05	0.5	1.0
1/2	2.1	—	0.37	2.1	0.90	2.4	14	0.33	—	5.5	0.05	0.5	1.0
3/4	1.7	0.92	0.35	1.7	0.90	2.1	10	0.28	6.1	3.7	0.05	0.5	1.0
1	1.5	0.78	0.34	1.5	0.90	1.8	9	0.24	4.6	3.0	0.05	0.5	1.0
1-1/4	1.3	0.65	0.33	1.3	0.90	1.7	8.5	0.22	3.6	2.7	0.05	0.5	1.0
1-1/2	1.2	0.54	0.32	1.2	0.90	1.6	8	0.19	2.9	2.5	0.05	0.5	1.0
2	1.0	0.42	0.31	1.0	0.90	1.4	7	0.17	2.1	2.3	0.05	0.5	1.0
2-1/2	0.85	0.35	0.30	0.85	0.90	1.3	6.5	0.16	1.6	2.2	0.05	0.5	1.0
3	0.80	0.31	0.29	0.80	0.90	1.2	6	0.14	1.3	2.1	0.05	0.5	1.0
4	0.70	0.24	0.28	0.70	0.90	1.1	5.7	0.12	1.0	2.0	0.05	0.5	1.0

Source: *Engineering Data Book* (HI 1979).

Table 2 K Factors—Flanged Welded Pipe Fittings

Nominal Pipe Dia., in.	90° Ell Reg.	90° Ell Long	45° Ell Long	Return Bend Reg.	Return Bend Long	Tee-Line	Tee-Branch	Glove Valve	Gate Valve	Angle Valve	Swing Check Valve
1	0.43	0.41	0.22	0.43	0.43	0.26	1.0	13	—	4.8	2.0
1-1/4	0.41	0.37	0.22	0.41	0.38	0.25	0.95	12	—	3.7	2.0
1-1/2	0.40	0.35	0.21	0.40	0.35	0.23	0.90	10	—	3.0	2.0
2	0.38	0.30	0.20	0.38	0.30	0.20	0.84	9	0.34	2.5	2.0
2-1/2	0.35	0.28	0.19	0.35	0.27	0.18	0.79	8	0.27	2.3	2.0
3	0.34	0.25	0.18	0.34	0.25	0.17	0.76	7	0.22	2.2	2.0
4	0.31	0.22	0.18	0.31	0.22	0.15	0.70	6.5	0.16	2.1	2.0
6	0.29	0.18	0.17	0.29	0.18	0.12	0.62	6	0.10	2.1	2.0
8	0.27	0.16	0.17	0.27	0.15	0.10	0.58	5.7	0.08	2.1	2.0
10	0.25	0.14	0.16	0.25	0.14	0.09	0.53	5.7	0.06	2.1	2.0
12	0.24	0.13	0.16	0.24	0.13	0.08	0.50	5.7	0.05	2.1	2.0

Source: *Engineering Data Book* (HI 1979).

Table 3 Approximate Range of Variation for K Factors

90° Elbow	Regular screwed	±20% above 2 in.	Tee	Screwed, line or branch	±25%
		±40% below 2 in.		Flanged, line or branch	±35%
	Long-radius screwed	±25%	Globe valve	Screwed	±25%
	Regular flanged	±35%		Flanged	±25%
	Long-radius flanged	±30%	Gate valve	Screwed	±25%
45° Elbow	Regular screwed	±10%		Flanged	±50%
	Long-radius flanged	±10%	Angle valve	Screwed	±20%
Return bend (180°)	Regular screwed	±25%		Flanged	±50%
	Regular flanged	±35%	Check valve	Screwed	±50%
	Long-radius flanged	±30%		Flanged	+200% –80%

Source: *Engineering Data Book* (HI 1979).

Example 1. Determine the pressure drop for 60°F water flowing at 4 fps through a nominal 1 in., 90° screwed ell.

Solution: From Table 1, the K for a 1 in., 90° screwed ell is 1.5.

$$\Delta p = 1.5 \times 62.4/32.2 \times 4^2/2 = 23.3 \text{ lb/ft}^2 \text{ or } 0.16 \text{ psi}$$

The loss coefficient for valves appears in another form as C_v, a dimensional coefficient expressing the flow through a valve at a specified pressure drop.

$$Q = C_v \sqrt{\Delta p} \tag{8}$$

where

Q = volumetric flow, gpm
C_v = valve coefficient, gpm at Δp = 1 psi
Δp = pressure drop, psi

See the section on Control Valve Sizing in Chapter 42 of the 2000 *ASHRAE Handbook—Systems and Equipment* for a more complete explanation of C_v.

Example 2. Determine the volumetric flow through a valve with C_v = 10 for an allowable pressure drop of 5 psi.

Solution: $Q = 10\sqrt{5} = 22.4$ gpm.

Alternative formulations express fitting losses in terms of equivalent lengths of straight pipe (Tables 4 and 5, Figure 4). Pressure loss data for fittings are also presented in Idelchik (1986).

Calculating Pressure Losses

The most common engineering design flow loss calculation selects a pipe size for the desired total flow rate and available or allowable pressure drop.

Because either formulation of fitting losses requires a known diameter, pipe size must be selected before calculating the detailed influence of fittings. A frequently used rule of thumb assumes that the design length of pipe is 50 to 100% longer than actual to account for fitting losses. After a pipe diameter has been selected on this basis, the influence of each fitting can be evaluated.

WATER PIPING

FLOW RATE LIMITATIONS

Stewart and Dona (1987) surveyed the literature relating to water flow rate limitations. This section briefly reviews some of their findings. Noise, erosion, and installation and operating costs all limit the maximum and minimum velocities in piping systems. If piping sizes are too small, noise levels, erosion levels, and pumping costs can be unfavorable; if piping sizes are too large, installation costs are excessive. Therefore, pipe sizes are chosen to minimize initial cost while avoiding the undesirable effects of high velocities.

A variety of upper limits of water velocity and/or pressure drop in piping and piping systems is used. One recommendation places a velocity limit of 4 fps for 2 in. pipe and smaller, and a pressure drop limit of 4 ft of water/100 ft for piping over 2 in. Other guidelines are based on the type of service (Table 4) or the annual operating hours (Table 5). These limitations are imposed either to control the levels of pipe and valve noise, erosion, and water hammer pressure or for economic reasons. Carrier (1960) recommends that the velocity not exceed 15 fps in any case.

Noise Generation

Velocity-dependent noise in piping and piping systems results from any or all of four sources: turbulence, cavitation, release of entrained air, and water hammer. In investigations of flow-related noise, Marseille (1965), Ball and Webster (1976), and Rogers (1953, 1954, 1956) reported that velocities on the order of 10 to 17 fps lie within the range of allowable noise levels for residential and commercial buildings. The experiments showed considerable variation in the noise levels obtained for a specified velocity. Generally, systems with longer pipe and with more numerous fittings and valves were noisier. In addition, sound measurements were taken under widely differing conditions; for example, some tests used plastic-covered pipe, while others did not. Thus, no detailed correlations relating sound level to flow velocity in generalized systems are available.

Table 4 Water Velocities Based on Type of Service

Type of Service	Velocity, fps	Reference
General service	4 to 10	a, b, c
City water	3 to 7	a, b
	2 to 5	c
Boiler feed	6 to 15	a, c
Pump suction and drain lines	4 to 7	a, b

[a]Crane Co. (1976). [b]Carrier (1960). [c]Grinnell Company (1951).

Table 5 Maximum Water Velocity to Minimize Erosion

Normal Operation, h/yr	Water Velocity, fps
1500	15
2000	14
3000	13
4000	12
6000	10

Source: Carrier (1960).

The noise generated by fluid flow in a pipe system increases sharply if cavitation or the release of entrained air occurs. Usually the combination of a high water velocity with a change in flow direction or a decrease in the cross section of a pipe causing a sudden pressure drop is necessary to cause cavitation. Ball and Webster (1976) found that at their maximum velocity of 42 fps, cavitation did not occur in straight 3/8 and 1/2 in. pipe; using the apparatus with two elbows, cold water velocities up to 21 fps caused no cavitation. Cavitation did occur in orifices of 1:8 area ratio (orifice flow area is one-eighth of pipe flow area) at 5 fps and in 1:4 area ratio orifices at 10 fps (Rogers 1954).

Some data are available for predicting hydrodynamic (liquid) noise generated by control valves. The International Society for Measurement and Control compiled prediction correlations in an effort to develop control valves for reduced noise levels (ISA 1985). The correlation to predict hydrodynamic noise from control valves is

$$SL = 10 \log C_v + 20 \log \Delta p - 30 \log t + 5 \tag{9}$$

where

SL = sound level, dB
C_v = valve coefficient, gpm/(psi)$^{0.5}$
Q = flow rate, gpm
Δp = pressure drop across valve, psi
t = downstream pipe wall thickness, in.

Air entrained in water usually has a higher partial pressure than the water. Even when flow rates are small enough to avoid cavitation, the release of entrained air may create noise. Every effort should be made to vent the piping system or otherwise remove entrained air.

Erosion

Erosion in piping systems is caused by water bubbles, sand, or other solid matter impinging on the inner surface of the pipe. Generally, at velocities lower than 100 fps, erosion is not significant as long as there is no cavitation. When solid matter is entrained in the fluid at high velocities, erosion occurs rapidly, especially in bends. Thus, high velocities should not be used in systems where sand or other solids are present or where slurries are transported.

Allowances for Aging

With age, the internal surfaces of pipes become increasingly rough, which reduces the available flow with a fixed pressure supply. However, designing with excessive age allowances may result in oversized piping. Age-related decreases in capacity depend on the type of water, type of pipe material, temperature of water, and type of system (open or closed) and include

- Sliming (biological growth or deposited soil on the pipe walls), which occurs mainly in unchlorinated, raw water systems.
- Caking of calcareous salts, which occurs in hard water (i.e., water bearing calcium salts) and increases with water temperature.
- Corrosion (incrustations of ferrous and ferric hydroxide on the pipe walls), which occurs in metal pipe in soft water. Because oxygen is necessary for corrosion to take place, significantly more corrosion takes place in open systems.

Allowances for expected decreases in capacity are sometimes treated as a specific amount (percentage). Dawson and Bowman (1933) added an allowance of 15% friction loss to new pipe (equivalent to an 8% decrease in capacity). The *HDR Design Guide* (1981) increased the friction loss by 15 to 20% for closed piping systems and 75 to 90% for open systems. Carrier (1960) indicates a factor of approximately 1.75 between friction factors for closed and open systems.

Obrecht and Pourbaix (1967) differentiated between the corrosive potential of different metals in potable water systems and concluded that iron is the most severely attacked, then galvanized steel, lead, copper, and finally copper alloys (i.e., brass). Hunter (1941) and Freeman (1941) showed the same trend. After four years of cold and hot water use, copper pipe had a capacity loss of 25 to 65%. Aged ferrous pipe has a capacity loss of 40 to 80%. Smith (1983) recommended increasing the design discharge by 1.55 for uncoated cast iron, 1.08 for iron and steel, and 1.06 for cement or concrete.

The Plastic Pipe Institute (1971) found that corrosion is not a problem in plastic pipe; the capacity of plastic pipe in Europe and the United States remains essentially the same after 30 years in use.

Extensive age-related flow data are available for use with the Hazen-Williams empirical equation. Difficulties arise in its application, however, because the original Hazen-Williams roughness coefficients are valid only for the specific pipe diameters, water velocities, and water viscosities used in the original experiments. Thus, when the *C*s are extended to different diameters, velocities, and/or water viscosities, errors of up to about 50% in pipe capacity can occur (Williams and Hazen 1933, Sanks 1978).

Water Hammer

When any moving fluid (not just water) is abruptly stopped, as when a valve closes suddenly, large pressures can develop. While detailed analysis requires knowledge of the elastic properties of the pipe and the flow-time history, the limiting case of rigid pipe and instantaneous closure is simple to calculate. Under these conditions,

$$\Delta p_h = \rho c_s V / g_c \tag{10}$$

where

Δp_h = pressure rise caused by water hammer, lb_f/ft^2
ρ = fluid density, lb_m/ft^3
c_s = velocity of sound in fluid, fps
V = fluid flow velocity, fps

The c_s for water is 4720 fps, although the elasticity of the pipe reduces the effective value.

Example 3. What is the maximum pressure rise if water flowing at 10 fps is stopped instantaneously?

Solution: $\Delta p_h = 62.4 \times 4720 \times 10/32.2 = 91{,}468 \text{ lb/ft}^2$
$= 635 \text{ psi}$

Other Considerations

Not discussed in detail in this chapter, but of potentially great importance, are a number of physical and chemical considerations: pipe and fitting design, materials, and joining methods must be appropriate for working pressures and temperatures encountered, as well as being suitably resistant to chemical attack by the fluid.

Other Piping Materials and Fluids

For fluids not included in this chapter or for piping materials of different dimensions, manufacturers' literature frequently supplies pressure drop charts. The Darcy-Weisbach equation, with the Moody chart or the Colebrook equation, can be used as an alternative to pressure drop charts or tables.

HYDRONIC SYSTEM PIPING

The Darcy-Weisbach equation with friction factors from the Moody chart or Colebrook equation (or, alternatively, the Hazen-Williams equation) is fundamental to calculating pressure drop in hot and chilled water piping; however, charts calculated from these equations (such as Figures 1, 2, and 3) provide easy determination of pressure drops for specific fluids and pipe standards. In addition, tables of pressure drops can be found in HI (1979) and Crane Co. (1976).

The Reynolds numbers represented on the charts in Figures 1, 2, and 3 are all in the turbulent flow regime. For smaller pipes and/or lower velocities, the Reynolds number may fall into the laminar regime, in which the Colebrook friction factors are no longer valid.

Most tables and charts for water are calculated for properties at 60°F. Using these for hot water introduces some error, although the answers are conservative (i.e., cold water calculations overstate the pressure drop for hot water). Using 60°F water charts for 200°F water should not result in errors in Δp exceeding 20%.

