Piping Systems & Pipeline

ASME B31 CODE SIMPLIFIED

- ✔ Various repair techniques, their advantages and shortcomings
- ✔ Step-by-step logic to be followed in making repair decisions and selecting the applicable repair
- ✔ Design, construction, inspection and testing rules, and their application to integrity assessment of operating systems

Piping Systems & Pipeline

ASME Code Simplified

Email: info@newcastleeng.com

Contents

Preface

Email: info@newcastleeng.com

The genesis of this book came from work done on another book about codes, *Pressure Vessels: The ASME Code Simplified*. It was fueled by several years working on the ASME B31 Pipe Codes as a volunteer for both the B31.3 Section Committee and the Mechanical Design Technical Committee. And it was honed by the many years spent in the designing and fabricating of products under the guidance and regulation of one or more of the various codes, both national and international, that are used to help make modern pressure technology systems safer.

This book also results from years of reflections on the questions I have encountered in my career—both my own questions and questions I have been asked, either as a company engineer or as a committee member. These questions come in three generic types, which can be classified as follows:

1. Does this passage in the Code mean …?
2. Why is …?
3. Which Code/Standard applies in …?

The actual code committees sometimes have a hard time answering the latter two questions, because answering them in detail would constitute or approach a consulting requirement. The first question can often be interpreted for a specific code or edition of that code. Since the Code is constantly being improved these questions cannot be answered generically.

The piping codes tend to spawn the third type of question because piping serves so many varied fluids, temperatures, pressures, materials, and risk factors. Code B31 ameliorates that problem by publishing section books dedicated to a particular category of those services.

The fact remains, as has often been said, "Once the pipe is in service, how does it know which code it was designed and built for?" It doesn't. But there are different solutions to those common problems and thus the various codes develop requirements that suit those solutions.

This development leads to the second question. Why does this section do it this way? This book's intention is to answer in some comprehensible way both the second and third questions. In the process of explaining those differences we have posited two general categories, those of buried piping and aboveground piping. Subdifferences within those categories are recognized.

The logic of the reasons for those differences as this author understands them is discussed. That logic comes from long experience and many hours of conversation with my colleagues. The final choice and explanation of the logic are mine and therefore any errors found in the logic are mine.

Where it is deemed helpful, the areas of the B31 Codes that address a specific issue are listed so that the reader may draw his or her own conclusions. The Codes stress that they are not handbooks and do not do away with sound engineering judgment. I would add that the information presented in this book is advisory only, gained from the author's long experience in the field. There is no obligation on the part of anyone to adhere to any recommendations made.

It must be remembered that there is no alternative to reading and understanding the ASME B31 Code. This book is not meant to replace the reading of the Code, but to clarify it.

Chapter

1

Piping History

Ancient Piping

Pipe or reasonable facsimiles to modern pipe began to appear as people started to live in towns. That move created the need to transfer water from the source, usually a stream or spring, to some central place in the town. Archaeologists have found earthenware pipes with flanged ends dating to 2700 B.C. These flanges were joined with asphalt rather than bolted, as are modern flanges.

Evidence of the use of metal pipe goes as far back as the 2400s B.C. This pipe was made from copper in Egypt. Many other archaeological finds confirm the use of pipe to convey water from sources as distant as several miles to the places of use.

Some of the most famous and longer delivery systems were built by the Romans. Their systems of aqueducts are well known. Many can still be seen today. The famous Pont du Gard in southern France is one of the more well-known pieces of evidence.

Less well known about Roman waterworks is the system of water delivery in the city itself. It is estimated that the Romans had as much as 250 mi of piping delivering water to private sources. By A.D. 97 they even appointed a water superintendent, Julius Frontinus.

The system included valves and stopcocks to control the flow of water. Among Julius Frontinus's accomplishments was the standardization of dimensions and materials. These materials were mainly made of lead and copper including alloys, or their near equivalent, that are still found as American Society of Testing Materials (ASTM) B67 today.

Pipe sizes were standardized and named as shown in Table 1.1 for the popular sizes up to approximately 4 in. in diameter.

TABLE 1.1 Selected Roman Pipe Sizes

Pipe name (Latin)	Pipe diameter, mm	Pipe diameter, in.
Quinaria	23	0.9
Senaria	28	1.1
Octonaria	37	1.4
Denaria	46	1.8
Duodenaria	55	2.1
Vicenaria	92	3.6

Most pipe was made from lead. It was made in a sheet and then rolled and welded, by melting the lapped lead. This resulted in more of a raindrop shape than a round one. The valves and other paraphernalia were usually made from bronze, as mentioned above, of a specific composition.

The metallurgy and dimensions were consistent with the standards that Frontinus had listed in his book. This could be considered the forerunner of the modern American Society of Mechanical Engineers (ASME) standards for piping.

There was little progress and possibly some slippage through the Middle Ages. Another interesting development in piping came in the early 1800s. At that time London began using gas lighting for street lamps. Those pipes were made by welding musket barrels end to end.

By this time steam engines began to be developed. Early steam engines were low-temperature and low-pressure devices; even so, they put more stringent requirements on both the boilers and the piping.

U.S. and ASME Code Development

Early steam engines operated at very low pressures compared to today's boilers, at about 10 psi (69 kPa). At that pressure the temperature would be right at 212°F (100°C). Even at these pressures and temperatures there were accidents. Those engines were not much more than teakettles with direct flame impingement as the source of power. They were not high-pressure devices. As late as the 1900s, the average operating pressures were under 100 psi. There was also a gradual development of unfired pressure vessels and piping.

By the late 1800s boiler explosions were becoming commonplace. One of the most memorable occurred on April 27, 1865. A steamship, the *Sultana*, was carrying a load of 2021 Union prisoners home from their prison camp in Vicksburg, Mississippi, up the river toward release and home. The ship was 7 mi north when the boiler exploded; 1547 of the passengers were killed. More people were killed in that explosion than in the San Francisco earthquake and fire of 1906.

This was one of many similar explosions. There were several more with increasing death and property destruction. From 1898 to 1905 in the United States the recorded number of boiler explosions was 3612. This averaged more than one explosion every day. The number of lives lost in those explosions was 7600. This is almost an average of $2\frac{1}{2}$ deaths per day.

Several of these explosions occurred in Massachusetts. On March 20, 1905, an explosion in Brockton, Massachusetts, killed 58 persons and injured 117 others. The legislature saw the need for action. In 1907 the state acted. It formed the Board of Boiler Rules. This was the first legislation on boiler design that was effective. In short order other states began to follow with their boiler rules:

1911 New York and Ohio

1913 New Jersey

1915 Indiana

1916 Pennsylvania

1917 California, Michigan, and Arkansas

1918 Delaware and Oklahoma

1920 Oregon

By 1920 fully 22 percent, or 11 of the states, had rules. However, no two sets of rules were exactly alike. This created problems with both boiler users and boiler manufacturers. A boiler built to one set of rules might not be acceptable under another set.

The ASME was the leading engineering society. It was being urged to create a common set of specifications to cover the design, construction, and operation of pressure-containing equipment. The result was that on February 13, 1915, Section 1, Power Boilers, was submitted to the ASME for approval. Other sections of the *Boiler and Pressure Vessel Code* (BPVC) came in rapid succession:

1921 Section III, Locomotive Boilers

1922 Section V, Minature Boilers

1923 Section VI, Heating Boilers

1924 Section II, Materials, and Section VI, Inspection

1925 Section VIII, Unfired Pressure Vessels

1926 Section VII, Care and Use of Boilers

Since that time the ASME has continued to develop and expand those rules.

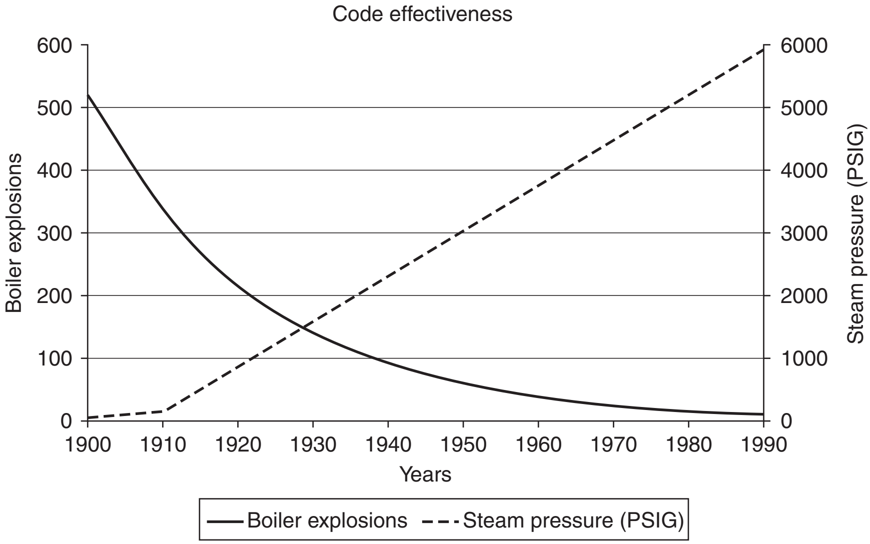

Figure 1.1 Effect of codes on pipeline safety.

The net result of the use of the boiler code and pressure vessel code development and implementation is shown in the graph of boiler explosions versus steam pressure per year. This is an exponentially smoothed portion, for boiler explosions, and a least-squares resolution of the peak steam pressures applied by the years shown. The graph shows that the effect of the Codes has been exemplary.

However, there had been no definite effort to develop rules for piping during those years. That history is explored in Chap. 2.

Chapter

2

Historical Development

The analogous piping standards came much later. The ASME requested that the American Standards Association (ASA) initiate a project for such a standard. The project B31 was initiated in 1926. The first issue was published in 1935 as the *American Tentative Standard Code for Pressure Piping*.

It became obvious that talents from several disciplines were necessary, as the volunteer committees who were attempting to write these codes struggled with the technical issues of piping systems. These codes were drawn from as many as 40 different engineering societies. Even though piping is similar visually, each use had its own technical requirements. This large diversity foreshadowed the resulting different book sections and committees.

During the period from 1942 to 1955 these codes evolved into B31.1, *American Standard Code for Pressure Piping*. It was published as B31.1. The ASA began to publish the various sections of the code as separate documents. As is stated in the introductions, these separate documents are published as different books for convenience. Each of the books is a part of the entire B31 piping code. The first of these separate books was ASA B31.8, *Transmission and Distribution Piping Systems*.

ASA B31.3 was published as a separate book in 1959. It superseded Section 3 of B31.1 of 1955. That code evolved into the *Petroleum Refinery Piping Code* in 1973. Subsequently, it further evolved into its current form as B31.3, *Process Piping*. As such, it encompasses petroleum refinery, chemical, cryogenic, and paper processing requirements. As the years progressed so did the names of the controlling committee. During the years 1967 to 1969 the ASA became the American National Standards Institute (ANSI). At all times the technical aspects were applied by ASME. Currently, the general titles are listed as an ASME code for xxx

An American National Standard. As discussed elsewhere, these codes are developed and monitored by ANSI in order to be a national standard.

Over the years other piping books have appeared. Currently they are

- B31.1, *Power Piping*
- B31.3, *Process Piping*
- B31.4, *Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids*
- B31.5, *Refrigeration Piping and Heat Transfer Components*
- B31.8, *Gas Transmission and Distribution Piping Systems*
- B31.8S, *Managing System Integrity of Gas Pipelines*
- B31.9, *Building Services Piping*
- B31.11, *Slurry Transportation Piping Systems*
- B31G-1991, *Manual for Determining the Remaining Strength of Corroded Pipelines*

The missing numerical code sections have existed but for various reasons are not current parts of the code. The piping code B31.7, *Nuclear Power Piping*, is no longer a B31 code. It has become an integral part of the *Boiler Code*, in Section III. Most of the elements of that code are included in and under the auspices of the nuclear code volunteers rather than the piping code members.

At one time a proposed B31.6, *Chemical Process Piping*, based on Case 49 of the B31 code, was in preparation. As it approached publication, the similarities to the *Petroleum Piping* code, now B31.3 *Process Piping*, were noted, and it was decided to include it in that book. As noted above, this book now contains several related types of piping systems.

A code variously called *Industrial Gas and Air Piping* or *Fuel Gas Piping*, B31.2, was published as a separate book for a time. That code was withdrawn as an American National Standard early in 1988.

This book points out in the various sections their intended purposes. Each code also states what it is *not* intended to do. The various committees try to keep those technical issues that are common to all pressure piping as similar as practicable. They also try to focus on the needs that make the particular area of concern for each of the recognized differences.

Basis of Each Code

In all cases it is a good idea for readers to check the actual code as well as the jurisdictions in which they are planning their installation before finally determining which code to use for the particular project. The code books clearly state it is the owner's responsibility to select the code that most nearly applies.

The codes list the considerations, generally as the limitations of the particular code section, the jurisdictional requirements, and the applicability of other codes. They also point out that in some instances different codes may apply to different parts of the installation. They list some specific codes and standards that may apply.

B31.1 power piping

This code is the original code and was a direct development out of the *Boiler* and subsequent codes. A boiler needs pipe, both internally and externally. The internal pipe would come under the rules of Section I and the external piping would come under B31.1. This piping is generally found in electric power generating stations. It is typically transporting steam or water under elevated temperatures and pressures. It may be used in other heating and steam uses.

Paragraph 101.3 enumerates what it does not apply to. It does *not* apply to

- Boilers, pressure vessel heaters, and components covered by the ASME *Boiler and Pressure Vessel Code* (BPVC)
- Building heating and distribution steam and condensate systems designed for 100 kPa (15 psig) or less
- Hot water heating systems designed for 200 kPa (30 psig) or less
- Piping for hydraulic or pneumatic tools
- Piping for marine and other installations under federal control
- Towers, building frames, and other similar structures

B31.3 process piping

This piping is typically found in petroleum refineries, chemical and pharmaceutical plants, and many other process plants and terminals. It has a high-pressure section. It recognizes different degrees of fluids safety concerns and imposes different rules on each. It has a nonmetallic section. It is generally considered the most broadly applicable code. Paragraph 300.1.3 lists the exclusions:

- Piping systems designed for pressures at or above 0 but less than 105 kPa (15 psig), provided they meet certain other requirements including temperature ranges
- Power boilers and piping required to meet B31.1

- Tubes and so forth that are internal to a heater enclosure
- Pressure vessels and certain other equipment and piping

B31.4 pipeline transportation systems for liquid hydrocarbons and other liquids

This code is for the type of pipelines that transport liquids between plants and terminals and pumping regulating and metering stations. One of the more well-known pipelines that is predominately under the auspices of this code is the Alaskan Pipeline from Prudhoe Bay in Alaska to Valdez.

Paragraph 400.1.2 lists the elements to which it does *not* apply:

- Auxiliary piping, e.g., water, air, or steam
- Pressure vessels, heat exchangers, and similar equipment
- Piping designed at or below 1 bar (15 psig) at any temperature
- Piping above 1 bar (15 psig) if temperature is below –30°C (–20°F) or above 120°C (250°F)
- Pipe, casing, or tubing used in oil well and related assemblies
- Petroleum refinery piping with certain exceptions
- Gas transmission and distribution lines
- Most proprietary equipment
- Ammonia refrigeration piping and carbon dioxide gathering and distribution systems

B31.5 refrigeration piping and heat transfer components

This is piping used for refrigerants and secondary coolants. It is to cover temperatures as low as –196°C (–320°F). There is a note explaining that the other codes may have requirements in their sections.

Paragraph 500.1.3 lists the elements to which the code does *not* apply:

- Any self-contained unit system that is subject to Underwriters Laboratories (UL) or a similar testing laboratory
- Water piping
- Piping designed for use not exceeding 105 kPa (15 psig)
- Pressure vessel and similar equipment but starting at the first joint of any piping for refrigerant piping that is connecting such equipment

B31.8 gas transmission and distribution piping systems

This code covers primarily gas transportation piping between sources and terminals. It includes gas metering, regulating, and gathering pipelines. It has rules about corrosion protection and with its supplement B31.8S covers the management of the integrity of such pipelines.

Paragraph 802.1.3 lists the elements to which it does *not* apply:

- Pressure vessels covered by the BPVC
- Piping with metal temperatures above 232°C (450°F) or below –29°C (–20°F)
- Piping beyond the outlet of the customer's meter assembly
- Piping in oil refineries with exceptions
- Vent piping for waste gases
- Wellhead assemblies
- Design and manufacture of proprietary equipment
- Design and manufacture of heat exchangers to Tubular Exchanger Manufacturers Association (TEMA) standards
- Liquid petroleum transportation systems, liquid slurry transportation piping, carbon dioxide transportation systems, and liquefied natural gas piping systems; it includes references to other documents for these types of systems

B31.9 building services piping

This code covers requirements for piping typically found in industrial, institutional, commercial, and public buildings. It is also found in many apartment residences. These piping systems do not typically require the sizes, pressures, and temperatures covered in B31.1 *Power Piping*.

This code in Paragraph 900.1.2 lists the types of building services that it is intended to cover including the material and size limits of that coverage. In a short Paragraph 900.1.3 it states essentially that it does not apply to those elements covered by the BPV Code.

B31.11 slurry transportation piping systems

The primary use of this code is for aqueous slurries between plants and terminals. It also covers use within those areas. One of the uses of these systems is in the mining industries in moving ores from the mines to elsewhere.

Paragraph 1100.1.2 states the elements to which it does *not* apply:

- Auxiliary piping such as for water, air, and similar liquids and gases
- Pressure vessels
- Piping designed for pressures below 103 kPa (15 psig) at any temperature
- Piping designed for pressures above 103 kPa (15 psig), when temperature is below –30°C (–20°F) or above 120°C (250°F)
- Piping within the battery limits of slurry processing plants and other nonstorage facilities
- Design and fabrication of proprietary items

A careful reading about each of the codes would find many similarities, especially for the elements to which the piping codes do not apply. They do not design for under 105 kPa (15 psig). They are for piping, not pressure vessels and other elements of a project that are covered by the ASME BPV Code.

It is interesting to note that several of the codes limit the temperatures for which their piping systems are to be designed or used. The reasons for this will be covered in greater detail as this book leads us through the specific rules of a particular portion of the code.

Code Organization

The ASME has a multilayered organization to help it achieve its goal of being the premier organization promoting the art, science, and practice of mechanical engineering throughout the world. As such it provides 600 codes and standards with 3500 active individuals acting as volunteers and working in 90 countries.

Many councils report to the Board of Governors. The Council on Codes and Standards has direct oversight of codes such as Code B31, which is the subject of this book. This book discusses only those boards and standards in direct line with the B31 codes. There is a Board on Pressure Technology under which the B31 codes, the BPV codes, and the B16 standards operate.

Individual codes each have separate committees. The B31 code comprises all the piping codes. They are separated into sections such as B31.1 and B31.3, to be sure that the guiding principles of the different piping system requirements are met. Those principles are as follows:

- The documents and responses are technically correct.
- The process produces consensus documents.

- Due process is maintained.
- ASME marks and copyrights are protected.

One of the more important ingredients in this code-building process is the maintenance of consensus. One might ask, What is consensus? The simple definition might be substantial agreement by affected interest categories. This would include consideration of differing views and attempted resolution of those views. This does not mean unanimity.

As discussed elsewhere, the public has an opportunity to become involved. There is also the opportunity to be a guest at a particular section meeting, and ultimately there is the right of appeal to go through the various levels.

Along with the principle of openness and transparency, all the pressure technology codes recognize certain basic interest groups as those from which balance needs to be sought. They are

- Manufacturers
- Constructors
- Enforcing authorities and inspection agencies
- Specialists with expert knowledge
- Designers
- Users or owners of equipment

Each of the B31 section books is composed of volunteer experts in the particular type of piping of that section e.g., B31.3 who do the actual writing. These committees are assisted by ASME staff appointed for a particular type of piping for which the section is generally intended. The ASME organized these committees to be in accord with the requirements of the American National Standards Institute (ANSI) rules for developing an American National Standard.

To be an American National Standard, a general procedure has to be followed under the guidance of ANSI. There are specific variations to the procedure. They have in common the following traits: They must be agreed to by a balanced set of users, engineering groups, manufacturers, and public parties. All objections must be addressed. When the proposed standard is considered in its final form, it is advertised and subject to general public comment. Having passed through that gauntlet, it can bear the title American National Standard.

To accomplish the above, the B31 committee comprises a standards committee which is made up of representatives from each of the sections and is kept in the aforementioned balance. Each section book has a main committee which is divided, in some manner, into specialties such as general concerns, design, fabrication, materials, and any other narrower group, such as nonmetallic piping, as may be thought to be required to complete the section's responsibilities.

In addition to these section committees, three committees represent specific disciplines:

- B31 Fabrication and Examination Committee
- B31 Materials Technical Committee
- B31 Mechanical Design Technical Committee

These B31 committees work on the issues of the discipline in a generic way rather than on the specific concern of the section groups. The members generally come from the section committees.

In addition, there are two other groups that perform the review and oversight function. One is the B31 Conference group, made up primarily of members from the jurisdictional authorities. The other is the B31 National Interest Review group, consisting of representatives from related or interested organizations such as the Valve Manufacturers Association. Each of the section books lists the individual members at the time of publishing in all these committees.

Each year the B31 committee has what may be called *code week*. In this week all the various section committees meet and conduct their business at one location. The various section committees may, at their discretion and need, meet at other times during the year. For instance, the B31.3 committee meets twice a year—once during the code week and at one other time approximately 6 months later. The other committees meet more or less often depending on their needs.

All these meetings are open to the public. When these meetings are attended, guests are invited to enter into the discussions, offer insights, or raise issues. Guests may not vote on business issues.

With the growing use of the Internet, interested parties can find out the time and places of the meetings via the Internet. The website is http://www.ASME.org. It is quite extensive and useful. As the volunteers have become more proficient, much of the business can be conducted more expeditiously. In an effort to become more responsive, the ASME developed a very useful Internet tool for members to use to conduct much of their business. A growing amount of the work is becoming available to the public. Much more is available to the volunteers.

Each section book has an appendix that explains and outlines how to ask the committees for information regarding the code. The most frequent type of communication regards an interpretation of the code as it is written. Each meeting of the sections, should there be such a request, addresses those questions as a first order of business.

As soon as an appropriate reply is identified, it is sent to the requester by the secretary of the section. The ASME will not perform consultation services, and if it is the opinion of the group that the question posed is of such a nature, such will be the response. There is a great effort to pose the questions in such a manner that the answer can be a simple yes or no. Anyone posing such a question should study the appropriate sections' rules to expedite any answer. These interpretations are published so that all who may have a similar question can receive guidance. These answers are specific to a year and edition of the relevant code book.

One can also ask for a revision or an addition to the Code. The boiler code uses code cases extensively, and there is a growing use of such devices in the piping codes. A code case is similar to a code revision, but it becomes effective immediately when it is published. In general, it may cover a temporary need and not become a permanent part of the code. However, the intention is to either incorporate the case into the code or let it die when the temporary situation is no longer in existence.

It is imperative for the user of the codes or standards to understand that these are living documents. Technology changes and allows the same margin of safety to be achieved in new ways. Some methods may become obsolete or require revision. Given the reduction in accidents as the pressures and service became more severe, it appears the system is working.

Definition of piping system

Each of the codes refers to a piping system. While the specific definition of a piping system may vary from codebook to codebook, this simplified book treats a piping system according to the following definition:

A piping system is a set of components including pipe, pipe fittings, flanges, bolting, gaskets, relief devices, and the pressure-retaining parts included in any stress analysis. It also includes the hangers, supports, and other equipment necessary to prevent overstressing of the pressure-retaining parts. It does not include the structure and equipment and foundations, except as they may affect the stress analysis.

This definition comes mainly from the B31.7 1969 edition. It captures the gist of the many separate definitions each book might offer as well as the reason for the codes. That reason is to define the design and fabrication of a system that offers a reasonable expectation of being safe when operated as intended. As noted in Chap. 1, that basic philosophy has reduced the number of resulting accidents in this pressure technology world over the years.

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Chapter

3

Metrication

The world uses the metric system. It is more accurate to call it SI, for *Système International*, than the metric system, even though there is little difference for ordinary users. Metrication is one of those things that we in the United States can't make up our minds about. Congress changes the timetable regularly. It is well known that the United States does not officially use SI units. Less well known is that a great portion of U.S. industry has already metricated. The auto industry uses metric measurements in most of its components. With the exception of some commodities, the food and beverage industries use metric measurements.

The rest of the world uses the metric system. Increasingly, if one wants to do business in that rest of the world, metric units of measure are required in the markings at least. Fortunately, for those who still think in U.S. Customary System (USCS) units, often a conversion to USCS units is printed in parentheses next to the metric measure. Since England has converted to the metric system, the system we use is now called the U.S. Customary System.

The metrication of pressure technology standards and codes is moving at deliberate speed to accomplish its goals. This is desired by both the community and the international users of these well-respected codes and standards. The conversion at the level of codes and standards is not easy.

Mathematical conversion is relatively simple if one has a handbook. However, since the basic units of measure in the USCS and SI are considerably different, manufacturers and users have problems keeping the precision at a consistent level. For example, 1 in. is 25.4 times the size of 1 mm. The same fraction of a millimeter, say ½, is considerably smaller and more precise than ½ in.

The B16 committee encountered some questions from both manufacturers and users when they converted the B16.5 flange standard. As a result, they drafted a letter explaining their philosophy of conversion. It is reproduced here as an example of the type of thinking that is required when converting from one system of measurement to another. (See Fig. 3.1.)

The ASME codes are being converted to list the metric unit as the preferred unit and the inch units as the parenthetical listing. But those codes have many calculations that are related to the allowable stresses of the materials, and those massive tables have yet to be converted in the hard sense. However, the B31 tables are circulating in draft form and should be adopted soon. The major units of measure in metric and USCS forms are shown in Table 3.1.

One of the key advantages of SI units is the use of base 10 and the relationships of the various units. To move from, say, kilopascals to megapascals or to bar, one merely has to move a decimal point. Try that kind of conversion in U.S. Customary System units, say, from pounds per square foot to pounds per square inch. It requires greater effort than that to move the decimal point.

Another issue concerns which standard size to use, i.e., from which country. Piping comes in standard sizes. There are different standards in different systems. The question is, Which standard size is to be utilized? The U.S. piping dimensions (as noted in ASME B36.19 and B36.10) are far from intuitive when one works with the actual dimensions. The ODs are not the same as the nominal designation in inches. The schedules follow an irregular pattern. Some of these points will be discussed further elsewhere. See App. B, Pipe Charts.

In the fittings and flange standards, there is the additional consideration regarding pressure-temperature ratings for these complex-shaped components. Generally speaking, the ASME and other standards have developed a *class* designation to define these ratings. The class designation leads to specific pressure-temperature ratings for a specific shape and a specific material. These class ratings reflect that allowed pressure can and will be different as the temperatures change over a broad range.

After much consideration it appears that the international community is deciding to utilize the pipe sizing as depicted by the dimensions in the ASME B36 standards noted above. To avoid as much confusion as possible, there is general agreement to harmonize on two dimensionless designators for pipe size. Both designators would mean the same actual pipe dimensions and in that sense are completely interchangeable. They do have some recognition level in their respective regions and therefore will probably be maintained as designators for some time.

Figure 3.1 ASME B16.5 metric values—a philosophy.

Introduction

The 2003 Edition of ASME B16.5, Pipe Flanges and Flanged Fittings NPS ½ through NPS 24 Metric/Inch Standard, contains millimeter dimensions and pressure-temperature ratings expressed in bar, with US Customary units in either parenthetical or separate forms. The purpose of this paper is to offer an explanation about why some of the conversions were made the way they were.

The cognizant committee had two primary goals during the development of the metric values shown in the 2003 Edition:

- The dimensions in mm should reflect the needed precision as much as possible.
- Flanges manufactured using existing forging dyes and machinery settings based on the inch dimensions should be able to meet the requirements for the metric dimensions.

Conversion from the Original Fraction

ASME B16.5 dimensions before 1977 were expressed mostly in fractional inches instead of decimal inches. For example, 1/8 in. was shown as 0.12 in. or 0.125 in., depending on the intended precision of the dimension, starting in the 1977 edition.

Millimeter dimensions were converted from the original fractional inch dimensions rather than the decimal inch dimensions. For example, 1/16 in., when converted from the fraction, converts to 1.6 mm. The decimal "equivalent" (0.06 in.) converts to 1.5 mm. Some of the conversions shown in the tables will appear to be incorrect when converted from the decimal inch dimensions.

1/16 in. was sometimes converted to nearest 0.1 mm, sometimes converted to the nearest 0.5 mm, and at other times converted to the nearest 1 mm. The conversion depended on the needed precision of the measurement. So the millimeter equivalent for 0.06 in. is sometimes 1.6 mm, sometimes 1.5 mm and at other times 2 mm.

Toleranced Dimensions

Dimensions that have tolerances are those that are considered to be needed for adequate fit-up and those important for integrity of the pressurized flanged joint. These dimensions were converted such that the metric dimensions are essentially the same as the US Customary dimensions, and the tolerances were selected such that the permitted deviations from the tabulated dimensions were nearly identical to those permitted by the US Customary dimensions.

Bolt circle diameter converted to nearest 0.1 mm. The committee believes this level of precision is needed to minimize problems with fit-up to other flanges, even though the tolerance on the dimension is 1.5 mm. Converting with less precision was expected to cause additional problems with centering metal gaskets as well.

Length through hub converted to nearest 1 mm. This dimension needs to be consistent in order to maintain overall dimensions for fabricated spools. The committee believes that maintaining this dimension to the nearest whole millimeter provides the needed precision.

Un-toleranced Dimensions

Dimensions that have no tolerances are those that need not have precision for fit-up and don't contribute significantly to the integrity of the pressurized flanged joint.

(*Continued*)

Figure 3.1 ASME B16.5 metric values—a philosophy. (*Continued*)

Examples of those dimensions and the philosophy used to create the millimeter dimensions are:

1/16" raised face converted to 2 mm instead of 1.6 mm. Raised faces measuring something different than 2 mm meet the requirements of the standard. Conversion to the nearest mm reflects the intended precision of the dimension.

Outside diameter of flanges converted to nearest 5 mm. For example, NPS $^3/_4$ Class 600 Flange outside diameter. The 4-5/8" was converted to the nearest 5 mm (115) instead of the nearest whole mm (117) or tenth mm (117.5). Outside diameters measuring other than 115 mm meet the requirements of the standard. Conversion to the nearest 5 mm reflects the intended precision of the dimension.

Bolt hole diameters expressed in fractional inches. Inch dimension bolt holes were retained for flanges manufactured to metric dimensions. Inch bolts are recommended for use with these flanges. Extensive dimensional compatibility studies exploring the possibility of using metric as well as inch dimensioned bolting with ASME B16.5 flanges were conducted. The studies revealed that providing dimensions that allowed for the use of metric as well as inch dimensioned bolts, especially when combined with metal gaskets, was impossible. This conclusion was supported by experience with some flanges manufactured to ISO 7005-1, Metallic flanges – Part 1: Steel flanges.

In Summary

The cognizant subcommittee did not intentionally change any of the requirements for dimensions in the 2003 Edition of ASME B16.5. The dimensions in mm reflect the needed precision as much as possible.

Flanges manufactured using existing forging dyes and machinery settings based on the inch dimensions should be able to meet the requirements for the metric dimensions. While acceptable dimension ranges are not precisely the same for the two units of measure, there is a significant amount of overlap. Still it is possible for a flange to meet the requirements in one system of units and not in the other.

TABLE 3.1 Comparison of Basic Measurements in the Two Systems

Measure	SI units	U.S. Customary System units
Linear	m (meters), mm (millimeters)	ft (feet), in. (inches)
Pressure	kPa (kilopascals), bar	psi (pounds per square inch)
Stress	MPa (megapascals)	psi
Force	N (newtons)	lbf (pounds force)
Moment	N · m (newton-meters)	in · lbf (inch-pounds)
Energy	J (joules)	ft · lbf (foot-pounds)
Temperature	°C (degrees Celsius)	°F (degrees Fahrenheit)

Abbreviated metric conversion chart

To convert inches to millimeters:	Multiply inches by 25.4
To convert pounds per square inch to kilopascals:	Multiply pounds per square inch by 6.8947
To convert degrees Celsius to degrees Fahrenheit:	Multiply degrees Celsius by 1.8 and add 32°

The U.S. designator is NPS. It has a rather easy interpretation as *nominal pipe size*. The alternate or metric designator is DN, which also has an easy translation to *diameter nominale* or *nominal diameter*.

The issue of how to define pressure ratings has a less clear-cut answer, but it appears that there is a slight but growing preference for a class-type designation with its implied disconnect from a specific pressure. The U.S. method employs its designator as the word *class* followed by some identifying number (say, 150, 3000, 3M, etc.). The alternate or metric practice has been to designate *PN* (for *pressure nominale*).

Since the word *class* does not lead the unknowledgeable to think of a specific pressure, the slight preference occurs. One would expect that allowing the terms to be interchangeable would allow conversion to proceed. This is what will be utilized in this book.

Other units such as millimeters and degrees of temperature are not in real question. There seems to be some small controversy over the use of bar or pascals, but ultimately this is like an argument over whether to write numbers in scientific notation or in Arabic notation. The relationship between bar and pascals is only decimal places apart.

The familiar U.S. standards are being converted to include metric units as fast as the process will allow. This is important as competition from the International Organization for Standards (ISO) standards and other national standards grows.

For reference, Table 3.2 shows the correspondence of the various nomenclatures. It is clear in the larger sizes that the DN designation is the NPS designation times 25. This is not exactly true for those sizes below NPS 4.

TABLE 3.2 NPS versus DN

NPS	DN	NPS	DN
1/2	15	18	450
3/4	20	20	500
1	25	22	550
1 1/4	32	24	600
1 1/2	40	26	650
2	50	28	700
2 1/2	65	30	750
3	80	32	800
3 1/2	90	34	850
4	100	36	900
5	125	38	950
6	150	40	1000
8	200	42	1050
10	250	44	1100
12	300	46	1150
14	350	48	1200
16	400	50	1250

While this book is about the ASME B31 piping codes, it is important to recognize that there is a major effort to standardize internationally. The ISO is the most well-known organization in those attempts, and the United States is deeply involved in this effort.

At present there are a host of national and area standards competing for the user community. Since reasons exist within the piping user community for having different codebooks (because different disciplines

TABLE 3.3 Common Conversion Factors: USC to Metric (SI)

Plane angle deg to rad	1.745 E–02	**Bending, torque**	
Length		kgf · m to N · m	9.806 E+00
in. to m	2.54 E–02	lbf-in. to N · m	1.129 E–01
ft to m	3.048 E–01	lbf-ft to N · m	1.355 E+00
mi to m	1.609 E+02	**Pressure (1)**	
Area		psi to bar	6.894 E–02
in.^{2} to m^2	6.451 E–04	Pa to bar	1 E–05
ft^2 to m^2	9.29 E–02	kPa to bar	1 E–02
Volume		**Stress**	
ft^3 to m^3	2.831 E–02	psi to MPa	6.894 E–03
U.S. gal to m^3	3.785 E–03	kips/in. to MPa	6.894 E+00
in.^{3} to m^3	1.638 E–05	N/mm^2 to MPa	1 E+00
oz (fluid, U.S.) to m^3	2.957 E–05	**Energy, work**	
L to m^3	1 E–03	Btu (11) to J	1.055 E+03
Velocity		Calorie (11) to J	4.186 E+00
ft/min to m/s	5.08 E–03	lbf-ft to J	1.355 E+00
ft/s to m/s	3.048 E–01	**Power**	
km/h to m/s	2.777 E–01	hp (550 ft lbf/s) to W	7.456 E+02
mi/h to m/s	4.47 E–01	**Temperature (1)**	
mi/h to km/h	1.609 E+00	°C to K	tK = tC + 273.15
Mass		°F to K	tK = (tF + 459.67)/1.8
oz (avoir) to kg	2.834 E–02	°F to °C	tC = (tF – 32)/1.8
lb (avoir) to kg	4.535 E–01	**Temperature interval**	
slug to kg	1.459 E+01	°C to K	1 E+00
Acceleration		°F to K or °C	5.555 E–01
ft/s^2 to m/s^2	3.048 E–01		
std. grav. to m/s^2	9.806 E+00		
Force			
kgf to N	9.806 E+00		
lbf to N	4.448 E+00		
poundal to N	1.382 E–01		

GENERAL NOTES:

(a) For other commonly used conversion factors refer to ASTM E 380.

(b) The factors are written as a number greater than 1 and less than 10 with six or fewer decimal places. The number is followed by the letter E (for exponent), a plus or minus symbol, and two digits that indicate the power of 10 by which the number must be multiplied to obtain the correct value. For example, 1.745 E–02 is 1.745×10^{-2} or 0.01745, and 25 degrees = 25×0.01745 or 0.436 radian.

SOURCE: (1) Extracted partially from ASME 51-1.137 B31.8-2003.

are involved in the use of pipe to achieve the needed results), there is also an argument for regional differences based on, if nothing else, installation inertia and local jurisdictional requirements.

By similar reasoning, there is much greater incentive to have common standards for use worldwide. When one international user was asked why we needed to add another similar standard or code to the current multiple standards, the reply was simple: "I specify such items in at least 30 different companies to do the same thing. I do not need or want to have the same 30 different specifications." Put even more simply, there is economy in volume.

One important goal of this code simplification effort is to contain existing codes within an ISO code. For instance, ISO Code 15469 is essentially Code B31.3. That standard says if you want to build this type of system, use B31.3. It recognizes that since this is a U.S. code, one may want to use something differently and allows this to be done under certain conditions. It has added some cautionary rules about piping layout that the committee felt Code B31.3 might not have addressed adequately for international purposes.

Such utilization of existing codes and standards within the international code movement speaks well for simplification. It will behoove the standards committees for the harder, older arts to recognize the benefits of such common standards.

This discussion of metrication recognizes the fact that the ASME piping codes are somewhat hybrid for international use. Their technical adequacy has been demonstrated for the piping systems in which they are used. The issue of different measuring systems is being addressed. These codes are, at the least, useful in many international areas. In fact they are used to some degree in international piping efforts.

In the breakdown of the exclusions listed by codes in Chap. 2, one cannot help but notice that 15 psig is universally agreed to as a lower limit for code applicability to a specific system. That same universal agreement does not follow as to which is the proper kilopascal equivalent.

There were three noted kilopascal numbers associated with 15 psig. These range from 100 to 105. Mathematical conversion gives 103.42 as the precise conversion. This brings us back to the statement earlier that metric conversion is not easy, and that is mostly relevant to precision.

Table 3.3 provides a chart of often-used conversion factors between USC and metric (SI) units.

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Chapter

4

Materials

General Considerations

There are many factors to consider in choosing piping materials. Most are beyond the scope of the codes. They include such things as availability, type of service, and fluid. The designer cannot begin the design for sizes until these decisions are made. The designer cannot apply the design rules until the decisions are made. The codes set out materials that the committees believe are appropriate for the services for which the code is intended.

In observing the overall aspects of the various codes, it becomes apparent that there are two major classifications. Those classifications are basically defined by their location. The first is *aboveground piping*, which is usually within the boundaries of a property or building. The second is *buried piping*, which usually goes through public rights-of-way and/or across rights-of-way on private land.

While it is true that the buried, or pipeline, type of piping may have areas that are aboveground and/or facilities such as terminals or pumping stations, the major portions would be belowground. The opposite is true of aboveground piping; it may have buried or similar portions, but the major portions are aboveground.

This fundamental difference can be attributed to most of the specific differences in the focus of the codes. The following division of the codes is somewhat arbitrary, as noted above, but defines the way this book will group the codes. That basic breakdown is as follows:

Aboveground codes	*Buried codes*
B31.1	B31.4
B31.3	B31.8
B31.5	B31.11
B31.9	

Materials Aboveground

Aboveground pipes have two basic characteristics. Generally, there is a wide range of fluids that the piping may contain, and they have a rather wide range of temperatures and pressures at which they operate. These two characteristic result in a broader set of materials being recognized by that code.

All the piping codes recognize ASTM materials. This is in contrast to the BPVC, which limits the materials of choice to the ASME materials. One can tell the difference between the two by the product form classification designation.

For metallic materials the ASTM designations have a single letter plus a number. That letter is A for ferrous materials and B for nonferrous materials. The ASME designations follow a similar scheme but add an S, such as SA or SB. A typical example for seamless pipe is A-106 in the ASTM designation and SA-106 in the ASME. In general, one may substitute an SA or SB material for the same A or B designation.

Often this raises the question, What is the difference between the two materials? Often the answer is nothing. In the SA or SB specification standard, there is a statement at the top of the page that says what it is based upon. Those statements follow two general patterns. Pattern 1 says that this specification is the same as A(B)-XXX and names a year of issue. Pattern 2 says this specification is the same as A(B)-XXX and names a year of issue. Then it states that there are the following exceptions.

The ASME Section II (materials) committee reviews the ASTM specifications and determines whether to adopt them for the SA standards. The B31 committees rely on the Section II committee to provide the allowable stress values for the ASTM materials that they want to adopt. That Section II group also assigns various other group letters or numbers to those materials for the B31 groups. The Section II materials group maintains a standard database of all those materials for the whole of the ASME code and standard groups.

This is an efficient system regarding data storage and similar items of information. It, by its very nature, means that there will always be some synchronization problems. The ASTM publishes new sets of specifications each year. The ASME committees, as mentioned, meet with some frequency each year. However, the process of publish, meet, review, approve or disapprove, and publish for the ASME version is seldom sequentially synchronized. Unless a particular ASTM specification doesn't change over an appropriate period, it is seldom exactly compatible with the current SA or SB specification.

Even a new specification from ASTM needs some period of time to be adapted to a particular ASME code. Each of the Code B31 sections has

TABLE 4.1 Appendices of Referenced Standards

Code	Appendix
B31.1	F
B31.3	E
B31.4	A
B31.5	A
B31.8	A
B31.9	C
B31.11	I

a way to tell which year or edition of the standard has been approved by the committee. Those take the form of appendices of referenced standards. See Table 4.1.

Aboveground codes

A great range of materials are specifically accepted by the codes. As might be expected, the aboveground codes list the greatest variety of materials as well as devote many pages to the posting of allowable stresses for the temperature ranges that they cover. Code B31.9 actually defers to B31.1 for materials that are not specifically listed in Code B31.9.

Each of the codes recognizes that there may be a desirable material from the user's standpoint that may not be listed in the code. With certain restrictions, the codes provide some means of utilizing that unlisted material. Those restrictions vary considerably from code to code. They are detailed in the chapters of the code that are, for example, numbered X23. The X stands for the code number in the sectional sequence; i.e., for B31.3 the number would be Paragraph 323, for B31.9 it would be 923, and so on for the aboveground codes.

Note that Code B31.1 has a nonmandatory Appendix VI that explains the approval of new materials. It lists the basic requirements and the actions that the requestor must take to request that material to be added:

- Chemical composition
- Mechanical properties
- Tensile data per ASTM E-21
- When creep properties are expected to rule, those data points at specific intervals
- If it is to be welded, welding data in accordance with ASME Section IX including the welding process, weld metal test data at expected temperature, any restrictions, the appropriate per heat and postweld heat treatment required, and toughness data

- Any special application or handling required
- Applicable product form
- Either the ASTM specification or the application to ASTM

If there are circumstances requiring quick action, a code case would be considered.

Codes for Buried Materials

The buried piping codes have significantly fewer listed materials. The range of fluids covered by those codes is relatively narrow, and it is normally handled by the carbon steels. They most often use American Petroleum Institute (API) 5LX-XX piping.

5LX piping starts at a 42,000 *specified minimum yield strength* (SMYS) and increases in increments to 80,000 SMYS. A similar ASTM material, A-106 has a 40,000 SMYS for its best grade. Since the required thickness or weight of pipe for a given length and material when designed is inversely proportional to the yield strength, it follows that if 80,000 SMYS is an appropriate material, it would require much less material weight. The cost, therefore, would be much less, all other things being equal.

Pipelines have miles of pipe in a particular project. Therefore, pipe can be ordered to a specific thickness to achieve volume production and discounts. ASTM pipe is usually made to standard thicknesses, the schedules of pipe that are listed. It is that standard thickness pipe that is usually used in aboveground piping.

In contrast to the miles in a pipeline project, plant or building projects are usually measured in feet, albeit sometimes thousands of feet. So it is often not economical to order a special thickness because a volume manufacturing price is not available unless one uses one of the standard thicknesses.

This is not to say that special pipe thicknesses are not utilized in unusual sizes and in severe process conditions. However, often some material other than the plain carbon steel is used.

It is not necessarily true that precautions do not have to be taken when the higher-strength pipe is used. As the yield strength raises, the ductility will generally lessen. It is a philosophy in ASME design that one wants as much ductility as possible. The theory is that a ductile material will bend or bulge before it breaks. So the high-strength materials are not simple choices based on first cost only. A brittle fracture is something to avoid, and the codes do have requirements covering that issue.

These codes for buried pipe recognize that the limited materials specified may create a problem for a particular project. B31.8 has a detailed listing recognizing the categories of piping; it lists specific

TABLE 4.2 Material Categories and Paragraphs Defining Qualification Requirements

Categories of material recognized by B31.8	Paragraph in B31.8 defining qualifications
Items conforming to a B31.8 specification referenced in the code	811.2
Items of a type of specification referenced in the code but that specifically do not conform to a standard referenced in the code	811.22 referencing 811.221 or 811.222
Items for which standards are referenced in the code but do not conform to that standard and are relatively unimportant	811.23
Items of a type for which no standard or specification is referenced (e.g., gas compressor)	811.24
Proprietary items	811.24
Unidentified or used pipe	811.25 referencing 817

categories and has a paragraph outlining the qualifications procedure, as shown in Table 4.2.

Codes B31.4 and B31.11 are much less flexible. They simply state that materials that do not conform to one of the listed standards shall be qualified by petitioning the code committee for complete approval. They further stipulate that the approval shall be obtained before the material may be used.

Material toughness (Charpy notch toughness or similar)

As pointed out above, the ASME basic philosophy is to use ductile materials. Metallic materials have varying degrees of ductility at varying temperatures. The ambient ductility is measured, among other ways, by the mechanical properties of percentage of elongation and/or reduction of area in the ASTM product form specification. This measurement is not necessarily done for all temperatures.

At some temperature-thickness combination, all metals tend to become brittle. This is known as the *null ductility* point. This tendency to become brittle can be measured by the *Charpy V notch* (CVN) test. The goal is to have some residual "toughness" as measured by this test at temperatures representative of the design conditions.

Some material specifications referenced include specific testing requirements that cover the CVN requirements. For instance, A-350 in ASTM is similar to the A-105 product form in chemistry and mechanical properties. However, A-350 includes a series of these tests. A specific grade of material has a certain CVN requirement at a certain temperature. API 5LX-XX has similar tests as part of the basic requirements.

The codes recognize that sometimes more is required. Code B31.3 has potential problems inherent in many of the processes for which that code has been written. It has an extensive set of requirements and supplemental tests addressing this issue. There are tests in addition to those required by the metal specification. They are shown in Fig. 4.1, which shows Table 323.2.2 of Code B31.3.

The requirements are based on the minimum design metal temperature. This is a temperature quite often different from the design temperature and is discussed in Paragraph 301.3.1 of Code B31.3.

Code B31.3 defines the minimum temperature without impact testing for carbon steel materials. This is based on the nominal thickness of the component. The code provides a graph for the user to consult (see Fig. 4.2). It is relatively easy to read. For instance, from the notes ASTM material A-381 is for curve B if normalized and for curve A if not. Assuming a 1-in. nominal thickness and a minimum metal temperature of 20°C (68°F) or higher, an impact test would not be required. If the temperature were between 20°C and 6°C (42°F), one could avoid an impact test by specifying a normalized, or quenched and tempered, material.

There is an advanced procedure for reducing the temperature readings from the chart. It is based on stress ratios and is somewhat less than simplified. See Paragraph 327.2 in the Code.

Once it is determined that an impact test is required, Code B31.3 sets out the testing requirements in Table 323.3.1, duplicated here in Fig. 4.3. In addition, B31.3 sets the acceptance requirements shown in Fig. 4.4.

Code B31.9, *Refrigeration*, takes a similar approach to the impact or Charpy testing. However, it is somewhat simpler within the code. Code B31.9 defers to BPVC Section VIII UG-84. That set of requirements is quite similar to the requirements of B31.3. Code B31.9, Paragraph 523.2.2, sets the reference to UG-84 and sets out a substitution for subparagraph UG-84(b)(2). That substitution is quite lengthy, and the reader is referred to Code B31.9 for the details.

The code has a simplified exemption table and temperature reduction chart and procedure. There is, however, no significant difference from the discussion in B31.3 above; one who is familiar with that can, given UG-84 and Code B31.9, address the impact and toughness issues for B31.9.

Interestingly, neither Code B31.1 or Code B31.9 makes any specific requirements for the toughness issue. Of course, the requirements of the chosen material specification would include any requirements that might be required. From the standpoint of those codes and their committees, those requirements would be sufficient.

