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 pressure 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:

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

where

Δp = pressure drop, Pa
f = friction factor, dimensionless (from Moody chart, Figure 13 in Chapter 2)
L = length of pipe, m
D = internal diameter of pipe, m
ρ = fluid density at mean temperature, kg/m^3
V = average velocity, m/s

This equation is often presented in specific energy form as

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

where

Δh = energy loss, m
g = acceleration of gravity, m/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 preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

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, m
μ = dynamic viscosity of fluid, Pa·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

$$\Delta p = 6.819L\left(\frac{V}{C}\right)^{1.852}\left(\frac{1}{D}\right)^{1.167}(\rho g) \tag{5}$$

or

$$\Delta h = 6.819L\left(\frac{V}{C}\right)^{1.852}\left(\frac{1}{D}\right)^{1.167} \tag{6}$$

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

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

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

Table 1 *K* Factors—Screwed Pipe Fittings

Nominal Pipe Dia., mm	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
10	2.5	—	0.38	2.5	0.90	2.7	20	0.40	—	8.0	0.05	0.5	1.0
15	2.1	—	0.37	2.1	0.90	2.4	14	0.33	—	5.5	0.05	0.5	1.0
20	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
25	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
32	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
40	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
50	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
65	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
80	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
100	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., mm	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
25	0.43	0.41	0.22	0.43	0.43	0.26	1.0	13	—	4.8	2.0
32	0.41	0.37	0.22	0.41	0.38	0.25	0.95	12	—	3.7	2.0
40	0.40	0.35	0.21	0.40	0.35	0.23	0.90	10	—	3.0	2.0
50	0.38	0.30	0.20	0.38	0.30	0.20	0.84	9	0.34	2.5	2.0
65	0.35	0.28	0.19	0.35	0.27	0.18	0.79	8	0.27	2.3	2.0
80	0.34	0.25	0.18	0.34	0.25	0.17	0.76	7	0.22	2.2	2.0
100	0.31	0.22	0.18	0.31	0.22	0.15	0.70	6.5	0.16	2.1	2.0
150	0.29	0.18	0.17	0.29	0.18	0.12	0.62	6	0.10	2.1	2.0
200	0.27	0.16	0.17	0.27	0.15	0.10	0.58	5.7	0.08	2.1	2.0
250	0.25	0.14	0.16	0.25	0.14	0.09	0.53	5.7	0.06	2.1	2.0
300	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 50 mm	Tee	Screwed, line or branch	±25%
		±40% below 50 mm		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 15°C water flowing at 1 m/s through a nominal 25 mm, 90° screwed ell.

Solution: From Table 1, the *K* for a 25 mm, 90° screwed ell is 1.5.

$\Delta p = 1.5 \times 1000 \times 1^2/2 = 750$ Pa

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

$$Q = A_v \sqrt{\Delta p / \rho} \tag{8}$$

where

Q = volumetric flow, m^3/s
A_v = valve coefficient, m^3/s at $\Delta p = 1$ Pa
Δp = pressure drop, Pa
ρ = density of fluid ≈ 1000 kg/m^3 for water at below 120°C

Example 2. Determine the volumetric flow through a valve with A_v = 0.00024 for an allowable pressure drop of 35 kPa.

Solution:

$Q = 0.00024\sqrt{35\ 000/1000} = 0.0014\ m^3/s = 1.4$ L/s

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 1.2 m/s for 50 mm pipe and smaller, and a pressure drop limit of 400 Pa/m for piping over 50 mm. 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 4.6 m/s 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 3 to 5 m/s 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, m/s	Reference
General service	1.2 to 3.0	a, b, c
City water	0.9 to 2.1	a, b
	0.6 to 1.5	c
Boiler feed	1.8 to 4.6	a, c
Pump suction and drain lines	1.2 to 2.1	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, m/s
1500	4.6
2000	4.4
3000	4.0
4000	3.7
6000	3.0

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 13 m/s, cavitation did not occur in straight pipe; using the apparatus with two elbows, cold water velocities up to 6.5 m/s caused no cavitation. Cavitation did occur in orifices of 1:8 area ratio (orifice flow area is one-eighth of pipe flow area) at 1.5 m/s and in 1:4 area ratio orifices at 3 m/s (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

$$SL = 10 \log A_v + 20 \log \Delta p - 30 \log t + 76.6 \tag{9}$$

where

SL = sound level, dB
A_v = valve coefficient, $m^3/(s \cdot \sqrt{Pa})$
Q = flow rate, m^3/s
Δp = pressure drop across valve, Pa
t = downstream pipe wall thickness, mm

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 30 m/s, 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,

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

where

Δp_h = pressure rise caused by water hammer, Pa
ρ = fluid density, kg/m^3
c_s = velocity of sound in fluid, m/s
V = fluid flow velocity, m/s

The c_s for water is 1439 m/s, although the elasticity of the pipe reduces the effective value.

Example 3. What is the maximum pressure rise if water flowing at 3 m/s is stopped instantaneously?

Solution: $\Delta p_h = 1000 \times 1439 \times 3 = 4.32$ MPa

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 15°C. 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 15°C water charts for 90°C 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 100 and 400 Pa/m of pipe. A value of 250 Pa/m 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 1.2 m/s for 50 mm pipe and under, and a pressure drop limit of 400 Pa/m for piping over 50 mm in diameter. Velocities in excess of 1.2 m/s 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 0.5 to 0.6 m/s or more in pipe 50 mm and under. Minimum velocities of 0.6 m/s are therefore recommended. For pipe sizes 50 mm and over, minimum velocities corresponding to a pressure loss of 75 Pa 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 1.25 L/s flow.

Solution: Enter Figure 1 at 1.25 L/s, read up to pipe size within normal design range (100 to 400 Pa/m), and select 40 mm. Velocity is 1 m/s and pressure loss is 300 Pa/m.

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 length of pipe for a 100 mm open gate valve at a flow velocity of approximately 1.33 m/s.

Solution: From Table 6, at 1.33 m/s, each elbow is equivalent to 3.2 m of 100 mm 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 3.2 × 0.5, or 1.6 m of 100 mm pipe.

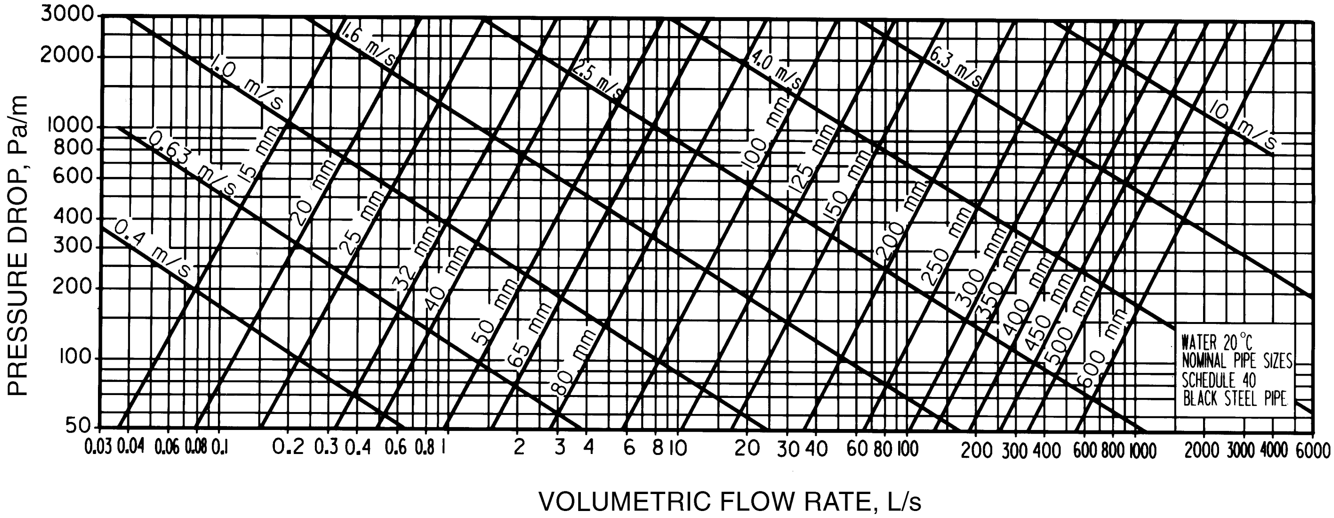


Fig. 1 Friction Loss for Water in Commercial Steel Pipe (Schedule 40)

