CHAPTER B2
DESIGN BASES

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The design of a piping system is a straightforward process. The technology is extensive and diverse. Piping design requires the application of theory from a number of engineering disciplines, including fluid mechanics, statics, dynamics, strength of materials, and physical metallurgy. It also requires the knowledge and application of a number of codes and standards.

This chapter identifies and explores the various facets of piping design and highlights parameters that must be considered in completing the design process. The design bases discussed here are generic and should be considered during the course of design of any piping system, regardless of its function. In some cases, a number of the parameters discussed may not be applicable; however, they all should be considered when the initial design concept for the system is formulated and developed.

DEFINITION OF THE TERM DESIGN BASES

Design bases are the physical attributes, loading and service conditions, environmental factors, and materials-related factors which must be considered in the detailed design of a piping system, to ensure its pressure integrity over its design life.

Physical Attributes

Physical attributes are those parameters that govern the size, layout, and dimensional limits or proportions of the piping system. Dimensional standards have been established for most piping components such as fittings, flanges, and valves, as well as for the diameter and wall thickness of standard manufactured pipe. Those standards are identified in the section ''Use of Codes and Standards in Piping System Design.'' Certain types of piping systems require special design practices for configuration control, to ensure constructibility, or in-service performance. For example, high-temperature, high-pressure piping systems are usually designed with weld joints spaced a minimum of one pipe diameter apart to facilitate radiographic examination of the joints. Steam and wet-gas systems are designed to maintain the pipe runs at some minimum pitch to ensure adequate drainage of condensate or other liquids that may separate from the gas stream. Pipelines that are subject to frequent plugging should be designed with adequate clearance, and mechanically joined, to allow for ready disassembly and maintenance.

Loading and Service Conditions

Loading conditions, or *loads,* are forces, moments, pressure changes, temperature changes, thermal gradients, or any other parameters that affect the state of stress of the piping system. Typical examples of loading conditions include internal pressure, piping system deadweight, steady-state or transient temperatures, wind loads, or snow and ice loads. Loads may be external to the piping system, such as environmental temperature changes or wind loads; or they may be internal to the system, such as internal fluid pressure or temperature changes.

Service conditions are combinations of loads or load sets that occur simultaneously, therefore the piping system must be designed to withstand their combined effects. Occasionally, the service conditions will be specified by the piping design code. Examples are found in the American Society of Mechanical Engineers' (ASME) Boiler and Pressure Vessel Code, Section III, *Nuclear Power Plants Components,* Paragraph NCA-2142, where service conditions are defined directly, and the ASME Code for Pressure Piping, ASME B31.1, *Power Piping,* Paragraph 104.8, where service conditions are specified under the topic of analysis of piping components due to the effects of sustained and occasional loads.

Where service conditions are not specified by a particular code, the designer should review the various loading conditions that the piping system is exposed to and formulate the combinations that must be considered in design. Reference to the most commonly used piping design codes listed in the section ''Use of Codes and Standards in Piping System Design'' will provide the designer with guidance in setting appropriate design stress limits.

Environmental Factors

When used within the context of this chapter, the term *environmental factors* refers to operating conditions that result in progressive physical or chemically induced deterioration of the piping system which can ultimately lead to a breach of the pressure boundary or a gross structural failure. Failures that are the result of environmental factors are usually slow to progress and frequently involve localized areas of the piping system. The most common examples of environmental factors include *corrosion, erosion,* and *physical damage.* While corrosion and erosion mechanisms can act independently, a combined reaction known as *erosion/corrosion,* or *flow-assisted corrosion,* frequently occurs in wet-steam and water piping systems constructed of plain-carbon steels.

Materials-Related Considerations

Materials-related considerations are the specific chemical, metallurgical, and physical properties of a piping system's material constituents that can ultimately determine its suitability for a particular service. Proper materials selection can be a crucial design consideration that will determine the adequacy of performance of a piping system where extremes of temperature, chemical attack, or erosion are significant factors in its operation.

Pressure Integrity

Pressure integrity is the maintenance of a leak-tight condition in piping systems' pressure-containing boundaries coincident with the control of the level of stress or strain within predefined criteria limits. Pressure integrity is not synonymous with leakage integrity; the latter is only an assurance of a leak-tight condition without regard for the state of stress or structural stability of the pressure boundary. Maintenance of the pressure integrity of a piping system, within predefined criteria limits, is a major objective of the design process.

USE OF CODES AND STANDARDS IN PIPING SYSTEM DESIGN

In practice, the assurance that the design and construction of a piping system will meet prescribed pressure-integrity requirements is achieved through the use of published codes and standards. Numerous codes and standards have been formulated and published by major interest groups of the piping and pressure vessel industry. The most widely used codes and standards for piping system design are published by the American Society of Mechanical Engineers. The *American National Standards Institute* (*ANSI*) accredits many of these codes and standards.

Differentiation between Codes and Standards

Codes and standards both provide criteria through which pressure integrity can be ensured and simplified design rules to ensure adherence to the criteria. Many designers and engineers think the terms *code* and *standard* are synonymous, or at least somewhat interchangeable, but this understanding is incorrect.

Codes. Piping *codes* provide specific design criteria such as permissible materials of construction, allowable working stresses, and load sets that must be considered in design. In addition, rules are provided to determine the minimum wall thickness and structural behavior due to the effects of internal pressure, deadweight, seismic loads, live loads, thermal expansion, and other imposed internal or external loads. Piping codes provide design rules for nonstandard components and for the reinforcement of openings in the pipe wall. They do not provide design rules for standard in-line components such as valves, flanges, and standard fittings; rather, they define the design requirements for these classes of components by reference to industry standards.

The use of specific codes for the design and construction of piping systems is frequently mandated by statute or regulations imposed by regulatory and enforcement agencies.

Typically codes are structured around technology or industry user lines. For example, ASME B31.1, *Power Piping,* covers piping systems in power plants, district heating plants, district distribution piping systems, and general industrial piping systems while ASME B31.3, *Process Piping,* is structured around the chemical, petroleum, and petrochemical industries. Any one of the above-named industrial facilities might have a pipeline with similar service requirements such as a high-pressure steam main, a boiler feedwater line, or a cooling water line. However, the requirements of the specific code, as influenced by the needs and experience of the user industry, will dictate the pipeline's design and construction requirements.

Many piping design and construction codes are listed in the section ''Reference Codes and Standards.'' The systems and subsystems covered by these codes are defined in their scope sections. The scope sections of all potentially applicable codes should be reviewed early in the design phase of a piping project to determine which code, or codes, should be applied to the piping design and construction.

In some cases, multiple codes may be required for the design and construction of the same piping system, depending upon its location. For example, a steam main serving a petrochemical plant from a major utility's district heating system would be designed and constructed to ASME B31.1, up to the petrochemical plant property line. The balance of the piping on the petrochemical plant's property would be designed to ASME B31.3. In the case of a natural gas main serving a utility powerhouse, the outdoor piping is designed and constructed to ASME B31.8 up to and including the meter set, and the in-plant piping is designed and constructed to ASME B31.1. For more details, refer to Chap. A4.

Sometimes, different piping systems within the same building or facility will be designed and constructed to different codes. For example, most of the piping systems in a utility power plant are designed and constructed to ASME B31.l. However, the building heating and air conditioning piping systems are designed and constructed to ASME B31.9, *Building Services Piping.*

Standards. *Standards* provide specific design criteria and rules for individual components or classes of components such as valves, flanges, and fittings. There are two general types of standards: dimensional and pressure integrity.

Dimensional standards provide configuration control parameters for components. The main purpose of dimensional standards is to ensure that similar components manufactured by different suppliers will be physically interchangeable. Conformity to a particular dimensional standard during the manufacture of a product does not imply that all such similarly configured products will provide equal performance. For example, two different styles of NPS 10 (DN 250) Class 150 flanged-end gate valves could be manufactured, in part, to ASME B16.10, *Face-to-Face and End-to-End Dimensions of Valves.* The valves would be physically interchangeable between mating flanges in a particular piping system. However, because of completely different seat and disk design, one valve might be capable of meeting far more stringent seat leakage criteria than the other.

Pressure-integrity standards provide uniform minimum-performance criteria. Components designed and manufactured to the same standards will function in an equivalent manner. For example, all NPS 10 (DN 250) Class 150 ASTM A105 flanges, which are constructed in accordance with ASME B16.5, *Pipe Flanges and Flanged Fittings,* have a pressure-temperature rating of 230 psig (1590 kPa gage) at 300°F (149°C).

Statute or regulation does not normally mandate standards; rather they are usually invoked by a construction code or purchaser's specification.

The ASME Pressure Classification System

The ASME pressure classification system meets the needs of industry by providing quantitative performance standards for a wide range of piping components, based upon a manageable number of operational variables. This system defines predetermined pressure-temperature ratings that components are designed to meet.

A number of different ASME standards for piping components provide pressure-temperature ratings. The standards in current use in the piping industry are listed in the section ''Reference Codes and Standards.'' In this section the pressure classification system in ASME B16.5, *Pipe Flanges and Flanged Fittings,*[1] will be used for illustration. However, the concepts covered are generally applicable to all the ASME pressure-integrity standards.

Flanges manufactured in accordance with ASME B16.5 are made from materials categorized into 34 material or material alloy groups. There are 8 carbon and low-alloy steel material groups, 10 high-alloy steel material groups, and 16 nonferrous metal groups. Within each of the 34 material groups is a subgroup listing of ASTM materials specifications for forgings, castings, and plates. In addition, acceptable bolting materials and bolting dimensional recommendations are specified. Partial listings of the various material groups, subgroups, and bolting materials are shown in Tables B2.1, B2.2a, and B2.2b. For the complete list, see ASME B16.5.

For any single material group, all flanges made from any material in the group, which carry the same ASME flange pressure class, have the same pressure-temperature rating.

ASME B16.5 provides seven pressure classes for flanges. They are Classes 150, 300, 400, 600, 900, 1500, and 2500. The pressure-temperature ratings for flanges representing all material groups are organized within 34 tables, one for each material group. Table B2.3 is adapted from ASME Standard B16.5 and is typical of the 34 flange rating tables. It provides the pressure-temperature ratings for flanges in material group 1.1. The table is organized with the pressure classes listed across the top and the maximum working temperatures along the left-hand border. The body of the table provides the pressure ratings for flanges from each pressure class, at the given temperature.

In practice, the use of ASME B16.5 to determine a flange rating is quite simple. The procedure is outlined below:

1. Determine the maximum operating pressure and temperature for the required flange.

2. Select a flange material and therefore a material group from one of the 34 listed material groups. Be aware that some of the qualifying notes concerning maximum operating temperatures for various materials may influence the final material selection.

3. Enter the appropriate material group table at the increment of temperature listed which is higher than the desired maximum operating temperature. Start with the Class 150 column and proceed to the right until a pressure rating for the desired temperature is found which equals or exceeds the required operating pressure. The column in which this condition is satisfied dictates the required pressure class and specifies the actual pressure-temperature rating of the flange.

Example B2.1. Assume that an ASTM A105 carbon-steel flange is required to satisfy a pressure rating of 1060 psig (7310 kPa gage) at 650°F (343°C). ASTM A105 is a material group 1.1 material. Entering Table B2.3 at a temperature of 650°F

TABLE B2.1 A Partial Listing of Materials Used for ASME B16.5 Flange Construction

Material group	Nominal designation	Applicable ASTM specifications		
		Forgings	Castings	Plates
1.1	C-Si C-Mn-Si	A 105 A 350 Gr. LF2	A 216 Gr. WCB	A 515 Gr. 70 A 516 Gr. 70 A 537 Cl. 1
1.2	C-Mn-Si 2½Ni 3½Ni	A 350 Gr. LF3	A 216 Gr. WCC A 352 Gr. LCC A 352 Gr. LC2 A 352 Gr. LC3	A 203 Gr. B A 203 Gr. E
1.3	C-Si C-Mn-Si 2½Ni 3½Ni		A 352 Gr. LCB	A 515 Gr. 65 A 516 Gr. 65 A 203 Gr. A A 203 Gr. D
1.4	C-Si C-Mn-Si	A 350 Gr. LF1 Cl. 1		A 515 Gr. 60 A 516 Gr. 60
1.5	C-½Mo	A 182 Gr. F1	A 217 Gr. WC1 A 352 Gr. LC1	A 204 Gr. A A 204 Gr. B
1.7	C-½Mo ½Cr-½Mo Ni-½Cr-½Mo ¾Ni-¾Cr-1Mo	A 182 Gr. F2	A 217 Gr. WC4 A 217 Gr. WC5	A 204 Gr. C
1.9	1Cr-½Mo 1¼Cr-½Mo 1¼Cr-½Mo-Si	A 182 Gr. F12 Cl. 2 A 182 Gr. F11 Cl. 2	A 217 Gr. WC6	A 387 Gr. 11 Cl. 2
1.10	2¼Cr-1Mo	A 182 Gr. F22 Cl. 3	A 217 Gr. WC9	A 387 Gr. 22 Cl. 2
1.13	5Cr-½Mo	A 182 Gr. F5 A 182 Gr. F5a	A 217 Gr. C5	
1.14	9Cr-1Mo	A 182 Gr. F91	A 217 Gr. C12	
2.1	18Cr-8Ni	A 182 Gr. F304 A 182 Gr. F304H	A 351 Gr. CF3 A 351 Gr. CF8	A 240 Gr. 304 A 240 Gr. 304H
2.2	16Cr-12Ni-2Mo 18Cr-13Ni-3Mo 19Cr-10Ni-3Mo	A 182 Gr. F316 A 182 Gr. F316H	A 351 Gr. CF3M A 351 Gr. CF8M A 351 Gr. CG8M	A 240 Gr. 316 A 240 Gr. 316H A 240 Gr. 317
2.3	18Cr-8Ni 16Cr-12Ni-2Mo	A 182 Gr. F304L A 182 Gr. F316L		A 240 Gr. 304L A 240 Gr. 316L
2.4	18Cr-10Ni-Ti	A 182 Gr. F321 A 182 Gr. F321H		A 240 Gr. 321 A 240 Gr. 321H

Source: Adapted from ASME B16.5, *Pipe Flanges and Flanged Fittings,* American Society of Mechanical Engineers, New York, 1996, Table 1A, p. 10.

TABLE B2.2a Bolting Materials Used with ASME B16.5 Flanges

Bolting materials*											
High strength*			Intermediate strength*			Low strength*			Nickel and special alloy*		
Spec. no.	Grade	Notes*	Spec. no.	Grade	Notes*	Spec. no.	Grade	Notes*	Spec. no.	Grade	Notes*
A 193	B7	. . .	A 193	B5	. . .	A 193	B8 Cl. 1	(6)	B 164	. . .	(7)(8)(9)
A 193	B16	. . .	A 193	B6	. . .	A 193	B8C Cl. 1	(6)			
			A 193	B6X	. . .	A 193	B8M Cl. 1	(6)	B 166	. . .	(7)(8)(9)
A 320	L7	(10)	A 193	B7M	. . .	A 193	B8T Cl. 1	(6)			
A 320	L7A	(10)	A 193	B8 Cl. 2	(11)	A 193	B8A	(6)	B 335	N10665	(7)
A 320	L7B	(10)	A 193	B8C Cl. 2	(11)	A 193	B8CA	(6)			
A 320	L7C	(10)	A 193	B8M Cl. 2	(11)	A 193	B8MA	(6)	B 408	. . .	(7)(8)(9)
A 320	L43	(10)	A 193	B8T Cl. 2	(11)	A 193	B8TA	(6)			
									B 473	. . .	(7)
A 354	BC	. . .	A 320	B8 Cl. 2	(11)	A 307	B	(12)			
A 354	BD	. . .	A 320	B8C Cl. 2	(11)				B 574	N10276	(7)
			A 320	B8F Cl. 2	(11)	A 320	B8 Cl. 1	(6)			
A 540	B21	. . .	A 320	B8M Cl. 2	(11)	A 320	B8C Cl. 1	(6)			
A 540	B22	. . .	A 320	B8T Cl. 2	(11)	A 320	B8M Cl. 1	(6)			
A 540	B23	. . .				A 320	B8T Cl. 1	(6)			
A 540	B24		A 449	. . .	(13)						
			A 453	651	(14)						
			A 453	660	(14)						

* For Notes, refer to ASME B16.5, Table 1B.

Source: Adapted from ASME B16.5, *Pipe Flanges and Flanged Fittings,* American Society of Mechanical Engineers, New York, 1996, Table 1B, p. 13.