Range of Usage of Pressure Drop Charts

General Design Range. The general range of pipe friction loss used for design of hydronic systems is between 1 and 4 ft of water per 100 ft of pipe. A value of 2.5 ft/100 ft represents the mean to which most systems are designed. Wider ranges may be used in specific designs if certain precautions are taken.

Piping Noise. Closed-loop hydronic system piping is generally sized below certain arbitrary upper limits, such as a velocity limit of 4 fps for 2 in. pipe and under, and a pressure drop limit of 4 ft per 100 ft for piping over 2 in. in diameter. Velocities in excess of 4 fps can be used in piping of larger size. This limitation is generally accepted, although it is based on relatively inconclusive experience with noise in piping. Water *velocity noise* is not caused by water but by free air, sharp pressure drops, turbulence, or a combination of these, which in turn cause cavitation or flashing of water into steam. Therefore, higher velocities may be used if proper precautions are taken to eliminate air and turbulence.

Air Separation

Air in hydronic systems is usually undesirable because it causes flow noise, allows oxygen to react with piping materials, and sometimes even prevents flow in parts of a system. Air may enter a system at an air-water interface in an open system or in an expansion tank in a closed system, or it may be brought in dissolved in makeup water. Most hydronic systems use air separation devices to remove air. The solubility of air in water increases with pressure and decreases with temperature; thus, separation of air from water is best achieved at the point of lowest pressure and/or highest temperature in a system. For more information, see Chapter 12, Hydronic Heating and Cooling System Design, of the 2000 *ASHRAE Handbook—Systems and Equipment*.

In the absence of venting, air can be entrained in the water and carried to separation units at flow velocities of 1.5 to 2 fps or more in pipe 2 in. and under. Minimum velocities of 2 fps are therefore recommended. For pipe sizes 2 in. and over, minimum velocities corresponding to a head loss of 0.75 ft/100 ft are normally used. Maintenance of minimum velocities is particularly important in the upper floors of high-rise buildings where the air tends to come out of solution because of reduced pressures. Higher velocities should be used in *downcomer* return mains feeding into air separation units located in the basement.

Example 4. Determine the pipe size for a circuit requiring 20 gpm flow.

Solution: Enter Figure 1 at 20 gpm, read up to pipe size within normal design range (1 to 4 ft/100 ft), and select 1-1/2 in. Velocity is 3.1 fps, which is between 2 and 4. Pressure loss is 2.9 ft/100 ft.

Valve and Fitting Pressure Drop

Valves and fittings can be listed in elbow equivalents, with an elbow being equivalent to a length of straight pipe. Table 6 lists equivalent lengths of 90° elbows; Table 7 lists elbow equivalents for valves and fittings for iron and copper.

Example 5. Determine equivalent feet of pipe for a 4 in. open gate valve at a flow velocity of approximately 4 fps.

Solution: From Table 6, at 4 fps, each elbow is equivalent to 10.6 ft of 4 in. pipe. From Table 7, the gate valve is equivalent to 0.5 elbows. The actual equivalent pipe length (added to measured circuit length for pressure drop determination) will be 10.6 × 0.5, or 5.3 equivalent feet of 4 in. pipe.

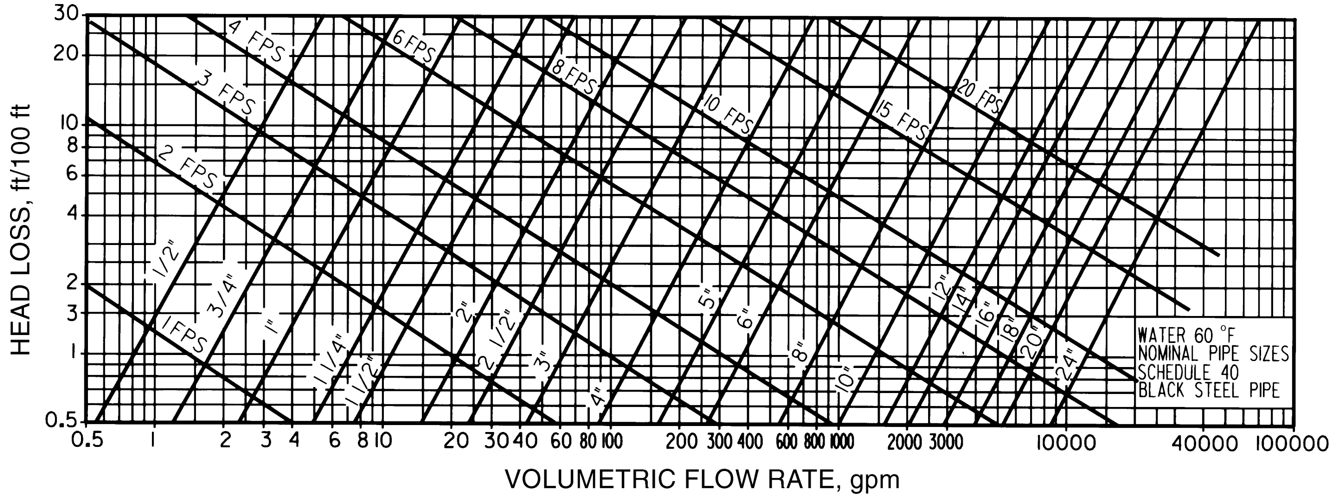


Fig. 1 Friction Loss for Water in Commercial Steel Pipe (Schedule 40)

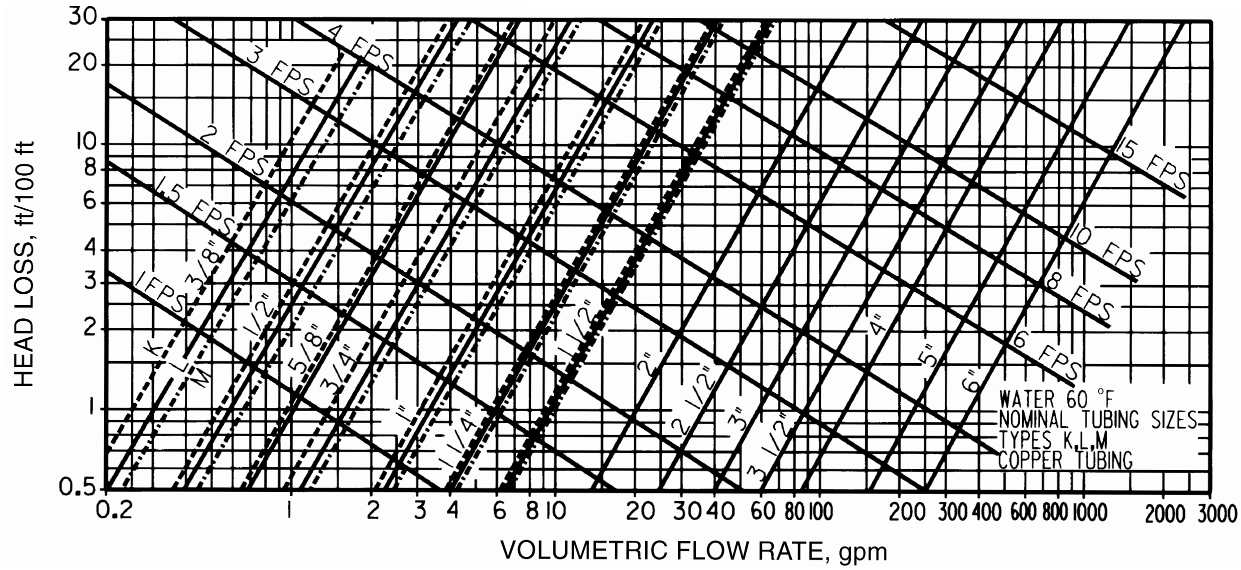


Fig. 2 Friction Loss for Water in Copper Tubing (Types K, L, M)

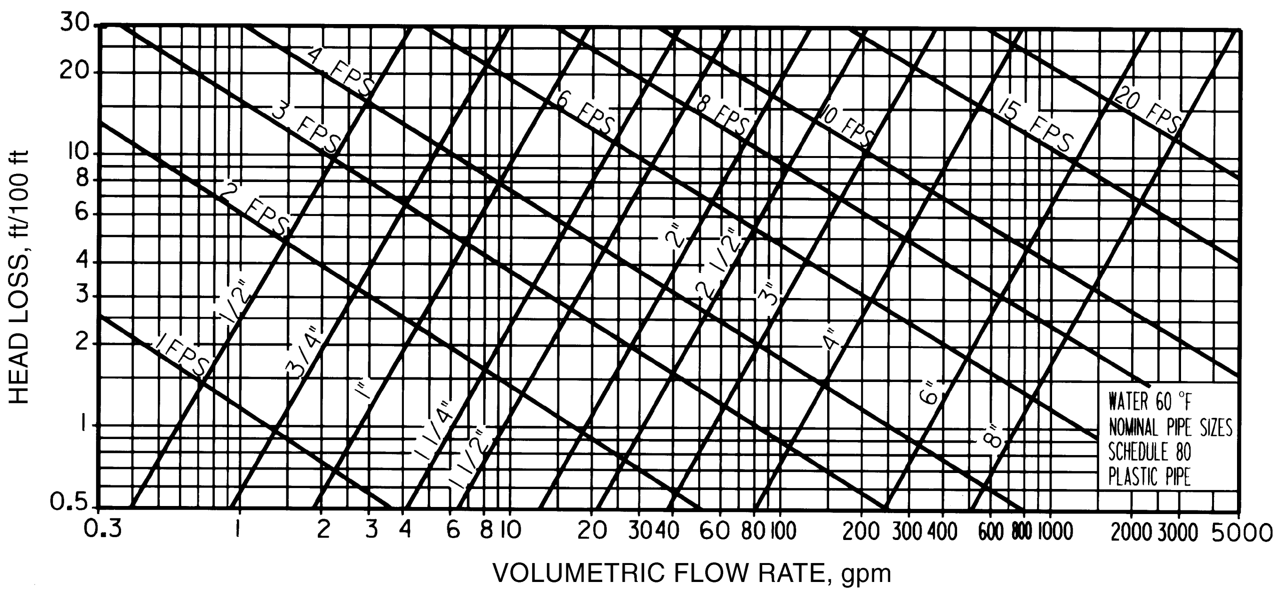


Fig. 3 Friction Loss for Water in Plastic Pipe (Schedule 80)

Table 6 Equivalent Length in Feet of Pipe for 90° Elbows

Veloc-ity, fps	Pipe Size														
	1/2	3/4	1	1-1/4	1-1/2	2	2-1/2	3	3-1/2	4	5	6	8	10	12
1	1.2	1.7	2.2	3.0	3.5	4.5	5.4	6.7	7.7	8.6	10.5	12.2	15.4	18.7	22.2
2	1.4	1.9	2.5	3.3	3.9	5.1	6.0	7.5	8.6	9.5	11.7	13.7	17.3	20.8	24.8
3	1.5	2.0	2.7	3.6	4.2	5.4	6.4	8.0	9.2	10.2	12.5	14.6	18.4	22.3	26.5
4	1.5	2.1	2.8	3.7	4.4	5.6	6.7	8.3	9.6	10.6	13.1	15.2	19.2	23.2	27.6
5	1.6	2.2	2.9	3.9	4.5	5.9	7.0	8.7	10.0	11.1	13.6	15.8	19.8	24.2	28.8
6	1.7	2.3	3.0	4.0	4.7	6.0	7.2	8.9	10.3	11.4	14.0	16.3	20.5	24.9	29.6
7	1.7	2.3	3.0	4.1	4.8	6.2	7.4	9.1	10.5	11.7	14.3	16.7	21.0	25.5	30.3
8	1.7	2.4	3.1	4.2	4.9	6.3	7.5	9.3	10.8	11.9	14.6	17.1	21.5	26.1	31.0
9	1.8	2.4	3.2	4.3	5.0	6.4	7.7	9.5	11.0	12.2	14.9	17.4	21.9	26.6	31.6
10	1.8	2.5	3.2	4.3	5.1	6.5	7.8	9.7	11.2	12.4	15.2	17.7	22.2	27.0	32.0

Table 7 Iron and Copper Elbow Equivalents[a]

Fitting	Iron Pipe	Copper Tubing
Elbow, 90°	1.0	1.0
Elbow, 45°	0.7	0.7
Elbow, 90° long turn	0.5	0.5
Elbow, welded, 90°	0.5	0.5
Reduced coupling	0.4	0.4
Open return bend	1.0	1.0
Angle radiator valve	2.0	3.0
Radiator or convector	3.0	4.0
Boiler or heater	3.0	4.0
Open gate valve	0.5	0.7
Open globe valve	12.0	17.0

Source: Giesecke (1926) and Giesecke and Badgett (1931, 1932a).
[a]See Table 6 for equivalent length of one elbow.

Tee Fitting Pressure Drop. Pressure drop through pipe tees varies with flow through the branch. Figure 4 illustrates pressure drops for nominal 1 in. tees of equal inlet and outlet sizes and for the flow patterns illustrated. Idelchik (1986) also presents data for threaded tees.

Different investigators present tee loss data in different forms, and it is sometimes difficult to reconcile results from several sources. As an estimate of the upper limit to tee losses, a pressure or head loss coefficient of 1.0 may be assumed for entering and leaving flows (i.e., $\Delta p = 1.0\rho V_{in}^2/2 + 1.0\rho V_{out}^2/2$).

