Liquid Process Piping -
Part 1: General Piping Design

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Chapter 3
General Piping Design

3-1. Materials of Construction

Most failures of liquid process systems occur at or within interconnect points - - the piping, flanges, valves, fittings, etc. It is, therefore, vital to select interconnecting equipment and materials that are compatible with each other and the expected environment. Materials selection is an optimization process, and the material selected for an application must be chosen for the sum of its properties. That is, the selected material may not rank first in each evaluation category; it should, however, be the best overall choice. Considerations include cost and availability. Key evaluation factors are strength, ductility, toughness, and corrosion resistance.

a. Strength

The strength of a material is defined using the following properties: modulus of elasticity, yield strength, and ultimate tensile strength. All of these properties are determined using ASTM standard test methods.

The modulus of elasticity is the ratio of normal stress to the corresponding strain for either tensile or compressive stresses. Where the ratio is linear through a range of stress, the material is elastic; that is, the material will return to its original, unstressed shape once the applied load is removed. If the material is loaded beyond the elastic range, it will begin to deform in a plastic manner. The stress at that deformation point is the yield strength. As the load is increased beyond the yield strength, its cross-sectional area will decrease until the point at which the material cannot handle any further load increase. The ultimate tensile strength is that load divided by the original cross-sectional area.

b. Ductility

Ductility is commonly measured by either the elongation in a given length or by the reduction in cross-sectional area when subjected to an applied load. The hardness of a material is a measure of its ability to resist deformation. Hardness is often measured by either of two standard scales, Brinell and Rockwell hardness.

c. Toughness

The toughness of a material is dependent upon both strength and ductility. Toughness is the capability of a material to resist brittle fracture (the sudden fracture of materials when a load is rapidly applied, typically with little ductility in the area of the fracture). Two common ASTM test methods used to measure toughness are the Charpy Impact and Drop-Weight tests. The Charpy brittle transition temperature and the Drop-Weight NDTT are important design parameters for materials that have poor toughness and may have lower operating temperatures. A material is subject to brittle, catastrophic failure if used below the transition temperature.

d. Corrosion Resistance

Appendix B provides a matrix that correlates process fluids, piping materials and maximum allowable process temperatures to assist in determining material suitability for applications.

e. Selection Process

Piping material is selected by optimizing the basis of design. First, eliminate from consideration those piping materials that:

- are not allowed by code or standard;
- are not chemically compatible with the fluid;
-have system rated pressure or temperatures that do not meet the full range of process operating conditions; and
- are not compatible with environmental conditions such as external corrosion potential, heat tracing requirements, ultraviolet degradation, impact potential and specific joint requirements.

The remaining materials are evaluated for advantages and disadvantages such as capital, fabrication and installation costs; support system complexity; compatibility to handle thermal cycling; and cathodic protection requirements. The highest ranked material of construction is then selected. The design proceeds with pipe sizing, pressure-integrity calculations and stress analyses. If the selected piping material does not meet those requirements, then the second ranked material is used and the pipe sizing, pressure-integrity calculations and stress analyses are repeated.

Example Problem 1:
Assume a recovered material process line that handles nearly 100% ethyl benzene at 1.20 MPa (174 psig) and 25°C (77°F) is required to be installed above ground. The piping material is selected as follows:

Solution:
Step 1. Above ground handling of a flammable liquid by thermoplastic piping is not allowed by ASME B31.3[1].

Step 2. Review of the Fluid/Material Corrosion Matrix (Appendix B) for ethyl benzene at 25°C (77°F) indicates that aluminum, Hastelloy C, Monel, TP316 stainless steel, reinforced furan resin thermoset and FEP lined pipe are acceptable for use. FKM is not available in piping.

Step 3. Reinforced furan resin piping is available to a system pressure rating of 689 kPa (100 psig)[2]; therefore, this material is eliminated from consideration. The remainder of the materials have available system pressure ratings and material allowable stresses greater than the design pressure.

Step 4. FEP lined piping is not readily available commercially. Since other material options exist, FEP lined piping is eliminated from consideration.

Step 5. The site specific environmental conditions are now evaluated to determine whether any of the remaining materials (aluminum, Hastelloy C, Monel or TP316 stainless steel) should be eliminated prior to ranking. The material is then selected based on site specific considerations and cost.

3-2. Design Pressure

After the piping system's functions, service conditions, materials of construction and design codes and standards have been established (as described in Chapter 2) the next step is to finalize the system operational pressures and temperatures. Up to this point, the system operating pressure has been addressed from a process requirement viewpoint to ensure proper operation of the system as a whole. At this point in the detail design of the piping system, it is necessary to ensure that the structural integrity of the pipe and piping system components is maintained during both normal and upset pressure and temperature conditions. In order to select the design pressure and temperature, it is necessary to have a full understanding and description of all operating processes and control system functions. The pressure rating of a piping system is determined by identifying the maximum steady state pressure, and determining and allowing for pressure transients.

a. Maximum Steady State Pressure

The determination of maximum steady state design pressure and temperature is based on an evaluation of specific operating conditions. The evaluation of conditions must consider all modes of operation. This is typically accomplished utilizing design references, codes and standards. An approach using the code requirements of ASME B31.3 for maximum pressure and temperature loads is used herein for demonstration.

Piping components shall be designed for an internal pressure representing the most severe condition of coincident pressure and temperature expected in normal operation.[3] This condition is by definition the one which results in the greatest required pipe thickness and the highest flange rating. In addition to hydraulic conditions based on operating pressures, potential back pressures, surges in pressures or temperature fluctuations, control system performance variations and process upsets must be considered. The system must also be evaluated and designed for the maximum external differential pressure conditions.

Piping components shall be designed for the temperature representing the most severe conditions described as follows:

- for fluid temperatures below 65°C (150°F), the metal design temperature of the pipe and components shall be taken as the fluid temperature.

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1 ASME B31.3, p. 95.
2 Schweitzer, Corrosion-Resistant Piping Systems, p. 140.
3 ASME B31.3, p. 11.

- for fluid temperatures above 65°C (150°F), the metal design temperature of uninsulated pipe and components shall be taken as 95% of the fluid temperature, except flanges, lap joint flanges and bolting shall be 90%, 85% and 80% of the fluid temperature, respectively.
- for insulated pipe, the metal design temperature of the pipe shall be taken as the fluid temperature unless calculations, testing or experience based on actual field measurements can support the use of other temperatures.
- for insulated and heat traced pipe, the effect of the heat tracing shall be included in the determination of the metal design temperature.[4]

In addition to the impact of elevated temperatures on the internal pressure, the impact of cooling of gases or vapors resulting in vacuum conditions in the piping system must be evaluated.

b. Pressure Transients

As discussed in Paragraph 2-5, short-term system pressure excursions are addressed either through code defined limits or other reasonable approaches based on experience. The ASME B31.3 qualification of acceptable pressure excursions states:

"*302.2.4 Allowances for Pressure and Temperature Variations. Occasional variations of pressure or temperature, or both, above operating levels are characteristic of certain services. The most severe conditions of coincident pressure and temperature during the variation shall be used to determine the design conditions unless all of the following criteria are met.*

(a) The piping system shall have no pressure containing components of cast iron or other nonductile metal.

(b) Nominal pressure stresses shall not exceed the yield strength at temperature (see para. 302.3 of this Code [ASME B31.3] and Sy data in [ASME] BPV Code, Section II, Part D, Table Y-1).

(c) Combined longitudinal stress shall not exceed the limits established in paragraph 302.3.6 [of ASME B31.3].

(d) The total number of pressure-temperature variations above the design conditions shall not exceed 1000 during the life of the piping system.

(e) In no case shall the increased pressure exceed the test pressure used under para. 345 [of ASME B31.3] for the piping system.

(f) Occasional variations above design conditions shall remain within one of the following limits for pressure design.

(1) Subject to the owner's approval, it is permissible to exceed the pressure rating or the allowable stress for pressure design at the temperature of the increased condition by not more than:

(a) 33% for no more than 10 hour at any one time and no more than 100 hour per year; or

(b) 20% for no more than 50 hour at any one time and no more than 500 hour per year.

The effects of such variations shall be determined by the designer to be safe over the service life of the piping system by methods acceptable to the owner. (See Appendix V [of ASME B31.3])

(2) When the variation is self-limiting (e.g., due to a pressure relieving event), and lasts no more than 50 hour at any one time and not more than 500 hour/year, it is permissible to exceed the pressure rating or the allowable stress for pressure design at the temperature of the increased condition by not more than 20%.

(g) The combined effects of the sustained and cyclic variations on the serviceability of all components in the system shall have been evaluated.

(h) Temperature variations below the minimum temperature shown in Appendix A [of ASME B31.3] are not permitted unless the requirements of para. 323.2.2 [of ASME B31.3] are met for the lowest temperature during the variation.

[4] ASME B31.3, pp. 11-12.

(i) The application of pressures exceeding pressure-temperature ratings of valves may under certain conditions cause loss of seat tightness or difficulty of operation. The differential pressure on the valve closure element should not exceed the maximum differential pressure rating established by the valve manufacturer. Such applications are the owner's responsibility."[5]

The following example illustrates a typical procedure for the determination of design pressures.

Example Problem 2:
Two motor-driven boiler feed pumps installed on the ground floor of a power house supply 0.05 m^3/s (793 gpm) of water at 177°C (350°F) to a boiler drum which is 60 m (197 ft) above grade. Each pump discharge pipe is 100 mm (4 in), and the common discharge header to the boiler drum is a 150 mm (6 in) pipe. Each pump discharge pipe has a manual valve that can isolate it from the main header. A relief valve is installed upstream of each pump discharge valve to serve as a minimum flow bypass if the discharge valve is closed while the pump is operating. The back pressure at the boiler drum is 17.4 MPa (2,520 psig). The set pressure of the relief valve is 19.2 MPa (2,780 psig), and the shutoff head of each pump is 2,350 m (7,710 ft). The piping material is ASTM A 106, Grade C, with an allowable working stress of 121 MPa (17,500 psi), over the temperature range of -6.7 to 343°C (-20 to 650°F). The corrosion allowance is 2 mm (0.08 in) and the design code is ASME B31.1 (Power Piping).

The design pressures for the common discharge header and the pump discharge pipes upstream of the isolation valve must be determined. Also the maximum allowable pressure is to be calculated assuming the relief valve on a pump does not operate when its discharge valve is closed.