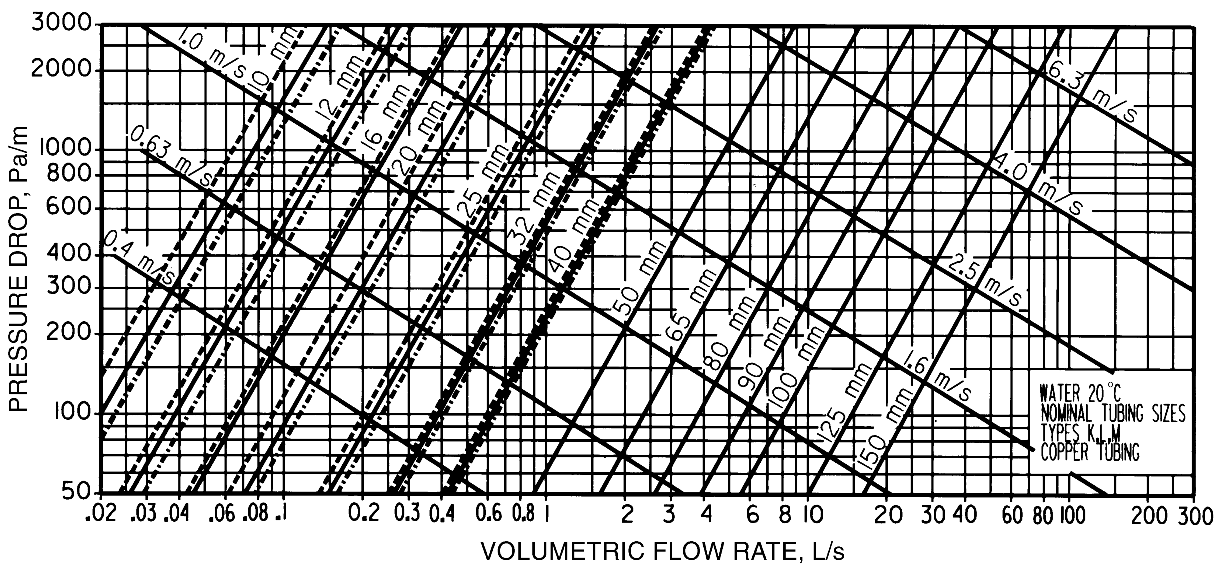


Fig. 2 Friction Loss for Water in Copper Tubing (Types K, L, M)

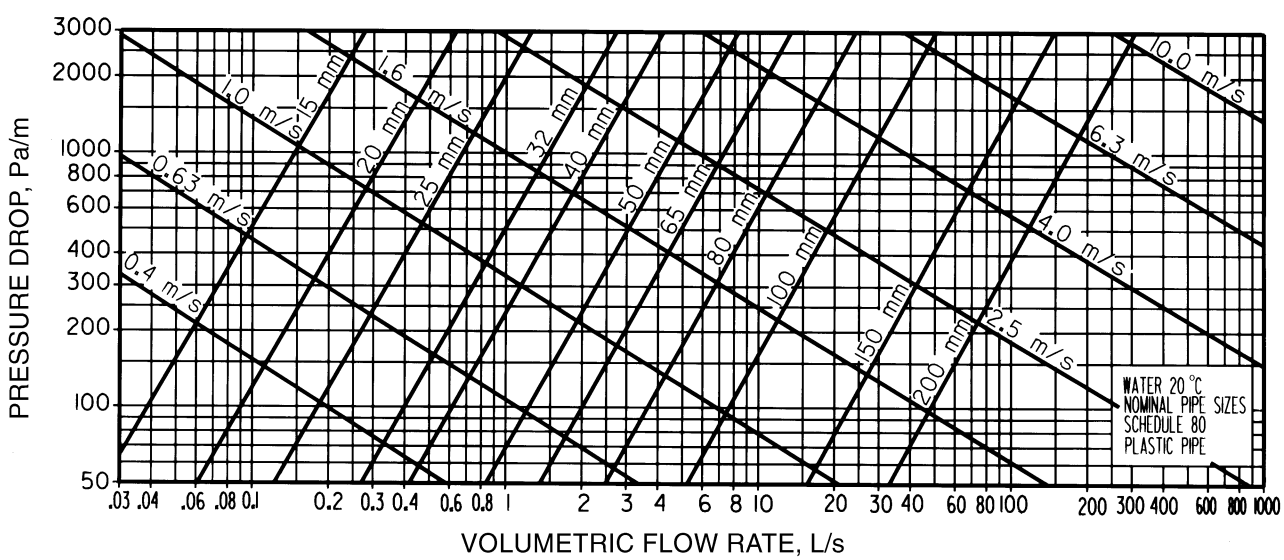


Fig. 3 Friction Loss for Water in Plastic Pipe (Schedule 80)

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

Velocity, m/s	Pipe Size, mm 15	20	25	32	40	50	65	90	100	125	150	200	250	300
0.33	0.4	0.5	0.7	0.9	1.1	1.4	1.6	2.0	2.6	3.2	3.7	4.7	5.7	6.8
0.67	0.4	0.6	0.8	1.0	1.2	1.5	1.8	2.3	2.9	3.6	4.2	5.3	6.3	7.6
1.00	0.5	0.6	0.8	1.1	1.3	1.6	1.9	2.5	3.1	3.8	4.5	5.6	6.8	8.0
1.33	0.5	0.6	0.8	1.1	1.3	1.7	2.0	2.5	3.2	4.0	4.6	5.8	7.1	8.4
1.67	0.5	0.7	0.9	1.2	1.4	1.8	2.1	2.6	3.4	4.1	4.8	6.0	7.4	8.8
2.00	0.5	0.7	0.9	1.2	1.4	1.8	2.2	2.7	3.5	4.3	5.0	6.2	7.6	9.0
2.35	0.5	0.7	0.9	1.2	1.5	1.9	2.2	2.8	3.6	4.4	5.1	6.4	7.8	9.2
2.67	0.5	0.7	0.9	1.3	1.5	1.9	2.3	2.8	3.6	4.5	5.2	6.5	8.0	9.4
3.00	0.5	0.7	0.9	1.3	1.5	1.9	2.3	2.9	3.7	4.5	5.3	6.7	8.1	9.6
3.33	0.5	0.8	0.9	1.3	1.5	1.9	2.4	3.0	3.8	4.6	5.4	6.8	8.2	9.8

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 25 mm 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., $\Delta p = 1.0\rho V_{in}^2/2 + 1.0\rho V_{out}^2/2$).