TABLE B2.2b Bolting Dimensional Standards Recommendations for Bolts Used with ASME B16.5 Flanges

Product	Carbon steel	Alloy steel
Stud bolts	ASME B18.2.1	ASME B18.2.1
Bolts smaller than ¾ in	ASME B18.2.1, square or heavy hex head	ASME B18.2.1, heavy hex head
Bolts equal to or larger than ¾ in	ASME B18.2.1, square or hex head	ASME B18.2.1, heavy hex head
Nuts smaller than ¾ in	ASME B18.2.2, heavy hex	ASME B18.2.2, heavy hex
Nuts equal to or larger than ¾ in	ASME B18.2.2, hex or heavy hex	ASME B18.2.2, heavy hex
Male threads	ASME B1.1, Cl. 2A course series	ASME B1.1, Cl. 2A course series up through 1 in; eight thread series for larger bolts
Female threads	ASME B1.1, Cl. 2B course series	ASME B1.1, Cl. 2B course series up through 1 in; eight thread series for larger bolts

Source: Adapted from ASME B16.5, *Pipe Flanges and Flanged Fittings,* American Society of Mechanical Engineers, New York, 1996, Table 1C, p. 14.

TABLE B2.3 Pressure-Temperature Ratings for ASME B16.5 Flanges Made from Material Group 1.1 Materials

	Working pressure by classes, psig†						
Class temps., °F	150	300	400	600	900	1500	2500
−20 to 100	285	740	990	1480	2220	3705	6170
200	260	675	900	1350	2025	3375	5625
300	230	655	875	1315	1970	3280	5470
400	200	635	845	1270	1900	3170	5280
500	170	600	800	1200	1795	2995	4990
600	140	550	730	1095	1640	2735	4560
650	125	535	715	1075	1610	2685	4475
700*	110	535	710	1065	1600	2665	4440
750*	95	505	670	1010	1510	2520	4200
800*	80	410	550	825	1235	2060	3430
850*	65	270	355	535	805	1340	2230
900*	50	170	230	345	515	860	1430
950*	35	105	140	205	310	515	860
1000*	20	50	70	105	155	260	430

* For notes concerning the use of carbon-steel materials covered by this table at temperatures over 700°F (371°C), see ASME B16.5, Table 2-1.1.

† To convert pressures to kPa gage, multiply tabular value by 6.9.

Source: Adapted from ASME B16.5, *Pipe Flanges and Flanged Fittings,* American Society of Mechanical Engineers, New York, 1996, Table 2-1.1, p. 15.

(343°C), a Class 600 flange is found to have a rating of 1075 psig (7420 kPa gage) at 650°F (343°C). Therefore, a Class 600 ASTM A105 flange is suitable for the stated conditions.

When using the tables, linear interpolation between listed temperatures and pressures may be employed to determine intermediate pressure ratings.

Reference Codes and Standards

The following listing identifies the codes and standards used for most design work done for modern power and industrial piping systems. It has been prepared as a ready reference.

The ASME Boiler and Pressure Vessel Code. This Code covers a wide variety of pressure-integrity-related design and construction applications. Certain sections of the Code provide rules for the design of piping systems:

Section I: *Rules for Construction of Power Boilers*

Section III: *Rules for Construction of Nuclear Plant Components*

Section IV: *Rules for Construction of Heating Boilers*

Section VIII: *Rules for Construction of Pressure Vessels*

The ASME Pressure Piping Codes. These codes are commonly used for the design of commercial power and industrial piping systems:

B31.1, *Power Piping*

B31.2, *Fuel Gas Piping*

B31.3, *Process Piping*

B31.4, *Liquid Transportation Systems for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia and Alcohols*

B31.5, *Refrigeration Piping*

B31.8, *Gas Transmission and Distribution Piping Systems*

B31.9, *Building Services Piping*

B31.11, *Slurry Transportation Piping Systems*

B31G, *Manual for Determining the Remaining Strength of Corroded Pipelines**
*ASME Guide for Transmission and Distribution Piping Systems**

The ASME Pressure-Integrity Standards. The standards listed below provide design and manufacturing criteria for many commonly used piping components:

B16.1, *Cast Iron Pipe Flanges and Flanged Fittings*

B16.3, *Malleable Iron Threaded Fittings*

B16.4, *Gray Iron Threaded Fittings*

B16.5, *Pipe Flanges and Flanged Fittings (NPS ½ Through NPS 24)*

B16.9, *Factory Made Wrought Steel Buttwelding Fittings*

B16.11, *Forged Fittings, Socket-Welding and Threaded*

B16.15, *Cast Bronze Threaded Fittings (Class 125 and 250)*

B16.18, *Cast Copper Alloy Solder Joint Pressure Fittings*

B16.22, *Wrought Copper and Copper Alloy Solder Joint Pressure Fittings*

B16.24, *Cast Copper Alloy Pipe Flanges and Flanged Fittings (Class 150, 300, 400, 600, 900, 1500, and 2500)*

B16.26, *Cast Copper Alloy Fittings for Flared Copper Tubes*

B16.28, *Wrought Steel Buttwelding Short Radius Elbows and Returns*

B16.33, *Manually Operated Metallic Gas Valves for Use in Gas Piping Systems up to 125 psig (Sizes ½ Through 2)*

B16.34, *Valves—Flanged, Threaded and Welding End*

B16.36, *Orifice Flanges*

B16.38, *Large Metallic Valves for Gas Distribution (Manually Operated, NPS 2½ to 12, 125 psig Maximum)*

B16.39, *Malleable Iron Threaded Pipe Unions, Classes 150, 250, and 300*

B16.42, *Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300*

B16.47, *Large Diameter Steel Flanges (NPS 26 Through NPS 60)*

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* Not an ASME code but listed here for the reader's convenience.

The ASME Dimensional Standards. Listed below are the most commonly used piping-related dimensional standards:

B1.20.1, *Pipe Threads, General Purpose (Inch)*

B1.20.3, *Dryseal Pipe Threads (Inch)*

B16.10, *Face-to-Face and End-to-End Dimensions of Valves*

B16.20, *Metallic Gaskets for Pipe Flanges—Ring Joint, Spiral Wound, and Jacketed*

B16.21, *Non-Metallic Flat Gaskets for Pipe Flanges*

B16.25, *Buttwelding Ends*

B36.10M, *Welded and Seamless Wrought Steel Pipe*

B36.19M, *Stainless Steel Pipe*

PIPING JOINTS

Joint design and selection can have a major impact on the initial installed cost, the long-range operating and maintenance cost, and the performance of the piping system. Factors that must be considered in the joint selection phase of the project design include material cost, installation labor cost, degree of leakage integrity required, periodic maintenance requirements, and specific performance requirements. In addition, since codes do impose some limitations on joint applications, joint selection must meet the applicable code requirements. In the paragraphs that follow, the above-mentioned considerations will be briefly discussed for a number of common pipe joint configurations. Figures illustrating many common piping system joints are shown in Chap. A2.

Butt-welded Joints

Butt-welding is the most common method of joining piping used in large commercial, institutional, and industrial piping systems. Material costs are low, but labor costs are moderate to high due to the need for specialized welders and fitters. Long-term leakage integrity is extremely good, as is structural and mechanical strength. The interior surface of a butt-welded piping system is smooth and continuous which results in low pressure drop. The system can be assembled with internal weld backing rings to reduce fit-up and welding costs, but backing rings create internal crevices, which can trap corrosion products. In the case of nuclear piping systems, these crevices can cause a concentration of radioactive solids at the joints, which can lead to operating and maintenance problems. Backing rings can also lead to stress concentration effects, which may promote fatigue cracks under vibratory or other cyclic loading conditions. Butt-welded joints made up without backing rings are more expensive to construct, but the absence of interior crevices will effectively minimize ‘‘crud’’ buildup and will also enhance the piping system's resistance to fatigue failures. Most butt-welded piping installations are limited to NPS 2½ (DN 65) or larger. There is no practical upper size limit in butt-welded construction. Butt-welding fittings and pipe system accessories are available down to NPS ½ (DN 15). However, economic penalties associated with pipe end preparation and fit-up, and special weld procedure qualifications normally preclude the use of butt-welded construction in sizes NPS 2 (DN 50) and under, except for those special cases where interior surface smoothness and the elimination of internal crevices are of paramount importance. Smooth external surfaces give butt-welded construction high aesthetic appeal.

Socket-welded Joints

Socket-welded construction is a good choice wherever the benefits of high leakage integrity and great structural strength are important design considerations. Construction costs are somewhat lower than with butt-welded joints due to the lack of exacting fit-up requirements and elimination of special machining for butt weld end preparation. The internal crevices left in socket-welded systems make them less suitable for corrosive or radioactive applications where solids buildup at the joints may cause operating or maintenance problems. Fatigue resistance is lower than that in butt-welded construction due to the use of fillet welds and abrupt fitting geometry, but it is still better than that of most mechanical joining methods. Aesthetic appeal is good.

Brazed and Soldered Joints

Brazing and soldering are most often used to join copper and copper-alloy piping systems, although brazing of steel and aluminum pipe and tubing is possible. Brazing and soldering both involve the addition of molten filler metal to a close-fitting annular joint. The molten metal is drawn into the joint by capillary action and solidifies to fuse the parts together. The parent metal does not melt in brazed or soldered construction. The advantages of these joining methods are high leakage integrity and installation productivity. Brazed and soldered joints can be made up with a minimum of internal deposits. Pipe and tubing used for brazed and soldered construction can be purchased with the interior surfaces cleaned and the ends capped, making this joining method popular for medical gases and high-purity pneumatic control installations.

Soldered joints are normally limited to near-ambient temperature systems and domestic water supply. Brazed joints can be used at moderately elevated temperatures. Most brazed and soldered installations are constructed using light-wall tubing; consequently the mechanical strength of these systems is low.

Threaded or Screwed Joints

Threaded or screwed piping is commonly used in low-cost, noncritical applications such as domestic water, fire protection, and industrial cooling water systems. Installation productivity is moderately high, and specialized installation skill requirements are not extensive. Leakage integrity is good for low-pressure, low-temperature installations where vibration is not encountered. Rapid temperature changes may lead to leaks due to differential thermal expansion between the pipe and fittings. Vibration can result in fatigue failures of screwed pipe joints due to the high stress intensification effects caused by the sharp notches at the base of the threads. Screwed fittings are normally made of cast gray or malleable iron, cast brass or bronze, or forged alloy and carbon steel. Screwed construction is commonly used with galvanized pipe and fittings for domestic water and drainage applications. While certain types of screwed fittings are available in up to NPS 12 (DN 300), economic considerations normally limit industrial applications to NPS 3 (DN 80). Screwed piping systems are useful where disassembly and reassembly are necessary to accommodate maintenance needs or process changes. Threaded or screwed joints must be used within the limitations imposed by the rules and requirements of the applicable code.

Grooved Joints

The main advantages of the grooved joints are their ease of assembly, which results in low labor cost, and generally good leakage integrity. They allow a moderate amount of axial movement due to thermal expansion, and they can accommodate some axial misalignment. The grooved construction prevents the joint from separating under pressure. Among their disadvantages are the use of an elastomer seal, which limits their high-temperature service, and their lack of resistance to torsional loading. While typical applications involve machining the groove in standard wall pipe, light wall pipe with rolled-in grooves may also be used. Grooved joints are used extensively for fire protection, ambient temperature service water, and low-pressure drainage applications such as floor and equipment drain systems and roof drainage conductors. They are a good choice where the piping system must be disassembled and reassembled frequently for maintenance or process changes.

Flanged Joints

Flanged connections are used extensively in modern piping systems due to their ease of assembly and disassembly; however, they are costly. Contributing to the high cost are the material costs of the flanges themselves and the labor costs for attaching the flanges to the pipe and then bolting the flanges to each other. Flanges are normally attached to the pipe by threading or welding, although in some special cases a flange-type joint known as a *lap joint* may be made by forging and machining the pipe end. Flanged joints are prone to leakage in services that experience rapid temperature fluctuations. These fluctuations cause high-temperature differentials between the flange body and bolting, which eventually causes the bolt stress to relax, allowing the joint to open up. Leakage is also a concern in high-temperature installations where bolt stress relaxation due to creep is experienced. Periodic retorquing of the bolted connections to reestablish the required seating pressure on the gasket face can minimize these problems. Creep-damaged bolts in high-temperature installations must be periodically replaced to reestablish the required gasket seating pressure. Flanged joints are commonly used to join dissimilar materials, e.g., steel pipe to cast-iron valves and in systems that require frequent maintenance disassembly and reassembly. Flanged construction is also used extensively in lined piping systems.

Compression Joints

Compression sleeve-type joints are used to join plain end pipe without special end preparations. These joints require very little installation labor and as such result in an economical overall installation. Advantages include the ability to absorb a limited amount of thermal expansion and angular misalignment and the ability to join dissimilar piping materials, even if their outside diameters are slightly different.

Disadvantages include the use of rubber or other elastomer seals, which limits their high-temperature application, and the need for a separate external thrust-resisting system at all turns and deadends to keep the line from separating under pressure. Compression joints are frequently used for temporary piping systems or systems that must be dismantled frequently for maintenance. When equipped with the proper gaskets and seals, they may be used for piping systems containing air, other gases, water, and oil; in both aboveground and underground service. Small-diameter compression fittings with all-metal sleeves may be used at elevated temperatures and pressures, when permitted by the rules and requirements of the applicable code. They are common in instrument and control tubing installations and other applications where high seal integrity and easy assembly and disassembly are desirable attributes.

LOADING CONDITIONS

In an earlier section, ''Definition of the Term *Design Bases,*'' loading conditions were identified as one of the five principal elements in the definition of the term *design bases*. This section will identify some of the more common loading conditions and discuss the way in which they are considered in design.

Loading conditions may be classified as either sustained or occasional. *Sustained loads* act on the piping system during all or at least the great majority of its operating time. These loads are time-invariant. Examples of sustained loads include the deadweight of the pipe plus its contents or the pressure load, including the effects of static head. *Occasional loads* are transient and act during relatively small percentages of the system's total operation time. Examples of occasional loads include surges due to pump start-up and shutdown or pressure depressions and/or peaks due to sudden valve actuations.

Design Pressure

The *design pressure* is the maximum sustained pressure that a piping system must contain without exceeding its code-defined allowable stress limits. In single-compartment systems the design pressure is the maximum differential pressure between the interior and exterior portions of the system. In multicompartment systems the design pressure is the maximum differential pressure between any two adjacent compartments. The design pressure is the pressure that results in the heaviest piping wall thickness and/or the highest component pressure rating. The design pressure is not to be exceeded during any normal steady-state operating mode of the piping system.

In formulating the design pressure, the designer must consider all potential pressure sources. Among the more common sources to be considered are

- The hydrostatic head due to differences in elevation between the high and low points in the system
- Back-pressure effects
- Friction losses
- The shutoff head of in-line pumps
- Frequently occurring pressure surges
- Variations in control system performance

Variations in System Pressure. As previously indicated, the system design pressure is the steady-state or sustained maximum pressure. Sustained conditions are those that remain constant over the majority of the total operating time. It is reasonable to expect that short-duration transient system pressure excursions in excess of the steady-state design pressure will occur during normal system operation. These transients, or *occasional pressure* excursions, may be tolerated without increasing the basic system design pressure, provided that the pressure increase does not exceed predefined limits and provided that the amount of time that the transients act does not exceed a specified percentage of the total system operating time.

A number of, but not all, piping design codes provide rules to account for overpressure transients. Among the codes that provide design criteria or guidance are

- The ASME Boiler and Pressure Vessel Code, Section III, *Rules for Construction of Nuclear Power Plant Components*
- Various sections of the ASME Code for Pressure Piping, including

  B31.1, *Power Piping*

  B31.3, *Process Piping*

  B31.4, *Liquid Transportation Systems for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia and Alcohols*

  B31.11, *Slurry Transportation Piping Systems*

The methods used to qualify overpressure conditions for service vary from code to code. The ASME Code, Section III, uses a rather complex approach in which the range of acceptable overpressure transients is related to the nature of the loading combinations being investigated. The loading combinations are known as *service conditions,* and depending upon their severity and frequency of occurrence, pressure transients of up to 2 times the design pressure may be tolerated. The interested reader is referred to Subsubarticles NB, NC, ND-3600 of Section III for the details. In contrast to the complex methods adopted by Section III, ASME B31.4 and ASME B31.11 allow pressure transients of up to 10 percent over the system design pressure without restricting the amount of time that the transients may act.