Solution:
Step 1. Determination of design pressure for the 150 mm (6 in) header is as follows. The specific volume of 177°C (350°F) saturated water is 0.001123 m^3/kg (0.01799 ft^3/lbm). The specific volume is corrected for the effects of compression to 17.2 MPa (2,500 psig) using steam tables:

$$\nu - \nu_f = -0.000013 \; m^3/kg \; (-0.00021 \; ft^3/lbm)$$

$$\nu_f \; at \; 177°C \; (350°F) = 0.001123 \; m^3/kg$$
$$(0.01799 \; ft^3/lbm), \; saturated$$

$$\nu \; at \; 17.2 \; MPa \; (2{,}500 \; psig)$$

$$= 0.001123 \; m^3/kg + (-0.000013 \; m^3/kg)$$

$$= 0.001110 \; m^3/kg \; (0.01778 \; ft^3/lbm),$$
$$compressed$$

where:
ν = specific volume of water, m^3/kg (ft^3/lbm)
ν_f = specific volume of feed water, m^3/kg (ft^3/lbm)

The static head above the pumps due to the elevation of the boiler drum is:

$$P_{st} = (60 \; m)\left(\frac{1}{0.001110 \; \frac{m^3}{kg}}\right)\left(9.81 \; \frac{m}{s^2}\right)$$

$$= 530 \; kPa \; (76.9 \; psig)$$

where:
P_{st} = static head, kPa (psig)

Step 2. The total discharge pressure at the pump exit is:

$$P = P_b + P_{st}$$
$$= 17.4 \; MPa + 0.530 \; MPa$$
$$= 17.9 \; MPa \; (2{,}600 \; psig)$$

where:
P = total discharge pressure, MPa (psig)
P_b = back pressure, MPa (psig)
P_{st} = static head, MPa (psig)

[5] ASME B31.3, pp. 13-14.

The design pressure for the 150 mm (6 in) header should be set slightly above the maximum operating pressure. Therefore the design pressure for the 150 mm (6 in) header is 18.3 MPa (2,650 psig).

Step 3. Determination of design pressure for the 100 mm (4 in) pipe is as follows. The set pressure of the relief valve is 19.2 MPa (2,780 psig). The design pressure of the 100 mm (4 in) pipe upstream of the pump discharge valve should be set at the relief pressure of the relief valve. Although not shown in this example, the design pressure should also take into account any over-pressure allowance in the relief valve sizing determination. Therefore, for this example, the design pressure for the 100 mm (4 in) pipe upstream of the pump isolation valves is 19.2 MPa (2,780 psig).

Step 4. The maximum allowable pressure in the 100 mm (4 in) pipe is compared to that which would be observed during relief valve failure. The probability that a valve will fail to open is low. It is recognized that variations in pressure and temperature inevitably occur.

"*102.2.4 Ratings: Allowance for Variation From Normal Operation. The maximum internal pressure and temperature allowed shall include considerations for occasional loads and transients of pressure and temperature.*"[6]

The calculated stress resulting from such a variation in pressure and/or temperature may exceed the maximum allowable stress from ASME B31.1 Appendix A by 15% if the event duration occurs less than 10% of any 24- hour operating period, or 20% if the event duration occurs less than 1% of any 24-hour operating period.[7] The occasional load criteria of ASME B31.1, paragraph 102.2.4, is applied, and it is assumed that the relief valve failure-to-open event occurs less than 1% of the time. Therefore, the allowable stress is 20% higher than the basic code allowable stress of 121 MPa (17,500 psi).

Step 5. The higher allowable stress is denoted as S':

$$S' = 1.20\ (S) = 1.20\ (121\ MPa) = 145\ MPa\ (21{,}000\ psi)$$

where:

S' = higher allowable stress, MPa (psi)
S = code allowable stress, MPa (psi)

Step 6. The maximum pressure rating of the 100 mm (4 in) pipe is calculated using the following equation[8]:

$$P_{max} = \frac{2\ S\ E\ (t_m - A)}{D_o - 2\ y\ (t_m - A)}$$

where:

P_{max} = maximum allowable pressure, MPa (psig)
S = code allowable stress, MPa (psi)
E = joint efficiency
t_m = pipe wall thickness, mm (in)
A = corrosion allowance, mm (in)
D_o = outside diameter of pipe, mm (in)
y = temperature-based coefficient, see ASME B31.1, for cast iron, non-ferrous metals, and for ferric steels, austenitic steels and Ni alloys less than 482°C (900°F), y = - 0.4.

Step 7. For this example, the value of S is set to equal to S' and E = 1.00 for seamless pipe. The pipe wall thickness is determined in accordance to pressure integrity, see Paragraph 3-3b, and is assumed equal to 87½% of the nominal wall thickness of schedule XXS pipe. Therefore:

$$t_m = 17.1\ mm\ (0.875) = 15.0\ mm\ (0.590\ in)$$

where

t_m = pipe wall thickness, mm (in)

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6 ASME B31.1, p. 13.
7 Ibid., p. 13.
8 Ibid., p. 17.

and

$$P_{max} = \frac{2(145\ MPa)(1.0)(15.0\ mm - 2\ mm)}{114.3\ mm - 2(0.4)(15.0\ mm - 2\ mm)}$$

$$= 36.3\ MPa\ (5{,}265\ psig)$$

where:
P_{max} = maximum allowable pressure, MPa (psig)

Step 8. Therefore, the maximum allowable pressure in the 100 mm (4 in) pipe section during a relief valve failure is 36.3 MPa (5,265 psig).

Another common transient pressure condition is caused by suddenly reducing the liquid flow in a pipe. When a valve is abruptly closed, dynamic energy is converted to elastic energy and a positive pressure wave is created upstream of the valve. This pressure wave travels at or near the speed of sound and has the potential to cause pipe failure. This phenomenon is called water hammer.

The maximum pressure rise is calculated by:

$$P_i = \rho\ \Delta V\ V_w\ n_1$$

where:
P_i = maximum pressure increase, MPa (psi)
ρ = fluid density, kg/m^3 (slugs/ft^3)
ΔV = sudden change in liquid velocity, m/s (ft/s)
V_w = pressure wave velocity, m/s (ft/s)
n_1 = conversion factor, 10^{-6} MPa/Pa for SI units (1 ft^2/144 in^2 for IP units)

The maximum time of valve closure that is considered sudden (critical) is calculated by:

$$t_c = \frac{2\ L}{V_w}$$

where:
t_c = critical time, s
L = length of pipe, m (ft)
V_w = pressure wave velocity, m/s (ft/s)

The velocity of the pressure wave is affected by the fluid properties and by the elasticity of the pipe. The pressure wave velocity in water is approximately 1,480 m/s (4,800 ft/s). For a rigid pipe, the pressure wave velocity is calculated by:

$$V_w = \left(\frac{E_s}{n_1\ \rho} \right)^{1/2}$$

where:
V_w = pressure wave velocity, m/s (ft/s)
E_s = fluid's bulk modulus of elasticity, MPa (psi)
ρ = fluid density, kg/m^3 (slugs/ft^3)
n_1 = conversion factor, 10^{-6} MPa/Pa for SI units (1 ft^2/144 in^2 for IP units)

Because of the potential expansion of an elastic pipe, the pressure wave for an elastic pipe is calculated by:

$$V_w = \left(\frac{E_s}{n_1\ \rho \left(1 + \frac{E_s\ D_i}{E_p\ t} \right)} \right)^{1/2}$$

where:
V_w = pressure wave velocity, m/s (ft/s)
E_s = fluid's bulk modulus of elasticity, MPa (psi)
ρ = fluid density, kg/m^3 (slugs/ft^3)
E_p = bulk modulus of elasticity for piping material, MPa (psi)
D_i = inner pipe diameter, mm (in)
t = pipe wall thickness, mm (in)
n_1 = conversion factor, 10^{-6} MPa/Pa for SI units (1 ft^2/144 in^2 for IP units)

If the valve is slowly closed (i.e., the time of closure is greater than the critical time), a series of small pressure waves is transmitted up the pipe and returning negative pressure waves will be superimposed on the small pressure waves and full pressure will not occur. The pressure developed by gradual closure of a value is:

$$P'_i = \frac{2\ \rho\ L\ V\ n_1}{t_v}$$

where:

P'_I = pressure increase, MPa (psi)
t_v = valve closure time
ρ = fluid density, kg/m^3 (slugs/ft^3)
L = length of pipe, m (ft)
V = liquid velocity, m/s (ft/s)
n_1 = conversion factor, 10^{-6} MPa/Pa for SI units (1 ft^2/144 in^2 for IP units)

CECER has a computer program, WHAMO, designed to simulate water hammer and mass oscillation in pumping facilities. The program determines time varying flow and head in a piping network which may includevalves, pumps, turbines, surge tanks and junctions arranged in a reasonable configuration. Transients are generated in the program due to any variation in the operation of pumps, valves, and turbines, or in changes in head.

Example Problem 3:
Water at 20°C (68°F) flows from a tank at a velocity of 3 m/s (9.8 ft/s) and an initial pressure of 275 kPa (40 psi) in a 50 mm (2 in) PVC pipe rated for 16 kgf/cm^2 (SDR 26); i.e., wall thickness is 4.7 mm (0.091 in for SDR 26). A valve 150 m (492 ft) downstream is closed. Determine the critical time of closure for the valve and the internal system pressure if the valve is closed suddenly versus gradually (10 times slower).

Solution:
Step 1. Velocity of the pressure wave assuming rigid pipe;

$$V_w = \left(\frac{E_s}{n_1 \ \rho} \right)^{1/2}$$

where:

V_w = pressure wave velocity, m/s (ft/s)
E_s = fluid's bulk modulus of elasticity; for water at 20°C (68°F) = 2,180 MPa (319,000 psi)
n_1 = conversion factor, 10^{-6} MPa/Pa for SI units (1 ft^2/144 in^2 for IP units)
ρ= fluid density, for water at 20°C (68°F) = 998.2 kg/m^3 (1.937 slugs/ft^3)

$$V_w = \left(\frac{2{,}180 \ MPa}{(10^{-6} \ MPa/Pa) \ (998.2 \ kg/m^3)} \right)^{1/2}$$
$$= 1{,}478 \ m/s \ (4{,}848 \ ft/s)$$

Step 2. Critical time for valve closure;

$$t_c = \frac{2 \ L}{V_w} = \frac{2 \ (150 \ m)}{1{,}478 \ m/s}$$
$$= 0.2 \ s$$

where:

t_c = critical time, s
L = Length of pipe, m (ft)
V_w = pressure wave velocity, m/s (ft/s)

Step 3. Maximum pressure rise (valve closure time < critical time, t_c);

$$P_i = \rho \ \Delta V \ V_w \ n_1$$

where:

P_i = maximum pressure increase, MPa (psi)
ρ = fluid density, kg/m^3 (slugs/ft^3)
ΔV = sudden change in liquid velocity, m/s (ft/s)
V_w = pressure wave velocity, m/s (ft/s)
n_1 = conversion factor, 10^{-6} MPa/Pa for SI units (1 ft^2/144 in^2 for IP units)

$$P_i = \left(998.2 \frac{kg}{m^3} \right) \left(3 \frac{m}{s} \right) \left(1{,}478 \frac{m}{s} \right) \left(10^{-6} \frac{MPa}{Pa} \right)$$
$$= 4.43 \ MPa \ (642 \ psi)$$

Therefore, maximum system pressure is

$$P_{\max} = 4.43 \ MPa + 275 \ kPa \ (10^{-3} \ MPa/kPa)$$
$$= 4.71 \ MPa \ (682 \ psig)$$

Step 4. Pressure increase with gradual valve closure (valve closure time = critical time, t_c, x 10 = 2s)

$$P'_i = \frac{2\ \rho\ L\ V\ n_1}{t_v}$$

where:

P'_I = pressure increase, MPa (psi)
t_v = valve closure time
ρ = fluid density, kg/m^3 ($slugs/ft^3$)
L = length of pipe, m (ft)
V = liquid velocity, m/s (ft/s)
n_1 = conversion factor, 10^{-6} MPa/Pa for SI units (1 $ft^2/144\ in^2$ for IP units)

$$P'_i = \frac{2\left(998.2\frac{kg}{m^3}\right)(150m)\left(3\frac{m}{s}\right)}{2\ s}\left(10^{-3}\frac{kPa}{Pa}\right)$$

$$= 449\ kPa\ (65\ psi)$$

Therefore, the maximum system pressure is 449 kPa + 275 kPa = 724 kPa (105 psig).