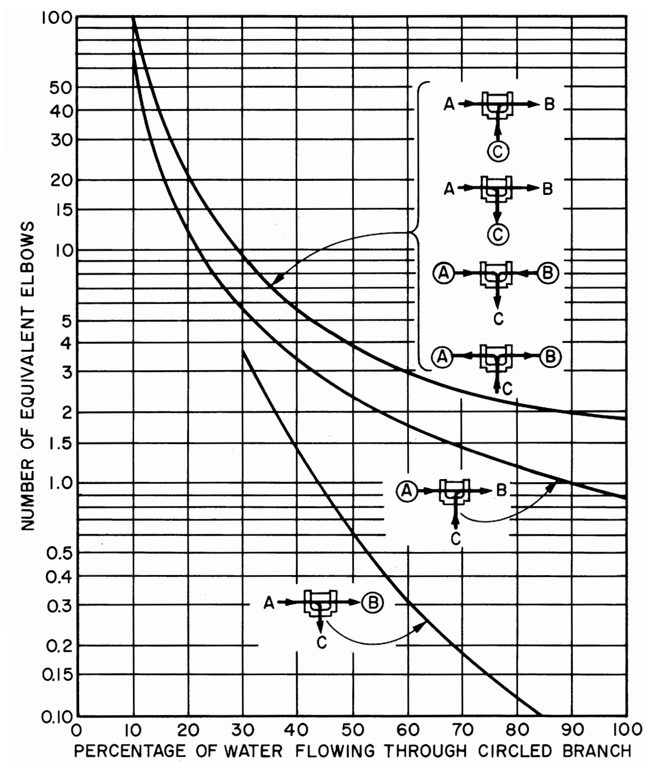
Example 6. Determine the pressure or energy losses for a 25 mm (all openings) threaded pipe tee flowing 25% to the side branch, 75% through. The entering flow is 1 L/s (1.79 m/s).

Solution: From Figure 4, bottom curve, the number of equivalent elbows for the through-flow is 0.15 elbows; the through-flow is 0.75 L/s (1.34 m/s); and the pressure loss is based on the exit flow rate. Table 6 gives the equivalent length of a 25 mm elbow at 1.33 m/s as 0.8 m. Using Equations (1) and (2) with friction factor f = 0.0263 and diameter D = 26.6 mm,

$$\Delta p = (0.15)(0.0263)(0.8/0.0266)(1000)(1.34^2)/2$$
$$= 0.107 \text{ kPa pressure drop, or}$$
$$\Delta h = (0.15)(0.0263)(0.8/0.0266)(1.34^2)/[(2)(9.8)]$$
$$= 0.0109 \text{ m loss}$$

From Figure 4, top curve, the number of equivalent elbows for the branch flow of 25% is 13 elbows; the branch flow is 0.25 L/s (0.45 m/s); and the pressure loss is based on the exit flow rate. Interpolating from Table 6 gives the equivalent of a 25 mm elbow at 0.45 m/s as 0.75 m. Using Equations (1) and (2) with friction factor f = 0.0334 and diameter = 26.6 mm,

$$\Delta p = (13)(0.0334)(0.75/0.0266)(1000)(0.45^2)/(2)$$
$$= 1.24 \text{ kPa pressure drop, or}$$
$$\Delta h = (13)(0.0334)(0.75/0.0266)(0.45^2)/[(2)(9.8)]$$
$$= 0.126 \text{ m loss}$$



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 litres per second 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, kPa (gage)[a]	Flow, L/s
Ordinary basin faucet	55	0.2
Self-closing basin faucet	85	0.2
Sink faucet—10 mm	70	0.3
Sink faucet—15 mm	35	0.3
Dishwasher	105 to 175	—[b]
Bathtub faucet	35	0.4
Laundry tube cock—8 mm	35	0.3
Shower	85	0.2 to 0.6
Ball cock for closet	105	0.2
Flush valve for closet	70 to 140	1.0 to 2.5[c]
Flush valve for urinal	105	1.0
Garden hose, 15 m, and sill cock	210	0.3

[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.

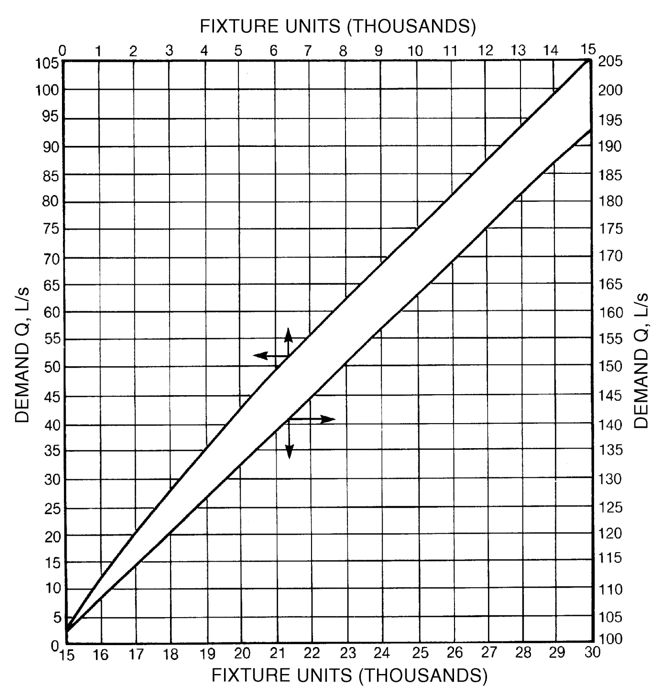


Fig. 5 Demand Versus Fixture Units, Mixed System, High Part of Curve
(Hunter 1941)

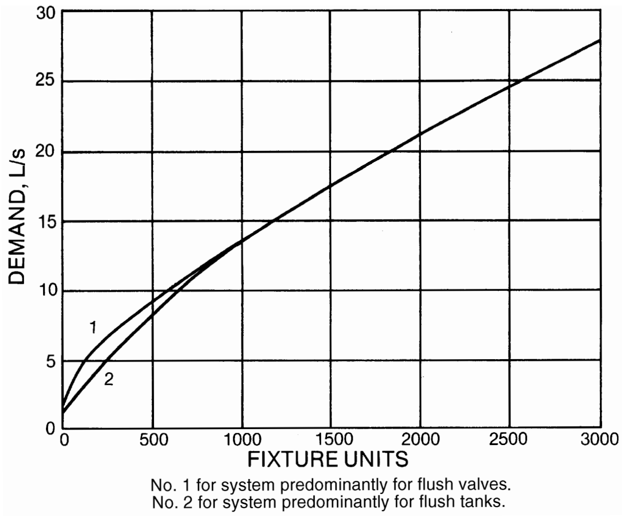


Fig. 6 Estimate Curves for Demand Load
(Hunter 1941)

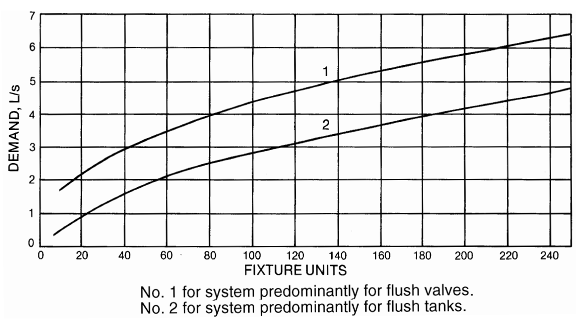


Fig. 7 Section of Figure 6 on Enlarged Scale

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 100 kPa may be ample for most flush valves, but reference should be made to the manufacturers' requirements. Some fixtures require a pressure up to 175 kPa. A minimum of 55 kPa 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 kPa) resulting from the difference in elevation between the street main and the highest fixture can be obtained by multiplying the difference in elevation in metres by the conversion factor 9.8.