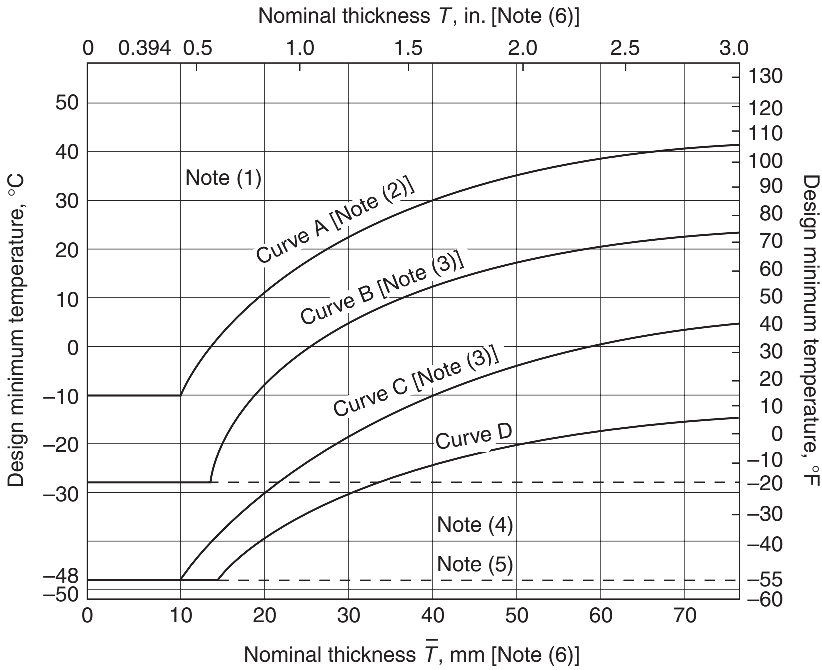
When one examines the stress tables in B31.1 and those in B31.3, one can get a flavor of the differences in material requirements between

TABLE 323.2.2
REQUIREMENTS FOR LOW TEMPERATURE TOUGHNESS TESTS FOR METALS
These Toughness Test Requirements Are in Addition to Tests Required by the Material Specification

	Type of Material	Column A Design Minimum Temperature at or Above Min. Temp. in Table A-1 or Fig. 323.2.2A		Column B Design Minimum Temperature Below Min. Temp. in Table A-1 or Fig. 323.2.2A
Listed materials	1 Gray cast iron	A-1 No additional requirements		B-1 No additional requirements
	2 Malleable and ductile cast iron; carbon steel per Note (1)	A-2 No additional requirements		B-2 Materials designated in Box 2 shall not be used.
		(a) Base Metal	(b) Weld Metal and Heat Affected Zone (HAZ) [Note (2)]	
	3 Other carbon steels; low and intermediate alloy steels; high alloy ferritic steels; duplex stainless steels	A-3 (a) No additional requirements	A-3 (b) Weld metal deposits shall be impact tested per para. 323.3 if design min. temp. < −29°C (−20°F), except as provided in Notes (3) and (5), and except as follows: for materials listed for Curves C and D of Fig. 323.2.2A, where corresponding welding consumables are qualified by impact testing at the design minimum temperature or lower in accordance with the applicable AWS specification, additional testing is not required.	B-3 Except as provided in Notes (3) and (5), heat treat base metal per applicable ASTM specification listed in para. 323.3.2; then impact test base metal, weld deposits, and HAZ per para. 323.3 [see Note (2)]. When materials are used at design min. temp. below the assigned curve as permitted by Notes (2) and (3) of Fig. 323.2.2A, weld deposits and HAZ shall be impact tested [see Note (2)].
	4 Austenitic stainless steels	A-4 (a) If: (1) carbon content by analysis > 0.1%; or (2) material is not in solution heat treated condition; then, impact test per para. 323.3 for design min. temp. < −29°C (−20°F) except as provided in Notes (3) and (6)	A-4 (b) Weld metal deposits shall be impact tested per para. 323.3 if design min. temp. < −29°C (−20°F) except as provided in para. 323.2.2 and in Notes (3) and (6)	B-4 Base metal and weld metal deposits shall be impact tested per para. 323.3. See Notes (2), (3), and (6).
	5 Austenitic ductile iron, ASTM A 571	A-5 (a) No additional requirements	A-5 (b) Welding is not permitted	B-5 Base metal shall be impact tested per para. 323.3. Do not use < −196°C (−320°F). Welding is not permitted.
Materials	6 Aluminum, copper, nickel, and their alloys; unalloyed titanium	A-6 (a) No additional requirements	A-6 (b) No additional requirements unless filler metal composition is outside the range for base metal composition; then test per column B-6	B-6 Designer shall be assured by suitable tests [see Note (4)] that base metal, weld deposits, and HAZ are suitable at the design min. temp.
Unlisted	7 An unlisted material shall conform to a published specification. Where composition, heat treatment, and product form are comparable to those of a listed material, requirements for the corresponding listed material shall be met. Other unlisted materials shall be qualified as required in the applicable section of column B.			

NOTES:
(1) Carbon steels conforming to the following are subject to the limitations in Box B-2; plates per ASTM A 36, A 283, and A 570; pipe per ASTM A 134 when made from these plates; and pipe per ASTM A 53 Type F and API 5L Gr. A25 buttweld.
(2) Impact tests that meet the requirements of Table 323.3.1, which are performed as part of the weld procedure qualification, will satisfy all requirements of para. 323.2.2, and need not be repeated for production welds.
(3) Impact testing is not required if the design minimum temperature is below −29°C (−20°F) but at or above −104°C (−155°F) and the Stress Ratio defined in Fig. 323.2.2B does not exceed 0.3 times *S*.
(4) Tests may include tensile elongation, sharp-notch tensile strength (to be compared with unnotched tensile strength), and/or other tests, conducted at or below design minimum temperature. See also para. 323.3.4.
(5) Impact tests are not required when the maximum obtainable Charpy specimen has a width along the notch of less than 2.5 mm (0.098 in.). Under these conditions, the design minimum temperature shall not be less than the lower of −48°C (−55°F) or the minimum temperature for the material in Table A-1.
(6) Impact tests are not required when the maximum obtainable Charpy specimen has a width along the notch of less than 2.5 mm (0.098 in.).

Figure 4.1 Code Table 323.2.2.



NOTES:

(1) Any carbon steel material may be used to a minimum temperature of –29°C (–20°F) for category D fluid service.

(2) X grades of API 5L, and ASTM A 381 materials, may be used in accordance with curve B if normalized or quenched and tempered.

(3) The following materials may be used in accordance with curve D if normalized:
(a) ASTM A 516 plate, all grades;
(b) ASTM A 671 pipe, grades CE55, CE60, and all grades made with A 516 plate;
(c) ASTM A 672 pipe, grades E55, E60, and all grades made with A 516 plate.

(4) A welding procedure for the manufacture of pipe or components shall include impact testing of welds and HAZ for any design minimum temperature below –29°C (–20°F), except as provided in Table 323.2.2, A-3(b).

(5) Impact testing in accordance with para. 323.3 is required for any design minimum temperature below –48°C (–55°F), except as permitted by note (3) in Table 323.2.2.

(6) For blind flanges and blanks, $\overline{T}$ shall be 1/4 of the flange thickness.

Figure 4.2 Code Fig. 323.2.2A showing minimum temperatures without impact testing for carbon steel materials.

those two otherwise similar codes. If one chooses the category of material talked about above in the discussion of the difference between materials A-105 and A-350, those materials are carbon steel forgings. In B31.1 there are two listings, A-105 and A-181. In B31.3 the exact category is carbon steel forgings and fittings, and there are eight grades listed: two grades of A-350, two grades of A-181, and two grades of A-234.

TABLE 323.3.1
IMPACT TESTING REQUIREMENTS FOR METALS

	Test Characteristics	Column A Materials Tested by the Manufacturer [See Note (1)] or Those in Table 323.2.2 Requiring Impact Tests Only on Welds	Column B Materials Not Tested by the Manufacturer or Those Tested But Heat Treated During or After Fabrication
Tests Materials	Number of tests	A-1 The greater of the number required by: (a) the material specification; or (b) the applicable specification listed in para. 323.3.2. See Note (2).	B-1 The number required by the applicable specification listed in para. 323.3.2. See Note (2).
	Location and orientation of specimens	A-2 As required by the applicable specification listed in para. 323.3.2.	
	Tests by	A-3 The manufacturer	B-3 The fabricator or erector
Tests on Welds in Fabrication or Assembly	Test piece for preparation of impact specimens	A-4 One required for each welding procedure, for each type of filler metal (i.e., AWS E-XXXX classification), and for each flux to be used. Test pieces shall be subjected to essentially the same heat treatment (including time at temperature or temperatures and cooling rate) as the erected piping will have received.	
	Number of test pieces [see Note (3)]	A-5 (a) One piece, thickness T, for each range of material thickness from $T/2$ to $T + 6.4$ mm ($\frac{1}{4}$ in.). (b) Unless required by the engineering design, pieces need not be made from each lot, nor from material for each job, provided that welds have been tested as required by Section 4 above, for the same type and grade of material (or for the same P-Number and Group Number in BPV Code, Section IX), and of the same thickness range, and that records of the tests are made available.	B-5 (a) One piece from each lot of material in each specification and grade including heat treatment [see Note (4)] unless; (b) materials are qualified by the fabricator or erector as specified in Sections B-1 and B-2 above, in which case the requirements of Section A-5 apply.
	Location and orientation of specimens	6 (a) Weld metal: across the weld, with notch in the weld metal; notch axis shall be normal to material surface, with one face of specimen ≤ 1.5 mm ($\frac{1}{16}$ in.) from the material surface. (b) Heat affected zone (HAZ): across the weld and long enough to locate notch in the HAZ after etching; notch axis shall be approximately normal to material surface and shall include as much as possible of the HAZ in the fracture.	
	Tests by	7 The fabricator or erector	

NOTES:

(1) A certified report of impact tests performed (after being appropriately heat treated as required by Table 323.2.2, item B-3) by the manufacturer shall be obtained as evidence that the material (including any welds used in its manufacture) meets the requirements of this Code, and that:
(a) the tests were conducted on specimens representative of the material delivered to and used by the fabricator or erector; or,
(b) the tests were conducted on specimens removed from test pieces of the material which received heat treatment separately in the same manner as the material (including heat treatment by the manufacturer) so as to be representative of the finished piping:

(2) If welding is used in manufacture, fabrication, or erection, tests of the HAZ will suffice for the tests of the base material.

(3) The test piece shall be large enough to permit preparing three specimens from the weld metal and three from the HAZ (if required) per para. 323.3. If this is not possible, preparation of additional test pieces is required.

(4) For purposes of this requirement, "lot" means the quantity of material described under the "Number of tests" provision of the specification applicable to the product term (i.e., plate, pipe, etc.) listed in para. 323.3.2.

Figure 4.3 Code Table 323.3.1.

Note that in B31.1 those grades are the only two listed as wrought fittings. To continue B31.3 finishes with A-105 and A-420.

Even by including or not including a material, Code B31.1 shows some lack of concern for impact testing. One more subtle difference lies in the emphasis of different codes.

Now the pipeline (buried pipe) codes do express concern about the toughness of materials that vary about as much as the aboveground piping. One of the concerns for pipelines is the propagation of the fracture, be it brittle or ductile. One of the phenomena of a pipeline fracture is

TABLE 323.3.5
MINIMUM REQUIRED CHARPY V-NOTCH IMPACT VALUES

Specified Minimum Tensile Strength	No. of Specimens [Note (2)]	Energy [Note (1)]			
		Fully Deoxidized Steels		Other Than Fully Deoxidized Steels	
		Joules	ft-lbf	Joules	ft-lbf
(a) Carbon and Low Alloy Steels					
448 MPa (65 ksi) and less	Average for 3 specimens	18	13	14	10
	Minimum for 1 specimen	16	10	10	7
Over 448 to 517 MPa (75 ksi)	Average for 3 specimens	20	15	18	13
	Minimum for 1 specimen	16	12	14	10
Over 517 but not incl. 656 MPa (95 ksi)	Average for 3 specimens	27	20	...	...
	Minimum for 1 specimen	20	15	...	...
		Lateral Expansion			
656 MPa and over [Note (3)]	Minimum for 3 specimens	0.38 mm (0.015 in.)			
(b) Steels in P-Nos. 6, 7, and 8	Minimum for 3 specimens	0.38 mm (0.015 in.)			

NOTES:
(1) Energy values in this Table are for standard size specimens. For subsize specimens, these values shall be multiplied by the ratio of the actual specimen width to that of a full-size specimen, 10 mm (0.394 in.).
(2) See para. 323.3.5(d) for permissible retests.
(3) For bolting of this strength level in nominal sizes M 52 (2 in.) and under, the impact requirements of ASTM A 320 may be applied. For bolting over M 52, requirements of this Table shall apply.

Figure 4.4 Code Table 323.3.5.

that it might propagate for a significant portion of those miles of pipeline, given no form of crack arrests.

Code B31.11 on slurry transportation makes little or no mention specifically of impact tests or toughness. As mentioned earlier, it recognizes API 5L pipe, which has within its specifications some consideration for toughness.

Along with its reliance on the basic specification of the pipe or material that it accepts, B31.4 has some basic concerns regarding both brittle and ductile fracture. These concerns are specifically related to carbon dioxide piping.

The reader should note that carbon dioxide, unlike many of the fluids contemplated for B31.4 piping, is a gas. Gas is inherently more dangerous to confine than a liquid. This is due to the relative compressibility of a gas compared to that of a liquid. In addition to the stored energy from the pressure, there is considerably more energy stored in the compression.

That fracture caution is seen in Paragraph 402.5 with subparagraphs 402.5.1 through 402.5.3. Subparagraph 402.5.1 simply states the designer shall provide reasonable protection so as to limit the occurrence and length of fracture. Then 402.5.2 states that the designer shall prevent brittle fracture by the selection of material, including invoking appropriate API 5L or similar supplementary requirements. The intent is to force any facture to be in the ductile range. Lastly, in 402.5.3 the designer shall minimize fracture by selection of pipe with appropriate fracture toughness and/or installing suitable fracture arrestors. This is accompanied by other considerations including the fracture toughness.

Code B31.8 has some specific requirements regarding fracture toughness. Remember that all the anticipated fluids are gas in the piping anticipated by this code. So the precautions are understandably more specific.

The requirements for mandatory specification to control fracture propagation are based on the amount of hoop stress designed into the system and the size of pipe used. That breakdown is as follows:

- For 16 NPS pipe and larger, the fracture criterion is mandatory when the hoop stress is 40 percent through 80 percent of the specified minimum yield strength.
- For sizes smaller than NPS 16, the hoop stresses must be over 72 percent through 80 percent for the criterion to be mandatory.

In subparagraph (1) the code specifies that the testing procedures shall be in accord with supplementary requirement S5 or S6 of API 5L. If the operating temperature is below 50°F, an appropriately lower test temperature shall be taken. Note that B31.8 has yet to be metricated. The lower test temperature shall be at or below the expected minimum metal temperature.

TABLE 4.3 Acceptable CVN Values for B31.8

Research programs	Formula
Batelle Columbus Laboratories (BCL) (AGA)	$CVN = 0.0108\ \sigma^2 R^{1/3} t^{1/3}$
American Iron and Steel Institute (AISI)	$CVN = 0.0345\ \sigma^{3/2} R^{1/2}$
British Gas Council (BGC)	$CVN = 0.0315\ \sigma\ R/t^{1/2}$
British Steel Corporation (BSC)	$CVN = 0.00119\ \sigma^2 R$

NOTE: For all equations

CVN = full-size Charpy V-notch absorbed energy, ft · lb
R = pipe radius, in.
t = wall thickness, in.
σ = hoop stress, ksi

There are alternative acceptance criteria in B31.8. They are as follows for brittle fracture:

- The shear appearance of the specimens shall be not less than 60 percent from each heat.
- The all-heat average for each order per diameter size and grade shall not be less than 80 percent.
- Drop weight tear testing is an alternative. If it is used, at least 80 percent of heats shall exhibit an appearance of 40 percent or greater shear.

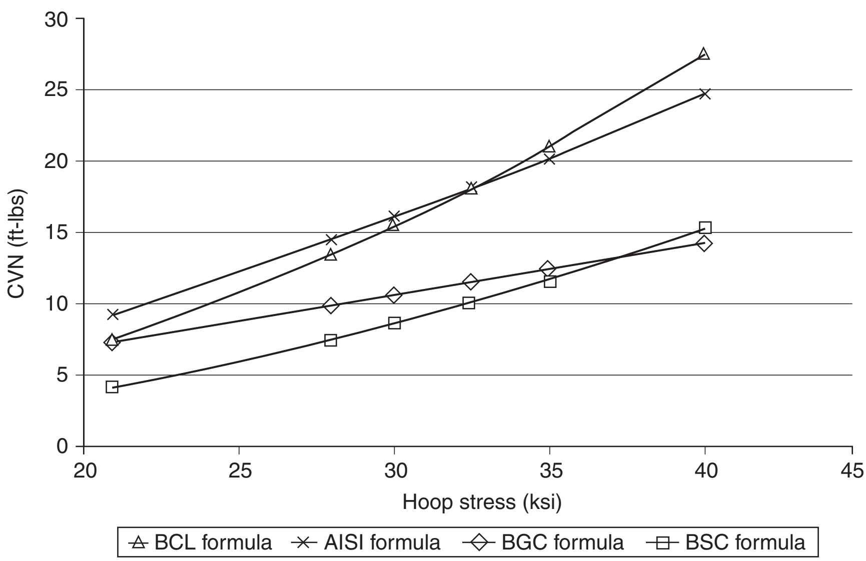


Figure 4.5 B31.8 CVN formula comparison.

Ductile fracture arrest requires that the test be done in accordance with supplementary requirement S5 of API 5L. The acceptance values are set by calculations using one of four equations developed in various research programs. Those equations are listed in Table 4.3.

You may wonder what the difference might be among the values for each of the formulas. Figure 4.5 is shown for reference. The pipe is chosen as NPS 16, $R = 8$; the t is chosen at 0.5 in., and the hoop stress is posited at 50 percent of six different SMYS values—42, 56, 60, 65, 70, and 80 ksi.

Nonmetallic materials

Not all the codes address nonmetallic materials, for example, B31.5 in the aboveground group and B31.4 and B31.11 in the buried group. In Code B31.3 nonmetallic materials are addressed in Chapter VII. In B31.1 they are addressed in a *nonmandatory* appendix, Appendix III. Both list allowable stresses for such materials in the appropriate stress appendices.

Codes B31.8 and B31.9 address such materials in paragraphs in the appropriate places where there is sufficient difference in the way the nonmetallic material is to be handled in the code that requires specific mention. Table 4.4 lists those paragraphs.

Note that where chapter numbers are listed as applying to nonmetallic pipe, mostly plastic, it does not mean the entire chapter is devoted to the nonmetallic pipe. These codes work in the nonmetallic pipe by including a portion which may be a subparagraph establishing a somewhat different requirement, caution, or guidance for those materials from the base material.

B31.8 includes a table, duplicated in Fig. 4.6, that provides a convenience to the code user in selecting thermoplastic pipe. The table uses a standard dimension ratio. When one is using the same thermoplastic material, one does not need to recalculate the required wall thickness

TABLE 4.4 Paragraphs Addressing Nonmetallic Materials

B31.9	B31.8
900	804
902	805
904	814
905	817
911	842
921.1.3(d)	849
923	
926	
934	

Table 842.32(c) Wall Thickness and Standard Dimension Ratio for Thermoplastic Pipe

Nominal Pipe Size	Outside Diameter, in.	Minimum Wall Thickness, in., for Standard Dimension Ratio, *R* 26	21	17	13.5	11
1/2	0.840	0.062	0.062	0.062	0.062	0.076
3/4	1.050	0.090	0.090	0.090	0.090	0.095
1	1.315	0.090	0.090	0.090	0.097	0.119
1 1/4	1.660	0.090	0.090	0.098	0.123	0.151
1 1/2	1.660	0.090	0.090	0.112	0.141	0.173
2	2.375	0.091	0.113	0.140	0.176	0.216
2 1/2	2.875	0.110	0.137	0.169	0.213	...
3	3.500	0.135	0.167	0.206	0.259	...
3 1/2	4.000	0.154	0.190	0.236	0.296	...
4	4.500	0.173	0.214	0.264	0.333	...
5	5.563	0.224	0.265	0.328	0.413	...
6	6.625	0.255	0.316	0.390	0.491	...

GENERAL NOTES:

(a) *Standard Dimension Ratio.* The Standard Dimension Ratio System enables the user to select a number of different sizes of pipe for a piping system, all of which will have the same design pressure. When plastic materials of the same design strengths are used, the same Standard Dimension Ratio may be used for all sizes of pipe instead of calculating a value of t for each size.

(b) Wall thicknesses above the line are minimum values and are not a function of the Standard Dimension Ratio.

Figure 4.6 Standard dimension ratio.

when one has the same design pressure. The standard dimension ratio is the outside diameter of the pipe divided by the wall thickness (e.g. 2.375/.091 = 26). All pipe with a 26 standard dimension ratio would be acceptable. Only one size pipe thickness needs to be computed; all sizes with the same ratio would work at that condition.

Chapter VII of B31.3 is in reality a subbook of B31.3. The practice of the B31.3 committee when it has a fluid or type of service that merits inclusion in the book is to create a chapter. There are four such chapters, or subbooks, in addition to the base book:

- Chapter VII, *Nonmetallic Piping and Piping Lined with Nonmetals* (A-XXX)
- Chapter VIII, *Piping for Category M Fluid Service, Metallic* (M-XXX)
- Chapter VIII, *Parts 11 though 20, Category M, Nonmetallic* (MA-XXX)
- Chapter IX, *High-Pressure Piping* (K-XXX)

Practice is to duplicate the paragraphs in the base code by number and to add a letter prefix to indicate in which chapter they are located; that nomenclature is indicated in parentheses in the above list as the letter prefix for that portion—XXX indicates a particular number. The code says that the comparably numbered paragraph in the base code applies either in its entirety or with an exception it may include the deletion of a subparagraph, or a substitute paragraph indicating what the requirements for that specific service will be.

Paragraph A342.4.2 lists some specific requirements for the anticipated materials. Those specific requirements are summarized here:

- Thermoplastics are prohibited aboveground when employed with flammable fluids. Safeguards are required for all but category D fluids.
- PVC and CPVC are prohibited from being used with compressed gas.
- Safeguarding is required for reinforced plastic mortar (RPM) in other than category D service.
- Safeguarding reinforced thermosetting resin (RTR) is required for use in toxic or flammable service. Temperature limits are recommended in the code.
- Safeguarding is required for borosilicate glass and porcelain when used in toxic or flammable service. Safeguarding against rapid temperature changes shall be employed in fluid services.

Appendix III of B31.1 carries the same title as Chapter VII in B31.3. It is essentially the same type of subbook mentioned for B31.3. However, it is organized differently and does not refer to paragraphs in the base book.

TABLE III-4.1.1
NONMETALLIC MATERIAL AND PRODUCT STANDARDS

Standard or Specification	Designation[1,2]
Nonmetallic Fittings	
Threaded Poly(Vinyl Chloride) (PVC) Plastic Pipe Fittings, Sch 80	ASTM D 2464-90
Poly(Vinyl Chloride) PVC Plastic Pipe Fittings, Schedule 40	ASTM D 2466-90a
Socket-Type Poly(Vinyl Chloride) (PVC) Plastic Pipe Fittings, Schedule 80	ASTM D 2467-90
Acrylonitrile-Butadiene-Styrene ABS Plastic Pipe Fittings, Schedule 40	ASTM D 2468-89
Thermoplastic Gas Pressure Pipe, Tubing, and Fittings	ASTM D 2513-90b
Reinforced Epoxy Resin Gas Pressure Pipe and Fittings	ASTM D 2517 (R1987)
Plastic Insert Fittings for Polyethylene (PE) Plastic Pipe	ASTM D 2609-90
Socket-Type Polyethylene Fittings for Outside Diameter-Controlled Polyethylene Pipe and Tubing	ASTM D 2683-90
Chlorinated Poly(Vinyl Chloride) CPVC Plastic Hot and Cold Water Distribution Systems	ASTM D 2846-90
Butt Heat Fusion Polyethylene (PE) Plastic Fittings for Polyethylene (PE) Plastic Pipe and Tubing	ASTM D 3261-90
Polybutylene (PB) Plastic Hot-Cold-Water Distribution Systems	ASTM D 3309-89a
Reinforced Thermosetting Resin (RTR) Flanges	ASTM D 4024-87
Threaded Chlorinated Poly(Vinyl Chloride) (CPVC) Plastic Pipe Fittings, Schedule 80	ASTM F 437-89a
Socket-Type Chlorinated Poly(Vinyl Chloride) (CPVC) Plastic Pipe Fittings, Schedule 40	ASTM F 438-89a
Socket-Type Chlorinated Poly(Vinyl Chloride) (CPVC) Plastic Pipe Fittings, Schedule 80	ASTM F 439-89
Electrofusion Type Polyethylene Fittings for Outside Diameter Controlled Polyethylene Pipe and Tubing	ASTM F 1055-98
Nonmetallic Pipe and Tube Products	
Polyethylene Line Pipe	API 15LE (1987)
Thermoplastic Line Pipe (PVC and CPVC)	API 15LP (1987)
Low Pressure Fiberglass Line Pipe	API 15LR (1986)
Concrete Sewer, Storm Drain, and Culvert Pipe	ASTM C 14-82
Acrylonitrile-Butadiene-Styrene (ABS) Plastic Pipe, Sch 40 and 80	ASTM D 1527-77 (1989)
Poly(Vinyl Chloride) (PVC) Plastic Pipe, Sch 40, 80 and 120	ASTM D 1785-89
Polyethylene (PE) Plastic Pipe, Schedule 40	ASTM D 2104-89
Polyethylene (PE) Plastic Pipe (SIDR-PR) Based on Controlled Inside Diameter	ASTM D 2239-89
Poly(Vinyl Chloride) (PVC) Pressure-Rated Pipe (SDR Series)	ASTM D 2241-89
Acrylonitrile-Butadiene-Styrene (ABS) Plastic Pipe (SDR-PR)	ASTM D2282-89
Machine-Made Reinforced Thermosetting-Resin Pipe	ASTM D 2310-80 (1986)
Polyethylene (PE) Plastic Pipe, Sch 40 and 80, Based on Outside Diameter	ASTM D 2447-89
Thermoplastic Gas Pressure Pipe, Tubing, and Fittings	ASTM D 2513-86A
Reinforced Epoxy Resin Gas Pressure Pipe and Fittings	ASTM D 2517 (R1987)
Polybutylene (PB) Plastic Pipe (SIDR-PR) Based on Controlled Inside Diameter	ASTM D 2662-89
Polybutylene (PB) Plastic Tubing	ASTM D 2666-89
Joints for IPS PVC Pipe Using Solvent Cement	ASTM D 2672-89
Polyethylene (PE) Plastic Tubing	ASTM D 2737-89
Chlorinated Poly(Vinyl Chloride) (CPVC) Plastic Hot- and Cold-Water Distribution System	ASTM D 2846-90
Filament-Wound "Fiberglass" (Glass-Fiber Reinforced Thermosetting-Resin) Pipe	ASTM D 2996-88
Centrifugally Cast Reinforced Thermosetting Resin Pipe	ASTM D 2997-90
Polybutylene (PB) Plastic Pipe (SDR-PR) Based on Outside Diameter	ASTM D 3000-89
Polyethylene (PE) Plastic Pipe (SDR-PR) Based on Controlled Outside Diameter	ASTM D 3035-89a
PB Plastic Hot-Water Distribution Systems	ASTM D 3309-89a
Chlorinated Poly(Vinyl Chloride) (CPVC) Plastic Pipe, Schedules 40 and 80	ASTM F 441-89
Chlorinated Poly(Vinyl Chloride) (CPVC) Plastic Pipe, (SDR-PR)	ASTM F 442-87
Plastic-Lined Ferrous Metal Pipe, Fittings, and Flanges [Note (3)]	ASTM F 1545-97
PVC Pressure Pipe, 4-inch through 12-inch, for Water	*AWWA C 900
AWWA Standard for Glass-Fiber-Reinforced Thermosetting-Resin Pressure Pipe	*AWWA C 950-88
Miscellaneous	
Standard Methods of Testing Vitrified Clay Pipe	ASTM C 301-87
Contact-Molded Reinforced Thermosetting Plastic (RTP) Laminates for Corrosion Resistant Equipment	ASTM C 582-87
Standard Definitions of Terms Relating to Plastics	ASTM D 297-81
Standard Abbreviations of Terms Relating to Plastics	ASTM D 1600-90
Threads 60° (Stub) for "Fiberglass" (Glass-Fiber-Reinforced Thermosetting-Resin) Pipe	*ASTM D 1694-91
Solvent Cements for Acrylonitrile-Butadiene-Styrene (ABS) Plastic Pipe and Fittings	ASTM D 2235-88
External Loading Properties of Plastic Pipe by Parallel-Plate Loading	ASTM D 2412-87
Solvent Cements for Poly(Vinyl Chloride) (PVC) Plastic Pipe and Fittings	ASTM D 2564-88
Heat-Joining Polyolefin Pipe and Fitting	ASTM D 2657-90
Obtaining Hydrostatic Design Basis for Thermoplastic Pipe Materials	ASTM D 2837-90
Making Solvent-Cemented Joints With Poly (Vinyl Chloride) (PVC) Pipe and Fittings	ASTM D 2855-90
Standard Test Method For External Pressure Resistance of Reinforced Thermosetting Resin Pipe	ASTM D 2924-86
Obtaining Hydrostatic or Pressure Design Basis for "Fiberglass" (Glass-Fiber-Reinforced Thermosetting-Resin) Pipe and Fittings	*ASTM D 2992-87
Joints for Plastic Pressure Pipes Using Flexible Elastomeric Seals	ASTM D 3139-89
Underground Installation of "Fiberglass" (Glass-Fiber Reinforced Thermosetting Resin) Pipe	ASTM D 3839-89
Design and Construction of Nonmetallic Enveloped Gaskets for Corrosive Service	ASTM F 336-87
Solvent Cements for Chlorinated Poly(Vinyl Chloride) (CPVC) Plastic Pipe and Fittings	ASTM F 493-89
Electrofusion Joining Polyolefin Pipe and Fitting	ASTM F 1290-98a
Plastic Pipe Institute (PPI) Technical Report Thermal Expansion and Contraction of Plastic Pipe	PPI TR21-88

NOTES:

(1) An asterisk (*) preceding the designation indicates that the standard has been approved as an American National Standard by the American National Standards Institute.

(2) Numbers in parentheses are reapproval dates.

(3) This standard contains no pressure-temperature ratings. Paragraph III-2.1.2(B.3) applies.

Figure 4.7 Code Table III-4.1.1.

For its material it has Code Table III-4.1.1 (see Fig. 4.7), which lists the standards and materials that are recognized by that appendix. Paragraph III-4.1.3 lists restrictions on the use of that table. It states that other references may be in the standards but do not apply except as in the context of the documents listed in the table. It further states that the rules of this appendix govern in the case of conflict. It does not allow the appendix to be other than for a piping system constructed for this code.

Chapter

5

Design

A Brief History and Discussion of Pressure Design for Piping

It must be remembered that the B31 codes are intended to be simplified codes. This means that the user of the codes can follow the equations and directions in the codes and design a system that has reasonable expectations of working for the assumed life of the design.

Nevertheless, the user of the codes will be well served to have a basic understanding of the underlying considerations. The codes generally allow more rigorous analysis when performed by a capable engineer who can demonstrate that the analysis meets the fundamental criterion of the code.

As this book progresses from section to section, a brief discussion of the background will be given. It is not intended to provide a textbook type of insight. It is intended to give a broader understanding of some of the underlying science, mathematics, and technology that go into "simplifying" a very complicated subject.

In their essential approach to pressure design, the codes recognize two types of stresses, called primary and secondary stresses. *Primary stresses* are those usually involved in the pressure design. *Secondary stresses* are those usually found in flexibility, or fatigue, analysis.

Primary stresses come from specifically applied loads, such as internal pressure on the pipe, external pressure, weight, wind, and earthquake loads. The magnitude of the load creates the stress in the element being calculated.

Secondary stresses result from some sort of restraint on the system. The most common is the load from thermal expansion between two anchor points. If the pipe is heated, it expands. Given no restraints, there are no stresses generated from that expansion. The pipe is free to expand.

In any piping system there are always some restraints, if only the equipment or building the pipe may be connected to. Assume a straight carbon steel pipe connected between two buildings 1 mi apart. Heat the pipe 100°F higher than its installation temperature. It will try to expand slightly more than 42 in. It is unlikely that the pipe will move the buildings; it can only bow up, creating stress. This is discussed more thoroughly in the sections on flexibility analysis.

To return to pressure design, although there are many other loads to consider, the most frequent one is the hoop stress. The stress limitations of the material chosen for the service conditions for which the designer is working will define the thickness of the pipe required.

The hoop stress calculation is the fundamental one in the codes. One way of calculating the thickness required is to establish the pressure rating of the system. All other components should have a pressure-temperature rating equal to, or higher than, that for which the pipe has been designed. If not, that component will limit the system's rating.

A Frenchman named Lamé who lived in the early 1800s defined an elegant stress theory for thick-walled pipe. It is somewhat more mathematically tedious than the current forms in the B31 codes. Those current equations are forms of what is known as the *Barlow hoop stress equation*.

This equation is developed on the assumption of a *thin wall*. In B31.3 this is defined as $t < D/6$. This condition is met by standard pipe. The basic equation is $t = pD/(2S)$; for the development of this equation, see App. D, which includes the rationale for the development of the modification. Modifcation came about as a result of trying to more closely approximate Lamé's more exact equation. It was further complicated by the increases in temperature and pressure, which caused the codes to change the limiting stress from a variation of ultimate tensile strength or yield to a creep mode.

So, except for the refinements mentioned above, the basic equations have been in existence and used for approximately 200 years. The ancients mentioned in Chap. 1, who were concerned mostly with lower-pressure water, did not have the refinements to work with the modern complexities.

Bases for Allowable Stresses

There are two distinct patterns in establishing allowable stresses within the B31 codes. Fortunately, they are divided by the arbitrary categorical differences between buried piping and aboveground piping. However, there are minor differences within each of the two major categories.

Buried piping allowed stresses

The buried piping group works exclusively with the specified minimum yield strength (SMYS). The temperature ranges over which those codes accept jurisdiction are relatively low. So as the temperature rises, the changes that occur in most materials have minimal effects. These codes handle that minimal effect by means of a temperature reduction factor. Both B31.4 and B31.11 limit their high-temperature range to a high of 120°C (250°F). So in effect their temperature reduction factor does not go below 1 and essentially disappears in their design formula.

These piping systems can cover a relatively large expanse of territory where the safety concerns vary. This is especially true of Code B31.8. Because of the volatility of the gases that are their intended fluid, they vary their design *location factor*. Once again, the two liquid codes set their factors at 1, so it disappears from the equations.

Each grade of piping can vary in its overall integrity due to the method of manufacture or type of material. So for each type of material, these codes have a factor assigned to the material type that reflects the amount of stress allowed. This is known as the *longitudinal joint factor*.

Codes B31.4 and B31.11 treat the factors as applying to the SMYS and thus develop a project allowable stress based on the factors including those hidden factors of 1. One of the curious results of this factor approach to establishing stresses is that the resulting wall thickness is the nominal wall thickness not the minimum.

Each code points out in its definition that the calculated wall thickness means the nominal wall thickness. The codes have made the allowance to include the expected manufacturing tolerances and other variations that may apply. All codes put the responsibility for any mechanical or corrosion allowances on the designer.

Such allowances are beyond the scope of the code and rightly belong with the specifics of whether any allowances, including thread and similar machining-type allowances, need to be made. The codes cannot determine in advance which material will be chosen for what fluid, or any other corrosive environment the pipe might see. Such determinations rightly belong in the project design.

The resulting design formulas for wall thickness or pressure differ only in form. The designer for B31.4 and B31.11 needs to make the calculation for the project allowable stress and then, using that stress, calculate the wall thickness. The B31.8 designer uses the SMYS directly in the thickness formula and includes the three factors. The net result can be the same depending on the factors used.

B31.8 has this variable design location factor. It is based on the type of geographic area in which the pipe is located at the point of design under consideration. If we remember the line may go for miles through all sorts of geographic areas, it becomes clear that even though every other condition may be the same, the required thickness of the pipe may vary throughout the system.

Those requirements are defined in a table, shown on the following page as Fig. 5.1, entitled "Design Factors for Steel Pipe Construction." Once the location class is determined, the design factor to use in the wall thickness formula can be determined from the chart.

The highest factor, i.e., the one that would give the thinnest pipe, is safe for pressure containment in all locations. However, studies have shown that certain activities around the pipeline can cause accidents. Long-term studies of the causes of failure of both liquid and gas pipelines have shown that around 33 percent of failures are caused by third-party damage. In the B31.8 systems, the code includes the distribution, and often the distribution is in areas where third-party activity can be relatively higher. Therefore, the B31.8 system has a more elaborate definition of class changes that might be said to be related to this activity in its establishment of location classes.

Another consideration in lowering these design factors is the expected concentration of people in the proximity of the line. This can be related to the potential severity of any failure and its attendant loss of people, etc. Paragraph 840 and all its subparagraphs in the code discuss these factors in greater detail.

To summarize, the first step is to establish an area ¼ mile to either side of the pipeline where the pipeline is the centerline of this strip. This area and consideration of future development within the strip are used to establish the location class. Table 5.1 gives a brief description of the location classes.

The code (B31.8) allows metal temperatures up to 450°F. It has a derating factor for temperatures of 250°F or higher. As noted in B31.4 and B31.11 where the maximum temperature is 250°F, up to that temperature the factor in B31.8 is 1. Above that temperature, the factor drops 0.033 for every 50°F up to 450°F, which is the upper limit in B31.8. Between the 50°F increments interpolation is specifically allowed.

TABLE 5.1 Location Class Description

	Facilities in any 1-mi section
Class 1, Divs. 1 and 2	Areas such as wasteland and farmland with comparable sparse populations. Divisions 1 and 2 relate to test pressures.
Class 2	Facilities for human occupancy of 10 to 46
Class 3	Facilities for human occupancy of 46 or more
Class 4	Multistory and extensive traffic and utilities

Table 841.114B Design Factors for Steel Pipe Construction

Facility	Location Class 1 Div. 1	Location Class 1 Div. 2	2	3	4
Pipelines, mains, and service lines [see para. 840.21(b)]	0.80	0.72	0.60	0.50	0.40
Crossings of roads, railroads without casing:					
(a) Private roads	0.80	0.72	0.60	0.50	0.40
(b) Unimproved public roads	0.60	0.60	0.60	0.50	0.40
(c) Roads, highways, or public streets, with hard surface and railroads	0.60	0.60	0.50	0.50	0.40
Crossings of roads, railroads with casing:					
(a) Private roads	0.80	0.72	0.60	0.50	0.40
(b) Unimproved public roads	0.72	0.72	0.60	0.50	0.40
(c) Roads, highways, or public streets, with hard surface and railroads	0.72	0.72	0.60	0.50	0.40
Parallel encroachment of pipelines and mains on roads and railroads:					
(a) Private roads	0.80	0.72	0.60	0.50	0.40
(b) Unimproved public roads	0.80	0.72	0.60	0.50	0.40
(c) Roads, highways, or public streets, with hard surface and railroads	0.60	0.60	0.60	0.50	0.40
Fabricated assemblies (see para. 841.121)	0.60	0.60	0.60	0.50	0.40
Pipelines on bridges (see para. 841.122)	0.60	0.60	0.60	0.50	0.40
Pressure/Flow Control and Metering Facilities (see para. 841.123)	0.60	0.60	0.60	0.50	0.40
Compressor station piping	0.50	0.50	0.50	0.50	0.40
Near concentration of people in Location Classes 1 and 2 [see para. 840.3(b)]	0.50	0.50	0.50	0.50	0.40

Figure 5.1 Code Table 841.114B.

Aboveground piping stresses

The aboveground piping group follows, to a degree, the Boiler and Pressure Vessel Code (BPVC) practice of establishing a specific stress for a specific material grade at a specific temperature. It was noted in the materials section that the Section 2 material committee in the BPVC set of committees is the source of the base data for all these recognized materials.

That source supplies the basic material, and each one of the B31 aboveground group sets its own acceptance criteria. For a number of years, the B31.1 group was in complete accord with the Section II stresses. This is not totally the case at present.

In the late 1990s there was a significant change in the BPVC allowable stresses. The method for determining the allowable stress from the tensile strength of the material was changed from $^1/_4$ times tensile strength to $^1/_{3.5}$ times tensile strength. This was done after a careful study concluded that this change could be made without impacting the safety of the affected pressure equipment. Note that the last change was made over 40 years ago, and it was from $^1/_5$ to $^1/_4$.

In its present tables, B31.1 still has published stresses that were computed using the older value of $^1/_4$ times tensile strength as one of its determinants. That committee is in the process of developing those higher allowable stresses for the tables. In the interim, they have published B31 Code Case 173. That case allows, with listed restrictions, the use of the factor of $^1/_{3.5}$ times tensile strength, where applicable. One of the major restrictions is that the material must be listed in Table 126.1 and have a corresponding ASME material as allowed in Paragraph 123.1 of Code B31.1.

B31.3 has consistently had higher allowable stresses than B31.1 or the BPVC with one exception. The B31.3 allowables do correlate with the Section VIII, Division 2, allowables. That section is essentially an alternative code to Division 1 with much higher requirements for analysis. It has been generally believed that the stresses were higher because the process industries had shorter expected plant life.

B31.5 and B31.9 have similar allowable stress patterns with somewhat fewer materials. Code B31.9 follows to a close degree Code B31.1. It allows substitution of B31.1 rules where greater analysis or effort is required in many instances. B31.5 follows the general pattern and uses the $^1/_4$ times tensile strength rule.

The buried group, as discussed above, works exclusively with the SMYS. These aboveground codes have other considerations. They do encompass considerably higher temperatures and therefore see changes in the material properties that do not occur in the pipeline groups. That essentially establishes a four-part test as the temperature increases.

It was discussed at some length in Chap. 4 how the various codes handle the lower temperatures. Each of the codes describe the bases of the stress allowed in the tables.

For purposes of simplicity, the specific rules of B31.3 are listed in Table 5.2 and in the paragraphs in the other codes which could, and do at present, have variations essentially based on the discussions of the BPVC changes which are working their way through the committees. The general intent of all is the same. At present, where a material has allowable uses at or above the creep range, the allowable stresses should continue to be the same regardless of the particular code one is searching.

The specific paragraphs that describe the details for establishing allowable stresses in each of the aboveground codes are as follows:

- B31.1 Paragraph 102.3, Code Case 173
- B31.3 Paragraph 302.3
- B31.5 Paragraph 502.3
- B31.9 Paragraph 902.3

It is pleasant to note that the attempt to have similar paragraphs in each of the books holds in this particular case.

It might be noted by the astute reader that in the upper temperatures, where creep begins to control the establishment of the allowable stress, there could be room for the designer use different values when non-time-dependent actions are contemplated. Certainly the committees

TABLE 5.2 Bases for Allowable Stress Tables of Materials

When all limits have been considered the lowest result shall be the control

Other than bolting, cast iron, or malleable iron	
Property	Limit
Tensile strength	The lower of ⅓ times tensile strength and ⅓ times tensile strength at temperature
Yield strength	The lower of ⅔ times yield strength or ⅔ of times yield at temperature
Creep	100% of average for a creep rate of 0.01% per 1000 h
Rupture	67% of average stress for rupture at end of 10,000 h
Rupture	80% of the minimum stress for rupture at end of 100,000 h

NOTE 1: For austenitic stainless and nickel alloy with similar stress-strain behavior, the criterion can change to ⅔ times yield strength and 90 percent of yield strength at temperature. However, this is not recommended for flanges or other locations where the distortion might cause a leak or malfunction. There are detailed instructions as to what to use in those cases.

NOTE 2: Structural materials shall be 0.92 times the lowest value obtained by following the above steps.

NOTE 3: Each code has specific rules which are similar.

writing the requirements did, and they will be noted as progress is made toward specific design criteria.

Design of pipe

The final determination allowing the designer to set the final pipe thickness requires that a design pressure and temperature be chosen. This is not as simple a task as it might seem. The buried pipe group is relatively less specific as to what design pressure is in the codes, whereas the aboveground piping group has specific definitions of these terms.

The form of the equations in either group of codes may be manipulated algebraically to calculate any one of the three variables—pressure, stress, or thickness—when the other two are known. This most often comes into play in the buried pipelines. It should be recalled that pipelines are intended to cover miles of terrain, with potential elevation changes, which would lead to variations from the base pressure.

B31.11 uses the term *maximum steady-state operating pressure*. It stipulates that pressure to be the sum of static head pressure, pressure to overcome friction losses, and any backpressure. The steady-state pressure is also used in B31.4. Code B31.8 uses a somewhat backward definition by invoking in Paragraph 803.212 the design pressure as the maximum pressure permitted by this code, as determined by the procedures applicable to the materials and locations involved.

Remember that in most cases there is not a temperature problem to be dealt with. Only Code B31.8 recognizes temperatures above 250°F and then to only 450°F where the reduction is less than 14 percent. In those portions of a line that are buried, the temperature would be very steady and would not fluctuate drastically.

Essentially the above discussion boils down to this: The design pressure must, of necessity, be outside the scope of the code to predetermine, as the line must be laid out to find elevation differences and must be sized to calculate friction losses, and the end conditions must be established to determine any backpressure.

The maximum allowable working pressure would be a calculation based on the materials, the thickness of the pipe, or the pressure rating of the component to determine that pressure. Then one must be sure to not operate above it.

There are surge-type rules and conditions for which the code defines limitations. In some situations because of some outside factor, the operating conditions have changed in such a way that the line has to have a lowered operating pressure. This lowering of pressure is sometimes permanent but often temporary until a particular condition can be corrected.

TABLE 5.3 Paragraphs Defining Design Temperature and Pressure

Code	Paragraph for design temperature	Paragraph for design pressure
B31.1	101.3	101.2
B31.3	301.3	301.2
B31.5	501.3	501.2
B31.9	901.3	901.2

NOTE 1: B31.1 and B31.9 have paragraph x01.1 wherein they invoke the requirement for the temperature and pressure to be established at the most severe coincident temperature and pressure and loading.
NOTE 2: B31.3 and B31.5 include the most severe condition in the paragraph on pressure.
NOTE 3: Unless otherwise computed, the metal temperature in the calculations is considered to be the same as the temperature of the fluid.

There is further discussion of these phenomena when we discuss the operation of the lines later in the book. Remember that one of the uses of pressure in a transportation line is the speed of delivery of the fluid. A lowering of pressure may reduce the amount of flow velocity but not the actual delivery volume.

A somewhat different situation exists in the aboveground groups. Each of those codes tends to establish both a design temperature and a pressure. The piping system may have many states during the course of operation. Many of these would have varying pressures and temperatures. And since the allowable stress would change with temperature, the thickness of the pipe or the pressure needs of the component could change. The defining paragraphs for design temperature and pressure for each of the codes are shown in Table 5.3.

The most severe coincident temperature and pressure is defined as that which causes the thickest pipe or the highest component rating. It becomes apparent that a system that has many states would have to be checked for all, to make that determination. This severe case sets the design criterion, but the system will have to be checked for all the other states. There are usually at least two states, operating and design.

Actual comparison

The base state of any piping system is the pressure rating of the straight pipe in the system. One can't compare all the ratings for each of the codes for all the temperatures. One can pick a set of conditions and a common material and size of pipe that would be possible to use in each code. Once that is done, a sense of technical difference can be determined.

Suppose one chose the following:

- Design temperature of 250°F
- Design pressure of 500 psig

- NPS 6 pipe
- ASTM A-106 C pipe material (E = 1)
- 0.0 corrosion or mechanical allowance

Table 5.4 gives the straight pipe wall formula from each code and calculates the required, or code, thickness.

Because the allowable stresses are different for the different codes, they are listed here (all are in psi): B31.1, 17,500; B31.3, 23,300; SMYS for the pipeline codes (B31.4, B31.8, B31.11), 40,000; B31.5, 17,500; and B31.9, 15,000. The formula for B31.3 Chapter IX is quite different. The intent is to provide a factor of not less than 2 on the pressure required to initiate yielding on the outside surface of the pipe. The factor c_o in that equation is for any corrosion or other allowance on the outside surface.

TABLE 5.4 Required Thickness in Inches for Each Code

Code	Formula	Calculated t	Comment
B31.1	$t_m = \dfrac{PD_o}{2(SE+Py)} + A$	0.094	6 std = 0.245 min
B31.3	$t_m = \dfrac{PD_o}{2(SE+Py)} + A$	0.070	6 std = 0.245 min
B31.3 (chapter IX)	$t = \dfrac{D_o - 2c_o}{2}\left[1 - \exp\left(\dfrac{-1.155P}{S}\right)\right]$	0.080	See discussion for comments
B31.4	$t = \dfrac{PD_o}{2SE(0.72)}$	0.057	6 std = 0.245 min
B31.5	$t_m = \dfrac{PD_o}{2(S+Py)} + A$	0.094	6 std = 0.245 min
B31.8	$t = \dfrac{PD_o}{2SFET}$	0.082	6 std = 0.245 min
B31.9	$t = \dfrac{PD_o}{2SE} + A$	0.109	6 std = 0.245 min actual is A-106 B
B31.11	$t = \dfrac{PD_o}{2SE(0.80)}$	0.051	6 std = 0.245

In this table,

P = pressure
D_o = outside diameter
F = B31.8 design factor (used compressor station = 0.5)
T = temperature derating factor of 1
A = corrosion/mechanical factor

The 500 psig in the example is not really considered a high pressure, which is why the chapter exists for comparison. If the pressure were 5000 psig, then the required wall thickness, all other things being equal, would be 0.72 in. versus 0.67 in. for the standard formula. Chapter IX has a higher margin due to the amount of damage from failure at the higher pressure.

It is easy to see that there are mathematical differences. If one is using ASME standard sizes, the same size pipe would probably be used in all cases. A different design factor for B31.8 would probably be higher, meaning thinner than the table calculated. Higher temperature often means thicker wall, as the allowable stresses would be lower.

Certainly, as discussed above, if one were buying miles of pipe, one might order less than standard wall pipe. However, the thicknesses needed for pressure design might be too thin for structural reasons and to meet some of the other design requirements. So the designer would have to take other factors into consideration.

It is most important to meet the wall thickness requirements. This simple example shows that it may not be the critical point in the design of the system. Larger-diameter pipe would result in the need for a thicker wall. So would a higher pressure. One begins to perceive the many avenues that the engineer has as options in pursuit of the optimal design.

External pressure design

While the standard pressure consideration is to design against internal pressure, situations in the operation may arise in which the pipe is subject to external pressure. This could be a vacuum in the line or a double pipe situation where the external pressure is greater than the internal. In offshore or underwater pipelines, this possibility must be considered.

External pressure, unlike internal pressure, becomes more damaging as the length of the unsupported surface increases. Naturally, because pipelines as a rule have longer lines than aboveground piping, this is a concern. However, it is also a concern in the other aboveground codes.

The B31 codes address their specific requirements by referencing Section VIII, Division 1 procedures. In that code section there is a detailed and somewhat complicated solution to determining the ability of the pipe to withstand the designed-for external pressure. The B31 codes have found that it is better to reference that technique than to either repeat it or develop a competing procedure.

Some codes specifically list the paragraphs; others require that the problem be addressed. The Section VIII procedure is therefore introduced by implication. Some of the codes list specific minor changes to ensure that the user makes adjustments to consider that piping system's requirements. Table 5.5 shows the paragraph and comments on the requirements by code.

TABLE 5.5 External Reference Paragraphs by Code B31

Code	Paragraph references	Comment
B31.1	104.1.3	Specific reference to Section VIII, Division 1 Ug 28-UG 30
B31.3	304.1.3, 304.2.4, and corresponding paragraphs in chapters VII & IX	Specific reference to Section VIII, Division 1, with adjustments
B31.4	401.2.3	Implicit reference with a shall requirement
B31.5	501.2.3, 504.1.3	Specific reference to Section VIII, Division 1, with adjustments
B31.8	A842.11	Implicit reference with a shall requirement
B31.9	904.1.2	Specific reference to Section VIII, Division 1
B31.11	1104.1.3	Implicit reference with a shall requirement

Design bends, miters, or elbows

A piping system of any extent will have bends or elbows to change the inline direction of the pipe. These changes may be two- or three-dimensional. There are differences in design technique as well as type of change involved.