For a more complex review of water hammer effects in pipes, refer to the references found in Appendix A, Paragraph A-4.

3-3. Sizing

The sizing for any piping system consists of two basic components fluid flow design and pressure integrity design. Fluid flow design determines the minimum acceptable diameter of the piping necessary to transfer the fluid efficiently. Pressure integrity design determines the minimum pipe wall thickness necessary to safely handle the expected internal and external pressure and loads.

a. Fluid Flow Sizing

The primary elements in determining the minimum acceptable diameter of any pipe network are system design flow rates and pressure drops. The design flow rates are based on system demands that are normally established in the process design phase of a project. Before the determination of the minimum inside diameter can be made, service conditions must be reviewed to determine operational requirements such as recommended fluid velocity for the application and liquid characteristics such as viscosity, temperature, suspended solids concentration, solids density and settling velocity, abrasiveness and corrosivity. This information is then used to determine the minimum inside diameter of the pipe for the network.

For normal liquid service applications, the acceptable velocity in pipes is 2.1 ± 0.9 m/s (7 ± 3 ft/s) with a maximum velocity limited to 2.1 m/s (7 ft/s) at piping discharge points including pump suction lines and drains. As stated, this velocity range is considered reasonable for normal applications. However, other limiting criteria such as potential for erosion or pressure transient conditions may overrule. In addition, other applications may allow greater velocities based on general industry practices; e.g., boiler feed water and petroleum liquids.

Pressure drops throughout the piping network are designed to provide an optimum balance between the installed cost of the piping system and operating costs of the system pumps. Primary factors that will impact these costs and system operating performance are internal pipe diameter (and the resulting fluid velocity), materials of construction and pipe routing.

Pressure drop, or head loss, is caused by friction between the pipe wall and the fluid, and by minor losses such as flow obstructions, changes in direction, changes in flow area, etc. Fluid head loss is added to elevation changes to determine pump requirements.

A common method for calculating pressure drop is the Darcy-Weisbach equation:

$$h_L = \left(\frac{f\ L}{D_i} + \Sigma K\right)\frac{V^2}{2\ g};\ \textit{loss coefficient method}$$

or

$$h_L = f\ \frac{(L + L_e)}{D_i}\ \frac{V^2}{2\ g};\ \textit{equivalent length method}$$

where:
- h_L = head loss, m (ft)
- f = friction factor
- L = length of pipe, m (ft)
- D_i = inside pipe diameter, m (ft)
- L_e = equivalent length of pipe for minor losses, m (ft)
- K = loss coefficients for minor losses
- V = fluid velocity, m/s (ft/sec)
- g = gravitational acceleration, 9.81 m/sec^2 (32.2 ft/sec^2)

The friction factor, f, is a function of the relative roughness of the piping material and the Reynolds number, R_e.

$$R_e = \frac{D_i \ V}{\nu}$$

where:
- R_e = Reynolds number
- D_i = inside pipe diameter, m (ft)
- V = fluid velocity, m/s (ft/s)
- ν = kinematic viscosity, m^2/s (ft^2/s)

If the flow is laminar (R_e < 2,100), then f is determined by:

$$f = \frac{64}{R_e}$$

where:
- f = friction factor
- R_e = Reynolds number

If the flow is transitional or turbulent (R_e > 2,100), then f is determined from the Moody Diagram, see Figure 3-1. The appropriate roughness curve on the diagram is determined by the ratio ϵ/D_i where ϵ is the specific surface roughness for the piping material (see Table 3-1) and D_i is the inside pipe diameter.

The method of equivalent lengths accounts for minor losses by converting each valve and fitting to the length of straight pipe whose friction loss equals the minor loss. The equivalent lengths vary by materials, manufacturer and size (see Table 3-2). The other method uses loss coefficients. This method must be used to calculate exit and entrance losses. The coefficients can be determined from Table 3-3.

Another method for calculating pressure drop is the Hazen-Williams formula:

$$h_L = (L + L_e) \left(\frac{V}{a \ C \ (D_i/4)^{0.63}} \right)^{1.85}$$

where:
- h_L = head loss, m (ft)
- L = length of pipe, m (ft)
- L_e = equivalent length of pipe for minor losses, m (ft)
- V = fluid velocity, m/s (ft/s)
- a = empirical constant, 0.85 for SI units (1.318 for IP units)
- C = Hazen-Williams coefficient
- D_i = inside pipe diameter, m (ft)

The Hazen-Williams formula is empirically derived and is limited to use with fluids that have a kinematic viscosity of approximately 1.12 x 10^{-6} m^2/s (1.22 x 10^{-5} ft^2/s), which corresponds to water at 15.6°C (60°F), and for turbulent flow. Deviations from these conditions can lead to significant error. The Hazen-Williams coefficient, C, is independent of the Reynolds number. Table 3-1 provides values of C for various pipe materials.

The Chezy-Manning equation is occasionally applied to full pipe flow. The use of this equation requires turbulent flow and an accurate estimate of the Manning factor, n, which varies by material and increases with increasing pipe size. Table 3-1 provides values of n for various pipe materials. The Chezy-Manning equation is:

$$h_L = \frac{V^2 \ n^2}{a \ (D_i/4)^{4/3}} \ (L + L_e)$$

where:
- h_L = head loss, m (ft)
- V = fluid velocity, m/s (ft/s)
- n = Manning factor
- a = empirical constant, 1.0 for SI units (2.22 for IP units)

Table 3-1
Pipe Material Roughness Coefficients

Pipe Material	Specific Roughness Factor, ϵ, mm (in)	Hazen-Williams Coefficient, C	Manning Factor, n
Steel, welded and seamless	0.061 (0.0002)	140	
Ductile Iron	0.061 (0.0002)	130	
Ductile Iron, asphalt coated	0.12 (0.0004)	130	0.013
Copper and Brass	0.61 (0.002)	140	0.010
Glass	0.0015 (0.000005)	140	
Thermoplastics	0.0015 (0.000005)	140	
Drawn Tubing	0.0015 (0.000005)		

Sources:
Hydraulic Institute, Engineering Data Book.
Various vendor data compiled by SAIC, 1998.

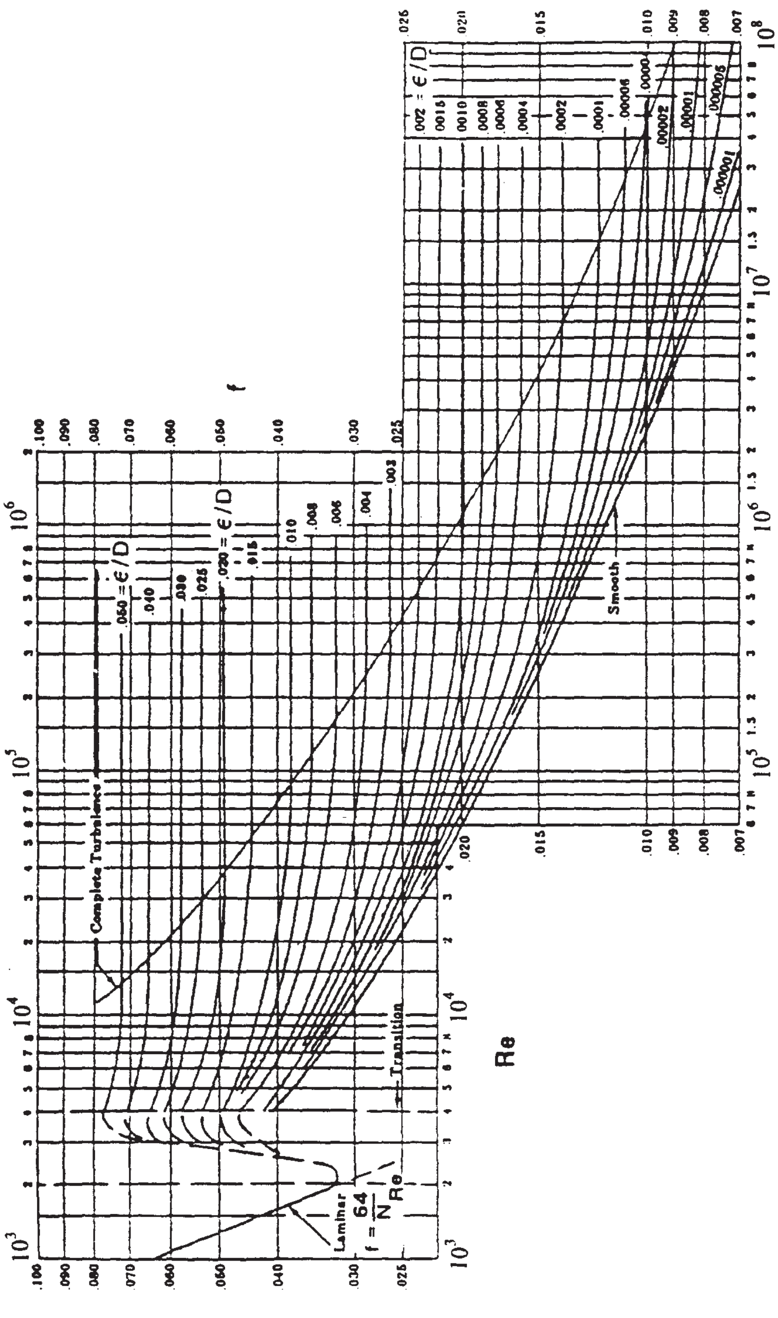

Figure 3-1. Moody Diagram
(Source: L.F. Moody, "Friction Factors for Pipe Flow," Transactions
of the ASME, Vol. 66, Nov. 1944, pp. 671-678, Reprinted by permission of ASME.)

Table 3-2
Estimated Pressure Drop for Thermoplastic Lined Fittings and Valves

Size mm (in)	Standard 90° elbow	Standard tee Through run	Standard tee Through branch	Plug Valve	Diaphragm Valve	Vertical Check Valve	Horizontal Check Valve
25 (1)	0.55 (1.8)	0.37 (1.2)	1.4 (4.5)	0.61 (2.0)	2.1 (7)	1.8 (6.0)	4.9 (16)
40 (1½)	1.1 (3.5)	0.70 (2.3)	2.3 (7.5)	1.3 (4.2)	3.0 (10)	1.8 (6.0)	7.0 (23)
50 (2)	1.4 (4.5)	0.91(3.0)	3.0 (10)	1.7 (5.5)	4.9 (16)	3.0 (10)	14 (45)
65 (2½)	1.7 (5.5)	1.2 (4.0)	3.7 (12)	N.A.	6.7 (22)	3.4 (11)	15 (50)
80 (3)	2.1 (7.0)	1.2 (4.1)	4.6 (15)	N.A.	10 (33)	3.7 (12)	18 (58)
100 (4)	3.0 (10)	1.8 (6.0)	6.1 (20)	N.A.	21 (68)	6.1 (20)	20 (65)
150 (6)	4.6 (15)	3.0 (10)	9.8 (32)	N.A.	26 (85)	9.4 (31)	46 (150)
200 (8)	5.8 (19)	4.3 (14)	13 (42)	N.A.	46 (150)	23 (77)	61 (200)
250 (10)	7.6 (25)	5.8 (19)	16 (53)	N.A.	N.A.	N.A.	N.A.
300 (12)	9.1 (30)	7.0 (23)	20 (64)	N.A.	N.A.	N.A.	N.A.