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.

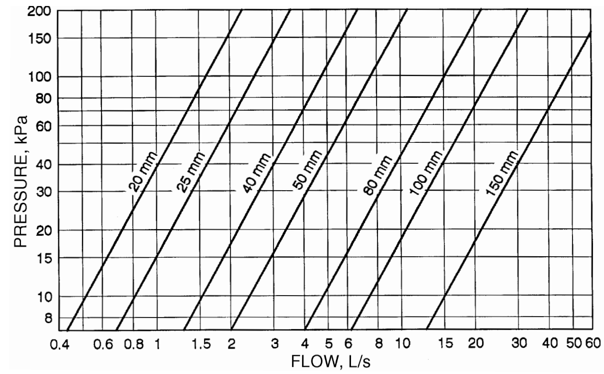
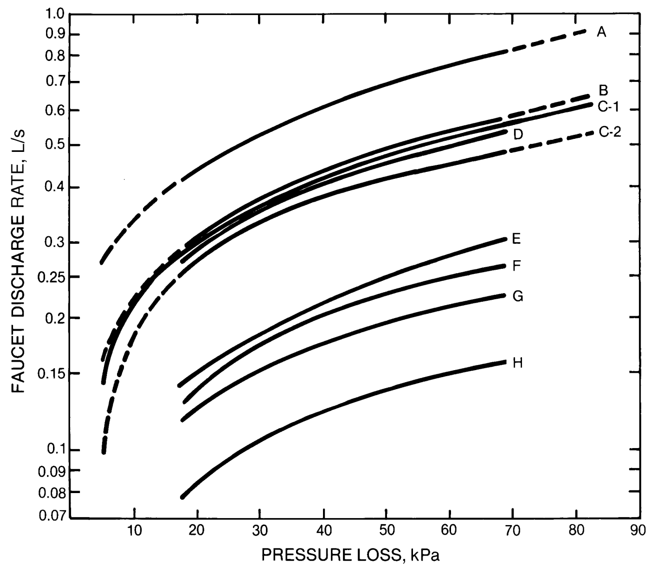


Fig. 8 Pressure Losses in Disk-Type Water Meters



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 25 mm 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 1.5 m/s; 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 50 to 175 kPa above atmospheric.
6. Calculate the approximate design value of the average pressure drop per unit length of pipe in equivalent length determined in step 4.

$$\Delta p = (p_s - 9.8H - p_f - p_m)/L \tag{11}$$

where

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

If the system is downfeed supply from a gravity tank, height of water in the tank, converted to kPa by multiplying by 9.8, replaces the street main pressure, and the term 9.8H 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 375 kPa; a height of topmost fixture (a urinal with flush valve) above street main of 15 m; an equivalent pipe length from water main to highest fixture of 30 m; 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 3.2 L/s. From Table 8, the minimum pressure required to operate the topmost fixture is 105 kPa. For a trial computation, choose the 40 mm meter. From Figure 8, the pressure drop through a 40 mm disk-type meter for a flow of 3.2 L/s is 45 kPa.

The pressure drop available for overcoming friction in pipes and fittings is $375 - 9.8 \times 15 - 105 - 45 = 78$ kPa.

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 15 m.

The permissible pressure loss per metre of equivalent pipe is 78/(30 + 15) = 1.7 kPa/m. A 40 mm 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 1.7 kPa/m, the size of branch (determined from Table 9 and Figures 1 and 7) is found to be 40 mm. 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	2.4 L/s	40 mm

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

Table 10 Allowable Number of 25 mm Flush Valves Served by Various Sizes of Water Pipe[a]

Pipe Size, mm	No. of 25 mm Flush Valves
32	1
40	2-4
50	5-12
65	13-25
75	26-40
100	41-100

[a]Two 20 mm flush valves are assumed equal to one 25 mm flush valve but can be served by a 25 mm pipe. Water pipe sizing must consider demand factor, available pressure, and length of run.

Velocities exceeding 3 m/s 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 1.5 m/s.

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 40 to 60 m/s, with a maximum of 75 m/s. 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.

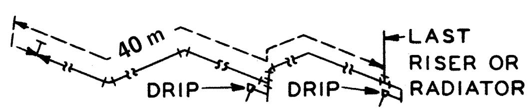
Table 11 Pressure Drops Used for Sizing Steam Pipe[a]

Initial Steam Pressure, kPa[b]	Pressure Drop, Pa/m	Total Pressure Drop in Steam Supply Piping, kPa
Vacuum return	30 to 60	7 to 14
101	7	0.4
108	30	0.4 to 1.7
115	30	3.5
135	60	10
170	115	20
205	225	30
310	450	35 to 70
445	450 to 1100	70 to 105
790	450 to 1100	105 to 170
1140	450 to 2300	170 to 210

[a]Equipment, control valves, and so forth must be selected based on delivered pressures.
[b]Subtract 101 to convert to pressure above atmospheric.

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 length of pipe for the run illustrated.



Measured length=	40 m
100 mm gate valve=	0.6 m
Four 100 mm elbows=	10.8 m
Two 100 mm tees=	11 m
Equivalent=	62.4 m

Sizing Charts

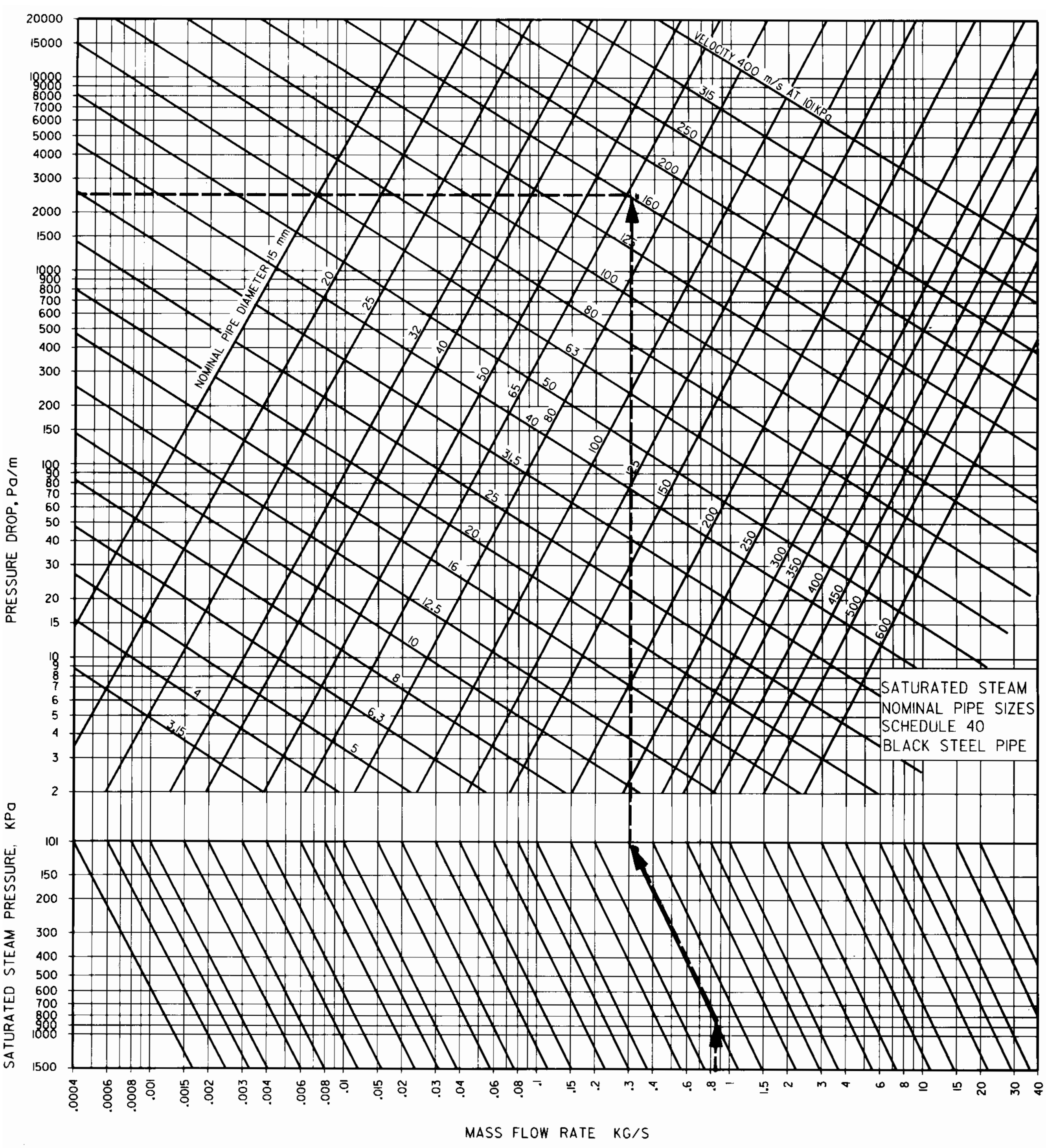
Figure 10 is the basic chart for determining the flow rate and velocity of steam in Schedule 40 pipe for various values of pressure drop per unit length, based on saturated steam at standard pressure (101.325 kPa). Using the multiplier chart (Figure 11), Figure 10 can be used at all saturation pressures between 101 and 1500 kPa (see Example 10).