ASME B31.1 and ASME B31.3 provide rules that are about midway in relative complexity from the extremes indicated above. As an example, the acceptance criteria for occasional loads specified in Paragraph 102.2.4 of the ASME B31.1 Code for Power Piping are reproduced below:

> **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.
>
> It is recognized that variations in pressure and temperature inevitably occur, and therefore the piping system except as limited by component standards referred to in Para. 102.2.1 or by manufacturers of components referred to in Para. 102.2.2, shall be considered safe for occasional short operating periods at higher than design pressure or temperature. For such variations, either pressure or temperature, or both, may exceed the design values if the computed circumferential pressure stress does not exceed the maximum allowable stress from Appendix A for the coincident temperature by:
>
> **A.** 15% if the event duration occurs less than 10% of any 24 hour operating period;
>
> or
>
> **B.** 20% if the event duration occurs less than 1% of any 24 hour operating period.[2]

Referring to Paragraph 104.1.2 of the ASME B31.1 code, one finds Eq. (4) for the maximum allowable pressure in a straight pipe[3]

$$P = \frac{2SE(t_m - A)}{D_o - 2y(t_m - A)} \tag{B2.1}$$

It can be seen from Eq. (B2.1) that the maximum allowable pressure P varies directly with the allowable stress S. Therefore, the net effect of Paragraph 102.2.4 is to allow short-term pressure excursions of from 15 to 20 percent in excess of the design pressure, as long as the respective time criteria are met.

As indicated above, not all piping codes provide rules for accepting transient pressure excursions in excess of the design pressure. Sections of the ASME Code for Pressure Piping which have no such rules include

- B31.5, *Refrigeration Piping*
- B31.8, *Gas Transmission and Distribution Piping Systems*
- B31.9, *Building Services Piping*

When designing to a code which has no rules for acceptance of overpressure transients, the designer must increase the design pressure to envelop the transient condition. If, however, no specific design code is being used as a basis for design of a project, the designer may make a reasonable engineering judgment concerning the handling of transient overpressure events. In the absence of any other governing criteria, the following may be considered:

> For transient pressure conditions that exceed the design pressure by 10 percent or less and act for no more than 10 percent of the total operating time, the transient may be neglected and the design pressure need not be increased. For transients whose magnitude or duration is greater than 10 percent of the design pressure or operating time, the design pressure should be increased to envelop the transient.

Determination of the Piping Wall Thickness. The determination of the piping wall thickness is one of the most important calculations of the piping system design process. In arriving at the final specification of the piping wall thickness, the designer must consider a number of important factors:

- Pressure integrity
- Allowances for mechanical strength, corrosion, erosion, wear, threading, grooving, or other joining processes
- Manufacturing variations (tolerance) in the wall thickness of commercial pipe
- Wall thickness reduction due to butt-welding of end preparation (counterboring)

While a number of different pipe wall thickness design formulas have been proposed over the years,[4] the ASME piping codes have adopted one or the other of the following formulas for pressure-integrity design:

$$t = \frac{PD}{2(SE + Py)} \qquad \text{(Ref. 5)} \tag{B2.2}$$

or

$$t = \frac{PD}{2SE} \qquad \text{(Ref. 6)} \tag{B2.3}$$

where t = design minimum wall thickness required to ensure pressure integrity, in
P = design pressure, psig
D = outside diameter of pipe, in
S = allowable stress, psi
E = weld joint efficiency factor (some codes also specify a casting quality factor F for cast piping materials)
y = dimensionless factor which varies with temperature

For the precise definition of the method by which either equation is used by the codes, the particular code of interest should be consulted.

Most construction codes require the provision of additional wall thickness, over and above that intended to ensure pressure integrity. This additional material allowance is provided in accordance with Eq. (B2.4):

$$t_m = t + c \qquad \text{(Ref. 7)} \tag{B2.4}$$

where t_m = minimum wall thickness required to satisfy the design rules of the code, in
t = wall thickness required to provide pressure integrity [see Eqs. (B2.2) and (B2.3)]
c = additional material allowance,† in

The additional material allowance c is made up of a number of individual allowances that are provided to address different loads or conditions the piping system will see during fabrication, installation, and operation. Each allowance is figured separately, and their sum is added to the pressure-integrity wall thickness to arrive at the final design minimum wall thickness. The major constituents of c include

- Wall thickness added to account for progressive deterioration or thinning of the pipe wall in service due to the effects of corrosion, erosion, and wear.
- Wall thickness added to account for material removed to facilitate joining of the various segments of the piping system. Typical joining methods include threading, grooving, and swagging. If a machining tolerance is required as a part of the joint manufacture, this tolerance must be accounted for in the most conservative manner.
- Wall thickness added to provide mechanical strength. This additional strength might be required to resist external operating loads or loads associated with shipping and handling

The effects of pressure result in pipe wall stresses in both the longitudinal and circumferential (*hoop stress*) directions. Typically, the circumferential stress is twice the longitudinal stress. Piping wall thickness selections made using hoop stress-type formulas, such as (B2.2) and (B2.3), result in excess-material availability in the longitudinal direction. In most cases, this excess material is adequate to resist bending stresses associated with the deadweight of the pipe, its contents, and in-line components such as valves, flanges, and piping specialties. In some cases, such as extremely long spans between pipe hangers and piping which is required to support unusually large concentrated loads, it may be necessary to increase the wall thickness to control bending stresses. Refer to Chap. B4.

† In some codes, the term A is used in lieu of c, but the intent is the same.

Once the design minimum wall thickness t_m is determined, the only remaining step is to specify the actual or purchase wall thickness.

Pipe is manufactured to one of two wall thickness dimensioning procedures: minimum wall thickness and nominal wall thickness.

Pipe purchased to a minimum wall thickness specification will be manufactured using special processes to control the wall thickness. These processes may include custommade dies, extra rolling passes, or final boring of the inside diameter. Most minimum wall pipe is custom-manufactured. The use of minimum wall pipe is normally limited to high-pressure, high-temperature applications where the savings in material weight is sufficient to offset the additional manufacturing cost.

Pipe purchased to a nominal wall thickness specification is manufactured in accordance with the dimensional criteria specified in ASME Standards B36.10M and B36.19M. These standards provide predetermined nominal wall thicknesses, or *schedules,* for various standard outside diameters of commercially manufactured pipe.

The tolerance on the wall thickness of pipe varies with the particular manufacturing process employed and with the relevant manufacturing specification. Rolled seamless and seam-welded (without filler metal) pipe has a normal wall thickness tolerance of +0, −12½ percent. Forged and bored pipe has a wall thickness tolerance of +⅛ in (3.2 mm), −0. Piping manufactured from rolled and welded plate has a wall thickness tolerance of −0.01 in (0.25 mm). There is no plus tolerance for this type of pipe. The ASTM (or ASME) specification for the particular piping material should be consulted to determine the wall thickness tolerance. Refer to App. E5.

When piping is to be joined by butt-welding, the pipe ends are frequently counterbored to facilitate fit-up. Counterbore dimensions for standard pipe wall thicknesses are given in ASME B16.25. It is important that the net minimum wall thickness resulting from the counterboring process be compared with the code minimum wall thickness t_m to be sure that an under-thickness condition does not occur at the joints.

Example B2.1 demonstrates how the previously discussed concepts associated with the design pressure may be applied to a typical problem.

Example B2.2—Problem Statement. Two motor-driven boiler feed pumps installed on the ground floor of a powerhouse supply 3000 gal/min (11,360 L/min) of water at 350°F (177°C) to a boiler drum which is 260 ft (79.3 m) above the pump discharge. Each pump discharge is NPS 8 (DN 200) pipe, and the common discharge header running up to the boiler drum is NPS 12 (DN 300) 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 relieve the excess pressure which the pump will develop if the discharge valve is closed while the pump is operating. The normal working pressure at the boiler drum is 2520 psig (17,390 kPa gage). The set pressure of the drum safety valve is 2600 psig (17,940 kPa gage) (MAWP of the drum) and the shutoff head of the pumps is 8700 ft (2652 m). The piping material is ASTM A106, Grade C with an allowable working stress of 17,500 psi (121 MPa) over the temperature range of −20°F (−29°C) to 650°F (343°C). The corrosion allowance is 0.08 in (2.0 mm), and the design code is ASME B31.1. A simplified system diagram is shown in Fig. B2.1.

Find the required nominal pipe wall thickness for the NPS 12 header (DN 300) (zone 1) and the NPS 8 (DN 200) pump discharge lines upstream of the isolation valve (zone 2). Also check the adequacy of the pipe wall thickness in zone 2, assuming the relief valve on one pump does not operate when the associated discharge valve is closed.

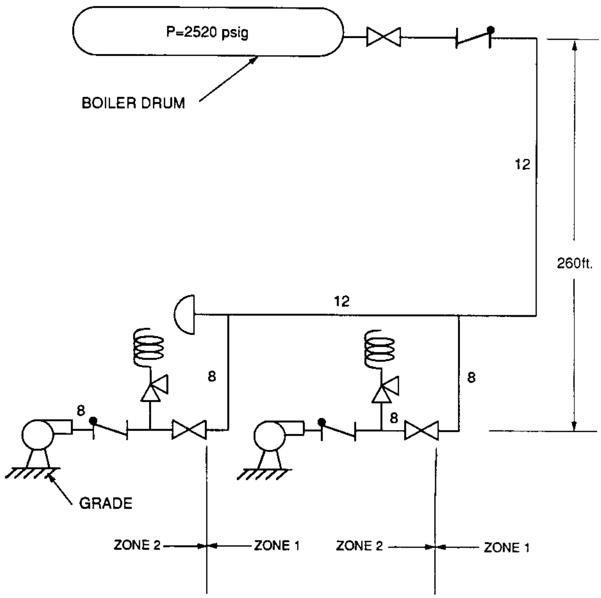

FIGURE B2.1 Simplified powerhouse boiler feed system.

Solution. Note that the piping material used in this example is A106, Grade C. A106, Grade C material has a higher allowable stress than A106, Grade B. The use of Grade C material results in thinner pipe when compared to Grade B. However, A106, Grade B is an acceptable material for this application.

The specific volume of 350°F (177°C) saturated water is 0.01799 ft^3/lbm (1.123 cm^3/g). Correcting the specific volume for the effects of compression to 2500 psig (17,250 kPa gage) results in a revised specific volume of 0.01778 ft^3/lbm (1.109 cm^3/g), or a density of 56.24 lb/ft^3 (901 kg/m^3). The static head P_{st} above the pumps due to the elevation of the boiler drum is

$$P_{st} = 260\,\text{ft} \times 56.24\,\frac{\text{lb}}{\text{ft}^3} \times \frac{1\,\text{ft}^2}{144\,\text{in}^2}$$

$$= 102\,\frac{\text{lb}}{\text{in}^2}\ (704\,\text{kPa gage})$$

Paragraph PG 61.1 of ASME Section I, *Power Boilers,* requires the piping in Zones 1 and 2 be designed for a pressure equal to the highest setting of any safety valve plus 3 percent. In this case the safety valve is set at 2600 psig (17,490 kPa gage). Considering the static head of 102 psi (704 kPa gage) and the 3 percent margin above the set pressure of 2600 psig (17,940 kPa gage), the minimum design pressure for piping is

$$2600 + 0.03(2600) + 102 = 2780\ \text{psig}\ (19{,}180\ \text{kPa gage})$$

ASME B31.1, Paragraph 122.1.3 (A.1), requires the design pressure of the feedwater piping from the boiler, up to and including the required stop valve and check valve, to exceed the maximum allowable working pressure by either 25 percent or 225 psi (1,553 kPa gage), whichever is less. In compliance with this requirement the design pressure for the piping must be greater than $2520 + 225 + 102 = 2847$ psig.

Considering the two code issues discussed above, a design pressure of 2850 psig (19,670 kPa gage) will be used for this piping.

ASME B31.1 uses the approach provided by Eqs. (B2.2) and (B2.4) to yield a combined minimum wall thickness function represented by Eq. (B2.5):

$$t = \frac{PD_o}{2(SE + Py)} \tag{B2.2}$$

$$t_m = t + A \tag{B2.4}$$

$$t_m = \frac{PD_o}{2(SE + Py)} + A \tag{B2.5}$$

The values of the variables are

$$\begin{aligned} P &= 2850\,\text{psig}\ (19{,}670\,\text{kPa gage}) \\ D_o &= 12.75\,\text{in}\ (324\,\text{mm}) \\ S &= 17{,}500\,\text{psi}\ (121\,\text{MPa}) \\ y &= 0.4 \\ A &= 0.08\,\text{in}\ (2.0\,\text{mm}) \\ E &= 1.0 \end{aligned}$$

Substituting in Eq. (B2.5) yields the following:

$$\begin{aligned} t_m &= \frac{2850(12.75)}{2(17{,}500 + 2850 \times 0.4)} + 0.08 \\ &= 1.055\,\text{in}\ (26.8\,\text{mm}) \end{aligned}$$

The commercial wall thickness tolerance for seamless rolled pipe is +0, −12½ percent; therefore, to determine the required nominal wall thickness, t_m must be divided by 0.875:

$$t_{\text{nom}} = \frac{1.055}{0.875} = 1.21\,\text{in}\ (30.7\,\text{mm})$$

Referring to ASME B36.10M, the nearest commercial NPS 12 (DN 300) pipe whose wall thickness exceeds 1.21 in is Schedule 160 with a nominal wall thickness of 1.312 in (33.3 mm). Therefore NPS 12 (DN 300), Schedule 160 pipe meeting the requirements of ASTM A106 Grade C is chosen for this application. Refer to App. E2 and E2M for pipe properties.

This calculation does not consider the effects of bending the pipe during fabrication. If bending during fabrication is planned, the required wall thickness may need to be increased depending upon the radius of the bend. Refer to the design code for the appropriate bending allowance.

Similarly, the required wall thickness for the NPS 8 (DN 200) pipe upstream of the isolation valves (zone 2) is calculated. The outside diameter of NPS 8 (DN 200) standard pipe is 8.625 in (219 mm). Substituting in Eq. (B2.5) yields:

$$t_m = \frac{2850(8.625)}{2(17{,}500 + 2850 \times 0.4)} + 0.08$$
$$= 0.739 \text{ in}$$

$$t_{\text{nom}} = 0.845 \text{ in } (21.5 \text{ mm})$$

The required nominal wall thickness is 0.845 in (21.5 mm). By referring to ASME B36.10M, or App. E2 or E2M, NPS 8 (DN 200), XXS pipe with a nominal wall thickness of 0.875 in (22.2 mm) is selected.

The only remaining step is to check the pipe in zone 2 to see if the wall thickness is adequate to withstand the pump's shutoff head. It is reasonable to assume that failure of a relief valve to operate is a very low-probability event. The occasional load criteria of ASME B31.1, Paragraph 102.2.4 will be invoked, and it will be assumed that the pump operation at shutoff head due to a failed relief valve occurs less than 1 percent of the time. Therefore, the allowable stress is 20 percent higher than the basic Code allowable stress of 17,500 psi (121 MPa). The higher allowable stress is denoted as S' where

$$S' = 1.20 \times S$$

$$S' = 1.20 \times 17{,}500 = 21{,}000 \text{ psi } (145 \text{ MPa})$$

The maximum pressure rating of the NPS 8 (DN 200), XXS pipe is calculated using Eq. (B2.1):

$$P = \frac{2SE(t_m - A)}{D_o - 2y(t_m - A)} \tag{B2.1}$$

For this evaluation, the value of S is set equal to S', and $E = 1.00$ for seamless pipe; t_m is assumed equal to 87½ percent of the nominal wall thickness of the pipe.

$$t_m = 0.875 \times 0.875 = 0.766 \text{ in } (19.5 \text{ mm})$$

then

$$P = \frac{2 \times 21{,}000\,(0.766 - 0.08)}{8.625 - 2 \times 0.4\,(0.766 - 0.08)}$$
$$= 3568 \text{ psig } (24{,}620 \text{ kPa gage})$$

The shutoff head of the pump was given as 8700 ft. The density of pressurized water at 350°F was previously determined to be 56.24 lb/ft^3. The pressure equivalent to the shutoff head may be calculated based upon this density.

$$P = 8700 \text{ ft} \times 56.24 \frac{\text{lb}}{\text{ft}^3} \times \frac{1 \text{ ft}^2}{144 \text{ in}^2}$$
$$= 3398 \text{ psig } (23{,}450 \text{ kPa gage})$$

Since the 3568 psig (24,620 kPa gage) occasional pressure rating of the NPS 8 (DN 200), XXS pipe exceeds the 3398 psig (23,450 kPa gage) shutoff head of the pump, the piping is adequate for the intended service.