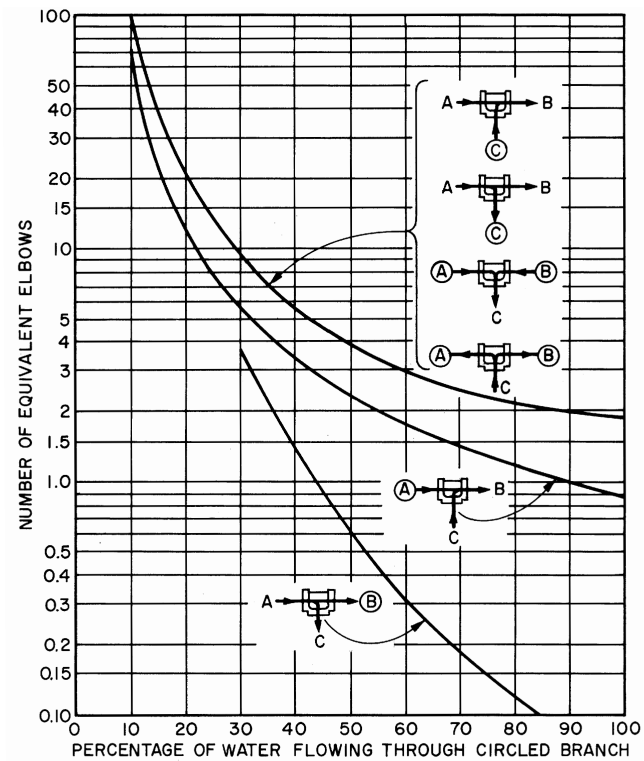
Example 6. Determine the pressure or head losses for a 1 in. (all openings) threaded pipe tee flowing 25% to the side branch, 75% through. The entering flow is 10 gpm (3.71 fps).

Solution: From Figure 4, bottom curve, the number of equivalent elbows for the through-flow is 0.15 elbows; the through-flow is 7.5 gpm (2.78 fps); and the pressure loss is based on the exit flow rate. Table 6 gives the equivalent length of a 1 in. elbow at 3 fps as 2.7 ft. Using Equations (1) and (2) with friction factor $f = 0.0290$ and diameter $D = 0.0874$ ft,

$$\Delta p = (0.15)(0.0290)(2.7/0.0874)(62.4/32.2)(2.78^2/2)$$
$$= 1.01 \text{ lb/ft}^2 = 0.00699 \text{ psi pressure drop, or}$$
$$\Delta h = (0.15)(0.0290)(2.7/0.0874)(2.78^2)/[(2)(32.2)]$$
$$= 0.0161 \text{ ft head loss}$$

From Figure 4, top curve, the number of equivalent elbows for the branch flow of 25% is 13 elbows; the branch flow is 2.5 gpm (0.93 fps); and the pressure loss is based on the exit flow rate. Table 6 gives the equivalent of a 1 in. elbow at 1 fps as 2.2 ft. Using Equations (1) and (2) with friction factor $f = 0.0350$ and diameter = 0.0874 ft,

$$\Delta p = (13)(0.0350)(2.2/0.0874)(62.4/32.2)(0.93^2/2)$$
$$= 9.60 \text{ lb/ft}^2 = 0.0667 \text{ psi pressure drop, or}$$
$$\Delta h = (13)(0.0350)(2.2/0.0874)(0.93^2)/[(2)(32.2)]$$
$$= 0.154 \text{ ft head loss}$$



Notes: 1. Chart is based on straight tees (i.e., branches A, B, and C are the same size).
2. Pressure loss in desired circuit is obtained by selecting the proper curve according to illustrations, determining the flow at the circled branch, and multiplying the pressure loss for the same size elbow at the flow rate in the circled branch by the equivalent elbows indicated.
3. When the size of an outlet is reduced, the equivalent elbows shown in the chart do not apply. Therefore, the maximum loss for any circuit for any flow will not exceed 2 elbow equivalents at the maximum flow occurring in any branch of the tee.
4. Top curve is average of 4 curves, one for each circuit shown.

Fig. 4 Elbow Equivalents of Tees at Various Flow Conditions
(Giesecke and Badgett 1931, 1932b)

SERVICE WATER PIPING

Sizing of service water piping differs from sizing of process lines in that design flows in service water piping are determined by the probability of simultaneous operation of a multiplicity of individual loads such as water closets, urinals, lavatories, sinks, and showers. The full flow characteristics of each load device are readily obtained from manufacturers; however, service water piping sized to handle all load devices simultaneously would be seriously oversized. Thus, a major issue in sizing service water piping is to determine the diversity of the loads.

The procedure shown in this chapter uses the work of R.B. Hunter for estimating diversity (Hunter 1940, 1941). The present-day plumbing designer is usually constrained by building or plumbing codes, which specify the individual and collective loads to be used for pipe sizing. Frequently used codes (including the BOCA *National Plumbing Code*, *Standard Plumbing Code*, *Uniform Plumbing Code*, and *National Standard Plumbing Code*) contain procedures quite similar to those shown here. The designer must be aware of the applicable code for the location being considered.

Federal mandates are forcing plumbing fixture manufacturers to reduce design flows to many types of fixtures, but these may not yet be included in locally adopted codes. Also, the designer must be aware of special considerations; for example, toilet usage at sports arenas will probably have much less diversity than the codes allow and thus may require larger supply piping than the minimum specified by the codes.

Table 8 gives the rate of flow desirable for many common fixtures and the average pressure necessary to give this rate of flow. The pressure varies with fixture design.

In estimating the load, the rate of flow is frequently computed in **fixture units**, which are relative indicators of flow. Table 9 gives the demand weights in terms of fixture units for different plumbing fixtures under several conditions of service, and Figure 5 gives the estimated demand in gallons per minute corresponding to any total number of fixture units. Figures 6 and 7 provide more accurate estimates at the lower end of the scale.

The estimated demand load for fixtures used intermittently on any supply pipe can be obtained by multiplying the number of each kind of fixture supplied through that pipe by its weight from Table 9, adding the products, and then referring to the appropriate curve of Figure 5, 6, or 7 to find the demand corresponding to the total fixture units. In using this method, note that the demand for fixture or supply outlets other than those listed in the table of fixture units is not yet included in the estimate. The demands for outlets (e.g., hose connections and air-conditioning apparatus) that are likely to impose continuous demand during heavy use of the weighted fixtures should be estimated separately and added to demand for fixtures used intermittently to estimate total demand.

The Hunter curves in Figures 5, 6, and 7 are based on use patterns in residential buildings and can be erroneous for other usages such as sports arenas. Williams (1976) discusses the Hunter assumptions and presents an analysis using alternative assumptions.

Table 8 Proper Flow and Pressure Required During Flow for Different Fixtures

Fixture	Flow Pressure, psig[a]	Flow, gpm
Ordinary basin faucet	8	3.0
Self-closing basin faucet	12	2.5
Sink faucet—3/8 in.	10	4.5
Sink faucet—1/2 in.	5	4.5
Dishwasher	15-25	—[b]
Bathtub faucet	5	6.0
Laundry tube cock—1/4 in.	5	5.0
Shower	12	3-10
Ball cock for closet	15	3.0
Flush valve for closet	10-20	15-40[c]
Flush valve for urinal	15	15.0
Garden hose, 50 ft, and sill cock	30	5.0

[a]Flow pressure is the pressure in the pipe at the entrance to the particular fixture considered.
[b]Varies; see manufacturers' data.
[c]Wide range due to variation in design and type of flush valve closets.

Table 9 Demand Weights of Fixtures in Fixture Units[a]

Fixture or Group[b]	Occupancy	Type of Supply Control	Weight in Fixture Units[c]
Water closet	Public	Flush valve	10
Water closet	Public	Flush tank	5
Pedestal urinal	Public	Flush valve	10
Stall or wall urinal	Public	Flush valve	5
Stall or wall urinal	Public	Flush tank	3
Lavatory	Public	Faucet	2
Bathtub	Public	Faucet	4
Shower head	Public	Mixing valve	4
Service sink	Office, etc.	Faucet	3
Kitchen sink	Hotel or restaurant	Faucet	4
Water closet	Private	Flush valve	6
Water closet	Private	Flush tank	3
Lavatory	Private	Faucet	1
Bathtub	Private	Faucet	2
Shower head	Private	Mixing valve	2
Bathroom group	Private	Flush valve for closet	8
Bathroom group	Private	Flush tank for closet	6
Separate shower	Private	Mixing valve	2
Kitchen sink	Private	Faucet	2
Laundry trays (1 to 3)	Private	Faucet	3
Combination fixture	Private	Faucet	3

Source: Hunter (1941).
[a]For supply outlets likely to impose continuous demands, estimate continuous supply separately, and add to total demand for fixtures.
[b]For fixtures not listed, weights may be assumed by comparing the fixture to a listed one using water in similar quantities and at similar rates.
[c]The given weights are for total demand. For fixtures with both hot and cold water supplies, the weights for maximum separate demands can be assumed to be 75% of the listed demand for the supply.

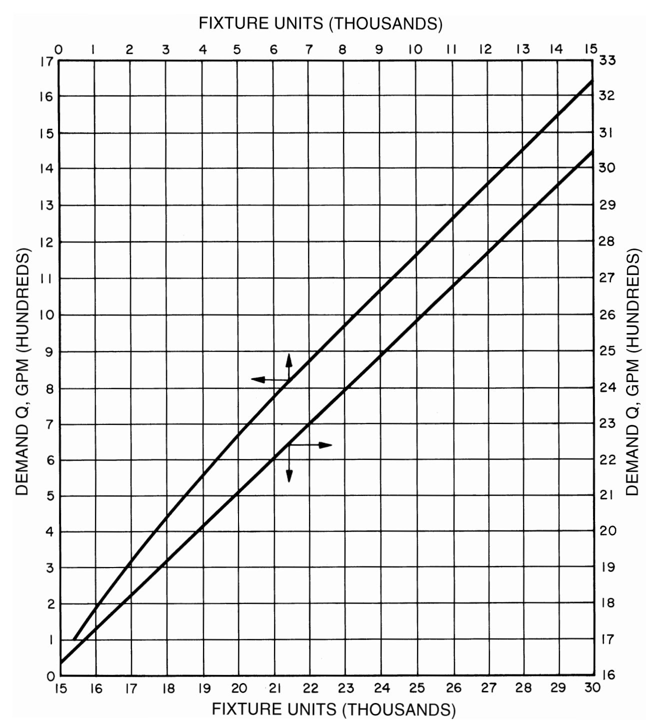
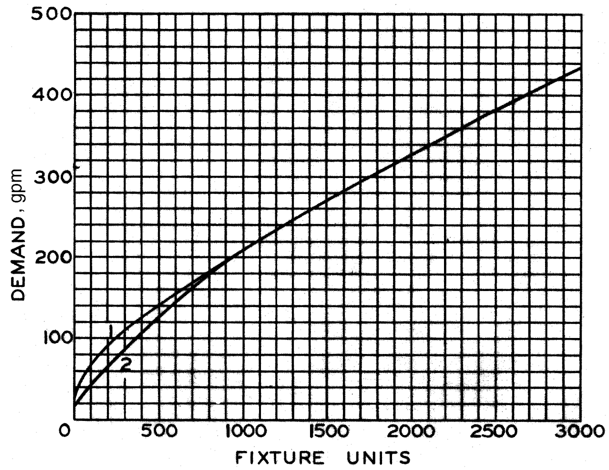
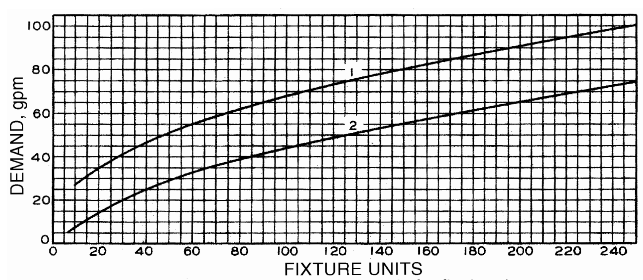


Fig. 5 Demand Versus Fixture Units, Mixed System, High Part of Curve
(Hunter 1941)



No. 1 for system predominantly for flush valves.
No. 2 for system predominantly for flush tanks.

Fig. 6 Estimate Curves for Demand Load
(Hunter 1941)



No. 1 for system predominantly for flush valves.
No. 2 for system predominantly for flush tanks.

Fig. 7 Section of Figure 6 on Enlarged Scale

So far, the information presented shows the *design rate of flow* to be determined in any particular section of piping. The next step is to determine the *size* of piping. As water flows through a pipe, the pressure continually decreases along the pipe due to loss of energy from friction. The problem is then to ascertain the minimum pressure in the street main and the minimum pressure required to operate the topmost fixture. (A pressure of 15 psig may be ample for most flush valves, but reference should be made to the manufacturers' requirements. Some fixtures require a pressure up to 25 psig. A minimum of 8 psig should be allowed for other fixtures.) The pressure differential overcomes pressure losses in the distributing system and the difference in elevation between the water main and the highest fixture.

The pressure loss (in psi) resulting from the difference in elevation between the street main and the highest fixture can be obtained by multiplying the difference in elevation in feet by the conversion factor 0.434.

Pressure losses in the distributing system consist of pressure losses in the piping itself, plus the pressure losses in the pipe fittings, valves, and the water meter, if any. Approximate design pressure losses and flow limits for disk-type meters for various rates of flow are given in Figure 8. Water authorities in many localities require compound meters for greater accuracy with varying flow; consult the local utility. Design data for compound meters differ from the data in Figure 8. Manufacturers give data on exact pressure losses and capacities.

Figure 9 shows the variation of pressure loss with rate of flow for various faucets and cocks. The water demand for hose bibbs or other large-demand fixtures taken off the building main frequently results in inadequate water supply to the upper floor of a building.