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

Pitch of Pipe, mm/m	20 Capacity	20 Maximum Velocity	25 Capacity	25 Maximum Velocity	32 Capacity	32 Maximum Velocity	40 Capacity	40 Maximum Velocity	50 Capacity	50 Maximum Velocity
20	0.4	2.4	0.9	2.7	1.5	3.4	2.5	3.7	5.4	4.6
40	0.5	3.4	1.1	3.7	2	4.3	3.3	4.9	6.8	5.5
80	0.7	4.0	1.5	4.6	2.5	5.2	4.2	5.8	8.7	7.3
120	0.8	4.3	1.6	5.2	3.1	6.1	4.7	6.7	10.5	8.2
170	0.9	4.9	1.9	5.8	3.4	6.7	5.3	7.3	11.7	9.1
250	1.0	5.2	2.2	6.7	3.9	7.6	5.9	7.9	12.5	9.8
350	1.2	6.7	2.4	7.3	4.2	7.9	6.4	8.5	12.9	9.8
420	1.3	6.7	2.6	7.6	4.9	9.4	7.5	10.1	14.5	10.1

(Nominal Pipe Diameter, mm: 20, 25, 32, 40, 50)

Source: Laschober et al. (1966). Capacity in g/s; velocity in m/s.



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 101 and 1500 kPa; see also Examples 9 and 10.

Fig. 10 Flow Rate and Velocity of Steam in Schedule 40 Pipe at Saturation Pressure of 101 kPa [0 kPa (gage)]

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 25 and 85 kPa (gage). The flow rates shown for 25 kPa can be used for saturated pressures from 7 to 41 kPa, and those shown for 85 kPa can be used for saturated pressures from 55 to 110 kPa 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 metre of equivalent length.

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

Nominal Pipe Diameter, mm	Length to Be Added to Run, m				
	Standard Elbow	Side Outside Tee[b]	Gate Valve[a]	Globe Valve[a]	Angle Valve[a]
15	0.4	0.9	0.1	4	2
20	0.5	1.2	0.1	5	3
25	0.7	1.5	0.1	7	4
32	0.9	1.8	0.2	9	5
40	1.1	2.1	0.2	10	6
50	1.3	2.4	0.3	14	7
65	1.5	3.4	0.3	16	8
80	1.9	4.0	0.4	20	10
100	2.7	5.5	0.6	28	14
125	3.3	6.7	0.7	34	17
150	4.0	8.2	0.9	41	20
200	5.2	11	1.1	55	28
250	6.4	14	1.4	70	34
300	8.2	16	1.7	82	40
350	9.1	19	1.9	94	46

[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.

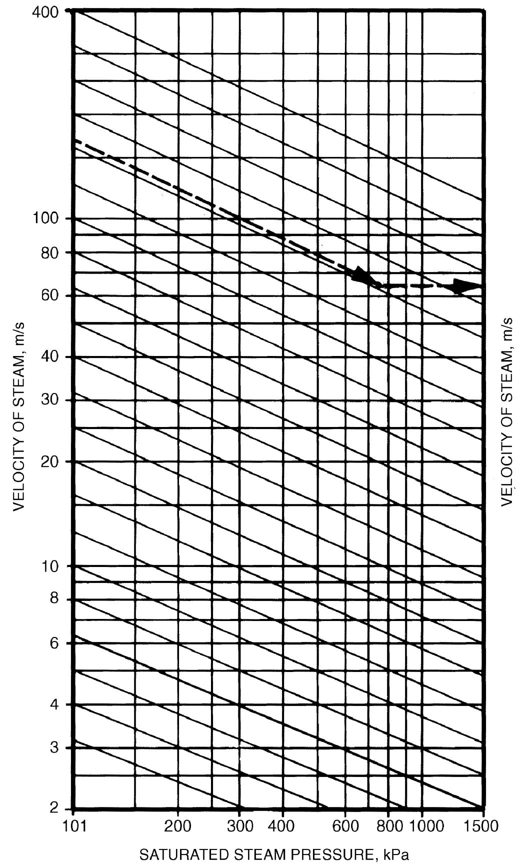


Fig. 11 Velocity Multiplier Chart for Figure 10

Table 14 Flow Rate of Steam in Schedule 40 Pipe

Nominal Pipe Size, mm	Pressure Drop, Pa/m													
	14 Pa/m		28 Pa/m		58 Pa/m		113 Pa/m		170 Pa/m		225 Pa/m		450 Pa/m	
	Sat. Press., kPa		Sat. Press., kPa		Sat. Press., kPa		Sat. Press., kPa		Sat. Press., kPa		Sat. Press., kPa		Sat. Press., kPa	
	25	85	25	85	25	85	25	85	25	85	25	85	25	85
20	1.1	1.4	1.8	2.0	2.5	3.0	3.7	4.4	4.5	5.4	5.3	6.3	7.6	9.2
25	2.1	2.6	3.3	3.9	4.7	5.8	6.8	8.3	8.6	10	10	12	14	17
32	4.5	5.7	6.7	8.3	9.8	12	14	17	18	21	20	25	29	35
40	7.1	8.8	11	13	15	19	22	26	27	33	31	38	45	54
50	14	17	20	24	29	36	42	52	53	64	60	74	89	107
65	22	27	33	39	48	58	68	83	86	103	98	120	145	173
80	40	48	59	69	83	102	121	146	150	180	174	210	246	302
90	58	69	84	101	125	153	178	214	219	265	252	305	372	435
100	81	101	120	146	178	213	249	302	309	378	363	436	529	617
125	151	180	212	265	307	378	450	536	552	662	643	769	945	1 080
150	242	290	355	422	499	611	718	857	882	1 080	1 060	1 260	1 500	1 790
200	491	605	702	882	1 020	1 260	1 440	1 800	1 830	2 230	2 080	2 580	3 020	3 720
250	907	1 110	1 290	1 590	1 890	2 290	2 650	3 280	3 300	4 030	3 780	4 660	5 380	6 550
300	1 440	1 730	2 080	2 460	2 950	3 580	4 160	5 040	5 170	6 240	6 050	7 250	8 540	10 200