The major difference between the types of bends is as much traditional as it is a major difference. The main result of the difference is the amount of space that it takes to achieve the bend.

Elbows are, by tradition, either long- or short-radius. The long radius is 1.5 times the pipe's nominal diameter. The short radius is equal to the pipe diameter. Historically short-radius elbows had a rating that was 80 percent of the pressure rating of the schedule of the matching pipe. That has recently changed as the manufacturers have rated their elbows at 100 percent.

Bends are traditionally pipe that is bent with a radius that is longer than 1.5 times the nominal pipe diameter of the long-radius elbow. Bends usually started at 3 times the nominal pipe diameter and went up from there. One of the major differences between bends is the method of manufacture and whether the bend is made hot or cold.

Miters are bends that are made from short, straight segments of metal. These segments are cut on some appropriate angle which is dependent on the number of segments utilized in making the bend. Miters are generally made for diameters of pipe that are larger than the radial bending machines or tooling is capable of making.

Each of these means of changing the pipe direction has limitations, and some of the codes impose limitations on their use. From a pressure design standpoint, the problem is that those components, either elbows or bends, have different thickness requirements due to pressure depending on the location in the bend. Many manufacturing techniques adjust for these different thickness requirements more or less automatically.

Others don't. The design of the miter takes these requirements into account.

Elbows are usually made in accord with some standard recognized by the B31 code. That standard would have some means of defining the pressure rating of that component. This book will discuss those issues in Chap. 8, which addresses listed and unlisted components.

Bends and miters in general do not have a defining standard, and the requirements are set out or defined in the code. There are some standards for the induction bending process. This standard was written for the pipeline industry, which uses bends extensively.

One of the universal sets of concerns regarding bends is the flattening that may occur as the pipe is bent. Caution is required regarding the possibility of needing a mandrel during the bending process to achieve the desired bend. The pipeline codes have a table that limits the bend radius by size. That table is the same for all three codes; the requirements are duplicated here as Table 5.6.

A smaller bend radius is permitted provided testing is done to prove that the requirements of other paragraphs are met. These include wall thickness, ovalizing limitations, wrinkles, and other requirements, including lowering of the allowable stress in certain cases. In all cases, tangents to the bend are specified.

There are specific limitations when a pipe has achieved its specified yield strength by cold working. This limitation is based on hot rather than cold bending. Should the temperature be beyond 600°F for 1 hour or 900°F for any length of time, the code recognizes that some of the strength acquired by the cold working has been lost and specifies a reduced allowable stress for calculations.

Codes B31.1 and B31.9 have similar requirements but do not include the large-radius bends as noted in the pipeline tables. These are not anticipated. Both those codes give guidance to the user as to how much thicker to order the pipe for such bends. Note that pipe for these codes is generally not assembled in the field. The thickness recommendations can be expressed as in Table 102.4.5 of B31.1. Or one can approximate it on a scientific calculator by the formula

$$\text{Extra thickness} = 1.64\ (\text{radius})^{-0.25}$$

TABLE 5.6 Minimum Field Cold Bends for B31.4, B31.8, and B31.11

Pipe size, NPS	Minimum bend radius, pipe diameter D
≤12	18D
14	21D
16	24D
18	27D
20+	30D

TABLE 5.7 Paragraphs for Design Requirements of Bends per Code

Code	Paragraphs
B31.1	104.21, 102.4.5
B31.3	304.2, 306.2, and corresponding A, M, MA, K
B31.4	404, 406, 434.7
B31.5	506, 504.3, 529.1
B31.8	841.23, 835.2, 842.414
B31.9	904.2.2, 902.4, 906.29
B31.11	1104.21, 1106.2

This formula does not give an exact match to the tables in the codes; however, it gives a proper order of magnitude. The reader will recall that pipe wall thickness does not necessarily fall in those calculated intervals.

Both B31.1 and B31.3 specify a method to calculate the theoretical wall requirement for the intrados, or inside, of the bend and one for the extrados, or outside, of the bend. The formula is somewhat more complex but is useful in more sophisticated bending processes. These may not require all the extra metal for the extrados. From a pressure design standpoint, it is more important to have the extra thickness on the intrados. This method gives a methodology to check whether the wall thickness is sufficient at any point in the bend. The reader who is interested can find the method in App. E.

The paragraphs which apply to bends in each of the codes are shown in Table 5.7.

Miter bends

Miters have a distinct definition in the codes. A *miter* is two or more straight sections of pipe matched and joined in a plane bisecting the angle of junctions so as to produce a change in direction. Most are familiar with the miter as the twin 45° cut that produces the square corner in a picture frame.

Miter bends are basically designed. There are two major types: those with multiple miters and those with a single miter. Those codes indicate that a 3° miter is not one that requires design consideration. Additionally, the design methodology given for either single or multiple miters as 22.5°. Although it is not a pressure-only requirement, the aboveground codes do recognize that the length of a single piece affects the overall safety of the miter.

As noted, the method of designing a miter is explained in App. E; within the limitations of the design method, the aboveground codes do not further limit the use of miters. Those miters will not be discussed further.

TABLE 5.8 Summary of Restrictions on Miter Bends by Pipeline Codes

Code	Paragraph	Max. hoop stress allowed, percent	Max. angle of miter allowed, deg.	Min. distance between miters, diameter D	Hoop stress re no restrictions
B31.4	406.2.2	20	12.5	$1D$	Less than 10%
B31.8	841.232	40	12.5	$1D$	Less than 10%, angle <45°
B31.11	1106.2.2	20	12.5	$1D$	Less than 10%

The pipeline codes do not necessarily define a design process for the miters, but they do place relatively severe limitations on their use. These limitations include the level of hoop stress allowed, the amount of the miter angle, and the minimum distance between miters. Those limitations are summarized and the relevant paragraphs are listed in Table 5.8.

Note that the lowest percent of yield strength allowed is 40 percent. That is rare, as it occurs in very high-density multistory B31.8 applications. The remainder of the factors on design SMYS range from 0.5 to 0.8.

Pressure design of intersections

Most piping systems require intersections. The codes recognize that there are design stress anomalies at that intersection. Many codes will assert, somewhat unnecessarily, that when a hole is cut into the pipe, the pipe is weakened. One pundit even mentioned that with the hole in it, the pipe would not hold pressure either. Piping systems by their nature are closed so that flow or pressure can be maintained.

When pressure is applied at an intersection, the smoothness of hoop stress is complicated by changes in direction. Figure 5.2 is a generic

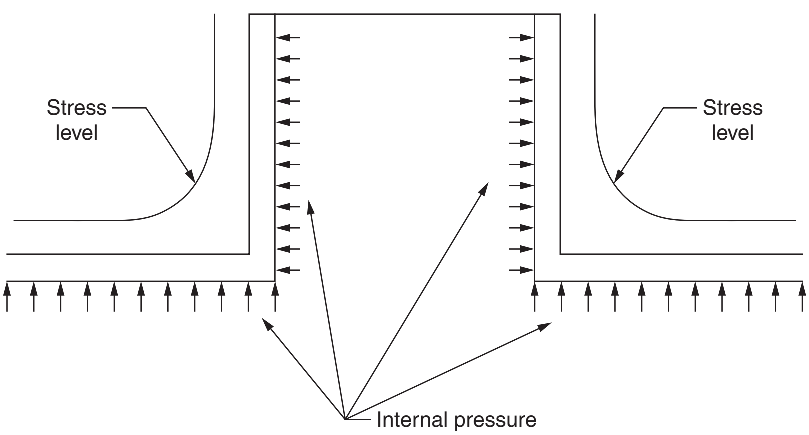


Figure 5.2 Relative stress levels in intersections.

picture showing what basically is happening in that intersection. The increased stress around the intersection has a high probability of placing the material in an overstressed condition.

The codes recognize that in some cases the higher stress will not be enough to cause a need for additional reinforcement. These cases are listed. In those listings, they referenced certain standard components that could provide sufficient reinforcement. Finally they give methods to achieve reinforcement.

As a way to show how this reinforcement works, a finite element check was made on a specific piping intersection. That intersection was given a pressure that was the maximum the weakest pipe could accept without becoming overstressed. The entire assembly was pressurized. Although the individual pipe was not overstressed, the intersection was. That is case 1 in Fig. 5.3.

Case 2 was built with a conventional pad reinforcement. Each code has rules, albeit slightly different, that tell one how to design such a pad. As the chart in Fig. 5.3 shows, that brings the stresses down at the intersection. And because the rules of the code are followed, one actually doesn't have to compute the stresses. The solution is acceptable by definition.

Case 3 was constructed by choosing a higher-schedule pipe for the header pipe. This is the pipe that is weakened. One will note that with this method the assembly is not overstressed. It is essentially the same mathematical solution as for the pad in case 2. However, because the

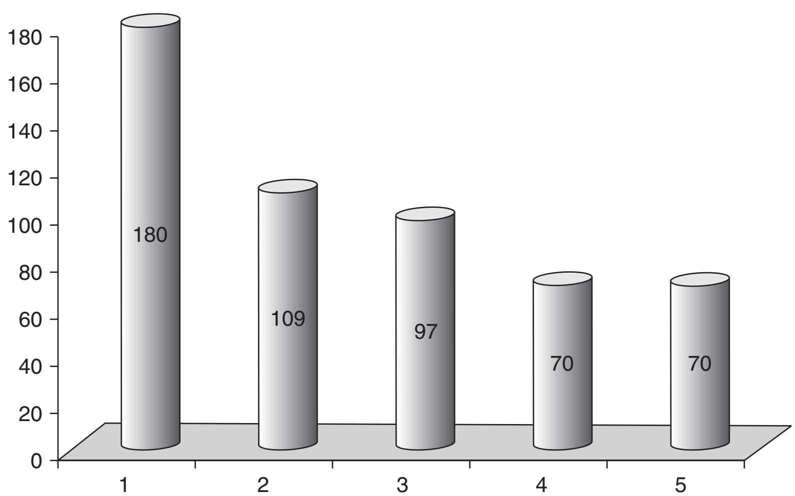


Figure 5.3 Stress level comparison by case.

pad is not a full encirclement or smooth pattern, it has slightly higher stresses which may be due to more local anomalies.

Cases 4 and 5 are models of weld-on and weld-in types of fitting. Figure 5.3 shows that with respect to pressure design, there are no differences in the effectiveness of the solution. The differences between the two are in other sections of analysis, and the decision of which solution to use is dependent on factors other than pressure design.

The actual figures of stress, etc., are not shown because they would be accurate only for a specific case. As such they are meaningless regarding actual numbers. The relationships of the various methods show that often the fitting-type solution gives the best stress pattern. It follows that cost and other design concerns would enter into the decision as to which actual solution to use.

The general way in which branch connections are proved is through the use of area replacement. *Area replacement* is a simplified mathematical technique to substantiate that there is sufficient reinforcement in the junction. Each of the codes has a slightly different set of area replacements. The paragraphs showing the specific rules for each code are listed in Table 5.9.

The pipeline codes for buried pipe are more restrictive regarding welded intersections. They have special rules which are discussed below. This is so primarily because they work with the higher-yield materials which are less ductile and therefore more prone to failure in the kinds of stress abnormalities present in such things as weld pads.

Code B31.3, Chapter IX, on high pressure, does not allow pad-type reinforcement. It requires insert-type fittings. These are the type depicted

TABLE 5.9 Paragraphs Giving Detailed Rules for Area Replacement

Code	Paragraph	Comment
B31.1	104.3.1 (D, E, F, G)	Both internal and external and welded and extruded.
B31.3	304.3.3 through 304.3.6, A304.3.1–2, K304.3.2–3	Some forms are not permitted in Chapters A and K.
B31.4	404.3.1	Some detailed differences between extruded and welded; see general discussion.
B31.5	504.3.1	Has some special rules for fittings from copper; also, there are different rules for welded and extruded.
B31.8	831.4 through 831.6, Appendices F, I, Figs. 12 and 13	Special rules as in B31.4.
B31.9	904.3.3, Fig. 904.3.3 A & B	Use B31.1 if needed; figures determine need.
B31.11	1104.3.1	Special rules as in B31.4 and B31.8.

TABLE 5.10 Typical Pipeline Rules for Reinforcement of Intersections

Ratio of design hoop stress to minimum specified yield strength in the header	Ratio of nominal branch diameter to nominal header diameter: 25% or less	More than 25% through 50%	More than 50%
20% or less	(g)	(g)	(h)
More than 20% through 50%	(d) (i)	(i)	(h) (i)
More than 50%	(c) (d) (e)	(b) (e)	(a) (e) (f)

NOTES:

(a) Smoothly contoured tees are preferred. Pads, partial saddles, or other localized reinforcement is prohibited.

(b) Smoothly contoured tees are preferred. It should be full encirclement but pads, partial saddles, or other localized reinforcement is allowed.

(c) It may be a full encirclement pad, saddle, or welding outlet. Edges should be tapered to the header thickness, and the legs of the fillet weld should not exceed that same thickness.

(d) No reinforcement calculations are required for openings of 2 in. or smaller, but for suitable protection, vibration and similar forces should be considered.

(e) Welds shall be equivalent to Appendix I, Figs. 12 and 13.

(f) Inside edges of opening are rounded to $\frac{1}{8}$-in. radius and tapers, as in note (c) above. All fillets shall be continuous.

(g) Reinforcement is not mandatory but may be required for over 100 psig and other circumstances.

(h) If the required reinforcement will extend more than halfway around the pipe, then full encirclement shall be used.

(i) Any type of reinforcement may be used.

in case 5 of Fig. 5.3, where the protection from pressure is essentially the same. Other requirements limit the use of case 3 and case 4 solutions.

The prohibition in Chapter IX of B31.3 is a more absolute version of the special rules in the pipeline codes. These special rules are related to the amount of hoop stress in the header and the ratio of the branch diameter to the header diameter. This ratio of branch to header diameter is often used in more rigorous analysis of intersections. As that ratio gets bigger, there is greater weakening and therefore greater need for reinforcement.

The specific rules of B31.8 are summarized here in Table 5.10. They are somewhat more restrictive than those in B31.4 and B31.11. As always for a specific instance, the actual code should be consulted.

Generic area replacement

Rather than show the specific diagrams in the body of the text, each diagram will be found in App. A. At this time, the generic area replacement will be discussed. Figure 5.4 is a generic area replacement diagram.

This generic replacement reduces to a simple set of three steps. It is the description, but each code and especially the differences in welded versus extruded procedures would require one to follow the details of the code.

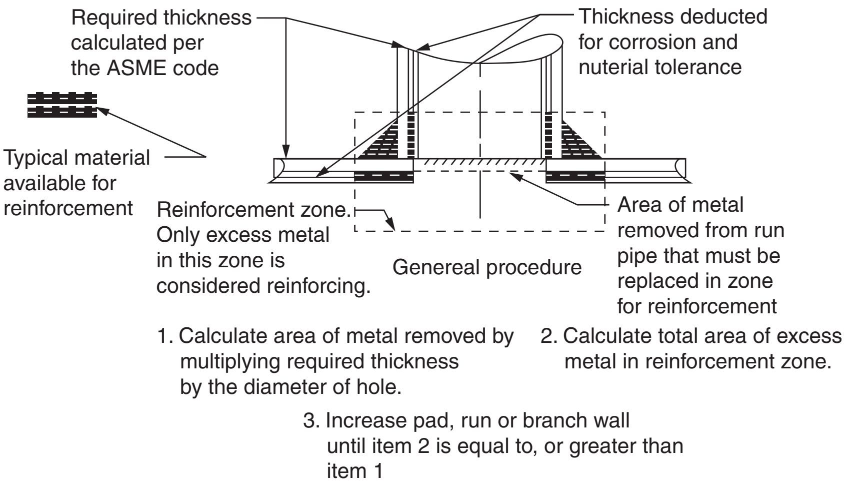


Figure 5.4 Generic area replacement diagram. (*Courtesy of HFI International Inc.*)

Flanges, closures, and blanks

The piping codes naturally use flanges in attaching pipe sections together. Each of the codes recognizes flange standards that are appropriate for its code section. These are discussed more thoroughly in Chap. 8 on such listed standards.

However, because there are so many uses for flanges, those standards cannot always cover the specific requirements for a flange. Often a special flange must be designed to meet the requirements of the project. The piping codes give rules for this design.

The most common flange design methodology is seen in App. II of Section VIII, Division 1. This is allowed as the base methodology.

TABLE 5.11 Paragraphs for Flange Design Requirements: Closures and Blanks

Code	Paragraph(s)	Comment
B31.1	104.4 (closures),104.5, 108	All subchapters, use of metric discussed
B31.3	304.4 (closures), 304.5 Main A and K, 308 Appendix F308.2–F312	Includes restrictions and prohibitions
B31.4	404.5, 408	All subchapters
B31.5	504.4 (closures), 504.5, 508.5	All subchapters
B31.8	831.2, 831.37 (closures)	All subchapters
B31.9	904.4 (closures), 904.5, 908	All subchapters
B31.11	1104.5, 1108	All subchapters

NOTE 1: Code B31.3 has an App. L for aluminum flanges which have no other U.S. standard.
NOTE 2: Code B31.8, App. I, includes a table giving dimensions for lightweight flanges that have a maximum pressure of 25 psig.

Exceptions, if deemed necessary to the procedure, are listed. Table 5.11 lists the paragraphs that spell out those requirements, if any exist.

For closures, the generally accepted procedures are spelled out in BPV Code, Section VIII, Division 1, Paragraphs UG 34. Each Code lists exceptions to the Section VIII rules as they might apply to the particular piping in that codes applications. Code B31.8 also recognizes that there could be closure heads attached to pipe. In that case, the closure rules would be the rules for heads in Section VIII, Division 1. Again, any special limitations are noted in the 831.373 sets of paragraphs.

Chapter

6

Flexibility Design

Brief History

Flexibility design is an important part of the design of piping systems. It is primarily utilized to handle thermal-type expansion in the B31 codes. The concept is relatively simple. As pipes are heated, they expand. As mentioned, when they are constrained, as they must be in some way in a piping system, stresses are developed.

No real piping system heats up only once and stays at a steady temperature. This results in changing stresses as it heats and cools. The changing stresses create a fatigue situation. *Fatigue* is the phenomenon whereby a structure, pipe in this case, will fail even though it does not ever see a stress that is above its static failure stress value.

The driving function of when this failure occurs basically is the number of times the stress cycles and how far it fluctuates. This failure mode was first noted by a Frenchman named J. V. Poncelet in the mid-1800s. Similar piping problems did not take precedence until the temperatures of piping began to climb with the advent of more powerful steam engines and the consequent need for higher temperatures and pressures.

One of the first papers on expansion of pipes in the United States was published in 1910. This coincides with the concerns being raised about boilers at that time. Since that time there have been many systems and methods to find the stresses involved in these increasingly complex systems.

The issue of flexibility is easy to explain in a simple system, one that is called a two-anchor system, say between a pump and a storage tank that has to go around some corner for layout reasons. Figure 6.1 shows the simple example of what happens to the pipe as it expands to some temperature.

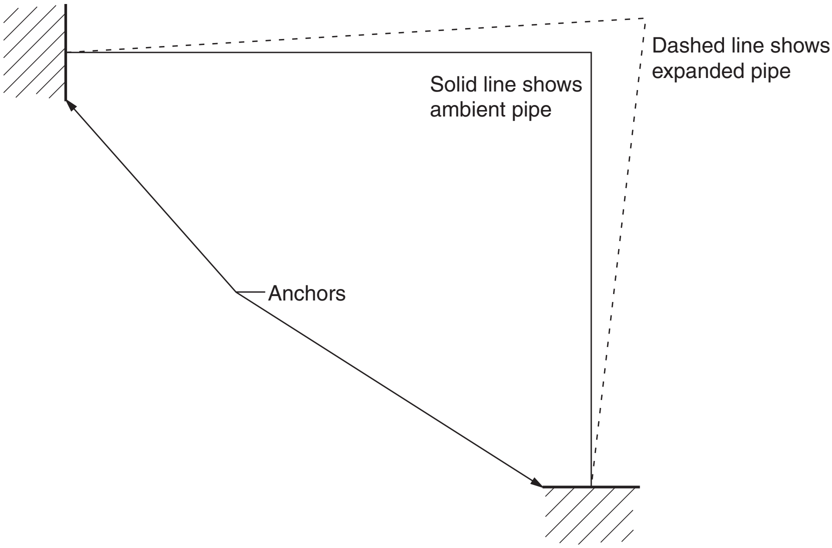


Figure 6.1 Simple two anchor flexibility example.

The business of flexibility analysis is to calculate the stresses that are allowed by a particular code for just such a system as this. Several methods have been utilized: charts, graphical analysis, simplified analysis, general analysis, mainframe computer analysis, and today's popular and almost universal personal computer (PC) analysis. Some of these others are still used.

To use many of these systems requires a great deal of skill and understanding of all the issues involved. The calculated stresses are not the same for every system, as they might be for pressure design. A discussion of the PC method is found later in this chapter.

Code requirements

The requirements of the two systems—aboveground and buried piping—are considerably different. The buried piping is basically in a stable-temperature environment when it is buried. Occasionally, the pipe has to come aboveground to pumping stations, valve stations, tanks, and related aboveground facilities. At those times, the piping can have more advanced flexibility needs.

The aboveground piping systems in general have extensive flexibility or fatigue problems. They also have many more complex piping routings and the attendant interactions as the pipes expand and contract. In many of the plants there are several different cycles. The temperature states, as well as the pressure states, may change frequently.

In any case, whether the pipe is buried or aboveground, the stress or piping engineer must have certain basics to work with. One of the first things to be developed is the number of cycles and the allowable stress.

One of the significant differences between flexibility analysis and pressure design is that flexibility analysis works on a stress range rather than on a specific stress. As noted earlier, these stresses are secondary-type stresses in that they are self-limiting. That is, the temperature change and expansion create the stresses. The allowable stress at higher temperature is usually not the same as that at lower temperature. One is called the cold allowable, denoted by S_c, and the other is the hot allowable, or S_h. Those stresses refer to the cycle under analysis.

The stress range S_a is computed from

$$S_a = f\,(1.25S_c + 0.25S_h)$$

f = stress range reduction factor discussed below

This formula is used and explained in all the codes except B31.4 and B31.11, where they define S_a differently and set a maximum range. B31.4 and B31.11 give a formula for longitudinal stress to calculate the amount of stress. Remember that these two codes limit the temperatures to which the code is applicable. At that limit there is no difference between hot and cold allowable specified minimum yield strength (SMYS).

Some of the codes—B31.1, B31.3, and B31.9 by reference to B31.1—allow an expansion of the stress range under certain conditions. Since B31.4 and B31.11 do not have that range concept available, expansion is not an option. B31.8 and B31.5 do not mention whether this option is available or not. In codes, if a particular action is not prohibited, one can generally make a case for it. The lack of mention in those two codes is most likely due to the fact that the option is seldom useful.

The option has to do with the longitudinal stresses created by sustained loads. General treatment of longitudinal stresses is discussed later in this chapter. In pressure design, the discussion centered on hoop stresses. These hoop stresses, from pressure, are twice the longitudinal stresses generated by pressure. However, longitudinal stresses can also result from the weight of pipe, contents, any insulation, attachments, and other similar loads. It is therefore possible to generate a longitudinal stress that is above an allowable static stress value. In such a case, that higher-than-allowable stress would be sufficient for the system not to be in compliance with the code.

It is somewhat more likely that the total, or additive, calculated longitudinal stresses are less than the static allowable value. If that is true in the hot condition, i.e., that the longitudinal stress in that condition is less than S_h, the difference may be added to the previously calculated S_a.

The two codes handle it differently. This difference may appear at first glance to give different results, but the difference is simply algebraic manipulation. They are presented as formulas in different paragraphs. The B31.3 formula (1b) is in Paragraph 302.3.5 and is $S_a = f\,[1.25\,(S_c + S_h) - S_L]$. The B31.1 formula (13a) is in Paragraph 104.8.3. It is for the expansion stress S_E and simply states that one should add the factor $f(S_h - S_L)$ to the previously calculated S_a. One can infer that if the expression within the parentheses is negative, S_a would be reduced. This would not be the case because if the expression becomes negative, the code is being violated.

Nevertheless, the additional S_a from the condition is the same in either instance. The apparent difficulty is pointed out as a reminder that the code must be read and interpreted carefully.

Table 6.1 lists the paragraphs that establish the acceptable requirements for all codes with respect to allowable amounts of stress in the flexibility and/or longitudinal calculations.

The stress range reduction factor f is defined and explained in the appropriate paragraphs. This factor is the means by which the codes recognize that the more cycles there are, the less of a stress range the pipe can take. The codes that find this factor applicable—primarily Codes B31.1, B31.3, and B31.5—give a method of calculating f to use in the stress reduction factor by a table, graph, or formula. That f formula is $f = 6.0(N)^{-0.2} \leq 1.0$, where N is some number of cycles from 7000 to 2 million. This range of cycles is under some effort to expand. Users are trying to utilize the concept for both less than 7000 cycles and something larger or other than thermal expansion stresses.

There are two fundamental types of cycles in any system. One is large-displacement and low-frequency. The other is small-displacement and high-frequency. The first is typical of thermal expansion cycles. The second is typical of vibration. Analysis by thermal techniques such as the code utilizes is not totally accurate for vibration. The code method is an analogous method that might be used under certain circumstances.

An example of that would be something like wave motion. It is not fast, as is the vibration excited by much mechanical equipment, but it has

TABLE 6.1 Code Paragraphs Establishing Allowable Flexibility Stresses

Code	Paragraph
B31.1	102.3.2, 104.8.3
B31.3	302.3.5, M302.3, K302.3, K308.4
B31.4	402.3.2, 419.6.4
B31.5	502.3.2, Fig. 502.3.5
B31.8	833.8
B31.9	B31.1, 102.3, 2902.3.2
B31.11	1119.6.4, 1102.3.2

many cycles. So the committees are moving toward allowing more cycles and a lower f factor.

The use of 7000 cycles for the factor of 1 is loosely based on an assumption of 1 thermal cycle per day for 20 years. Code B31.5 in Fig. 502.3.5 is refreshingly explicit. The number of cycles is set at 7300, and note 1 says it is based on the assumption of 365 days for 20 years. One puckishly wonders if refrigerators are aware of leap years.

Another interesting aspect is the answer to this question: What about the situation in which one has an accumulation of a different number of cycles with displacement stress ranges different from each other.

The formula given to calculate N requires that first all the stress ranges be calculated and then the maximum computed range S_E and its attendant N_E. All the other ranges less than the maximum are defined as S_i, and as an attendant N_iS_E is defined a little later.

The N formula is

$$N = N_E + \sum \left(\frac{S_i}{S_E} \right)^5 N_i$$

where $i = 1, 2, \ldots, n$. The reader will note that each code has a minimum f which is accepted. There are footnotes stating that lower numbers are the responsibility of the designer. That designer is also cautioned that at elevated temperatures, some materials may have reduced fatigue life.

When the allowable stress range can be computed, the actual stress analysis of the specific system can be completed.

The codes require a calculation for the longitudinal stresses. This stress is a stress that is hard to define precisely. The codes vary in their treatment of longitudinal stresses much more than in that of the hoop stress. While there is a difference in hoop treatment, it is basically all based on the same fundamental set of equations. As noted earlier, when the systems have similar design conditions, there is little differences in practical results. The differences in longitudinal stress calculations are far more sensitive to the difference in the application's sensitivity to those *sustained loadings*; as usual, the greater differences are between the aboveground piping and the buried piping.

The most direct definition of longitudinal stress can be found in Code B31.8. Its Chapter 833, "Design for Longitudinal Stress," is quite specific. It starts by giving guidance as to the difference between restrained and unrestrained piping. That is summarized very roughly thus: If the pipeline is in a situation or has a design that allows it to move around, it is unrestrained. Chapter 833.1(b) says piping in which soil or supports prevent axial displacement of flexure at bends is *restrained*. Chapter 833.1(c) says piping that is freed to displace axially or flex at bends is *unrestrained*.

The rest of that paragraph gives examples that will help in determining which piping is restrained and which isn't.

The code then gives specific formulas that define the components that make up longitudinal stress:

$$S_p = 0.3S_H$$

where S_H is hoop stress in restrained lines.

$$S_p = 0.5S_H$$

where S_H is hoop stress in unrestrained lines.

$$S_t = E\alpha(T_1 - T_2)$$

where E = elastic modulus at ambient
α = coefficient of thermal expansion
T_1 = temperature of installation, tie-in, or burial
T_2 = warmest or coldest operating temperature (note both warmer and colder need to be examined)

$$S_B = \frac{M}{Z}$$

where M = bending moment at the pipe cross section and Z = pipe's section modulus. If the cross section is at a fitting or component, the M used is M_R:

$$M_R = \sqrt{(0.75i_o M_o)^2 + (0.75i_i M_i)^2 + M_t^2}$$

where i = stress intensification factor, subscripts i and o mean in plane and out plane, and t is for torsion.

$$0.75\,i \geq 1$$

$$S_X = \frac{R}{A}$$

where A = pipe cross-sectional area and R = an external axial force. Then the longitudinal stress for restrained pipe is

$$S_L = S_P + S_T + S_X + S_B$$

The maximum value allowed is 0.95(SMYS) times the temperature reduction factor. Paragraphs 833.4 and 833.5 give procedures for modifiying the S_L calculation for restrained pipe under certain conditions where biaxial stress might be a factor.

The longitudinal stress for unrestrained pipe allows the dropping of the S_T term. The reader will recall that unrestrained pipe does not develop any stress from expansion. When this term is dropped, the maximum value drops to 0.75(SMYS) times the temperature reduction factor.

This is compared to the calculation of longitudinal stress in Codes B31.4 and B31.11 where the formula is much simpler. It is $S_L = E\alpha(T_1 - T_2) - \upsilon S_H$ for restrained lines. The formula for unrestrained lines will be discussed in the section on expansion stresses.

The aboveground lines have a little asymmetry in their makeup; also Code B31.5 does not address longitudinal stress. It apparently is not an issue in refrigeration. Code B31.9 defers to Code B31.1. It should be pointed out that the bulk of B31.9 piping does not require sophisticated analysis. When a building services system is one that the designer believes requires such analysis, for instance, a steam heat system, it is most often similar to those anticipated for B31.1.

This leaves us with Code B31.1 and Code B31.3 to discuss. The mystery seems to be why B31.3 doesn't publish a formula of any kind. This has been a source of debate in the B31.3 committee. At present the committee is considering a code case to provide such a formula. The current version will include an axial term and a torsion term which could be used. The code case approach is, as mentioned above, usable as soon as it has passed the appropriate voting of the committee, while the earliest a code version can be published would be in 2006.

The question is left as to what the designer should do. An inquiry to the committee would elicit a response that the code does not address that issue or the dreaded "this is consulting" response. Neither answers the real question. Many experienced designers use the formula from B31.1. Some will add the terms being considered in the code case as the situation demands. This is one reason the wise owner will choose an experienced designer.

This leaves the formula as published in B31.1 for the longitudinal stresses. This formula is found in Paragraph 104.8.1:

$$S_L = \frac{PD_o}{4t_n} + \frac{0.75iM_A}{Z} \leq 1.0S_H$$

Note: If one is using metric units, merely multiply the numerator of the second term by 1000 and of course use consistent units throughout that calculation.

In Codes B31.4 and B31.11, there is a specific prohibition to using the mechanical allowances that would be included in t_n, which is t nominal of the B31.1 equation. Code B31.1 by default takes the larger of the in-plane and out-plane stress intensification factors, so for their purposes there is only one. This is a simplification. The moment M_A is defined as the resultant moment which would include all moments regardless of their orientation.

The 1.0 multiplier for S_H becomes understandable when one reads further in the paragraph. If there is an occasional load, for instance, a wind load or a seismic load, the equation is modified by adding a third term which is the same as the second term except that the moment due to the occasional event is included. At that time, the multiplier for S_H becomes 1.15 for those where the event is no more than 8 hours at any one time and no more than 800 hours for any one year. It becomes 1.2 if the load acts for no more than 1 hour at any one time and for no more than 80 hours during a year.

Code B31.3 has a more complex method of handling this type of problem. If the limiting criteria are met, the designer can have a lower design temperature or pressure, leading to a higher initial allowable stress in the design. The paragraph in conjunction with App. V and the owner's concurrence may be invoked. We will not discuss this procedure further in this book.

Once the foregoing determinations have been made, one can proceed to the actual flexibility analysis. There are two other considerations to be taken first. In B31.4 and B31.11 and for most of the B31.8 piping, nothing more is done after the longitudinal stresses are computed and found to be within the maximum values.

The code gives a method of determining if further analysis needs to be done for all the codes except B31.4 and B31.11. The reader is reminded that Code B31.9 defers to Code B31.1. Those remaining codes—B31.1, B31.3, B31.5, and B31.8—give a set of criteria to determine if formal flexibility analysis is required.

That analysis involves a three-step process to make the determination:

1. Does it duplicate or replace a system without significant change to a system with a successful service record?
2. Can it be readily judged adequate by comparison to a previously analyzed system?
3. Is it a uniform size that has no more than two anchor points and no intermediate restraints and meets the requirements of the empirical limits of the equation below?

There is a general warning that the equation is empirical and no general proof can be given as to its accuracy or conservatism. Many limitations are listed in that warning, and it is recommended that the user read it in its entirety before using it. It can be found in Code B31.1 as 119.7.1; in B31.3 at 319.4.1; and in B31.8 at 833.7.

The formula is

$$\frac{Dy}{(L-U)^2} \leq K$$

where D = ouside diameter of pipe
y = resultant of total displacement strains to be absorbed
L = developed length of piping between anchors
U = straight-line distance between anchors

and

$$K = \begin{cases} 20{,}800\dfrac{S_A}{E_A} & \text{B31.3, metric} \\ 30\dfrac{S_A}{E_A} & \text{B31.3, USCS} \\ 0.03 & \text{B31.1, USCS} \\ 208.3 & \text{B31.1, metric} \\ 0.03 & \text{B31.8} \\ 30\dfrac{S_A}{E_c} & \text{B31.5} \end{cases}$$

The E modulus in B31.3 is at 21°C, and in B31.5, E_c is cold. All other terms are as defined in the particular code.

This might be best demonstrated by using the illustration in the brief history and adding dimensions. See Fig. 6.1 (on p. 62) and test the following:

10 NPS pipe, 10.75 dia. ASTM A-106 B
Temperature 70 to 800°F
Modulus is 29.7×10^6
S_c is 20,000 psi, S_h is 10,800 psi
Number of cycles is 7000

Thermal expansion α is 7.75 E^{-6}
Horizontal leg is 12 ft
Vertical leg is 13 ft

The calculation is as follows:

$$y = (12 + 13) \times 7.75E(10^{-6}) \times (800 - 70) = 0.141$$

$$L = 12 + 13 = 25$$

$$U = \sqrt{12^2 + 13^2} = 17.7$$

$$D = 10.75$$

$$S_A = 1 \times (1.25 \times 20{,}000 + 0.25 \times 10{,}800) = 25{,}002$$

$$\text{Target} = \frac{30 \times 25{,}002}{29.5\ E^6} = 0.0282$$

At that computation is 0.0284 it doesn't pass; but change the vertical leg to 14, and it passes as the computation changes to 0.0276.

This example shows the kinds of things that can be done as one thinks about the system. It is obvious that one would rarely have this simple an example, but one could extend the number of legs. The only real criteria are the two anchor points and the uniformity with no intermediate restraints. In theory, at least, one could go from anchor to anchor and make these determinations. There are other simplified methods one could use, as discussed. Some are more conservative and others, less. Using a different method, one could get different answers regarding whether to analyze.

At any rate, assuming one has determined by some means that more formal analysis is required, the analysis begins. In that analysis the designer looks at the expansion stresses developed at all points in the system and determines whether they pass or fail the S_A test.

Stress intensification

The piping codes use *stress intensification factors* (SIFs) to simplify the calculations. This is particularly useful when one is looking at small incidents or making decisions rather than modeling full systems. These factors were developed in the early 1950s by A. R. C. Markl and his team, including E. C. Rodabaugh at the Tube Turns facility in Kentucky.

SIFs may be one of the less understood factors in the piping codes. They are certainly different from the methods used in the Nuclear Piping Code and in BPV, Section VIII, Division 2, fatigue analysis. It is useful to explain them in detail.

First the team tested pieces of welded pipe and seamless pipe. They tested several pieces and analyzed the data. They determined that there was a significant difference in the cycles to failure of plain straight pipe and two pieces of pipe butt-welded together. A look at Fig. 6.2 will show that the difference at 40,000 psi was 100,000 cycles for plain pipe and butt-welded pipe. From those data they developed a formula for the S-N curve for the pipe. The formula that was developed was based on the butt-welded pipe, and the intensification for that pipe was arbitrarily set at 1. This is different from the polished bar tests used in Section VIII, Division 2; that line can be seen in the figure. The formula that was developed is

$$iS = 245{,}000N^{-0.2}$$

where i is the SIF, S is the stress, and N is the number of cycles to failure.

This formula is used today for testing and developing new SIFs or reconfirming old ones. A test machine similar to the one in Fig. 6.3 is used. The machine in the picture is set to check an out-plane fitting,

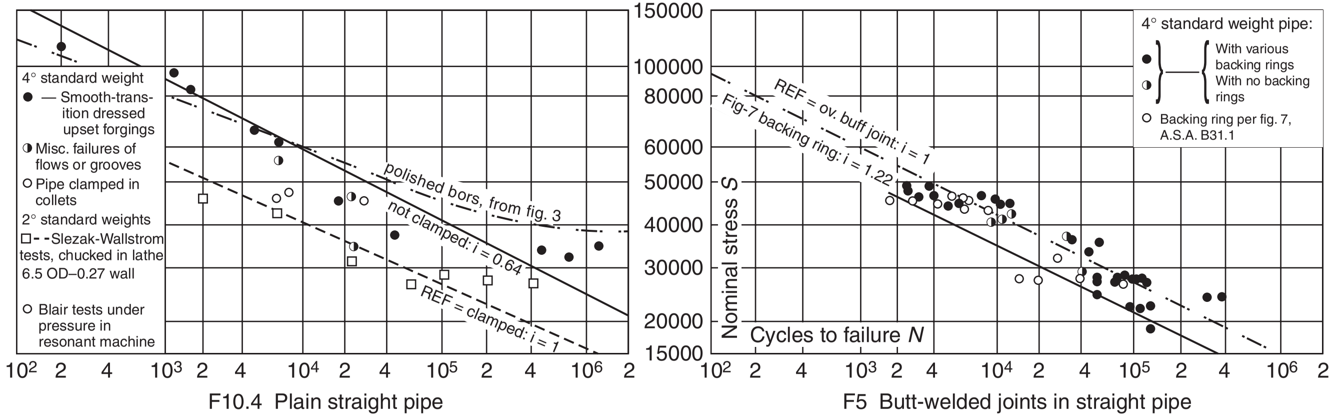


Figure 6.2 Fatigue curves of plain vs welded pipe compared.

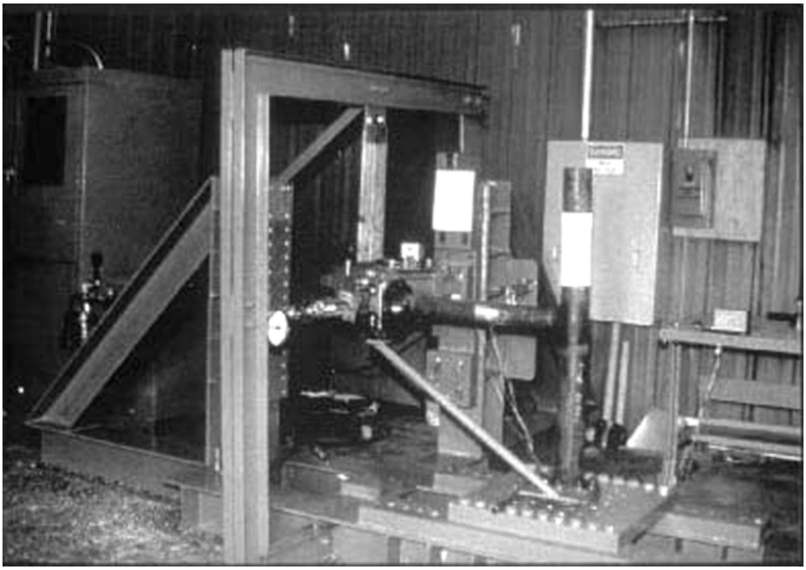


Figure 6.3 SIF test machine (*Courtesy WFI*).

because the hydraulic ram cycles the components transverse to the header, which is the vertical piece. To test for an in-plane factor, the hydraulic system is switched to cycle the components along the axis of the header or vertical pipe.

The test is a load deflection test. This means that for whatever is tested, a series of measured loads are applied, and the deflections caused by that load are recorded. Then the test technician sets the cycles to a certain deflection and counts the number of cycles to failure. A detailed description of the process can be found in the Weld Research Council Bulletin 392. It was written by Rodabaugh.

Once the test is complete, it is a matter of determining the factor i by mathematical substitution. And if $i > 1$, determine a formula that will produce that i for others to use. At present, the preferred form is to develop a flexibility characteristic and from that characteristic to compute an SIF.

The codes publish the SIF formula in their books. They are all basically built from the test data of Markl and from theoretical analysis. Most codes will allow the designer to use a different formula than the one they list, provided there is objective evidence. The designer is allowed to use analogy for geometries that are not in those listings. Some manufacturers of fittings have tested their products and can offer those objective data.

As was noted, Code B31.1 shows in its listing only the higher of the two, in-plane or out-plane, intensification factors. Code B31.3 shows both. Usually the out-plane factor is higher. Table 6.2 gives the paragraph or appendix for each code that lists the SIFs that it uses.

TABLE 6.2 SIFs by Code

Code	Location
B31.1	Appendix D
B31.3	Appendix D
B31.4	Table 419.6.4
B31.5	Table 519.3.6
B31.8	Appendix E
B31.9	Defers to Code B31.1
B31.11	Figure 1119.6.4(c)

The Code B31.1 appendix is reproduced here as Fig. 6.4, as it has the most complete set of factors. As mentioned, B31.3 does give both in-plane and out-plane factors. The other codes do not include as many different geometries. Similar geometries have similar SIFs. There are several notes to these factor charts. It is highly advisable to read the notes as they do give some very specific warnings, prohibitions, and additional information.

Expansion stress

The only item left in the calculation of stress for the flexibility analysis is the expansion stress or the stress due to expansion. This stress is calculated at several cross sections throughout the piping system. The most obvious higher stresses will be at intersections, changes in direction, restraints, and anchor points.

The formula for the expansion stress is

$$S_E = \sqrt{S_b^2 + 4S_t^2}$$

where the S_b is the resultant bending stress, which has the formula

$$S_b = \sqrt{\frac{(i_i M_i)^2 + (i_o M_o)^2}{Z}}$$

where S_E = expansion stress
S_t = torsional stress = $\frac{M_t}{2Z}$
S_b = resultant bending stress
i, o = in-plane or out-plane
i = stress intensification factor
M = moment
Z = section modulus

TABLE D-1
FLEXIBILITY AND STRESS INTENSIFICATION FACTORS

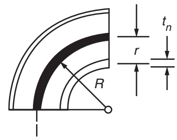
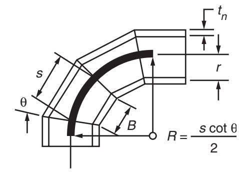
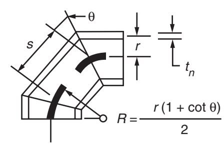
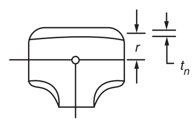
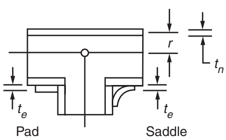
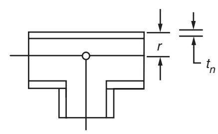
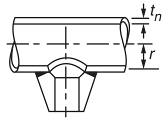
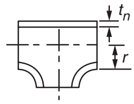
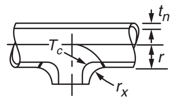
Description	Flexibility Characteristic h	Flexibility Factor k	Stress Intensification Factor i	Sketch
Welding elbow or pipe bend [Notes (1), (2), (3), (9), (13)]	$\dfrac{t_n R}{r^2}$	$\dfrac{1.65}{h}$	$\dfrac{0.9}{h^{2/3}}$	
Closely spaced miter bend [Notes (1), (2), (3), (13)] $s < r(1 + \tan\theta)$ $B \geq 6\ t_n$ $\theta \leq 22\tfrac{1}{2}$ deg.	$\dfrac{s t_n \cot\theta}{2r^2}$	$\dfrac{1.52}{h^{5/6}}$	$\dfrac{0.9}{h^{2/3}}$	
Widely spaced miter bend [Notes (1), (2), (4), (13)] $s \geq r(1 + \tan\theta)$ $\theta \leq 22\tfrac{1}{2}$ deg.	$\dfrac{t_n (1 + \cot\theta)}{2r}$	$\dfrac{1.52}{h^{5/6}}$	$\dfrac{0.9}{h^{2/3}}$	
Welding tee per ASME B16.9 [Notes (1), (2), (10)]	$\dfrac{4.4t_n}{r}$	1	$\dfrac{0.9}{h^{2/3}}$	
Reinforced fabricated tee [Notes (1), (2), (5), (10)]	$\dfrac{\left(t_n + \dfrac{t_e}{2}\right)^{5/2}}{r(t_n)^{3/2}}$	1	$\dfrac{0.9}{h^{2/3}}$	
Unreinforced fabricated tee [Notes (1), (2), (10)]	$\dfrac{t_n}{r}$	1	$\dfrac{0.9}{h^{2/3}}$	

Figure 6.4 An example of Code SIF tables from Appendix D of B31.1.

TABLE D-1
FLEXIBILITY AND STRESS INTENSIFICATION FACTORS (CONT'D)

Description	Flexibility Characteristic h	Flexibility Factor k	Stress Intensification Factor i	Sketch
Branch welded-on fitting (integrally reinforced) per MSS SP-97 [Notes (1), (2)]	$\frac{3.3t_n}{r}$	1	$\frac{0.9}{h^{2/3}}$	
Extruded outlet meeting the requirements of para. 104.3.1(G) [Notes (1), (2)]	$\frac{t_n}{r}$	1	$\frac{0.9}{h^{2/3}}$	
Welded-in contour insert with $r_x \geq D_{ob}/8$, $T_c \geq 1.5\ t_n$ [Notes (1), (2)]	$4.4\ \frac{t_n}{r}$	1	$\frac{0.9}{h^{2/3}}$	

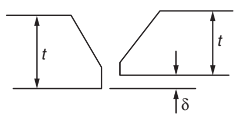
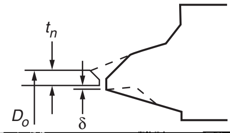
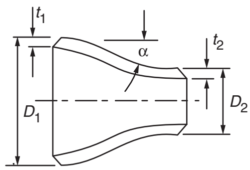
Description	Flexibility Factor k	Stress Intensification Factor i	Sketch
Branch connection [Notes (1), (6)]	1	For checking branch end $1.5\left(\frac{R_m}{t_{nh}}\right)^{2/3}\left(\frac{r'_m}{R_m}\right)^{1/2}\left(\frac{t_{nb}}{t_{nh}}\right)\left(\frac{r'_m}{r_p}\right)$	See Fig. D-1
Butt weld [Note (1)] $t \geq 0.237$ in., $\delta_{max} \leq \frac{1}{16}$ in., and $\delta_{avg}/t \leq 0.13$	1	1.0 [Note (12)]	
Butt weld [Note (1)] $t \geq 0.237$ in., $\delta_{max} \leq \frac{1}{8}$ in., and δ_{avg}/t = any value	1	1.9 max. or $[0.9 + 2.7(\delta_{avg}/t)]$, but not less than 1.0 [Note (12)]	
Butt weld [Note (1)] $t \geq 0.237$ in., $\delta_{max} \leq \frac{1}{16}$ in., and $\delta_{avg}/t \leq 0.33$	1		
Fillet welds	1	2.1; or 1.3 for fillet welds as defined in Note (11)	See Figs. 127.4.4(A), 127.4.4(B), and 127.4.4(C)

Figure 6.4 (*Continued*)

TABLE D-1
FLEXIBILITY AND STRESS INTENSIFICATION FACTORS (CONT'D)

Description	Flexibility Factor k	Stress Intensification Factor i	Sketch
Tapered transition per para. 127.4.2(B) and ASME B16.25 [Note (1)]	1	1.9 max. or $1.3 + 0.0036\frac{D_o}{t_n} + 3.6\frac{\delta}{t_n}$	
Concentric reducer per ASME B16.9 [Note (7)]	1	2.0 max. or $0.5 + 0.01\alpha\left(\frac{D_2}{t_2}\right)^{1/2}$	
Threaded pipe joint, or threaded flange	1	2.3	...
Corrugated straight pipe, or corrugated or creased bend [Note (8)]	5	2.5	...

NOTES:

(1) The following nomenclature applies to Table D-1:

- B = length of miter segment at crotch, in. (mm)
- D_o = outside diameter, in.
- D_{ob} = outside diameter of branch, in. (mm)
- R = bend radius of elbow or pipe bend, in. (mm)
- r = mean radius of pipe, in. (mm) (matching pipe for tees)
- r_x = external crotch radius of welded-in contour inserts, in. (mm)
- s = miter spacing at center line, in. (mm)
- T_c = crotch thickness of welded-in contour inserts, in. (mm)
- t_n = nominal wall thickness of pipe, in. (mm) (matching pipe for tees)
- t_r = reinforcement pad or saddle thickness, in. (mm)
- α = reducer cone angle, deg.
- δ = mismatch, in. (mm)
- θ = one-half angle between adjacent miter axes, deg.

(2) The flexibility factors k and stress intensification factors i in Table D-1 apply to bending in any plane for fittings and shall in no case be taken less than unity. Both factors apply over the effective arc length (shown by heavy center lines in the sketches) for curved and miter elbows, and to the intersection point for tees. The values of k and i can be read directly from Chart D-1 by entering with the characteristic h computed from the formulas given.

(3) Where flanges are attached to one or both ends, the values of k and i in Table D-1 shall be multiplied by the factor c given below, which can be read directly from Chart D-2, entering with the computed h: one end flanged, $c = h^{1/6}$; both ends flanged, $c = h^{1/3}$.

(4) Also includes single miter joints.

(5) When $t_c > 1.5t_n$, $h = 4.05t_n/r$.

(6) The equation applies only if the following conditions are met.