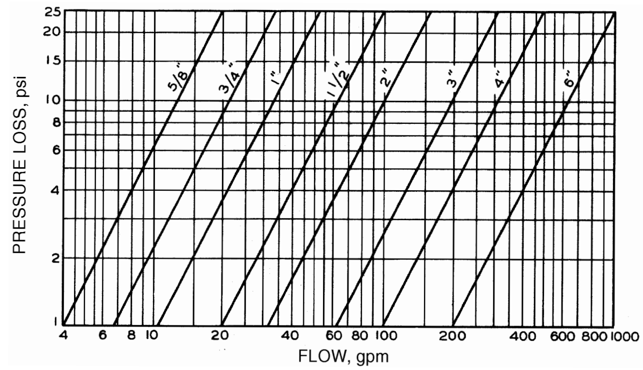
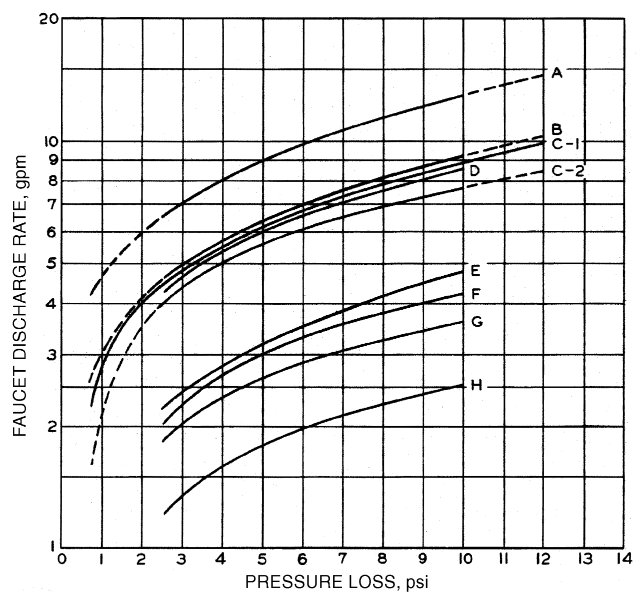


Fig. 8 Pressure Losses in Disk-Type Water Meters



A. 1/2 in. laundry bibb (old style)
B. Laundry compression faucet
C-1. 1/2 in. compression sink faucet (mfr. 1)
C-2. 1/2 in. compression sink faucet (mfr. 2)
D. Combination compression bathtub faucets (both open)
E. Combination compression sink faucet
F. Basin faucet
G. Spring self-closing faucet
H. Slow self-closing faucet
(Dashed lines indicate recommended extrapolation)

Fig. 9 Variation of Pressure Loss with Flow Rate for Various Faucets and Cocks

This condition can be prevented by sizing the distribution system so that the pressure drops from the street main to all fixtures are the same. An ample building main (not less than 1 in. where possible) should be maintained until all branches to hose bibbs have been connected. Where the street main pressure is excessive and a pressure reducing valve is used to prevent water hammer or excessive pressure at the fixtures, the hose bibbs should be connected ahead of the reducing valve.

The principles involved in sizing upfeed and downfeed systems are the same. In the downfeed system, however, the difference in elevation between the overhead supply mains and the fixtures provides the pressure required to overcome pipe friction. Because friction pressure loss and height pressure loss are not additive, as in an upfeed system, smaller pipes may be used with a downfeed system.

Plastic Pipe

The maximum safe water velocity in a thermoplastic piping system under most operating conditions is typically 5 fps; however, higher velocities can be used in cases where the operating characteristics of valves and pumps are known so that sudden changes in flow velocity can be controlled. The total pressure in the system at any time (operating pressure plus surge of water hammer) should not exceed 150% of the pressure rating of the system.

Procedure for Sizing Cold Water Systems

The recommended procedure for sizing piping systems is outlined below.

1. Sketch the main lines, risers, and branches, and indicate the fixtures to be served. Indicate the rate of flow of each fixture.
2. Using Table 9, compute the demand weights of the fixtures in fixture units.
3. Determine the total demand in fixture units and, using Figure 5, 6, or 7, find the expected demand.
4. Determine the equivalent length of pipe in the main lines, risers, and branches. Because the sizes of the pipes are not known, the exact equivalent length of various fittings cannot be determined. Add the equivalent lengths, starting at the street main and proceeding along the service line, the main line of the building, and up the riser to the top fixture of the group served.
5. Determine the average minimum pressure in the street main and the minimum pressure required for the operation of the topmost fixture, which should be 8 to 25 psi.
6. Calculate the approximate design value of the average pressure drop per 100 ft of equivalent length of pipe determined in step 4.

$$\Delta p = (p_s - 0.434H - p_f - p_m)100/L \tag{11}$$

where

Δp = average pressure loss per 100 ft of equivalent length of pipe, psi
p_s = pressure in street main, psig
p_f = minimum pressure required to operate topmost fixture, psig
p_m = pressure drop through water meter, psi
H = height of highest fixture above street main, ft
L = equivalent length determined in step 4, ft

If the system is downfeed supply from a gravity tank, height of water in the tank, converted to psi by multiplying by 0.434, replaces the street main pressure, and the term $0.434H$ is added instead of subtracted in calculating Δp. In this case, H is the vertical distance of the fixture below the bottom of the tank.

7. From the expected rate of flow determined in step 3 and the value of Δp calculated in step 6, choose the sizes of pipe from Figure 1, 2, or 3.

Example 7. Assume a minimum street main pressure of 55 psig; a height of topmost fixture (a urinal with flush valve) above street main of 50 ft; an equivalent pipe length from water main to highest fixture of 100 ft; a total load on the system of 50 fixture units; and that the water closets are flush valve operated. Find the required size of supply main.

Solution: From Figure 7, the estimated peak demand is 51 gpm. From Table 8, the minimum pressure required to operate the topmost fixture is 15 psig. For a trial computation, choose the 1-1/2 in. meter. From Figure 8, the pressure drop through a 1-1/2 in. disk-type meter for a flow of 51 gpm is 6.5 psi.

The pressure drop available for overcoming friction in pipes and fittings is $55 - 0.434 \times 50 - 15 - 6.5 = 12$ psi.

At this point, estimate the equivalent pipe length of the fittings on the direct line from the street main to the highest fixture. The exact equivalent length of the various fittings cannot be determined since the pipe sizes of the building main, riser, and branch leading to the highest fixture are not yet known, but a first approximation is necessary to tentatively select pipe sizes. If the computed pipe sizes differ from those used in determining the equivalent length of pipe fittings, a recalculation using the computed pipe sizes for the fittings will be necessary. For this example, assume that the total equivalent length of the pipe fittings is 50 ft.

The permissible pressure loss per 100 ft of equivalent pipe is $12 \times 100/(100 + 50) = 8$ psi or 18 ft/100 ft. A 1-1/2 in. building main is adequate.

The sizing of the branches of the building main, the risers, and the fixture branches follows these principles. For example, assume that one of the branches of the building main carries the cold water supply for 3 water closets, 2 bathtubs, and 3 lavatories. Using the permissible pressure loss of 8 psi per 100 ft, the size of branch (determined from Table 9 and Figures 1 and 7) is found to be 1-1/2 in. Items included in the computation of pipe size are as follows:

Fixtures, No. and Type	Fixture Units (Table 9 and Note c)		Demand (Figure 7)	Pipe Size (Figure 1)
3 flush valves	3×6 =	18		
2 bathtubs	$0.75 \times 2 \times 2$ =	3		
3 lavatories	$0.75 \times 3 \times 1$ =	2.25		
Total	=	23.25	38 gpm	1-1/2 in.

Table 10 is a guide to minimum pipe sizing where flush valves are used.

Table 10 Allowable Number of 1 in. Flush Valves Served by Various Sizes of Water Pipe[a]

Pipe Size, in.	No. of 1 in. Flush Valves
1-1/4	1
1-1/2	2-4
2	5-12
2-1/2	13-25
3	26-40
4	41-100

[a]Two 3/4 in. flush valves are assumed equal to one 1 in. flush valve but can be served by a 1 in. pipe. Water pipe sizing must consider demand factor, available pressure, and length of run.

Velocities exceeding 10 fps cause undesirable noise in the piping system. This usually governs the size of larger pipes in the system, while in small pipe sizes, the friction loss usually governs the selection because the velocity is low compared to friction loss. Velocity is the governing factor in downfeed systems, where friction loss is usually neglected. Velocity in branches leading to pump suctions should not exceed 5 fps.

If the street pressure is too low to adequately supply upper-floor fixtures, the pressure must be increased. Constant or variable speed booster pumps, alone or in conjunction with gravity supply tanks, or hydropneumatic systems may be used.

Flow control valves for individual fixtures under varying pressure conditions automatically adjust the flow at the fixture to a predetermined quantity. These valves allow the designer to (1) limit the flow at the individual outlet to the minimum suitable for the purpose, (2) hold the total demand for the system more closely to the required minimum, and (3) design the piping system as accurately as is practicable for the requirements.

STEAM PIPING

Pressure losses in steam piping for flows of dry or nearly dry steam are governed by Equations (1) through (7) in the section on Pressure Drop Equations. This section incorporates these principles with other information specific to steam systems.

Pipe Sizes

Required pipe sizes for a given load in steam heating depend on the following factors:

- The initial pressure and the total pressure drop that can be allowed between the source of supply and the end of the return system
- The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow
- The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit

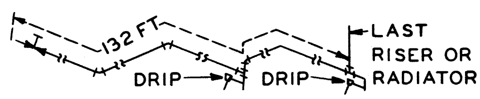
Initial Pressure and Pressure Drop. Table 11 lists pressure drops commonly used with corresponding initial steam pressures for sizing steam piping.

Several factors, such as initial pressure and pressure required at the end of the line, should be considered, but it is most important that (1) the total pressure drop does not exceed the initial gage pressure of the system (and in practice it should never exceed one-half the initial gage pressure); (2) the pressure drop is not great enough to cause excessive velocities; (3) a constant initial pressure is maintained, except on systems specially designed for varying initial pressures (e.g., subatmospheric pressure), which normally operate under controlled partial vacuums; and (4) for gravity return systems, the pressure drop to the heating units does not exceed the water column available for removing condensate (i.e., the height above the boiler water line of the lowest point on the steam main, on the heating units, or on the dry return).

Maximum Velocity. For quiet operation, steam velocity should be 8000 to 12,000 fpm, with a maximum of 15,000 fpm. The lower the velocity, the quieter the system. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counterflowing water may (1) produce objectionable sound, such as water hammer, or (2) result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which these disturbances take place is a function of (1) pipe size; (2) the pitch of the pipe if it runs horizontally; (3) the quantity of condensate flowing against the steam; and (4) the freedom of the piping from water pockets that, under certain conditions, act as a restriction in pipe size. Table 12 lists maximum capacities for various size steam lines.

Equivalent Length of Run. All tables for the flow of steam in pipes based on pressure drop must allow for pipe friction, as well as for the resistance of fittings and valves. These resistances are generally stated in terms of straight pipe; that is, a certain fitting produces a drop in pressure equivalent to the stated number of feet of straight run of the same size of pipe. Table 13 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter, the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe. A common sizing method is to assume the length of run and to check this assumption after pipes are sized. For this purpose, the length of run is usually assumed to be double the actual length of pipe.

Example 8. Using Table 13, determine the equivalent length in feet of pipe for the run illustrated.



Measured length=	132.0 ft
4 in. gate valve=	1.9 ft
Four 4 in. elbows=	36.0 ft
Two 4 in. tees=	36.0 ft
Equivalent=	205.9 ft

Table 11 Pressure Drops Used for Sizing Steam Pipe[a]

Initial Steam Pressure, psig	Pressure Drop per 100 ft	Total Pressure Drop in Steam Supply Piping
Vacuum return	2 to 4 oz/in^2	1 to 2 psi
0	0.5 oz/in^2	1 oz/in^2
1	2 oz/in^2	1 to 4 oz/in^2
2	2 oz/in^2	8 oz/in^2
5	4 oz/in^2	1.5 psi
10	8 oz/in^2	3 psi
15	1 psi	4 psi
30	2 psi	5 to 10 psi
50	2 to 5 psi	10 to 15 psi
100	2 to 5 psi	15 to 25 psi
150	2 to 10 psi	25 to 30 psi

[a]Equipment, control valves, and so forth must be selected based on delivered pressures.

Sizing Charts

Figure 10 is the basic chart for determining the flow rate and velocity of steam in Schedule 40 pipe for various values of pressure drop per 100 ft, based on 0 psig saturated steam. Using the multiplier chart (Figure 11), Figure 10 can be used at all saturation pressures between 0 and 200 psig (see Example 10).