Notes:
1. Flow rate is in g/s at initial saturation pressures of 25 and 85 kPa (gage). Flow is based on Moody friction factor, where the flow of condensate does not inhibit the flow of steam.
2. The flow rates at 25 kPa cover saturated pressure from 7 to 41 kPa, and the rates at 85 kPa cover saturated pressure from 55 to 110 kPa 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, mm	Capacity, g/s — Two-Pipe System — Condensate Flowing Against Steam — Vertical	Capacity, g/s — Two-Pipe System — Condensate Flowing Against Steam — Horizontal	Capacity, g/s — One-Pipe Systems — Supply Risers Upfeed	Capacity, g/s — One-Pipe Systems — Radiator Valves and Vertical Connections	Capacity, g/s — One-Pipe Systems — Radiator and Riser Runouts
A	B[a]	C[b]	D[c]	E	F[b]
20	1.0	0.9	0.8	—	0.9
25	1.8	1.8	1.4	0.9	0.9
32	3.9	3.4	2.5	2.0	2.0
40	6.0	5.3	4.8	2.9	2.0
50	12	11	9.1	5.3	2.9
65	20	17	14	—	5.3
80	36	25	25	—	8.2
90	49	36	36	—	15
100	64	54	48	—	23
125	132	99	—	—	35
150	227	176	—	—	69
200	472	378	—	—	—
250	882	718	—	—	—
300	1450	1200	—	—	—
400	2770	2390	—	—	—

Notes:
1. For one- or two-pipe systems in which condensate flows against the steam flow.
2. Steam at average pressure of 7 kPa (gage) is used as a basis of calculating capacities.

[a]Do not use Column B for pressure drops of less than 13 Pa per metre of equivalent run. Use Figure 10 or Table 13 instead.
[b]Pitch of horizontal runouts to risers and radiators should be not less than 40 mm/m. Where this pitch cannot be obtained, runouts over 2.5 m 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 9 Pa per metre of equivalent run, except on sizes 80 mm 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 150 m, and the initial pressure must not exceed 14 kPa above atmospheric?

Solution: It is assumed, if the measured length of the longest run is 150 m, that when the allowance for fittings is added, the equivalent length of run does not exceed 300 m. Then, with the pressure drop not over one-half of the initial pressure, the drop could be 7 kPa or less. With a pressure drop of 7 kPa and a length of run of 300 m, the drop would be 23 Pa/m; if the total drop were 3.5 kPa, the drop would be 12 Pa/m. 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 [100 to 1000 kPa (gage)]. 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 0.85 kg/s, an initial steam pressure of 800 kPa, and a pressure drop of 2.5 kPa/m, 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 0.85 kg/s, and move vertically to the horizontal line at 800 kPa.

Table 16 Return Main and Riser Capacities for Low-Pressure Systems, g/s

	Pipe Size, mm	7 Pa/m Wet	7 Pa/m Dry	7 Pa/m Vac.	9 Pa/m Wet	9 Pa/m Dry	9 Pa/m Vac.	14 Pa/m Wet	14 Pa/m Dry	14 Pa/m Vac.	28 Pa/m Wet	28 Pa/m Dry	28 Pa/m Vac.	57 Pa/m Wet	57 Pa/m Dry	57 Pa/m Vac.	113 Pa/m Wet	113 Pa/m Dry	113 Pa/m Vac.
	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y
Return Main	20	—	—	—	—	—	5	—	—	13	—	—	18	—	—	25	—	—	36
Return Main	25	16	8	—	18	9	18	22	10	22	32	13	31	44	14	44	—	—	62
Return Main	32	27	16	—	31	19	31	38	21	38	54	27	54	76	30	76	—	—	107
Return Main	40	43	26	—	50	30	49	60	33	60	85	43	85	120	48	120	—	—	169
Return Main	50	88	59	—	102	67	103	126	72	126	176	93	179	252	104	252	—	—	357
Return Main	65	149	96	—	199	109	171	212	120	212	296	155	300	422	171	422	—	—	596
Return Main	80	237	184	—	268	197	275	338	221	338	473	284	479	674	315	674	—	—	953
Return Main	90	347	248	—	416	277	410	504	315	504	693	407	716	1010	451	1010	—	—	1424
Return Main	100	489	369	—	577	422	567	693	473	693	977	609	984	1390	678	1390	—	—	1953
Return Main	125	—	—	—	—	—	993	—	—	1220	—	—	1730	—	—	2440	—	—	3440
Return Main	150	—	—	—	—	—	1590	—	—	1950	—	—	2770	—	—	3910	—	—	5519
Riser	20	—	6	—	—	6	18	—	6	22	—	6	31	—	6	44	—	—	62
Riser	25	—	14	—	—	14	31	—	14	38	—	14	54	—	14	76	—	—	107
Riser	32	—	31	—	—	31	49	—	31	60	—	31	85	—	31	120	—	—	169
Riser	40	—	47	—	—	47	103	—	47	126	—	47	179	—	47	252	—	—	357
Riser	50	—	95	—	—	95	171	—	95	212	—	95	300	—	95	422	—	—	596
Riser	65	—	—	—	—	—	275	—	—	338	—	—	479	—	—	674	—	—	953
Riser	80	—	—	—	—	—	410	—	—	504	—	—	716	—	—	1010	—	—	1424
Riser	90	—	—	—	—	—	564	—	—	693	—	—	984	—	—	1390	—	—	1953
Riser	100	—	—	—	—	—	993	—	—	1220	—	—	1730	—	—	2440	—	—	3440
Riser	125	—	—	—	—	—	1590	—	—	1950	—	—	2772	—	—	3910	—	—	5519

2. Follow along inclined multiplier line (upward and to the left) to horizontal 101 kPa line. The equivalent mass flow at 101 kPa is about 0.30 kg/s.
3. Follow the 0.30 kg/s line vertically until it intersects the horizontal line at 2500 Pa/m pressure drop. Nominal pipe size is 65 mm. The equivalent steam velocity at 101 kPa is about 165 m/s.
4. To find the steam velocity at 800 kPa, locate the value of 165 m/s on the ordinate of the velocity multiplier chart (Figure 11) at 101 kPa.
5. Move along the inclined multiplier line (downward and to the right) until it intersects the vertical 800 kPa pressure line. The velocity is about 65 m/s.

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.