  (*a*) The reinforcement area requirements of para. 104.3 are met.

  (*b*) The axis of the branch pipe is normal to the surface of run pipe wall.

  (*c*) For branch connections in a pipe, the arc distance measured between the centers of adjacent branches along the surface of the run pipe is not less than three times the sum of their inside radii in the longitudinal direction or is not less than two times the sum of their radii along the circumference of the run pipe.

  (*d*) The inside corner radius r_1 (see Fig. D-1) is between 10% and 50% of t_{nh}.

  (*e*) The outer radius r_2 (see Fig. D-1) is not less than the larger of $T_b/2$, $(T_b + y)/2$ [shown in Fig. D-1 sketch (c)], or $t_{nh}/2$.

  (*f*) The outer radius r_3 (see Fig. D-1) is not less than the larger of:

    (1) $0.002\theta d_o$;

    (2) $2(\sin\theta)^3$ times the offset for the configurations shown in Fig. D-1 sketches (a) and (b).

  (*g*) $R_m/t_{nh} \leq 50$ and $rt_m/R_m \leq 0.5$.

(7) The equation applies only if the following conditions are met:

  (*a*) Cone angle α does not exceed 60 deg., and the reducer is concentric.

  (*b*) The larger of D_1/t_1 and D_2/t_2 does not exceed 100.

  (*c*) The wall thickness is not less than t_1 throughout the body of the reducer, except in and immediately adjacent to the cylindrical portion on the small end, where the thickness shall not be less than t_2.

(8) Factors shown apply to bending; flexibility factor for torsion equals 0.9.

(9) The designer is cautioned that cast butt welding elbows may have considerably heavier walls than those of the pipe with which they are used. Large errors may be introduced unless the effect of these greater thicknesses is considered.

(10) The stress intensification factors in the Table were obtained from tests on full size outlet connections. For less than full size outlets, the full size values should be used until more applicable values are developed.

Figure 6.4 (*Continued*)

TABLE D-1 (CONT'D)

NOTES (CONT'D):

(11) A stress intensification factor of 1.3 may be used for socket weld fitting if toe weld blends smoothly with no undercut in pipe wall as shown in the concave, unequal leg fillet weld of Fig. 127.4.4(A).

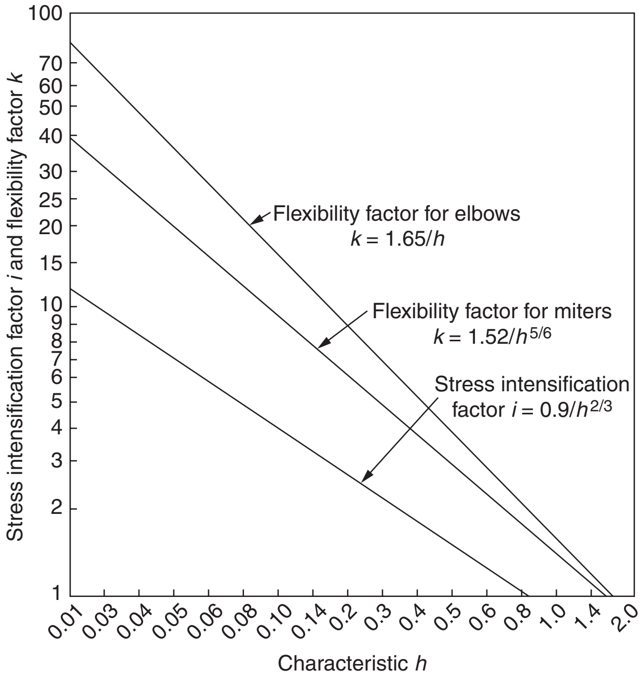
(12) The stress intensification factors apply to girth butt welds between two items for which the wall thicknesses are between $0.875t$ and $1.10t$ for an axial distance of $\sqrt{D_o t}$. D_o and t are nominal outside diameter and nominal wall thickness, respectively. δ_{avg} is the average mismatch or offset.

(13) In large diameter thin-wall elbows and bends, pressure can significantly affect magnitudes of k and i. Values from the table may be corrected by dividing k by

$$\left[1 + 6\left(\frac{P}{E_c}\right)\left(\frac{r}{t_n}\right)^{7/3}\left(\frac{R}{r}\right)^{1/3}\right]$$

and dividing i by

$$\left[1 + 3.25\left(\frac{P}{E_c}\right)\left(\frac{r}{t_n}\right)^{5/2}\left(\frac{R}{r}\right)^{2/3}\right]$$



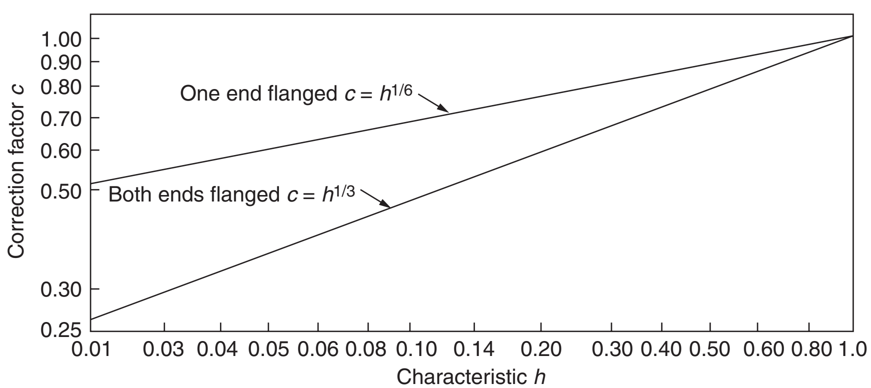


CHART D-2 CORRECTION FACTOR c

Figure 6.4 (*Continued*)

The B31.1 methodology is somewhat more complex in that it only uses one SIF and applies it to the resultant moments, which are defined as $\sqrt{M_x^2 + M_y^2 + M_z^2}$. Code B31.1 has a somewhat more complex method of identifying the effective modulus in a branch, and it is explained in detail in the appropriate paragraph, all of which is listed in Table 6.3.

For guidance, the codes, especially the aboveground codes, give you sketches to show where and how the moments relate to the piping. Those are duplicated here for the reader to determine. The rules are that the locations at the pipe intersection or at the change of direction are where the stresses should be checked. See Fig. 6.5.

Now that this is all defined, what are the moments? The codes are silent on the method of calculating those moments and stresses; they do point out that sometimes it is also important to check the movement. Since pipe might bump into some structure, methods are given for checking some reactions at restraints and anchor spots, but no other methodology is offered.

Throughout this chapter we have pointed out that many methods have been tried. Fortunately for the analyst today, there are many PC programs that do all the "dirty" work. Unfortunately, the analyst still has to know what he or she is doing regarding the input data.

The codes themselves are beginning to give some direction as to how these computer programs might work. The 2004 edition Code B31.3 has a new appendix that is a simple piping problem run on several popular computer programs. The results are not to be taken as precise.

A sketch and discussion of that code Appendix are duplicated here with discussion and to show what information is needed for these programs and what one might expect. In addition, the same problem is run as a B31.1 or B31.3 problem with slightly different results due to the idiosyncrasies of the various programs. These results are courtesy of Coade; their program is Caesar II. There are many programs that do the same kind of analysis; each will have slightly different methods. The author has experience with Caesar II; the problem was run on several

TABLE 6.3 Expansion and Bending Stress Methodology Locations

Code	Location
B31.1	104.8.3,104.8.4
B31.3	319.4.4
B31.4	419.6.4
B31.5	519.4.5
B31.8	833.7, 833.8
B31.9	Defers to B31.1
B31.11	1119.6.4, 1119.7

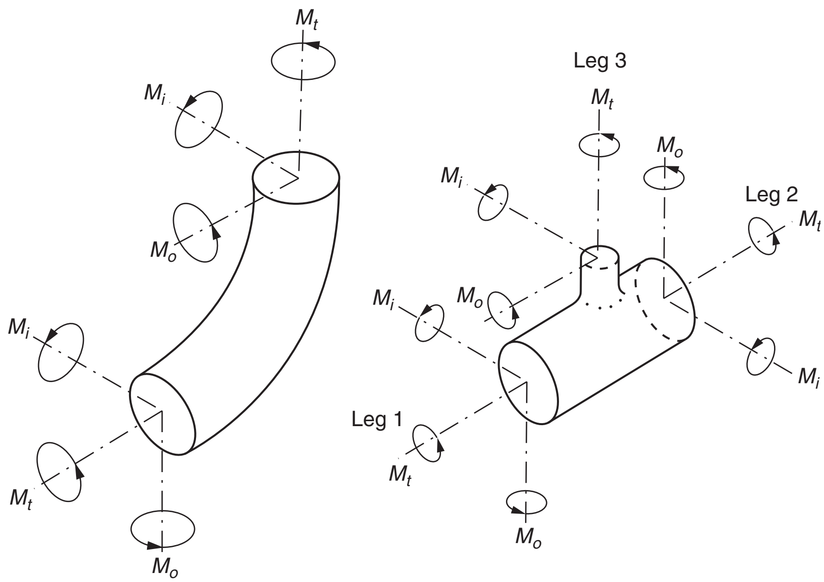


Figure 6.5 Typical moment to pipe relationships.

different programs for B31.3, and all the results were within the accuracy required.

As stated, the problem is a very simple Z arrangement, as shown in Fig. 6.6. The input for the program is duplicated here. One will note the use of node numbers. These are used to tell the computer that things are the same between node numbers, so it treats that section equally.

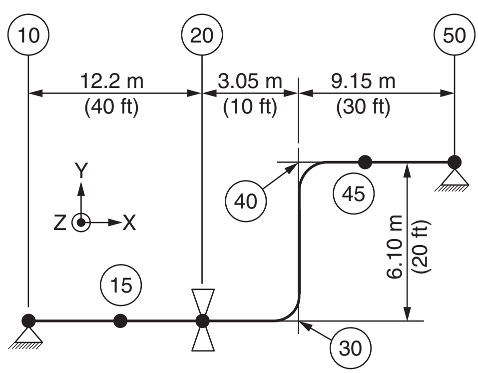


Figure 6.6 S301.1 Simple Code compliant model.

From 10 To 15 DX = 20.000 ft.

PIPE

Dia = 16.000 in. Wall = .375 in. Insul = 5.000 in. Cor = .0625 in.

GENERAL

T1 = 500 F T2 = 30 F P1 = 500.0000 lb./sq.in. Mat = (106)A106 B
E = 29,500,000 lb./sq.in. v = .292 for (B31.1 v = .300) Density = .2830 lb./cu.in.
Insul = .0064 lb./cu.in. Fluid = .0361111 lb./cu.in.

RESTRAINTS

Node 10 ANC

ALLOWABLE STRESSES

B31.3 (2002)	Sc = 20,000 lb./sq.in.	Sh1 = 18,900 lb./sq.in.
Sh2 = 20,000 lb./sq.in.	Sh3 = 20,000 lb./sq.in.	Sh4 = 20,000 lb./sq.in.
Sh5 = 20,000 lb./sq.in.	Sh6 = 20,000 lb./sq.in.	Sh7 = 20,000 lb./sq.in.
Sh8 = 20,000 lb./sq.in.	Sh9 = 20,000 lb./sq.in.	

ALLOWABLE STRESSES

B31.1 (2003)	Sc = 15,000 lb./sq.in.	Sh1 = 15,000 lb./sq.in.
Sh2 = 15,000 lb./sq.in.	Sh3 = 15,000 lb./sq.in.	Sh4 = 15,000 lb./sq.in.
Sh5 = 15,000 lb./sq.in.	Sh6 = 15,000 lb./sq.in.	Sh7 = 15,000 lb./sq.in.
Sh8 = 15,000 lb./sq.in.	Sh9 = 15,000 lb./sq.in.	

From 15 To 20 DX = 20.000 ft.

RESTRAINTS

Node 20 Y

From 20 To 30 DX = 10.000 ft.

BEND at “TO” end

Radius = 24.000 in. (LONG) Bend Angle = 90.000 Angle/Node @1 = 45.00 29
Angle/Node @2 = .00 28

From 30 To 40 DY = 20.000 ft.

BEND at “TO” end

Radius = 24.000 in. (LONG) Bend Angle = 90.000 Angle/Node @1 = 45.00 39
Angle/Node @2 = .00 38

From 40 To 45 DX = 10.000 ft.
From 45 To 50 DX = 20.000 ft.

RESTRAINTS

Node 50 ANC

The careful reader will note that for B31.3 the Poisson ratio (v) is slightly different. B31.1 uses 0.3 which is a common simplification for that ratio. B31.3 uses the more accurate ratio for this specific material. The Poisson ratio is the relationship of the shrinkage (or expansion) in the transverse direction when one pushes or pulls on the longitudinal direction.

The reader will also note the change in allowable stresses for the same conditions. This has been discussed above. The more conservative allowable is the choice of each committee for their section.

Next is a set of what Caesar II calls the setup parameters. These are switches that the user sets inside the program that give the program direction regarding what to do as it progresses through the analysis. They could have considerable effect. These are basically the user's options.

How to determine what the actual settings should be is beyond the scope of this description. However, one will be pointed out to give some idea of what sorts of things the user might need to determine. There is a setting that asks, Are the tees B16.9 tees? This is because if a tee is fully in compliance with B16.9 and has the proper crotch thickness and transition radius, the SIF for that tee is lower than for any other type of tee. The training or user manual of the particular program that is being used would explain in detail what these settings mean and give guidance as to how to set them. For brevity the B31.3 parameters will be shown here.

SETUP FILE PARAMETERS

CONNECT GEOMETRY THRU CNODES =	YES
MIN ALLOWED BEND ANGLE =	5.00000
MAX ALLOWED BEND ANGLE =	95.0000
BEND LENGTH ATTACHMENT PERCENT =	1.00000
MIN ANGLE TO ADJACENT BEND PT =	5.00000
LOOP CLOSURE TOLERANCE =	1.00000 in.
THERMAL BOWING HORZ TOLERANCE =	0.100000E–03
AUTO NODE NUMBER INCREMENT =	10.0000
Z AXIS UP =	NO

USE PRESSURE STIFFENING =	DEFAULT
ALPHA TOLERANCE =	0.500000E–01
RESLD-FORCE =	NO
HGR DEF RESWGT STIF =	0.100000E+13 lb./in.
DECOMP SNG TOL =	0.100000E+11
BEND AXIAL SHAPE =	YES
FRICT STIF =	0.100000E+07 lb./in.
FRICT NORM FORCE VAR =	0.150000
FRICT ANGLE VAR =	15.0000
FRICT SLIDE MULT =	1.00000
ROD TOLERANCE =	1.00000
ROD INC =	2.00000
INCORE NUMERICAL CHECK =	NO
OUTCORE NUMERICAL CHECK =	NO
DEFAULT TRANS RESTRAINT STIFF =	0.100000E+13 lb./in.
DEFAULT ROT RESTRAINT STIFF =	0.100000E+13 in.lb./deg
IGNORE SPRING HANGER STIFFNESS =	NO
MISSING MASS ZPA =	EXTRACTED
MIN WALL MILL TOLERANCE =	12.5000
WRC-107 VERSION =	MAR 79 1B1/2B1
WRC-107 INTERPOLATION =	INTERACTIVE
DEFAULT AMBIENT TEMPERATURE =	70.0000 F
BOURDON PRESSURE =	NONE
COEFFICIENT OF FRICTION (MU) =	0.000000
INCLUDE SPRG STIF IN HGR OPE =	NO
REDUCED INTERSECTION =	B31.1(POST1980)
USE WRC329	NO
NO REDUCED SIF FOR RFT AND WLT	NO
B31.1 REDUCED Z FIX =	YES
CLASS 1 BRANCH FLEX	NO
ALL STRESS CASES CORRODED =	NO
ADD TORSION IN SL STRESS =	DEFAULT
ADD F/A IN STRESS =	DEFAULT
OCCASIONAL LOAD FACTOR =	0.000000
DEFAULT CODE =	B31.3
B31.3 SUS CASE SIF FACTOR =	0.750000
ALLOW USERS BEND SIF =	NO

USE SCHNEIDER	NO
YIELD CRITERION STRESS =	VON MISES
USE PD/4T	NO
BASE HOOP STRESS ON ? =	ID
APPLY B318 NOTE2 =	NO
LIBERAL EXPANSION ALLOWABLE =	NO
STRESS STIFFENING DUE TO PRESSURE =	NONE
B31.3 WELDING/CONTOUR TEE IS B16.9	NO
PRESSURE VARIATION IN EXP CASE =	DEFAULT
USE FRP SIF =	YES
USE FRP FLEX =	YES
BS 7159 Pressure Stiffening =	Design Strain
FRP Property Data File =	CAESAR.FRP
FRP Emod (axial) =	0.320000E+07 lb./sq.in.
FRP Ratio Gmod/Emod (axial) =	0.250000
FRP Ea/Eh*Vh/a =	0.152727
FRP Laminate Type =	THREE
FRP Alpha =	12.0000 F
FRP Density =	0.600000E–01 lb./cu.in.
EXCLUDE f2 FROM UKOOA BENDING =	NO

This program also gives the user control of certain execution parameters and they are shown next.

EXECUTION CONTROL PARAMETERS

Rigid/ExpJt Print Flag...............	1.000
Bourdon Option..........................	.000
Loop Closure Flag	.000
Thermal Bowing Delta Temp.....	.000 F
Liberal Allowable Flag...............	.000
Uniform Load Option.................	.000
Pressure Stiffening Option.........	.000
Ambient Temperature................	70.000 F
Plastic (FRP) Alpha....................	12.000
Plastic (FRP) GMOD/EMODa....	.250
Plastic (FRP) Laminate Type.....	3.000
Eqn Optimizer............................	.000

Node Selection....................	.000
Eqn Ordering......................	.000
Collins................................	.000
Degree Determination........	.000
User Eqn Control..............	.000

The next set of data is computed data by the program. The first data are the bend SIFs. This illustrates one of the differences between B31.3 and B31.1. As discussed, B31.1 uses the highest SIF of in-plane and/or out-plane. This can be expected to affect the expansion stresses in the final analysis. Both sets are shown here. The *K* values are flexibility factors used in calculating the SIFs; see the appendices.

B31.3

BEND SIF & FLEXIBILITY VALUES

SIFs IN/OUT of Plane

Flexibilities IN/OUT of plane

BEND	TYPE	SIFi	SIFo	Ki	Ko
30. 0	Flanges	2.61997	2.18331	9.50719	9.50719
40. 0	Flanges	2.61997	2.18331	9.50719	9.50719

B31.1

BEND SIF & FLEXIBILITY VALUES

SIFs IN/OUT of Plane

Flexibilities IN/OUT of plane

BEND	TYPE	SIFi	SIFo	Ki	Ko
30.0	Flanges	2.61997	2.61997	9.50719	9.50719
40.0	Flanges	2.61997	2.61997	9.50719	9.50719

As one would expect, the pipe properties are the same for both codes as the pipe is the same. These include things like the weight of insulation, fluid, and pipe. Also the *y* factor used in calculating hoop stress is calculated from the table as shown in the appendix and the various books; since they are the same, only one is shown.

PIPE PROPERTIES #1

FROM	TO	PIPE WT	INSUL WT	FLUID WT	TB ALPHA1	TB ALPHA2	TB ALPHA3	y
		WEIGHTS (lb./ft.)			THERMAL BOWING EXP (in./in./F)			
10.	15.	62.545	25.198	79.146	0.000E+00	0.000E+00	0.000E+00	.400
15.	20.	62.545	25.198	79.146	0.000E+00	0.000E+00	0.000E+00	.400
20.	30.	62.545	25.198	79.146	0.000E+00	0.000E+00	0.000E+00	.400
30.	40.	62.545	25.198	79.146	0.000E+00	0.000E+00	0.000E+00	.400
40.	45.	62.545	25.198	79.146	0.000E+00	0.000E+00	0.000E+00	.400
45.	50.	62.545	25.198	79.146	0.000E+00	0.000E+00	0.000E+00	.400

From this, the weight and center of gravities of the system are calculated. The reader will note that this program allows the inclusion of refractory. When this is used, it is usually internal to protect a carbon pipe from very high temperatures.

CENTER OF GRAVITY REPORT

	Total Wght (lb.)	X cg (ft.)	Y cg (ft.)	Z cg (ft.)
Pipe	6147.1	41.9	8.0	0.0
Insulation	2476.6	41.9	8.0	0.0
Refractory	0.0	0.0	0.0	0.0
Fluid	7778.7	41.9	8.0	0.0
Pipe+Insl+Refrty	8623.7	41.9	8.0	0.0
Pipe+Fluid	13,925.8	41.9	8.0	0.0
Pipe+Insl+Refrty+Fluid	16,402.4	41.9	8.0	0.0

The thermal coefficients are calculated from internal tables to the program. These are calculated by section. Now there is a little difference between B31.1 and B31.3 for the thermal coefficients. Each book has an appendix giving the thermal properties. The BPV Code also gives thermal coefficients. Most of the discrepancy is due to the fact that when a change is made in the BPV, Section II, which is the major control of material properties for all of ASME, it takes time before each piping code section can agree with the change.

The differences will make a difference in the calculated amount of expansion and therefore in the calculated stresses. This is an example of the fact that such things as these coeffiecients are not precise and one would find different ones at different sources. In point of fact the coefficients change with the relative temperatures. Most are based on a reference temperature of 70°F in USCS and 20°C in metric units.

There are only two temperatures in this sample problem so, only two coefficients are needed for each nodal section. Both code coefficients are given here as they are different.

THERMAL EXPANSION (in./in.)

B31.1		250°	20°	B31.3		250°	20°
10.	15.	0.0031	–0.0003	10	15.	0.0030	–0.0002
15.	20.	0.0031	–0.0003	15.	20.	0.0030	–0.0002
20.	30.	0.0031	–0.0003	20.	30.	0.0030	–0.0002
30.	40.	0.0031	–0.0003	30.	40.	0.0030	–0.0002
40.	45.	0.0031	–0.0003	40.	45.	0.0030	–0.0002
45.	50.	0.0031	–0.0003	45.	50.	0.0030	–0.0002

The run calculates the loads, forces and moments on the restraints and anchors by what is known as a load case. There are three in this problem. They are:

CASE 1 (OPE) W+T1+P1

CASE 2 (OPE) W+T2+P1

CASE 3 (SUS) W+P1

where OPE is an operating case and the letters signify as follows: W = weight, T1 and T2 are temperatures, P1 is pressure, and SUS is sustained. The reader will note that there are differences in both the loads and the moments. These are due to those differences in the codes.

B31.1

10		Rigid ANC					
1	OPE	–6169.	–2867.	0.	0.	0.	–16,004.
2	OPE	–297.	–2601.	0.	0.	0.	–12,470.
3	SUS	–735.	–2621.	0.	0.	0.	–12,734.
20		Rigid Y					
1	OPE	0.	–14,328.	0.	0.	0.	0.
2	OPE	0.	–10,067.	0.	0.	0.	0.
3	SUS	0.	–10,385.	0.	0.	0.	0.
50		Rigid ANC					
1	OPE	6169.	793.	0.	0.	0.	–37,532.
2	OPE	297.	–3734.	0.	0.	0.	33,224.
3	SUS	735.	–3396.	0.	0.	0.	27,940.

B31.3

10		Rigid ANC					
1	OPE	–5961.	–2858.	0.	0.	0.	–15,878.
2	OPE	–321.	–2602.	0.	0.	0.	–12,485.
3	SUS	–735.	–2621.	0.	0.	0.	–12,734.

20		Rigid Y					
1	OPE	0.	–14,177.	0.	0.	0.	0.
2	OPE	0.	–10,085.	0.	0.	0.	0.
3	SUS	0.	–10,385.	0.	0.	0.	0.
50		Rigid ANC					
1	OPE	5961.	632.	0.	0.	0.	–35,026.
2	OPE	321.	–3715.	0.	0.	0.	32,929.
3	SUS	735.	–3396.	0.	0.	0.	27,939.

The computer then tells you whether this stress check passed or didn't pass the code requirements. The reader will remember that the two codes have different acceptance criteria. So both will be shown here.

The reader will also note that the hoop stresses are different for the two codes. Since they use the same formula to calculate this stress, the question is, Why? The program uses the minimum wall in B31.3 and nominal wall in B31.1.

****CODE STRESS CHECK PASSED

PIPING CODE: B31.1 –2001, September 30, 2003

HIGHEST STRESSES: (lb./sq.in.)

CODE STRESS %:	80.2	@NODE	20
STRESS:	12,026.8	ALLOWABLE:	15,000.
BENDING STRESS:	7068.3	@NODE	20
TORSIONAL STRESS:	0.0	@NODE	29
AXIAL STRESS:	4918.6	@NODE	15
HOOP STRESS:	10,161.1	@NODE	15
VON MISES STRESS:	6581.0	@NODE	20

****CODE STRESS CHECK PASSED

PIPING CODE: B31.3 –2002, April 30, 2002

HIGHEST STRESSES: (lb./sq.in.)

CODE STRESS %:	76.0	@NODE	20
STRESS:	14,357.4	ALLOWABLE:	18,900.
BENDING STRESS:	8381.6	@NODE	20
TORSIONAL STRESS:	0.0	@NODE	30
AXIAL STRESS:	5975.7	@NODE	15
HOOP STRESS:	12,291.9	@NODE	15
VON MISES STRESS:	6554.1	@NODE	20

The next thing the program does is calculate the stresses for case 3, sustained: this is what is known as the expansion stress and has some interesting results. Because the allowable stresses are different, the results are different.

The careful reader will note that even though there are higher stresses in the B31.1 calculation method due to the differences mentioned, it was the difference in allowable stresses between the two codes that caused the failure call in B31.1. It was not the slight difference in parameters. This can be taken as an example of the fact that thought must be given to the results and the numerical comparisons.

One might expect that there are many more results that could be reported in this experimental run and comparison. They would be right. But the purpose of the example was to give an understanding of the flexibility design process, and that purpose has been met.

**** CODE STRESS CHECK FAILED

PIPING CODE: B31.1 -2001, September 30, 2003

HIGHEST STRESSES: (lb./sq.in.)

CODE STRESS %:	117.2	@NODE	39
STRESS:	26,371.8	ALLOWABLE:	22,500.
BENDING STRESS:	26,371.8	@NODE	39
TORSIONAL STRESS:	0.0	@NODE	40
AXIAL STRESS:	399.3	@NODE	29
HOOP STRESS:	0.0	@NODE	15
VON MISES STRESS:	14966.6	@NODE	39

****CODE STRESS CHECK PASSED

PIPING CODE: B31.3 -2002, April 30, 2002

HIGHEST STRESSES: (lb./sq.in.)

CODE STRESS %:	85.2	@NODE	39
STRESS:	25,326.9	ALLOWABLE:	29,725.
BENDING STRESS:	25,326.9	@NODE	39
TORSIONAL STRESS:	0.0	@NODE	39
AXIAL STRESS:	383.4	@NODE	29
HOOP STRESS:	0.0	@NODE	15
VON MISES STRESS:	14,373.6	@NODE	39

As one might expect, there are differences when one is calculating similar stresses in buried piping. The B31.1 has a nonmandatory appendix, Appendix VII, which goes into some detail for those differences. That appendix also gives references to other books from which the outlined procedure was drawn.

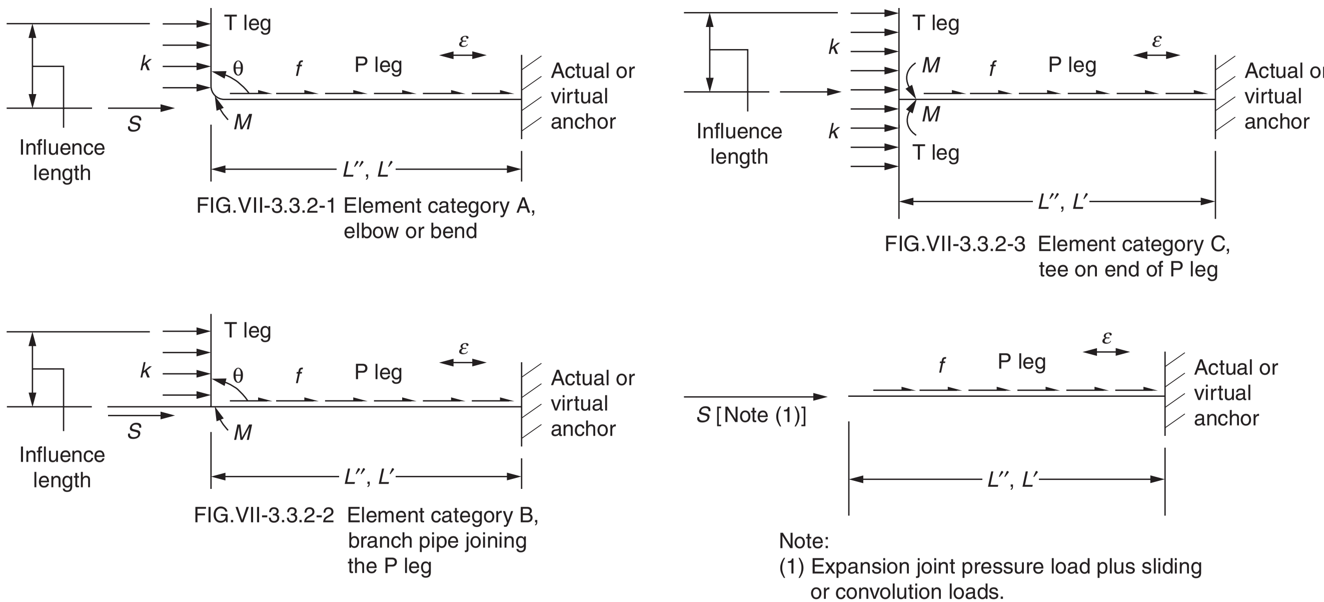


Figure 6.7 B31.1 Figures of element in flexibility of buried piping.

As the appendix points out, most of the B31.1 code is "written for piping suspended in open space and supported at local points," whereas the buried pipe is "supported, confined, and restrained continuously by the passive effects of the backfill and the trench bedding." So they wrote the nonmandatory section to show one way to handle the different problem. It should be pointed out that some programs do not use the methods as described in this appendix.

The appendix mentions that all pipe stress computer programs that have the buried piping options require the following factors to be input in addition to the ones mentioned previously in the example problem:

- Location of the virtual anchor
- The soil spring rate
- Influence length

The *virtual anchor* is defined as "a point or region along the axis of a buried pipe where there is no relative motion at the point pipe/soil interface." The soil spring rate is calculated using the modulus of subgrade reaction, which is the "rate of change of soil bearing stress with respect to compressive deformation of the soil," and the *influence length* is "that portion of a transverse pipe run which is deflected or 'influenced' by pipe thermal expansion along the axis of the longitudinal run."

The appendix suggests that the way to model the buried pipe is to divide the pipe into elements of convenient length and to impose a transverse spring at the center of each element which simulates the passive soil resistance. And it further suggests using the program's user manual for guidance.

Figure 6.7 shows you the basic categories A, B, C and D of pipe elements that are recognized as useful by this methodology. It further develops three subcategories of these basic categories which are based on the opposite end of the element from the one being analyzed.

The appendix establishes a series of formulas and worked examples which can guide the user in developing the three additional factors for the analysis. This includes a method of determining the element lengths and number of elements. It points out that the allowable stress in buried pipe may be higher that that which would be allowed in aboveground or open piping. That formula is given as $S_c =< S_A + S_h$, where the components have the definition given for open piping.

The reader interested in becoming better acquainted with this method is referred to the appendix for the details and example problems.

Chapter

7

Pipe Support

Buried Piping

As noted in the Chap. 6 on flexibility, there is a vast difference between the types of support engendered in buried piping and in aboveground piping. One must be reminded that there is aboveground piping in the pipelines. The pipelines have to come up at some point in time or space. They must have pumping or compressor stations, they must have valves, and they must start and stop.

Note that one of the better-known pipelines has significant portions aboveground—the Alaskan pipeline. This is so because in transporting crude oil, as it does from Prudhoe to Valdez, the pipe itself can become relatively warm. The incoming oil is above 100°F (38°C). Many of the areas that the pipeline crosses are substantially permafrost. The heated pipe would melt the permafrost and cause the pipeline to sink. Unique heat-transfer radiators and supports were developed to solve this problem, but they do not fall within the scope of this book.

Buried pipe requirements

The pipeline codes have specific requirements for the burial of pipe. The major requirement has to do with the amount of cover or fill over the pipe. That requirement is listed in tables duplicated here. Codes B31.4 and B31.11 have the same requirements. These tables are different in that one lists the metric requirements in millimeters (B31.11) and the other (B31.4) lists the metric requirements in meters. Only the millimeter table is duplicated; see Fig. 7.1.

Code B31.8 has a more complex table, shown in Fig. 7.2. One will recall that gas transmission lines are frequently found in dense population areas, and the table reflects a more conservative cover requirement for those areas.

Table 1134.6(a) Minimum Cover for Buried Pipelines

Location	For Normal Excavation, in. (mm)	For Rock Excavation Requiring Blasting or Removal by Equivalent Means, in. (mm)
Industrial, commercial, and residential areas	36 (914)	24 (610)
River and stream crossings	48 (1 219)	18 (457)
Drainage ditches at roadways and railroads	36 (914)	24 (610)
Any other area	30 (762)	18 (457)

Figure 7.1 Minimum cover per B31.4 & 11.

While this requirement addresses cover above the pipe, there is no mention of maximum ditch depth. There are cautions regarding that as well as provisions for situations in which the minimum coverage cannot be achieved. A newly installed pipeline is required to give minimum clearance between other underground structures. That clearance can be as close as 2 in. from drainage tile to a minimum of 12 in. in most cases; the specific paragraphs for these requirements are listed in Table 7.1.

For the oil and slurry pipelines, a series of general statements discuss the width of the ditch; e.g., the depth shall be below the ordinary level of cultivation. Code B31.8 has much greater discussion. The possibility of catastrophic damage has been well documented by past accidents on those gas lines, so the rules are more specific. There is a requirement for casings when the lines go under railroads or other vehicular passages such as roads.

In all cases, the codes specify that if the minimum cover cannot be met, then additional protection must be provided. In Paragraph 841.145,

	Cover, in.		
		For Rock Excavation [Note (1)]	
Location	For Normal Excavation	Pipe Size NPS 20 and Smaller	Pipe Size Larger Than NPS 20
Class 1	24	12	18
Class 2	30	18	18
Classes 3 and 4	30	24	24
Drainage ditch at public roads and railroad crossings (all locations)	36	24	24

NOTE:
(1) Rock excavation is excavation that requires blasting.

Figure 7.2 Minimum cover per B31.8.

TABLE 7.1 Paragraphs Covering Ditching for Buried Pipe

Code	Paragraphs
B31.4	434.6
B31.11	1134.6
B31.8	841.13, 841.14 subparagraphs 141 through 145, 862.117, 862.218

Code B31.8 identifies some acceptable types of requirements, including a list of additional guidance from other sources. Those additional elements of protection and sources of further guidance are listed here for convenience.

- API RP 1102, *Steel Pipelines Crossing Railroads and Highways*
- GRI report 91/0284, "Guidelines for Pipelines Crossing Highways"
- Gas Piping Technology Committee, Appendix G15
- Concrete or steel barrier above the pipe
- Concrete slab on each side extending above the pipe
- Damage-resistant coating, e.g., concrete
- Extra depth of ditch
- High-visibility warning tape
- Casing
- Heavier wall thickness than required by design factor F

Periodic surveillance and patrol of the pipeline is required as well as an educational program to teach the public ways to recognize and handle emergencies. These programs are used for notification regarding impending excavation in the area.

Note that the *Code of Federal Regulations*, Part 193, also controls pipelines. In general, these regulations follow the pipeline codes. They do have precedence in case of conflict and should be followed, as should all jurisdictional requirements.

Aboveground piping support

All piping must be supported. There are pipe support criteria for the aboveground standards. Basically the three pipeline codes do not address such support; they are written for the buried piping discussed above. Support rules are diverse among the four aboveground codes. All have support sections in varying degrees of complexity.

A common starting point is that the support has to be attached to a structure, frame, etc. Each designer/engineer must ensure that the structure is able to absorb the loads that the pipe will impose on it and that the support device is able to transmit that load properly.

Pipe support is an integral part of piping design and, in fact, flexibility analysis. Any experienced stress analyst will explain that relocating the supports and changing the type of support will significantly affect the stresses developed by the system. Code B31.3 implies in Paragraph 321.2 that the location and design of those elements may be based on engineering judgment and simple calculations. The fact remains that they are a part of the analysis, especially when one is using the computer programs that are now so common.

Along with reference to the appropriate paragraphs for each code, some of the more relevant points may be discussed. A good starting point may be the list of objectives set out in Code B31.3. There is a similar but shorter list in Code B31.9. The objectives of the layout and design are to prevent these occurrences:

- Piping stresses in excess of those permitted by the code
- Leakage at joints
- Excessive loads and moments to connected equipment
- Excessive stresses in the supporting elements
- Resonance with any vibration
- Excessive interference with expansion or contraction otherwise sufficient
- Unintentional disengagement from the support
- Excessive sagging in piping
- Excessive distortion or sagging of piping subject to creep
- Excessive heat flow subjecting supporting elements to temperatures outside their design limits

The materials used in the supporting elements should be those of the code in question or compatible with those materials. There are specific limits to the allowable stresses that are imposed on the supports in both the operation and the hydrotesting of the system.

Cast iron is generally acceptable for compression-loaded situations. However, impact or similar loads are discouraged. Nodular or malleable iron is somewhat less restricted. And in B31.3, wood or other material is acceptable provided it has appropriate design considerations pertaining to flammability, durability, and such relevant conditions.

Where corrosion is a consideration, coating protection is allowed. Temperatures that will be transmitted to the support are limited. Code B31.1 requires that the design of the elements be in accordance with MSS SP-58, *Pipe Hangers and Supports: Materials, Design and Manufacture*. This is considered the best available standard on the subject worldwide. The Manufacturers Standardization Society (MSS) has other pipe hangar Standard Practices that are also useful. MSS SP-69, Pipe Hangers and Supports: Selection and Application, has recently been approved as an American National Standard by the American National Standards Institute (ANSI).

The uninitiated or even the somewhat experienced piper can be amazed at the number of different types of supports that exist as more or less standard. The basic categories are rigid, variable supports, constant force, and guides. Figure 7.3 is a sample of the types of supports regularly employed for the appropriate situations. It comes from the aforementioned SP-69. This figure shows 59 generic types of hangers.

Once one has selected the pipe hangers, routed the pipe around the various obstacles, and added any loops or other flexibility considerations, one needs to locate the hangers. In general, one would like to add as few hangers as possible to meet all the requirements in the objectives. The primary consideration is the excessive sag and/or slope that would occur with piping supported by hangers placed too far apart.

It is possible to calculate these for each piping system. One would need to calculate the weights of the completed and operating system. In addition to the weight of the pipe, there would be insulation, if any, the process fluid, and other equipment such as valves. Recall that this is also done in calculating the piping stresses. Once that is done the typical method of calculating the sag between hangers would be to treat the section of pipe between the supports as a simple beam with a uniform load.

There are at least three considerations in this calculation:

- The deflection or sag experienced by the simulated beam
- The force that will be applied to the supports
- Which end conditions apply for the calculation

The forces are relatively simple, since it is a simple beam. That formula is

$$F = \frac{WL}{2}$$

where W = weight of piping per linear inch
L = length of pipe, in.
Z = section modulus of pipe (used later in stress calculations)
∂ = deflection, the result of stress calculation (see Table 7.2)

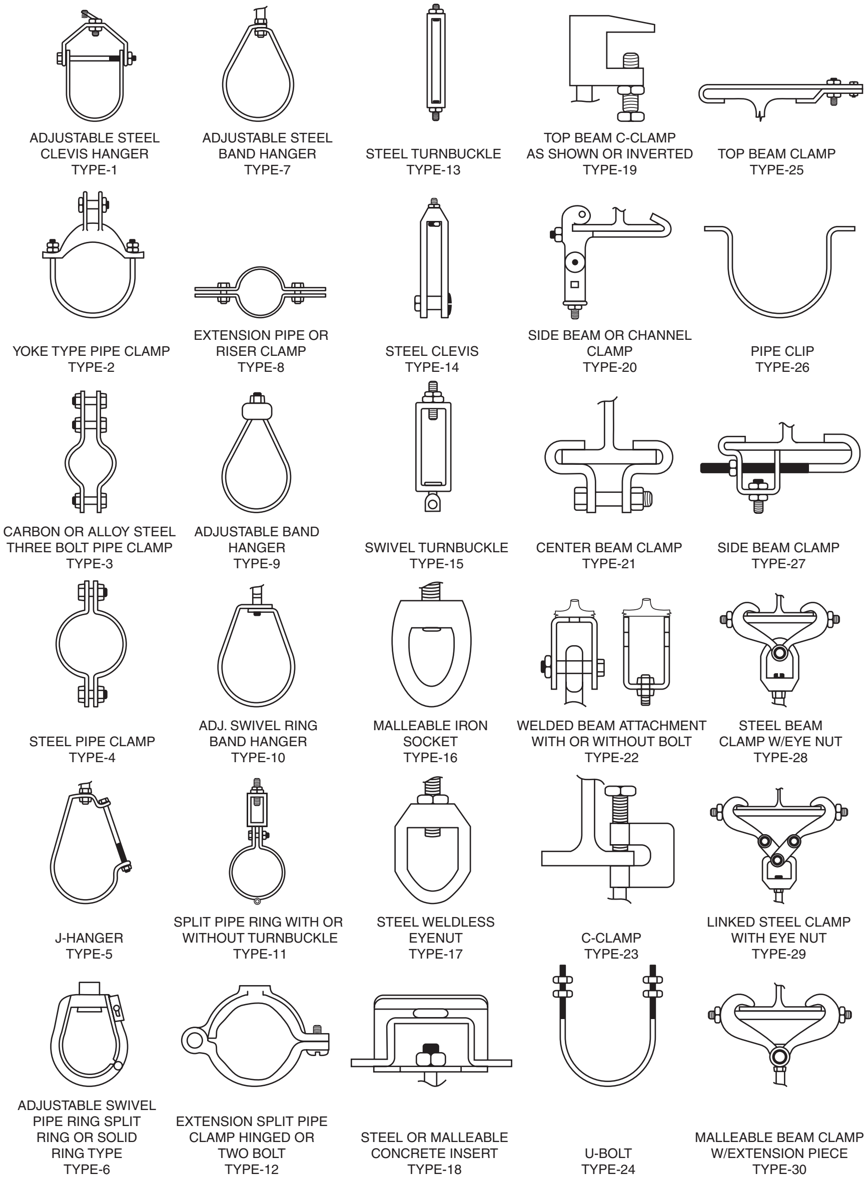


Figure 7.3 Types of supports and hangers. (*From ANSI/MSS, Standard Practice 69*).

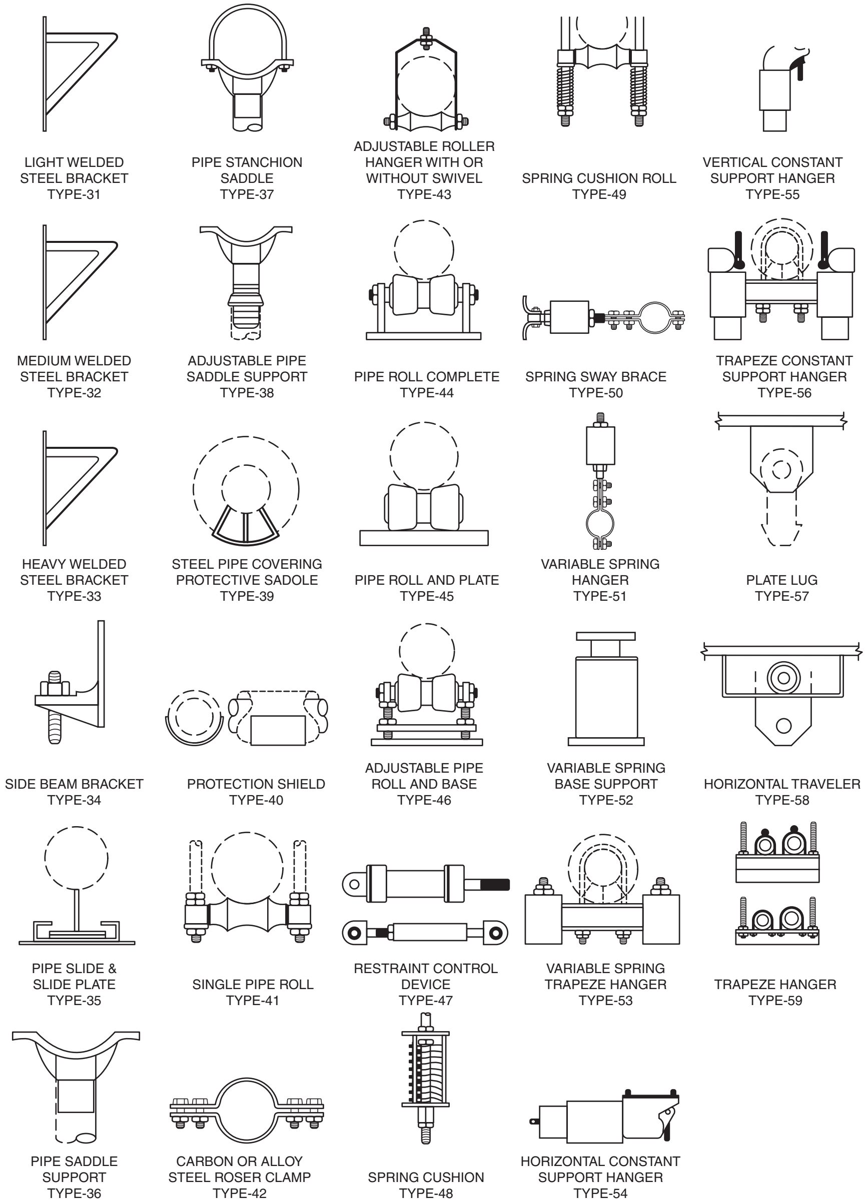


Figure 7.3 (*Continued*)

The end connections need to be considered before the stresses can be determined. In a simple beam situation, two conditions are reasonable to consider for the simplified calculations we are discussing. One is the simply supported case. The other is the fixed case where the pipe is considered rigidly clamped in such a way that the ends cannot move. The real situation is very rarely either, so it is quite common to average the two. Table 7.2 shows the formulas by type.

Note that the deflections can be calculated by other means. In most cases this would be done by the computer program where the hangers and types are the input data for the calculations. This type of consideration is to make the preliminary determination of location. Many specifications or simplifications have a fixed maximum deflection. It is a simple algebraic manipulation of the formula to determine an approximate length for the portion of the system under consideration. That manipulation results in the following formula, which is based on the hybrid formula.

$$L = \sqrt{\frac{10ZS}{W}}$$

The additional symbol S is for the allowable stress that is chosen for the conditions and the material.

Most pipe systems do not consist solely of horizontal runs. There are guidelines to follow that may help in more complicated situations:

- Place supports as close as possible to concentrated loads such as valves or other heavy components.
- Reduce the span lengths where directional changes occur.
- If the run is vertical, the supporting structure is the determining factor in placement and a support should be placed on the upper portion of the vertical run, to lessen the chances of buckling.

TABLE 7.2 Formulas for Pipe as Simple Beam, Deflection

Type of end	Formula
Simply supported	$\partial = \frac{WL^2}{8Z}$
Fixed-end support	$\partial = \frac{WL^2}{12Z}$
Hybrid approximation	$\partial = \frac{WL^2}{10Z}$

Even so, the problem of location of supports could be quite labor-intensive to calculate for a system of any extent. Therefore many span tables have been developed to reduce the computation. There is a table in Code B31.1 that covers certain sizes of pipe with certain contents. A similar but more extensive table appears in MSS SP-69. That table is duplicated here as Fig. 7.4. Note that this table is for standard-weight pipe and vapor service. Heavier pipe or adding insulation or changing stresses or limiting deflection would change the span. One can only say that it is a starting place.

Code B31.9 is a code that has a certain penchant for making simplifying charts. The reader will recall the charts in Appendix A for area replacement. Figure 7.5 is a chart from the B31.9 series of charts which show that code's span charts for thermoplastic and drawn copper tubing. These are items not found in the B31.1 or Fig. 7.4.

Code B31.9 covers many buildings and therefore goes into much greater detail about the pipe support design. Note that the buildings have considerable pipe that would have vertical runs, and they tend to have unique problems. They could have many more attachments of pipe supports to concrete so they have requirements for concrete and embedded studs.

Many of the pipe supports would have what are called *hanger rods*, in which the attachment is over the pipe and the weight must be supported by those rods. Typically these are threaded rods, and the requirements include calculation of the strength of the rods or threads. Code B31.1 gives the most extensive table of the carrying capacities of these rods, and that table is duplicated here for convenience (see Fig. 7.6). The general rule is that the strength is calculated on the root strength. The table gives the capacity for a specific material rod. It also gives the root area so that users can calculate the capacity of a different material, given its allowable stress.

The design and placement of pipe supporting elements is learned through experience rather than based on scientific procedures. As one gains experience, the supports can be placed more accurately. Following the general guidelines given in the codes and this book can help tremendously. The specific paragraphs by code are shown in Table 7.3. Only the aboveground codes are listed.