The design procedures presented in the foregoing problem are valid for steel or other code-approved wrought materials. They are not 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).

Design Temperature

The *design temperature* is the temperature at which the allowable stresses for all pressure-retaining parts of the piping system are assigned. The design temperature must be equal to or greater than the maximum sustained temperature that the pressure-retaining components will experience during all normal and expected abnormal modes of operation.

The design temperature of the system's pressure-retaining metal parts is normally assumed equal to the maximum free-stream fluid temperature. The effects of any internal or external heat sources such as heat tracing must be considered, as must any temperature excursions occurring as a result of control system error. The design temperature should be set at or above the peak of these temperature excursions.

While the pressure-integrity design is based upon the design temperature, most other thermally related aspects of the design are based upon the normal operating temperature. The *normal operating temperature* is the temperature achieved by the system fluid while the system is operating in full-load, steady-state, nontransient conditions. It is lower than the design temperature. The normal operating temperature is used as the basis for all thermal design analyses that relate to the structural integrity of the piping system, including the thermal flexibility analysis, the spring hanger sizing and setting calculations, and the thermally induced anchor movement calculations. If a system has more than one ''normal'' operating mode (i.e., the system runs at different temperatures or has branches that run at different temperatures for different operating modes), then multiple thermal analysis calculations at all normal operating temperatures may be necessary to fully qualify the design.

Deadweight

The *deadweight* (self-weight) of a piping system consists of the sum of the distributed loads from the weight of the pipe itself, its thermal insulation, and/or other uniformly applied covering materials, plus the sum of the weights of any permanently installed concentrated loads such as valves, strainers, or other in-line appurtenances.

External loads on the piping system such as wind loads, snow and ice loads, and the weight of the fluid contents are considered as live loads. They are distinct from deadweight in that live loads may be variable both in magnitude and/or in the percentage of the total system operating time during which they act. An additional distinction is that the effects of live loads may be removed from the piping system while those of deadweight may not (without dismantling the system, of course).

Both the ASME Boiler and Pressure Vessel Code, Section I, *Power Boilers,* and the ASME B31.1 Code, *Power Piping,* require that the effects of deadweight and other sustained loads be considered in verifying the pressure integrity of components. Subpart PG-22 of ASME Section I, *Power Boilers,* provides the following generalized rule:

> Stresses due to hydrostatic head shall be taken into account in determining the minimum thickness required unless noted otherwise. Additional stresses imposed by effects other than working pressure or static head, which increase the average stress by more than 10% of the allowable working stress, shall also be taken into account. These effects include the weight of the component and its contents, and the method of support.[8]

ASME B31.1, *Power Piping,* specifies more definitive rules to account for the effects of deadweight. Paragraph 104.8 provides closed-form equations to evaluate the effect of the simultaneous application of the internal pressure, deadweight, and other sustained loads on the design of a piping system. Equation 11 is used to calculate the piping system stress and to compare the calculated stress with code acceptance criteria. Details of this analytical approach are discussed in Chap. B4.

Wind Load

The majority of all piping system installations are indoors where the effects of wind loading can be neglected. However, there are sufficient numbers of outdoor piping installations where wind loading can be a significant design factor. *Wind load*, like deadweight, is a uniformly distributed load that acts along the entire length, or that portion of the piping system that is exposed to the wind. The difference is that while deadweight loads are oriented in the downward vertical direction, wind loads are horizontally oriented and may act in any arbitrary direction. Since wind loads are oriented in the horizontal direction, the regular deadweight support system of hangers and anchors may have little or no ability to resist these loads. Consequently, when wind loading is a factor, a separate structural evaluation and wind load support system design is required.

Determination of the magnitude of the wind loadings is based upon empirical procedures developed for the design of buildings and other outdoor structures. Analysis of piping system stresses and support system loads is accomplished by using techniques that are similar to those applied for deadweight design. Details of these procedures are discussed in Chap. B4.

Snow and Ice Loads

Snow and ice loads, like wind loads, need to be considered in the design of piping systems which are installed outdoors, particularly if the installation is made in the northern latitudes. Since snow and ice loads act in the vertical direction, they are treated the same as deadweight loads. In design, they are simply added as distributed loads in the deadweight analysis, as discussed earlier in the section ‘‘Deadweight’’ and in Chap. B4.

Snow Loads. ANSI/ASCE 7–95, *Minimum Design Loads for Buildings and Other Structures,*[9] provides recommendations and data for developing design loadings due to snow. The methods used in this standard are generally applicable to sloping or horizontal flat surfaces such as building roofs or grade slabs. While the methods of ANSI/ASCE 7–95 are completely appropriate for extended flat surfaces, they may be too conservative for application to a smooth, round, horizontal pipe which will tend to shed most of the snowfall which may land upon it. The data provided in ANSI/ASCE 7–95 can, however, be used as part of a rational method to estimate the maximum probable snow load on an outdoor piping system, and the following procedure may be adopted for piping.

Table B2.4 provides ground snow loading data for 204 locations where those data are measured. The column marked 2% Annual Probability represents the loading associated with the maximum probable snowfall that is likely to occur in a 50-year period. Based upon the data from Table B2.4, the following relationship for the design snow load for outdoor piping systems may be used:

$$W_s = \frac{1}{2} D_o S_{50} \tag{B2.6}$$

where W_s = design snow load to be added to other distributed loads acting on pipe, lb/ft
D_o = outside diameter of pipe or insulation, ft
S_{50} = 2 percent probability snow loading for nearest appropriate location from Table B2.4, lb/ft^2

This formula assumes the snow remaining on the pipe will take the shape of an equilateral triangle whose base equals the outside diameter of the pipe.

Ice Loads. Ice storms are sporadic in the frequency of their occurrence and in their intensity. Weather records dating back to the turn of the 20th century for a typical midwestern state relate instances of ice storm deposits of ⅛ in (3.2 mm) to 4 in (102 mm) in thickness. *The American Weather Book*[10] cites examples of ice accumulations of up to 8 in (203 mm) in northern Idaho (1961) and 6 in (152 mm) in northwest Texas (1940) and New York State (1942).

The paper ''Estimated Glaze Ice and Wind Loads at the Earth's Surface for the Contiguous United States''[11] documents the results of a comprehensive study of ice storm records over the 50-winter period from 1919–1920 through 1968–1969. A statistical evaluation of the data indicates that 50-year maximum probable values for ice deposits from a single storm vary over the range of 2 in (51 mm) to 3 in (76 mm), with the larger accumulations occurring in the upper midwest, far west, and northeast and the smaller accumulations occurring in the lower midwest and southeast. Given the relative infrequency of ice storms, this range probably represents a reasonable value for design. It is suggested that the designer contact local weather or agricultural authorities to determine whether any better region-specific data exist.

Once the appropriate design thickness is determined, the following formula may be used to estimate the unit loadings on an exposed pipeline due to ice accumulation:

$$W/L = 1.36t(D_o + t) \tag{B2.7}$$

where W/L = unit loading on pipe, lb/ft
D_o = outside diameter of pipe or insulation lagging, in
t = assumed iced covering thickness, in

Table B2.5 provides a tabulation of ice loadings based upon Eq. (B2.7) for piping systems up to 30 in (762 mm) in outside diameter and ice thicknesses up to 3 in (76 mm).

TABLE B2.4 Ground Snow Loads at 204 National Weather Service Locations at Which Measurements Are Made

Location	Ground snow load* (lb/ft^2) Years of record	Maximum observed	2% Annual probability
ALABAMA			
Birmingham	40	4	3
Huntsville	33	7	5
Mobile	40	1	1
ARIZONA			
Flagstaff	38	88	48
Tucson	40	3	3
Winslow	39	12	7
ARKANSAS			
Fort Smith	37	6	5
Little Rock	24	6	6
CALIFORNIA			
Bishop	31	6	8
Blue Canyon	26	213	242
Mt. Shasta	32	62	62
Red Bluff	34	3	3
COLORADO			
Alamosa	40	14	14
Colorado Springs	39	16	14
Denver	40	22	18
Grand Junction	40	18	16
Pueblo	33	7	7
CONNECTICUT			
Bridgeport	39	21	24
Hartford	40	23	33
New Haven	17	11	15
DELAWARE			
Wilmington	39	12	16
GEORGIA			
Athens	40	6	5
Atlanta	39	4	3
Augusta	40	8	7
Columbus	39	1	1
Macon	40	8	7
Rome	28	3	3
IDAHO			
Boise	38	8	9
Lewiston	37	6	9
Pocatello	40	12	10
ILLINOIS			
Chicago—O'Hare	32	25	17
Chicago—Midway	26	37	22
Moline	39	21	19
Peoria	39	27	15
Rockford	26	31	19
Springfield	40	20	21
INDIANA			
Evansville	40	12	17
Fort Wayne	40	23	20
Indianapolis	40	19	22
South Bend	39	58	41

TABLE B2.4 Ground Snow Loads at 204 National Weather Service Locations at Which Measurements Are Made (*Continued*)

Location	Ground snow load* (lb/ft^2)		
	Years of record	Maximum observed	2% Annual probability
IOWA			
Burlington	11	15	17
Des Moines	40	22	22
Dubuque	39	34	32
Sioux City	38	28	28
Waterloo	33	25	32
KANSAS			
Concordia	30	12	17
Dodge City	40	10	14
Goodland	39	12	15
Topeka	40	18	17
Wichita	40	10	14
KENTUCKY			
Covington	40	22	13
Jackson	11	12	18
Lexington	40	15	13
Louisville	39	11	12
LOUISIANA			
Alexandria	17	2	2
Shreveport	40	4	3
MAINE			
Caribou	34	68	95
Portland	39	51	60
MARYLAND			
Baltimore	40	20	22
MASSACHUSETTS			
Boston	39	25	34
Nantucket	16	14	24
Worcester	33	29	44
MICHIGAN			
Alpena	31	34	48
Detroit City	14	6	10
Detroit Airport	34	27	18
Detroit—Willow	12	11	22
Flint	37	20	24
MICHIGAN (*Continued*)			
Grand Rapids	40	32	36
Houghton Lake	28	33	48
Lansing	35	34	36
Marquette	16	44	53
Muskegon	40	40	51
Sault Ste. Marie	40	68	77
MINNESOTA			
Duluth	40	55	63
International Falls	40	43	44
Minneapolis-St. Paul	40	34	51
Rochester	40	30	47
St. Cloud	40	40	53
MISSISSIPPI			
Jackson	40	3	3
Meridian	39	2	2
MISSOURI			
Columbia	39	19	20
Kansas City	40	18	18
St. Louis	37	28	21
Springfield	39	14	14
MONTANA			
Billings	40	21	15
Glagsow	40	18	19
Great Falls	40	22	15
Havre	26	22	24
Helena	40	15	17
Kalispell	29	27	45
Missoula	40	24	22
NEBRASKA			
Grand Island	40	24	23
Lincoln	20	15	22
Norfolk	40	28	25
North Platte	39	16	13
Omaha	25	23	20

TABLE B2.4 Ground Snow Loads at 204 National Weather Service Locations at Which Measurements Are Made (*Continued*)

Location	Ground snow load* (lb/ft^2) Years of record	Maximum observed	2% Annual probability
NEBRASKA (*Continued*)			
Scottsbluff	40	10	12
Valentine	26	26	22
NEVADA			
Elko	12	12	20
Ely	40	10	9
Las Vegas	39	3	3
Reno	39	12	11
Winnemucca	39	7	7
NEW HAMPSHIRE			
Concord	40	43	63
NEW JERSEY			
Atlantic City	35	12	15
Newark	39	18	15
NEW MEXICO			
Albuquerque	40	6	4
Clayton	34	8	10
Roswell	22	6	8
NEW YORK			
Albany	40	26	27
Binghamton	40	30	35
Buffalo	40	41	39
NYC—Kennedy	18	8	15
NYC—LaGuardia	40	23	16
Rochester	40	33	38
Syracuse	40	32	32
NORTH CAROLINA			
Asheville	28	7	14
Cape Hatteras	34	5	5
Charlotte	40	8	11
Greensboro	40	14	11
Raleigh-Durham	36	13	14
NORTH CAROLINA (*Continued*)			
Wilmington	39	14	7
Winston-Salem	12	14	20
NORTH DAKOTA			
Bismark	40	27	27
Fargo	39	27	41
Williston	40	28	27
OHIO			
Akron-Canton	40	16	14
Cleveland	40	27	19
Columbus	40	11	11
Dayton	40	18	11
Mansfield	30	31	17
Toledo	36	10	10
Youngstown	40	14	10
OKLAHOMA			
Oklahoma City	40	10	8
Tulsa	40	5	8
OREGON			
Astoria	26	2	3
Burns	39	21	23
Eugene	37	22	10
Medford	40	6	6
Pendleton	40	9	13
Portland	39	10	8
Salem	39	5	7
Sexton Summit	14	48	64
PENNSYLVANIA			
Allentown	40	16	23
Erie	32	20	18
Harrisburg	19	21	23

TABLE B2.4 Ground Snow Loads at 204 National Weather Service Locations at Which Measurements Are Made (*Continued*)

Location	Ground snow load* (lb/ft^2) Years of record	Maximum observed	2% Annual probability
PENNSYLVANIA (*Continued*)			
Philadelphia	39	13	14
Pittsburgh	40	27	20
Scranton	37	13	18
Williamsport	40	18	21
RHODE ISLAND			
Providence	39	22	23
SOUTH CAROLINA			
Charleston	39	2	2
Columbia	38	9	8
Florence	23	3	3
Greenville-Spartanburg	24	6	7
SOUTH DAKOTA			
Aberdeen	27	23	43
Huron	40	41	46
Rapid City	40	14	15
Sioux Falls	39	40	40
TENNESSEE			
Bristol	40	7	9
Chattanoga	40	6	6
Knoxville	40	10	9
Memphis	40	7	6
Nashville	40	6	9
TEXAS			
Abilene	40	6	6
Amarillo	39	15	10
Austin	39	2	2
Dallas	23	3	3
El Paso	38	8	8
Fort Worth	39	5	4
Lubbock	40	9	11
Midland	38	4	4
San Angelo	40	3	3
San Antonio	40	9	4
Waco	40	3	2
Wichita Falls	40	4	5
UTAH			
Milford	23	23	14
Salt Lake City	40	11	11
Wendover	13	2	3
VERMONT			
Burlington	40	43	36
VIRGINIA			
Dulles Airport	29	15	23
Lynchburg	40	13	18
National Airport	40	16	22
Norfolk	38	9	10
Richmond	40	11	16
Roanoke	40	14	20
WASHINGTON			
Olympia	40	23	22
Quillayute	25	21	15
Seattle-Tacoma	40	15	18
Spokane	40	36	42
Stampede Pass	36	483	516
Yakima	39	19	30
WEST VIRGINIA			
Beckley	20	20	30
Charleston	38	21	18
Elkins	32	22	18
Huntington	30	15	19
WISCONSIN			
Green Bay	40	37	36
La Crosse	16	23	32
Madison	40	32	35
Milwaukee	40	34	29
WYOMING			
Casper	40	9	10
Cheyenne	40	18	18
Lander	39	26	24
Sheridan	40	20	23

* To convert ground loadings to kN/m^2, multiply tabular value by 0.0479.

Source: Adapted from ANSI/ASCE 7-95, *Minimum Design Loads for Buildings and Other Structures,* American Society of Civil Engineers, New York, 1996, Table C7-1.

TABLE B2.5 Weight Loadings for Ice Coatings on Horizontal Pipelines

Pipe or lagging actual outside diameter D_o [in (mm)]	Ice coating thickness [in (mm)]					
	¼ (6.4)	½ (12.7)	1 (25.4)	1½ (38)	2 (51)	3 (76)
2 (51)	0.8	1.7	4.0	7.1	10.9	20.4
2½ (64)	0.9	2.0	4.8	8.2	12.2	22.4
3 (76)	1.1	2.4	5.4	9.2	13.6	24.5
4 (102)	1.5	3.1	6.8	11.2	16.3	28.6
6 (152)	2.1	4.4	9.5	15.3	21.8	36.7
8 (203)	2.8	5.8	12.2	19.4	27.2	44.9
10 (254)	3.5	7.1	15.0	23.5	32.6	53.1
12 (305)	4.2	8.5	17.7	27.5	38.1	61.2
14 (356)	4.8	9.9	20.4	31.6	43.5	69.4
16 (406)	5.5	11.2	23.1	35.7	49.0	77.5
18 (457)	6.2	12.6	25.8	39.8	54.4	85.7
20 (508)	6.9	13.9	28.6	43.9	59.8	93.8
22 (559)	7.6	15.3	31.3	47.9	65.3	102
24 (610)	8.3	16.7	34.0	52.0	70.7	110
26 (660)	8.9	18.0	36.7	56.1	76.2	118
28 (711)	9.6	19.4	39.4	60.2	81.6	127
30 (762)	10.3	20.7	42.2	64.3	87.0	135

Note: Ice thickness *t*, in. Loadings are in pounds per foot. To convert loadings to kg/m, multiply tabular loadings by 1.488.