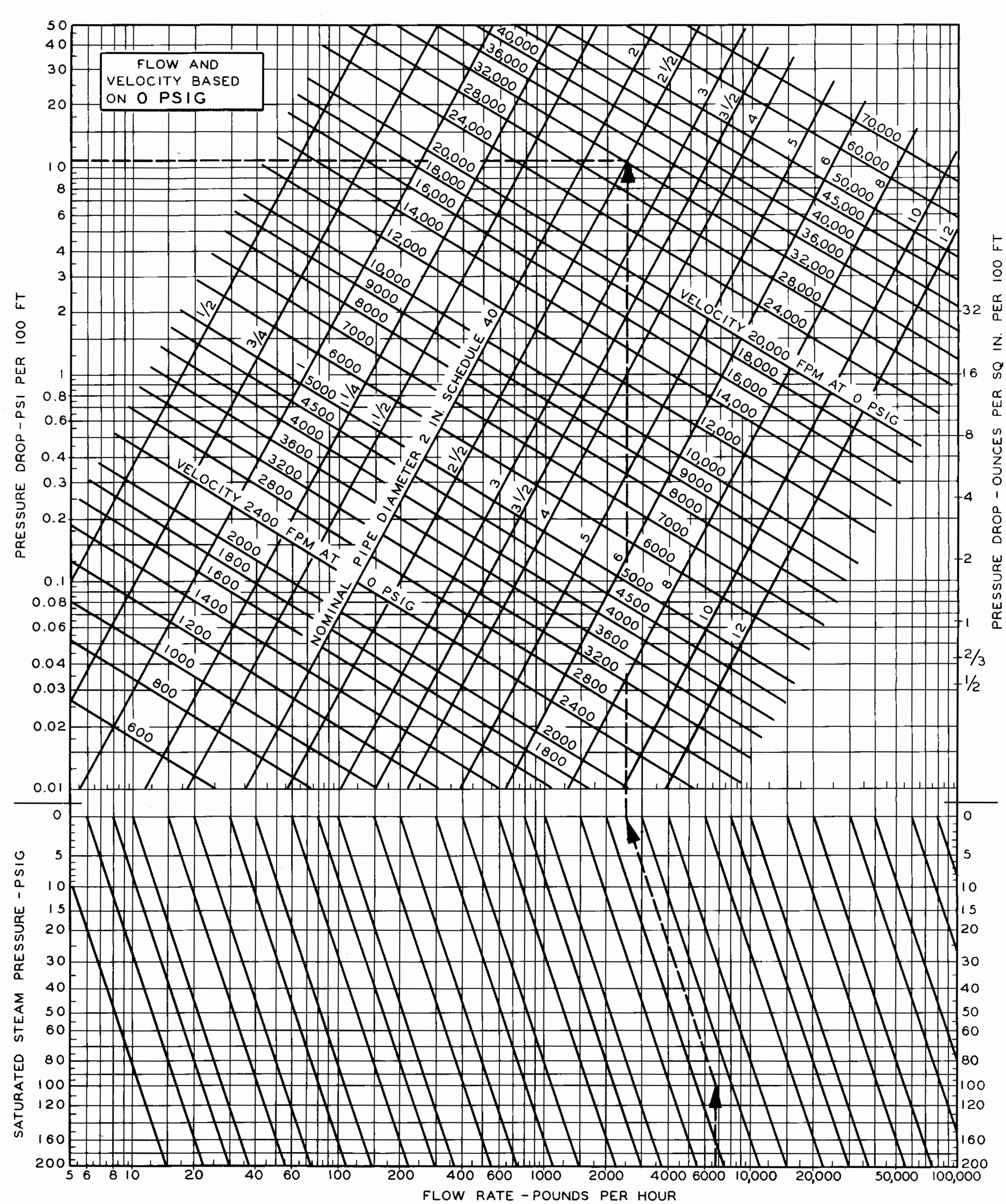
Figures 10A through 10D present charts for sizing steam piping for systems of 30, 50, 100, and 150 psig at various pressure drops. These charts are based on the Moody friction factor, which considers the Reynolds number and the roughness of the internal pipe surfaces; they contain the same information as the basic chart (Figure 10) but in a more convenient form.

Table 12 Comparative Capacity of Steam Lines at Various Pitches for Steam and Condensate Flowing in Opposite Directions

Pitch of Pipe, in/10 ft	Nominal Pipe Diameter, in.									
	3/4		1		1-1/4		1-1/2		2	
	Capacity	Maximum Velocity	Capacity	Maximum Velocity	Capacity	Maximum Velocity	Capacity	Maximum Velocity	Capacity	Maximum Velocity
1/4	3.2	8	6.8	9	11.8	11	19.8	12	42.9	15
1/2	4.1	11	9.0	12	15.9	14	25.9	16	54.0	18
1	5.7	13	11.7	15	19.9	17	33.0	19	68.8	24
1-1/2	6.4	14	12.8	17	24.6	20	37.4	22	83.3	27
2	7.1	16	14.8	19	27.0	22	42.0	24	92.9	30
3	8.3	17	17.3	22	31.3	25	46.8	26	99.6	32
4	9.9	22	19.2	24	33.4	26	50.8	28	102.4	32
5	10.5	22	20.5	25	38.5	31	59.2	33	115.0	33

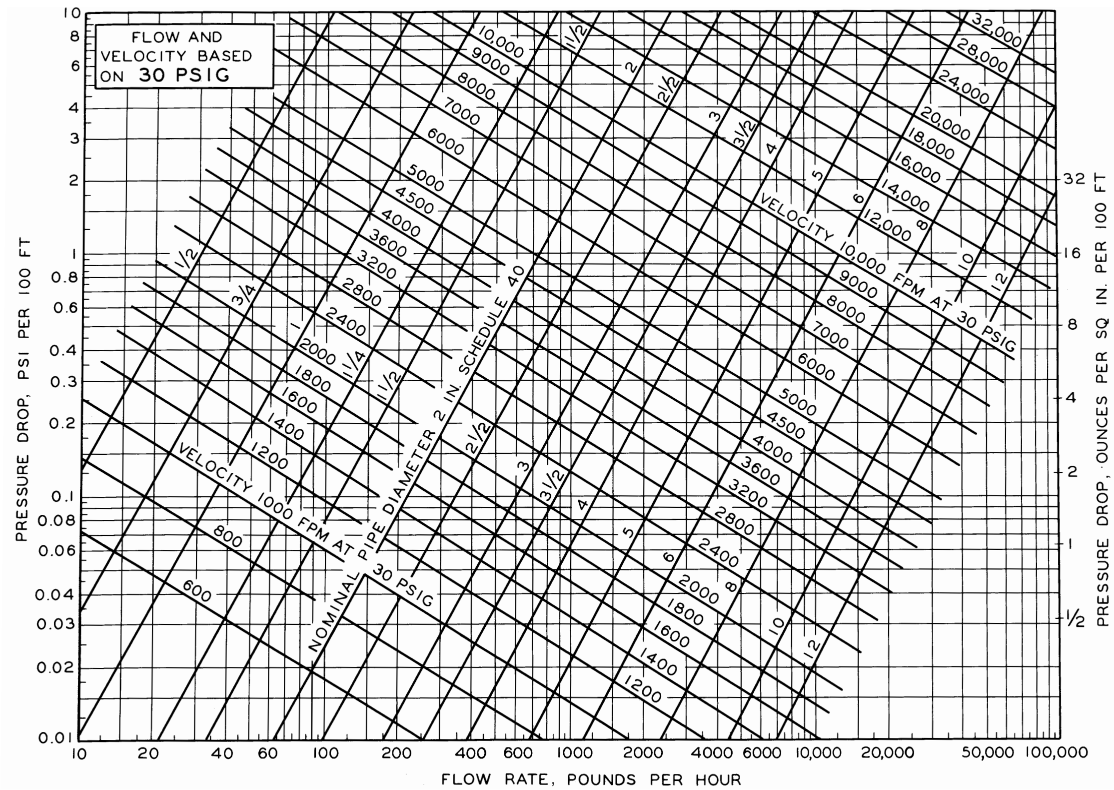
Source: Laschober et al. (1966).

Velocity in fps; capacity in lb/h.



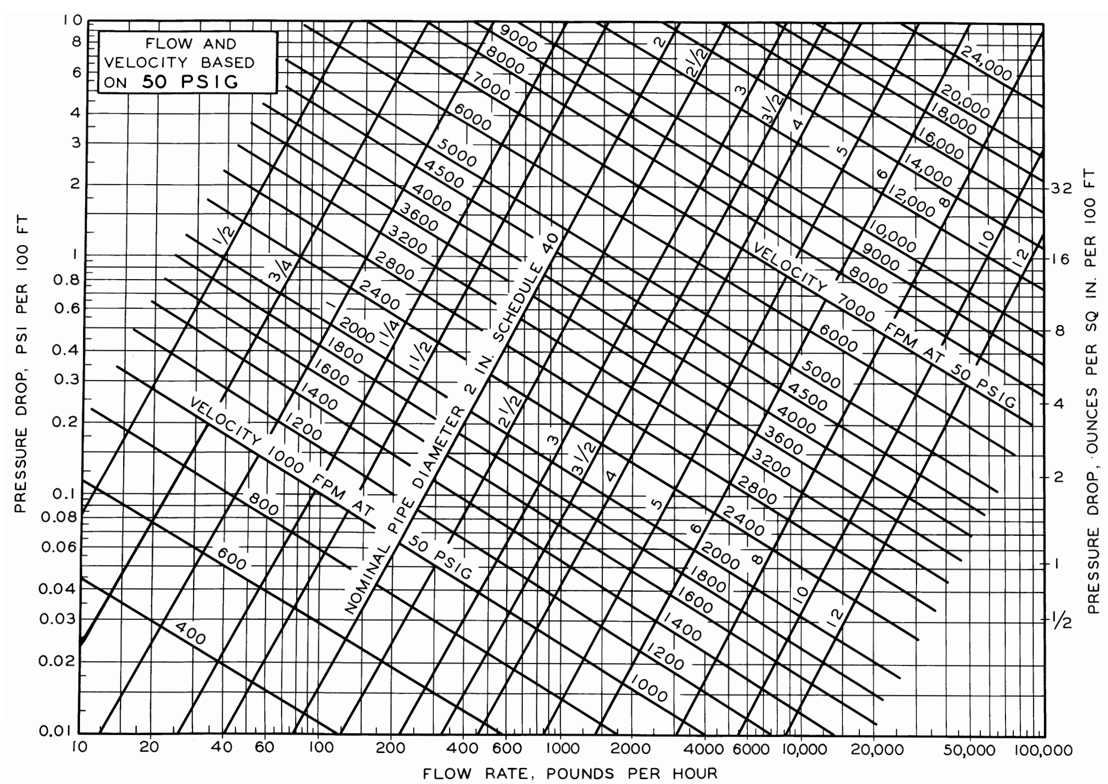
Notes: Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.
See Figure 11 for obtaining flow rates and velocities of all saturation pressures between 0 and 200 psig; see also Examples 9 and 10.

Fig. 10 Flow Rate and Velocity of Steam in Schedule 40 Pipe at Saturation Pressure of 0 psig



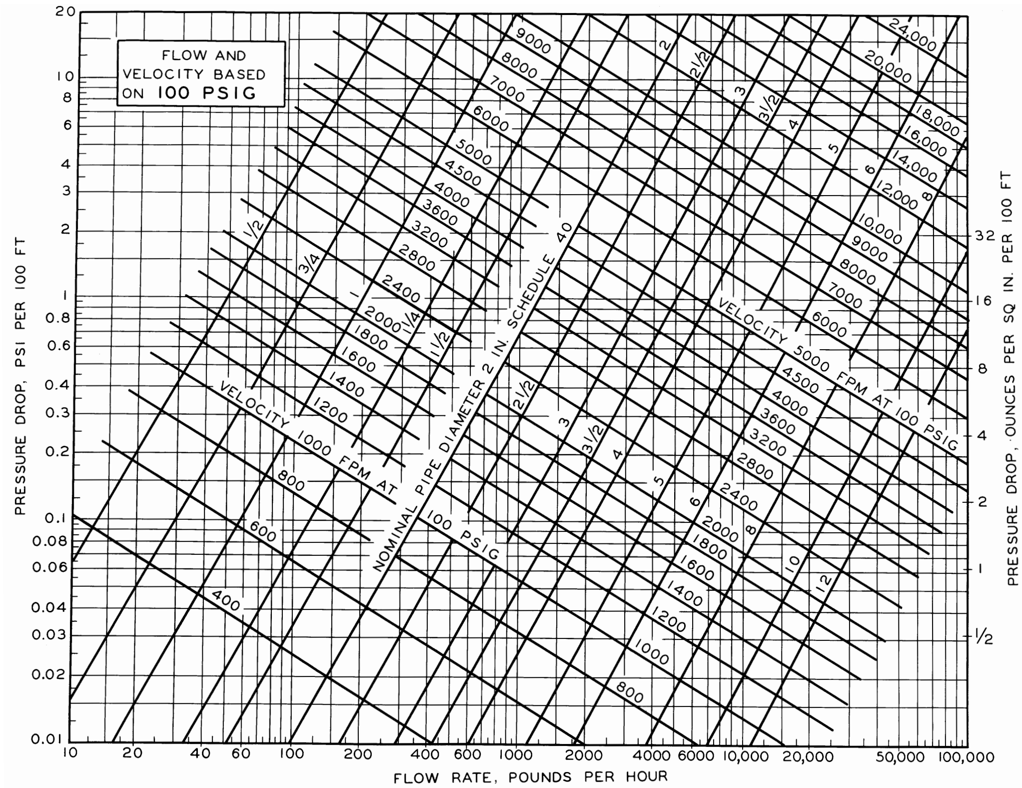
Notes: Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.
May be used for steam pressures from 23 to 37 psig with an error not exceeding 9%.

Fig. 10A Flow Rate and Velocity of Steam in Schedule 40 Pipe at Saturation Pressure of 30 psig



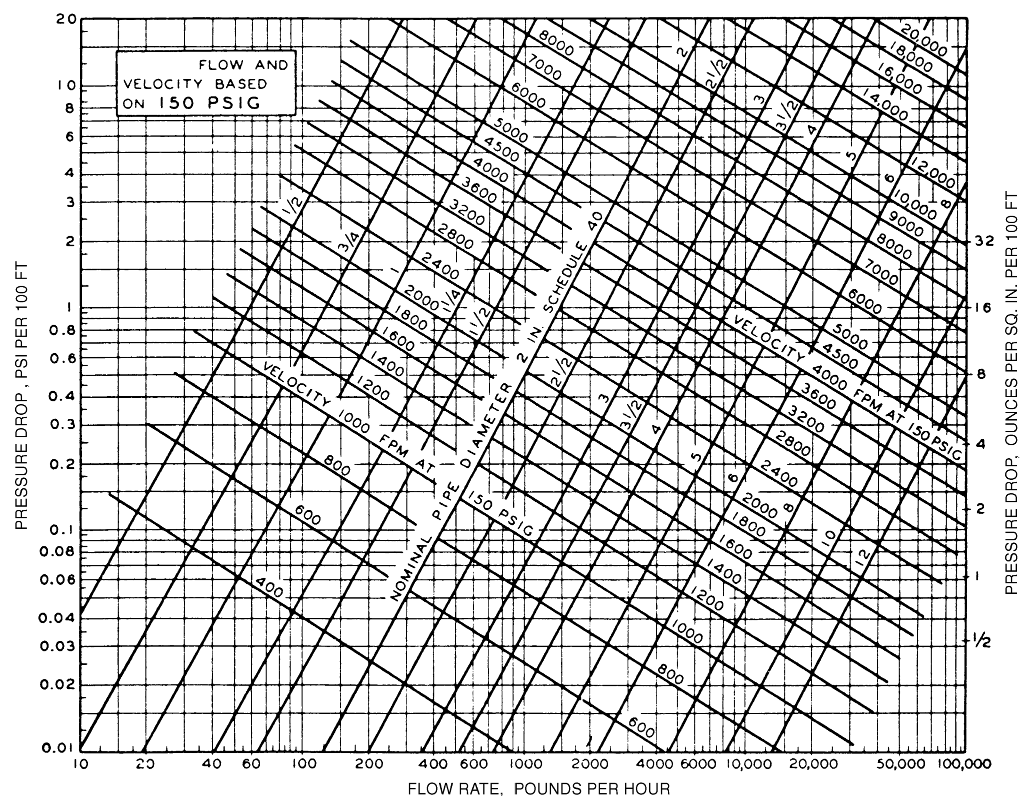
Notes: Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.
May be used for steam pressures from 40 to 60 psig with an error not exceeding 8%.