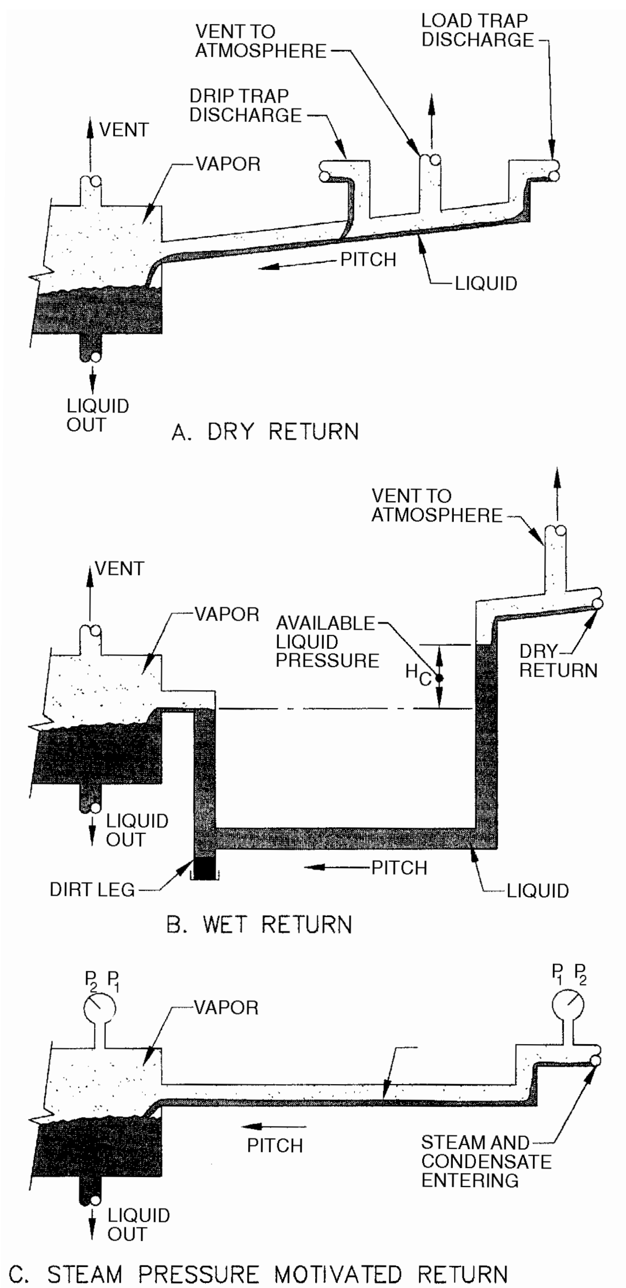


Fig. 12 Types of Condensate Return Systems

Two-Pipe Systems

When steam is used for heating a liquid to 102°C 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**:

$$Q = \frac{1.00Ar^{2/3}S^{1/2}}{n} \qquad (12)$$

where

Q = volumetric flow rate, m^3/s
A = cross-sectional area of conduit, m^2
r = hydraulic radius of conduit, m
n = coefficient of roughness (usually 0.012)
S = slope of conduit, m/m

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, mm	Condensate Flow, g/s[a,b] Condensate Line Slope 0.5%	1%	2%	4%
15	5	7	10	13
20	10	14	20	29
25	19	27	39	54
32	40	57	80	113
40	60	85	121	171
50	117	166	235	332
65	189	267	377	534
80	337	476	674	953
100	695	983	1390	1970
125	1270	1800	2540	3590
150	2070	2930	4150	5860

[a] Flow is in g/s of 82°C 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 pressure difference between the entering section of the flooded line and the leaving section. It is common practice, in addition to providing for the fluid pressure 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 pressures. 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, mm	Condensate Flow, g/s[a,b]							
	Condensate Pressure, Pa/m							
	50	100	150	200	250	300	350	400
15	13	19	24	28	32	35	38	41
20	28	41	51	60	68	74	81	87
25	54	79	98	114	129	142	154	165
32	114	165	204	238	267	294	318	341
40	172	248	308	358	402	442	479	513
50	334	482	597	694	779	857	928	994
65	536	773	956	1 110	1 250	1 370	1 480	1 590
80	954	1 370	1 700	1 970	2 210	2 430	2 630	2 810
100	1 960	2 810	3 470	4 030	4 520	4 960	5 370	5 750
125	3 560	5 100	6 290	7 290	8 180	8 980	9 720	10 400
150	5 770	8 270	10 200	11 800	13 200	14 500	15 700	16 800

[a] Flow is in g/s of 82°C 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, mm	Supply Pressure = 35 kPa Return Pressure = 0 kPa			Supply Pressure = 100 kPa Return Pressure = 0 kPa			Supply Pressure = 210 kPa Return Pressure = 0 kPa			Supply Pressure = 340 kPa Return Pressure = 0 kPa		
	$\Delta p/L$, Pa/m											
	15	60	240	15	60	240	15	60	240	15	60	240
	Flow Rate, g/s											
15	30	66	139	12	26	57	8	16	35	5	12	25
20	64	141	302	26	57	120	16	35	74	11	25	53
25	126	271	572	50	108	229	32	67	141	23	48	101
32	265	567	1 200	106	227	479	66	140	295	47	101	212
40	399	854	1 790	160	343	718	98	210	442	71	151	318
50	786	1 680	a	315	670	a	194	412	a	140	296	a
65	1 260	2 680	a	508	1 070	a	312	662	a	224	476	a
80	2 270	4 790	a	907	1 920	a	559	1 180	a	402	848	a
100	4 690	9 830	a	1 880	3 940	a	1 160	2 420	a	839	1 740	a
150	13 900	a	a	5 580	a	a	3 440	a	a	2 470	a	a
200	28 800	a	a	11 600	a	a	7 110	a	a	5 100	a	a

Pipe Dia. D, mm	Supply Pressure = 690 kPa Return Pressure = 0 kPa			Supply Pressure = 1030 kPa Return Pressure = 0 kPa			Supply Pressure = 690 kPa Return Pressure = 100 kPa			Supply Pressure = 1030 kPa Return Pressure = 100 kPa		
	$\Delta p/L$, Pa/m											
	15	60	240	15	60	240	15	60	240	15	60	240
	Flow Rate, g/s											
15	4	8	17	3	6	14	7	15	33	5	12	25
20	8	17	37	6	14	29	15	33	71	12	25	53
25	15	33	69	13	26	57	30	63	134	23	49	101
32	32	68	142	25	55	117	63	134	277	48	101	212
40	48	102	214	39	83	176	95	202	418	72	152	315
50	95	200	a	77	164	a	185	391	813	141	296	617
65	151	321	a	123	265	a	299	630	1 300	227	476	983
80	272	573	a	222	467	a	533	1 120	a	403	845	a
100	562	1180	a	459	961	a	1 100	2 290	a	834	1 740	a
150	1 660	a	a	1 360	a	a	3 260	6 750	a	2 470	5 120	a
200	3 450	a	a	2 820	a	a	6 730	13 900	a	5 100	10 500	a

[a]For these sizes and pressure losses, the velocity is above 35 m/s. 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

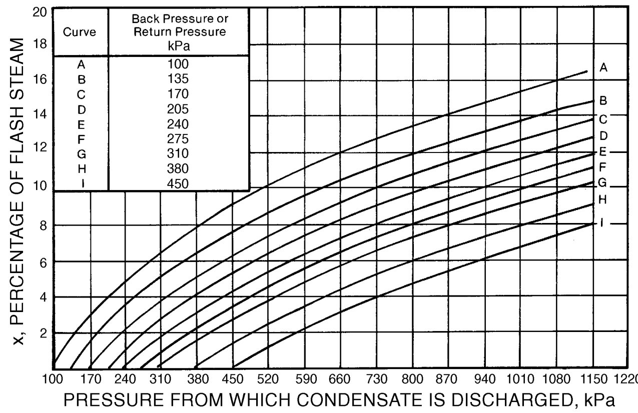


Fig. 13 Working Chart for Determining Percentage of Flash Steam (Quality)