Notes:
Data is for water expressed as equal length of straight pipe in m (ft)
N.A. = Part is not available from source.
Source:
"Plastic Lined Piping Products Engineering Manual", p. 48.

Table 3-3
Minor Loss Coefficients (K)

Minor loss	Description	K
Pipe Entrance	sharp edged inward projected pipe rounded	0.5 1.0 0.05
Pipe Exit	all	1.0
Contractions	sudden gradual, $\phi < 22°$ gradual, $\phi > 22°$	$0.5\,[1 - (\beta^2)^2]$ $0.8\,(\sin \phi)\,(1 - \beta^2)$ $0.5\,(\sin \phi)^{0.5}\,(1 - \beta^2)$
Enlargements	sudden gradual, $\phi < 22°$ gradual, $\phi > 22°$	$[1 - (\beta^2)^2]^2$ $2.6\,(\sin \phi)\,(1 - \beta^2)^2$ $(1 - \beta^2)^2$
Bends	90° standard elbow 45° standard elbow	0.9 0.5
Tee	standard, flow through run standard, flow through branch	0.6 1.8
Valves	globe, fully open angle, fully open gate, fully open gate, ½ open ball, fully open butterfly, fully open swing check, fully open	10 4.4 0.2 5.6 4.5 0.6 2.5

Notes:

ϕ = angle of convergence/divergence

β = ratio of small to large diameter

Sources:

Hydraulic Institute, "Pipe Friction Manual, 3rd Ed.

Valve data from Crane Company, "Flow of Fluids," Technical Paper 410; reprinted by permission of the Crane Valve Group.

D_i = inside pipe diameter, m (ft)
L = length of pipe, m (ft)
L_e = equivalent length of pipe for minor losses, m (ft)

It is common practice in design to use higher values of ϵ and n and lower values of C than are tabulated for new pipe in order to allow for capacity loss with time.

Example Problem 4:
An equalization tank containing water with dissolved metals is to be connected to a process tank via above grade piping. A pump is required because the process tank liquid elevation is 30 m (98.4 ft) above the equalization tank level.

The piping layout indicates that the piping system requires:

- 2 isolation valves (gate);
- 1 swing check valve;
- 5 standard 90° elbows; and
- 65 m (213.5 ft) of piping.

The process conditions are:

- T = 25°C (77 °F); and
- Q = 0.05 m^3/s (1.77 ft^3/s).

The required piping material is PVC. The design program now requires the pipe to be sized and the pressure drop in the line to be determined in order to select the pump.

Solution:
Step 1. Select pipe size by dividing the volumetric flow rate by the desired velocity (normal service, V = 2.1 m/s).

$$A = \pi \frac{D_i^2}{4} = \frac{Q}{V}$$

$$D_i = \left[\frac{4}{\pi} \frac{0.05\ m^3/s}{2.1\ m/s}\right]^{0.5} \left(1000\ \frac{mm}{m}\right)$$

$$= 174\ mm\ (6.85\ in)$$

Step 2. From Table 1-1, select 150 mm (6 in) as the actual pipe size and calculate actual velocity in the pipe.

$$V = \frac{Q}{A} = \frac{Q}{\frac{\pi}{4} D_i^2}$$

$$= \frac{0.05\ m^3/s}{\frac{\pi}{4} (0.150\ m)^2}$$

$$= 2.83\ m/s\ (9.29\ ft/s)$$

Step 3. At 25°C, ν = 8.94 x 10^{-7} m^2/s. So the Darcy-Weisbach equation is used to calculate the pressure drop through the piping.

$$h_L = \left(\frac{f\ L}{D_i} + \Sigma K\right) \frac{V^2}{2\ g}$$

Step 4. Determine the friction factor, f, from the Moody Diagram (Figure 3-1) and the following values.

$$R_e = \frac{D_i\ V}{\nu} = \frac{(0.150\ m)(2.83\ m/s)}{8.94\ x\ 10^{-7}\ m^2/s}$$

$$= 4.75\ x\ 10^5\ -\ turbulent\ flow$$

$$\epsilon = 1.5\ x\ 10^{-6}\ m\ from\ Table\ 3-1$$

$$\epsilon/D_i = \frac{1.5\ x\ 10^{-6}\ m}{0.150\ m} = 0.00001;$$

therefore, f = 0.022 from Figure 3-1.

Step 5. Determine the sum of the minor loss coefficients from Table 3-3:

minor loss	K
entry	0.5
2 gate valves	0.2x2
check valve	2.5
5 elbows	0.35x5
exit	1.0
sum	6.15

Step 6. Calculate the head loss.

$$h_L = \left(\frac{f\ L}{D_i} + \Sigma K \right) \frac{V^2}{2\ g}$$

$$= \left[\frac{(0.022)(65\ m)}{0.150\ m} + 5.15 \right] \frac{(2.83\ m/s)^2}{2\ (9.81\ m/s^2)}$$

$$= 6.4\ m\ (21\ ft)$$

Step 7. The required pump head is equal to the sum of the elevation change and the piping pressure drop.

$$P_{head} = 30\ m + 6.4\ m = 36.4\ m$$

The prediction of pressures and pressure drops in a pipe network are usually solved by methods of successive approximation. This is routinely performed by computer applications now. In pipe networks, two conditions must be satisfied: continuity must be satisfied (the flow entering a junction equals the flow out of the junction); and there can be no discontinuity in pressure (the pressure drop between two junctions are the same regardless of the route).

The most common procedure in analyzing pipe networks is the Hardy Cross method. This procedure requires the flow in each pipe to be assumed so that condition 1 is satisfied. Head losses in each closed loop are calculated and then corrections to the flows are applied successively until condition 2 is satisfied within an acceptable margin.

b. Pressure Integrity

The previous design steps have concentrated on the evaluation of the pressure and temperature design bases and the design flow rate of the piping system. Once the system operating conditions have been established, the minimum wall thickness is determined based on the pressure integrity requirements.

The design process for consideration of pressure integrity uses allowable stresses, thickness allowances based on system requirements and manufacturing wall thickness tolerances to determine minimum wall thickness.

Allowable stress values for metallic pipe materials are generally contained in applicable design codes. The codes must be utilized to determine the allowable stress based on the requirements of the application and the material to be specified.

For piping materials that are not specifically listed in an applicable code, the allowable stress determination is based on applicable code references and good engineering design. For example, design references that address this type of allowable stress determination are contained in ASME B31.3 Sec. 302.3.2. These requirements address the use of cast iron, malleable iron, and other materials not specifically listed by the ASME B31.3.

After the allowable stress has been established for the application, the minimum pipe wall thickness required for pressure integrity is determined. For straight metallic pipe, this determination can be made using the requirements of ASME B31.3 Sec. 304 or other applicable codes. The determination of the minimum pipe wall thickness using the ASME B31.3 procedure is described below (see code for additional information). The procedure and following example described for the determination of minimum wall thickness using codes other than ASME B31.3 are similar and typically follow the same overall approach.

$$t_m = t + A$$

where:

t_m = total minimum wall thickness required for pressure integrity, mm (in)
t = pressure design thickness, mm (in)
A = sum of mechanical allowances plus corrosion allowance plus erosion allowance, mm (in)

Allowances include thickness due to joining methods, corrosion/erosion, and unusual external loads. Some methods of joining pipe sections result in the reduction of wall thickness. Joining methods that will require this allowance include threading, grooving, and swagging. Anticipated thinning of the material due to effects of corrosion or mechanical wear over the design service life of the pipe may occur for some applications. Finally, site-specific conditions may require additional strength to account for external operating loads (thickness allowance for mechanical strength due to external loads). The stress associated with these loads should be considered in conjunction with the stress associated with the pressure integrity of the pipe. The greatest wall thickness requirement, based on either pressure integrity or external loading, will govern the final wall thickness specified. Paragraph 3-4 details stress analyses.

Using information on liquid characteristics, the amount of corrosion and erosion allowance necessary for various materials of construction can be determined to ensure reasonable service life. Additional information concerning the determination of acceptable corrosion resistance and material allowances for various categories of fluids is contained in Paragraph 3-1a.

The overall formula used by ASME B31.3 for pressure design minimum thickness determination (t) is:

$$t = \frac{P\ D_o}{2\ (S\ E + P\ y)}$$

where:

P = design pressure, MPa (psi)
D_o = outside diameter of the pipe, mm (in)
S = allowable stress, see Table A-1 from ASME B31.3, MPa (psi)
E = weld joint efficiency or quality factor, see Table A-1A or Table A-1B from ASME B31.3
y = dimensionless constant which varies with temperature, determined as follows:
For $t < D_o/6$, see table 304.1.1 from ASME B31.3 for values of y
For $t \geq D_o/6$ or $P/SE > 0.385$, then a special consideration of failure theory, fatigue and thermal stress may be required or ASME B31.3 also allows the use of the following equation to calculate y:

$$y = \frac{D_i + 2A}{D_o + D_i + 2A}$$

where:

D_i = inside diameter of the pipe, mm (in)
D_o = outside diameter of the pipe, mm (in)
A = sum of mechanical allowances plus corrosion allowance plus erosion allowance, mm (in)

Example Problem 5:
In order to better illustrate the process for the determination of the minimum wall thickness, the example in Paragraph 3-2b will be used to determine the wall thickness of the two pipes. For the 150 mm (6 in) header, the values of the variables are:

P = 18.3 MPa (2650 psig)
D_o = 160 mm (6.299 in)
S = 121 MPa (17,500 psi)
Assume t <12.75 in/6, so y = 0.4 from ASME B31.3
A = 2 mm (0.08 in)
E = 1.0

Solution:
Step 1. Determine the minimum wall thickness.

$$t_m = t + A$$

$$t = \frac{P\ D_o}{2\ (S\ E + P\ y)}$$

Therefore,

$$t_m = \frac{P\ D_o}{2\ (S\ E + P\ y)} + A$$

$$= \frac{(18.3\ MPa)(160\ mm)}{2[(121\ MPa)(1.0) + (18.3\ MPa)(0.4)]} + 2\ mm$$

$$= 13.4\ mm\ (0.528\ in)$$

Step 2. The commercial wall thickness tolerance for seamless rolled pipe is +0, -12½%; therefore, to determine the nominal wall thickness, the minimum wall thickness is divided by the smallest possible thickness allowed by the manufacturing tolerances.

$$t_{NOM} = \frac{13.4 \ mm}{1.0 - 0.125} = 15.3 \ mm \ (0.603 \ in)$$

Step 3. Select a commercially available pipe by referring to a commercial specification. For U.S. work ANSI B36.10M/B36.10 is used commercially; the nearest commercial 150 mm (6 in) pipe whose wall thickness exceeds 15.3 mm (0.603 in) is Schedule 160 with a nominal wall thickness of 18.3 mm (0.719 in). Therefore, 150 mm (6 in) Schedule 160 pipe meeting the requirements of ASTM A 106 Grade C is chosen for this application. This calculation does not consider the effects of bending. If bending loads are present, the required wall thickness may increase.