Fig. 10B Flow Rate and Velocity of Steam in Schedule 40 Pipe at Saturation Pressure of 50 psig



Notes: Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.
May be used for steam pressures from 85 to 120 psig with an error not exceeding 8%.

Fig. 10C Flow Rate and Velocity of Steam in Schedule 40 Pipe at Saturation Pressure of 100 psig



Notes: Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.
May be used for steam pressures from 127 to 180 psig with an error not exceeding 8%.

Fig. 10D Flow Rate and Velocity of Steam in Schedule 40 Pipe at Saturation Pressure of 150 psig

LOW-PRESSURE STEAM PIPING

Values in Table 14 (taken from Figure 10) provide a more rapid means of selecting pipe sizes for the various pressure drops listed and for systems operated at 3.5 and 12 psig. The flow rates shown for 3.5 psig can be used for saturated pressures from 1 to 6 psig, and those shown for 12 psig can be used for saturated pressures from 8 to 16 psig with an error not exceeding 8%.

Both Figure 10 and Table 14 can be used where the flow of condensate does not inhibit the flow of steam. Columns B and C of Table 15 are used in cases where steam and condensate flow in opposite directions, as in risers or runouts that are not dripped. Columns D, E, and F are for one-pipe systems and include risers, radiator valves and vertical connections, and radiator and riser runout sizes, all of which are based on the critical velocity of the steam to permit the counterflow of condensate without noise.

Return piping can be sized using Table 16, in which pipe capacities for wet, dry, and vacuum return lines are shown for several values of pressure drop per 100 ft of equivalent length.

Table 13 Equivalent Length of Fittings to Be Added to Pipe Run

Nominal Pipe Diameter, in.	Length to Be Added to Run, ft: Standard Elbow	Side Outlet Tee[b]	Gate Valve[a]	Globe Valve[a]	Angle Valve[a]
1/2	1.3	3	0.3	14	7
3/4	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1-1/4	3.0	6	0.6	29	15
1-1/2	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2-1/2	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3-1/2	8	15	1.6	80	40
4	9	18	1.9	92	45
5	11	22	2.2	112	56
6	13	27	2.8	136	67
8	17	35	3.7	180	92
10	21	45	4.6	230	112
12	27	53	5.5	270	132
14	30	63	6.4	310	152

[a]Valve in full-open position.
[b]Values apply only to a tee used to divert the flow in the main to the last riser.

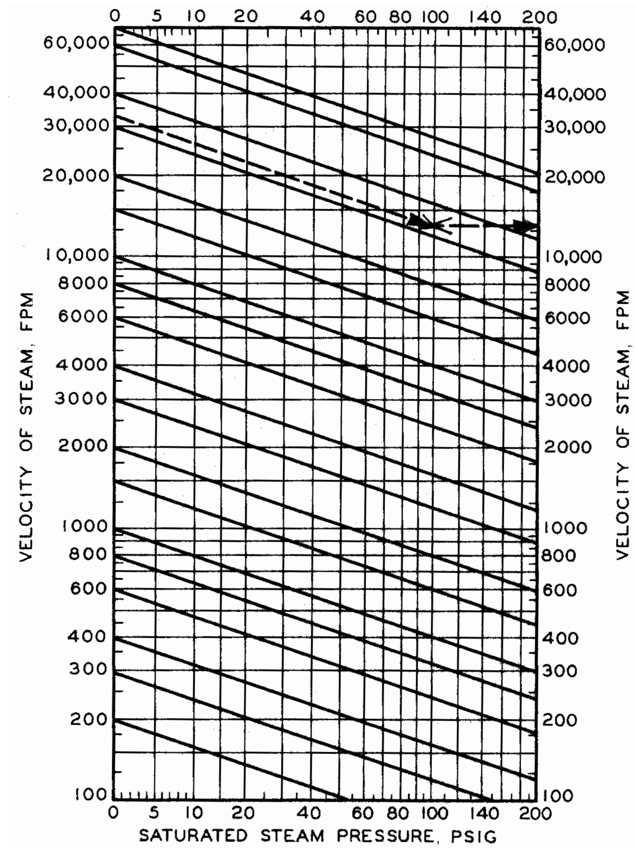


Fig. 11 Velocity Multiplier Chart for Figure 10

Table 14 Flow Rate of Steam in Schedule 40 Pipe

Nominal Pipe Size, in.	1/16 psi (1 oz/in^2) Sat. Press., psig 3.5	12	1/8 psi (2 oz/in^2) Sat. Press., psig 3.5	12	1/4 psi (4 oz/in^2) Sat. Press., psig 3.5	12	1/2 psi (8 oz/in^2) Sat. Press., psig 3.5	12	3/4 psi (12 oz/in^2) Sat. Press., psig 3.5	12	1 psi Sat. Press., psig 3.5	12	2 psi Sat. Press., psig 3.5	12
3/4	9	11	14	16	20	24	29	35	36	43	42	50	60	73
1	17	21	26	31	37	46	54	66	68	82	81	95	114	137
1-1/4	36	45	53	66	78	96	111	138	140	170	162	200	232	280
1-1/2	56	70	84	100	120	147	174	210	218	260	246	304	360	430
2	108	134	162	194	234	285	336	410	420	510	480	590	710	850
2-1/2	174	215	258	310	378	460	540	660	680	820	780	950	1,150	1,370
3	318	380	465	550	660	810	960	1,160	1,190	1,430	1,380	1,670	1,950	2,400
3-1/2	462	550	670	800	990	1,218	1,410	1,700	1,740	2,100	2,000	2,420	2,950	3,450
4	640	800	950	1,160	1,410	1,690	1,980	2,400	2,450	3,000	2,880	3,460	4,200	4,900
5	1,200	1,430	1,680	2,100	2,440	3,000	3,570	4,250	4,380	5,250	5,100	6,100	7,500	8,600
6	1,920	2,300	2,820	3,350	3,960	4,850	5,700	6,800	7,000	8,600	8,400	10,000	11,900	14,200
8	3,900	4,800	5,570	7,000	8,100	10,000	11,400	14,300	14,500	17,700	16,500	20,500	24,000	29,500
10	7,200	8,800	10,200	12,600	15,000	18,200	21,000	26,000	26,200	32,000	30,000	37,000	42,700	52,000
12	11,400	13,700	16,500	19,500	23,400	28,400	33,000	40,000	41,000	49,500	48,000	57,500	67,800	81,000

Pressure Drop per 100 ft of Length.

Notes:
1. Flow rate is in lb/h at initial saturation pressures of 3.5 and 12 psig. Flow is based on Moody friction factor, where the flow of condensate does not inhibit the flow of steam.
2. The flow rates at 3.5 psig cover saturated pressure from 1 to 6 psig, and the rates at 12 psig cover saturated pressure from 8 to 16 psig with an error not exceeding 8%.
3. The steam velocities corresponding to the flow rates given in this table can be found from Figures 10 and 11.

Table 15 Steam Pipe Capacities for Low-Pressure Systems

Nominal Pipe Size, in.	Capacity, lb/h: Two-Pipe System, Condensate Flowing Against Steam, Vertical	Two-Pipe System, Condensate Flowing Against Steam, Horizontal	One-Pipe Systems: Supply Risers Upfeed	Radiator Valves and Vertical Connections	Radiator and Riser Runouts
A	B[a]	C[b]	D[c]	E	F[b]
3/4	8	7	6	—	7
1	14	14	11	7	7
1-1/4	31	27	20	16	16
1-1/2	48	42	38	23	16
2	97	93	72	42	23
2-1/2	159	132	116	—	42
3	282	200	200	—	65
3-1/2	387	288	286	—	119
4	511	425	380	—	186
5	1,050	788	—	—	278
6	1,800	1,400	—	—	545
8	3,750	3,000	—	—	—
10	7,000	5,700	—	—	—
12	11,500	9,500	—	—	—
16	22,000	19,000	—	—	—

Notes:
1. For one- or two-pipe systems in which condensate flows against the steam flow.
2. Steam at an average pressure of 1 psig is used as a basis of calculating capacities.

[a] Do not use Column B for pressure drops of less than 1/16 psi per 100 ft of equivalent run. Use Figure 10 or Table 13 instead.
[b] Pitch of horizontal runouts to risers and radiators should be not less than 0.5 in/ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one pipe size larger than that called for in this table.
[c] Do not use Column D for pressure drops of less than 1/24 psi per 100 ft of equivalent run except on sizes 3 in. and over. Use Figure 10 or Table 13 instead.

Example 9. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft, and the initial pressure must not exceed 2 psig?

Solution: It is assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added, the equivalent length of run does not exceed 1000 ft. Then, with the pressure drop not over one-half of the initial pressure, the drop could be 1 psi or less. With a pressure drop of 1 psi and a length of run of 1000 ft, the drop per 100 ft would be 0.1 psi; if the total drop were 0.5 psi, the drop per 100 ft would be 0.05 psi. In both cases, the pipe could be sized for a desired capacity according to Figure 10.

On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is adequate; if it is more, an unusual number of fittings is probably involved, and either the lines must be straightened, or the next larger pipe size must be tried.

HIGH-PRESSURE STEAM PIPING

Many heating systems for large industrial buildings use high-pressure steam (15 to 150 psig). These systems usually have unit heaters or large built-up fan units with blast heating coils. Temperatures are controlled by a modulating or throttling thermostatic valve or by face or bypass dampers controlled by the room air temperature, fan inlet, or fan outlet.

Use of Basic and Velocity Multiplier Charts

Example 10. Given a flow rate of 6700 lb/h, an initial steam pressure of 100 psig, and a pressure drop of 11 psi/100 ft, find the size of Schedule 40 pipe required and the velocity of steam in the pipe.

Solution: The following steps are illustrated by the broken line on Figures 10 and 11.

1. Enter Figure 10 at a flow rate of 6700 lb/h, and move vertically to the horizontal line at 100 psig.

Table 16 Return Main and Riser Capacities for Low-Pressure Systems, lb/h

	Pipe Size, in.	1/32 psi (1/2 oz/in²) Drop per 100 ft			1/24 psi (2/3 oz/in²) Drop per 100 ft			1/16 psi (1 oz/in²) Drop per 100 ft			1/8 psi (2 oz/in²) Drop per 100 ft			1/4 psi (4 oz/in²) Drop per 100 ft			1/2 psi (8 oz/in²) Drop per 100 ft		
		Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y
Return Main	3/4	—	—	—	—	—	42	—	—	100	—	—	142	—	—	200	—	—	283
Return Main	1	125	62	—	145	71	143	175	80	175	250	103	249	350	115	350	—	—	494
Return Main	1-1/4	213	130	—	248	149	244	300	168	300	425	217	426	600	241	600	—	—	848
Return Main	1-1/2	338	206	—	393	236	388	475	265	475	675	340	674	950	378	950	—	—	1,340
Return Main	2	700	470	—	810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000	—	—	2,830
Return Main	2-1/2	1,180	760	—	1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350	—	—	4,730
Return Main	3	1,880	1,460	—	2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350	—	—	7,560
Return Main	3-1/2	2,750	1,970	—	3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000	—	—	11,300
Return Main	4	3,880	2,930	—	4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000	—	—	15,500
Return Main	5	—	—	—	—	—	7,880	—	—	9680	—	—	13,700	—	—	19,400	—	—	27,300
Return Main	6	—	—	—	—	—	12,600	—	—	15,500	—	—	22,000	—	—	31,000	—	—	43,800
Riser	3/4	—	48	—	—	48	143	—	48	175	—	48	249	—	48	350	—	—	494
Riser	1	—	113	—	—	113	244	—	113	300	—	113	426	—	113	600	—	—	848
Riser	1-1/4	—	248	—	—	248	388	—	248	475	—	248	674	—	248	950	—	—	1,340
Riser	1-1/2	—	375	—	—	375	815	—	375	1,000	—	375	1,420	—	375	2,000	—	—	2,830
Riser	2	—	750	—	—	750	1,360	—	750	1,680	—	750	2,380	—	750	3,350	—	—	4,730
Riser	2-1/2	—	—	—	—	—	2,180	—	—	2,680	—	—	3,800	—	—	5,350	—	—	7,560
Riser	3	—	—	—	—	—	3,250	—	—	4,000	—	—	5,680	—	—	8,000	—	—	11,300
Riser	3-1/2	—	—	—	—	—	4,480	—	—	5,500	—	—	7,810	—	—	11,000	—	—	15,500
Riser	4	—	—	—	—	—	7,880	—	—	9680	—	—	13,700	—	—	19,400	—	—	27,300
Riser	5	—	—	—	—	—	12,600	—	—	15,500	—	—	22,000	—	—	31,000	—	—	43,800

2. Follow along inclined multiplier line (upward and to the left) to horizontal 0 psig line. The equivalent mass flow at 0 psig is about 2500 lb/h.
3. Follow the 2500 lb/h line vertically until it intersects the horizontal line at 11 psi per 100 ft pressure drop. Nominal pipe size is 2-1/2 in. The equivalent steam velocity at 0 psig is about 32,700 fpm.
4. To find the steam velocity at 100 psig, locate the value of 32,700 fpm on the ordinate of the velocity multiplier chart (Figure 11) at 0 psig.
5. Move along the inclined multiplier line (downward and to the right) until it intersects the vertical 100 psig pressure line. The velocity as read from the right (or left) scale is about 13,000 fpm.