Seismic (Earthquake) Loads

Under certain circumstances it is necessary or desirable to design a piping system to withstand the effects of an earthquake. Although the applications are not extensive, piping system seismic design technology is well developed and readily accessible. Many currently available piping stress analysis computer programs are capable of performing a detailed seismic structural and stress analysis, in addition to the traditional deadweight and thermal flexibility analyses. Most of these programs run on desktop microcomputers.

Because of the higher construction costs and design complexities introduced by the application of seismic design criteria, this type of work is normally done only in response to specific regulatory, code, or contractual requirements. An overview of these applications is outlined in the following paragraphs.

Nuclear Power Plants. Title 10, Part 50, of the Code of Federal Regulations requires that safety-related piping systems* in nuclear power plants be designed to withstand the effects of certain severe natural phenomena, including earthquakes.[12]

In general, seismic analysis of nuclear power plant piping is done to demonstrate that the piping system satisfies one of two specific objectives:

* A simplified definition of a safety-related system is one whose failure could result in a reduced capacity to mitigate the effects of an accident or which could ultimately result in the uncontrolled release of radioactivity into the environment. See Refs. 13 and 14.

Operability. Under this objective, the design of the piping system is such that it will retain its pressure-integrity status and remain capable of performing its design function before, during, and after the occurrence of a postulated seismic event at the plant site. Piping systems designed to meet operability criteria must normally comply with code-specified stress limits during the postulated earthquake.

Structural Integrity. Piping designed to this objective is not required to remain functional or to retain its pressure integrity during or after an earthquake. The only requirement is that the piping system retain its gross structural integrity so that it does not deflect excessively or cause the generation of secondary missiles. Either condition could cause impact and subsequent unacceptable damage to adjacent safety-related structures, systems, or components. Piping designed under this classification is normally allowed to attain stress levels, due to seismic excitation, well in excess of normal code limits. Of all modern industrial and commercial applications, nuclear power plant piping systems represent the largest single class of seismically qualified piping systems in service. It is estimated that the typical nuclear power unit contains approximately 100,000 ft of seismically qualified piping.[15] It is safe to say that the nuclear piping industry is the largest single contributor to the technology of seismic design of piping systems.

General Building Codes. Some building codes have rules devoted to the seismic design of buildings and other structures. Two examples of such codes are The Uniform Building Code[16] and The BOCA National Building Code.[17] In addition to these two codes, ANSI/ASCE 7–95, *Minimum Design Loads for Buildings and Other Structures,*[9] provides extensive guidance in the area of seismic design. Note that while both of the above-mentioned codes are intended to be national in scope and applicability, neither is mandatory unless specifically adopted by local statute or ordinance. Both ANSI/ASCE 7–95 and the BOCA code are specifically applicable to all piping systems NPS 2½ (DN 65) and larger in nominal pipe size, with smaller limits of applicability stated for piping systems in boiler rooms [NPS 1¼ (DN 32) and larger] and natural gas piping [NPS 1 (DN25) and larger]. The Uniform Building Code is only applicable to fire protection sprinkler piping systems. Since both the Uniform Building Code and the BOCA code are more specific to the design of buildings and related structures, their application to piping systems requires some interpretation on the part of the designer. It is therefore recommended that the piping designer who intends to apply these codes maintain a close liaison with jurisdictional authorities.

Contractual Arrangements. Under certain specific circumstances, an agreement might be reached to seismically design a piping system; this agreement would be made between the owner-operator of the system and another organization which has a vested financial interest in it, such as an insurance carrier. An example might involve the construction of a pipeline which carries a hazardous fluid through a seismically active region where no statutory design requirement exists. Under such circumstances the insurance carrier might require the owner to seismically design the line to limit the risk of rupture during a seismic event. Should a piping designer become involved in such an arrangement, every effort should be made to ensure that the design criteria are carefully specified, understood, and agreed to by all parties prior to starting design work.

Methods of Analysis. There are three methods of analysis in common use for the seismic design of piping systems: the *static coefficient method,* the *response spectra modal analysis method,* and the *time history analysis method.* The static coefficient method is the easiest to apply, but due to simplifying assumptions, it provides a very conservative design. The response spectra modal analysis method is about midway in complexity and provides a lesser degree of conservatism. This is the method used for the majority of piping system analysis and design. The time history analysis method is the least conservative and the most difficult to apply. This approach is used only when the most exacting (and least conservative) results are required. All three methods are discussed, in detail, in Chap. B4.

Effects of Seismic Analysis on Overall Design. The design costs associated with seismic analysis of a piping system go far beyond the simple costs of analyzing the piping system for ''just one more load.'' A piping system is usually a subordinate part of a larger structure, typically a building. Prior to analyzing the piping system to determine its seismic behavior, the analyst must first develop the forcing function. This involves a detailed analysis of the building structure itself to determine its response to the ground motion associated with the postulated earthquake. Development of the postulated ground motion normally requires the consideration of actual earthquake-induced ground motion data taken from geologically similar sites where earthquakes have actually been experienced. It can be seen that the seismic analysis of the piping system is really just the top layer of a multitier design exercise that requires the consideration of the dynamic interaction of a number of complex structures. Such an exercise requires a considerable expenditure of human resources and computer time.

Seismic qualification of a piping system also leads to greatly increased construction costs. Some of the principal contributors to those costs include the following:

Higher Loads. If the frequency content of the seismic excitation forces is coincident with the natural frequencies of the piping system, resonant amplification of the forcing function loads will occur. The resulting support system loads will be much higher than corresponding loads caused by deadweight effects alone. These higher loads translate to heavier, and consequently more expensive, supporting structures.

Multiple Load Paths. The multidirectional load characteristics of an earthquake acting on seismically qualified piping systems invariably result in the application of upward and/or lateral reactions on the building structure which would not be present in a typical static design.

Special Supporting Devices. Seismic design of high-temperature piping systems represents an especially challenging exercise. Thermal expansion effects require that the piping system be flexibly supported to allow for free thermal growth. The dynamic aspects of the design usually require that the piping system be rigidly supported during the earthquake to transfer the seismic loads back into the building structure. The simultaneous consideration of these diametrically opposite requirements results in the need to use a significant number of specialized (and therefore expensive) pipe support devices called *snubbers.* Snubbers lock up and carry load when subjected to the rapidly varying vibratory loads associated with an earthquake, yet remain free to permit thermal movement of the piping system during the relatively slow expansion or contraction caused by temperature changes.

Hydraulic Transient Loads

Of all the loading conditions that a piping system may experience in service, hydraulic transients are among the most damaging. The most common form of damage caused by hydraulic transient loads is the failure of pipe supports and supporting structures. However, occasional breaches of pressure integrity are also experienced, particularly where large-diameter thin-walled pipe is involved.

Two common types of hydraulic transient loads are waterhammer and relief valve discharge. These two load sets are discussed in detail below.

Waterhammer. If the velocity of water or other liquid flowing in a pipe is suddenly reduced, a pressure wave results which travels up and down the piping system at the speed of sound in the liquid. Depending upon the initial velocity and physical properties of the liquid and the mechanical properties of the piping system, the peak value of the pressure wave may exceed the steady-state pressure.

Waterhammer frequently occurs in systems that are subject to rapid changes in fluid flow rate, including systems with rapidly actuated valves, fast-starting pumps, and check valves. It is most severe in systems which convey fully condensed liquids; however, it is possible to develop waterhammer-type pressure transients in systems containing two-phase fluids and gases, although the magnitude of the pressure rise for these systems will generally be lower. The techniques used to calculate the magnitude of a waterhammer-induced pressure rise are discussed in Chap. B8.

Waterhammer must be considered in the design of those systems where it is likely to occur. For systems designed to codes that provide higher allowable stress criteria for occasional loads, the waterhammer-induced peak pressure should be evaluated under that loading category. For systems designed to codes which do not provide alternative design criteria for occasional loads, the design pressure may be set high enough to envelop the waterhammer-induced peak pressure. The designer is cautioned that this approach can result in an extremely conservative design, which may be prohibitively expensive. Consequently consideration of alternatives may be required.

Relief Valve Discharge Loads. Because of their rapid opening characteristics and generally high flow rates, the actuation of relief valves frequently results in the application of significant loads to the associated piping system. These loads are caused by the differential pressures across the valves, differential pressure between the valve discharge and the downstream discharge piping, and differential pressure between the discharge piping and the receiver or atmosphere. In addition, momentum effects caused by velocity changes at high flow rates result in secondary loads, which act on the attached (or nonattached) piping. These secondary loads must be reacted to by the supporting structures. Actual pipe stress levels in inadequately designed relief valve installations can exceed code-allowable values, and failures of such installations are common. Depending upon the pressure and temperature conditions of the system fluid, such failures can represent a personnel safety hazard as well as a costly economic issue.

To assist the piping designer in developing a safe and functional relief valve installation, Appendix II of ASME B31.1, *Power Piping,* was developed and issued.[18] This nonmandatory appendix to the code provides an extensive treatment of the relevant factors which must be considered to produce a successful design. Relief valve discharge loadings typically occur during a very small percentage of the total system operating time; consequently, they can be treated as occasional loads. The design for load combinations that include these hydraulic transients can therefore be based upon the higher code-defined stress limits.

Acoustically Induced Vibration Loads

When a piping system is exposed to fluctuating pressure disturbances, or pulsations, it frequently responds by vibrating. The magnitude and nature of the piping system vibration are dependent upon the frequency and energy content of the excitation. Low- to moderate-level periodic excitation, such as the pressure pulsations from positive-displacement or constant-speed centrifugal pumps, will not ordinarily excite significant levels of response in the piping system as long as the excitation frequencies are well removed from the natural vibrating frequencies of the pipe. If the pulsation frequency of the disturbance coincides with the natural frequencies of the piping system, however, resonant vibration can occur. Resonant response normally results in vibratory amplitudes many times that which would occur if the disturbance did not coincide with the natural frequencies of the piping system. Broad-spectrum or random excitation of the type associated with cavitation, bubble collapse, and extreme pressure reductions can also lead to resonant vibration. This type of vibration is known as *self-excited vibration.* The piping system draws energy from the broad-spectrum excitation and responds by vibrating at its own fundamental or harmonic natural frequencies.

Resonant response, whether due to the effects of fixed frequency or random excitation, can lead to unacceptable piping system damage. Cyclical stress reversals associated with resonant vibration can result in short-term fatigue failures, which may occur after only a few hours or days of operation. Reduction in the cyclical stress levels and attendant failures can be accomplished through a number of approaches.

When the excitation is a constant-frequency disturbance, ''decoupling'' the vibrating piping system from the source of excitation can often be accomplished by changing the frequency of the excitation. If the disturbance comes from a positive-displacement pump, changing the running speed will change the disturbance frequency. If a centrifugal pump is involved, a change in running speed or in the number of vanes on the impeller may have a beneficial effect.

Changing the natural response frequencies of the piping system can also mitigate the effects of fixed-frequency vibration. Again, the objective is to decouple or ''detune'' the piping system relative to the disturbance. This can often be accomplished by adding supplemental bracing to the pipe or by breaking the system into smaller segments by introducing flexible elements.

When the excitation is broad spectrum or random in nature, detuning by changing the piping system natural frequencies is usually not effective in solving the vibration problem. The modified piping system will continue to draw energy from the broad-spectrum excitation and will vibrate at its new natural frequencies. Mitigation of this class of problems usually requires the reduction of the energy level of the excitation or the ''strengthening'' of the piping system.

A large number of broad-spectrum vibration problems are the result of high-differential pressure reduction systems.[19] Excitation (noise) reduction for these types of systems can often be accomplished by the use of low-noise cage-type pressure-reducing valves or multiple orifices arranged for staged pressure reduction. In other cases, where the excitation is the result of turbulence, geometric changes to the piping system to smooth out the flow or reduce average and local velocities can have a beneficial effect.

The objective of the strengthening process is the reduction of piping system stresses to a level where fatigue failures are substantially eliminated. Much can be accomplished by the elimination of stress concentrations through the removal of geometric discontinuities. Examples of these discontinuities include hanger lugs, insulation supports, and small pipe (vent, drain, test, etc.) connections. Additionally, where changes in section are required, they should be effected by gradual, smooth changes in contour and generous fillet radii.

Special attention should be paid to all in-line welding done on pipelines that are subject to fluid-induced vibratory loadings. The use of inert-gas root pass welding with filler metal addition is recommended. This technique reduces the potential for the formation of critical root defects, which can lead to crack initiation and propagation. Radiographic and ultrasonic examinations done in excess of the minimum requirements may prove cost-effective in identifying stress-intensifying volumetric defects. Where such examinations are planned, weld backing rings should not be used since they complicate the job of interpreting the examination results.

Finally, the use of pipe wall thickness in excess of that required for pressure-integrity design alone has been found to be beneficial in mitigating the effects of fluid transient loads.

An experience was encountered which required the replacement of the main turbine bypass piping on a large nuclear unit because of multiple short-term acoustically induced vibration failures. The original system was made up of NPS 30 (DN 750) and NPS 24 (DN 600) × ⅜ in (10-mm) wall pipe. The original design of the piping wall thickness was based upon pressure-integrity considerations alone. Several pressure boundary failures were experienced at pipe support lugs, hanger clamps, and vent and drain connections after only a few hours of operation. The replacement material was NPS 30 (DN 750) × 1-in (25.4-mm) wall and NPS 24 (DN 600) × 1¼-in (31.5-mm) wall. In addition, special attention was paid to the elimination of nonaxisymmetric discontinuities and minor welded attachments. Lugs

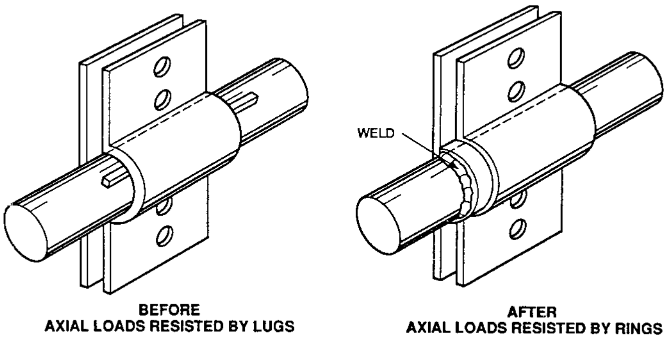

FIGURE B2.2 Use of shear rings instead of lugs to reduce localized pipe wall stress.

were replaced by rings, as shown in Fig. B2.2, and unnecessary small pipe connections were removed. The upgraded system has been operated extensively without experiencing any further pressure boundary failures.

Since systems subject to flow-induced vibratory loads usually see those loads over much of their service life, their design should be based upon sustained loading criteria with no increase permitted in allowable stress.

Relative Anchor Movements

Every piping system requires some type of support system to function properly. The piping system can be supported from a building or other structure on traditional pipe hangers or from the ground on piers or bents. It even can be supported from another piping system. As long as all the piping system's support points remain motionless relative to the piping system and relative to one another, the system is unaffected. However, if some of the piping system's supports move relative to the pipe or relative to one another, the piping system will attempt to follow that motion and will experience a change in its state of stress. This condition is called *relative anchor movement.*

Relative movements of a piping system's supports can be caused by a number of phenomena. Some of the more common causes include

- Thermal expansion-related movement of the connection point on a larger piping system, where the subject system is attached
- Earthquake-induced relative movements of the various points on a building's structure where the subject piping system is supported
- Thermal expansion or mechanically induced movements of a piping connection (nozzle) on a machine, pressure vessel, or heat exchanger

The amount of stress, or more properly stated, the change in stress, that a piping system experiences from relative anchor movements is a function of two variables: the magnitude of the anchor movement and the stiffness of the system. As one might expect, larger movements will result in greater changes in stress. Moreover, for a given magnitude of movement, stiffer piping systems will experience greater changes in stress than those that are less stiff. In general, systems that are shorter, have fewer changes in direction, and are made up of larger-diameter pipe are stiffer than those for which the opposite conditions are true.