Table 20 Flash Steam from Steam Trap on Pressure Drop

Supply Pressure, kPa (gage)	Return Pressure, kPa (gage)	x, Fraction Vapor, Mass Basis	V_c, Fraction Vapor, Volume Basis
35	0	0.016	0.962
103	0	0.040	0.985
207	0	0.065	0.991
345	0	0.090	0.994
690	0	0.133	0.996
1030	0	0.164	0.997
690	103	0.096	0.989
1030	103	0.128	0.992

Table 21 Estimated Return Line Pressures

Pressure Drop, Pa/m	Pressure in Return Line, Pa (gage)	
	200 kPa (gage) Supply	1000 kPa (gage) Supply
30	3.5	9
60	7	18
120	14	35
180	21	52
240	28	70
480	—	138

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 Litres per Second

Nominal Iron Pipe Size, mm	Internal Diameter, mm	Length of Pipe, m											
		5	10	15	20	25	30	35	40	45	50	55	60
8	9.25	0.19	0.13	0.11	0.09	0.08	0.07	0.07	0.06	0.06	0.06	0.05	0.05
10	12.52	0.43	0.29	0.24	0.20	0.18	0.16	0.15	0.14	0.13	0.12	0.12	0.11
15	15.80	0.79	0.54	0.44	0.37	0.33	0.30	0.28	0.26	0.24	0.23	0.22	0.21
20	20.93	1.65	1.13	0.91	0.78	0.69	0.63	0.58	0.54	0.50	0.47	0.45	0.43
25	26.14	2.95	2.03	1.63	1.40	1.24	1.12	1.03	0.96	0.90	0.85	0.81	0.77
32	35.05	6.4	4.4	3.5	3.0	2.7	2.4	2.2	2.1	1.9	1.8	1.7	1.7
40	40.89	9.6	6.6	5.3	4.5	4.0	3.6	3.3	3.1	2.9	2.8	2.6	2.5
50	52.50	18.4	12.7	10.2	8.7	7.7	7.0	6.4	6.0	5.6	5.3	5.0	4.8
65	62.71	29.3	20.2	16.2	13.9	12.3	11.1	10.2	9.5	8.9	8.4	8.0	7.7
80	77.93	51.9	35.7	28.6	24.5	21.7	19.7	18.1	16.8	15.8	14.9	14.2	13.5
100	102.26	105.8	72.7	58.4	50.0	44.3	40.1	36.9	34.4	32.2	30.4	28.9	27.6

Note: Capacity is in litres per second at gas pressures of 3.5 kPa (gage) or less and a pressure drop of 75 kPa; density = 0.735 kg/m^3.

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) density of the gas.

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 energy 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 60 m of pipe based on a density of 0.735 kg/m^3. Capacities for pressures less than 10 kPa may also be determined by the following equation from NFPA/IAS *National Fuel Gas Code*:

$$Q = 0.0001 d^{2.623} (\Delta p / CL)^{0.541} \tag{17}$$

where

Q = flow rate at 15°C and 101 kPa, L/s
d = inside diameter of pipe, mm
Δp = pressure drop, Pa
C = factor for viscosity, density, and temperature
  = $0.00223(t + 273)s^{0.848}\mu^{0.152}$
t = temperature, °C
s = ratio of density of gas to density of air at 15°C and 101 kPa
μ = viscosity of gas, μPa·s (12 for natural gas, 8 for propane)
L = pipe length, m

Gas service in buildings is generally delivered in the "low-pressure" range of 1.7 kPa (gage). The maximum pressure drop allowable in piping systems at this pressure is generally 125 Pa 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., 15, 35, or 70 kPa (gage)]. 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 34 kPa for distillate grade fuels and 50 kPa 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 34 kPa, or pump shaft seals may fail. Figure 14 shows a typical oil circulating loop.

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 70 kPa. 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.

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.

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.

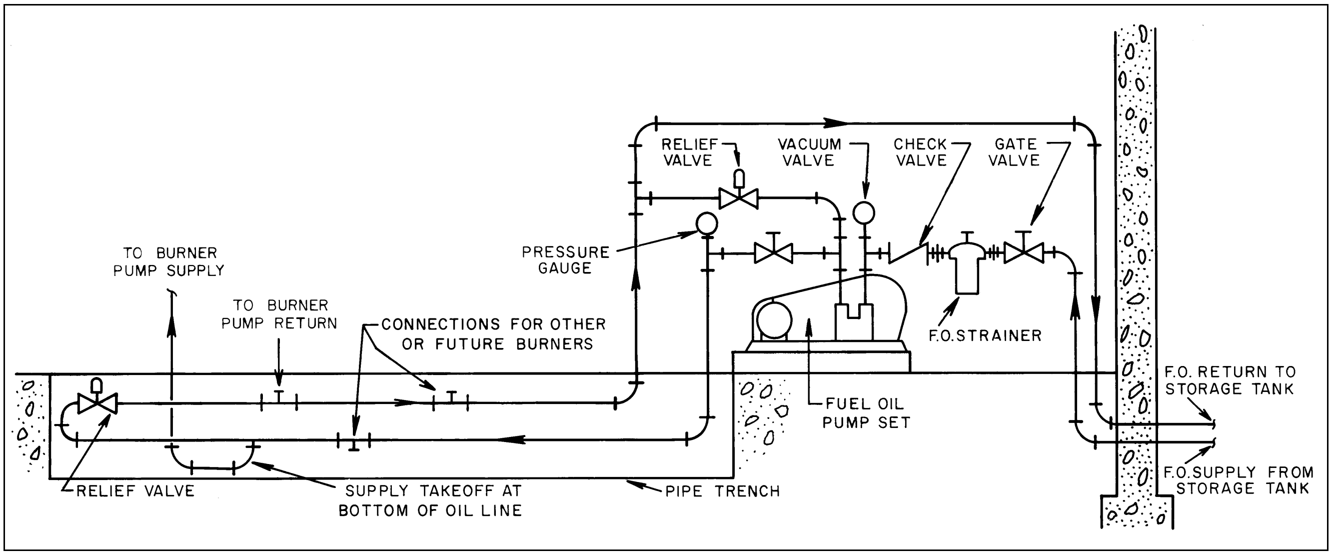


Fig. 14 Typical Oil Circulating Loop

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

Pumping Rate, L/h	Length of Run in Metres at Maximum Suction Lift of 4.5 kPa									
	10	20	30	40	50	60	70	80	90	100
50	40	40	40	50	50	50	65	65	65	80
100	40	40	50	50	65	65	65	65	80	80
200	40	50	50	50	65	65	65	80	80	80
300	50	50	65	65	65	80	80	80	80	80
400	50	50	65	65	80	80	80	80	80	100
500	50	65	65	65	80	80	80	80	100	100
600	65	65	65	80	80	80	100	100	100	100
700	65	65	65	80	80	100	100	100	100	100
800	65	65	80	80	100	100	100	100	100	100

Notes:
1. Sizes (in millimetres) are nominal.
2. Pipe sizes smaller than 25 mm ISO are not recommended for use with residual grade fuel oils.
3. 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, L/h	Length of Run in Metres at Maximum Suction Lift of 9.0 kPa									
	10	20	30	40	50	60	70	80	90	100
50	15	15	15	15	15	20	20	20	25	25
100	15	15	15	15	20	20	20	20	25	25
200	15	20	20	20	20	20	25	25	25	25
300	15	20	20	20	20	25	25	25	25	32
400	20	20	20	20	25	25	25	25	32	32
500	20	25	25	25	25	25	32	32	32	32
600	20	25	25	25	25	32	32	32	32	50
700	20	25	25	25	25	32	32	32	50	50
800	20	25	25	25	32	32	32	32	50	50

Note: Sizes (in millimetres) are nominal.

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