Note: Steps 1 through 5 would be rearranged or reversed if different data were given.

STEAM CONDENSATE SYSTEMS

The majority of steam systems used in heating applications are two-pipe systems, in which the two pipes are the "steam" pipe and the "condensate" pipe. This discussion is limited to the sizing of the condensate lines in two-pipe systems.

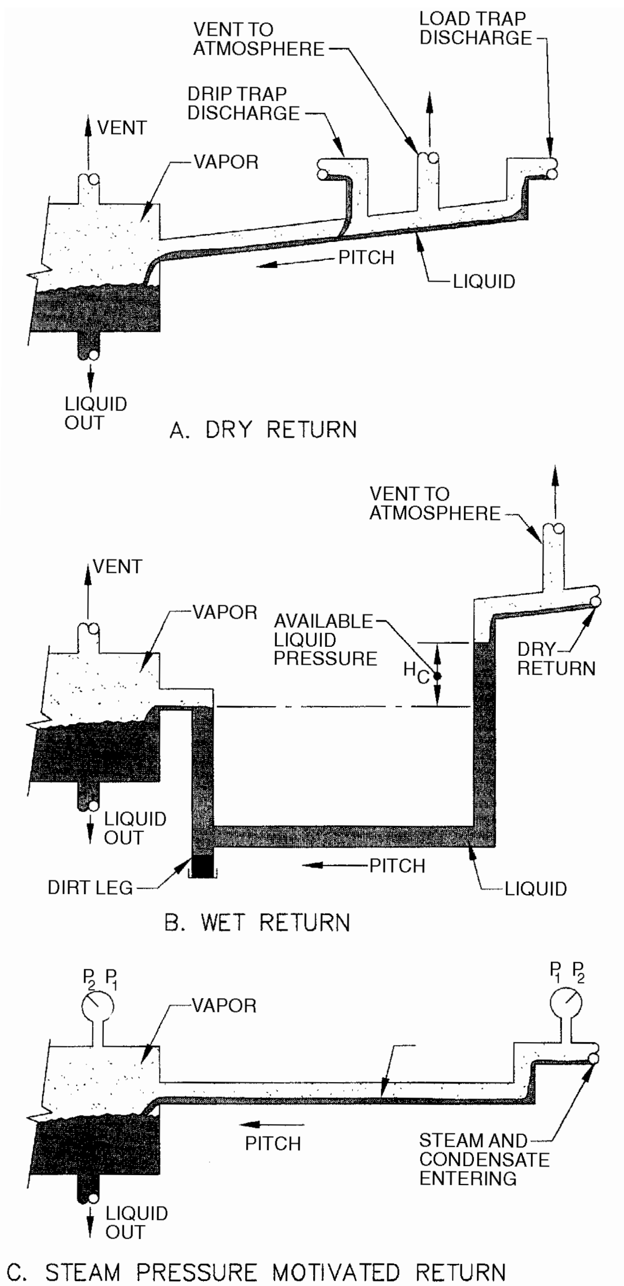


Fig. 12 Types of Condensate Return Systems

Two-Pipe Systems

When steam is used for heating a liquid to 215°F or less (e.g., in domestic water heat exchangers, domestic heating water converters, or air-heating coils), the devices are usually provided with a steam control valve. As the control valve throttles, the absolute pressure in the load device decreases, removing all pressure motivation for flow in the condensate return system. In order to ensure the flow of steam condensate from the load device through the trap and into the return system, it is necessary to provide a vacuum breaker on the device ahead of the trap. This ensures a minimum pressure at the trap inlet of atmospheric pressure plus whatever liquid leg the designer has provided. Then, to ensure flow through the trap, it is necessary to design the condensate system so that it will never have a pressure above atmospheric in the condensate return line.

Vented (Open) Return Systems. To achieve this pressure requirement, the condensate return line is usually vented to the atmosphere (1) near the point of entrance of the flow streams from the load traps, (2) in proximity to all connections from drip traps, and (3) at transfer pumps or feedwater receivers.

With this design, the only motivation for flow in the return system is gravity. Return lines that are below the liquid level in the downstream receiver or boiler and are thus filled with liquid are called wet returns; those above the liquid level have both liquid and gas in the pipes and are called dry returns.

The dry return lines in a vented return system have flowing liquid in the bottom of the line and gas or vapor in the top (Figure 12A). The liquid is the condensate, and the gas may be steam, air, or a mixture of the two. The flow phenomenon for these dry return systems is open channel flow, which is best described by the **Manning equation**:

$$Q = \frac{1.49Ar^{2/3}S^{1/2}}{n} \qquad (12)$$

where

Q = volumetric flow rate, cfs
A = cross-sectional area of conduit, ft^2
r = hydraulic radius of conduit, ft
n = coefficient of roughness (usually 0.012)
S = slope of conduit, ft/ft

Table 17 is a solution to Equation (12) that shows pipe size capacities for steel pipes with various pitches. Recommended

Table 17 Vented Dry Condensate Return for Gravity Flow Based on Manning Equation

Nominal Diameter, in. IPS	Condensate Flow, lb/h[a,b]			
	Condensate Line Slope, in/ft			
	1/16	1/8	1/4	1/2
1/2	38	54	76	107
3/4	80	114	161	227
1	153	216	306	432
1-1/4	318	449	635	898
1-1/2	479	677	958	1,360
2	932	1,320	1,860	2,640
2-1/2	1,500	2,120	3,000	4,240
3	2,670	3,780	5,350	7,560
4	5,520	7,800	11,000	15,600
5	10,100	14,300	20,200	28,500
6	16,500	23,300	32,900	46,500

[a] Flow is in lb/h of 180°F water for Schedule 40 steel pipes.
[b] Flow was calculated from Equation (12) and rounded.

practice is to size vertical lines by the maximum pitch shown, although they would actually have a capacity far in excess of that shown. As the pitch increases, hydraulic jump that could fill the pipe and other transient effects that could cause water hammer should be avoided. Flow values in Table 17 are calculated for Schedule 40 steel pipe, with a factor of safety of 3.0, and can be used for copper pipes of the same nominal pipe size.

The flow characteristics of **wet return lines** (Figure 12B) are best described by the Darcy-Weisbach equation [Equation (1)]. The motivation for flow is the fluid head difference between the entering section of the flooded line and the leaving section. It is common practice, in addition to providing for the fluid head differential, to slope the return in the direction of flow to a collection point such as a dirt leg in order to clear the line of sediment or solids. Table 18 is a solution to Equation (1) that shows pipe size capacity for steel pipes with various available fluid heads. Table 18 can also be used for copper tubing of equal nominal pipe size.

Nonvented (Closed) Return Systems. For those systems in which there is a continual steam pressure difference between the point where the condensate enters the line and the point where it

Table 18 Vented Wet Condensate Return for Gravity Flow Based on Darcy-Weisbach Equation

Nominal Diameter, in. IPS	Condensate Flow, lb/h[a,b]							
	Condensate Head, ft per 100 ft							
	0.5	1	1.5	2	2.5	3	3.5	4
1/2	105	154	192	224	252	278	302	324
3/4	225	328	408	476	536	590	640	687
1	432	628	779	908	1,020	1,120	1,220	1,310
1-1/4	901	1,310	1,620	1,890	2,120	2,330	2,530	2,710
1-1/2	1,360	1,970	2,440	2,840	3,190	3,510	3,800	4,080
2	2,650	3,830	4,740	5,510	6,180	6,800	7,360	7,890
2-1/2	4,260	6,140	7,580	8,810	9,890	10,900	11,800	12,600
3	7,570	10,900	13,500	15,600	17,500	19,300	20,900	22,300
4	15,500	22,300	27,600	32,000	35,900	39,400	42,600	45,600
5	28,200	40,500	49,900	57,900	64,900	71,300	77,100	82,600
6	45,800	65,600	80,900	93,800	105,000	115,000	125,000	134,000

[a] Flow is in lb/h of 180°F water for Schedule 40 steel pipes.
[b] Flow was calculated from Equation (1) and rounded.

Table 19 Flow Rate for Dry-Closed Returns

Pipe Dia. *D*, in.	Supply Pressure = 5 psig Return Pressure = 0 psig			Supply Pressure = 15 psig Return Pressure = 0 psig			Supply Pressure = 30 psig Return Pressure = 0 psig			Supply Pressure = 50 psig Return Pressure = 0 psig		
	$\Delta p/L$, psi/100 ft											
	1/16	1/4	1	1/16	1/4	1	1/16	1/4	1	1/16	1/4	1
	Flow Rate, lb/h											
1/2	240	520	1,100	95	210	450	60	130	274	42	92	200
3/4	510	1,120	2,400	210	450	950	130	280	590	91	200	420
1	1,000	2,150	4,540	400	860	1,820	250	530	1,120	180	380	800
1-1/4	2,100	4,500	9,500	840	1,800	3,800	520	1,110	2,340	370	800	1,680
1-1/2	3,170	6,780	14,200	1,270	2,720	5,700	780	1,670	3,510	560	1,200	2,520
2	6,240	13,300	a	2,500	5,320	a	1,540	3,270	a	1,110	2,350	a
2-1/2	10,000	21,300	a	4,030	8,520	a	2,480	5,250	a	1,780	3,780	a
3	18,000	38,000	a	7,200	15,200	a	4,440	9,360	a	3,190	6,730	a
4	37,200	78,000	a	14,900	31,300	a	9,180	19,200	a	6,660	13,800	a
6	110,500	a	a	44,300	a	a	27,300	a	a	19,600	a	a
8	228,600	a	a	91,700	a	a	56,400	a	a	40,500	a	a

Pipe Dia. *D*, in.	Supply Pressure = 100 psig Return Pressure = 0 psig			Supply Pressure = 150 psig Return Pressure = 0 psig			Supply Pressure = 100 psig Return Pressure = 15 psig			Supply Pressure = 150 psig Return Pressure = 15 psig		
	$\Delta p/L$, psi/100 ft											
	1/16	1/4	1	1/16	1/4	1	1/16	1/4	1	1/16	1/4	1
	Flow Rate, lb/h											
1/2	28	62	133	23	51	109	56	120	260	43	93	200
3/4	62	134	290	50	110	230	120	260	560	93	200	420
1	120	260	544	100	210	450	240	500	1,060	180	390	800
1-1/4	250	540	1,130	200	440	930	500	1,060	2,200	380	800	1,680
1-1/2	380	810	1,700	310	660	1,400	750	1,600	3,320	570	1,210	2,500
2	750	1,590	a	610	1,300	a	1,470	3,100	6,450	1,120	2,350	4,900
2-1/2	1,200	2,550	a	980	2,100	a	2,370	5,000	10,300	1,800	3,780	7,800
3	2,160	4,550	a	1,760	3,710	a	4,230	8,860	a	3,200	6,710	a
4	4,460	9,340	a	3,640	7,630	a	8,730	18,200	a	6,620	13,800	a
6	13,200	a	a	10,800	a	a	25,900	53,600	a	19,600	40,600	a
8	27,400	a	a	22,400	a	a	53,400	110,300	a	40,500	83,600	a

[a] For these sizes and pressure losses, the velocity is above 7000 fpm. Select another combination of size and pressure loss.

leaves (Figure 12C), Table 16 or Table 19, as applicable, can be used for sizing the condensate lines. Although these tables express condensate capacity without slope, common practice is to slope the lines in the direction of flow to a collection point similar to wet returns to clear the lines of sediment or solids.

When saturated condensate at pressures above the return system pressure enters the return (condensate) mains, some of the liquid flashes to steam. This occurs typically at drip traps into a vented return system or at load traps leaving process load devices that are not valve-controlled and typically have no subcooling. If the return main is vented, the vent lines will relieve any excessive pressure and prevent a back pressure phenomenon that could restrict the flow through traps from valved loads; the pipe sizing would be as described above for vented dry returns. If the return line is not vented, the flash steam results in a pressure rise at that point and the piping could be sized as described above for closed returns, and in accordance with Table 16 or Table 19, as applicable.

The passage of the fluid through the steam trap is a throttling or constant enthalpy process. The resulting fluid on the downstream side of the trap can be a mixture of saturated liquid and vapor. Thus, in nonvented returns, it is important to understand the condition of the fluid when it enters the return line from the trap.