Certain phenomena result in a loading case in which both the magnitude and direction of the piping system's terminal movements are known, such as movements resulting from the thermal expansion of a pipe. In such a case, the known magnitudes and directions of the anchor movement are input to the piping system stress analysis, and the attendant stress levels are predicted.

There are other cases, however, in which only the magnitude of the anchor movement is known. Examples include earthquake-induced anchor movements or the movements of a building due to wind loading. In this case, the magnitude of the movement is input into analysis as known, but the direction is assumed such that the worst-case change in the state of stress of the piping system under study results. This approach ensures that the piping system stress analysis is conducted in the most conservative manner.

ENVIRONMENTAL FACTORS

The generalized definition of the term *stress,* when used in a structural connotation, is force per unit area. The mathematical equation that relates the stress σ to the load F and the load-resisting area A is

$$\sigma = \frac{F}{A} \tag{B2.8}$$

The loss of a piping system's pressure or structural integrity is invariably the result of its having attained a higher-than-acceptable state of stress. In the preceding section, it was shown that such an increase in the state of stress of a piping system could result from the application of one or more external (or internal) loads. In terms of Eq. (B2.8), it can be said that loading conditions or loads increase the state of stress of the system by increasing the value of the numerator F of the equation. However, there is a second mechanism through which the state of stress of a piping system may increase, perhaps to the point at which a failure occurs. That mechanism is the loss, or deterioration, of the load-resisting area A.

In this chapter the various mechanisms that result in the loss or deterioration of the viable load-resisting base material of a piping system are titled *environmental factors.* An attribute that is common to all environmental factors is that they effectively shorten the useful life of the piping system compared to what it would be if the factors were not present. The following discussion is limited to four specific environmental factors: corrosion, erosion, physical damage, and erosion-corrosion.

Corrosion

Within the context of this chapter, *corrosion* is the loss of load-carrying material in the pipe wall due to an electrochemical reaction between the piping material and the process fluid, or the environment.

Corrosion is normally accounted for in design by the provision of additional material in the pipe wall, the use of a suitable coating or lining, or the specification of a corrosion-resistant material. Frequently, the method used to deal with corrosion depends upon the corrosion rate.

For steel corroding in water, the corrosion rate is strongly influenced by the amount of oxygen present and by the temperature. The effect of these variations

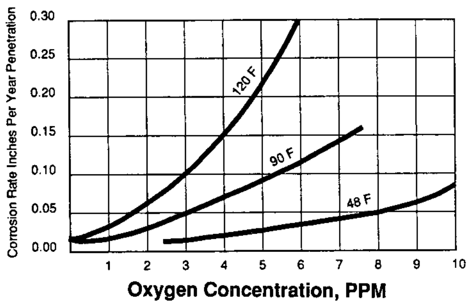

FIGURE B2.3 Effects of oxygen concentration on the corrosion of low-carbon steel in tap water at different temperatures. (*Reprinted by permission from* Corrosion Basics: An Introduction, *National Association of Corrosion Engineers, Houston, 1984, Figure 8.1, p. 149.*)

is shown in Fig. B2.3. Similar data exist in the literature for a wide variety of piping system materials and corrodants. Through the use of these data, the corrosion rate for a given process can often be estimated.

When corrosion is anticipated to occur at a slow, regular rate and this rate can be reliably predicted, it may be accommodated by the provision of excess material in the pipe wall known as the *corrosion allowance.* This excess material will be consumed over the design life of the piping system and therefore cannot be counted upon to serve any other purpose such as pressure integrity, structural strength, or mechanical strength. The relationship of the corrosion allowance to the other components of the total pipe wall thickness was discussed earlier in the section ‘‘Design Pressure.’’

In cases in which the corrosion rate is prohibitively high or would result in unacceptable contamination of the process fluid, a *pipe lining* or corrosion-resistant piping material may be specified.

Linings tend to be fragile; therefore, their use is limited to applications in which abrasion or other physical injury is not likely to occur. Many linings are temperature-sensitive and cannot be used in extreme-temperature service. Most lined pipe is fabricated from plain-carbon steel, although fiberglass-reinforced plastic pipe with an integral, chemically resistant plastic lining is available. Refer to Chaps. B9–B12 and D2. Joining methods are a significant factor in lined pipe design, and the joints must not allow any corrosion-sensitive material to be exposed to the process stream.

Abrasion or other physical attack may damage a lining and expose the corrosion-sensitive substrate. In such cases, a homogeneous material is frequently warranted. If the original corrosion-resistant surface is physically damaged, the remaining material will continue to resist corrosion. Many corrosion-resistant materials have good high-temperature strength properties and as such are used where linings or plain-carbon steels will not survive. Many near-ambient-temperature, high-corrosion-rate applications can be successfully and economically accommodated by the use of plastic pipe.

Erosion

Erosion is the wearing away of a surface by abrasion. The abrasion may be the result of particles suspended in the fluid stream, or it may be the result of direct action by the fluid itself.

When the erosion rate is small and consistent, and reliable quantitative data concerning that erosion rate are available, an *erosion allowance* may be provided in the design. The erosion allowance is analogous to the corrosion allowance discussed earlier in the sections ''Design Pressure'' and ''Corrosion.'' Excess material, over and above that required for pressure integrity and structural and mechanical strength, is provided. This excess material is allowed to waste away over the design life of the piping system.

When the erosion process is not readily quantified, a more qualitative approach to design is normally taken. One approach is to specify special erosion-resistant piping system materials. High-hardness materials are generally effective in resisting erosion. An alternative approach is to modify the piping system geometry to minimize or eliminate turbulent flow, direct pipe wall impingement, and vortex flow, all of which increase piping system erosion.

Physical Damage

Physical damage or abuse also can be a significant factor in the design of piping systems. This is particularly true of low-pressure, thin-walled piping, which has little resistance to external loadings.

Direct buried pipe is subject to damage from soil pressure and loads from overhead traffic. Uniformly distributed soil pressure loads can normally be estimated with reasonable accuracy and the pipe designed for these loads using methods described in the literature.[20] The effects of heavy concentrated overhead loads cannot be accounted for as easily. Consequently, pipelines which run under heavily trafficked roads or railroad tracks are frequently run through oversized sleeves or conduits which prevent the imposed loads from being directly transmitted to the pipe.

Piping systems of all sizes that carry important services, toxic fluids, or high-pressure, high-temperature fluids should be physically protected from impact from passing motor vehicle traffic, including such vehicles as industrial forklift trucks. The preferred method of protection is to route the piping outside the reach of passing traffic. Where this is not possible, substantial barriers should be erected to protect the piping from impact.

Small-diameter piping takeoffs from large headers such as vents, drains, and instrumentation source connections are particularly prone to damage from unspecified external loads. Common design practice is to make the small piping from the header out to the first isolation valve at least one schedule heavier than called for by the pressure design. Similarly, the first isolation or *root valve* is normally made one or two pressure classes heavier than called for by pressure design considerations. These steps may make the small lines durable enough to resist random impact or other undefined external loadings that can occur during shipping, construction, or operation.

Erosion-Corrosion (Flow-Assisted Corrosion)

When iron or steel corrodes in water, a soluble oxide layer called *magnetite* is formed. During steady-state conditions, the magnetite attains a constant protective thickness which promotes a uniform corrosion rate. If the magnetite layer is ''swept'' by a water film deposited by wet steam or by a locally high-velocity jet in a liquid stream, the dissolution rate of the magnetite increases. This results in an increase in the localized corrosion rate and an attendant loss of metal from the surface. Since the sweeping away of the oxide layer is an essential part of this corrosion process, it has been named *erosion-corrosion* [also known as *flow-assisted corrosion* (*FAC*)].

A number of factors have been found to affect the rate of erosion-corrosion (FAC) in piping systems. In wet-steam systems, percentage of moisture, material composition, pH and water chemistry, temperature, oxygen level, and flow path geometry have all been found to be significant.[21] In water piping systems, piping material, temperature, pH and oxygen level, and flow path geometry all affect the rate of erosion-corrosion.[22] Among the variables cited above, two that can be readily controlled in design are the piping system materials and the flow path geometry.

Carbon steel is known to be highly susceptible to erosion-corrosion (FAC). Both chromium-molybdenum (Cr-Mo) and austenitic stainless steels are significantly less susceptible. The EPRI publication ''Erosion/Corrosion in Nuclear Power Plant Steam Piping: Causes and Inspection Program Guidelines''[21] cites a study that indicates the rate of erosion-corrosion (FAC) in Cr-Mo wet-steam piping is one-tenth that of carbon steel. Low-alloy Cr-Mo materials can usually be substituted for carbon steel without any other significant design changes. The materials are readily welded to each other, and both have similar physical properties such as tensile and yield strength, density, and thermal expansion coefficient.

The substitution of austenitic stainless steels will normally require some additional engineering. These materials have a thermal expansion rate that averages 50 percent greater than that of plain-carbon steel. Accordingly, increases in terminal reactions and predicted pipe support movements can be expected. These may require that the pipe be rerouted or that different pipe support components be provided.

Piping system geometry plays an important role in mitigating the effects of erosion-corrosion (FAC). High, localized velocities, vortex flow, jet, and direct stream impingement all increase the rate of magnetite dissolution and therefore increase the rate of erosion-corrosion. Gradual transitions in flow section pipe size and geometric changes to smooth out variations in flow velocity and the provision of shallow-angle intersections will all have a beneficial effect. The removal of discontinuities such as weld backing rings and sharp edges at branch connections will also reduce erosion-corrosion (FAC). The areas immediately downstream of valves and flow measurement orifices are frequently prone to erosion-corrosion due to vortex formation. These areas will benefit from the addition of flow liners or the substitution of erosion-corrosion-resistant alloys.

MATERIALS-RELATED CONSIDERATIONS

The variety of piping system materials currently in use is extensive and continually growing. The purpose of this section is to provide a brief overview of the common engineering properties of those materials and to describe how those properties influence the design process. For the most part, discussions of specific material characteristics will be limited to plain-carbon and low-alloy steel piping materials. Many of the concepts discussed, however, are applicable to virtually all piping materials.

Strength

Most piping design codes relate the allowable working stresses for materials to their yield strength or ultimate tensile strength at the working temperature. For example, the *allowable working stresses* for materials used for construction in accordance with ASME B31.1, *Power Piping,* are developed using rules defined in the ASME Boiler and Pressure Vessel Code, Section II, *Materials.* At any temperature below the creep range, those rules require that the allowable working stress be set at a value no greater than the lowest of the following alternatives[23]:

- One-fourth of the specified minimum tensile strength at room temperature
- One-fourth of the tensile strength at operating temperature
- Two-thirds of the specified minimum yield strength at room temperature
- Two-thirds of the yield strength at operating temperature

As the temperature of most common pressure-retaining materials increases from ambient, their tensile and yield strengths decrease. Application of the above rules ensures that the decreasing strength of piping materials, with increasing temperature, is reflected in the allowable stresses used for design.

At temperatures within the creep range, the allowable working stress is set at a value equal to the lowest of the following[23]:

- 100 percent of the average stress for a creep rate of 0.01 percent/1000 h
- 67 percent of the average stress for rupture at the end of 100,000 h
- 80 percent of the minimum stress for rupture at the end of 100,000 h

When carbon steels are exposed to temperatures greater than 775°F (413°C) for long periods, the carbide phase may convert to graphite. *Graphitization* causes steels to experience brittle fracture at stress levels well below their short-term rupture strength. In recognition of this phenomenon, the ASME B31.1, *Power Piping,* Code provides the following warning statement in the allowable-stress tables:

> Upon prolonged exposure to temperatures above 775°F (413°C), the carbide phase of carbon steel may be converted to graphite.[24]

For temperatures in excess of 775°F (413°C), chromium-molybdenum low-alloy steels or high-alloy stainless steels may be used. These steels offer almost complete freedom from graphitization and enhanced creep-rupture resistance. ASME B31.1 allows the use of these materials at temperatures up to 1200°F (649°C).

Toughness

Toughness, or *ductility*, is the ability of a material to resist impact, to withstand repeated reversals of stress, or to absorb energy when stressed beyond the elastic limit. Steel is normally considered to be a ductile material. Contrary to expectation, however, steels sometimes rupture without prior evidence of distress. Under certain conditions, steel may shatter just as glass. In piping, however, this behavior generally occurs only at low temperatures.

The *transition temperature* for any steel is the temperature above which the steel behaves in a predominantly ductile manner and below which it behaves in a predominantly brittle manner. Steel with a high transition temperature is more likely to behave in a brittle manner during fabrication or in service. It follows that a steel with a low transition temperature is more likely to behave in a ductile manner. Therefore, steels with low transition temperatures are generally preferred for service involving severe stress concentrations, impact loading, low operating temperatures, or a combination of all three.

Table B2.6 indicates the low-temperature limitations of various piping materials.

TABLE B2.6 Low-Temperature Operating Limits for Selected Piping Materials

Low-temp. limit, °F (°C)	Material and suitable ASTM designation	Comments
0 (−18)	Mild steel (A53, A135)	No requirements other than suitable pressure rating
−20 (−29)	Mild steel (A53, A135)	Reduce pressure rating 1% for each 1°F below zero, or Charpy impact test, 15 ft·lb at design temperature
−50 (−46)	Killed or limited carbon steel (A333, Gr. 1)	Charpy impact test, 15 ft·lb at design temperature
−150 (−101)	3½% Ni-steel (A333, Gr. 3)	Charpy impact test, 15 ft·lb at design temperature
−325 (−198)	Austenitic stainless steel (types 304, 316, etc.)	Limited carbon content
No limit	Nonferrous copper, brass, aluminum	Aluminum, copper, brass

Low-alloy steels may be used at low temperatures 0°F (−18°C) when they have a Charpy keyhole impact value of at least 15 ft · lb (2.1 kg · m) at the lowest design temperature. Austenitic stainless steels with limited carbon content, copper and copper alloys, and aluminum do not experience transitions in impact strength from ductile to brittle fracture and, therefore, may be used for low temperatures without pressure-rating penalties.

Low-temperature piping is generally covered with insulation which, in addition to limiting heat transfer, helps provide protection from external impact. This, however, is not sufficient insurance against the type of damage that could result if a pipe should fracture.

Additional perspectives on materials strength and toughness are provided in Chap. A3.

Corrosion Resistance

Considered as a material property, *corrosion resistance* is a measure of a piping system material's relative inertness to chemical attack from a specific process fluid at the system's normal operating temperature, or its environment (see earlier section ''Corrosion''). The importance of considering the system's operating temperature cannot be overemphasized. It is well known that many chemical reactions are highly temperature-dependent. A particular piping system material could be virtually immune to chemical attack by a specific corrodant at one temperature, while prone to excessive attack by the same corrodant at a higher temperature.

Within this context, then, it is clear that there is no such thing as a universally corrosion-resistant material. All common piping system materials react with some process fluids (corrodants) at certain temperatures. Therefore, when one is pursuing a ''corrosion-resistant'' material for a specific application, the objective is to identify a material whose corrosion rate in the presence of a specific corrodant is negligible, or at least acceptable, over the design life of the piping system.

It is important also to consider the effect corrosion may have on the process fluid. Under certain conditions, the dissolution of the base metal or the corrosion products into the process stream may require economic or technical considerations that go beyond the piping system's pressure-containing parts. In some cases, the major consideration in choosing a piping system material may be the preservation of the chemical purity of the process fluid. Such is usually the case in choosing piping system materials that handle food products and piping used in many chemical process operations.

THERMAL INSULATION

Whenever the surface temperature of a piping system differs significantly from that of its surrounding environment, the potential need for an insulation system exists. An insulation system serves three principal purposes:

- The significant reduction in the transfer of thermal (heat) energy to or from the surface of the piping system
- The prevention of moisture formation and collection on the surface of the piping system due to condensation

- The prevention of potentially injurious personnel contact with the surface of the piping system

The reduction in heat transfer to or from the surface of a piping system will minimize the gain or loss in temperature of the process fluid, thus maximizing the capability of the fluid to perform its intended function. Minimizing heat exchange between the piping system and the environment also minimizes the unwanted heating or cooling of the environment. This improves the comfort level for the inhabitants, or improves the operating conditions for equipment.