Step 4. For the 100 mm (4 in) header, the outside diameter of 100 mm (4 in) pipe = 110 mm (4.331 in). Therefore:
.

$$t_m = \frac{P \ D_o}{2 \ (S \ E + P \ y)} + A$$

$$= \frac{(19.2 \ MPa)(110 \ mm)}{2[(121 \ MPa)(1.0) + (19.2 \ MPa)(0.4)]} + 2 \ mm$$

$$= 10.2 \ mm \ (0.402 \ in)$$

$$t_{NOM} = \frac{10.2 \ mm}{1.0 - 0.125} = 11.7 \ mm \ (0.459 \ in)$$

The required nominal wall thickness is 11.7 mm (0.459 in).

Step 5. Select a commercially available pipe by referring to a commercial standard. Using ANSI B36.10M/B36.10, XXS pipe with a nominal wall thickness of 17.1 mm (0.674 in) is selected.

Step 6. Check whether the wall thickness for the selected 100 mm (4 in) schedule XXS pipe is adequate to withstand a relief valve failure. The shutoff head of the pump was given as 2,350 m (7,710 ft), and the specific volume of pressurized water at 177°C (350°F) was previously determined to be 0.001110 m^3/kg (0.01778 ft^3/lbm). The pressure equivalent to the shutoff head may be calculated based upon this specific volume.

$$P = (2{,}350 \ m)\left(\frac{1}{0.001110 \ \frac{m^3}{kg}}\right)\left(9.81 \ \frac{m}{s^2}\right)$$

$$= 20.8 \ MPa \ (3{,}020 \ psig)$$

Step 7. Since the previously determined maximum allowable pressure 36.3 MPa (5,265 psig) rating of the XXS pipe exceeds the 20.8 MPa (3,020 psig) shutoff head of the pump, the piping is adequate for the intended service.

The design procedures presented in the forgoing problem are valid for steel or other code-approved wrought materials. They would not be valid for cast iron or ductile iron piping and fittings. For piping design procedures which are suitable for use with cast iron or ductile iron pipe, see ASME B31.1, paragraph 104.1.2(b).

3-4. Stress Analysis

After piping materials, design pressure and sizes have been selected, a stress analysis is performed that relates the selected piping system to the piping layout (Paragraph 2-6) and piping supports (Paragraph 3-7). The analysis ensures that the piping system meets intended service and loading condition requirements while optimizing the layout and support design. The analysis may result in successive reiterations until a balance is struck between stresses and layout efficiency, and stresses and support locations and types. The stress analysis can be a simplified analysis or a computerized analysis depending upon system complexity and the design code.

a. Code Requirements

Many ASME and ANSI codes contain the reference data, formulae, and acceptability limits required for the stress analysis of different pressure piping systems and services. ASME B31.3 requires the analysis of three stress limits: stresses due to sustained loads, stresses due to displacement strains, and stresses due to occasional loads. Although not addressed by code, another effect resulting from stresses that is examined is fatigue.

b. Stresses due to Sustained Loads

The stress analysis for sustained loads includes internal pressure stresses, external pressure stresses and longitudinal stresses. ASME B31.3 considers stresses due to internal and external pressures to be safe if the wall thickness meets the pressure integrity requirements (Paragraph 3-3b). The sum of the longitudinal stresses in the piping system that result from pressure, weight and any other sustained loads do not exceed the basic allowable stress at the maximum metal temperature.

$$\Sigma S_L \leq S_h$$

where:
S_L = longitudinal stress, MPa (psi)
S_h = basic allowable stress at maximum material temperature, MPa (psi), from code (ASME B31.3 Appendix A).

The internal pressure in piping normally produces stresses in the pipe wall because the pressure forces are offset by pipe wall tension. The exception is due to pressure transients such as water hammer which add load to pipe supports. The longitudinal stress from pressure is calculated by:

$$S_L = \frac{P\ D_o}{4\ t}$$

where:
S_L = longitudinal stress, MPa (psi)
P = internal design pressure, MPa (psi)
D_o = outside pipe diameter, mm (in)
t = pipe wall thickness, mm (in)

The longitudinal stress due to weight is dependent upon support locations and pipe spans. A simplified method to calculate the pipe stress is:

$$S_L = 0.1\ \frac{W\ L^2}{n\ Z}$$

where:
S_L = longitudinal stress, MPa (psi)
W = distributed weight of pipe material, contents and insulation, N/m (lbs/ft)
L = pipe span, m (ft)
n = conversion factor, 10^{-3}m/mm (1 ft/12 in)
Z = pipe section modulus, mm^3 (in^3)

$$Z = \frac{\pi}{32}\ \frac{D_o^4 - D_i^4}{D_o}$$

where:
D_o = outer pipe diameter, mm (in)
D_i = inner pipe diameter, mm (in)

c. Stresses due to Displacement Strains

Constraint of piping displacements resulting from thermal expansion, seismic activities or piping support and terminal movements cause local stress conditions. These localized conditions can cause failure of piping or supports from fatigue or over-stress, leakage at joints or distortions. To ensure that piping systems have sufficient flexibility to prevent these failures, ASME B31.3 requires that the displacement stress range does not exceed the allowable displacement stress range.

$$S_E \leq S_A$$

where:
S_E = displacement stress range, MPa (psi)
S_A = allowable displacement stress range, MPa (psi)

$$S_A = f\ [1.25\ (S_c + S_h) - S_L]$$

where:

S_A = allowable displacement stress range, MPa (psi)
f = stress reduction factor
S_c = basic allowable stress of minimum material temperature, MPa (psi), from code (ASME B31.3 Appendix A)
S_h = basic allowable stress at maximum material temperature, MPa (psi), from code (ASME B31.3 Appendix A)
S_L = longitudinal stress, MPa (psi)

$$f = 6.0 \ (N)^{-0.2} \leq 1.0$$

where:

f = stress reduction factor
N = equivalent number of full displacement cycles during the expected service life, < 2 x 10^6.

$$S_E = (S_b^2 + 4S_t^2)^{0.5}$$

where:

S_E = displacement stress range, MPa (psi)
S_b = resultant bending stress, MPa (psi)
S_t = torsional stress, MPa (psi)

$$S_b = \frac{[(i_i \ M_i)^2 + (i_o \ M_o)^2]^{0.5}}{n \ Z}$$

where:

S_b = resultant bending stress, MPa (psi)
i_i = in plane stress intensity factor (see Table in code, ASME B31.3 Appendix D)
M_i = in plane bending moment, N-m (lb-ft)
i_o = out plane stress intensity factor (see table in code, ASME B31.3 Appendix D)
M_o = out plane bending moment, N-m (lb-ft)
n = conversion factor, 10^{-3}m/mm (1 ft/12 in)
Z = Section modulus, mm^3 (in^3)

$$Z = \frac{\pi}{32} \ \frac{D_o^4 - D_i^4}{D_o}$$

where:

D_o = outer pipe diameter, mm (in)
D_i = inner pipe diameter, mm (in)

$$S_t = \frac{M_t}{2 \ Z \ n}$$

where:

S_t = torsional stress, MPa (psi)
M_t = torsional moment, N-m (lb-ft)
Z = section modulus, mm^3 (in^3)
n = conversion factor, 10^{-3}m/mm (1 ft/12 in)

A formal flexibility analysis is not required when: (1) the new piping system replaces in kind, or without significant change, a system with a successful service record; (2) the new piping system can be readily judged adequate by comparison to previously analyzed systems; and (3) the new piping system is of uniform size, has 2 or less fixed points, has no intermediate restraints, and meets the following empirical condition.[9]

$$\frac{D_o \ Y}{(L - L_s)^2} \leq K_1$$

where:

D_o = outside pipe diameter, mm (in)
Y = resultant of total displacement strains, mm (in)
L = length of piping between anchors, m (ft)
L_s = straight line distance between anchors, m (ft)
K_1 = constant, 208.3 for SI units (0.03 for IP units)

d. Stresses due to Occasional Loads

The sum of the longitudinal stresses due to both sustained and occasional loads does not exceed 1.33 times the basic allowable stress at maximum material temperature.

[9] ASME B31.3, p. 38.

$$\Sigma\ S'_L \leq 1.33\ S_h$$

where:

S'_L = longitudinal stress from sustained and occasional loads, MPa (psi)

S_h = basic allowable stress at maximum material temperature, MPa (psi), from code (ASME B31.3 Appendix A)

The longitudinal stress resulting from sustained loads is as discussed in Paragraph 3-4b. The occasional loads that are analyzed include seismic, wind, snow and ice, and dynamic loads. ASME B31.3 states that seismic and wind loads do not have to be considered as acting simultaneously.

e. Fatigue

Fatigue resistance is the ability to resist crack initiation and expansion under repeated cyclic loading. A material's fatigue resistance at an applied load is dependent upon many variables including strength, ductility, surface finish, product form, residual stress, and grain orientation.

Piping systems are normally subject to low cycle fatigue, where applied loading cycles rarely exceed 10^5. Failure from low cycle fatigue is prevented in design by ensuring that the predicted number of load cycles for system life is less than the number allowed on a fatigue curve, or S-N curve, which correlates applied stress with cycles to failure for a material. Because piping systems are generally subject to varying operating conditions that may subject the piping to stresses that have significantly different magnitudes, the following method can be used to combine the varying fatigue effects.

$$U = \Sigma \frac{n_i}{N_i}$$

$$U < 1.0$$

where:

U = cumulative usage factor

n_i = number of cycles operating at stress level i

N_i = number of cycles to failure at stress level i as per fatigue curve.

The assumption is made that fatigue damage will occur when the cumulative usage factor equals 1.0.

3-5. Flange, Gaskets and Bolting Materials

ANSI, in association with other technical organizations such as the ASME, has developed a number of predetermined pressure-temperature ratings and standards for piping components. Pipe flanges and flanged fittings are typically specified and designed to ASME B16.5 for most liquid process piping materials. The primary exception to this is ductile iron piping, which is normally specified and designed to AWWA standards. The use of other ASME pressure-integrity standards generally conforms to the procedures described below.

a. Flanges

Seven pressure classes -- 150, 300, 400, 600, 900, 1,500 and 2500 -- are provided for flanges in ASME B16.5. The ratings are presented in a matrix format for 33 material groups, with pressure ratings and maximum working temperatures. To determine the required pressure class for a flange:

Step 1. Determine the maximum operating pressure and temperature.
Step 2. Refer to the pressure rating table for the piping material group, and start at the class 150 column at the temperature rating that is the next highest above the maximum operating temperature.
Step 3. Proceed through the table columns on the selected temperature row until a pressure rating is reached that exceeds the maximum operating pressure.
Step 4. The column label at which the maximum operating pressure is exceeded at a temperature equal to or above the maximum operating temperature is the required pressure class for the flange.