TABLE 7.3 General Guidelines per Code

Code	Paragraphs
B31.1	121, all subparagraphs 121.1 through 121.9
B31.1	321, all subparagraphs 321.1 through 321.4, A321, M321, MA321, K321 and all subparagraphs
B31.5	520, all subparagraphs
	521, all subparagraphs
B31.9	920, all subparagraphs
	921, all subparagraphs

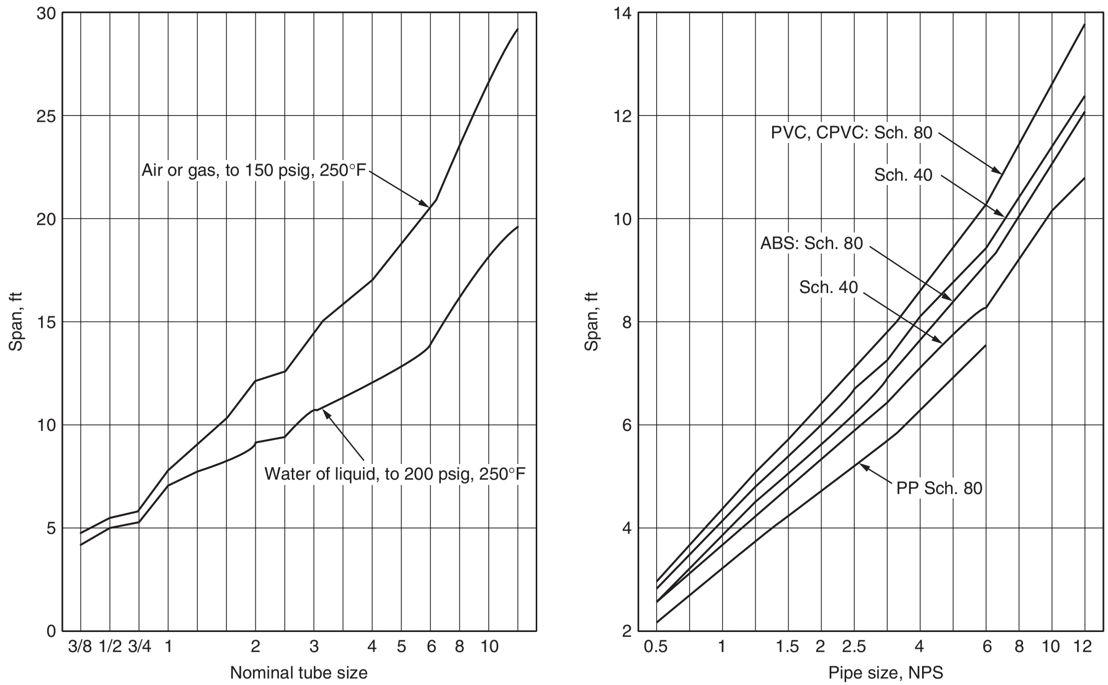
TABLE 3 - Maximum Horizontal Pipe Hanger and Support Spacing

NOMINAL PIPE OR TUBE SIZE		1 STD WT STEEL PIPE WATER SERVICE		2 STD WT STEEL PIPE VAPOR SERVICE		3 COPPER TUBE WATER SERVICE		4 COPPER TUBE VAPOR SERVICE	
in.	mm	ft	m	ft	m	ft	m	ft	m
1/4	(6)					5	1.5	5	1.5
3/8	(10)	7	2.1	8	2.4	5	1.5	6	1.8
1/2	(15)	7	2.1	8	2.4	5	1.5	6	1.8
3/4	(20)	7	2.1	9	2.7	5	1.5	7	2.1
1	(25)	7	2.1	9	2.7	6	1.8	8	2.4
1 1/4	(32)	7	2.1	9	2.7	7	2.1	9	2.7
1 1/2	(40)	9	2.7	12	3.7	8	2.4	10	3.0
2	(50)	10	3.0	13	4.0	8	2.4	11	3.4
2 1/2	(65)	11	3.4	14	4.3	9	2.7	13	4.0
3	(80)	12	3.7	15	4.6	10	3.0	14	4.3
3 1/2	(90)	13	4.0	16	4.9	11	3.4	15	4.6
4	(100)	14	4.3	17	5.2	12	3.7	16	4.9
5	(125)	16	4.9	19	5.8	13	4.0	18	5.5
6	(150)	17	5.2	21	6.4	14	4.3	20	6.1
8	(200)	19	5.8	24	7.3	16	4.9	23	7.0
10	(250)	22	6.7	26	7.9	18	5.5	25	7.6
12	(300)	23	7.0	30	9.1	19	5.8	28	8.5
14	(350)	25	7.6	32	9.8				
16	(400)	27	8.2	35	10.7				
18	(450)	28	8.5	37	11.3				
20	(500)	30	9.1	39	11.9				
24	(600)	32	9.8	42	12.8				
30	(750)	33	10.1	44	13.4				

Column	Material	Spacing
5	FIRE PROTECTION	FOLLOW REQUIREMENTS OF THE NATIONAL FIRE PROTECTION ASSOCIATION. SEE SECTION 14.
6	DUCTILE IRON PIPE	20 ft (6.1 m) MAX SPACING; MIN OF ONE (1) HANGER PER PIPE SECTION CLOSE TO THE JOINT BEHIND THE BELL AND AT CHANGE OF DIRECTION AND BRANCH CONNECTIONS. FOR PIPE SIZES SIX (6) INCHES (150 mm) AND UNDER, INSTALLED ON ASME B31 PROJECTS, THAT ARE SUBJECTED TO LOADINGS OTHER THAN WEIGHT OF PIPE AND CONTENTS, THE SPAN SHOULD BE LIMITED TO THE MAXIMUM SPACING FOR WATER SERVICE STEEL PIPE.
7	CAST IRON SOIL	10 ft (3.0 m) MAX SPACING; MIN OF ONE (1) HANGER PER PIPE SECTION CLOSE TO JOINT ON THE BARREL. ALSO AT CHANGE OF DIRECTION AND BRANCH CONNECTIONS.
8	GLASS	8 ft (2.4 m) MAX SPACING, FOLLOW PIPE MANUFACTURER'S RECOMMENDATIONS SEE SECTION 17.
9	PLASTIC	FOLLOW PIPE MANUFACTURER'S RECOMMENDATIONS FOR MATERIAL AND SERVICE CONDITION. SEE SECTION 18.
10	FIBERGLASS REINFORCED	FOLLOW PIPE MANUFACTURER'S RECOMMENDATIONS FOR MATERIAL AND SERVICE CONDITION. SEE SECTION 19.

NOTE: (1) FOR SPACING SUPPORTS INCORPORATING TYPE 40 SHIELDS, SEE TABLE 5.
(2) DOES NOT APPLY WHERE SPAN CALCULATIONS ARE MADE OR WHERE THERE ARE CONCENTRATED LOADS BETWEEN SUPPORTS SUCH AS FLANGES, VALVES, SPECIALTIES, ETC., OR CHANGES IN DIRECTION REQUIRING ADDITIONAL SUPPORTS.
(3) UNBALANCED FORCES OF HYDROSTATIC OR HYDRODYNAMIC ORIGIN (THRUST FORCES) UNLESS RESTRAINED EXTERNALLY CAN RESULT IN PIPE MOVEMENT AND SEPARATION OF JOINTS IF THE JOINTS OF THE SYSTEM ARE NOT OF A RESTRAINED JOINT DESIGN. SEE SECTION 13.3.

Figure 7.4 MSS Table 3. (*From ANSI/MSS, Standard Practice 69.*)



(a) Drawn Tamper-ASTM B88, Type L, Copper Tube

(b) Thermoplastic Pipe

GENERAL NOTES:
(a) Based on pipe at 73°F with water and insulation. Closer spacing required at higher temperatures.
(b) Use shields on all hangers to avoid point loading of pipe.

Figure 7.5 Support spans for copper and thermoplastic pipe (Code Fig. 921.1.3D).

TABLE 121.7.2(A)
CARRYING CAPACITIES OF THREADED ASTM A 36, A 575, AND A 576 HOT ROLLED CARBON STEEL

Nominal Rod Diameter, in.	Root Area of Coarse Thread		Max. Safe Load at Rod Temp. of 650°F (345°C)	
	sq in.	sq mm	lb	kg
3/8	0.068	43.87	610	276
1/2	0.126	81.29	1,130	512
5/8	0.202	130.3	1,810	821
3/4	0.302	194.8	2,710	1 229
7/8	0.419	270.3	3,770	1 710
1	0.552	356.1	4,960	2 250
1 1/4	0.889	573.5	8,000	3 629
1 1/2	1.293	834.2	11,630	5 275
1 3/4	1.744	1 125	15,690	7 117
2	2.292	1 479	20,690	9 385
2 1/4	3.021	1 949	27,200	12 338
2 1/2	3.716	2 397	33,500	15 195
2 3/4	4.619	2 980	41,600	18 869
3	5.621	3 626	50,600	22 952
3 1/4	6.720	4 335	60,500	27 442
3 1/2	7.918	5 108	71,260	32 323
3 3/4	9.214	5 945	82,900	37 640
4	10.608	6 844	95,500	43 360
4 1/4	12.100	7 806	108,900	49 440
4 1/2	13.690	8 832	123,200	55 930
4 3/4	15.379	9 922	138,400	62 830
5	17.165	11 074	154,500	70 140

GENERAL NOTE: Tabulated loads are based on an allowable tensile stress of 12,000 psi (82.7 MPa) reduced by 25% using the root thread area as a base. [Refer to para. 121.2(A).]

Figure 7.6 Code Table 121.7.2(A).

Chapter

8

Listed and Unlisted Components

Each of the codes has a set of listed standards. These are standards that the code committee has researched and has accepted as being compliant with the code. Essentially when one uses one of those standards, no further action is needed to be in accordance with the code. This saves the user much time and effort in proving that the component is an appropriate use for that code, provided that it is in compliance with that standard.

In addition, many of the standards are what is commonly called *dimensional standards*. For the components covered by a given standard, the designers have a prechosen set of dimensions to make their layouts and drawings with. This is another time-saving device for the user. It goes without saying that if the manufacturers can produce in quantity with specific dimensions, the cost of the component will be reduced to a practical minimum.

This in no way prohibits the use of components that are not covered by a standard. In fact, some components are not covered by a standard but do, in fact, see frequent use and have developed a volume market of their own. Some are new inventions or developments that the developer would prefer to keep out of the completely standard market because they are enjoying a relative monopoly for the current moment.

Given that the code is a product of pressure technology, one of the concerns is the pressure-temperature ratings of the components. There are several ways this is accomplished in satisfying that pressure rating requirement.

Each system, be it vessel or piping, has some base pressure-temperature rating. That is essentially the pressure-temperature rating of the weakest member of the system. This can be translated to the assertion that no minor component (valve, fitting, or flange) should be the weakest link.

The base system rating, then, should be the main containment component, that is, pipe in a piping system. When one establishes the size, schedule, or wall thickness and material of the pipe, the base pressure rating of that system is established. It should be pointed out here that the actual highest pressure-temperature combination is usually determined by the process and its designers. Each of the codes goes to some length to define the design pressure and temperature, as discussed earlier.

The process people also work out the excursions from the normal operation that will occur and the various cases or combinations that can arise over the life of that system. Basically, the final design temperature and design pressure are that combination that gives the most critical result in terms of stresses and forces. It is commonly called the *concurrent* temperature and pressure that requires the thickest-wall pipe or highest rating of the components.

The next step is to determine what materials to use. This is a function of the fluid that is being processed and contained. That, too, will be considered as given, as will the volume of fluid that must be piped. When these are established, the sizes of the various components required to complete the piping layout can be ascertained.

The piping codes offer a simplified method to achieve the margin of safety appropriate for the components covered in that code. That basic method is the area replacement method discussed above and in App. A in detail. All codes allow some more robust form of analysis regarding the techniques that a designer may use, provided she or he can demonstrate their validity. Most codes utilize as their basic geometry a surface of revolution. Pipes are cylinders. Spherical or near-spherical surfaces are also employed, usually as end coverings. There exist cases of other shapes such as rectangular and flat. These nonrevolution-type shapes require special consideration. They will not be addressed here.

The pressure rating of a system starts with the pipe. Any flange, fitting, or valve must eventually fit into or on piping in that system. Given the prior work, the next consideration is to determine the pressure-retaining capability of the component of that size and material at that temperature. This pressure rating is one of the things provided by many standards.

The controlling principle is that no component should make the pressure-temperature rating of the system less than the design pressure-temperature rating of the pipe. The various standards rate their components in different manners to accomplish that goal. For listed components there are three basic methods of pressure-temperature rating. Each accomplishes the goal of allowing the designer to chose the proper component for the particular system.

It goes without saying that the first consideration is the size of the component. It must be attached to the matching pipe by some means.

Usually this means welding and sometimes by threads. On rare occasions, there are other mechanical means of attachment. However, many of those means are proprietary and not covered by the codes.

Those pressure-temperature rating systems are listed below. After those systems are covered, the means to provide a pressure-temperature rating for a given code are discussed. Note that when a code accepts the standard as meeting the code, that standard's method of pressure rating has been accepted by that code. Those basic systems of rating are:

- Rating by pressure-temperature chart
- Rating by some form of proof test
- Rating per some specific size pipe

Rating by Pressure-Temperature Chart

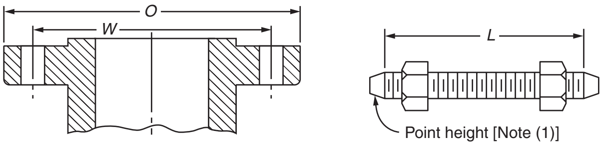
This is the most straightforward method. A chart of some form is provided, and one merely picks the component that has the appropriate rating for the system. There are many ways the chart can be provided.

The most familiar are the charts provided by Standard B16.5, the flange standard. That same chart is used for B16.34, Flanged Valves also. This chart may also be the most complicated in terms of how to determine what. This standard has divided the flanges into pressure classes. Every flange in a particular class has the same dimensions. So whatever material the flange is to be made from must have the same dimensions as a flange in that class of any other material recognized by the standard. The standard publishes a page of controlled dimensions of a particular class.

Figure 8.1 shows one of the dimensional pages. In late 2003, a metric version of the standard was published. The code has maintained the USCS dimensions also. As discussed in Chap. 3 on metrication, those dimensions are not exact mathematical conversions from one dimensional system to another. In fact, the letter shown in Fig. 3.1 was generated due to questions regarding this inexact mathematical conversion. This figure shows both the metric and USCS dimensions. The B16.5 tables with prefixes (Table F15 and Table F16) are in USCS.

In reading the charts, one might notice that the bolt-holes, and therefore the anticipated bolts, are in USCS dimensions for both system. This is so because metric bolting does not have what we call *heavy hex* nuts, and to use metric bolting, one would have to add something like a washer in order for the bolt's nuts to squeeze the flanges properly. This was done in an old existing ISO Standard 7005 and was found not to be acceptable to the user group.

CLASS 600 PIPE FLANGES AND FLANGED FITTINGS



Flange **Stud Bolt With Nuts**

Table 15 Templates for Drilling Class 600 Flanges

1	2	3	4	5	6	7	8	9
		Drilling [Notes (2), (3)]				Length of Bolts, *L* [Notes (1), (4)]		
Nominal Pipe Size NPS	Outside Diameter of Flange, *O*	Diameter of Bolt Circle, *W*	Diameter of Bolt Holes in.	Number of Bolts	Diameter of Bolts in.	7 mm Raised Face	Male and Female/ Tongue and Groove	Ring Joint
1/2	95	66.7	5/8	4	1/2	75	70	75
3/4	115	82.6	3/4	4	5/8	90	85	90
1	125	88.9	3/4	4	5/8	90	85	90
1 1/4	135	98.4	3/4	4	5/8	95	90	95
1 1/2	155	114.3	7/8	4	3/4	110	100	110
2	165	127.0	3/4	8	5/8	110	100	110
2 1/2	190	149.2	7/8	8	3/4	120	115	120
3	210	168.3	7/8	8	3/4	125	120	125
3 1/2	230	184.2	1	8	7/8	140	135	140
4	275	215.9	1	8	7/8	145	140	145
5	330	266.7	1 1/8	8	1	165	160	165
6	355	292.1	1 1/8	12	1	170	165	170
8	420	349.2	1 1/4	12	1 1/8	190	185	195
10	510	431.8	1 3/8	16	1 1/4	215	210	215
12	560	489.0	1 3/8	20	1 1/4	220	215	220
14	605	527.0	1 1/2	20	1 3/8	235	230	235
16	685	603.2	1 5/8	20	1 1/2	255	250	255
18	745	654.0	1 3/4	20	1 5/8	275	265	275
20	815	723.9	1 3/4	24	1 5/8	285	280	290
24	940	838.2	2	24	1 7/8	330	325	335

GENERAL NOTES:

(a) Dimensions of Table 15 are in millimeters, except for diameters of bolts and bolt holes, which are expressed in inch units. For dimensions in inch units, refer to Annex F, Table F15.

(b) For other dimensions, see Table 16.

NOTES:

(1) Length of stud bolt does not include the height of the points. See para 6.10.2.

(2) For flange bolt holes, see para. 6.5.

(3) For spot facing, see para 6.6.

(4) Bolt lengths not shown in table may be in accordance with Annex D. See para. 6.10.2.

Figure 8.1 Sample dimensional charts for class 600 from B16.5.

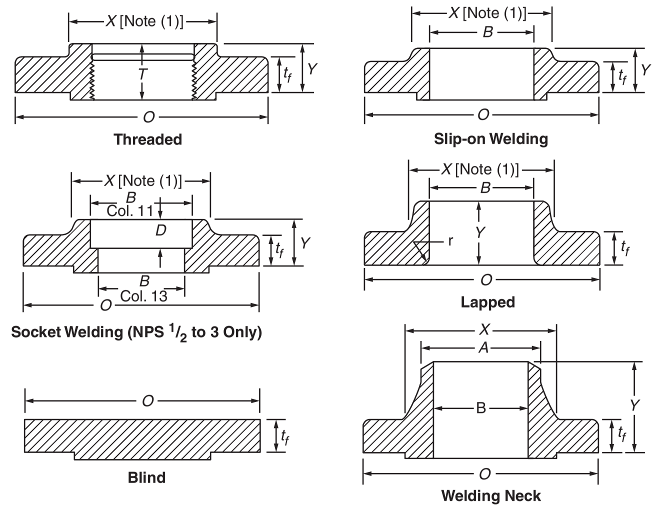


Table 16 Dimensions of Class 600 Flanges

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
					Length Through Hub			Thread Length Threaded Flange Min., T [Note (3)]	Bore			Corner Radius of Bore of Lapped Flange and Pipe, r	Counter bore Threaded Flange Min., Q	Depth of Socket, D
Nominal Pipe Size NPS	Outside Diameter of Flange, O	Thickness of Flange, Min., t_f	Diameter of Hub, X	Hub Diameter Beginning of Chamfer Welding Neck, A [Note (2)]	Threaded/ Slip-on/ Socket Welding, Y	Lapped, Y	Welding Neck, Y		Slip-on Min., B	Lapped Min., B	Welding Neck, B			
1/2	95	14.3	38	21.3	22	22	52	16	22.2	22.9	To be	3	23.6	10
3/4	115	15.9	48	26.7	25	25	57	16	27.7	28.2	specified	3	29.0	11
1	125	17.5	54	33.4	27	27	62	18	34.5	34.9	by	3	35.8	13
1 1/4	135	20.7	64	42.2	29	29	67	21	43.2	43.7	Purchaser	5	44.4	14
1 1/2	155	22.3	70	48.3	32	32	70	23	49.5	50.0		6	50.6	16

Figure 8.1 *(Continued)*

Table 16 Dimensions of Class 600 Flanges (Cont'd)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
					Length Through Hub			Thread Length Threaded Flange Min., T [Note (3)]	Bore			Corner Radius of Bore of Lapped Flange and Pipe, r	Counter bore Threaded Flange Min. Q	
Nominal Pipe Size NPS	Outside Diameter of Flange, O	Thickness of Flange, Min. t_f	Diameter of Hub, X	Hub Diameter Beginning of Chamfer Welding Neck, A [Note (2)]	Threaded/ Slip-on/ Socket Welding, Y	Lapped, Y	Welding Neck, Y		Slip-on/ Socket Welding Min., B	Lapped, Min., B	Welding Neck/ Socket Welding, B			Depth of Socket, D
2	165	25.4	84	60.3	37	37	73	29	61.9	62.5		8	63.5	17
$2\frac{1}{2}$	190	28.6	100	73.0	41	41	79	32	74.6	75.4		8	76.2	19
3	210	31.8	117	88.9	46	46	83	35	90.7	91.4		10	92.2	21
$3\frac{1}{2}$	230	35.0	133	101.6	49	49	86	40	103.4	104.1		10	104.9	...
4	275	38.1	152	114.3	54	54	102	42	116.1	116.8		11	117.6	...
5	330	44.5	189	141.3	60	60	114	48	143.8	144.4		11	144.4	...
6	355	47.7	222	168.3	67	67	117	51	170.7	171.4		13	171.4	...
8	420	55.6	273	219.1	76	76	133	58	221.5	222.2		13	222.2	...
10	510	63.5	343	273.0	86	111	152	66	276.2	277.4		13	276.2	...
12	560	66.7	400	323.8	92	117	156	70	327.0	328.2		13	328.6	...
14	605	69.9	432	355.6	94	127	165	74	359.2	360.2		13	360.4	...
16	685	76.2	495	406.4	106	140	178	78	410.5	411.2		13	411.2	...
18	745	82.6	546	457.0	117	152	184	80	461.8	462.3		13	462.0	...
20	815	88.9	610	508.0	127	165	190	83	513.1	514.4		13	512.8	...
24	940	101.6	718	610.0	140	184	203	93	616.0	616.0		13	614.4	...

GENERAL NOTES:

(a) Dimensions of Table 16 are in millimeters, except for diameter of bolts and bolt holes, which are in inch units. For dimensions in inch units, refer to Annex F, Table F16.

(b) For tolerance, see para. 7.

(c) For facings, see para. 6.4.

(d) For flange bolt holes, see para. 6.5 and Table 15.

(e) For spot facing, see para. 6.6.

(f) For reducing threaded and slip-on flanges, see Table 6.

(g) Blind flanges may be made with or without hubs at the manufacturer's option.

(h) For reducing welding neck flanges, see para. 6.8.

NOTES:

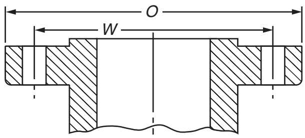
(1) This dimension is for large end of hub, which may be straight or tapered. Taper shall not exceed 7 deg on threaded, slip-on, socket-welding, and lapped flanges. This dimension is defined as the diameter at the intersection between the hub taper and the back face of the flange.

(2) For welding end bevel, see para. 6.7.

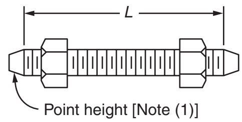
(3) For thread of threaded flanges, see para. 6.9.

Figure 8.1 *(Continued)*

CLASS 600 PIPE FLANGES AND FLANGED FITTINGS



Flange



Stud Bolt With Nuts

Table F15 Templates for Drilling Class 600 Flanges

1	2	3	4	5	6	7	8	9
		Drilling [Notes (2), (3)]				Length of Bolts, *L* [Notes (1), (4)]		
Nominal Pipe Size	Outside Diameter of Flange, *O*	Diameter of Bolt Circle, *W*	Diameter of Bolt Holes	Number of Bolts	Diameter of Bolts	Raised Face 0.25 in.	Male and Female/ Tongue and Groove	Ring Joint
1/2	3.75	2.62	5/8	4	1/2	3.00	2.75	3.00
3/4	4.62	3.25	3/4	4	5/8	3.50	3.25	3.50
1	4.88	3.50	3/4	4	5/8	3.50	3.25	3.50
1 1/4	5.25	3.88	3/4	4	5/8	3.75	3.50	3.75
1 1/2	6.12	4.50	7/8	4	3/4	4.25	4.00	4.25
2	6.50	5.00	3/4	8	5/8	4.25	4.00	4.25
2 1/2	7.50	5.88	7/8	8	3/4	4.75	4.50	4.75
3	8.25	6.62	7/8	8	3/4	5.00	4.75	5.00
3 1/2	9.00	7.25	1	8	7/8	5.50	5.25	5.50
4	10.75	8.50	1	8	7/8	5.75	5.50	5.75
5	13.00	10.50	1 1/8	8	1	6.50	6.25	6.50
6	14.00	11.50	1 1/8	12	1	6.75	6.50	6.75
8	16.50	13.75	1 1/4	12	1 1/8	7.50	7.25	7.75
10	20.00	17.00	1 3/8	16	1 1/4	8.50	8.25	8.50
12	22.00	19.25	1 3/8	20	1 1/4	8.75	8.50	8.75
14	23.75	20.75	1 1/2	20	1 3/8	9.25	9.00	9.25
16	27.00	23.75	1 5/8	20	1 1/2	10.00	9.75	10.00
18	29.25	25.75	1 3/4	20	1 5/8	10.75	10.50	10.75
20	32.00	28.50	1 3/4	24	1 5/8	11.25	11.00	11.50
24	37.00	33.00	2	24	1 7/8	13.00	12.75	13.25

GENERAL NOTES:
(a) Dimensions are in inches.
(b) For other dimensions, see Table F16.

NOTES:
(1) Length of stud bolt does not include the height of the points. See para. 6.10.2.
(2) For flange bolt hole, see para. 6.5.
(3) For spot facing, see para. 6.6.
(4) Bolt lengths not shown in table may be determined in accordance with Annex D. See para. 6.10.2.

Figure 8.1 (*Continued*)

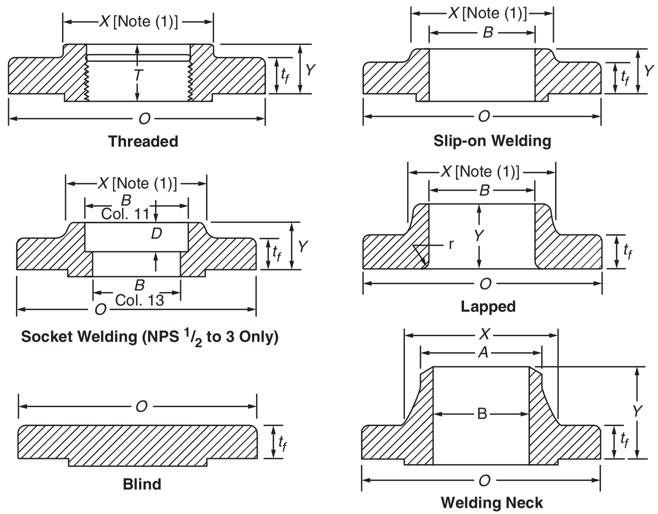


Table F16 Dimensions of Class 600 Flanges

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
					Length Through Hub				Bore					
Nominal Pipe Size	Outside Diameter of Flange, O	Thickness of Flange Min., t_f	Diameter of Hub, X	Hub Diameter Beginning of Chamfer Welding Neck, A [Note (2)]	Threaded Slip-on Socket Welding, Y	Lapped, Y	Welding Neck, Y	Thread Length Threaded Flange Min. T [Note (3)]	Slip-on Socket Welding Min., B	Lapped Min., B	Welding Neck/ Socket Welding, B	Corner Radius of Bore of Lapped Flange and Pipe, r	Counter-bore Threaded Flange Min., Q	Depth of Socket, D
½	3.75	0.56	1.50	0.84	0.88	0.88	2.06	0.62	0.88	0.90	To be	0.12	0.93	0.38
¾	4.62	0.62	1.88	1.05	1.00	1.00	2.25	0.62	1.09	1.11	specified	0.12	1.14	0.44
1	4.88	0.69	2.12	1.32	1.06	1.06	2.44	0.69	1.36	1.38	by	0.12	1.41	0.50
1¼	5.25	0.81	2.50	1.66	1.12	1.12	2.62	0.81	1.70	1.72	Purchaser	0.19	1.75	0.56
1½	6.12	0.88	2.75	1.90	1.25	1.25	2.75	0.88	1.95	1.97		0.25	1.99	0.62

Figure 8.1 (*Continued*)

Table F16 Dimensions of Class 600 Flanges (Cont'd)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
					Length Through Hub				Bore					
Nominal Pipe Size	Outside Diameter of Flange, O	Thickness of Flange Min., t_f	Diameter of Hub, X	Hub Diameter Beginning of Chamfer Welding Neck, A [Note (2)]	Threaded Slip-on Socket Welding, Y	Lapped, Y	Welding Neck, Y	Thread Length Threaded Flange Min., T [Note (3)]	Slip-on Socket Welding Min., B	Lapped Min., B	Welding Neck/ Socket Welding, B	Corner Radius of Bore of Lapped Flange and Pipe, r	Counter-bore Threaded Flange Min., Q	Depth of Socket, D
2	6.50	1.00	3.31	2.38	1.44	1.44	2.88	1.12	2.44	2.46		0.31	2.50	0.69
$2\frac{1}{2}$	7.50	1.12	3.94	2.88	1.62	1.62	3.12	1.25	2.94	2.97		0.31	3.00	0.75
3	8.25	1.25	4.62	3.50	1.81	1.81	3.25	1.38	3.57	3.60		0.38	3.63	0.81
$3\frac{1}{2}$	9.00	1.38	5.25	4.00	1.94	1.94	3.38	1.56	4.07	4.10		0.38	4.13	...
4	10.75	1.50	6.00	4.50	2.12	2.12	4.00	1.62	4.57	4.60		0.44	4.63	...
5	13.00	1.75	7.44	5.56	2.38	2.38	4.50	1.88	5.66	5.69		0.44	5.69	...
6	14.00	1.88	8.75	6.63	2.62	2.62	4.62	2.00	6.72	6.75		0.50	6.75	...
8	16.50	2.19	10.75	8.63	3.00	3.00	5.25	2.25	8.72	8.75		0.50	8.75	...
10	20.00	2.50	13.50	10.75	3.38	4.38	6.00	2.56	10.88	10.92		0.50	10.88	...
12	22.00	2.62	15.75	12.75	3.62	4.62	6.12	2.75	12.88	12.92		0.50	12.94	...
14	23.75	2.75	17.00	14.00	3.69	5.00	6.50	2.88	14.14	14.18		0.50	14.19	...
16	27.00	3.00	19.50	16.00	4.19	5.50	7.00	3.06	16.16	16.19		0.50	16.19	...
18	29.25	3.25	21.50	18.00	4.62	6.00	7.25	3.12	18.18	18.20		0.50	18.19	...
20	32.00	3.50	24.00	20.00	5.00	6.50	7.50	3.25	20.20	20.25		0.50	20.19	...
24	37.00	4.00	28.25	24.00	5.50	7.25	8.00	3.62	24.25	24.25		0.50	24.19	...

GENERAL NOTES:
(a) Dimensions are in inches.
(b) For tolerances, see para. 7.
(c) For facings, see para. 6.4.
(d) For flange bolt holes, see para. 6.5 and Table F15.
(e) For spot facing, see para. 6.6.
(f) For reducing threaded and slip-on flanges, see Table F6.
(g) Blind flanges may be made with or without hubs at the manufacturer's option.
(h) For reducing welding neck flanges, see para. 6.8.

NOTES:
(1) This dimension is for large end of hub, which may be straight or tapered. Taper shall not exceed 7 deg on threaded, slip-on, socket-welding, and lapped flanges.
(2) For welding end bevel, see para. 6.7.
(3) For threads in threaded flanges, see para. 6.9.

Figure 8.1 *(Continued)*

Then the materials that are recognized by the standard are divided up into material groups. Those groups are shown with the materials in that group also listed. And a pressure-temperature chart for each group is given. Figure 8.2 shows a group listing with its attendant materials. Group 2.8 is the group on this page. And the two pressure-temperature charts in metric (bars) and USCS (psig) are shown.

The standard allows interpolation when a given temperature on the chart is between the shown temperatures. This means that if one had a pressure of 640 psig at a temperature of 350°F, one could use a class 300 flange from the group 2.8 material. Conversely, if the pressure were 643 psig at that temperature, it would not be in compliance with the rating of the standard.

Table 2-2.8 Pressure–Temperature Ratings for Group 2.8 Materials

Nominal Designation	Forgings	Castings	Plates
20Cr–18Ni–6Mo	A 182 Gr. F44	A 351 Gr. CK3MCuN	A 240 Gr. S31254
22Cr–5Ni–3Mo–N	A 182 Gr. F51 (1)		A 240 Gr. S31803 (1)
25Cr–7Ni–4Mo–N	A 182 Gr. F53 (1)		A 240 Gr. S32750 (1)
24Cr–10Ni–4Mo–V		A 351 Gr. CE8MN (1)	
25Cr–5Ni–2Mo–3Cu		A 351 Gr. CD4MCu (1)	
25Cr–7Ni–3.5Mo–W–Cb		A 351 Gr. CD3MWCuN (1)	
25Cr–7Ni–3.5Mo–N–Cu–W	A 182 Gr. F55 (1)		A 240 Gr. S32760 (1)

Working Pressures by Classes, bar

Class Temp., °C	150	300	400	600	900	1500	2500
−29 to 38	20.0	51.7	68.9	103.4	155.1	258.6	430.9
50	19.5	51.7	68.9	103.4	155.1	258.6	430.9
100	17.7	50.7	67.5	101.3	152.0	253.3	422.2
150	15.8	45.9	61.2	91.9	137.8	229.6	382.7
200	13.8	42.7	56.9	85.3	128.0	213.3	355.4
250	12.1	40.5	53.9	80.9	121.4	202.3	337.2
300	10.2	38.9	51.8	77.7	116.6	194.3	323.8
325	9.3	38.2	50.9	76.3	114.5	190.8	318.0
350	8.4	37.6	50.2	75.3	112.9	188.2	313.7
375	7.4	37.4	49.8	74.7	112.1	186.8	311.3
400	6.5	36.5	48.9	73.3	109.8	183.1	304.9

NOTE:
(1) This steel may become brittle after service at moderately elevated temperatures. Not to be used over 315°C.

Figure 8.2 ASME Code Tables 2-2.8, 1A, and F2-2.8.

Table 1A List of Material Specifications (Cont'd)

Material Group	Nominal Designation	Pressure–Temperature Rating Table	Applicable ASTM Specifications [Note (1)] Forgings	Castings	Plates
2.4	18Cr–10Ni–Ti	2-2.4	A 182 Gr. F321 A 182 Gr. F321H		A 240 Gr. 321 A 240 Gr. 321H
2.5	18Cr–10Ni–Cb	2-2.5	A 182 Gr. F347 A 182 Gr. F347H A 182 Gr. F348 A 182 Gr. F348H		A 240 Gr. 347 A 240 Gr. 347H A 240 Gr. 348 A 240 Gr. 348H
2.6	23Cr–12Ni	2-2.6			A 240 Gr. 309H
2.7	25Cr–20Ni	2-2.7	A 182 Gr. F310		A 240 Gr. 310H
2.8	20Cr–18Ni–6Mo 22Cr–5Ni–3Mo–N 25Cr–7Ni–4Mo–N 24Cr–10Ni–4Mo–V 25Cr–5Ni–2Mo–3Cu 25Cr–7Ni–3.5Mo–W–Cb 25Cr–7Ni–3.5Mo–N–Cu–W	2-2.8	A 182 Gr. F44 A 182 Gr. F51 A 182 Gr. F53 A 182 Gr. F55	A 351 Gr. CK3MCuN A 351 Gr. CE8MN A 351 Gr. CD4MCu A 351 Gr. CD3MWCuN 	A 240 Gr. S31254 A 240 Gr. S31803 A 240 Gr. S32750 A 240 Gr. S32760
2.9	23Cr–12Ni 25Cr–20Ni	2-2.9			A 240 Gr. 309S A 240 Gr. 310S
2.10	25Cr–12Ni	2-2.10		A 351 Gr. CH8 A 351 Gr. CH20	
2.11	18Cr–10Ni–Cb	2-2.11		A 351 Gr. CF8C	
2.12	25Cr–20Ni	2-2.12		A 351 Gr. CK20	
3.1	35Ni–35Fe–10Cr–Cb	2-3.1	B 462 Gr. N08020		B 463 Gr. N08020
3.2	99.0Ni	2-3.2	B 160 Gr. N02200		B 162 Gr. N02200
3.3	99.0Ni–Low C	2-3.3	B 160 Gr. N02201		B 162 Gr. N02201
3.4	67Ni–30Cu 67Ni–30Cu–S	2-3.4	B 564 Gr. N04400 B 164 Gr. N04405		B 127 Gr. N04400
3.5	72Ni–15Cr–8Fe	2-3.5	B 564 Gr. N06600		B 168 Gr. N06600
3.6	33Ni–42Fe–21Cr	2-3.6	B 564 Gr. N08800		B 409 Gr. N08800
3.7	65Ni–28Mo–2Fe 64Ni–29.5Mo–2Cr–2Fe–Mn–W	2-3.7	B 462 Gr. N10665 B 462 Gr. N10675		B 333 Gr. N10665 B 333 Gr. N10675
3.8	54Ni–16Mo–15Cr 60Ni–22Cr–9Mo–3.5Cb 62Ni–28Mo–5Fe 70Ni–16Mo–7Cr–5Fe 61Ni–16Mo–16Cr 42Ni–21.5Cr–3Mo–2.3Cu 55Ni–21Cr–13.5Mo 55Ni–23Cr–16Mo–1.6Cu	2-3.8	B 462 Gr. N10276 B 564 Gr. N06625 B 335 Gr. N10001 B 573 Gr. N10003 B 574 Gr. N06455 B 564 Gr. N08825 B 462 Gr. N06022 B 462 Gr. N06200		B 575 Gr. N10276 B 443 Gr. N06625 B 333 Gr. N10001 B 434 Gr. N10003 B 575 Gr. N06455 B 424 Gr. N08825 B 575 Gr. N06022 B 575 Gr. N06200
3.9	47Ni–22Cr–9Mo–18Fe	2-3.9	B 572 Gr. N06002		B 435 Gr. N06002
3.10	25Ni–46Fe–21Cr–5Mo	2-3.10	B 672 Gr. N08700		B 599 Gr. N08700

Figure 8.2 (*Continued*)

Table F2-2.8 Pressure–Temperature Ratings for Group 2.8 Materials

Nominal Designation	Forgings	Castings	Plates
20Cr–18Ni–6Mo	A 182 Gr. F44	A 351 Gr. CK3MCuN	A 240 Gr. S31254
22Cr–5Ni–3Mo–N	A 182 Gr. F51 (1)		A 240 Gr. S31803 (1)
25Cr–7Ni–4Mo–N	A 182 Gr. F53 (1)		A 240 Gr. S32750 (1)
24Cr–10Ni–4Mo–V		A 351 Gr. CE8MN (1)	
25Cr–5Ni–2Mo–3Cu		A 351 Gr. CD4MCu (1)	
25Cr–7Ni–3.5Mo–W–Cb		A 351 Gr. CD3MWCuN (1)	
25Cr–7Ni–3.5Mo–N–Cu–W	A 182 Gr. F55 (1)		A 240 Gr. S32760 (1)

Working Pressures by Classes, psig

Class Temp., °F	150	300	400	600	900	1500	2500
100	290	750	1000	1500	2250	3750	6250
200	260	745	990	1490	2230	3720	6200
300	230	665	890	1335	2000	3335	5560
400	200	615	820	1230	1845	3070	5120
500	170	580	775	1160	1740	2905	4840
600	140	555	740	1115	1670	2785	4640
650	125	545	730	1095	1640	2735	4560
700	110	540	725	1085	1625	2710	4520
750	95	530	710	1065	1595	2660	4430

NOTE:
(1) This steel may become brittle after service at moderately elevated temperatures. Not to be used over 600°F.

Figure 8.2 (*Continued*)

There is an Appendix B to B16.5 that explains how the pressures and temperatures are rated. The question has been asked: How are the flanges designed? They are not really designed. The dimensions exist and, absent an error, are not changed. They are rated per the appendix. If one were to use a standard flange calculation method, given the dimensions, it might or might not pass that particular design methodology. Therein lies the beauty of using a flange that complies with the standard—no calculation is required.

Other chart rating standards may not be as complex as the B16.5 charts, but the principles are the same. The designer compares the component that is to be used, including size and material, to the design pressure and temperature of the chart. If it is equal to or lower than the value in the charts, it is acceptable.

Rating by Proof Test

Testing a specific configuration to establish a pressure-temperature rating is quite logical. All proof is "in the pudding." Testing was especially important early on in writing codes and standards for configurations that were difficult to analyze because of their shape and the analytical capabilities available. As analysis capability has grown, the importance of the tests continues; they are used to prove the analytical technique. One should always be able to tie any analytical technique back to actual results to establish that the theory or technique is an acceptable model of what will actually happen.

The grandfather codes—Section I, Power Boilers, and Section VIII, Division 1—included a testing methodology for designs that could not be verified by the "simplified techniques" of those standards. Paragraph UG-101 of Section VIII, Division 1, is the starting basis for much testing methodology. This is consistent with the fact that within the industry each vessel tends to be a unique design. Or, a particular size component has extensive use at the pressure-temperature ratings proved by the test results. There are quite restrictive conditions that would allow a test to be extended to components of exactly the same design.

For years, conservatism required that the successful pressure in the test (most often a bursting pressure) be divided by 5 to establish the working pressure at which that component would be rated—essentially a margin of 25 percent over the computed stresses methodology. Recently, the allowable stresses have been raised, in the material-dependent range, by a little over 14 percent. Accordingly, a test-rated pressure would allow a correspondingly higher working pressure. Actually, it allows a slightly higher increase of 20 percent. In any event, these tests are limited in that they only qualify a specific size and geometry, which is not quite economically feasible in more standardized, repeatable shapes and geometries.

The potential of developing standard components for the multitude of possible configurations and materials that could be proposed and available for use in piping type applications presented several considerations to the standards writers and/or the manufacturers. They can be summarized thus:

- Tests are relatively expensive.
- Tests take considerable time.
- The number of possible combinations of sizes and materials is conservatively in the thousands, if not millions.
- There undoubtedly is some linearity of results between tests, such as for a tee shape.

Given the above, the approach of expanding the coverage of results of single tests to include a range of *similarly proportioned* components evolved. It starts with the concept of relating the pressure-temperature rating of a component to the matching pipe that it is proposed to be mated with.

This rating-by-test methodology works primarily with butt-welding end fittings. The B16.9 (Factory Made Buttwelding End Fittings) tests may be considered the model for these. The simpler *Barlow equation* is used to establish that a design was usable at a pressure-temperature rating that would equal that of the matching pipe.

It does this by establishing a target pressure that a component must be proved to exceed in order to know that it has a pressure-temperature rating equal to that of the matching pipe. In B16.9 that formula in its pressure form is

$$P = \frac{2St}{D}$$

where P = computed proof of test pressure (or target pressure)
S = actual tensile strength of test fitting, determined on a specimen representative of test fitting, which shall meet tensile strength requirements of applicable material
t = *nominal* pipe wall thickness of pipe that the fitting marking identifies
D = specified outside diameter of matching pipe

The other major standards that offer this test use the same basic formula and definitions with any exceptions as follows for MSS SP-75, CSA Z245.11, (a Canadian standard), and MSS SP-97:

1. MSS SP-75—the same definitions
2. CSA Z245.11—the same with a constant 2000 rather than 2 because of metric units
3. MSS SP-97—the same formula, but S is the actual tensile strength of the tested header rather than the fitting. (See comment below on the tensile differences.)

The in-line fittings (other than SP-97) require the actual tensile strength of the fitting. This is a control of the test. If a tester used the tensile strength of the attached pipe, one could possibly pick a pipe of strength near the minimum tensile and a fitting as high as possible. This would tend to distort the test in favor of the fitting. Under that scenario, the target would be the lowest possible. It might not be an appropriate representation of actual field conditions.

The welded-on fitting (SP-97) has placed the control on the header. The logical reasons are the same, albeit somewhat reverse. If the fitting were in the equation, it would be possible to pick a lower-strength fitting and a higher-strength pipe (if the pipe were not in the equation) and distort the test in favor of the fitting.

One only need look at the amount of material in such a fitting to know that a fitting of that type offers more material for reinforcement than, say, a tee of the same size. All other things being equal, that fitting would be stronger than the same-size tee. Note that all other things are rarely equal. So the methodology offers the same philosophy. It behooves the tester to make the fittings and the pipe tested reasonably close in material properties, to ensure a valid test.

Some may question why actual tensile strengths are required when the codes only allow use of the minimum properties in calculating walls or pressure ratings of pipe. By requiring the actual tensile properties in the test equation, one has effectively taken the specific material out of the equation and allowed the use of a test of one material grade to be used for others.

The underlying consideration is that it is a test of geometry. Given that geometry, the pressure-retaining capacity is directly proportional to its tensile properties. That this is true of hollow cylinders is easily proved. The concept follows, albeit with less easily proven (by analysis) other geometries. It has been, and can be, demonstrated by examination of actual tests.

One point of contention that might arise in testing based on tensile strength is that essentially the allowable stresses are often based on yield. In some very ductile materials, this could create severe distortion. In certain operating components such as valves, this could create potential inoperability.

The test methods also put requirements on the various test assemblies regarding the length of the pipe and /or the distance to the end closures. These are intended to ensure that the remainder of the assembly does not offer any stiffening “help” to the fitting being tested by virtue of imposing end effects.

All tests are considered successful if the test is conducted to at least 105 percent of the calculated proof pressure. Careful examination will reveal, in most cases, that there is a margin greater than 5 percent.

1. The use of actual tensile strengths also includes a margin versus the code, in that it increases the test pressure some amount over the minimum for that material.
2. The use of nominal wall provides, in most cases, a mill tolerance increase over the wall thickness allowed by the code.

It is a good practice for the tester to take the test to ultimate failure or burst. This can give some indication of final strength. One of the more important considerations is rather loosely defined; that is the effective geometric control. The standards stipulate that in some manner the tested geometry shall be representative of production.

There is an implication that when the geometry changes, the test should be readministered. This is not an explicit requirement. Nor is there any real guidance for what might be a significant change. The best protection for the user is to take advantage of the necessity of some proof of design and the implied requirement for records. This requirement is explicitly set out where the manufacturer is required to have the analysis or record of the test available for examination.

One important result of a record of the proof is that one manufacturer cannot copy some dimensions of another manufacturer and claim compliance. The geometric dimensions that would be most critical to the pressure-temperature rating are generally not in the standard. For example, the B16.9 standard controls lengths and ends dimensions, whereas the strength-enhancing characteristics would be in the crotches of the tees and intrados of the elbows.

Even more silent in the standards is the necessity for the control of the manufacturing process and design process to ensure that what was tested is what is delivered. The best practice, for the user, is to examine the record of the tests plus the quality and design control procedures of the selected manufacturer. There are, in draft and proposal forms, testing procedures in the codes and standards that address these concerns explicitly. Until such time as the procedures are approved, the user is reminded of the old Latin phrase *caveat emptor* (buyer beware). One should use the tools available to ensure compliance. One representative report of one manufacturer is included in Appendix C, as an example of the type of consideration of the tests.

The goal was to make tests more universal and cover more than the tested component. Then a fair question is: What sizes or ranges does a successful test cover? As noted earlier, the idea of changing from a Section VIII, Division 1, UG-101 type of test, where the test is only good for the specific size, was to get a range of sizes from one test. The use of actual tensile strengths allows a range of materials, as noted before. The size range covered in Code B16.9 is in NPS sizes. A test of one size will apply to *similarly proportioned* NPS sizes from ½ to twice the size of the tested fitting. The rules are similar in the other standards.

The tabular dimensions of the different sizes would still apply and would be considered as similar proportions. The CSA standard includes the words that for tees and crosses the dimensions and tolerances must follow those of the appropriate tables in the standards. Nevertheless, *similarly proportioned* implies that scaling rules apply. Note that the critical proportionality dimensions would therefore be not only those of the standard but also those that would be critical to the pressure integrity.

If one were to test for a range of fittings that covered sizes from NPS ½ to 48, the down ½ and up 2 rule would require several tests at a minimum. See the example set in Table 8.1 as one way to accomplish this.

The reader will note that the sample test series covers overlapping NPSs in the middle ranges. There is strong preference for this sort of coverage. One may even want to add a test of NPS 6, which would cover the range from 3 through 12. Those sizes are the most used. Overlapping tests increase the probability that the conclusions are acceptable.

It is important to note that MSS SP-97 uses four paragraphs to cover the same topic regarding the test as the other standards cover in one. This is so because SP-97 does not really control the header to which the fitting is welded. This takes greater consideration as it is outside the control of the manufacturer or the standard. Therefore, the standard makes distinctions regarding the run-to-branch ratio D_{run}/D_b is. The general rules apply with those ratios equal to or higher than the test ratio. In any event, more than one test would be required.

Quite often the governing code requires interpolation. The suggested set of tests to make a particular family of parts rather automatically gives interpolation between sizes. When a user is reviewing the manufacturer's documentation, these are the sorts of elements of that manufacturer's proof program that give the security to the examiner that the manufacturer does it properly.

The size ranges allowed ratio would cover the same schedule or wall thickness proportionality in NPS ranges, but also the standards allow extrapolation of wall thickness proportionality. If this were not a consideration, a single size test would also require

$$\frac{2St}{D} = P = \frac{2St}{D}$$

tests of all the schedules included in a size catalog. The rule is to allow a range of t/D ratios.

A close examination of this relationship shows that when the material is the same (same allowable stresses), the t/D ratio is directly proportional to the allowed pressure. If one were to make a change in the

Table 8.1 Suggested Size for Proof Test to Cover Size Range from ½ to 48 NPS Size Fittings

Tested NPS	NPS size covered
1	½ through 2
4	2 through 8
12	6 through 24
24	12 through 48

t/*D* ratio of the pipe, its pressure rating would change. If one were to change a pipe diameter and the *t* so that *t*/*D* was the same, then the pressure rating would be the same. If a fitting is considered linearly acceptable across a size range, the changes in *t*/*D* ratios applying to the pipes of those size ranges are also applicable.

Let us examine an example of a tested fitting.

1. Fitting size is NPS 2.
2. What is the applicable size range covered by the fitting test?
3. Tested fitting pipe is Schedule 40 (0.154 nominal wall).
4. What is the *t*/*D* ratio?

Dimensional data: NPS 2, Schedule 40, pipe OD = 2.375 in. (60.3 mm), Schedule 40 wall 0.154 in. (3.9 mm). Thus

Size range would be NPS 1 to 4.

t/*D* would be 0.064.

The allowed changes in *t*/*D* are from ½ to 3 times.

The minimum *t*/*D* would be 0.032.

The maximum *t*/*D* would be 0.194.

The maximum wall thickness that the tested 2-in. fitting could mate with would be 0.460 in. (11.7 mm). This is larger than XXS strong wall for that pipe size, so the applicable schedule covered would be to XXS. (Note that the minimum *t*/*D* of 0.034 would yield a wall thickness of 0.076, which is slightly larger than 0.065 nominal wall of NPS 2 of Schedule 5.) So the fitting would be slightly stronger if it were made with the 0.076 wall and included an appropriate transition for welding it to Schedule 5 pipe.

One might ask: What about at temperature? The fitting is rated to match the pipe. The tests are normally at ambient temperature, and the allowable stresses at that temperature are used. This, then rates the fitting with the pipe at that temperature. As the temperature of the service for which the pipe is intended is changed enough to change the allowable stress of the pipe, that same change in mechanical properties would occur in a fitting of comparable material to the pipe. The analogy would still hold.

As one changes the *t*/*D* ratio, it is clear from the Barlow formula that the resulting allowed pressure will change for a given allowed stress. This sometimes can cause confusion when one is using a larger *t*/*D* ratio than the tested ratio. Using the test example above, consider a pipe of a material having a 15,000 allowable stress. That pipe would have a maximum pressure rating of (for simplification the Barlow formula is used)

$$2 \times 15{,}000 \times 0.154 \times \frac{0.875}{2.375} = 1700 \text{ psi approx.}$$

Note that the pressure required to pass the test would be much higher.