The condition of the condensate downstream of the trap can be expressed by the quality x, defined as

$$x = \frac{m_v}{m_l + m_v} \tag{13}$$

where

m_v = mass of saturated vapor in condensate
m_l = mass of saturated liquid in condensate

Likewise, the volume fraction V_c of the vapor in the condensate is expressed as

$$V_c = \frac{V_v}{V_l + V_v} \tag{14}$$

where

V_v = volume of saturated vapor in condensate
V_l = volume of saturated liquid in condensate

The quality and the volume fraction of the condensate downstream of the trap can be estimated from Equations (13) and (14), respectively.

$$x = \frac{h_1 - h_{f_2}}{h_{g_2} - h_{f_2}} \tag{15}$$

$$V_c = \frac{x v_{g_2}}{v_{f_2}(1 - x) + x v_{g_2}} \tag{16}$$

where

h_1 = enthalpy of liquid condensate entering trap evaluated at supply pressure for saturated condensate or at saturation pressure corresponding to temperature of subcooled liquid condensate
h_{f_2} = enthalpy of saturated liquid at return or downstream pressure of trap
h_{g_2} = enthalpy of saturated vapor at return or downstream pressure of trap
v_{f_2} = specific volume of saturated liquid at return or downstream pressure of trap
v_{g_2} = specific volume of saturated vapor at return or downstream pressure of trap

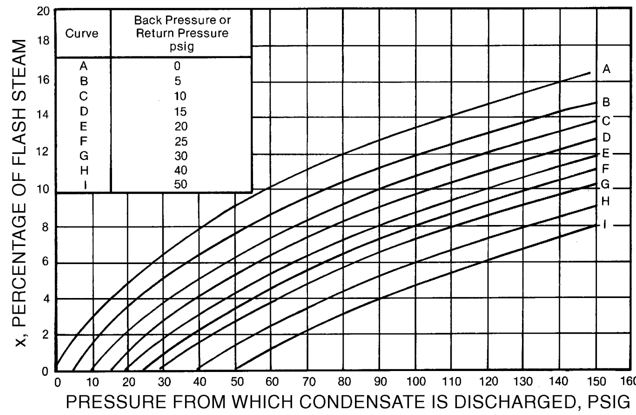


Fig. 13 Working Chart for Determining Percentage of Flash Steam (Quality)

Table 20 Flash Steam from Steam Trap on Pressure Drop

Supply Pressure, psig	Return Pressure, psig	x, Fraction Vapor, Mass Basis	V_c, Fraction Vapor, Volume Basis
5	0	0.016	0.962
15	0	0.040	0.985
30	0	0.065	0.991
50	0	0.090	0.994
100	0	0.133	0.996
150	0	0.164	0.997
100	15	0.096	0.989
150	15	0.128	0.992

Table 21 Estimated Return Line Pressures

Pressure Drop, psi/100 ft	Pressure in Return Line, psig	
	30 psig Supply	150 psig Supply
1/8	0.5	1.25
1/4	1	2.5
1/2	2	5
3/4	3	7.5
1	4	10
2	—	20

Table 20 presents some values for quality and volume fraction for typical supply and return pressures in heating and ventilating systems. Note that the percent of vapor on a mass basis x is small, while the percent of vapor on a volume basis V_c is very large. This indicates that the return pipe cross section is predominantly occupied by vapor. Figure 13 is a working chart to determine the quality of the condensate entering the return line from the trap for various combinations of supply and return pressures. If the liquid is subcooled entering the trap, the saturation pressure corresponding to the liquid temperature should be used for the supply or upstream pressure. Typical pressures in the return line are given in Table 21.

One-Pipe Systems

Gravity one-pipe air vent systems in which steam and condensate flow in the same pipe, frequently in opposite directions, are considered obsolete and are no longer being installed. See Chapter 33 of the 1993 *ASHRAE Handbook—Fundamentals* or earlier ASHRAE Handbooks for descriptions of and design information for one-pipe systems.

Table 22 Maximum Capacity of Gas Pipe in Cubic Feet per Hour

Nominal Iron Pipe Size, in.	Internal Diameter, in.	Length of Pipe, ft													
		10	20	30	40	50	60	70	80	90	100	125	150	175	200
1/4	0.364	32	22	18	15	14	12	11	11	10	9	8	8	7	6
3/8	0.493	72	49	40	34	30	27	25	23	22	21	18	17	15	14
1/2	0.622	132	92	73	63	56	50	46	43	40	38	34	31	28	26
3/4	0.824	278	190	152	130	115	105	96	90	84	79	72	64	59	55
1	1.049	520	350	285	245	215	195	180	170	160	150	130	120	110	100
1-1/4	1.380	1,050	730	590	500	440	400	370	350	320	305	275	250	225	210
1-1/2	1.610	1,600	1,100	890	760	670	610	560	530	490	460	410	380	350	320
2	2.067	3,050	2,100	1,650	1,450	1,270	1,150	1,050	990	930	870	780	710	650	610
2-1/2	2.469	4,800	3,300	2,700	2,300	2,000	1,850	1,700	1,600	1,500	1,400	1,250	1,130	1,050	980
3	3.068	8,500	5,900	4,700	4,100	3,600	3,250	3,000	2,800	2,600	2,500	2,200	2,000	1,850	1,700
4	4.026	17,500	12,000	9,700	8,300	7,400	6,800	6,200	5,800	5,400	5,100	4,500	4,100	3,800	3,500

Note: Capacity is in cubic feet per hour at gas pressures of 0.5 psig or less and a pressure drop of 0.3 in. of water; specific gravity = 0.60.

GAS PIPING

Piping for gas appliances should be of adequate size and installed so that it provides a supply of gas sufficient to meet the maximum demand without undue loss of pressure between the point of supply (the meter) and the appliance. The size of gas pipe required depends on (1) maximum gas consumption to be provided, (2) length of pipe and number of fittings, (3) allowable pressure loss from the outlet of the meter to the appliance, and (4) specific gravity of the gas.

Gas consumption in ft^3/h is obtained by dividing the Btu input rate at which the appliance is operated by the average heating value of the gas in Btu/ft^3. Insufficient gas flow from excessive pressure losses in gas supply lines can cause inefficient operation of gas-fired appliances and sometimes create hazardous operations. Gas-fired appliances are normally equipped with a data plate giving information on maximum gas flow requirements or Btu input as well as inlet gas pressure requirements. The gas utility in the area of installation can give the gas pressure available at the utility's gas meter. Using the information, the required size of gas piping can be calculated for satisfactory operation of the appliance(s).

Table 22 gives pipe capacities for gas flow for up to 200 ft of pipe based on a specific gravity of 0.60. Capacities for pressures less than 1.5 psig may also be determined by the following equation from NFPA/IAS *National Fuel Gas Code*:

$$Q = 2313 d^{2.623} \left(\frac{\Delta p}{CL} \right)^{0.541} \qquad (17)$$

where

Q = flow rate at 60°F and 30 in. Hg, cfh
d = inside diameter of pipe, in.
Δp = pressure drop, in. of water
C = factor for viscosity, density, and temperature
$= 0.00354(t + 460)s^{0.848}\mu^{0.152}$
t = temperature, °F
s = ratio of density of gas to density of air at 60°F and 30 in. Hg
μ = viscosity of gas, centipoise (0.012 for natural gas, 0.008 for propane)
L = pipe length, ft

Gas service in buildings is generally delivered in the "low-pressure" range of 7 in. of water. The maximum pressure drop allowable in piping systems at this pressure is generally 0.5 in. of water but is subject to regulation by local building, plumbing, and gas appliance codes (see also the NFPA/IAS *National Fuel Gas Code*).

Where large quantities of gas are required or where long lengths of pipe are used (e.g., in industrial buildings), low-pressure limitations result in large pipe sizes. Local codes may allow and local gas companies may deliver gas at higher pressures (e.g., 2, 5, or 10 psig). Under these conditions, an allowable pressure drop of 10% of the initial pressure is used, and pipe sizes can be reduced significantly. Gas pressure regulators at the appliance must be specified to accommodate higher inlet pressures. NFPA/IAS (1992) provides information on pipe sizing for various inlet pressures and pressure drops at higher pressures.

More complete information on gas piping can be found in the *Gas Engineers' Handbook* (1970).

FUEL OIL PIPING

The pipe used to convey fuel oil to oil-fired appliances must be large enough to maintain low pump suction pressure and, in the case of circulating loop systems, to prevent overpressure at the burner oil pump inlet. Pipe materials must be compatible with the fuel and must be carefully assembled to eliminate all leaks. Leaks in suction lines cause pumping problems that result in unreliable burner operation. Leaks in pressurized lines create fire hazards. Cast-iron or aluminum fittings and pipe are unacceptable. Pipe joint compounds must be selected carefully.

Oil pump suction lines should be sized so that at maximum suction line flow conditions, the maximum vacuum will not exceed 10 in. Hg for distillate grade fuels and 15 in. Hg for residual oils. Oil supply lines to burner oil pumps should not be pressurized by circulating loop systems or aboveground oil storage tanks to more than 5 psi, or pump shaft seals may fail. A typical oil circulating loop system is shown in Figure 14.

In assembling long fuel pipe lines, care should be taken to avoid air pockets. On overhead circulating loops, the line should vent air at all high points. Oil supply loops for one or more burners should be the continuous circulation type, with excess fuel returned to the storage tank. Dead-ended pressurized loops can be used, but air or vapor venting is more problematic.

Where valves are used, select ball or gate valves. Globe valves are not recommended because of their high pressure drop characteristics.

Oil lines should be tested after installation, particularly if they are buried, enclosed, or otherwise inaccessible. Failure to perform this test is a frequent cause of later operating difficulties. A suction line can be hydrostatically tested at 1.5 times its maximum operating pressure or at a vacuum of not less than 20 in. Hg. Pressure or vacuum tests should continue for at least 60 min. If there is no noticeable drop in the initial test pressure, the lines can be considered tight.

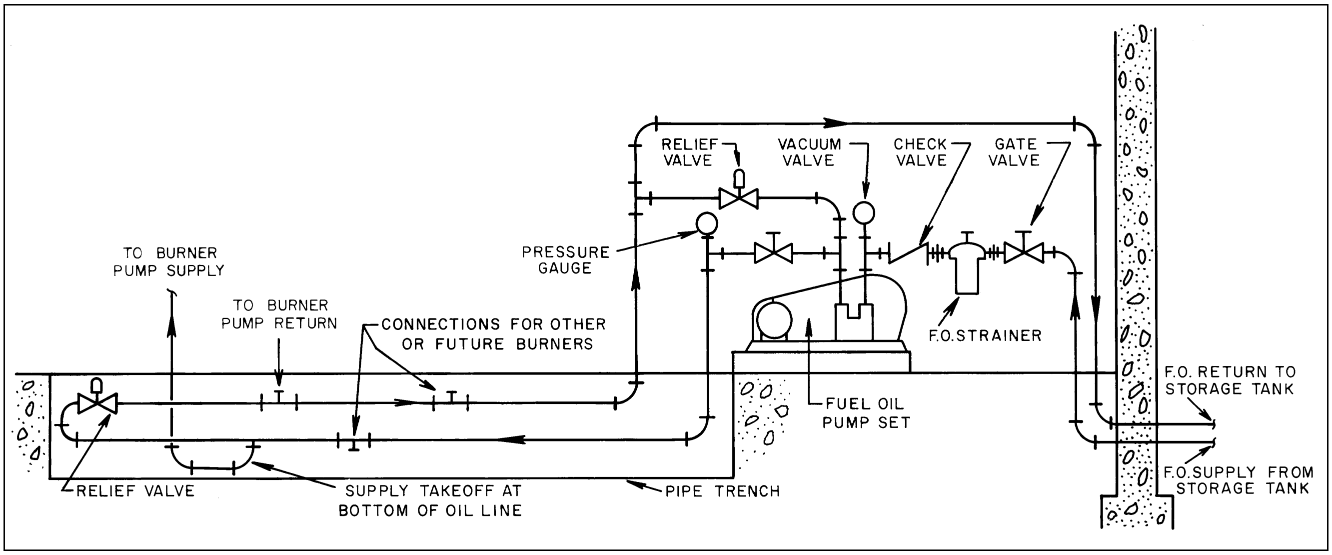


Fig. 14 Typical Oil Circulating Loop

Pipe Sizes for Heavy Oil

Tables 23 and 24 give recommended pipe sizes for handling No. 5 and No. 6 oils (residual grades) and No. 1 and No. 2 oils (distillate grades), respectively.

Table 23 Recommended Nominal Size for Fuel Oil Suction Lines from Tank to Pump (Residual Grades No. 5 and No. 6)

Pumping Rate, gph	Length of Run in Feet at Maximum Suction Lift of 15 ft									
	25	50	75	100	125	150	175	200	250	300
10	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	2	2	2-1/2	2-1/2
40	1-1/2	1-1/2	1-1/2	2	2	2-1/2	2-1/2	2-1/2	2-1/2	3
70	1-1/2	2	2	2	2	2-1/2	2-1/2	2-1/2	3	3
100	2	2	2	2-1/2	2-1/2	3	3	3	3	3
130	2	2	2-1/2	2-1/2	2-1/2	3	3	3	3	4
160	2	2	2-1/2	2-1/2	2-1/2	3	3	3	4	4
190	2	2-1/2	2-1/2	2-1/2	3	3	3	4	4	4
220	2-1/2	2-1/2	2-1/2	3	3	3	4	4	4	4

Notes:
1. Pipe sizes smaller than 1 in. IPS are not recommended for use with residual grade fuel oils.
2. Lines conveying fuel oil from pump discharge port to burners and tank return may be reduced by one or two sizes, depending on piping length and pressure losses.

Table 24 Recommended Nominal Size for Fuel Oil Suction Lines from Tank to Pump (Distillate Grades No. 1 and No. 2)

Pumping Rate, gph	Length of Run in Feet at Maximum Suction Lift of 10 ft									
	25	50	75	100	125	150	175	200	250	300
10	1/2	1/2	1/2	1/2	1/2	1/2	1/2	3/4	3/4	1
40	1/2	1/2	1/2	1/2	1/2	3/4	3/4	3/4	3/4	1
70	1/2	1/2	3/4	3/4	3/4	3/4	3/4	1	1	1
100	1/2	3/4	3/4	3/4	3/4	1	1	1	1	1-1/4
130	1/2	3/4	3/4	1	1	1	1	1	1-1/4	1-1/4
160	3/4	3/4	3/4	1	1	1	1	1-1/4	1-1/4	1-1/4
190	3/4	3/4	1	1	1	1	1-1/4	1-1/4	1-1/4	2
220	3/4	1	1	1	1	1-1/4	1-1/4	1-1/4	1-1/4	2

Storage tanks and piping and pumping facilities for delivering the oil from the tank to the burner are important considerations in the design of an industrial oil-burning system.

The construction and location of the tank and oil piping are usually subject to local regulations and National Fire Protection Association (NFPA) *Standards* 30 and 31.

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