Example Problem 6:
A nickel pipe, alloy 200, is required to operate at a maximum pressure of 2.75 MPa (399 psi) and 50°C (122°F).

Solution:
Nickel alloy 200 forged fitting materials are manufactured in accordance with ASTM B 160 grade

N02200 which is an ASME B16.5 material group 3.2. Entering Table 2-3.2 in ASME B16.5 at 200 degrees F, the next temperature rating above 50 °C (122 °F), a class 400 flange is found to have a 3.31 MPa (480 psi) rating and is therefore suitable for the operating conditions.

Care should be taken when mating flanges conforming to AWWA C110 with flanges that are specified using ASME B16.1 or B16.5 standards. For example, C110 flanges rated for 1.72 MPa (250 psi) have facing and drilling identical to B16.1 class 125 and B16.5 class 150 flanges; however, C110 flanges rated for 1.72 MPa (250 psi) will not mate with B16.1 class 250 flanges.[10]

b. Gaskets

Gaskets and seals are carefully selected to insure a leak-free system. A wide variety of gasket materials are available including different metallic and elastomeric products. Two primary parameters are considered, sealing force and compatibility. The force that is required at this interface is supplied by gasket manufacturers. Leakage will occur unless the gasket fills into and seals off all imperfections.

The metallic or elastomeric material used is compatible with all corrosive liquid or material to be contacted and is resistant to temperature degradation.

Gaskets may be composed of either metallic or nonmetallic materials. Metallic gaskets are commonly designed to ASME B16.20 and nonmetallic gaskets to ASME B16.21. Actual dimensions of the gaskets should be selected based on the type of gasket and its density, flexibility, resistance to the fluid, temperature limitation, and necessity for compression on its inner diameter, outer diameter or both. Gasket widths are commonly classified as group I (slip-on flange with raised face), group II (large tongue), or group III (small tongue width). Typically, a more narrow gasket face is used to obtain higher unit compression, thereby allowing reduced bolt loads and flange moments.

Consult manufacturers if gaskets are to be specified thinner than 3.2 mm (1/8 in) or if gasket material is specified to be something other than rubber.[11] For non-metallic gaskets, installation procedures are critical. The manufacturer's installation procedures should be followed exactly.

The compression used depends upon the bolt loading before internal pressure is applied. Typically, gasket compressions for steel raised-face flanges range from 28 to 43 times the working pressure in classes 150 to 400, and 11 to 28 times in classes 600 to 2,500 with an assumed bolt stress of 414 MPa (60,000 psi). Initial compressions typically used for other gasket materials are listed in Table 3-4.

Table 3-4
Gasket Compression

Gasket Material	Initial Compression, MPa (psi)
Soft Rubber	27.6 to 41.4 (4,000 to 6,000)
Laminated Asbestos	82.7 to 124 (12,000 to 18,000)
Composition	207 (30,000)
Metal Gaskets	207 to 414 (30,000 to 60,000)

Note: These guidelines are generally accepted practices. Designs conform to manufacturer's recommendations.
Source: SAIC, 1998

In addition to initial compression, a residual compression value, after internal pressure is applied, is required to maintain the seal. A minimum residual gasket compression of 4 to 6 times the working pressure is standard practice. See Paragraph 3-5c, following, for determination of bolting loads and torque.

10 AWWA C110, p. ix-x.
11 Ibid., p. 44.

c. Bolting Materials

Carbon steel bolts, generally ASTM A 307 grade B material, should be used where cast iron flanges are installed with flat ring gaskets that extend only to the bolts. Higher strength bolts may be used where cast iron flanges are installed with full-face gaskets and where ductile iron flanges are installed (using ring or full-face gaskets).[12] For other flange materials, acceptable bolting materials are tabulated in ASME B16.5. Threading for bolts and nuts commonly conform to ASME B1.1, Unified Screw Threads.

The code requirements for bolting are contained in Sections III and VIII of the ASME Boiler and Pressure Vessel Code. To determine the bolt loads in the design of a flanged connection that uses ring-type gaskets, two analyses are made and the most severe condition is applied. The two analyses are for operating conditions and gasket seating.

Under normal operating conditions, the flanged connection (i.e., the bolts) resists the hydrostatic end force of the design pressure and maintains sufficient compression on the gasket to assure a leak-free connection. The required bolt load is calculated by[13]:

$$W_{m1} = 0.785\ G^2\ P + (2\ b)(3.14\ G\ m\ P)$$

where:

W_{m1} = minimum bolt load for operating conditions, N (lb)
G = gasket diameter, mm (in)
= mean diameter of gasket contact face when seating width, b, ≤ 6.35 mm (0.25 in), or
= outside diameter of gasket contact face less 2 b when seating width, b, > 6.35 mm (0.25 in)
P = design pressure, MPa (psi)
b = effective gasket seating width, mm (in), see code (e.g., ASME Section VIII, Appendix 2, Table 2-5.2)
m = gasket factor, see Table 3-5

The required bolt area is then:

$$A_{m1} = \frac{W_{m1}}{S_b}$$

where:

A_{m1} = total cross-sectional area at root of thread, mm^2 (in^2)
W_{m1} = minimum bolt load for operating conditions, N (lb)
S_b = allowable bolt stress at design temperature, MPa (psi), see code (e.g. ASME Section VIII, UCS-23)

Gasket seating is obtained with an initial load during joint assembly at atmosphere temperature and pressure. The required bolt load is:

$$W_{m2} = 3.14\ b\ G\ y$$

where:

W_{m2} = minimum bolt load for gasket seating, N (lbs)
b = effective gasket seating width, mm (in), see code (e.g., ASME Section VIII, Appendix 2, Table 2-5.2)
G = gasket diameter, mm (in)
= mean diameter of gasket contact face when seating width, b, ≤ 6.35 mm (0.25 in)
= outside diameter of gasket contact face less 2b when seating width, b > 6.35 mm (0.25 in)
y = gasket unit seating load, MPa (psi), see Table 3-5

The required bolt area is then:

$$A_{m2} = \frac{W_{m2}}{S_a}$$

where:

A_{m2} = total cross-sectional area at root thread, mm^2 (in^2)
W_{m2} = minimum bolt load for gasket seating, N (lbs)
S_a = allowable bolt stress at ambient temperature, MPa (psi), see code (e.g. ASME Section VIII, UCS-23)

12 AWWA C110, p. 44.
13 ASME Section VIII, pp. 327-333.

Table 3-5
Gasket Factors and Seating Stress

Gasket Material	Gasket Factor, m	Minimum Design Seating Stress, y, MPa (psi)
Self-energizing types (o-rings, metallic, elastomer)	0	0 (0)
Elastomers without fabric		
below 75A Shore Durometer	0.50	0 (0)
75A or higher Shore Durometer	1.00	1.38 (200)
Elastomers with cotton fabric insertion	1.25	2.76 (400)
Elastomers with asbestos fabric insertion (with or without wire reinforcement		
3-ply	2.25	15.2 (2,200)
2-ply	2.50	20.0 (2,900)
1-ply	2.75	25.5 (3,700)
Spiral-wound metal, asbestos filled		
carbon	2.50	68.9 (10,000)
stainless steel, Monel and nickel-based alloys	3.00	68.9 (10,000)
Corrugated metal, jacketed asbestos filled or asbestos inserted		
soft aluminum	2.50	20.0 (2,900)
soft copper or brass	2.75	25.5 (3,700)
iron or soft steel	3.00	31.0 (4,500)
Monel or 4% to 6% chrome	3.25	37.9 (5,500)
stainless steels and nickel-based alloys	3.50	44.8 (6,500)
Corrugated metal		
soft aluminum	2.75	25.5 (3,700)
soft copper or brass	3.00	31.0 (4,500)
iron or soft steel	3.25	37.9 (5,500)
Monel or 4% to 6% chrome	3.50	44.8 (6,500)
stainless steels and nickel-based alloys	3.75	52.4 (7,600)
Ring joint		
iron or soft steel	5.50	124 (18,000)
Monel or 4% to 6% chrome	6.00	150 (21,800)
stainless steels and nickel-based alloys	6.50	179 (26,000)

Notes:
This table provides a partial list of commonly used gasket materials and contact facings with recommended design values m and y. These values have generally proven satisfactory in actual service. However, these values are recommended and not mandatory; consult gasket supplier for other values.

Source:
ASME Section VIII of the Boiler and Pressure Vessel Code, Appendix 2, Table 2-5.1, Reprinted by permission of ASME.

The largest bolt load and bolt cross-sectional area controls the design. The bolting is selected to match the required bolt cross-sectional area by:

$$A_s = 0.7854 \left(D - \frac{0.9743}{N} \right)^2$$

where:
A_s = bolt stressed area, mm^2 (in^2)
D = nominal bolt diameter, mm (in)
N = threads per unit length, 1/mm (1/in)

The tightening torque is then calculated using the controlling bolt load[14]:

$$T_m = W_m \, K \, D \, n$$

where:
T_m = tightening torque, N-m (in-lb)
W_m = required bolt load, N (lb)
K = torque friction coefficient
= 0.20 for dry
= 0.15 for lubricated
D = nominal bolt diameter, mm (in)
n = conversion factor, 10^{-3} m/mm for SI units (1.0 for IP units)

3-6. Pipe Identification

Pipes in exposed areas and in accessible pipe spaces shall be provided with color band and titles adjacent to all valves at not more than 12 m (40 ft) spacing on straight pipe runs, adjacent to directional changes, and on both sides where pipes pass through wall or floors. Piping identification is specified based on CEGS 09900 which provides additional details and should be a part of the contract documents. Table 3-6 is a summary of the requirements

a. Additional Materials

Piping systems that carry materials not listed in Table 3-6 are addressed in liquid process piping designs in accordance with ANSI A13.1 unless otherwise stipulated by the using agency. ANSI A13.1 has three main classifications: materials inherently hazardous, materials of inherently low hazard, and fire-quenching materials. All materials inherently hazardous (flammable or explosive, chemically active or toxic, extreme temperatures or pressures, or radioactive) shall have yellow coloring or bands, and black legend lettering. All materials of inherently low hazard (liquid or liquid admixtures) shall have green coloring or bands, and white legend lettering. Fire-quenching materials shall be red with white legend lettering.

3-7. Piping Supports

Careful design of piping support systems of above grade piping systems is necessary to prevent failures. The design, selection and installation of supports follow the Manufacturers Standardization Society of the Valve and Fitting Industry, Inc. (MSS) standards SP-58, SP-69, and SP-89, respectively. The objective of the design of support systems for liquid process piping systems is to prevent sagging and damage to pipe and fittings. The design of the support systems includes selection of support type and proper location and spacing of supports. Support type selection and spacing can be affected by seismic zone(see Paragraph 2-5b).

a. Support Locations

The locations of piping supports are dependent upon four factors: pipe size, piping configuration, locations of valves and fittings, and the structure available for support. Individual piping materials have independent considerations for span and placement of supports.