1. Assume the fitting had an actual tensile stress of 60,000. This would be appropriate if the allowed stress were based on 1.4 times the tensile strength, as it is in the current Code B31.1.
2. The 0.875 factor would be eliminated because the test requires a nominal wall in the calculation.
3. The 105 percent is the minimum accepted pressure for a "good" test.
4. The tested minimum pressure would be as shown in the calculation

$$1.05 \times 2 \times 60{,}000 \times \frac{0.154}{2.375} = 8170 \text{ psi approx.}$$

If one were to check the *t/D* ratio of 0.194 as the maximum thickness pipe that the fitting is approved for, the following would result:

1. The maximum schedule is XXS or 0.436 (lower than 0.194×2.375).
2. All other things are equal.
3. The maximum calculated pressure for the system is

$$2 \times 15{,}000 \times 0.436 \times \frac{0.875}{2.375} = 4815 \text{ psi approx.}$$

4. That is still below the pressure at which the fitting was tested.
5. Make the same calculation for the largest size, NPS 4.
6. The maximum wall (0.194×4.5) is 0.873, so the XXS nominal is 0.674.
7. Pressure would be

$$2 \times 15{,}000 \times 0.674 \times \frac{0.875}{4.5} = 3930 \text{ psi appx.}$$

8. This is also still below the successful test pressure.

One can see from this that the allowances which change size and the *t/D* range, in general, will keep the fittings bursting or target pressure within the pressure to which the fitting has been successfully exposed by test. This puts the burden on the user/specifier to do several things:

1. Properly specify the pipe.
2. Only mix materials between the fitting and the pipe with great care. The analogies of the burst do not necessarily hold when one has different materials with different material property curves.
3. Ensure that the manufacturer
   a. has performed appropriate tests on the fitting geometry.
   b. maintains the geometric proportionality to ensure that the allowances in the standard are applicable.
4. Be sure that the governing code or regulation recognizes the standard for which the fitting has been tested.

This is a somewhat harder methodology for the user to assure herself or himself of the validity of the particular product that is being purchased than the chart method. In the chart type of standard, the fitting is found to be in compliance with the standard when a physical inspection of the materials used and the dimensions of the product shows the fitting to meet the requirements of the standard.

A fitting from a proof test standard would require that the user assure himself or herself not only that the material and dimensional requirements of the standard have been met, but also that the proof of design requirements have been met. For the purposes of guidance in what to look for as assurance that a manufacturer has met the requirements of the tests, a copy of a test report from one manufacturer is included in App. C.

Note that in those standards which use pressure rating tables such as the B16.34 valve standard, there are clauses which specify that certain areas require special consideration for thicker walls or other design considerations not within the scope of the standard. These clauses generally make the manufacturer responsible. It would not be unreasonable for the user to ask: How have these considerations been handled? Quite often it will be by the methods of testing or correlated analysis described above.

Rating of Class by Pipe Schedule

The most well-known applications of this method of pressure rating are Code B16.11 and MSS SP-97. They both cover fittings that have a class rating. They are the socket weld and/or threaded fittings standards.

The methodology rates the fittings by establishing correspondence to a schedule of the pipe. The OD of the pipe, including threads, is nominally the same for a given NPS, so there is no effective control of the schedule of pipe that will fit into a specific size of socket or female threaded fittings. The solution the standards developers came to was to rate the fittings for one schedule of pipe. If the user uses a schedule of pipe in the socket or thread that class of fitting is not rated for, then the user has to determine the rating of the assembly.

B16.11 (Paragraph 8.1) specifically states that a proof test for standard materials is not required. SP-97 has ambiguous requirements.

It requires proof of design by mathematical analysis and/or proof test at the manufacturer's option. No exceptions are listed for the socket welding or threaded configuration.

Both standards publish tables to define the correspondence of the pressure rating of the fittings to the specific schedule of pipe for that class. Table 2 of B16.11 is the most explanatory and is duplicated in Fig. 8.3 for convenience. Since there are no specific wall thickness dimensions for certain small pipe in the class 160 and XXS schedules of pipe for $^3/_8$ and smaller pipe, the standard has created an equivalent wall thickness that is shown in Table 3. It is included in Fig. 8.3.

The classes listed are merely names or ways to designate in which pressure class a particular fitting falls. Those classes are classes 2000, 3000, 6000, and 9000. Class 2000 (in Code B16.11) is limited to threaded fittings.

TABLE 2 CORRELATION OF FITTINGS CLASS WITH SCHEDULE NUMBER OR WALL DESIGNATION OF PIPE FOR CALCULATION OF RATINGS

Class Designation of Fitting	Type of Fitting	Pipe Used for Rating Basis [Note (1)] Schedule No.	Wall Designation
2000	Threaded	80	XS
3000	Threaded	160	...
6000	Threaded	...	XXS
3000	Socket-welding	80	XS
6000	Socket-welding	160	...
9000	Socket-welding	...	XXS

NOTE:
(1) This table is not intended to restrict the use of pipe of thinner or thicker wall with fittings. Pipe actually used may be thinner or thicker in nominal wall than that shown in Table 2. When thinner pipe is used, its strength may govern the rating. When thicker pipe is used (e.g., for mechanical strength), the strength of the fitting governs the rating.

TABLE 3 NOMINAL WALL THICKNESS OF SCHEDULE 160 AND DOUBLE EXTRA STRONG PIPE

		Schedule 160		XXS	
DN	NPS	mm	in.	mm	in.
6	⅛	3.15	0.124	4.83	0.190
8	¼	3.68	0.145	6.05	0.238
10	⅜	4.01	0.158	6.40	0.252

Figure 8.3 B16.11 Code Tables 2 and 3.

That is rarely used in industry primarily due to the fact that class 3000 is readily available and will handle a class 2000 condition. Class 9000 (in Code B16.11) is limited to certain sizes of socket welding end fittings. MSS SP-97 recognizes only classes 3000 and 6000.

The reader is directed to Note 1 of Table 2 (see Fig. 8.3). It clearly states the use of pipe other than the schedule for which a class is correlated requires some additional work to determine which component of the assembly controls the pressure rating of that system.

Inherent in those tables is the assumption that the fitting is of the same material as the pipe for which it is intended. This assumption is most important in respect to the mechanical properties of the materials. The piping codes require adjustment to the size and amount of reinforcing material used when that material has a lower allowable stress than the attached pipe. The adjustment is made to require more of the lower-strength material. While not specifically discussed in the standards, it is accomplished by deferring to the governing code or regulation.

A casual reading of the tables might lead one to believe that if one uses a certain wall thickness or schedule, then the proper fitting to use is the class associated with that fitting. Note 1 states that is not the intent. It then describes that the only effect of using different wall thicknesses than those in the table may be that a different part of the system governs the rating. This may translate to "If you do that, the responsibility is yours."

What is the meaning of that? It is known that the system has at least three and maybe more components in its assembled state. Assume a Code B16.11 elbow. In its assembled state it will have two pieces of pipe and one elbow. Each of those elements has a pressure-temperature rating. Each pipe, as pointed out elsewhere, has a pressure-temperature rating based on its wall thickness. And in either the threaded case or the socket weld case, any wall thickness can be assembled to the socket or the thread.

In the case of the standard, it says that the fitting has a rating equal to a specific wall thickness of a piece of straight pipe of the same material. It may be different from that of the pipe utilized in the assembly. What the note is saying is that if one "mixes" wall thicknesses, one has to do some calculation to be sure what the pressure-temperature rating of the assembly will be.

This leads to another point of confusion with these tables in Code B16.11. One can easily note that there is a difference in the wall ratings for threaded and for socket-welded fittings. For example, a class 3000 socket fitting is rated as an S80 wall thickness, while the same class 3000 is rated as an S160 fitting. Why is the class 3000 rated so much heavier? The real question is: Is it rated heavier and, if so, by how much?

Remember that while rating a piece of pipe, one has to allow for any mechanical allowances before one can define the pressure rating of that piece of pipe. Threading requires a reduction of the usable wall thickness for pressure consideration equal to the thread depth. To achieve the same pressure rating at a specific temperature of threaded pipe, one has to start with a thicker pipe, at least the thread depth thicker to achieve that rating.

For illustrative purposes, an example is given.

1. Use 1 NPS pipe (OD of 1.315).
2. The S80 nominal wall is 0.179.
3. The S160 nominal wall is 0.250.
4. They both have a manufacturer's (pipe) wall tolerance of 12.5 percent.
5. One NPS threads NPT have a thread depth of 0.070. Note that the thread depth of 1 NPS B1.20.1 threads as given in that standard is *h*, and that is defined as 0.0695, which is rounded to 0.070.
6. The allowed stress for this material is 20,000 psi.
7. For simplicity the Barlow equations are used.
8. The allowable pressure of the Schedule 80 pipe is

$$P = \frac{2(0.179 - 0.125 \times 0.179) \times 20{,}000}{1.315} = 4764 \text{ psi}$$

9. The allowable pressure for the Schedule 160, less both the manufacturer's tolerance and the thread allowance, would be

$$P = \frac{2(0.250 \times 0.875 - 0.070) \times 20{,}000}{1.135} = 4525 \text{ psi}$$

As can be seen, the difference between the two actual pressures is, for all practical purposes, eliminated by the reduction in wall thickness due to the threads. The difference in schedules is only due to mechanical allowances within the scope of using standard walls. In any event, the class by pipe schedule type of pressure-temperature rating allows the user to calculate, for his or her system, which class is appropriate. What appears to be a simple match of schedules is not quite that straightforward, because schedules simply do not match up. The notes in tables point to user responsibility when not matching schedules. These are the three main ways in which standards listed by the codes rate the components within the standard.

Standards Recognized

As mentioned in Chap. 4 on materials, the B31 codes recognize many ASTM standards as their basic materials. There are, of course, many other standards that they recognize. The location of the lists and the editions of those standards are shown in Table 4.10. One should always refer to the specific table and code when one is going through the selection process. In general they are listed by standards writing organization.

Not all standards are recognized by all codes. It is somewhat tedious to look through the standards list in the books when seeking a solution to a particular piping problem. For that reason, an abbreviated table is developed here. Table 8.2 reclassifies the standards per type, e.g., flange and fitting. It states the standards writing organization and some brief discussion of what the standard covers.

Unlisted Standards

Sometimes one wants to use a component that is not found in the listed standards because either there is actually no standard that covers that component or the standard that does cover that component is not listed. Typically the B31 codes call these components *special* designed components. These codes offer solutions in various ways.

They have discussions of what is required. Table 8.3 lists the specific code paragraphs that cover the particular codes requirements for those unlisted components.

In general these paragraphs allow some proof test and/or engineering calculations. The specific codes have some limitations specific to the service of their intended use. Chapter IX of Code B31.3 is for high pressure and is more restrictive; e.g., bellows-type expansion joints are prohibited. The further one strays from a standardized component, the more important it is to check the specific requirements.

The most complete description of what is required, including documentation, is found in the B31.1 and B31.3 codes. Paragraph 104.7.3 is duplicated here for convenience (Fig. 8.4).

It is fair to assume that area replacement type of calculation is not specifically intended to be used. It is not ruled out either. The expression *engineering calculations* usually means that it might be difficult to prove by using area replacement only.

One such consideration might be a wye-type connection. This connection is used fairly often in piping layouts, particularly in situations where a boiler might be serving more than one turbine of the division of the flow required for the process.

It is relatively difficult to use straightforward area replacement. Of course, more rigorous analysis such as finite element analysis is always

TABLE 8.2 Acceptable Standards per Code

			B31.1	B31.3	B31.4	B31.5	B31.8	B31.9	B31.11
Flange Standards									
B16.5	ASME	Pipe Flanges & Flanged Fitting	Y	Y	Y	Y	Y	Y	Y
B16.1	ASME	Cast Iron Flanges	Y	Y	N	N	Y	Y	N
B16.24	ASME	Cast Copper Alloy Pipe Flanges	Y	Y	N	Y	Y	Y	N
B16.36	ASME	Steel Orifice Flanges	N	Y	N	N	N	N	N
B16.42	ASME	Ductile Iron Pipe Flanges	Y	Y	N	N	Y	Y	N
B16.47	ASME	Large Diameter Steel Flanges	Y	Y	N	N	N	N	N
SP-6	MSS	Standard Flange Face Finishes	Y	Y	Y	Y	Y	Y	Y
SP-9	MSS	Spot Facing for Flanges	Y	Y	N	Y	N	N	N
SP-44	MSS	Steel Pipe Line Flanges	N	Y	Y	N	Y	N	Y
SP-51	MSS	Class 150 LW Corrosion Cast Flanges	Y	Y	N	Y	N	Y	N
SP-65	MSS	High Pressure Tens Gasket Flanges	N	Y	N	N	N	N	N
C207	AWWA	Steel Pipe Flanges for Waterworks to 144 in.	Y	Y	N	N	N	Y	N
B16.20	ASME	Metallic Flange Gaskets	Y	Y	Y	N	Y	N	Y
B16.21	ASME	Non-Metallic Flange Gaskets	Y	Y	Y	N	N	Y	Y
Valve Standards									
B16.33	ASME	Manual Operated Metallic Gas Valves	N	N	N	N	Y	Y	N
B16.34	ASME	Valves Flanged, Threads and Welding End	Y	Y	N	Y	Y	Y	Y
B16.38	ASME	Large Manually Operated Metallic Gas Valves	N	N	N	N	Y	N	N
B16.40	ASME	Manually Operated Thermoplastic Valves	N	N	N	N	Y	N	N
SP-42	MSS	Cl 150 Gale Globe Angle & Check Valves	Y	Y	N	Y	N	Y	N
SP-67	MSS	Butterfly Valves	Y	N	Y	N	N	Y	Y
SP-70	MSS	Cast Iron Gate Valves	N	Y	Y	Y	Y	Y	Y
SP-71	MSS	Gray Iron Swing Check Valves	N	Y	Y	N	Y	Y	Y
SP-72	MSS	Ball Valves for General Service	N	Y	N	N	N	Y	N
SP-78	MSS	Cast Iron Plug Valve	N	N	Y	N	Y	Y	Y
SP-80	MSS	Bronze Gate Globe Angle & Check Valves	Y	Y	N	Y	N	Y	N
SP-81	MSS	Stainless Bonnetless Knife Gate Valves	N	Y	N	N	N	N	N

(Continued)

TABLE 8.2 Acceptable Standards per Code (*Continued*)

			B31.1	B31.3	B31.4	B31.5	B31.8	B31.9	B31.11
SP-85	MSS	Cast Iron Globe, Angle	N	Y	N	N	N	Y	N
SP-88	MSS	Diaphragm Type Valve	N	Y	N	N	N	Y	N
SP-105	MSS	Instrument Valves for Code Applications	Y	Y	N	N	N	N	N
C500	AWWA	Metal-Seated Gate Valves for Water	Y	Y	N	Y	N	Y	N
C504	AWWA	Rubber Seated Butterfly Valves	Y	Y	N	N	N	N	N
C509	AWWA	Resilient-Seated Gate Valves for Water	Y	N	N	N	N	N	N
API-526	API	Flanged Steel Pressure-Relief Valves	N	Y	N	N	N	N	N
API-594	API	Wafer and Wafer Plug Check Valves	N	Y	N	N	N	Y	N
API-599	API	Metal Plug Valves	N	Y	N	N	N	N	N
API-600	API	Steel Gate Valves	N	Y	Y	Y	N	N	Y
API-602	API	Compact Steel Gate Valve	N	Y	Y	N	N	N	Y
API-603	API	Corrosion Resistant Gate Valves	N	N	Y	N	N	N	Y
API-608	API	Metal Ball Valves	N	Y	N	N	N	N	N
API-609	API	Lug and Wafer Butterfly Valves	N	Y	N	N	N	Y	N
Fittings									
B16.3	ASME	Malleable Iron Threaded Fittings	Y	Y	N	Y	N	Y	N
B16.4	ASME	Grey Iron Threaded Fittings	Y	Y	N	Y	N	Y	N
B16.9	ASME	Wrought Steel Butt Welding Fittings	Y	Y	Y	Y	Y	Y	Y
B16.11	ASME	Forged Steel Butt Welding Fittings	Y	Y	N	Y	Y	Y	Y
B16.14	ASME	Ferrous Bushing, Plugs with Pipe Threads	Y	Y	N	Y	N	Y	N
B16.15	ASME	Cast Bronze Threaded Fittings	Y	Y	N	Y	N	Y	N
B16.18	ASME	Solder Joint Pressure Fitting	N	Y	N	Y	N	Y	N
B16.22	ASME	Wrought Copper Pressure Fittings	Y	Y	N	Y	N	Y	N
B16.26	ASME	Copper Fittings for Flared Tubing	N	Y	N	N	N	Y	N
B16.39	ASME	Malleable Iron Unions	N	Y	N	N	N	Y	N
B16.48	ASME	Steel Line Blanks	Y	Y	N	N	N	N	N
SP-43	MSS	Wrought Stainless Butt-Welding Fittings	Y	Y	N	Y	N	Y	N

SP-75	MSS	High Test Fittings	Y	Y	Y	N	Y	N	Y
SP-79	MSS	Socket Welding Reducer Inserts	Y	Y	N	N	N	N	N
SP-83	MSS	Class 3000 Steel Pipe Unions	N	Y	N	N	N	Y	N
SP-95	MSS	Swage Nipples and Bull Plugs	Y	Y	N	N	N	N	N
SP-97	MSS	Integrally Reinforce Forged Branch Outlets	Y	Y	N	Y	N	N	N
SP119	MSS	Belled End Socket Welding Fittings	N	Y	N	N	N	N	N
C110/A21.1	AWWA	Ductile and Gray Iron Fittings	Y	N	N	Y	N	Y	N
A21.14	AWWA	Ductile Iron Fittings for Gas	N	N	N	N	Y	Y	N
C151/A21.53	AWWA	Ductile Iron Compact Fittings	Y	N	N	N	N	Y	N
C208	AWWA	Dimensions, Fabricated Steel Water Fittings	Y	N	N	N	N	Y	Y
API-6A	API	Wellhead Equipment	N	N	Y	N	Y	N	Y
API-6D	API	Pipeline End Closures, Connectors and Swivels	N	N	Y	N	Y	N	N
SAE J513	SAE	Refrigeration Tube Fittings	N	Y	N	Y	N	Y	N
SAE J514	SAE	Hydraulic Tube Fittings	N	Y	N	N	N	Y	N
SAE J518	SAE	Hydraulic Flanged Tube Pipe & Hose Connections	N	Y	N	N	N	N	N
Quality Standards									
B16.10	ASME	Face to Face Dimensions for Valves	Y	Y	Y	Y	N	Y	Y
B16.25	ASME	Butt Welding Ends	Y	N	Y	Y	N	Y	Y
B46.1	ASME	Surface Texture	N	Y	N	N	N	N	N
TDP-1	ASME	Practices to Prevent Turbine Water Damage	Y	N	N	N	N	N	N
SP-25	MSS	Standard Marking Systems	Y	Y	Y	Y	Y	Y	Y
SP-53	MSS	Quality Standard—Magnetic Particle	Y	N	N	N	N	N	N
SP-54	MSS	Quality Standard—Radiographic	Y	N	N	N	N	N	N
SP-55	MSS	Quality Standard Visual for Surface Irregularity	Y	N	N	N	N	N	Y
SP-61	MSS	Hydrostatic Testing of Steel Valves	Y	N	Y	N	N	N	Y
SP-93	MSS	Quality Standard Dye Penetrantt Method	Y	N	N	N	N	N	N

(*Continued*)

TABLE 8.2 Acceptable Standards per Code (*Continued*)

			B31.1	B31.3	B31.4	B31.5	B31.8	B31.9	B31.11
SP-94	MSS	Quality Standard Ultrasonic	Y	N	N	N	N	N	N
API 5B	API	Specification for Line Pipe Threads	N	Y	Y	N	N	N	Y
CP-189	ASNT	Qualification & Certification of NDE Personnel	Y	N	N	N	N	N	N
SNT-TC-1A	ASNT	Qualification & Certification of NDE Personnel	Y	Y	N	N	N	N	N
QC1	AWS	Qualification & Certification of Welding Inspectors	Y	N	N	N	N	N	N
ES-16	PFI	Access Hole Plugs for Radiographic Inspection	Y	N	N	N	N	N	N
mrO1-75	Nace	Sulfide Stress Cracking	N	Y	Y	N	N	N	N
Pipe Supports and Hangers									
SP-58	MSS	Hangers Materials, Design, and Manufacture	Y	Y	N	Y	N	Y	Y
SP-69	MSS	Hangers and Supports Selection & Application	Y	N	Y	N	N	Y	Y
SP-89	MSS	Hangers and Supports Fabrication & Installation	Y	N	N	N	N	Y	N
SP-90	MSS	Guidelines on Terminology of Hangers and Supports	N	N	N	N	N	Y	N
Pipe and Joints									
5L	API	Line Pipe	Y	Y	Y	Y	Y	Y	Y
B36.10M	ASME	Welded and Seamless Wrought Steel Pipe	Y	Y	Y	Y	Y	Y	Y
B36.19M	ASME	Stainless Steel Pipe	Y	Y	Y	Y	N	Y	Y
C111/A21.11	AWWA	Rubber-Gaskets Joints for Pressure Pipe	Y	Y	N	N	Y	N	N
C115/A21.15	AWWA	Flanged Pipe with Threaded Flanges	Y	Y	N	N	Y	N	N
C150/A21.50	AWWA	Thickness Design of Ductile Iron Pipe	Y	Y	N	N	N	Y	N
C151/A21.51	AWWA	Centrifugally Cast Ductile Pipe for Water	Y	Y	N	N	N	N	N
A21.52	AWWA	Centrifugally Cast Ductile Pipe For Gas	N	N	N	N	Y	Y	N
C101	AWWA	Thickness Design of Cast Iron Pipe	N	N	N	N	Y	N	N
C200	AWWA	Steel Water Pipe	Y	Y	N	N	N	N	N
C300	AWWA	Reinforced Concrete Pressure Pipe	Y	Y	N	N	N	Y	N
C301	AWWA	Prestressed Concrete Pressure Pipe	Y	Y	N	N	N	Y	N

C302	AWWA	Reinforced Concrete Pressure Pipe Non-cylinder	Y	Y	N	N	N	Y	N
C304	AWWA	Design of Prestressed Concrete Pipe	Y	N	N	N	N	N	N
C600	AWWA	Installation of Ductile Iron Water Mains	Y	N	N	N	N	Y	N
C606	AWWA	Grooved and Shoulder Joints	Y	N	N	N	N	Y	N
C900	AWWA	PVC Pressure Pipe 4-12	Y	N	N	N	N	Y	N
NFPA	NFPa	Standard for Fire Hose Connections	Y	Y	N	N	N	N	N
SP-73	MSS	Brazing Joints for Copper Alloy Solder Joints							
Threads and Hardware									
			Y	N	N	Y	N	Y	Y
B1.20.3	ASME	Dryseal Pipe Threads (inch)	Y	Y	Y	N	N	N	N
B18.22M	ASME	Washers Metric Plain	Y	N	N	N	N	N	N
B1.1	ASME	Unified Inch Screw Threads	Y	Y	Y	Y	Y	Y	Y
B1.13 M	ASME	Metric Screw Threads M Profile	Y	N	N	N	N	N	Y
B1.20.1	ASME	Pipe Threads General Purpose (inch) 20	Y	Y	Y	Y	Y	Y	N
B1.20.7	ASME	Hose Coupling Screw Threads	N	Y	N	N	N	Y	N
B18.2.1	ASME	Square and Hex Bolts and Screws (inch)	Y	Y	N	Y	Y	Y	N
B18.2.2	ASME	Square and Hex Nuts (inch)	Y	Y	N	Y	Y	Y	N
B18.2.4.6M	ASME	Hex Nuts Heavy, Metric	Y	N	N	N	N	N	N
B18.2.3.5M	ASME	Metric Hex Bolts	Y	N	N	N	N	N	N
B18.2.3.6	ASME	Metric Heavy Hex Bolts	Y	N	N	N	N	N	N
B18.21.1	ASME	Lock Washers (inch)	Y	N	N	N	N	N	N
B18.22.1	ASME	Plain Washers	Y	N	N	N	N	N	N

TABLE 8.3 Code Paragraphs Setting Requirements for Unlisted Components

Code	Paragraphs
B31.1	102.2.2,104.7,104.7.2
B31.3	302.2.2, 302.2.3, 304.7.1,304.7.2; A307.4.2, A304.7.3; K304.7.2, K304.7.3, K304.7.4
B31.4	402.2.1, 402.2.2, 404.7, 423.1
B31.5	502.2.5, 0504.7
B31.8	831.36
B31.9	902.2.1, 902.2.2, 904.7.1, 904.7.2
B31.11	1102.2.2, 1104.7, 1123.1(b)

104.7.2 Specially Designed Components. The pressure design of components not covered by the standards listed in Table 126.1 or for which design formulas and procedures are not given in this Code shall be based on calculations consistent with the design criteria of this Code. These calculations shall be substantiated by one or more of the means stated in (A), (B), (C), and (D) below:

(A) extensive, successful service experience under comparable conditions with similarly proportioned components of the same or similar material;

(B) experimental stress analysis, such as described in the ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, Appendix 6;

(C) proof test in accordance with either ASME B16.9, MSS SP-97, or the ASME Boiler and Pressure Vessel Code, Section I. A-22; and

(D) detailed stress analysis, such as finite element method, in accordance with the ASME Boiler and Pressure Vessel Code, Division 2, Appendix 4, except that the basic material allowable stress from the Allowable Stress Tables of Appendix A shall be used in place of S_m.

For any of (A) through (D) above, it is permissible to interpolate between sizes, wall thicknesses, and pressure classes and to determine analogies among related materials.

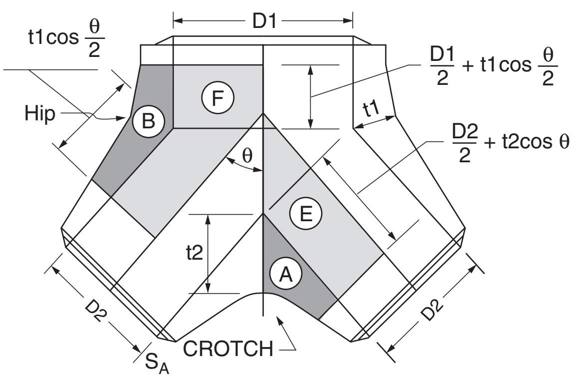
Calculations and documentation showing compliance with this paragraph shall be available for the owner's approval, and, for boiler external piping, they shall be available for the Authorized Inspector's review.

Figure 8.4 Paragraph 104.7.2 of Code B31.3.

an option. There are simpler analyses that have successfully been used to accomplish the required proof. One such analysis is commonly called *pressure area*. A simplified chart of how to accomplish that sort of analysis is given in Fig. 8.5. This is based on the concept that the two most highly stressed areas of the configuration are in the crotch and the hip of the wye configuration. Two calculations are made to equate the area exerting pressure influence on those areas and the metal resisting that pressure influence. It is based on unit depths of material and pressure. It assumes that if the resisting metal stress as calculated by the method is below the acceptable stress for the service, then the wye has sufficient margin. Many wyes have been designed using this method. They have been successful in the service they were designed to work in. It thus can be asserted that it meets the criteria as outlined in the paragraph cited. In some cases, finite element analysis has confirmed that the procedure is acceptable.

There are many commonly used fittings that fall into the same position that the wyes described above have. Some of those fittings are listed with a description of the general way they are proved to meet the codes. This discussion gives guidance to the reader as to an approach to these less-than-standard fittings.

- Anchor flanges are used mainly at the end of pipelines to absorb any expansion loads. A modified version of bolted flange calculations, such as in Section VIII, Appendix 2, is most often used. This eliminates the bolting process and includes the loads generated by the expansion of pipe.



$$S_A \geqq \frac{P(E + 0.5A)}{A} \qquad S_A \geqq \frac{P(F + 0.5B)}{B}$$

Figure 8.5 Diagram showing pressure area method of proving wyes.

- An example sketch of insert-type fittings can be seen in Figure K328.5.4 of B31.3. Many of the manufacturers have proof tests of their designs of this type of fitting. These tests mirror the type of test in B16.9 or MSS SP-97.
- There are laterals that are a wye with only one leg at an angle, rather than both legs. The standard methodology of proving this type of fitting follows the concepts of the wye methodology discussed in detail above.
- Target tees are usually made from a tee. Essentially, one leg of the tee is blocked in some manner to give a large amount of sacrificial material for an erosive condition. This proof involves proof of pressure retention for the unblocked portion. This can be a proof test of the unblocked tee, and a mathematical indication that the blocked portion wil not "blow out."
- There are many proprietary designs of components in which the proof of code compatibility can be left to the manufacturer.

The marrying of components to the main system and establishment of compliance with the code requirements are a major part of the work of design. The establishment of accepted standards becomes an important time-saver for the designer. This leaves the question: What are these accepted standards? Also a major concern should be which edition of the standard is acceptable.

Any given standard must be reviewed periodically to determine if it is still valid. The most common review period is five years. With hundreds, if not thousands, of standards to review, it is rather a blessing that they are not all on the same schedule. The B31 codes have recognized that it is a daunting task to keep these up to date. They usually have at least two tables in their book for this purpose. One lists the standards by the standards writing body. The other is usually an appendix that lists the approved edition of the standard.

The codes recognize that everywhere a particular standard is mentioned throughout the code, it would be difficult to mention the edition. They therefore put that information in an appendix and describe them in paragraphs. Table 8.4 lists those locations. By convention, the paragraph that has the same base number as the table also discusses the issue. Another table of commonly used standards is developed which gives the listing across codes. This table does not include the ASTM material standards. For convenience the names and addresses of the standardization body are also included as a part of that table.

The table that is broken down by type of standard valve, flange, etc., and showing which codes have that standard listed is Table 8.2. The

TABLE 8.4 Tables of Listed Standards

CODE	Tables, paragraphs
B31.1	Tables 123.1, 126.1 Appendix F
B31.3	Tables 323.1, 326.1, A326.1, M326, and K326; Appendix E
B31.4	423.1, 426.1, Appendix A
B31.5	526.1, Appendix A
B31.8	831, Appendix A
B31.9	Table 926.1, Appendix C
B31.11	Tables 1123, 1126.1, Appendix I

reader is reminded that not having a particular standard recognized by a certain code does not mean that the product of that standard cannot be used in that code. It means that the committee has not found enough reason for use in the intended service to recognize the code. The user must find other means to establish that the component meets the requirements of the code.

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Chapter

9

Fabrication

Historical Comments

When one begins to assemble the piping system that has been specified and designed, welding becomes a big part of the fabrication process. It is customary for aboveground piping to be made in welding shops as spools which can be shipped to the assembly and erection site.

ASME has a Section IX, Welding and Brazing Qualifications. It is a very complete section of some 200 plus pages. It is defined by all the piping books as the method used to ensure that the welding is in compliance with the code. There are exceptions listed to meet the specific needs of the particular section's concerns.

Ordinarily, that is the only outside welding reference accepted. However, the buried or pipeline codes have two related problems for which they allow an alternative. Codes B31.4, B31.8, and B31.11 all allow qualification through API 1104, *Welding of Pipelines and Related Equipment*, or ASME Section IX. The scope of this book does not include discussion of the qualifications embodied in these two different standards. However, readers will have enough information to determine whether they should investigate one over the other.

The first difference, and one of the major ones, between the two standards can be deduced by reading the scope of API 1104, including its title. That scope clearly states it is developed for welds in carbon and low-alloy steel piping, whereas ASME Section IX covers essentially all grades of material anticipated in the ASME codes. This includes austenitic steels, nickel alloys, and many other variations. It also includes brazing which is not anticipated in API 1104.

In addition, the standard establishes procedures for nondestructive testing and sets acceptance standards for tests that take the material to destruction or one of the NDE tests. It is in SI or inch-pound units and, as might be expected, requires each system to be used independently, as discussed in Chapter 3, Metrication.

As an example of the difference in complexity, one can compare the accepted weld rods and electrodes of the two standards. ASME Section IX in Paragraphs QW 431 through QW 433 sets out the ASME SFA* numbers, which correlate to AWS classifications, in sets of seven types:

- QW 432.1 Steel and Steel Alloys
- QW 432.2 Aluminum and Aluminum Base Alloys
- QW 432.3 Copper and Copper Base Alloys
- QW 432.4 Nickel and Nickel Base Alloys
- QW 432.5 Titanium and Titanium Alloys
- QW 432.6 Zirconium and Zirconium Alloys
- QW 432.7 Hard-Facing Weld Metal Overlay

By comparison, API 1104 in its Paragraph 4.2.2.1 states that all its recognized filler metal will conform to one of the following specifications:

- AWS A5.1
- AWS A5.2
- AWS A5.5
- AWS A5.17
- AWS A5.18
- AWS A5.20
- AWS A5.28
- AWS A5.29

To be accurate, the A5.1 specification may have several classifications inside it which would be listed separately in ASME Section IX. The comparison is given so that readers can develop an understanding of the difference in the two approaches.

The inclusion of the NDE tests and acceptance criteria is a further simplification of presentation. The piping codes that refer to API 1104 have included those criteria by reference. The aboveground codes either spell out their specific criteria, which most do to some extent, or refer to something like ASME Section V, Non-Destructive Examination.

The last consideration might be that in the construction of pipelines a considerable amount of the welding is done in the field. Sections of pipe

*The F represents filler metal to differentiate from the SA or base metals.

are often welded and laid in the ditch on site. This simpler and smaller standard can be used at the site, and many of the references required can be consulted at the location.

So one might want to use the API 1104 welding standard where appropriate. It should be reaffirmed that the final quality of the weld is, in the opinion of the Code B31 committees who have included the option in their code, not significantly different, assuming the requirements of the standard used are met. Some of the requirements are in the area of examination or test of actual welds, and these will be covered in greater detail when the requirements for examination and inspection are discussed.

Welding Requirements

Regardless of the weld standard used, certain basic requirements must be met when an organization provides welds for a piping system. The most basic of these is that the weld be done with a Qualified Weld Procedure Specification (WPS). It shall be performed by a qualified welder.

A WPS is the document that lists the parameters to be used in construction of a weldment in accordance with the code. One set of parameters that is extremely important is the fit-up and gap between the two pieces to be welded, along with the weld preparation of the ends to be welded. Figure 9.1 shows pictures of such a fit-up done in a weld fabrication shop. Figure 9.2 shows how welders must work with smaller pipe to make a particular spool work. There are many parameters to be listed, and they vary in two ways. First, there are differences in the welding process, such as gas-metal arc welding (GMAW) or gas tungsten-arc welding (GTAW).

In ASME Section IX, there is a listing of the parameters required for each of the several specifications that can be chosen. Similar requirements are set out in API 1104. These parameters are further broken into essential and nonessential variables. An essential variable is one that, when changed, requires a requalification of the procedure. Figure 9.3 is a copy of QW-256 listing the variables required for an ASME Section IX WPS for GTAW with the chapters in Section IX that spell out the details required by that section.

As has been discussed, a similar WPS might be required for the API 1104 standard. It would be somewhat simpler, but WPS would require some proof that it is appropriate. The proof is generally known as the procedure qualification record (PQR) or, in API 1104, the procedure qualification test. Figure 9.4 is a sample of the requirements as defined in API 1104 to qualify the WPS. Remembering that API 1104 is written specifically for pipe and the ASME is for all kinds of welds, one can see that this figure is somewhat more representative of the work required. It is not the exact requirements of Section IX.

Figure 9.1 Weld fit-up and gap examples. (*Photograph courtesy of TEAM Industries, Inc., Kaukauna, WI.*)

Figure 9.2 Weld fit-up and gap examples. (*Photograph courtesy of TEAM Industries, Inc., Kaukauna, WI.*)

PROCEDURE QUALIFICATIONS

QW-256
WELDING VARIABLES PROCEDURE SPECIFICATIONS (WPS)
Gas Tungsten-Arc Welding (GTAW)

Paragraph			Brief of Variables	Essential	Supplementary Essential	Nonessential
QW-402 Joints	.1	ϕ	Groove design			X
	.5	+	Backing			X
	.10	ϕ	Root spacing			X
	.11	±	Retainers			X
QW-403 Base Metals	.5	ϕ	Group Number		X	
	.6		*T* Limits		X	
	.7		*T/t* Limits > 8 in. (203 mm)	X		
	.8	ϕ	*T* Qualified	X		
	.11	ϕ	P-No. qualified	X		
	.13	ϕ	P-No. 5/9/10	X		
QW-404 Filler Metals	.3	ϕ	Size			X
	.4	ϕ	F-Number	X		
	.5	ϕ	A-Number	X		
	.12	ϕ	AWS class.		X	
	.14	±	Filler	X		
	.22	±	Consum. insert			X
	.23	ϕ	Filler metal product form	X		
	.30	ϕ	*t*	X		
	.33	ϕ	AWS class.			X
	.50	±	Flux			X
QW-405 Positions	.1	+	Position			X
	.2	ϕ	Position		X	
	.3	ϕ	↑↓ Vertical welding			X
QW-406 Preheat	.1		Decrease > 100°F (56°C)	X		
	.3		Increase > 100°F (56°C) (IP)		X	
QW-407 PWHT	.1	ϕ	PWHT	X		
	.2	ϕ	PWHT (T &T range)		X	
	.4		*T* Limits	X		
QW-408 Gas	.1	±	Trail or ϕ comp.			X
	.2	ϕ	Single, mixture, or %	X		
	.3	ϕ	Flow rate			X
	.5	±	or ϕ Backing flow			X
	.9	−	Backing or ϕ comp.	X		
	.10	ϕ	Shielding or trailing	X		

Figure 9.3 Sample of some requirements for a WPS from ASME IX.

QW-256
WELDING VARIABLES PROCEDURE SPECIFICATIONS (WPS)
Gas Tungsten-Arc Welding (GTAW) (CONT'D)

Paragraph		Brief of Variables		Essential	Supplementary Essential	Nonessential
QW-409 Electrical Characteristics	.1	>	Heat input		X	
	.3	±	Pulsing I			X
	.4	ϕ	Current or polarity		X	X
	.8	ϕ	I & E range			X
	.12	ϕ	Tungsten electrode			X
QW-410 Technique	.1	ϕ	String/weave			X
	.3	ϕ	Orifice, cup, or nozzle size			X
	.5	ϕ	Method cleaning			X
	.6	ϕ	Method back gouge			X
	.7	ϕ	Oscillation			X
	.9	ϕ	Multi to single pass/side		X	X
	.10	ϕ	Single to multi electrodes		X	X
	.11	ϕ	Closed to out chamber	X		
	.15	ϕ	Electrode spacing			X
	.25	ϕ	Manual or automatic			X
	.26	±	Peening			X

Legend:

+ Addition	> Increase/greater than	↑ Uphill	← Forehand	ϕ Change
− Deletion	< Decrease/less than	↓ Downhill	→ Backhand	

Figure 9.3 (*Continued*)

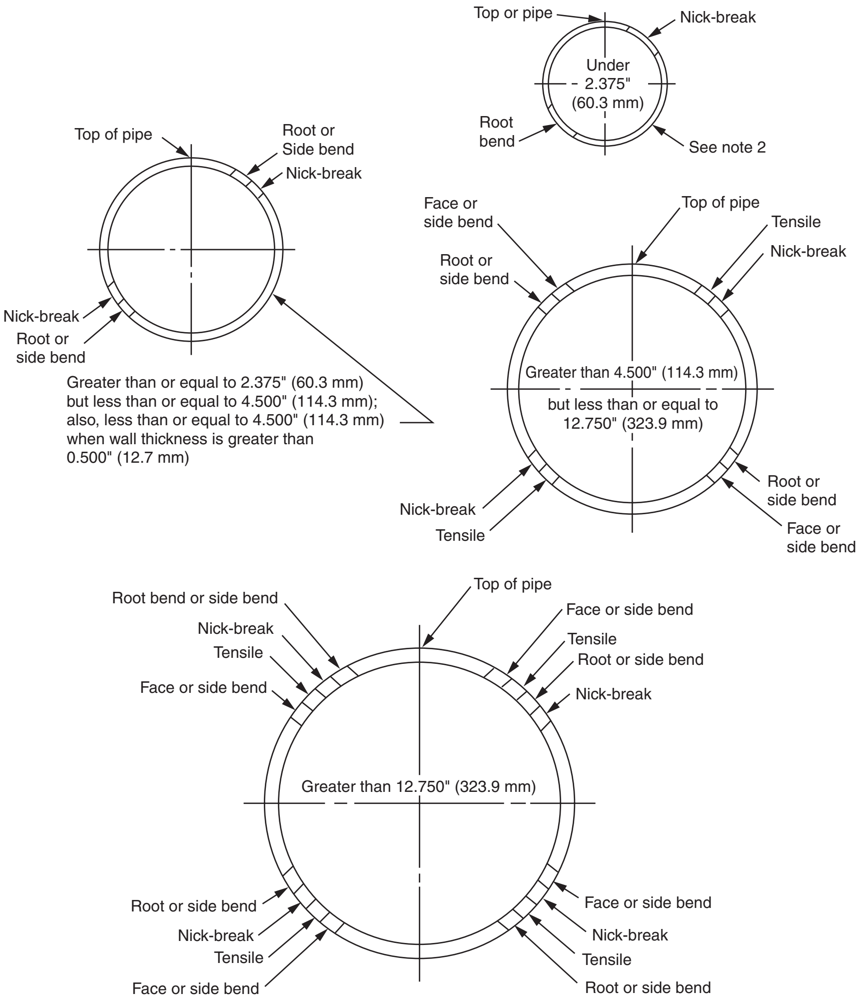
The requirement to use a qualified welder is somewhat simpler. First the welder who made the WPS that passed the PQR is automatically qualified. A welder using the procedure can be qualified with a simpler test. One of the major requirements is that a record be kept of the

Table 2—Type and Number of Test Specimens for Procedure Qualification Test

Outside Diameter of Pipe		Number of Specimens					
Inches	Millimetres	Tensile Strength	Nick-Break	Root Bend	Face Bend	Side Bend	Total
		Wall Thickness ≤ 0.500 inch (12.7 mm)					
< 2.375	< 60.3	0[b]	2	2	0	0	4[a]
2.375–4.500	60.3–114.3	0[b]	2	2	0	0	4
> 4.500–12.750	114.3–323.9	2	2	2	2	0	8
> 12.750	> 323.9	4	4	4	4	0	16
		Wall Thickness > 0.500 inch (12.7 mm)					
≤ 4.500	≤ 114.3	0[b]	2	0	0	2	4
> 4.500–12.750	> 114.3–323.9	2	2	0	0	4	8
> 12.750	> 323.9	4	4	0	0	8	16

[a]One nick-break and one root-bend specimen shall be taken from each of two test welds, or for pipe less than or equal to 1.315 inches (33.4 mm) in diameter, one full-section tensile-strength specimen shall be taken.
[b]For materials with specified minimum yield strengths greater than 42,000 psi (290 MPa), a minimum of one tensile test shall be required.

Figure 9.4 Sample of some requirements for procedure qualification from API 1104.



NOTES:

1. At the company's option, the locations may be rotated, provided they are equally spaced around the pipe; however, specimens shall not include the longitudinal weld.
2. One full-section tensile specimen may be used for pipe with a diameter less than or equal to 1.315 in. (33.4 mm).

Figure 9.4 (*Continued*)

welder's use of that procedure. And to remain qualified, the welder must have used it successfully within a specified time range.

It is common practice on projects to require approval of the WPS that is intended to be used for a particular weld or set of welds. A copy of the WPS and the attendant PQR is submitted to the owner project manager by the welding organization. It is reviewed and approved, or suggested changes, which may require requalification, are made. Once the approval process is completed, the welding may begin.

In an operation of any size, that is, one of many welders, a tremendous recordkeeping job is required to keep the process under control. In recent years, software has been developed that does the recordkeeping in the computer. Accurate input is required, but the massive paperwork job is reduced.

Another type of relief evolved in the late 1990s, known as *standard welding procedure specifications* (SWPSs). The exact rules can be found in Article V of Section IX as well as a listing of the SWPSs that are accepted. There are limitations to their use. These limitations should be checked in the section or the particular codebook which may add other limitations. However, where such limitations do not preclude the use, the SWPS would potentially provide relief to the welding organization regarding the cost and time of developing a WPS.

Also note that specific codes have requirements for the development of similar proof of joining methods that are appropriate for nonmetals. These are generally joined by a socket type of arrangement and the use of an adhesive appropriate to the specific nonmetal. These requirements will be mentioned at least by reference to the appropriate paragraph in the B31 code that specifies requirements.

Welding requirements for buried pipe

While B31.8 identifies the issues regarding welding and fabrication details, B31.4 and B31.11 also set requirements for construction welding and assembly. All have concerns about getting the pipes to be used to the location where they will be installed.

The general requirement is to transport the pipe in accordance with API RP 5L1. However, B31.8 also allows API RP5LW, and B31.11 allows API RP 5L5 or 6. These are API-recommended practices for means of protecting pipe in shipment.

The purpose is to eliminate accidents that will distort, dent, flatten, gouge, or notch the pipe. It is also to protect any coating that may have been applied prior to shipping. Code B31.8 in Paragraph 816 specifically requires an additional hydrostatic test, with the severity being determined by the location class in which the pipe is to be installed.

All three codes allow the reuse of pipe or unknown pipe with restrictions. Code B31.8 has the most comprehensive requirements, and they are summarized in Table 9.1. Codes B31.4 and B31.11 have requirements listed in their Paragraphs 405.2.1 and 1105.2.1, respectively.

There are additional rules prescribing the conditions in which to reuse ductile iron pipe and plastic piping. These rules are found in Paragraph 817.2 for ductile iron pipe and Paragraph 817.3 for plastic pipe.

TABLE 9.1 B31.8 Requirements for Reused Pipe or Unknown Pipe

Requirement	New or used pipe, unknown specification	Used pipe, known specification
Inspection	Note 1	Note 1
Bending properties	Note 2	NA
Thickness	Note 3	Note 3
Longitudinal joint factor	Note 4	Note 4
Weldability	Note 5	NA
Surface defects	Note 6	Note 6
Yield strength	Note 7	NA
S value	Note 8	NA
Hydrostatic test	Note 9	Note 9

NOTES: NA means not required.

1. Pipe shall be cleaned and inspected for roundness, straightness, and defects.
2. NPS 2 and smaller shall be subject to a specified bend test with no cracks or weld openings. For pipe larger than NPS 2, flattening test(s) per appendix H of this Code shall be made. The number of tests varies according to the number of lengths (a standard length is 20 feet) in the lot of pipe to be used.
3. Measure thickness at quarter points of each end. Determine the number of tests and type variable according to knowledge of the lot of pipe.
4. If known, *E* from Table 841.115A may be used; otherwise, it is specified as 0.6 for NPS 4 and smaller, 0.8 for larger than NPS 4.
5. Basically, this is a weld test per API 1104 with the most severe conditions met in the field. Several specific caveats regarding amount of pipe, size of pipe, or chemical tests are laid out in Section IX.
6. Pipe is to be examined and qualified per Paragraph 841.24. This paragraph gives specific instructions for the inspection and repair of this type of defect up to and including the cutting out and replacing of pipe deemed unrepairable by the examination and criteria.
7. The value of 24,000 psi yield shall be used, or a specific set of tensile tests shall be performed as prescribed by API 5L with a specified but variable number of tests depending on lot size.
8. Use the 24,000-psi strength. Or if the alternate material tests are made, use the lesser of 80 percent of the average of the tests or the minimum value of any yield strength test. In no case is it more than 52,000 psi.
9. Either a length-by-length or an after-installation hydrostatic test is carried out before placement in service.

Each of the pipeline codes sets out descriptions and requirements regarding the damage found when one is examining and inspecting pipe prior to installation. Those paragraphs are listed in Table 9.2.