Most insulating systems used aboveground consist of preformed components that are mechanically attached to the pipe. Low-temperature insulation is frequently made of expanded cellular plastic or foam rubber materials. Moderate-temperature insulations are frequently made from glass-fiber products. High-temperature insulation is usually made of preformed cementations or refractory materials or blankets made from ceramic fibers. Insulation used for buried pipe is frequently in loose granular form, so it can be poured loosely into the trench to surround the pipe and isolate it from the ground environment.

If the surface temperature of a piping system is less than the dew point of the surrounding air, water vapor in the air will condense on the surface of the pipe. This condition can be detrimental. The *condensation* can collect and drip onto surfaces below the pipe, thus doing damage. The condensate can also saturate the piping insulation, thus significantly increasing its thermal conductivity and reducing its insulating capability. To prevent condensation on an insulated pipe, the airborne water vapor must be prevented from reaching the pipe surface. This is normally accomplished by providing a *vapor barrier* at the outer surface of the insulation. An adequate vapor barrier may be constructed from a well-fitted metal jacket, an extruded plastic or rubber coating, or a spiral-wrapped impervious tape coating. Whatever the form, the vapor barrier must prevent the airborne water vapor from entering the pores of the insulation and migrating toward the cool pipe surface, where condensation can occur.

Extremely hot or cold piping systems can pose a contact safety hazard to personnel in the vicinity. Surface temperatures above 135°F (57°C) can cause severe burns to unprotected skin, and temperatures below approximately 20°F (−7°C) can cause freeze damage. Thermal insulation can be designed such that the insulation surface temperature is maintained in a safe range.

Frequently piping systems that are otherwise uninsulated will have insulation installed in accessible areas to provide personnel protection.

For a complete treatment of the engineering principles involved in designing a thermal insulation system for piping, see Chap. B7.

SIZING OF A PIPING SYSTEM

The term *sizing of a piping system* refers to the completion of two independent design functions: the fluid flow design and the pressure-integrity design. The purpose of the fluid flow design is to determine the minimum acceptable inside diameter of the various segments of the piping system. The purpose of the pressure-integrity design is to determine the minimum acceptable pipe wall thickness and the pressure ratings of the in-line components.

System Fluid Flow Design

The objective of the fluid flow design is to determine the minimum acceptable inside diameter of each segment of the piping system that will accommodate the design flow rate while maintaining the pressure drop and flow velocity within reasonable limits.

Most piping systems use pumps to develop the pressure or head required to maintain the system design flow rates. Piping system pressure drops must be maintained within reasonable values to limit the installed size of the system pumps and their prime movers. Pump and prime-mover size limitations are necessary to control initial system construction costs and continuing system operating costs. The optimum pipe size is based on an economic tradeoff between the installed capital cost of the piping system and the sum of the capital plus lifetime operating costs of the pumping system.

System flow velocities are limited by design to avoid a number of potential operating problems. These problems have already been discussed in previous sections of this chapter. In the absence of any other formal or more limiting criteria, the flow velocities given for water in Table B2.7 and for steam in Table B2.8 are considered reasonable for normal industrial applications.

TABLE B2.7 Reasonable Design Velocities for Water Flowing through Pipes

Service condition	Reasonable velocity, ft/s (m/s)
Boiler feed	8–15 (2.5–4.6)
Pump suction and drain lines	4– 7 (1.2–2.1)
General service	4–10 (1.2–3.0)
City water	to 7 (to 2.1)

Source: Crane Technical Paper 410, *Flow of Fluids through Valves, Fittings, and Pipe.* The Crane Company, New York, 1985, pp. 3–6.

The detailed fluid flow design of a piping system requires the consideration of a number of fluid parameters including flow rate, viscosity, density, and pipe wall frictional drag. Further discussions of this aspect of the pipe sizing process are provided in Chap. B8.

TABLE B2.8 Reasonable Design Velocities for Steam Flowing through Pipes

Condition of steam	Pressure P [psig (kPa)]	Service	Reasonable velocity V	
			ft/min	m/s
Saturated	0–25 (173)	Heating (short lines)	4,000– 6,000	20– 31
Saturated	25 (173) and up	Powerhouse equipment, process piping, etc.	6,000–10,000	31– 51
Superheated	200 (1380) and up	Boiler and turbine leads, etc.	7,000–20,000	36–100

Source: Crane Technical Paper 410, *Flow of Fluids through Valves, Fittings, and Pipe,* The Crane Company, New York, 1985, pp. 3–16.

Pressure-Integrity Design

The pressure-integrity design of a piping system normally requires the consideration of at least two issues. The first is the determination of the minimum or nominal pipe wall thickness, and the second is the determination of the pressure rating of the in-line components, such as fittings and valves.

Determination of Pipe Wall Thickness

After the fluid design is complete and the minimum inside diameters of the various segments of the piping system are determined, the piping pressure-integrity design may proceed. The major steps in the process are as follows:

1. Using the minimum inside diameter determined from the fluid flow evaluation, select the next-larger standard nominal or outside diameter (OD) size pipe from the listings provided in ASME B36.10M for standard wrought steel pipe or B36.19M for stainless-steel pipe (see earlier section ''The ASME Pressure Classification System'').
2. Based upon the fluid and service, select a suitable piping material, and if necessary, determine the required corrosion, erosion, joining, or mechanical strength allowances.
3. Using equations provided in the design code, calculate the required minimum wall thickness to provide for pressure integrity and allowances.
4. Refer to ASME B36.10M or B36.19M to select an appropriate nominal wall thickness or schedule. Refer to App. E2 and E2M.
5. Confirm that the standard manufacturing tolerance will not reduce the nominal wall thickness selected in step 4 below the minimum required, as calculated in step 3.
6. Confirm that the inside diameter of the pipe selected, based upon the nominal wall thickness selection of step 4, is compatible with the minimum inside-diameter requirements obtained from the fluid flow evaluation.

The process described above is demonstrated in the following example:

Example B2.3. A carbon-steel pipe having a required minimum inside diameter of 11.2 in (284 mm) is to transport water at 700 psig (4830 kPa gage) and 90°F (32°C). The design code is ASME B31.1, and the design life is 8 years. The water has a nominal oxygen content of 1 ppm. Butt-welded construction is used.

Evaluation. An economical grade of seam-welded carbon-steel pipe (ASTM A53 Grade A) is selected. From ASME B31.1, Appendix A, Table A-1, the allowable working stress at 90°F (32°C) is 10,200 psi (70.4 MPa). From Fig. B2.3, the corrosion rate is estimated at 0.02 in (0.5 mm) per year. The pressure-integrity design will be based upon ASME B31.1, Paragraph 104.1.2, equation (3)[25]:

$$t_m = \frac{PD_o}{2(SE + Py)} + A \qquad \text{(B2.5)}$$

From ASME B36.10M, NPS 12 (DN 300) [12.75-in (324-mm) OD] is tentatively selected.

Using the stated 8-year design life and 0.02 in/yr corrosion rate, the total corrosion allowance of $8 \times 0.02 = 0.16$ in (4 mm) is calculated. Butt-welded construction is specified; therefore, no additional wall thickness allowance for joining (threading, grooving, etc.) is required.

From ASME B31.1, Table 104.1.2(A), $y = 0.4$ is selected for ferritic steels at temperatures at or below 900°F (482°C).

Equation (B2.5) may now be used to calculate the required minimum wall thickness:

$$t_m = \frac{PD_o}{2(SE + Py)} + A \tag{B2.5}$$

$$= \frac{700 \times 12.75}{2(10{,}200 + 0.4 \times 700)} + 0.16$$

$$= 0.586 \text{ in } (14.9 \text{ mm})$$

From ASME B36.10M, under the listings for NPS 12 (DN 300), Schedule 80 pipe with a nominal wall thickness of 0.688 in (17.5 mm) is tentatively selected.

The wall thickness tolerance for ASTM A53 pipe, which is +0, −12½ percent, is checked next:

$$0.688 \times 0.875 = 0.602 \text{ in}$$

$$0.602 \text{ in } (15.3 \text{ mm}) > 0.586 \text{ in } (14.9 \text{ mm})$$

Finally, the nominal inside diameter is checked against the minimum flow diameter:

$$12.75 - 2(0.688) = 11.374 \text{ in}$$

$$11.374 \text{ in } (289 \text{ mm}) > 11.2 \text{ in } (284 \text{ mm})$$

The problem requirements are satisfied; NPS 12 (DN 300) seam welded Schedule 80 pipe meeting ASTM Specification A53 Grade A is acceptable.

The previous example did not consider the effects of bending on the pipe wall. In most instances the pressure design will dominate in the determination of pipe wall thickness. However, if the pipe span between supports is unusually long or if the pipe has a very heavy in-line component, such as a valve, then the longitudinal bending stress may dominate the design. This facet of the design is considered in the piping stress analysis discussion of Chap. B4.

To complete this chapter, five more example problems are presented. They demonstrate the concepts developed and bring them together to show how the design of a simple piping system might proceed.

Determining the Pressure Class for In-Line Components

The first two examples provided here demonstrate the process used to determine the pressure classification for in-line components. The first demonstrates the selection process for a standard flange; the second demonstrates the selection process for a special-class valve.

Example B2.4. An NPS 16 (DN 400) carbon-steel pipeline operates at 840 psig (5800 kPa gage) and 740°F (393°C). Select a standard weld-neck flange for the service.

Evaluation. Table B2.1 lists various materials of construction for standard pipe flanges. Under Material Group 1.1, ASTM Specification A105, *Forgings, Carbon Steel, for Piping Components,* is listed. Next refer to Table B2.3, which lists ASME pressure-temperature ratings for Material Group 1.1 flanges. Noted that a Class 600 flange has a pressure-temperature rating of 1010 psig (6970 kPa gage) at 750°F (399°C). Since this rating exceeds the requirements of 840 psig (5800 kPa gage) at 740°F (393°C), this flange is acceptable.

Example B2.5. An NPS 12 (DN 300) butt-welding end gate valve is required to operate at 2350 psig (16,220 kPa gage) and 1015°F (546°C). The valve material is ASTM A217 Grade WC9. Determine the appropriate ASME pressure classification.

Evaluation. Tables B2.9a and B2.9b list the pressure-temperature ratings for standard and special class valves of ASTM A217 Grade WC9.

There are two correct answers to this problem. The first and simplest answer is to select a standard Class 4500 valve from Table B2.9a. This valve has a pressure-temperature rating of 2625 psig (18,040 kPa gage) at 1050°F (566°C) and obviously meets the stated requirements. However, this valve may prove to be a very expensive alternative since Class 4500 valves are massively constructed, and valve prices vary according to the weight of the material used in their construction.

The second alternative is to consider the Special Class 2500 valves whose ratings are provided in Table B2.9b. *Special-class valves* undergo mandatory nondestructive examinations and, if necessary, defect repairs to allow them to qualify for higher pressure-temperature ratings. For a more detailed discussion of special-class valves, the reader is referred to Section 8 of ASME B16.34.[26] To determine whether a Special Class 2500 valve will meet the requirements of Example B2.3, a linear interpolation of the ratings in Table B2.9b is required. The process is illustrated below:

Temperature, °F	*Pressure, psig*
1000	2715
1015	P
1050	1820

$$\frac{1050 - 1015}{1050 - 1000} = \frac{P - 1870}{2715 - 1870}$$

$$\frac{35}{50} = \frac{P - 1870}{845}$$

$$\begin{aligned} P &= 1870 + \frac{35}{50} \times 845 \\ &= 2462 \text{ psig } (16{,}990 \text{ kPa gage}) \end{aligned}$$

Since the interpolated pressure rating of 2462 psig (16,990 kPa gage) is greater than the specified requirement of 2350 psig (16,220 kPa gage), a Special Class 2500 valve will satisfy the requirements of Example B2.5.

Determining the Design Conditions and Pressure Class of a Piping System

To minimize procurement complications and storage and handling problems during the construction phase, piping systems are frequently designed for the maximum

TABLE B2.9a Pressure-Temperature Ratings for Standard Class Valves Made of ASTM A217 Grade WC9 Body Material*

Temperature (°F)	Working pressure by classes (psig)†							
	150	300	400	600	900	1500	2500	4500
−20 to 100	290	750	1,000	1,500	2,250	3,750	6,250	11,250
200	260	750	1,000	1,500	2,250	3,750	6,250	11,250
300	230	730	970	1,455	2,185	3,640	6,070	10,925
400	200	705	940	1,410	2,115	3,530	5,880	10,585
500	170	665	885	1,330	1,995	3,325	5,540	9,965
600	140	605	805	1,210	1,815	3,025	5,040	9,070
650	125	590	785	1,175	1,765	2,940	4,905	8,825
700	110	570	755	1,135	1,705	2,840	4,730	8,515
750	95	530	710	1,065	1,595	2,660	4,430	7,970
800	80	510	675	1,015	1,525	2,540	4,230	7,610
850	65	485	650	975	1,460	2,435	4,060	7,305
900	50	450	600	900	1,350	2,245	3,745	6,740
950	35	375	505	755	1,130	1,885	3,145	5,665
1000	20	260	345	520	780	1,305	2,170	3,910
1050	20‡	175	235	350	525	875	1,455	2,625
1100	20‡	110	145	220	330	550	915	1,645
1150	20‡	70	90	135	205	345	570	1,030
1200	20‡	40	55	80	125	205	345	615

* For special limitations placed on the materials covered by this table, see ASME B16.34, Table 2-10A.
† To convert working pressures to kPa gage, multiply tabular values by 6.9.
‡ For welding-end valves only. Flanged-end valve ratings terminate at 1000°F.
Source: Adapted from ASME B16.34, *Valves—Flanged, Threaded, and Welding End,* American Society of Mechanical Engineers, New York, 1996, Table 2-10A, p. 45.

conditions permitted for each pressure class. This allows conservatism, which can accommodate changes in design conditions as a result of design development and minimizes the need to specify and buy different piping for each individual application. In addition, this approach provides an added allowance in the event of unexpected deterioration of the pipe wall thickness in service.

The following examples provide an illustration of determining the design pressure and design temperature for a piping system. They also provide insight into the method of establishing the pressure-temperature rating or pressure class of an entire piping system.

Example B2.6

Fluid:	Water
Normal conditions:	350 psig (2415 kPa gage) @ 350°F (177°C)
Maximum conditions:	(1) 375 psig (2588 kPa gage) @ 390°F (199°C)
	(2) 435 psig (3002 kPa gage) @ 375°F (191°C)

TABLE B2.9b Pressure-Temperature Ratings for Special Class Valves Made of ASTM A217 Grade WC9 Body Material*

Temperature (°F)	Working pressure by classes (psig)† 150	300	400	600	900	1500	2500	4500
−20 to 100	290	750	1,000	1,500	2,250	3,750	6,250	11,250
200	290	750	1,000	1,500	2,250	3,750	6,250	11,250
300	285	740	990	1,485	2,225	3,705	6,180	11,120
400	280	725	965	1,450	2,175	3,620	6,035	10,865
500	275	720	960	1,440	2,160	3,600	6,000	10,800
600	275	720	960	1,440	2,160	3,600	6,000	10,800
650	275	715	955	1,430	2,145	3,580	5,965	10,735
700	275	710	955	1,425	2,135	3,555	5,930	10,670
750	265	690	920	1,380	2,070	3,450	5,750	10,350
800	260	675	895	1,345	2,020	3,365	5,605	10,095
850	245	645	855	1,285	1,930	3,215	5,355	9,645
900	230	600	800	1,200	1,800	3,000	5,000	9,000
950	180	470	630	945	1,415	2,355	3,930	7,070
1000	125	325	435	650	975	1,630	2,715	4,885
1050	85	220	290	435	645	1,095	1,820	3,280
1100	55	135	185	275	410	685	1,145	2,055
1150	35	85	115	170	255	430	715	1,285
1200	25	50	70	105	155	255	430	770

* For special limitations placed on the materials covered by this table, see ASME B16.34, Table 2-10B.
† To convert working pressures to kPa gage, multiply tabular values by 6.9.
Source: Adapted from ASME B16.34, *Valves—Flanged, Threaded, and Welding End,* American Society of Mechanical Engineers, New York, 1996, Table 2-1.10B, p. 46.

Condition 1 has a maximum duration of 3 h. Condition 2 has a maximum duration of 10 min in any 24-h operating period.

Pipe sizes: NPS 6 (DN 150), NPS 10 (DN 250), NPS 14 (DN 350)

Evaluation. The piping system being considered is designed in accordance with ASME B31.1; however, the approach discussed below can be used to design a piping system in accordance with other codes.