Pipe size relates to the maximum allowable span between pipe supports. Span is a function of the weight that the supports must carry. As pipe size increases, the weight of the pipe also increases. The amount of fluid which the pipe can carry increases as well, thereby increasing the weight per unit length of pipe.

The configuration of the piping system affects the location of pipe supports. Where practical, a support should be located adjacent to directional changes of piping. Otherwise, common practice is to design the length of piping between supports equal to, or less than,

[14] Schweitzer, Corrosion-Resistant Piping Systems, p. 9.

Table 3-6
Color Codes for Marking Pipe

MATERIAL	LETTERS AND BAND	ARROW	LEGEND
Cold Water (potable)	Green	White	POTABLE WATER
Fire Protection Water	Red	White	FIRE PR. WATER
Hot Water (domestic)	Green	White	H. W.
Hot Water recirculating (domestic)	Green	White	H. W. R.
High Temp. Water Supply	Yellow	Black	H. T. W. S
High Temp. Water Return	Yellow	Black	H.T.W.R.
Boiler Feed Water	Yellow	Black	B. F.
Low Temp. Water Supply (heating)	Yellow	Black	L.T.W.S.
Low Temp. Water Return (heating)	Yellow	Black	L.T.W.R.
Condenser Water Supply	Green	White	COND. W.S.
Condenser Water Return	Green	White	COND. W.R.
Chilled Water Supply	Green	White	C.H.W.S.
Chilled Water Return	Green	White	C.H.W.R.
Treated Water	Yellow	Black	TR. WATER
Chemical Feed	Yellow	Black	CH. FEED
Compressed Air	Yellow	Black	COMP. AIR
Natural Gas	Blue	White	NAT. GAS
Freon	Blue	White	FREON
Fuel Oil	Yellow	Black	FUEL OIL
Steam	Yellow	Black	STM.
Condensate	Yellow	Black	COND.

Source: USACE, Guide Specification 09900, Painting, General, Table 1.

75% of the maximum span length where changes in direction occur between supports. Refer to the appropriate piping material chapters for maximum span lengths.

As discussed in Chapter 10, valves require independent support, as well as meters and other miscellaneous fittings. These items contribute concentrated loads to the piping system. Independent supports are provided at each side of the concentrated load.

Location, as well as selection, of pipe supports is dependent upon the available structure to which the support may be attached. The mounting point shall be able to accommodate the load from the support. Supports are not located where they will interfere with other design considerations. Some piping materials require that they are not supported in areas that will expose the piping material to excessive ambient temperatures. Also, piping is not rigidly anchored to surfaces that transmit vibrations. In this case, pipe supports isolate the piping system from vibration that could compromise the structural integrity of the system.

b. Support Spans

Spacing is a function of the size of the pipe, the fluid conveyed by piping system, the temperature of the fluid and the ambient temperature of the surrounding area. Determination of maximum allowable spacing, or span between supports, is based on the maximum amount that the pipeline may deflect due to load. Typically, a deflection of 2.5 mm (0.1 in) is allowed, provided that the maximum pipe stress is limited to 10.3 MPa (1,500 psi) or allowable design stress divided by a safety factor of 4[15], whichever is less. Some piping system manufacturers and support system manufacturers have information for their products that present recommended spans in tables or charts. These data are typically empirical and are based upon field experience. A method to calculate support spacing is as follows:

$$l = n \left(m \, C' \, \frac{Z \, S}{W} \right)^{0.5}$$

where:

l = span, m (ft)
n = conversion factor, 10^{-3} m/mm (1 ft/12 in)
m = beam coefficient, see Table 3-7
C′ = beam coefficient = 5/48 for simple, one-span beam (varies with beam type)
Z = section modulus, mm^3 (in^3)
S = allowable design stress, MPa (psi)
W = weight per length, N/mm (lb/in)

$$Z = \frac{\pi}{32} \, \frac{D_o^4 - D_i^4}{D_o}$$

where:

Z = section modulus, mm^3 (in^3)
D_o = outer pipe diameter, mm (in)
D_i = inner pipe diameter, mm (in)

Table 3-7
Beam Coefficient (m)

m	Beam Characteristic
76.8	simple, single span
185.2	continuous, 2-span
144.9	continuous, 3-span
153.8	continuous, 4 or more span

Note: These values assume a beam with free ends and uniform loads. For piping systems with a fixed support, cantilever beam coefficients may be more appropriate.
Source: Manual of Steel Construction, pp. 2-124 to 2-127.

The term W, weight per length, is the uniformly distributed total weight of the piping system and includes the weight of the pipe, the contained fluid, insulation and

[15] Schweitzer, Corrosion-Resistant Piping Systems, p. 5.

jacket, if appropriate. Due to the many types of insulation, the weight must be calculated after the type of insulation is selected; see Chapter 11 for insulation design. The following formula can be used to determine the weight of insulation on piping:

$$W_i = \pi \; K \; \delta \; T_i \; (D_o + T_i)$$

where:

W_i = weight of insulation per length, N/mm (lbs/in)
δ = insulation specific weight, N/m^3 (lbs/ft^3)
K = conversion factor, 10^{-9} m^3/mm^3 (5.79 x 10^{-4} ft^3/in^3)
T_i = insulation thickness, mm (in)
D_o = outer pipe diameter, mm (in)

Proper spacing of supports is essential to the structural integrity of the piping system. An improperly spaced support system will allow excessive deflection in the line. This can cause structural failure of the piping system, typically at joints and fittings. Excessive stress can also allow for corrosion of the pipe material by inducing stress on the pipe and, thereby, weakening its resistance to corrosive fluids.

The amount of sag, or deflection in a span, is calculated from the following equation:

$$y = \frac{W \; (l/n)^4}{m \; E \; I}$$

where:

y = deflection, mm (in)
W = weight per length, N/mm (lb/in)
l = span, m (ft)
n = conversion factor, 10^{-3} m/mm (1 ft/12 in)
m = beam coefficient, see Table 3-7.
E = modulus of elasticity of pipe material, MPa (psi)
I = moment of inertia, mm^4 (in^4)

$$I = \frac{\pi}{64} \; (D_o^4 - D_i^4)$$

where:

I = moment of inertia, mm^4 (in^4)
D_o = outer pipe diameter, mm (in)
D_i = inner pipe diameter, mm (in)

Improper spacing of supports can allow fluids to collect in the sag of the pipe. Supports should be spaced and mounted so that piping will drain properly. The elevation of the down-slope pipe support should be lower than the elevation of the lowest point of the sag in the pipe. This is determined by calculating the amount of sag and geometrically determining the difference in height required.

$$h = \frac{(l/n)^2 \; y}{0.25 \; (l/n)^2 - y^2}$$

where:

h = difference in elevation of span ends, mm, (in)
l = span, m (ft)
n = conversion factor, 10^{-3} m/mm (1 ft/12 in)
y = deflection, mm (in)

c. Support Types

The type of support selected is equally important to the design of the piping system. The stresses and movements transmitted to the pipe factor in this selection. Pipe supports should not damage the pipe material or impart other stresses on the pipe system. The basic type of support is dictated by the expected movement at each support location.

The initial support design must address the load impact on each support. Typically, a moment-stress calculation is used for 2-dimensional piping, and a simple beam analysis is used for a straight pipe-run.

If a pipe needs to have freedom of axial movement due to thermal expansion and contraction or other axial movement, a roller type support is selected. If minor axial and transverse (and minimal vertical) movements are expected, a hanger allowing the pipe to ‘swing’ is selected. If vertical movement is required, supports with springs or hydraulic dampers are required. Other structural requirements and conditions that have the potential to affect piping systems and piping support systems are analyzed. Pipes that connect to heavy tanks or pass under footings are protected from differential settlement by flexible couplings. Similarly, piping attached to vibrating or rotating equipment are also attached with flexible couplings.

d. Selection of Support Types

The selection of support types is dependent upon four criteria: the temperature rating of the system, the mechanism by which the pipe attaches to the support, protective saddles that may be included with the support, and the attachment of the support to the building or other structures. Support types are most commonly classified in accordance with MSS SP-58. Figure 3-2 displays some of the support types applicable to liquid process piping systems. The selection of the appropriate support type is made according to MSS SP-69. Table 3-8 provides guidance for process system temperatures.

Some piping systems utilize protective saddles between the pipe and the support member. This is done to minimize the stress on the pipe from point loads. In addition, pipe insulation requires protection from supports. Saddles support piping without damaging insulation.

The method by which the supports attach to buildings or other structures is addressed by the design. Typical pipe supports are in the form of hangers, supporting the pipe from above. These hangers may be attached to a ceiling, beam, or other structural member. Pipelines may be supported from below as well, with pipe stanchions or pipe racks. Pipe supports may be rigidly attached to a structure, or allow for a pivoting axial motion, depending on the requirements of the system.

Table 3-8
Support Type Selection for Horizontal Attachments: Temperature Criteria

Process Temperature, °C (°F)	Typical MSS SP-58 Types	Application
A-1. Hot Systems 49 to 232°C (120 to 450°F)	2, 3, 24, 1, 5, 7, 9, 10, 35 through 38, 59, 41, 43 through 46, 39, 40	clamps hangers sliding rollers insulation protection
B. Ambient Systems 16 to 48°C (60 to 119°F)	3, 4, 24, 26, 1, 5, 7, 9, 10, 35 through 38, 59, 41, 43 through 46, 39, 40	clamps hangers sliding rollers insulation protection
C-1. Cold Systems 1 to 15°C (33 to 59°F)	3, 4, 26, 1, 5, 7, 9, 10, 36 through 38, 59, 41, 43 through 46, 40	clamps hangers sliding rollers insulation protection

Source:
MSS SP-69, pp. 1, 3-4.