Each of the codes establishes acceptable weld joint designs for the welds. They are basically the same for any code, and the major pictures of acceptable design are replicated in Fig. 9.5. These include equal

TABLE 9.2 Paragraphs on Damage

Code	Paragraphs
B31.4	434.4, 434.5
B31.8	841.24 and all subparagraphs Nonmandatory Appendix R
B31.11	1134.4, 1134.5

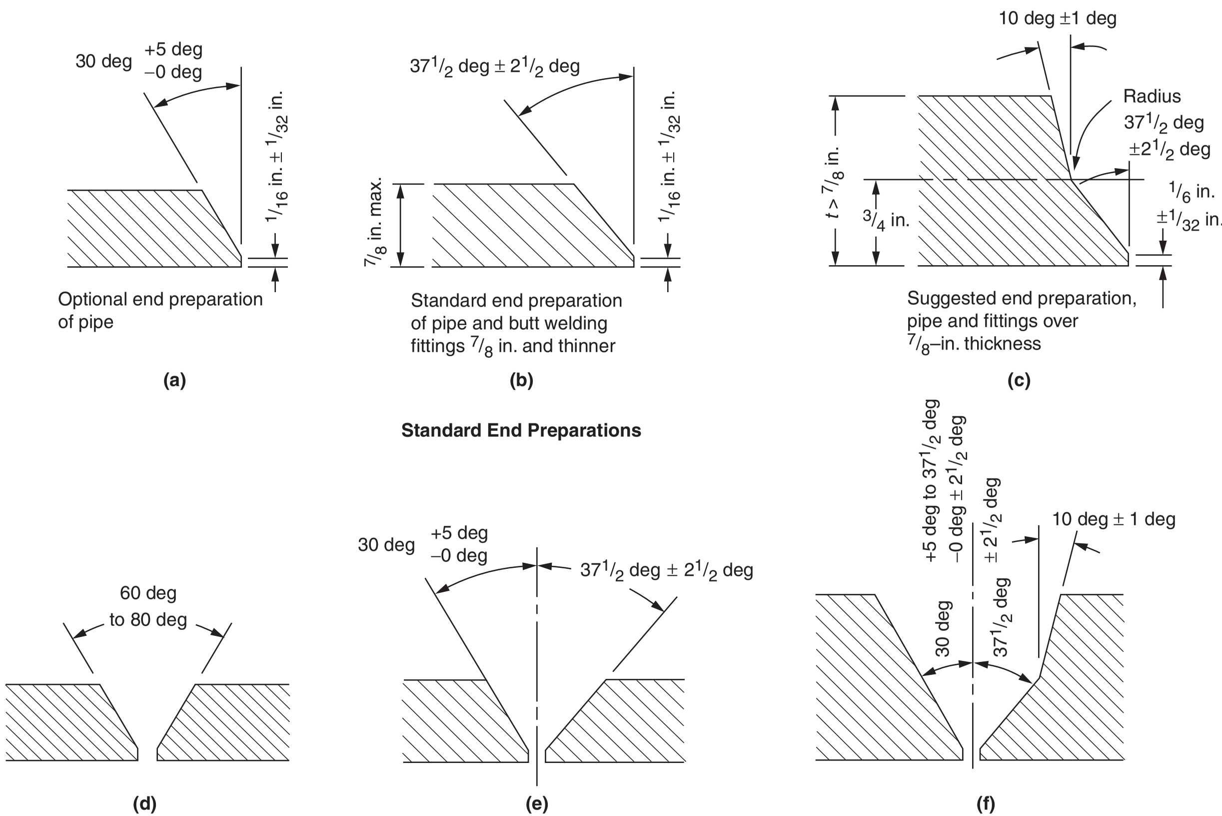
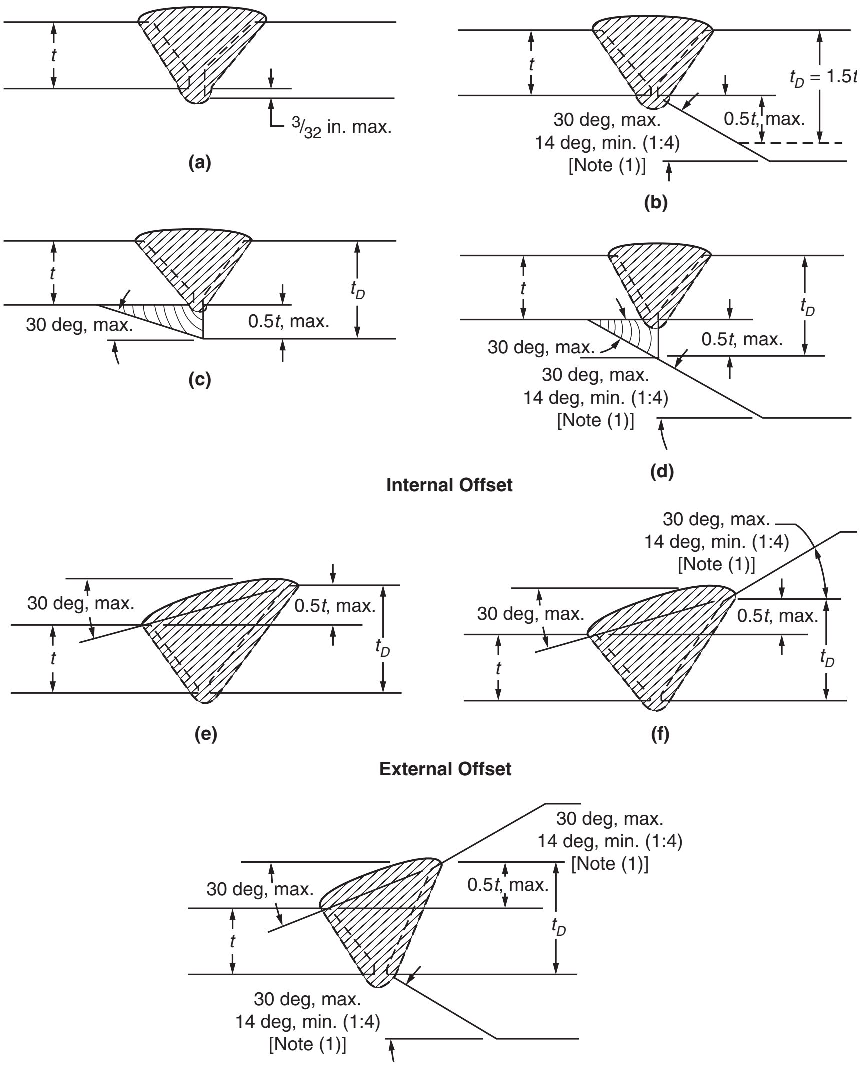


Figure 9.5 Some examples of weld preparations and types of welds that meet Code requirements. In any code others may be acceptable.



(g) Combination Offset

NOTE:

(1) No minimum when materials joined have equal specified minimum yield strengths.

Figure 9.5 (*Continued*)

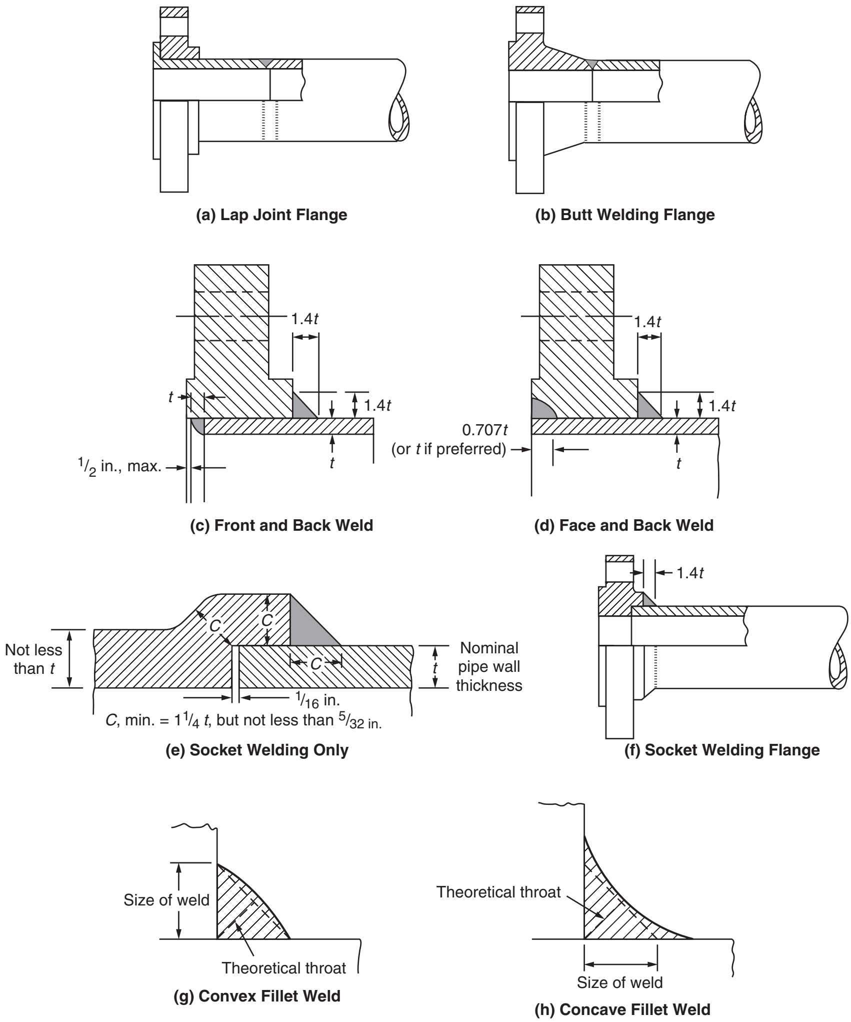
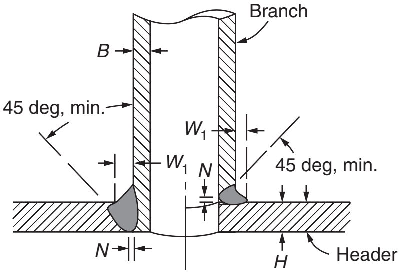


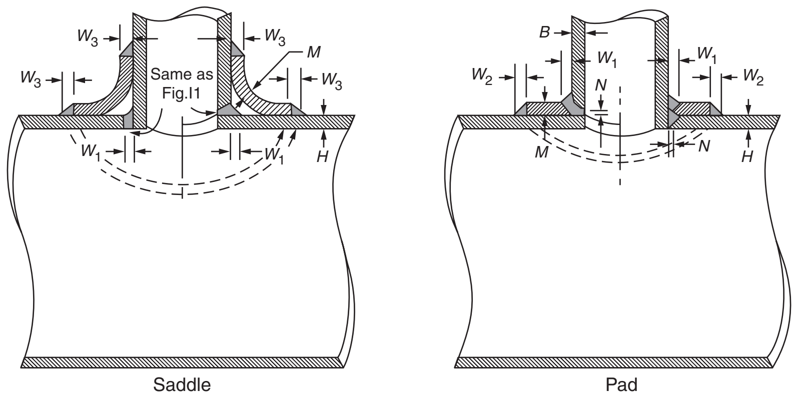
Figure 9.5 (*Continued*)



GENERAL NOTES:

(a) When a welding saddle is used, it shall be inserted over this type of connection.
(b) $W_1 = {}^{3B}/_8$, but not less than $^1/_4$ in.
(c) $N = {}^1/_{16}$ in. min., $^1/_8$ in. max., unless back welded or backing strip is used.

Fig. I1 Welding Details for Openings Without Reinforcement Other Than That in Header and Branch Walls

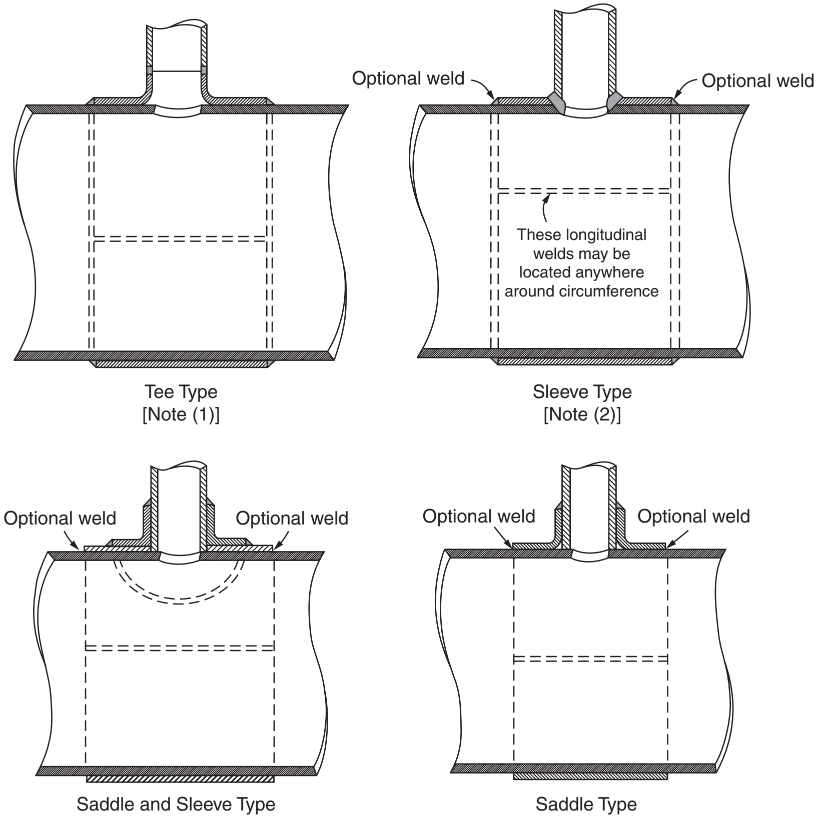


W_1 min. = $^{3B}/_8$, but not less than $^1/_4$ in.
W_2 min. = $^M/_2$, but not less than $^1/_4$ in.
W_3 min. = M, but not greater than H
$N = {}^1/_{16}$ in. min., unless back welded or backing strip is used

GENERAL NOTES:

(a) All welds to have equal leg dimensions, and a minimum throat = 0.707 X leg dimension.
(b) If M is thicker than H, the reinforcing member shall be tapered down to the header wall thickness.
(c) Provide hole in reinforcement to reveal leakage in buried welds and to provide venting during welding and heat treatment. [See para. 831.41 (h).]

Figure 9.5 (*Continued*)



NOTES:

(1) Since fluid pressure is exerted on both sides of pipe metal under tee, the pipe metal does not provide reinforcement.

(2) Provide hole in reinforcement to reveal leakage in buried welds and to provide venting during welding and heat treatment. [See para. 831.41 (h).] Not required for tee type.

Figure 9.5 (*Continued*)

wall-weld preparations which are most common in pipe-to-pipe butt welds. It includes unequal wall-weld preparations which are most common when a piece of equipment or fitting is attached to a pipe of higher yield strength. And the codes specify ways to attach flanges of the different styles. There are acceptable reinforcement types of welds. These come in both local reinforcement and full encirclement types. The local type has specific restrictions as outlined in the design portion.

Important aspects of the welding process are involved in preheating the specimen to be welded and stress-relieving that weld after the weld is completed. Both of these processes are expensive, especially when the welding is done in the field, and so the requirement is minimized whenever possible. Both reduce the probability of subsequent failure.

Because pipeline or buried pipe is made of carbon or low-carbon alloy, the requirements are relatively simple to follow. This is considerably different from aboveground piping, which can be made from myriad materials and therefore has more complex requirements. Any preheat, interpass temperature, or postweld heat treatment should be spelled out in the WPS.

The amount of preheat is specifically required when the carbon content by ladle analysis is above 0.32 percent. The pipelines use a very simple carbon equivalent expression, which is C + ¼ Mn. If that is above 0.65 percent, preheat is again required. The control of the interpass temperature is less specific. These elements work to reduce the amount of strain that is put in by the heating and cooling that would occur in the welding process by keeping those temperatures more nearly steady.

Postweld heat treatment is used to relax any residual stresses that have built up due to the weld process and to some extent the quickness of the cooling when welding is completed. The requirement for postweld is based on the amount of effective throat of the weld. Basically, when that throat is at 1¼ in. or larger, postweld heat treatment is expected. There are demonstrable ways that postweld can be waived. They require specific demonstration that it is acceptable. Whenever postweld heat treatment is specified, follow Section VIII for the material involved.

The actual ditching, and requirements for that, were covered in the discussion on buried pipe support.

Welding for aboveground piping

To reaffirm, Codes B31.1, B31.3, B31.5, and B31.9 do *not* offer the option of using API 1104 as the qualification for welding standard. The only acceptable standard is ASME BPV, Section IX. The previous discussion covered the general requirements of both.

The main difference is that Section IX covers many more types of materials, processes, including brazing, and filler metals to accommodate all the materials that might be utilized by these and the BPV codes. A secondary difference is that the majority of welds for aboveground piping are done in a weld fabrication shop rather than in the field. Certainly, as in most cases, some field welding is done in aboveground piping, and some shop welding is done in buried piping codes. Figure 9.6 shows a typical fabrication shop and the variety of spools of pipe it might contain.

One of the interesting things about Section IX is that it does not identify the materials specifically. It groups materials of like weldability,

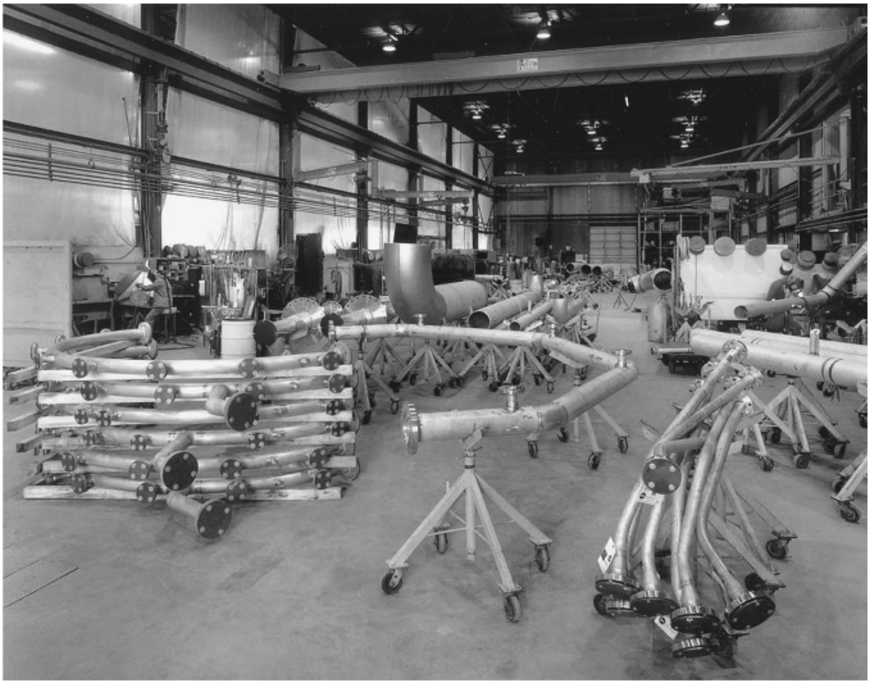


Figure 9.6 Typical large fabrication shop floor. (*Photograph courtesy of TEAM Industries, Inc., Kaukauna, WI*).

composition, brazeability, and mechanical properties into P numbers and group numbers within those P numbers. For materials that are acceptable to the B31 codes but not in Section II, the main BPV materials book, it assigns S numbers and groups within the S numbers. The S numbers are not mandatory but are used by B31.

This assigning of materials into groups does not mean that the materials in a particular group may be substituted for one another. It does mean that if a welding organization has a successful WPS within a particular group, the company can use it for other materials that have the same P/S number. This constitutes reducing the number of separate WPS, PQR, and welder qualifications that must be developed. Figure 9.7 reproduces an example page from Section IX which shows how this system is listed.

This constitutes the salient factors that were not covered in the general discussion previously. It pays to repeat that the B31 codes, including the pipeline codes, recognize ASME, Section IX, as governing the welding requirements and in particular for the aboveground codes.

All the aboveground codes (ABCs) allow backing rings with limitations. The main limitation is that any preparation for the rings may not reduce

QW/QB-422 FERROUS P-NUMBERS AND S-NUMBERS
Grouping of Base Metals for Qualification

GENERAL NOTE: To convert from ksi to MPa, multiply tensile strength in table by 6.9.

				Welding				Brazing			
Spec. No.	Type or Grade	UNS No.	Minimum Specified Tensile, ksi	P-No.	Group No.	S-No.	Group No.	P-No.	S-No.	Nominal Composition	Product Form
SA-36	...	K02600	58	1	1	...	...	101	...	C–Mn–Si	Plate, bar, & shapes
SA-53	Type F	...	48	1	1	...	...	101	...	C	Furnace welded pipe
SA-53	Type S, Gr. A	K02504	48	1	1	...	...	101	...	C	Smls. pipe
SA-53	Type E, Gr. A	K02504	48	1	1	...	...	101	...	C	Resistance welded pipe
SA-53	Type E, Gr. B	K03005	60	1	1	...	...	101	...	C–Mn	Resistance welded pipe
SA-53	Type S, Gr. B	K03005	60	1	1	...	...	101	...	C–Mn	Smls. pipe
SA-105	...	K03504	70	1	2	...	...	101	...	C–Si	Flanges & fittings
SA-106	A	K02501	48	1	1	...	...	101	...	C–Si	Smls. pipe
SA-106	B	K03006	60	1	1	...	...	101	...	C–Si	Smls. pipe
SA-106	C	K03501	70	1	2	...	...	101	...	C–Si	Smls. pipe
A 108	1015 CW	G10150	60	...	...	1	1	...	101	C	Bar
A 108	1018 CW	G10180	60	...	...	1	1	...	101	C	Bar
A 108	1020 CW	G10200	60	...	...	1	1	...	101	C	Bar
SA-134	SA283 Gr. A	...	45	1	1	...	...	101	...	C	Welded pipe
SA-134	SA283 Gr. B	...	50	1	1	...	...	101	...	C	Welded pipe
SA-134	SA283 Gr. C	K02401	55	1	1	...	...	101	...	C	Welded pipe
SA-134	SA283 Gr. D	K02702	60	1	1	...	...	101	...	C	Welded pipe
SA-134	SA285 Gr. A	K01700	45	1	1	...	...	101	...	C	Welded pipe
SA-134	SA285 Gr. B	K02200	50	1	1	...	...	101	...	C	Welded pipe
SA-134	SA285 Gr. C	K02801	55	1	1	...	...	101	...	C	Welded pipe
SA-135	A	...	48	1	1	...	...	101	...	C	E.R.W. pipe
SA-135	B	...	60	1	1	...	...	101	...	C	E.R.W. pipe
A 139	A	...	48	...	...	1	1	...	101	C	Welded pipe
A 139	B	K03003	60	...	...	1	1	...	101	C	Welded pipe
A 139	C	K03004	60	...	...	1	1	...	101	C	Welded pipe
A 139	D	K03010	60	...	...	1	1	...	101	C	Welded pipe
A 139	E	K03012	66	...	...	1	1	...	101	C	Welded pipe
A 148	90–60	...	90	...	...	4	3	...	103	...	Castings

Figure 9.7 Sample page showing how section IX groups various materials.

the wall thickness required by the design process for that code. The same allowance with some limitations is given for consumable inserts.

End preparations for the welding are major considerations, and they all follow a general pattern which is summarized here. Table 9.3 gives the paragraphs that outline any specific limitations or requirements.

- Reasonably smooth or arc cutting is accepted.
- B16.25 is referenced as to acceptable end preparation dimensions; B31.3 shows some additional J bevels.
- Boring to align the ends may not result in less than minimum thickness.
- Appropriate analysis weld metal may be deposited on the ID or OD to give sufficient metal for machining.
- Surfaces shall be clean and free of detrimental material for welding.
- Inside diameters shall be aligned as accurately as possible, preferably within 2.0 mm or 1⁄16 in. See Fig. 9.8; note that B31.3 allows the WPS to define misalignment. The figure is taken from B31.5 as typical.
- WPS defines the root opening.
- For socket weld, the fit should conform to the applicable standard, and the maximum diameter clearance should be 2.0 mm or 0.80 in. or less. A 1⁄16-in. bottom gap is required. See Fig. 9.9 from B31.9 taken as typical.
- The weld profile for unequal-thickness butt joints is also found in B16.25 and is shown in B31.1, which has a few tweaks for that but is typical of B16.25 in the main. See Fig. 9.10, showing a weld maximum envelope.

TABLE 9.3 Applicable Weld Requirement Paragraphs by Code

Code	Paragraphs
B31.1	127.1, 127.2, 127.3, 127.4, 127.5, 127.6, 128 and all subparagraphs for brazing
B31.3	328 and all subparagraphs for welds A328, A329, and all subparagraphs for plastic M328 and all subparagraphs K328 and all subparagraphs
B31.5	527 and all subparagraphs 528 and all subparagraphs for brazing and soldering
B31.9	527 and all subparagraphs 528 and all subparagraphs for brazing and soldering

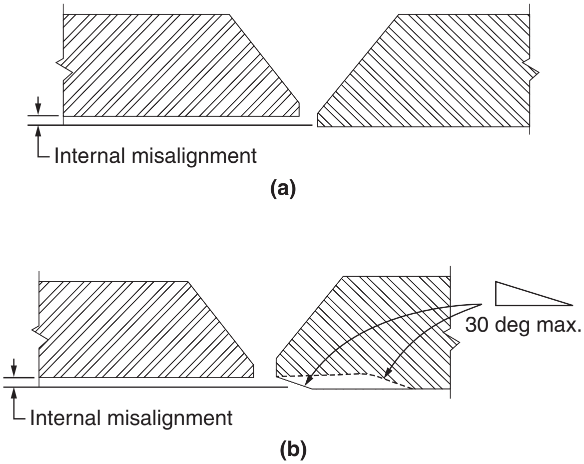


Figure 9.8 Fig. 527.2.1-B Internal trimming for butt welding of piping components with internal misalignment.

- Branch connections set on the pipe shall be contoured to fit within the parameters of the WPS. Those inserted shall be similarly fitted to at least the inside surface of the pipe. Their openings shall fit within the lesser of 3.2 mm, ⅛ in., or $0.5T_b$. This is spelled out in B31.3, which probably has more branches for their anticipated processes.
- Fillet welds may vary from convex to concave.
- Butt welds shall be complete-penetration welds.

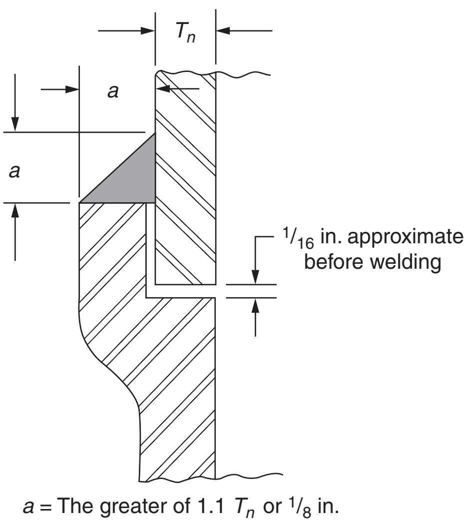
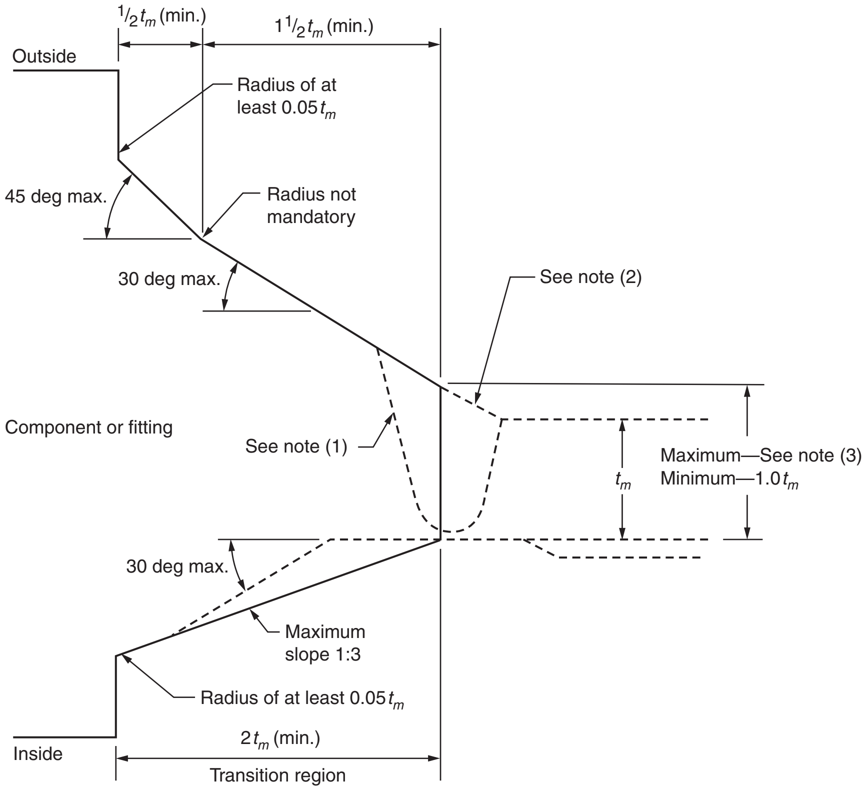


Figure 9.9 Typical socket weld diagram.



GENERAL NOTES:

(a) The value of t_m is whichever of the following is applicable:
 (1) as defined in para. 104.1.2(A);
 (2) the minimum ordered wall thickness of the cylindrical welding end of a component or fitting (or the thinner of the two) when the joint is between two components.

(b) The maximum envelope is defined by solid lines.

NOTES:

(1) Weld is shown for illustration only.

(2) The weld transition and weld reinforcement shall comply with paras. 127.4.2(B) and (C.2) and may be outside the maximum envelope.

(3) The maximum thickness at the end of the component is:
 (a) the greater of (t_m + 0.15 in.) or 1.15t_m when ordered on a minimum wall basis;
 (b) the greater of (t_m + 0.15 in.) or 1.10t_{nom} when ordered on a nominal wall basis.

Figure 9.10 Fig. 127.4.2 Welding end transition—maximum envelope.

- No welding is done if there is impingement of the weld area with rain, snow, or sleet.
- Weld surface shall be sufficiently free of coarse ripples and other surface imperfections that would hinder any nondestructive examination.
- Limits are placed on reinforcement and undercut and will be discussed in Chap. 10 on inspection, examination, and testing.
- Any seal welds of threaded joints shall cover the threads and be done by a qualified welder.

The above paragraphs also include the WPS and welder record requirements. There are descriptions of several acceptable weld configurations that have not been given in this discussion. It is important to note that a particular sketch or picture showing a specific weld configuration does not prohibit another equally strong weld. The use of the words *some acceptable* is meant to give guidelines.

This is also true of the shapes shown in the drawings in the codes and this book. Those drawings may be taken as generic and not specific. It is not meant to restrict the shapes to only those shown. The ASME Code sets out the means and methods that will produce an acceptably safe result.

Preheat requirement of ABCs

As shown in the discussion of welding for buried pipe, certain compositions of carbon pipe would require some preheat. As might be expected when one is dealing with many varieties of materials beyond carbon steel, those requirements would have greater variety. The expectation is correct.

The ABCs follow the P/S number groupings to set the preheat requirements. Each book has a somewhat different way of defining when and what the preheat requirements are. For instance, B31.1 does it subparagraph by subparagraph. Paragraphs 131.4 through 131.5 spell out a preferred temperature for a P group and then tell how the preheat may be modified, if desired.

It must be remembered that the preheat temperature is an essential variable in the WPS. The temperature used shall be set in the WPS, and if that preheat is lowered by more than 56°C or 100°F, the procedure must be requalified.

B31.3 has a table showing the preheat temperatures the committee either recommends or requires. Those temperatures are listed as minimum. As can be deduced from the essential variable requirement for retesting, if the temperature is lowered, a higher temperature might be desirable. It would be logical to set the preheat at the 100° above the minimum to forgo dropping below the minimum and stay within the restrictions of Section IX. That chart is reproduced here for convenience as Fig. 9.11.

Code B31.5 has a similar table, 531.2.1. That table includes both the preheat requirements and the postweld heat treatment requirements, if any. Fewer materials are used in B31.5, so its table is less complete than the one in B31.3. Both refer to P/S numbers and have basically the same temperatures, given the caveats mentioned in the discussion of B31.3.

Code B31.9 has a very short paragraph, Paragraph 931, stating that the materials recognized by that code do not require heat treatment for welding processes. It then states that if the design requires heat treatment, the

TABLE 330.1.1
PREHEAT TEMPERATURES

Base Metal P-No. or S-No. [Note (1)]	Weld Metal Analysis A-No. [Note (2)]	Base Metal Group	Nominal Wall Thickness		Specified Min. Tensile Strength, Base Metal		Min. Temperature			
							Required		Recommended	
			mm	in.	MPa	ksi	°C	°F	°C	°F
1	1	Carbon steel	< 25	< 1	≤ 490	≤ 71	...	...	10	50
			≥ 25	≥ 1	All	All	...	...	79	175
			All	All	> 490	> 71	...	...	79	175
3	2, 11	Alloy steels, Cr ≤ ½%	< 13	< ½	≤ 490	≤ 71	...	...	10	50
			≥ 13	≥ ½	All	All	...	...	79	175
			All	All	> 490	> 71	...	...	79	175
4	3	Alloy steels, ½% < Cr ≤ 2%	All	All	All	All	149	300	...	...
5A, 5B, 5C	4, 5	Alloy steels, 2¼% ≤ Cr ≤ 10%	All	All	All	All	177	350	...	...
6	6	High alloy steels martensitic	All	All	All	All	...	...	149[3]	300[3]
7	7	High alloy steels ferritic	All	All	All	All	...	...	10	50
8	8, 9	High alloy steels austenitic	All	All	All	All	...	...	10	50
9A, 9B	10	Nickel alloy steels	All	All	All	All	...	...	93	200
10	...	Cr-Cu steel	All	All	All	All	149–204	300–400	...	...
10I	...	27Cr steel	All	All	All	All	149[4]	300[4]	...	...
11A SG 1	...	8Ni, 9Ni steel	All	All	All	All	...	...	10	50
11A SG 2	...	5Ni steel	All	All	All	All	10	50	...	...
21–52	...	...	All	All	All	All	...	...	10	50

NOTES:
(1) P-Number or S-Number from BPV Code, Section IX, QW/QB-422.
(2) A-Number from Section IX, QW-442.
(3) Maximum interpass temperature 316°C (600°F).
(4) Maintain interpass temperature between 177°–232°C (350°F–450°F).

Figure 9.11 Sample preheat requirements from B31.3.

WPS should reflect that requirement. So basically the code is silent on the subject of preweld and postweld heat treatment requirements.

In any welding, it is preferable to make a complete weld and let that weld cool per its requirements. And during that welding time, the preheat shall be maintained. In some cases, there is a preference to have a maximum interpass temperature. The WPS would state that also.

In a real-world situation, there comes a time when the weld process must be interrupted. Both B31.1 and B31.3 address the issue of what to do in such circumstances. Code B31.3 acknowledges that it happens and that steps must be taken to prevent detrimental effects to the piping. When restart is begun, the preheat temperature must be achieved. (B31.3 Paragraph 330.2.4).

Because B31.1 generally works with heavier and more sensitive materials, it addresses the problem of interrupted welding more thoroughly. Paragraph 131.6 spells out those requirements. For specific P numbers, it requires that the minimum preheat temperature be maintained at all times unless certain requirements are met.

In any case, if a preheat is required, then a means of determining that it has been accomplished is required. It is good that temperature-indicating crayons exist. One can choose the proper crayon, mark the pipe, and when that temperature is reached, there is an indication. Thermo-couples and pyrometers are also allowed. See Paragraph 131.3 in B31.1 or Paragraph 330.1.3 in B31.3.

Postweld heat treatment

Postweld heat treatment may be required by the codes because of the P group into which a material falls or a determined thickness or both. Recall that B31.9 as a code does not require postweld heat treatment for the materials it recognizes. That code defers to the WPS if any postweld is required. In any case, if a postweld heat treatment is performed, it must be reflected in the WPS.

The other three codes—B31.1, B31.3, and B31.5—produce a chart that states that code's temperature and holding time for a particular P group of materials. The chart also states, if applicable, when the thickness governs. A material thickness in a particular situation may not govern while a thicker material in that same group might.

There is a major difference in this requirement between B31.1 and B31.3. Paragraphs 132.4.1 and 132.4.2 of B31.1 are quite simple. The determining thickness is defined as the lesser of

- The thickness of the weld
- The thicker of the two materials being joined

They further define the thickness of the weld as follows:

- For groove welds, the thicker of the two adjoining ends after weld preparation, including any ID machining
- For fillet welds, the throat thickness of the weld
- For partial penetration welds, the depth of the weld groove
- For repair welds, the depth of the cavity being repaired
- For branch welds, they refer to Code Figure 127.4.8 and provide formulas for calculating that thickness for each of the details in that figure.

Code B31.3 in Paragraph 331.1.3 calls for the determination of the governing thickness with some exceptions, which often preclude postweld heat treatment for certain applications. First it asserts the thickness to be that of the thicker component measured at the joint except as follows:

- For branch connections, metal (other than weld metal) that is added as reinforcement shall not be considered until that thickness is twice the minimum material thickness requiring heat treatment.
- It refers to code Figure 328.5.4D and gives a formula for calculating the weld thickness in accord with the various sketches.
- It defines some exceptions by size of fillet throat by P number for pipe DN 50, NPS 2, and smaller plus when attaching external non pressure containing parts.
- It allows use of austenitic welding materials on ferritic materials provided the service conditions will not adversely affect the weldment.
- Paragraphs K331.1.1 and K331.1.3 have more stringent requirements.

Code B31.5 is less specific about the governing thickness. In Chapter 531.3.8 it specifies that the thicker of the pipes being joined determines the code requirements. It does allow that for attachments such as lugs, the fillet size may be used.

As a rule, a furnace type of postweld heat treatment is preferred. However, all three codes do allow *local heat treatement*. In doing so, they require that the entire joint be covered and heated, and they define a zone around that joint to be included. The respective paragraphs are as follows:

- For B31.1, it is Paragraph 132.7.
- For B31.3, it is Paragraph 331.2.6.
- For B31.5, it is Paragraph 531.3.9.

Code B31 includes some helpful guidance which, while not specific in the other two codes, can be applied because the temperatures defined are given as minimums. The code points out that while the temperature given as minimum may be exceeded, it should not be above the lower critical temperature.

Code B31.1 provides a table that provides guidance in that respect. As an example B31.1 gives a holding temperature range of 1100 to 1200°F for P-1 material. The table specifies a temperature of 1340°F as the lower critical temperature. That table of lower critical temperature guidance is reproduced here as Fig. 9.12 for convenience.

One of the contentions of the postweld heat treatment is the verification of the temperature at which it is held. In some more stringent specifications, it is required that the metal temperature be verified by attaching thermocouples to the metal at specific spots. The codes do not specify to this degree. For a furnace, the temperature chart of the furnace gives information in this respect, and the time to ramp to the temperature correlates with the metal temperature.

The holding time at the temperature is something that is defined by the codes. This time is related to the governing thickness, which has been discussed. It is computed based on the soaking time it would take for the metal surface temperature to reach completely through the metal thickness. It is an important aspect of getting the residual stresses out of the piping being heat-treated.

A portion of the B31.1 and B31.3 figures that are provided is reproduced here as Fig. 9.13 so the reader can determine by comparison the differences in the two committees' approaches. The B31.5 chart does not include the materials that are common to this comparison.

B31.1 TABLE 129.3.2 APPROXIMATE LOWER CRITICAL TEMPERATURES

Material	Approximate Lower Critical Temperature (1), °F (°C)
Carbon Steel (P-No. 1)	1340 (725)
Carbon Molybdenum Steel (P-No. 3)	1350 (730)
1Cr–½Mo (P-No 4, Gr. No. 1)	1375 (745)
1¼Cr–½Mo (P-No. 4, Gr. No. 2)	1430 (775)
2¼Cr–1Mo, 3Cr–1Mo (P-No. 5A)	1480 (805)
5Cr–½Mo (P-No. 5B, Gr. No. 1)	1505 (820)
9Cr–½Mo (P-No. 5B, Gr. No. 2)	1490 (810)

NOTE:

(1) These values are intended for guidance only. The user may apply values obtained for the specific material in lieu of these values.

Figure 9.12 B31.1 Guidance on lower critical temperature.

TABLE 331.1.1
REQUIREMENTS FOR HEAT TREATMENT

Base Metal P-No. or S-No. [Note (1)]	Weld Metal Analysis A-Number [Note (2)]	Base Metal Group	Nominal Wall Thickness		Specified Min. Tensile Strength, Base Metal		Metal Temperature Range		Holding Time			Brinell Hardness, [Note (4)] Max.
									Nominal Wall [Note (3)]		Min. Time, hr	
			mm	in.	MPa	ksi	°C	°F	min/mm	hr/in.		
1	1	Carbon steel	≤ 19	$\le 3/4$	All	All	None	None	...	...	...	...
			> 19	$> 3/4$	All	All	593–649	1100–1200	2.4	1	1	...
3	2, 11	Alloy steels,	≤ 19	$\le 3/4$	≤ 490	≤ 71	None	None	...	...	...	...
		Cr $\le 1/2$%	> 19	$> 3/4$	All	All	593–718	1100–1325	2.4	1	1	225
			All	All	> 490	> 71	593–718	1100–1325	2.4	1	1	225
4[10]	3	Alloy steels,	≤ 13	$\le 1/2$	≤ 490	≤ 71	None	None	...	...	...	...
		$1/2$% < Cr $\le$ 2%	> 13	$> 1/2$	All	All	704–746	1300–1375	2.4	1	2	225
			All	All	> 490	> 71	704–746	1300–1375	2.4	1	2	225
5A,[10] 5B,[10] 5C[10]	4, 5	Alloy steels, ($2\frac{1}{4}$% $\le$ Cr $\le$ 10%)										
		$\le$ 3% Cr and $\le$ 0.15% C	≤ 13	$\le 1/2$	All	All	None	None	...	...	...	...
		$\le$ 3% Cr and $\le$ 0.15% C	> 13	$> 1/2$	All	All	704–760	1300–1400	2.4	1	2	241
		> 3% Cr or > 0.15% C	All	All	All	All	704–760	1300–1400	2.4	1	2	241
6	6	High alloy steels martensitic	All	All	All	All	732–788	1350–1450	2.4	1	2	241
		A 240 Gr. 429	All	All	All	All	621–663	1150–1225	2.4	1	2	241
7	7	High alloy steels ferritic	All	All	All	All	None	None	...	...	...	...
8	8, 9	High alloy steels austenitic	All	All	All	All	None	None	...	...	...	...
9A, 9B	10	Nickel alloy steels	≤ 19	$\le 3/4$	All	All	None	None	...	...	...	...
	...		> 19	$> 3/4$	All	All	593–635	1100–1175	1.2	$1/2$	1	...
10	...	Cr-Cu steel	All	All	All	All	760–816 [Note (5)]	1400–1500 [Note (5)]	1.2	$1/2$	$1/2$	...

Figure 9.13 Samples of differing heat treat requirements between B31.3 and B31.1.

TABLE 132
POSTWELD HEAT TREATMENT (CONT'D)

P-Number from Appendix A	Holding Temperature Range, °F (°C)	Holding Time Based on Nominal Thickness	
		Up to 2 in. (50 mm)	Over 2 in. (50 mm)
P-No. 9A Gr. No. 1	1100 (600) to 1200 (650)	1 hr/in. (25 mm) 15 min minimum	2 hr plus 15 min for each additional inch over 2 in. (50 mm)

GENERAL NOTES:

(A) PWHT is not mandatory for P-No. 9A material when welds on pipe or attachment welds to pipe comply with all of the following conditions:
   (1) a nominal material thickness of ½ in. (13.0 mm) or less;
   (2) a specified carbon content of the material to be welded of 0.15% or less;
   (3) a minimum preheat of 250°F (120°C) is maintained during welding.

(B) When it is impractical to PWHT at the temperature range specified in Table 132, it is permissible to perform the PWHT of this material at lower temperatures for longer periods of time in accordance with Table 132.1 but the minimum PWHT shall not be less than 1000°F (550°C).

P-Number from Appendix A	Holding Temperature Range, °F (°C)	Holding Time Based on Nominal Thickness	
		Up to 2 in. (50 mm)	Over 2 in. (50 mm)
P-No. 9B Gr. No. 1	1100 (600) to 1175 (630)	1 hr/in. (25 mm) 15 min minimum	2 hr plus 15 min for each additional inch over 2 in. (50 mm)

GENERAL NOTES:

(A) PWHT of P-No. 9B material is not mandatory for a nominal material thickness of ⅝ in. (16.0 mm) or less provided the Welding Procedure Qualification has been made using material of thickness equal to or greater than the production weld.

(B) When it is impractical to PWHT at the temperature range specified in Table 132, it is permissible to perform the PWHT of this material at lower temperatures for longer periods of time in accordance with Table 132.1, but the minimum PWHT temperature shall not be less than 1000°F (550°C).

P-Number from Appendix A	Holding Temperature Range, °F (°C)	Holding Time Based on Nominal Thickness	
		Up to 2 in. (50 mm)	Over 2 in. (50 mm)
P-No. 10H Gr. No. 1	...	...	...

GENERAL NOTE:

(A) Postweld heat treatment is neither required nor prohibited. If any heat treatment is performed after forming or welding, it shall be performed within the temperature range listed below for the particular alloy, followed by a rapid cool:

Alloy S31803	1870°F–2010°F
Alloy S32550	1900°F–2050°F

Figure 9.13 (*Continued*)

Assembly

There is some assembly of components in any piping system. When the codes talk about assembly, they are referencing things such as putting together spools of pipe that are flanged or connected in other mechanical ways. The codes express concern about connecting piping to equipment such as pumps.

The most frequent set of requirements is in regard to assembling flanges. It is relatively well known that flanges are susceptible to leakage due to improper assembly, misalignment, nonuniform tightening, improper gasket seating, and many other inadequacies in this assembly process.

Code B16.5, the ubiquitous flange standard, describes quite concisely the problem: "A flange joint is composed of separate and independent, although inter-related components: the flanges, the gasket, and the bolting, which are assembled by another influence, the assembler. Proper controls must be exercised in the selection and application for all these elements to attain a joint that has acceptable leak tightness." There is an ASME publication, ASME PCC-1, that discusses special techniques, such as controlled bolt tightening.

There are requirements for alignment of piping both for flange and for connection to equipment. The special requirements for flanges include the requirement that the gasket seating requires special attention and that the bolt should extend so as to get full threads on the nuts.

For all four codebooks, the appropriate paragraphs to review to determine specific requirements have the same relative number:

- B31.1 is Paragraph 135 and all subparagraphs.
- B31.3 is Paragraph 335 and all subparagraphs.
- B31.5 is Paragraph 535 and all subparagraphs.
- B31.9 is Paragraph 935 and all subparagraphs.

Code B31.9 contains requirements for things such as repair of joints that leak. It also contains guidance and requirements for borosilicate-lined pipe. These and other additional requirements may be attributed to the fact that building services are as much about plumbing as about welding pipe spools together. It is one place where local plumbing codes are most likely to have jurisdiction.

Chapter

10

Inspection, Examination, and Testing

General

This may be the least popular activity in the entire process of constructing a piping system. In another sense, it may be the most important to the owner and to the community around the system. The "best" design is not sufficient to ensure that an appropriate margin exists in the final facility.

History contains many cases of subsequent failure when the design was more than adequate. Investigation showed that the execution of the construction was not performed in accordance with the design. The codes recognize that this is an important issue—to ensure the actual construction is per the design. It is accomplished by separating the checking of the construction process into two responsibilities. One is examination by the fabricator; the other is inspection, which is the owner's responsibility.

Note that pipelines which have an oversight jurisdiction by the *Code of Federal Regulation* (CFR) have additional formal requirements. Also, certain parts of the B31.1 code are classified as boiler external piping and have in their requirements the oversight and inspection by authorized inspectors from the National Board.

Each of the codes has a nonmandatory appendix that suggests that the owner may invoke a quality standard or program which could also add documentation requirements. The ISO 9000 program is used as an example. One could also use something like the ASME quality program as outlined in the BPV Code section.

These programs do not necessarily impose additional requirements. They are a method of establishing process and procedure by which the fabricator demonstrates adherence to those procedures to ensure that all the checks and balances required are met. And it must be done in a traceable manner.

Some of the descriptions of code requirements liken the examinations, which are done by the fabricator, to quality control. They then liken the inspections, which are the owner's responsibility, to quality assurance. It should be understood that the requirements of the codes can be supplemented and added to by the contract. If so, the additional requirements should be agreed upon before the work begins.

The pipeline and buried piping codes do not make the separation between inspection and examination as explicit as do the aboveground codes. This is understood by the additional requirements imposed by the CFR.

An interesting note as to the importance of the inspector is that the codes established the requirements of the inspector and have done so from early editions. In the 2002 edition of B31.3 they also added requirements for the designer. These are similar to the qualifications of the inspector. The two sets of B31.3 requirements for designer and inspector are set out in Table 10.1 for comparison.

As one can see, they are related in many ways. A further definition of related experience includes design calculations for pressure, sustained and occasional loads, and flexibility analysis. In the case of the inspector, there is a further chapter that does not relieve the manufacturer, fabricator, or erector of the responsibility for

- Providing materials workmanship to the code requirements
- Performing all the required examinations
- Preparing the records of examinations and tests for the inspector

TABLE 10.1 Comparison of Inspector and Designer Requirements

Inspector	Designer
Designated by owner as agent. Is not representative or employee of fabricator, manufacturer, or erector unless they are owners.	Person(s) in charge of the engineering design and experienced in use of code. At least one of the following qualifications or approved by owner.
Qualifications	**Qualifications**
Not less than 10 years' experience in industrial pressure piping. Each 20 percent of completion toward engineering degree shall be considered 1 year of experience, up to 5 years total.	Experience in related pressure piping design; 4-year engineering degree + 5 years' experience. PE license in local jurisdiction and experience 2-year engineering associates degree + 10 years' experience. 15 years' experience.
Responsible for qualification of anyone he/she delegates to do inspection.	For B31.3, experience should be in B31.3 code.

Each of the codes recognizes that its approach is not the only acceptable approach to any problem. In various degrees, the codes allow the qualified designer to use other methods. These methods shall be consistent with the criteria of that code. Any such deviation requires acceptance by the owner.

It is understood that such methods may require more stringent construction, examination, and testing than expressed for the more simplified approach in the section book. That should be detailed in the engineering design when applicable. Code B31.3 in Paragraph 300(3) explicitly sets out that requirement.

Buried Piping Requirements

The requirements for buried piping are different enough from those of aboveground piping that they are again discussed separately. In many cases, the same general concerns are examined. Sometimes, the acceptance criteria are different; at other times, the details of a particular test are different.

The examination of welds is one case in which many of the differences are due to the field erection and welding tendencies of the more common method of construction. The rules reflect such concerns. The relevant paragraphs are

- For B31.4, Paragraph 434.8.5
- For B31.8, Paragraph 826
- For B31.11, Paragraph 1134.8.5

In B31.4 or B31.8 all welds shall be visually inspected. This inspection must be performed by an inspector qualified through training or experience. Code B31.11 is silent on visual inspection requirements. The general inspection requirements are to be met by nondestructive methods. Or, welds can be selected and designated by the inspector for removal and destructively tested.

Each of the codes specifies that the acceptance criteria and methods must be in accord with API 1104. As mentioned in the discussion on welding, this standard includes procedures and acceptance criteria for the major forms of nondestructive testing. The standard is considered as complete in the requirements. Chapters 8, 9, 10, and 11 give the necessary requirements. An abbreviated table of contents, showing those chapters, is included here for reference (see Table 10.2).

These standards have specific requirements for when and what to examine regarding the location or type of weld in the pipeline. Table 10.3 shows the paragraphs listed and summarized by Code.

TABLE 10.2 Abbreviated Table of Contents for API-1104 Showing Testing Subjects

8.1	Rights of Inspection
8.2	Methods of Inspection
8.3	Qualification of Inspection Personnel
8.4	Certification of Nondestructive Testing Personnel
9	ACCEPTANCE STANDARDS FOR NONDESTRUCTIVE TESTING
9.1	General
9.2	Rights of Rejection
9.3	Radiographic Testing
9.4	Magnetic Particle Testing
9.5	Liquid Penetrant Testing
9.6	Ultrasonic Testing
9.7	Visual Acceptance Standards for Undercutting
10	REPAIR AND REMOVAL OF DEFECTS
10.1	Authorization for Repair
10.2	Repair Procedure
10.3	Acceptance Criteria
10.4	Supervision
10.5	Welder
11	PROCEDURES FOR NONDESTRUCTIVE TESTING
11.1	Radiographic Test Methods
11.2	Magnetic Particle Test Method
11.3	Liquid Penetrant Test Method
11.4	Ultrasonic Test Methods

TABLE 10.3 Examination Requirements by Code for Buried Pipelines

Factor	Code B31.4	Code B31.8	Code B31.11
Hoop stress < 10% (SMYS)	Not addressed	Repair or remove visual defects	Not addressed
Hoop stress > 20% Exceptions listed below	10% of daily welds 100% of circumference	Location class 1, 10% Location class 2, 15% Location class 3, 40% Location class 4, 75%	10% of daily welds 100% of circumference
Hoop stress > 20% in populated areas	100%	See above	100%
Crossings of all types	100%	See above	100%
Offshore, old girth welds on used pipe; tie-in welds not tested	100%	Not addressed	100%
Pipe < NPS 6, hoop stress < 40% (SMYS)	Not addressed	Not mandatory except visual and approved	Not addressed

Defects as found must be removed or repaired. This shall be in accordance with the requirements of API 1104. It shall be done by qualified welders.