The fluid and the temperature dictate the use of carbon-steel piping. Assume the following materials:

Pipe: ASTM A106 GR B

Valve body: ASTM A216, WCB

Flanges: ASTM A105

Determine the pressure-temperature ratings for all conditions. The ratings are determined from the pressure-temperature tables of ASME B16.5[1] and ASME

B16.34.[26] The flange and valve materials are in material group 1.1; refer to ASME B16.5, Table 2-1.1, and ASME, B16.34, Table 2-1.1.

350 psig (2415 kPa gage) @ 350°F (177°C)—Class 300
375 psig (2588 kPa gage) @ 390°F (199°C)—Class 300
435 psig (3002 kPa gage) @ 375°F (191°C)—Class 300

Since Class 300 is required for each condition, the design conditions should be selected so as not to exceed the pressure-temperature ratings of Class 300. Otherwise, the design conditions will be overly conservative.

Determine design conditions from normal and maximum conditions. The design conditions are selected to ensure that the minimum wall thickness requirements of ASME B31.1 are met. This requires consideration of two factors: pressure and temperature.

Pressure. The greater the pressure, the greater the required wall thickness of the pipe. The design pressure must be selected so that each of the following requirements is satisfied:

- The design pressure shall be not less than the maximum sustained operating pressure (MSOP) within the piping system including the effects of static head (ASME B31.1, Paragraph 101.2.2).
- The design pressure shall be of sufficient magnitude that the stress resulting from a variation in pressure and/or temperature in the piping system does not exceed the allowable stress by more than 15 percent during 10 percent of any 24-h operating period, or by more than 20 percent during 1 percent of any 24-h operating period (see ASME B31.1, Paragraph 102.2.4).

Maximum condition 1 will cause a stress in the pipe wall which is less than 15 percent over the stress caused by the normal condition pressure

$$\frac{375}{350} < 1.15$$

But the duration exceeds 10 percent of a 24-h operating period. Therefore, maximum condition 1 must be considered as a sustained condition, which requires that the design pressure not be less than 375 maximum condition. Condition 2 has a duration of less than 1 percent of a 24-h operating period and will not cause a stress greater than 20 percent over the allowable stress using a design pressure of 375 psig.

$$\frac{435}{375} < 1.2$$

Therefore, maximum condition 2 can be treated as an occasional condition.

The minimum acceptable design conditions are

$$\text{Design pressure} = 375\ \text{psig}\ (2588\ \text{kPa gage})$$

$$\text{Design temperature} = 390^\circ\text{F}\ (199^\circ\text{C})$$

Example B2.7

Fluid:	Steam
Normal conditions:	400 psig (2760 kPa gage) @ 600°F (316°C)
Maximum conditions:	575 psig (3970 kPa gage) @ 600°F (316°C)

This condition occurs in less than 1 percent of any 24-h operating period.

Pipe sizes: NPS 12 (DN 300), NPS 18 (DN 450)

Evaluation. The fluid and the temperatures allow the use of carbon steel. Assume the following materials:

Pipe: ASTM A106, Gr. B
Valve body: ASTM A216 WCB
Flanges: ASTM A105

Determine the pressure-temperature ratings. With the help of ASME B16.5, Table 2-1.1, and ASME B16.34,[26] Table 2-1.1, the suitable classes for the normal and maximum conditions are established as follows:

400 psig (2760 kPa gage) @ 600°F (316°C)—Class 300
575 psig (3970 kPa gage) @ 600°F (316°C)—Class 400

The normal condition requires Class 300 flanges and valves while the maximum condition requires Class 400 flanges and valves. The maximum permissible (sustained) pressure for Class 300 at 600°F (316°C) is 550 psig (3,800 kPa gage). This pressure may be exceeded in the same manner as discussed in Example B2.4 (15 percent for 10 percent of the time; 20 percent for 1 percent of the time). Thus, the peak pressure that the flanges and valves may be exposed to is greater than the system maximum of 575 psig (3,970 kPa gage). Therefore, Class 300 can be used.

Determine the design conditions such that 575 psig exceeds the design pressure by not more than 20 percent.

$$\text{Design pressure} \times 1.2 = 575\,\text{psig (3970 kPa gage)}$$

$$\text{Design pressure} = \frac{575}{1.2} = 479\ \text{psig (3310 kPa gage)}$$

The minimum design conditions are

$$\text{Design pressure} = 479\,\text{psig (3310 kPa gage)}$$

$$\text{Design temperature} = 600^\circ\text{F (316}^\circ\text{C)}$$

This piping can also be designed for the maximum design condition permitted for Class 300 flanges made from material group 1.1 per ASME B16.5. The maximum design conditions, per B16.5, Table 2-1.1, are

$$\text{Design pressure} = 550\,\text{psig (3800 kPa gage)}$$

$$\text{Design temperature} = 600^\circ\text{F (316}^\circ\text{C)}$$

The piping may also be designed for the minimum design conditions shown above (479 psig at 600°F). Sometimes this can result in substantial savings in material, fabrication, and installation costs. This is particularly true for high-pressure and high-temperature applications that require the use of low- and high-alloy steels.

Design of Piping for Internal and External Pressure

Example B2.8. An NPS 24 (DN 600) seamless steel pipeline carries purified water from an onshore water treatment plant to an offshore island-sited nuclear power plant. The line runs vertically down in an open shaft to a depth of 120 ft (36.6 m) below grade. It then runs horizontally 100 ft (30.5 m) below the surface of a seawater strait that separates the two facilities. The discharge pressure of the pumping system that transfers the water is 350 psig (2415 kPa gage) at ambient temperature. The material is ASTM A106, Grade B, and the internal corrosion allowance is 0.065 in (1.7 mm). The line is coated to prevent external corrosion. At times, the line is shut down and drained for maintenance. Under these conditions it must withstand the external pressure exerted by the seawater, without collapse.

Determine the required wall thickness to safely contain the water at the internal design pressure, and verify that this thickness is adequate to withstand the external pressure. The design code is ASME B31.1.

Solution. The pipeline design will be developed initially for the internal pressure condition. It will then be checked for the external pressure.

The internal design pressure has two components: the pump discharge pressure and the static head due to the vertical run to 120 ft below grade. The head pressure is

$$\begin{aligned} P_h &= 62.4\,\frac{\text{lb}}{\text{ft}^3} \times \frac{1\ \text{ft}^2}{144\ \text{in}^2} \times 120\ \text{ft}\ (36.6\ \text{m}) \\ &= 52\ \text{psig}\ (360\ \text{kPa gage}) \end{aligned}$$

The internal design pressure is therefore 350 + 52 = 402 psig. The minimum wall thickness based upon this pressure is determined by using Eq. (B2.5):

$$t_m = \frac{PD_o}{2(SE + Py)} + A \tag{B2.5}$$

The values of the variables are

P = design pressure = 402 psig (2780 kPa gage)

D_o = outside diameter of pipe = 24 in (610 mm)

S = allowable stress = 15,000 psi (103.4 MPa)
(ASME B31.1, Table A.1 at −20°F (−29°C) to 650°F (343°C)

E = 1.0 seamless pipe

y = 0.4 ASME B31.1, Table 104.1.2(A)

A = 0.065 in (1.7 mm)

Substituting these values yields

$$\begin{aligned} t_m &= \frac{402 \times 24}{2(15{,}000 \times 1 + 402 \times 0.4)} + 0.065 \\ &= 0.318\ \text{in} + 0.065\ \text{in} \\ &= 0.383\ \text{in} \end{aligned}$$

The commercial wall thickness tolerance on ASTM A106 pipe is +0, −12½ percent; therefore the nominal wall thickness is determined by dividing the minimum wall thickness by 0.875.

$$t_{\text{nom}} = \frac{0.383}{0.875} = 0.438\ \text{in}\ (11.1\ \text{mm})$$

The next-larger standard pipe wall thickness for ASTM A106, per ASME B36.10M, is 0.500 in (12.7 mm). This nominal thickness is accepted preliminarily, and will be investigated for its adequacy to withstand the external pressure condition.

ASME B31.1, Paragraph 104.1.3, invokes the ASME Boiler and Pressure Code, Section VIII, Division 1, *Pressure Vessels*, Subsections UG-28 through UG-30,[27] for the external pressure design of straight pipe. This subsection provides a series of empirical procedures for the external pressure design of shells and tubes. They may be stiffened or unstiffened. The procedures rely on equations presented in UG-28 to UG-30, and a series of external pressure design charts given in ASME Section II, Part D, Subpart 3.[28]

This example problem is a basic case involving an unstiffened straight tube under external pressure, and simplifying assumptions have been made. The reader is encouraged to study Subsections UG-28 through UG-30 in their entirety, prior to attempting the solution of this class of design problems.

The nomenclature is

A = a geometric factor determined from ASME Section II, Part D, Subpart 3, Fig. G. It is used to enter the applicable material chart given in subpart 3. A simplified version of Fig. G, which is applicable to this sample problem, is shown in Fig. B2.4.

B = a factor determined from applicable material chart of ASME Section II, Part D, for maximum design metal temperature, psi. A simplified version of the material chart which is applicable to this sample problem is shown in Fig. B2.5.

D_o = outside diameter of cylindrical shell course or tube, in

L = total length of tube between tube sheets, or design length of vessel between lines of support*

P = external design pressure, psi

P_a = calculated value of maximum allowable external working pressure, for assumed value of t, psi

t = minimum required thickness of a cylindrical shell or tube

t_s = nominal thickness of a cylindrical shell or tube, in

Detailed Procedure. The following procedure applies to cylindrical shells or tubes whose diameter-to-thickness ratio D_o/t is greater than 10.

- For this example the minimum required wall thickness t is taken as the commercial minimum wall thickness, less the corrosion allowance.

$$\begin{aligned} t &= t_s \times (\text{tolerance factor}) - A \\ &= 0.500 \times 0.875 - 0.065 \\ &= 0.3725 \text{ in } (9.5 \text{ mm}) \end{aligned}$$

* See ASME Section VIII, UG-28(b).

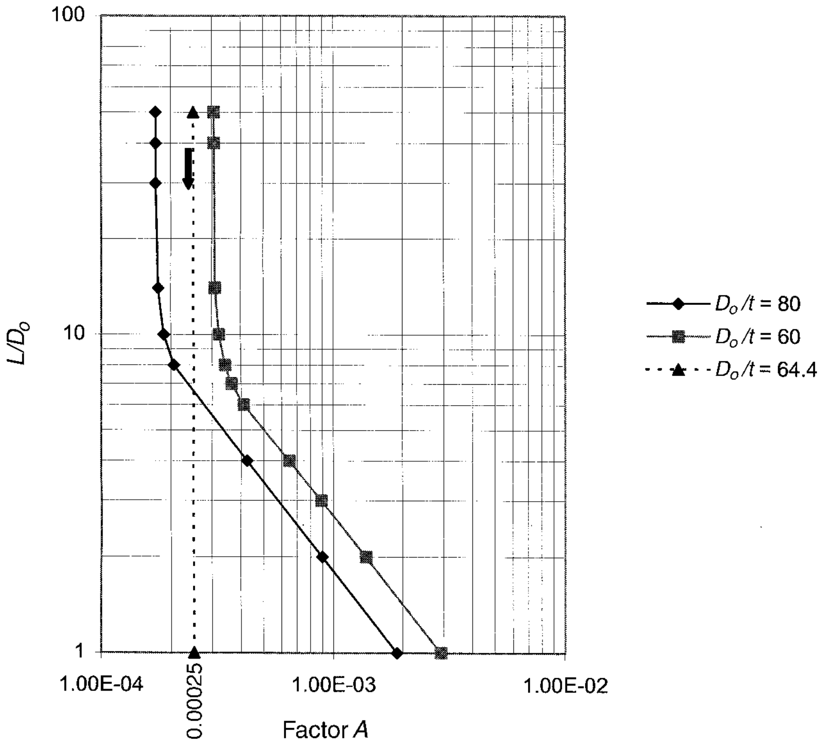

FIGURE B2.4 External pressure design factor A versus L/D_o. (*Adapted from ASME Section VIII, Division I,* Rules for the Construction of Pressure Vessels, *American Society of Mechanical Engineers, New York, 1998.*)

- For the assumed thickness t, determine the ratios L/D_o and D_o/t. If L/D_o is greater than 50, assume $L/D_o = 50$.

$$L/D_o = 50 \qquad L = \infty;\ \text{no tube supports}$$

$$D_o/t = 24.0/0.3725 = 64.4$$

- Using the values for L/D_o and D_o/t, proceed to Fig. B2.4 and determine the value of A. From Fig. B2.4, at $L/D_o = 50$ and $D_o/t = 64.4$, A equals 0.00025.
- Using the value of A found above, proceed to the material chart shown in Fig. B2.5 to determine the value of B. From Fig. B2.5, at the value of $A = 0.00025$; B equals 3600.
- Use the following formula to calculate the maximum allowable external working pressure which may act on the pipe:

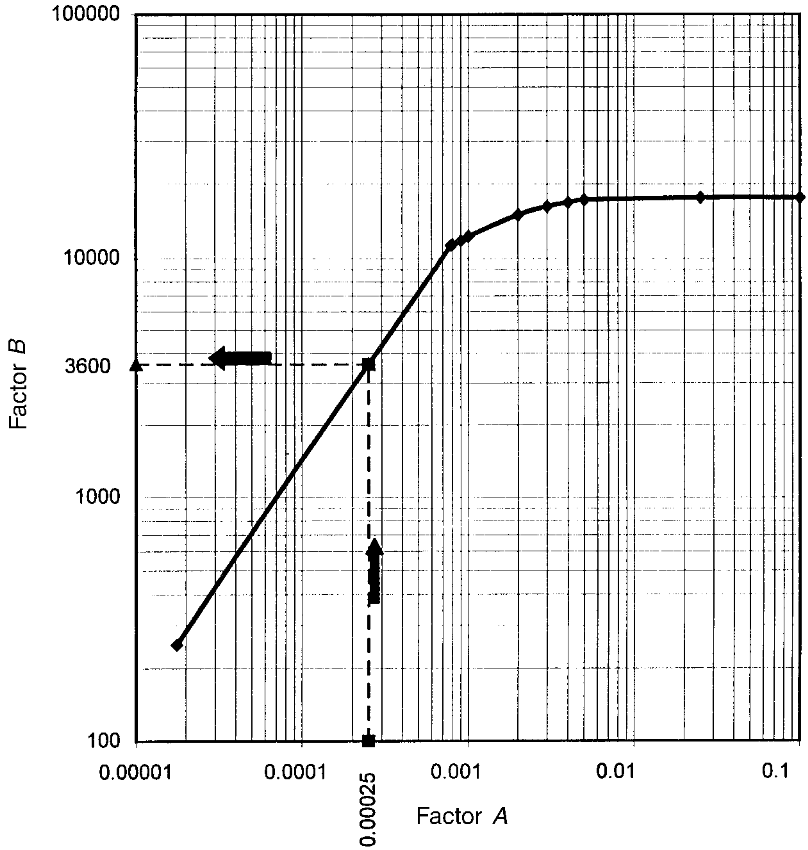

FIGURE B2.5 External pressure design factor A versus factor B. (*Adapted from ASME Section VIII, Division I,* Rules for the Construction of Pressure Vessels, *American Society of Mechanical Engineers, New York, 1998.*)

$$P_a = \frac{4B}{3(D_o/t)}$$

$$= \frac{4(3600)}{3(64.4)}$$

$$= 74.5 \text{ psig } (514 \text{ kPa gage})$$

This maximum allowable external working pressure must be compared with the actual external pressure due to the submergence in seawater, to determine whether the design is adequate.

The submergence depth is given as 100 ft, and the density of seawater is taken as 64.0 lb/ft^3. The seawater pressure P_{sw} acting on the outside of the pipe is

$$P_{sw} = 64 \frac{\text{lb}}{\text{ft}^3} \times \frac{1\ \text{ft}^2}{144\ \text{in}^2} \times 100\ \text{ft}$$
$$= 44.4\ \text{psig (306 kPa gage)}$$

Since the maximum allowable external working pressure for the pipe P_a exceeds the seawater pressure acting on the pipe P_{sw}, the design is acceptable.

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27. ASME Boiler and Pressure Vessel Code, Section VIII, Division I, *Rules for Construction of Pressure Vessels,* American Society of Mechanical Engineers, New York, 1998, Subsections UG-28 through UG-30.

28. ASME Boiler and Pressure Vessel Code, Section II, *Materials,* Part D—*Properties,* American Society of Mechanical Engineers, New York, 1998, Subpart 3.