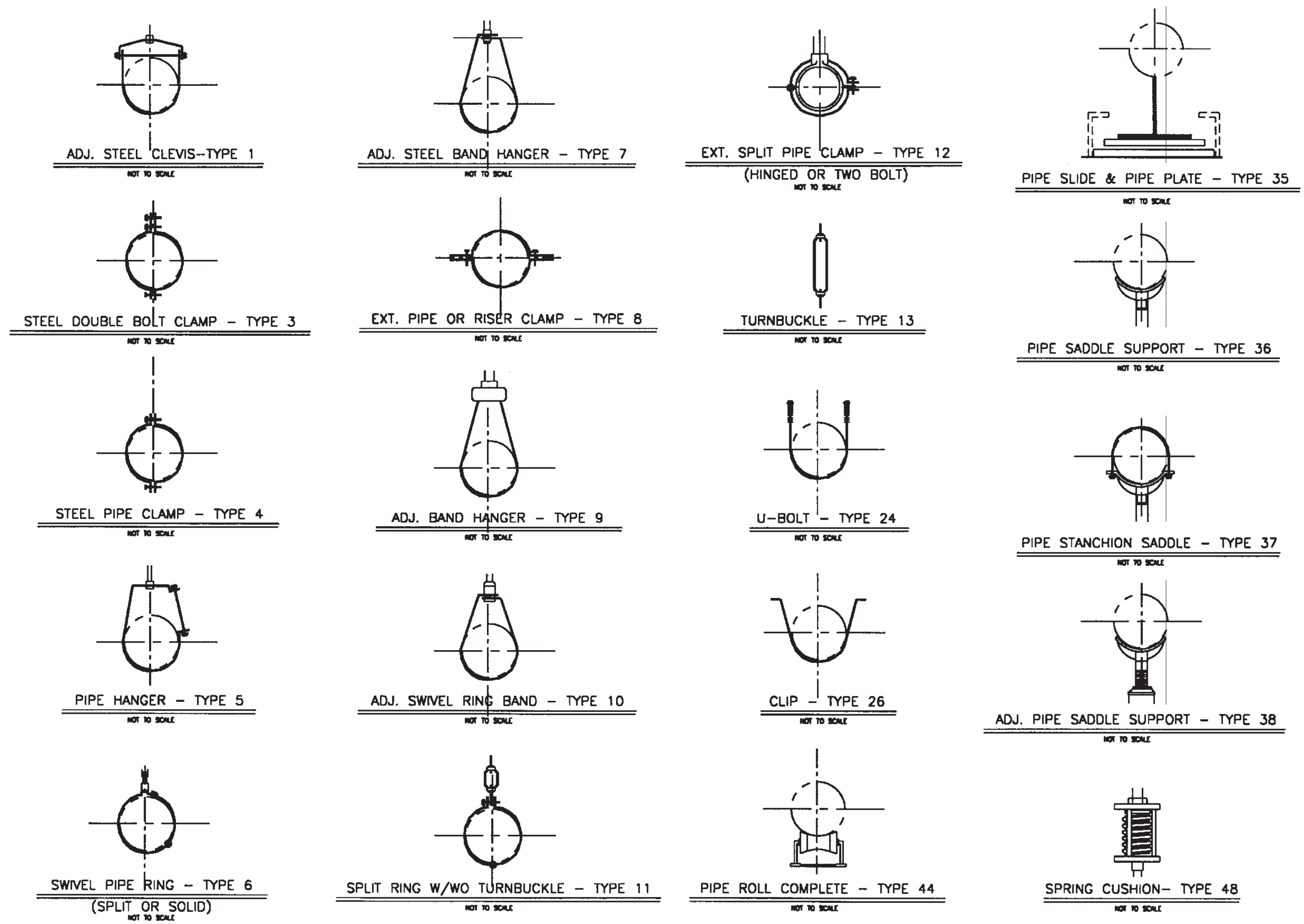

Figure 3-2. Pipe Supports for Ambient Applications
(Source: MSS SP-69, Pipe Hangers and Supports - Selection and Application, pp. 5-6)

Some piping systems require adjustable pipe supports. One reason for this requirement is the cold spring action. Cold spring is the action whereby a gap is left in the final joint of a piping run to allow for thermal expansion of the pipeline. This action results in the offset of all points along the piping system, including the attachments to pipe supports, and requires that supports be adjustable to accommodate this offset. From a maintenance consideration, cold springing should be avoided if possible through proper thermal expansion and stress analyses.

Vertical adjustment is also usually necessary for pipe supports. Settlement, particularly in new construction, may result in an improper deflection of the elevation of a pipe support. To maintain the proper slope in the pipeline, thereby avoiding excessive sag between supports and accumulation of the product being carried by the pipe, the possibility of vertical adjustment is accommodated in the design of pipe supports.

e. Coatings

Installation of piping systems in corrosive environments may warrant the specification of a protective coating on pipe supports. The coating may be metallic or non-metallic; MSS SP-58 is used to specify coatings. Support manufacturers can provide specific recommendations for coatings in specific environments, particularly for nonmetallic coatings. In addition, compatibility between the support materials and piping system materials is reviewed to avoid galvanic action. Electrical isolation pads or different support materials are sometimes required.

3-8. Testing and Flushing

This section addresses the requirements for pressure and leak testing of piping systems. In addition to these types of tests, welding procedures, welders and qualifications of welding operators must conform with the welding and nondestructive testing procedures for pressure piping specified in CEGS 05093, Welding Pressure Piping.

a. Test Procedure

A written test procedure is specified and utilized to perform a leak test. The procedure should prescribe standards for reporting results and implementing corrective actions, if necessary. Review items for preparing the test plans and procedures include:

(1) Determination of the test fluid.
(2) Comparison of the probable test fluid temperature relative to the brittle fracture toughness of the piping materials (heating the test fluid may be a solution).
(3) Depending upon the test fluid, placement of temporary supports where permanent supports were not designed to take the additional weight of the test fluid.
(4) Depending upon the test fluid, location of a relief valve to prevent excessive over-pressure from test fluid thermal expansion. No part of the system will exceed 90% of its yield strength.
(5) Isolation of restraints on expansion joints.
(6) Isolation of vessels, pumps and other equipment which may be over stressed at test pressure.
(7) Location of the test pump and the need for additional pressure gauges.
(8) Accessibility to joints for inspection (some codes require that the weld joints be left exposed until after the test). All joints in the pipe system must be exposed for inspection.
(9) Prior to beginning a leak test, the pipe line should be inspected for defects and errors and omissions.

Testing of piping systems is limited by pressure. The pressure used to test a system shall not produce stresses at the test temperature that exceed the yield strength of the pipe material. In addition, if thermal expansion of the test fluid in the system could occur during testing, precautions are taken to avoid extensive stress.

Testing of piping systems is also limited by temperature. The ductile-brittle transition temperature should be noted and temperatures outside the design range avoided. Heat treatment of piping systems is performed prior to leak testing. The piping system is returned to its ambient temperature prior to leak testing.

In general, piping systems should be re-tested after repairs or additions are made to the system. If a leak is detected during testing and then repaired, the system should be re-tested. If a system passes a leak test, and a component is added to the system, the system should be re-tested to ensure that no leaks are associated with the new component.

The documented test records required for each leak test are specified. The records are required to be standardized, completed by qualified, trained test personnel and retained for a period of at least 5 years. Test records include:

- date of the test;
- personnel performing the test and test location;
- identification of the piping system tested;
- test method, fluid/gas, pressure, and temperature; and
- certified results.

Flushing of a piping system prior to leak testing should be performed if there is evidence or suspicion of contaminants, such as dirt or grit, in the pipeline. These contaminants could damage valves, meters, nozzles, jets, ports, or other fittings. The flushing medium shall not react adversely or otherwise contaminate the pipeline, testing fluid, or service fluid. Flushing should be of sufficient time to thoroughly clean contaminants from every part of the pipeline.

b. Preparation

Requirements for preparation of a leak test are also specified. All joints in the piping system are exposed for the leak test in order to allow the inspector to observe the joints during the test to detect leaks. Specified leak test requirements provide for temporary supports. Temporary supports may be necessary if the test fluid weighs more than the design fluid.

c. Hydrostatic Leak Test

The fluid used for a typical hydrostatic leak test is water. If water is not used, the fluid shall be non-toxic and be non-flammable. The test pressure is greater than or equal to 1.5 times the design pressure.

$$P_T \geq 1.5\ P$$

where:
P_T = test pressure, MPa (psi)
P = design pressure, MPa (psi)

For cases in which the test temperature is less than the design temperature, the minimum test pressure is[16]:

$$P_T = \frac{1.5\ P\ S_T}{S}$$

and

$$\frac{S_T}{S} \leq 6.5$$

where:
P_T = test pressure, MPa (psi)
P = design pressure, MPa (psi)
S_T = stress at test temperature, MPa (psi)
S = stress at design temperature, MPa (psi)

For a typical liquid process piping system with temperatures approximately ambient and low pressure, the S_T/S ratio equals 1.0. If the test pressure would produce an S_T in excess of the material yield strength, then the test pressure may be reduced to limit S_T below the yield strength.

The time period required by ASME B31.3 for a hydrostatic leak test is at least ten (10) minutes, but normally one (1) hour is used.

d. Pneumatic Leak Test

Pneumatic leak tests are not recommended for liquid process piping systems and are only used when the liquid residue left from a hydrostatic test has a hazard potential. The test fluid for a pneumatic leak test is a gas. The gas shall be non-flammable and non-toxic. The hazard of released energy stored in a compressed gas shall be considered when specifying a pneumatic leak test. Safety must be considered when recommending a gas for use in this test.

The test temperature is a crucial consideration for the pneumatic leak test. Test temperature shall be considered

[16] ASME B31.3, p. 83.

when selecting the pipe material. Brittle failure is a consideration in extremely low temperatures for some materials. The energy stored in a compressed gas, combined with the possibility of brittle failure, is an essential safety consideration of the pneumatic leak test.

A pressure relief device shall be specified when recommending the pneumatic leak test. The pressure relief device allows for the release of pressure in the piping system that exceeds a set maximum pressure. The set pressure for the pressure relief device shall be 110% of the test pressure, or 345 kPa (50 psi) above test pressure, whichever is lower.

The test pressure for a pneumatic leak test is 110% of the design pressure. The pressure shall gradually increase to 50% of the test pressure or 170 kPa (25 psig), whichever is lower, at which time the piping system is checked. Any leaks found are then fixed before retesting. The test shall then proceed up to the test pressure before examining for leakage.

e. Initial Service Leak Test

An initial service leak test is permitted by ASME B31.3 with the concurrence of the using agency. This test is a preliminary check for leakage at joints and connections. If this test is performed, and all observed leaks are repaired, it is permissible to omit joint and connection examination during the hydrostatic (or pneumatic) leak tests. The initial service leak test is limited to piping systems subject to Category D fluid service only.

A Category D fluid is defined as non-flammable, non-toxic, and not damaging to human tissues. For this system the operating pressure is less than 1.035 MPa (150 psi), and the operating temperature range is between -29°C (-20°F) to 186°C (366°F)[17].

Typically, the service fluid is used for the initial service leak test. This is possible for a Category D fluid. During the test, the pressure in the piping system should be gradually increased to operating pressure. The piping system is then inspected for leaks.

f. Sensitive Leak Test

A sensitive leak test is required for all Category M fluids (optional for Category D fluids) using the Gas and Bubble Test Method of the ASME Boiler and Pressure Vessel Code, Section V, Article 10, or equivalent. The test pressure for the sensitive leak test is 25% of the design pressure or 105 kPa (15 psig), whichever is lower.

Category M fluid service is one in which the potential for personnel exposure is judged to be possible, and in which a single exposure to a small quantity of the fluid (caused by leakage) can produce serious and irreversible personnel health damage upon either contact or breathing.[18]

g. Non-Metallic Piping Systems

Testing requirements, methods, and recommendations for plastic, rubber and elastomer, and thermoset piping systems are the same as those for metallic piping systems, with the following exceptions. The hydrostatic leak test method is recommended and a pneumatic leak test is only performed with the permission of the using agency. The test pressure shall not be less than 1.5 times the system design pressure. However, the test pressure is less than the lowest rated pressure of any component in the system.

$$P_T \geq 1.5\ P$$

$$\textit{and}$$

$$P_T < P_{\min}$$

where:

P_T = test pressure, MPa (psi)
P = system design pressure, MPa (psi)
P_{min} = lowest component rating, MPa (psi)

h. Double Containment and Lined Piping Systems

Testing requirements, methods, and recommendations for double containment and lined piping systems are identical to those pertaining to the outer (secondary) pipe material.

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17 ASME B31.3, p. 5.
18 Ibid., p. 5.