Testing of buried pipe

All pipelines shall be tested for integrity after construction and before putting the line into operation. As might be expected, the tests vary according to use. The B31.4 and B31.11 test requirements are somewhat less stringent than those of B31.8, so they will be discussed separately. The relevant paragraphs are 437 and 1137.

These paragraphs include many details and should be perused before one conducts specific tests on components or unknown material. These elements are discussed in Paragraphs 437 and 1137, including subparagraphs and above and are defined as qualification tests for the various elements.

The major tests are different depending on whether the pipeline is to be operated above 20 percent of SMYS or below. There are some standard concerns. If equipment is not to be tested, it shall be disconnected or otherwise isolated during the test. In no case shall the test pressure be above that stipulated for the weakest element in the component's standard. Appropriate controls shall be tested to ensure that they will function correctly during the test.

There are several requirements or rules to be followed during the conduct of the test. This is applicable for systems at the 20 percent or above SMYS operating level. These rules are as follows:

- The test pressure shall be 1.25 times internal design pressure and held for not less than 4 hours.
- If the components are visually inspected during the test, no further tests are required for them.
- If for some reason a component is not visually inspected, the test pressure is lowered to 1.1 times the internal design pressure and held at that pressure for not less than 4 hours.
- API RP 1110 may be used for guidance.
- Water shall be used; there are exceptions listed.
- The pipeline may not be offshore.
- Each building within 300 ft must be unoccupied during the test unless the hoop stress is between 20 percent and 50 percent of the SMYS.
- For B31.4, the test section must be regularly patrolled and communication maintained.

TABLE 10.4 Test Requirements when Operating Pressure Produces Less than 20% SMYS

Factor	Hydrostatic	Pneumatic
Duration	1 hour	1 hour
Test pressure	1.25 times internal design pressure	100 psi (7 bar) or lesser of hoop stress at 25% (SMYS)

- Provisions for thermal expansion relief shall be made if the test section is subject to them.
- In cold weather the line and all components shall be drained to avoid damage due to freezing.
- Carbon dioxide lines shall be dewatered after the test to avoid any formation of corrosive compounds from CO_2 and water.

If the system is to be operated at less than 20 percent of the SMYS, a lesser test is allowed. That test may be hydrostatic or pneumatic. Those requirements are shown in Table 10.4.

These two codes require records in the files of the operating company to be kept for the life of the facility (Paragraphs 437.7 and 1137.7, respectively). These records include material specifications, route maps, as-built condition, pipe size, grade, wall thickness, weld seam (if any), manufacturer, coatings, and test data. The carbon dioxide line also requires toughness requirements.

Tests for B31.8

The requirements for B31.8 are somewhat more complex. This is primarily due to the location class varieties that are invoked. Also note that the break point is at a hoop stress of 30 percent of SMYS rather than 20 percent. There is an additional requirement for a pipeline that operates at less than 30 percent of hoop stress but above 100 psi, and another for a pipeline that operates below 100 psi.

Recognizing this compound set of requirements, the code published a table from which the base test requirements can be determined. Then the variations are set from that table. That base table is Table 841.322(f), reproduced here for convenience as Fig. 10.1. The standard length of the test is at least 2 hours.

The second category is at a hoop stress of less than 30 percent but an operating pressure greater than 100 psi. In this case, the first concern would be for pipes that are in class 1 locations. Then if the hoop stress is at 20 percent or more and the test medium is gas or air, the test pressure shall be in the range from 100 psi to one that produces a hoop stress of 20 percent of SMYS. As an alternative, the pressure may

Table 841.322(f) Test Requirements for Pipelines and Mains to Operate at Hoop Stresses of 30% or More of the Specified Minimum Yield Strength of the Pipe

1	2	3	4	5
Location Class	Permissible Test Fluid	Pressure Test Prescribed: Minimum	Pressure Test Prescribed: Maximum	Maximum Allowable Operating Pressure, the Lesser of
1 Division 1	Water	1.25 × m.o.p.	None	t.p. ÷ 1.25 or d.p.
1 Division 2	Water	1.1 × m.o.p.	None	t.p. ÷ 1.1 or d.p.
	Air	1.1 × m.o.p.	1.1 × d.p.	
	Gas	1.1 × m.o.p.	1.1 × d.p.	
2	Water	1.25 × m.o.p.	None	t.p. ÷ 1.25 or d.p.
	Air	1.25 × m.o.p.	1.25 × d.p.	
3 and 4 [Note (1)]	Water	1.40 × m.o.p.	None	t.p. ÷ 1.40 or d.p.

d.p. = design pressure
m.o.p. = maximum operating pressure (not necessarily the maximum allowable operating pressure)
t.p. = test pressure

GENERAL NOTE: This table defines the relationship between test pressures and maximum allowable operating pressures subsequent to the test. If an operating company decides that the maximum operating pressure will be less than the design pressure, a corresponding reduction in prescribed test pressure may be made as indicated in the Pressure Test Prescribed, Minimum, column. If this reduced test pressure is used, however, the maximum operating pressure cannot later be raised to the design pressure without retesting the line to the test pressure prescribed in the Pressure Test Prescribed, Maximum, column. See paras. 805.214, 845.213, and 845.214.

NOTE:
(1) For exceptions, see para. 841.322(d).

Figure 10.1 B31.8 Code Table 841.322(f).

be set at the 20 percent level and the test section walked and visibly checked.

For location class 2, 3, or 4, the code reverts to Table 841.322(f) with the following exceptions. If gas or air is used as the test medium, the maximum hoop stress may be set at the percentage of SMYS shown in Table 10.5.

If a pipeline is to operate at less than 100 psi, the rules are less stringent. Note that these would most likely be in gas mains or a distribution system. It is quite common for those pressures to be considerably less than 100 psi.

Code B31.8 in Paragraph 841.352 states that gas may be used as the test medium and that the maximum pressure available in the distribution system at the time of the test is acceptable. Even more important to the

TABLE 10.5 B31.8 Code Pressure Settings for Gas or Air Tests

Test medium	Class 2 max hoop stress, %	Class 3 max hoop stress, %	Class 4 max hoop stress, %
Air	75	50	40
Gas	30	30	30

tester is that a soap bubble test may be used to locate the leaks. This posits that all joints are accessible during the test.

In every case, should a leak be found, it shall be located and eliminated except that in Paragraphs 841.341 and 841.342 some guidance is given. Paragraph 841.342 cautions that the test procedure shall be selected to disclose all leaks, taking the volumetric content of the section and its location. These code words recognize that some distribution lines constitute a very small volume of gas which would not require a highly pecise test. If, however, it is in an area of high population density, the precision of the test should be increased. It further points out that such selection requires responsible and experienced judgment rather than numerical precision.

Paragraph 841.341 allows that "if it can be determined that no undue hazard to the public safety exists," it may not be necessary to locate and eliminate the leak. It should be pointed out that it is highly probable that these connections can be in plastic or threaded metallic pipe, and some leaks may be very small.

Examination and Inspection in Aboveground Codes

Aboveground piping by its very nature has more changes of direction, or branches, and is subject to more of the type of fluctuating loads that require flexibility anaylsis. For these reasons, the examination requirements tend to be more specific in these codes.

In addition, these codes recognize larger variety of examination and testing methods. One very apparent difference between buried and aboveground piping is that the aboveground codes spell out that which is often left to API-1104 in the buried pipelines. B31.3 addresses a greater variety of processes and fluids than any of the other ABC codes. As a result, it establishes a greater number of requirements.

A common requirement of all the ABCs is to define the qualifications of the nondestructive examiners (NDEs). The requirements of the owner's inspector have already been discussed. Each code requires that the NDEs be experienced and skilled in the methods they employ in the particular test. Each code has a different set of words to define this skill level, but B31.1 has the most general way that encompasses the requirements of the other three codes.

In Paragraph 136.3.2, entitled Qualifications of NDE Personnel, code requires that the personnel be certified as qualified for each method by a program established by the employer of the examiner. It then lists a set of minimum requirements:

- Instruction in the fundamentals of the method
- On-the-job training sufficient to assimilate the required knowledge
- Optical examination, at least yearly, to determine that capability
- Oral or written examination to verify the ability to perform the tests and interpret the results
- Keeping active in a method; if inactive for a year, recertification
- Recertification if the procedures have significantly changed

The paragraph then states that an alternative to the above as applicable would be SNT-TC-1A, or CP 189. It also allows AWS QC-1 for visual examination of welds. While the other codes do not go to the extent described above, the SNT-TC-1A is mentioned as acceptable evidence of the qualifications of NDEs.

Another universally accepted source of guidance and methodology, especially in establishing written procedures which are required as a means of proof of method, is BPV section V. They, when invoked, would also contain the acceptance criteria. B31.3 uses these articles to define its requirements, which are as follows:

- All articles found in BPVC, Section V
- For visual examination, Article 9
- For magnetic particle examination, Article 7
- For liquid penetrant examination, Article 6
- For radiographic examination, Article 2
- For ultrasonic examination, Article 5
- For eddy current testing (Chapter 9 requirement), Article 8
- For any special methods, the engineering design must specify.

Code B31.1 Table 136.4 (see Fig. 10.2) defines the minimum requirements by type of weld. Then in the paragraphs listed below the charts, it gives descriptions of the indications by type of examination and provides acceptance criteria to comply with the code. Those paragraphs are too extensive to be duplicated here. The reader looking for guidance and the acceptance standards in those areas is referred to the code itself. Figure 10.2 and the list of paragraphs in Table 10.6 will guide the reader to those.

Code B31.5 has a similar set of acceptance criteria and displays them in the paragraph mode. However, it sets acceptance more by type of joint or connection than by type of NDE. It does not recognize all types of NDE (Table 10.7).

TABLE 136.4
MANDATORY MINIMUM NONDESTRUCTIVE EXAMINATIONS FOR PRESSURE WELDS OR WELDS TO PRESSURE RETAINING COMPONENTS

Type Weld	Piping Design Conditions and Nondestructive Examination: Temperatures Over 750°F (400°C) and at All Pressures	Temperatures Between 350°F (175°C) and 750°F (400°C) Inclusive With All Pressures Over 1025 psig [7100 kPa (gage)]	All Others
Butt welds (girth and longitudinal) [Note (1)]	RT or UT for over NPS 2. MT or PT for NPS 2 and less [Note (2)]	RT or UT for over NPS 2 with thickness over $^3/_4$ in. (19.0 mm); VT for all sizes with thickness $^3/_4$ in. (19.0 mm) or less.	Visual for all sizes and thicknesses
Welded branch connections (size indicated is branch size) [Notes (3) and (4)]	RT or UT for over NPS 4. MT or PT for NPS 4 and less [Note (2)]	RT or UT for branch over NPS 4 and thickness of branch over $^3/_4$ in. (19.0 mm) MT or PT for branch NPS 4 and less with thickness of branch over $^3/_4$ in. (19 mm) VT for all sizes with branch thickness $^3/_4$ in. (19.0 mm) or less	VT for all sizes and thicknesses
Fillet, socket, attachment, and seal welds	PT or MT for all sizes and thicknesses [Note (5)]	VT for all sizes and thicknesses	VT for all sizes and thicknesses

GENERAL NOTES:
(A) All welds shall be given a visual examination in addition to the type of specific nondestructive examination specified
(B) NPS — nominal pipe size
(C) RT — radiographic examination; UT — ultrasonic examination; MT — magnetic particle examination; PT — liquid penetrant examination; VT — visual examination
(D) For nondestructive examinations of the pressure retaining component, refer to the standards listed in Table 126.1 or manufacturing specifications.
(E) Acceptance standards for nondestructive examinations performed are as follows: MT — see para. 136.4.3; PT — see para. 136.4.4; VT — see para. 136.4.2; RT — see para. 136.4.5; UT — see para. 136.4.6.

NOTES:
(1) The thickness of butt welds is defined as the thicker of the two abutting ends after end preparation.
(2) RT may be used as an alternative to PT or MT when it is performed in accordance with para. 136.4.5.
(3) RT or UT of branch welds shall be performed before any nonintegral reinforcing material is applied.
(4) In lieu of volumetric examination (RT, UT) of welded branch connections when required above, surface examination (PT, MT) is acceptable and, when used, shall be performed at the lesser of one-half of the weld thickness or each $^1/_2$ in. (12.5 mm) of weld thickness and all accessible final weld surfaces.
(5) Fillet welds not exceeding $^1/_4$ in. (6 mm) throat thickness which are used for the permanent attachment of nonpressure retaining parts are exempt from the PT or MT requirements of the above Table.

Figure 10.2 Code Table 136.4.

TABLE 10.6 Paragraphs in B31.1 for Acceptance and Indications

Examination type	Paragraph number
Visual	136.4.2
Magnetic particle	136.4.3
Liquid penetrant	136.4.4
Radiography	136.4.5
Ultrasonic	136.4.6

Code B31.9 also uses the paragraph mode to set the acceptance criteria. This table (Table 10.8) has a somewhat longer list as the criteria change from type of joint to type of joint.

Code B31.3 defines the extent of its required examination in Paragraph 341.4.1. It has a list, summarized below, explaining that required extent.

- Visually sufficient materials selected at random to ensure they meet specifications and are defect-free
- At least 5 percent of welds, which must represent each welder's work
- 100 percent of longitudinal welds unless made in accordance with a listed specification; if the weld joint factor is to be 0.90, then spot radiography that includes at least 1 percent of the length of those welds
- Random examination of the mechanical, including threaded joints to ensure compliance with Paragraph 335
- When pneumatic testing is expected, 100 percent examination to ensure compliance with Paragraph 335
- Random examination, including alignment and supports of erection and finished piping to find deviations from design intent
- Not less than 5 percent of girth welds by random radiography, with requirements to maximize coverage of each intersection with a longitudinal weld including the areas to be examined
- Not less that 5 percent of brazed joints
- Certification to the inspector that the above examinations have been carried out

TABLE 10.7 B31.5 Acceptance Criteria

What acceptance joint or method	Paragraph number
Welded joints	527
Brazed and soldered	528
Mechanical joints	535
Radiographic	536.6.3

TABLE 10.8 B31.9 Paragraphs for Acceptance

Joint acceptance	Paragraph number
Girth and groove welds	936.6.1
Fillet welds	936.6.2
Brazed and soldered	936.6.3
Threaded joints	936.6.4
Caulked and leaded joints	936.6.5
Flanged joints	936.6.6
Flared and flareless compression joints	936.6.7
Proprietary joints	936.6.8
Solvent, adhesive, and heat fusion joints	936.6.9

When either the spot or random examination finds a defect, the code spells out a specific procedure to provide further examination. The aim of this examination is to determine whether the only remedy is to repair the initial defect. That is spelled out in Paragraph 341.3.4. A flowchart showing the resampling process is seen in Fig. 10.3.

Paragraph 341.3.2 refers to a comprehensive figure that details the acceptance criteria. Paragraph 341.4 sets out the extent of examination required. Figure 10.4 is reproduced here for convenience.

As expected, there are different criteria for the separate sections in the book for the different types of product or levels, such as the category M fluids or high pressure in Chapter IX. The base codebook sets out some additional requirements for catergory D fluids, as well as for what is classified as severe cyclic service.

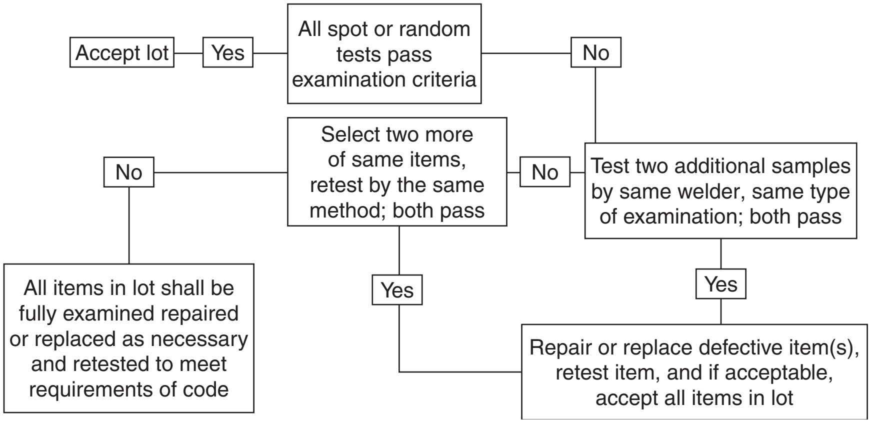


Figure 10.3 Spot or random flow chart per B31.3 Paragraph 341.3.4.

TABLE 341.3.2
ACCEPTANCE CRITERIA FOR WELDS AND EXAMINATION METHODS FOR EVALUATING WELD IMPERFECTIONS

Criteria (A to M) for Types of Welds and for Service Conditions [Note (1)]														Examination Methods				
Normal and Category M Fluid Service				Severe Cyclic Conditions				Category D Fluid Service										
Type of Weld				Type of Weld				Type of Weld										
Girth, Miter Groove & Branch Connection [Note (4)]	Longitudinal Groove [Note (2)]	Fillet [Note (3)]		Girth, Miter Groove & Branch Connection [Note (4)]	Longitudinal Groove [Note (2)]	Fillet [Note (3)]		Girth and Miter Groove	Longitudinal Groove [Note (2)]	Fillet [Note (3)]	Branch Connection [Note (4)]		Weld Imperfection	Visual	Radiography		Magnetic Particle	Liquid Penetrant
A	A	A	...	A	A	A	...	A	A	A	A		Crack	✓	✓	...	✓	✓
A	A	A	...	A	A	A	...	C	A	N/A	A	...	Lack of fusion	✓	✓	...	...	...
B	A	N/A	...	A	A	N/A	...	C	A	N/A	B	...	Incomplete penetration	✓	✓	...	...	...
E	E	N/A	...	D	D	N/A	...	N/A	N/A	N/A	N/A	...	Internal porosity	...	✓	...	...	...
G	G	N/A	...	F	F	N/A	...	N/A	N/A	N/A	N/A	...	Internal slag inlcusion, tungsten inclusion, or elongated indication	...	✓	...	...	...
H	A	H	...	A	A	A	...	I	A	H	H	...	Undercutting	...	✓	...	...	...
A	A	A	...	A	A	A	...	A	A	A	A	...	Surface porosity or exposed slug inclusion [Note (6)]	✓	...	...	...	...
N/A	N/A	N/A	...	J	J	J	...	N/A	N/A	N/A	N/A	...	Surface finish	✓	...	...	...	...
K	K	N/A	...	K	K	N/A	...	K	K	N/A	K	...	Concave root surface (suck up)	✓	✓	...	...	...
L	L	L	...	L	L	L	...	M	M	M	M	...	Weld reinforcement or internal protrusion	✓	...	...	...	...

Figure 10.4 Code Table 341.3.2.

Criterion Value Notes for Table 341.3.2

Symbol	Criterion: Measure	Acceptable Value Limits [Note (6)]	
A	Extent of imperfection	Zero (no evident imperfection)	
B	Depth of incomplete penetration Cumulative length of incomplete penetration	≤ 1 mm (1/32 in.) and ≤ $0.2\overline{T}_w$ ≤ 38 mm (1.5 in.) in any 150 mm (6 in.) weld length	
C	Depth of lack of fusion and incomplete penetration Cumulative length of lack of fusion and incomplete penetration [Note (7)]	≤ $0.2\overline{T}_w$ ≤ 38 mm (1.5 in.) in any 150 mm (6 in.) weld length	
D	Size and distribution of internal porosity	See BPV Code, Section VIII, Division 1, Appendix 4	
E	Size and distribution of internal porosity	For $\overline{T}_w$ ≤ 6 mm (1/4 in.), limit is same as D For $\overline{T}_w$ > 6 mm (1/4 in.), limit is 1.5 × D	
F	Slag inclusion, tungsten inclusion, or elongated indication Individual length Individual width Cumulative length	 ≤ $\overline{T}_w/3$ ≤ 2.5 mm (3/32 in.) and ≤ $\overline{T}_w/3$ ≤ $\overline{T}_w$ in any $12\overline{T}_w$ weld length	
G	Slag inclusion, tungsten inclusion, or elongated indication Individual length Individual width Cumulative length	 ≤ $2\overline{T}_w$ ≤ 3 mm (1/8 in.) and ≤ $\overline{T}_w/2$ ≤ $4\overline{T}_w$ in any 150 mm (6 in.) weld length	
H	Depth of undercut	≤ 1 mm (1/32 in.) and ≤ $\overline{T}_w/4$	
I	Depth of undercut	≤ 1.5 mm (1/16 in.) and ≤ [$\overline{T}_w/4$ or 1 mm (1/32 in.)]	
J	Surface roughness	≤ 500 min. Ra per ASME B46.1	
K	Depth of root surface concavity	Total joint thickness, incl. weld reinf., ≥ $\overline{T}_w$	
L	Height of reinforcement or internal protrusion [Note (8)] in any plane through the weld shall be within limits of the applicable height value in the tabulation at right, except as provided in Note (9). Weld metal shall merge smoothly into the component surfaces.	For $\overline{T}_w$, mm (in.) ≤ 6 (1/4) > 6 (1/4), ≤ 13 (1/2) > 13 (1/2), ≤ 25 (1) > 25 (1)	Height, mm (in.) ≤ 1.5 (1/16) ≤ 3 (1/8) ≤ 4 (5/32) ≤ 5 (3/16)
M	Height of reinforcement or internal protrusion [Note (8)] as described in L. Note (9) does not apply.	Limit is twice the value applicable for L above	

X = required examination NA = not applicable . . . = not required

Figure 10.4 *(Continued)*

TABLE 341.3.2 (CONT'D)

GENERAL NOTES:

(a) Weld imperfections are evaluated by one or more of the types of examination methods given, as specified in paras. 341.4.1, 341.4.2, 341.4.3 and M341.4, or by the engineering design.
(b) N/A the Code does not establish acceptance criteria or does not require evaluation of this kind of imperfection for this type of yield.
(c) * Alternative Leak Test requires examination of these welds, see para. 345.9.
(d) ✓ examination method generally used for evaluating this kind of weld imperfection
(e) . . . examination method not generally used for evaluating this kind of weld imperfection

NOTES:

(1) Criteria given are for required examination. More stringent criteria may be specified in the engineering design. See also paras. 341.5 and 341.5.3.
(2) Longitudinal groove weld includes straight and spiral seam. Criteria are not intended to apply to welds made in accordance with a standard listed in Table A-1 or Table 326.1.
(3) Fillet weld includes socket and seal welds, and attachment welds for slip-on flanges, branch reinforcement, and supports.
(4) Branch connection weld includes pressure containing welds in branches and fabricated laps.
(5) These imperfections are evaluated only for welds ≤ 5 mm ($^3/_{16}$ in.) in nominal thickness.
(6) Where two limiting values are separated by "and," the lesser of the values determines acceptance. Where two sets of values are separated by "or," the larger value is acceptable. $\bar{T}_w$ is the nominal wall thickness of the thinner of two components joined by a butt weld.
(7) Tightly butted unfused root faces are unacceptable.
(8) For groove welds, height is the lesser of the measurements made from the surfaces of the adjacent components; both reinforcement and internal protrusion are permitted in a weld. For fillet welds, height is measured from the theoretical throat, Fig. 328.5.2A; internal protrusion does not apply.
(9) For welds in aluminum alloy only, internal protrusion shall not exceed the following values:
    (a) for thickness ≤ 2 mm ($^5/_{64}$ in.): 1.5 mm ($^1/_{16}$ in.)
    (b) for thickness > 2 mm and ≤ 6 mm ($^1/_4$ in.): 2.5 mm ($^3/_{32}$ in.)
    For external reinforcement and for greater thicknesses, see the tabulation for Symbol L.

Figure 10.4 *(Continued)*

TABLE A341.3.2 ACCEPTANCE CRITERIA FOR BONDS

Kind of Imperfection	Thermoplastic			RTR and RPM [Note (1)]
	Hot Gas Welded	Solvent Cemented	Heat Fusion	Adhesive Cemented
Cracks	None permitted	Not applicable	Not applicable	Not applicable
Unfilled areas in joint	None permitted	None permitted	None permitted	None permitted
Unbonded areas in joint	Not applicable	None permitted	None permitted	None permitted
Inclusions of charred material	None permitted	Not applicable	Not applicable	Not applicable
Unfused filler material inclusions	None permitted	Not applicable	Not applicable	Not applicable
Protrusion of material into pipe bore, % of pipe wall thickness	Not applicable	Cement, 50%	Fused material, 25%	Adhesive, 25%

NOTE:
(1) RTR = reinforced thermosetting resin; RPM = reinforced plastic mortar

Figure 10.5 Code Table A341.3.2.

The different criteria for nonmetallic pipe are spelled out in Code Table A341.3.2 comparable to Code Table 341.3.2, which is duplicated here as Fig. 10.5. Note that there are at least four different methods of bonding nonmetallic pipe. These are mainly visual-type inspections.

The tables of acceptance criteria or requirements for the examination of other types of service are listed in Table 10.9.

Leak testing

Before a system is put into service, all the aboveground codes require a leak test. There might be another name used—B31.1 calls it a pressure test. This is a test for leaks at pressure. It is the final test before the system is put into service, and as the old saying goes, "The proof is in the pudding." There is simply too great a chance that something has been missed in the overall process to not perform this final test.

TABLE 10.9 References to Examination for Other Service in B31.3

Type of service	Paragraph or table
Category D fluid service	Paragraph 341.4.2
Severe cyclic service	Paragraph 341.4.3
Extent of examination for nonmetallic	Paragraph A341.4.1
Extent of examination categrory M	Paragraph M341.4
Acceptance criteria for Chapter IX	Table K341.3.2

This test is most generally a hydrostatic test. This test is usually done with water. However, it is interesting to note that B31.5 generally uses gas or refrigerant as the test medium. There will be more discussion of this a little later. However, due to the nature of the refrigerant fluid that generally is used in this type of service, Paragraph 538.5 has a specific caution that water or water solutions should not be used in refrigerant piping [538.5(d)].

This concern for the effects of the service fluid in piping that cannot be purged of all the effects of water or water solutions led the codes to allow other means of testing. The most common allowed alternative is some gas. Often this gas is air, but it can be a nontoxic gaseous substance. Other restrictions, such as having a lower flammable limit above 13 percent, are imposed.

The main concern regarding the use of these compressible gases is the fact that they are compressible. This means that much more energy is stored as the pressure rises. If such an amount of stored energy should fail during the test, an explosion of high magnitude might occur. This is also the reason for many of the other restrictions. While it is true that media such as water are "incompressible" and therefore not dangerous, a short discussion with any person who was present at the failure of one of these tests would convince the reader that any pressurized test has a degree of potential danger to the area in which it is done.

For the purpose of understanding the level of concern regarding pneumatic-type testing, an example is given here. The energy stored in the piping system is a function of the pressure and the specific heat ratio k of the gas being compressed. It is also dependent on volume. As an example of the potential, an example of a hypothetical situation is worked out for the reader.

Let us assume that a piping system consisting of 10 NPS Schedule 40 pipe is being tested. The owner opts not to do a hydrostatic test and wants a pneumatic one. The design pressure is 800 psig, which by material selection is below the upper limit of pressure that the pipe could handle. In this system there is 3000 lin ft of this pipe. The B31.3 paragraph sets the pneumatic test pressure of such a system at 110 percent of the design pressure, or 880 psig. It is common to use atmospheric pressure as 15 psi.

The calculations are as follows:

- The inner diameter of 10 NPS Schedule 40 pipe is 10.20 in. (0.85 ft).
- Therefore, the volume of the test is

$$V = \frac{\pi}{4}(0.85)^2 \times 3000 = 1702 \text{ ft}^2$$

- For air, it is a safe approximation to use 1.4 for k.

- The stored energy formula, under the assumption that the pipe is not moving and is being tested at a relatively low elevation, would be

$$\text{Stored energy} = \left(\frac{k}{k-1}\right) PV \left[1 - \left(\frac{P_a}{P}\right)^{k-1/k}\right] (144)$$

Substituting the values, we get

$$\text{Stored energy} = \left(\frac{1.4}{1.4-1}\right)(895)\times 1702 \left[1 - \left(\frac{15}{880P}\right)^{14-1/0.4}\right](144)$$

$$= 5.29 \times 10^8 \text{ ft} \cdot \text{lb}$$

- The number in foot-pounds doesn't give one a feeling for how explosive that is. If it is converted to TNT by using the heat of combustion of TNT of 5.066×10^6 ft · lbs/lb, it becomes 105 lb of TNT.

It is hoped that this example will give the reader an understanding of the power involved in pneumatic testing and will encourage every effort to be made to avoid that type of test. One could talk about the B31.5 requirements, but one should remember that the volume of refrigerant piping would be considerably smaller than in the example. In normal-size tubing and at the same lineal feet, it would be several thousand times less volume and consequently less explosive.

There are rather universal precautions taken in preparing for a hydrostatic test, and they are summarized below. The actual paragraphs with the detailed requirements are listed by code in Table 10.10.

- All joints including welds and bonds must be exposed. The specific relaxation of insulated joints that may be made is provided and outlined in the specific code.
- Temporary supports are made to the piping, if required, because the design was for less fluid weight than the test fluid.
- Any expansion joints must be restrained or isolated so as not to harm them.

TABLE 10.10 Paragraphs for Hydrostatic Test Requirements Code

Code	Paragraphs
B31.1	137 and all subparagraphs through 137.4.5
B31.3	345 and all subparagraphs through 345.4.3
B31.5	538 and all subparagraphs and 539 for records
B31.9	937 and all subparagraphs through 937.3.5

- A flange joint that is isolated to protect other equipment with a blank need not be tested.
- Test records are required.
- The test pressure is in general 1.5 times the design pressure. But in no case may it be higher than that of a nonisolated component or any other. As a note, this 1.5 has been lowered to 1.3 in BPVC due to the higher allowed stresses. If one is using one of the higher stresses, this should be checked because the test should stress nothing above the yield strength. Code B31.3 has a temperature-adjusted pressure formula, discussed later.
- In general, high points in the system should be vented, and at low points, a drain should be provided.
- Protection for the personnel conducting the test should be provided.
- The test gauges and pumps and all test equipment should be reviewed and, if necessary, calibrated.

As mentioned above, B31.3 has a temperature-adjusted formula that relates the test pressure to the allowable stress at the design temperature and the allowable stress at the test temperature. This is to ensure that the tests duplicate, as nearly as possible, the actual design conditions. A caution is added that at no time should the test pressure take the pipe to a stress level that would be above the yield strength at temperature. It allows the test pressure to be lowered in that case to a pressure that will not exceed yield at temperature. That is detailed in Paragraph 345.2. See Table 10.10.

As discussed earlier, these are leak tests. If the pipe system is large, it may take a long time to examine each joint for leaks. The codes recognize this. They also recognize that the value of 1.5 times the test pressure may be somewhat dangerous over a long time due to expansion from warming etc. Even though the testers are cautioned to take precautions, such trouble may result.

Thus the codes state that the test need only be held at that pressure for a specified length of time. This time is usually 10 min, but each code gives its specific time requirement in the paragraphs above. At the end of that time, the tester is allowed to relieve the pressure to a specified lower pressure and hold it there for sufficient time to examine all the joints.

Pneumatic tests

As indicated in earlier discussion, pneumatic or compressible gas tests are inherently more dangerous than the hydrostatic test. As a result, the codes set somewhat different requirements for them. As before, the specific requirements are outlined in the paragraphs listed in the pneumatic testing table (Table 10.11).

TABLE 10.11 Pneumatic Test Paragraphs by Code

Code	Pneumatic test paragraphs
B31.1	137.5 though 137.5.5 137.6 for halide tests
B31.3	345.5 for pneumatic tests 345.6 through 345.95 for other alternatives
B31.5	538 and all subparagraphs and 539 for records. Code does not allow water (hydrostatic test of refrigerant).
B31.9	937.4 for pneumatic test 937.5.1 and 937.5.2 for initial service test

Each of the codes has somewhat different procedures for these pneumatic-type tests, and one should check the specifics. The generic procedure is as follows:

- Some pressure-relieving device is required to forestall any type of runaway pressure event.
- The pressure is brought to generally one-half of the test pressure, but not to exceed some specified level, and held for a preliminary leak test. This is done to find gross leakages.
- The pressure is then raised in steps, generally by a percentage of the test pressure.
- After reaching each step, a hold point is required to allow the strains to equalize and the piping system to settle into an equilibrium.
- Once the test pressure is reached, it is held for the specified time.
- It is then reduced to some specified pressure, and the complete examination of all joints is performed.
- Some codes explicitly allow soap bubble–type tests to discover the leaks.

The codes may also give alternatives to either the pneumatic or the hydrostatic leak and pressure tests. They are listed in the paragraphs specified in Table 10.11. While the tests are required, assuming proper fabrication and examination along with an adequate design, they need not be considered unreasonably dangerous. One should follow the procedures and take all the precautions given in the requirements and guidance.

Chapter

11

Special Considerations for Pipelines

Throughout this book the difference between buried piping and aboveground piping has been one of the major ways to differentiate between the codes that anticipate pipeline service and the codes that do not. There is another major difference: Pipelines are also under the jurisdiction of the Code of Federal Regulations, Title 49, Parts 186 to 190.

To readers considering involvement in pipeline work, I recommend that they peruse this set of federal rules. This book does not discuss them in detail since the two major codes—B31.4 and B31.8—are basically written to cover the requirements set out in those rules. But, as in all cases, there can be differences between specific requirements. These differences may not create any problems; however, they would have to be reconciled.

A subtle, but important difference between the two arises because of these differences. A pipeline is rather like a highway. The major titles include the word *transportation* in their names, implying the same analogy. As transportation systems they cover a wide range of territory as opposed to a specific location. The aboveground codes basically are piping systems that have one address and location, however large that location may be. These imply several different requirements.

It is not insignificant that they in general are buried, which creates another set of problems to be considered. Just like travel along a highway, the fluid being transported would, after some distance, require additional driving force. They need compressor stations and pumping stations. And in modern times, at least, they often have a source that is offshore, requiring underwater piping that would have different although similar considerations.

It needs to be pointed out that not all pipelines, or not all sections of a pipeline, are buried. Some sections and facilities would be aboveground. By the same token, in the aboveground codes there may be some sections that may be buried for any number of reasons. The majority of pipelines are buried, and most of aboveground piping is aboveground.

This chapter will address those rather unique conditions of the pipelines in detail. Those conditions fall into major catergories:

- Facilities, including compressor/pumping stations and any required storage or control systems to isolate sections
- Operation and maintenance programs, including reporting and methods of determining the life of the various sections of pipeline
- Corrosion control including the transportation of more highly corrosive fluids
- Offshore pipeline differences
- Managing system integrity of B31.8S, Gas Pipeline Supplement

It might be pointed out that B31.1 has some nonmandatory, appendices on some of the subjects above. Since they are nonmandatory, this book will not discuss them in greater detail. They are mentioned for reference:

- Appendix IV, Corrosion Control for ASME B31.1 Power Piping Systems
- Appendix V, Recommended Practice for Operation, Maintenance, and Modification of Power Piping Systems
- Appendix VII, Procedures for the Design of Restrained Underground Piping (some of the features in this appendix were included in the discussion of supports in that chapter)

Facilities

The main requirement for the location of mainline or transmission line valves is the location and accessibility of those valves, should there be an accident or repair and replacement event, if that particular section of the line has to be isolated and, in the case of gas, "blown down." Blowdown is a way to empty the isolated section of the line of the hazardous material.

The reader might expect that the requirements become more stringent as the potential for damage increases. That is the case. Both B31.4 and B31.11 in their respective paragraphs, 434.15 and 1134.15, consider that the location shall be chosen so as to facilitate maintenance and limit the hazard and damage from accidental discharge. Code B31.4, Paragraph

434.15.2(e), specifically states that when LPG or liquid anhydrous ammonia is transported, block valves shall be spaced at 12 km (7.5 mi) maximum along the line. There are also specific locations which require a block or check valve.

Such lack of specificity is not seen in B31.8. Paragraph 846.1, entitled Required Spacing of Valves, does just that by type of line. For transmission line, a set of factors for considerations shall be taken into account. Those considerations are

- Continuous accessibility to the valves, which is considered primary
- Conservation of gas
- Time to blow down the isolated section
- Continuity of service
- Necessary operating flexibility
- Expected future development, which would change the location factor
- Significant natural conditions that could have adverse effects

Subsequently they list a maximum spacing according to the location class. These maximums are listed in Table 11.1. As an interesting aside to this maximum spacing list in B31.8, the corresponding paragraph in CFR 49, Part 192, is Paragraph 192.179. The description of the requirements is quite different but essentially winds up with the same maximum spacing. The wording in this paragraph follows this pattern. Each point on the pipe in a class X location line must be within class X, meaning that the pipe may go *X* distance in either direction from the valve. That works out to the Code B31.8 distance between valves. This can be considered an example of how carefully one should work with two separately written documents, even two that are covering the same subject, such as valve spacing in B31.8 and in CFR 49, Part 192.

These transmission valves may be installed aboveground, in a vault, and be accessible. They shall be protected from damage and tampering.

TABLE 11.1 B31.8 Maximum Transmission Line Spacing

Class	Required maximum spacing
4	5 mi (4 km)
3	10 mi (6.4 km)
2	15 mi (12 km)
1	20 mi (16 km)

NOTE: The spacing may be adjusted slightly to permit continuous accessibility.

As one travels across the country, one occasionally notices a set of pipelines above the ground and surrounded by a fence with a lock, which would meet such requirements.

Should the valves or regulators be in vaults, there are specific requirements. These include venting, ventilations, and sealing, as well as drainage from potential underground water. These requirements vary from no specific requirements for a vault of less than 75 ft^3 to very specific requirements when the vault is over 200 ft^3. They are all enumerated in Paragraph 847.3.4.

Compressor Stations and Gas Holders

While the code does not give the same level of requirements for the structure of a compressor station, it does enumerate some requirements. These are invoked in Paragraph 843 and all its subparagraphs. Those requirements are summarized here for convenience.

- Buildings should be located to minimize any communication of fire from adjacent property that is not controlled by the operating company.
- If piping is above 2 NPS, the materials shall be noncombustible and meet NFPA 220.
- They shall have at least two exits, and no exit shall be more than 75 ft from any point on the operating floor, measured along the centerline of aisles or walkways.
- Swinging doors shall open outward, and no exit shall require a key from the inside.
- Any fenced areas shall have a minimum of two gates, and any gate shall be unlocked when the area is occupied.
- Electrical equipment shall meet NFPA 70.
- When condensable vapors are present, the compressor shall be protected from liquefaction and have facilities to remove accumulation of liquid.
- Such removal equipment shall meet BPVC Section VII or location class 4 of B31.8.
- Fire protection is in accordance with the American Insurance Association.
- Emergency shutdown requirements are specific in subparagraph 843.431.
- Compressors shall have overspeed stops.
- Pressure relief is discussed in this book in a separate section.

- Gas detection and alarm systems are required unless specific items are met. These systems, when installed, have requirements specified in Paragraph 843.48.
- Piping within the station requirements is specified in Paragraphs 843.5 and all subparagraphs.

There are similar requirements for storage fields of gas. There are two types of storage container. The first type is called a *pipe-type holder*, and the second is called a *bottle-type holder*. As might be expected, the pipe-type holders are made from pipe and pipe components while the bottle-type holders are generally made with integral drawn, forged, or spun end closures and tested in the manufacturer's plant.

All these storage fields are required to be fenced, to prevent access by unauthorized persons. They have spacing requirements between the fence and the storage containers as well as between the storage containers. When the storage is at an operating pressure of 1000 psi or more, the clearance between the fence and storage containers is a minimum of 100 ft. When it is less than 1000 psi, that clearance is 25 ft.

Design factors by location are based on whether the minimum clearance is the 100 or the 25 and are shown in Table 11.2.

In addition, a formula is given for determining the minimum clearance distance between storage containers:

$$C = \frac{3DPF}{1000}$$

where C = clearance distance between containers, in.
D = outside diameter
F = design factor per Table 11.2
P = maximum allowable operating pressure, psig

In addition, there are requirements regarding the depth to which the containers shall be buried. Bottle-type containers shall be buried below the frostline, but in no case closer than 24 in. to the surface. Pipe-type containers shall be buried not less than 24 in. below the surface.

TABLE 11.2 Design Factors for Calculating Distance Between Gas Storage Containers

Class location	Minimum clearance of 25 ft	Minimum of 100 ft
1	0.72	0.72
2	0.60	0.72
3	0.60	0.60
4	0.40	0.40

Subparagraph 844.41 gives special conditions for the manufacture of bottle-type containers. These define materials, welding restrictions, hydrostatic test requirements, and after-installation testing.

If gas contains more than 0.1 gr of hydrogen sulfide per 100 standard ft^3, specific requirements for mitigation are outlined in subparagraph 844.5. These include cautions about corrosion and safe operation of the storage field.

Pumping Stations and Tank Farms

Code B31.4 is written basically for liquids and B31.8 for gases, so while the requirements are similar, the specifics are different. Liquids are pumped and stored in tanks at little pressure over the hydraulic pressure of storage.

As was the case with B31.8, general precautions, summarized here, are spelled out in subparagraphs 434.20 through 434.24.4 and 435 and all subparagraphs. For B31.11, those paragraphs are 1134.2 through 1134.23 and 1135 and all subparagraphs.

- Construction of facilities shall be planned and specified.
- All such work shall be done by knowledgeable workers.
- Location shall be on fee or leased property located so as not to be subject to fire communication from adjacent structures or property.
- All piping of such facilities shall be in accordance with B31.4 or B31.11, depending on the service for which the construction is being accomplished.
- Fire protection and dikes or firewalls shall be in accordance with NFPA 30 for B31.4 and with good engineering practice for B31.11.
- Electrical installation shall be in accordance with NFPR 70 and API RP 500C.
- In B31.4, if vapor pressure is approximately atmospheric, the tanks shall be in accordance with API 650, API 12 B, and API 12F.
- Code B31.11 uses API 650, API 12 D, API 12 F, or AWWA D100.
- In B31.4, if vapor pressure is 0.5 psig but not exceeding 15 psig, only API 650 may be utilized.
- In B31.4 if vapor pressure is over 15 psi, BPVC, Section VIII, Division 1 or 2, shall be used.
- Any buried holders used for storage shall be built to B31.4.
- Strainers and filters shall be built to the same criteria as the piping system to which they are installed.

Operation and Maintenance

One might ask: Why are there operation and maintenance requirements set out in the base code for the pipelines but not for the other codes? It certainly is not true that aboveground pipelines do not require maintenance. In fact, the ASME has committees working on what they call *post construction* codes or rules. The difference is that the pipelines, because of their quasi-public operation, as evidenced by the existence of such requirements of CFR 49, have certain ongoing requirements that are not necessarily in accord with the systems anticipated by the other codes.

It is true that B31.1, as mentioned, does have nonmandatory appendices, which are not code requirements, but options the owners could adapt or follow. Code B31.3 attempts to keep the line of demarcation clean between the new construction which they aim to cover and post-construction or operation and maintenance which they do not cover. That stance leads one to the question: What happens when one puts in a new piece or section? The other codes—B31.5 and B31.9—remain somewhat silent on the issue.

Even when the pipeline codes do have chapters on operation and maintenance, the codes recognize the problem of prescription of details for each case of the operation and maintenance. Each code specifically points out its chapters on the procedures. Each points out that the operating company is to develop procedures based on the provisions of that code, including the company's experience and knowledge. This is shown in the subtitles for the main chapter of each code. Those chapters are 450, 850, and 1150. The 4, 8, and 11 portion are the respective codes B31.4, B31.8, and B31.11. That subtitle addresses those procedures that affect the safety of the particular systems.

It is true that there are differences in presentation and detail that are predicated on the base service that the codes address. It seems intuitive that gas transmission requires more detailed safety concerns than does liquid hydrocarbon transport, and both exceed the concerns of slurry which quite often are water and some solid substance. Nevertheless, there are certain common denominators.

This book will give a general overview of those common denominators and discuss in some detail the more specific requirements. The general pattern is not explicitly the same in the codes. It is up to the operating companies to develop the explicit procedures. Possibly Code B31.8 has the more explicit discussion of the basic requirements, and it will be used as the template for the discussion of the common concerns. One could also reference the appropriate chapters in CFR 49, Parts 186 through 199.

The basic requirements as listed in B31.8 850.2 seem to be a good starting platform. They are summarized here.

- Each company shall have a written plan covering operation and maintenance and the scope and intent of the code in question.
- Each company shall have a written emergency plan covering failures or other emergencies.
- Each company shall operate in conformance to the above plans.
- Each company shall modify the plans periodically as required by experience and changes in conditions.
- Each company shall provide comprehensive training for employees to prepare them for their functional responsibilities.
- Each company shall keep records for the proper administration of the plans.
- The operating plan should include items recommended in the code, address the various sections in terms of their hazard, and address periodic inspections.
- Emergency plans shall include, among other things, a system for receiving and responding to emergencies, a prompt response procedure for each type of emergency, a method for disseminating information to the public, a method for safe restoration of service, and the reporting and documenting of the emergency.
- Each company shall establish and maintain liaison with appropriate police, fire, and other public officials during the emergency.
- Educational programs are particularly important for gas. They should educate the public on how to recognize and report such an emergency.
- Again for gas lines, Paragraph 850.6 lists requirements for preventing accidental ignition of the gas.
- Each company shall outline precautions against damage to the pipeline if blasting occurs in the area of the pipeline, including defining safe areas for such blasting work.

As mentioned, the degree and detail of such plans and procedures will vary with the code. In fact, the emphasis does vary with the code. This book uses the requirements of B31.8 as the guideline since they are basically more stringent in accordance with the potential for damage.

Major portions of the chapters define repairs and how they should be disposed. Those repair requirements include many details and are best understood by reading the appropriate paragraphs in the code. Accordingly, they are listed in Table 11.3. Since the codes have different levels of detail, they are listed separately.

Each of the pipelines has provisions for periodic patrolling of the pipeline. This is to address two major concerns: to determine the condition of the pipeline and to observe whether there has been any change in the

TABLE 11.3 Paragraphs for Repair Detail

Code B31.4	Code B31.8	Code B31.11
451.6.1, 451.6.2, 451.6.3, 451.9, 452	851.4, 851.41, 851.42, 851.43, 851.45, 851.51, 851.52, 852.52, 852.7, 853 through 853.5, Appendix L	1151.6.2, 1151.6.3, 1151.9, 1152.9

areas adjacent to the pipelines, such as construction or other significant events that might affect the pipeline. At that point, the plans would establish any required action depending on the results.

Code B31.4 states that the intervals between patrolling shall not exceed 2 weeks for areas that are not designated as industrial, commercial, or residential. The interval shall not exceed 1 week in those areas. Code B31.11 puts the patrolling intervals for that service at no more than 1 month. Code B31.8 has sliding intervals according to the class location of the line. They are given in Table 11.4.

In performing inspections and checking corrosion pits in the pipelines, the problem is always this: What amount of corrosion requires action? The paragraphs mentioned in Table 11.3 spell out the action required for specific measurements of corrosion. If the depth of the corrosion found is below a certain minimum—which is different in each code—no action is required, unless the length of the corrosion pit is longer than a calculated amount. Each of the codes has a formula for determining the allowable longitudinal length of corrosion for those instances where the depth of the corrosion is less than the minimum depth.

Although a dent is not corrosion, if it is too large, it can cause problems also. The criteria for dents are rather like some of the criteria for bends. There is a limit to how much strain the dent causes as it bends the surface. Code B31.8 has an Appendix R which discusses how to estimate the amount of strain in a bend. This is useful information for almost any dent regardless of the code. It gives some methods for calculating when a dent has gone too far.

This approach is also outlined in Code B31.G. The B31.G code book has charts that preclude the use of the formula and calculation of L which may be valuable in the field. There is also discussion of the background for the approach which shows the justification of the safety level

TABLE 11.4 Patrol Frequency for B31.8 Pipelines

Location class	Interval
1 and 2	At least once every year
3	At least once every 6 months
4	At least once every 3 months

represented. It has some basic computer programs which were used in the generation of those charts. They would also be helpful for users when they are dealing with a set of variables not in the charts.

There is a difference in the B31.8 formula in one of the constants. There is a slight difference in the algebraic presentation which does not change the numerical answer. The change in constant is noted in the formula presented here. The general rule is to repair the corrosion pit as long as the length is less than the L calculated. If not, the pipe needs to be cut out and replaced, or in some cases a full encirclement sleeve is allowed. The formula as given in B31.8 is presented here. Codes B31.G, B31.4, and B31.11 present it as two formulas. The sketch from B31.G is presented in Fig. 11.1.

$$L = 1.12\left[Dt_n\left(\frac{d/t_n}{1.1d/t_n - x}\right)^2 - 1\right]^{1/2}$$

where L = measured extent allowed, which should be less than L calculated
D = outside diameter of pipe
t_n = nominal wall of pipe
d = measured maximum depth of corrosion
x = factor which is 0.11 in B31.8 and 0.15 in all other books

Note that the calculated amount $1.12\sqrt{Dxt_n}$ is limited to 4 in all books but B31.8.

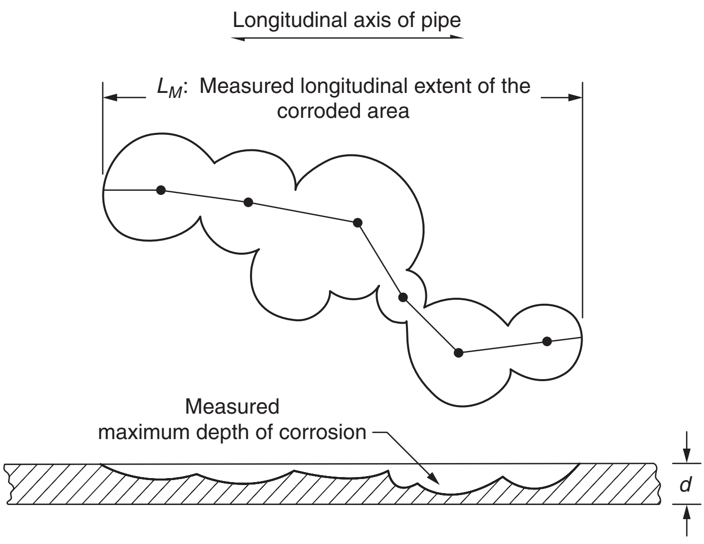


Figure 11.1 Typical sketch showing how to measure a corrosion pit.

As the operating company goes through these processes, one option is to reduce the maximum operating pressure (MAOP). In some cases for B31.8 this may come about as the area through which the pipeline travel changes to the extent that a change in location class will be required. This is discussed in Paragraph 854 and a guideline table, Table 854.1(c), which helps in determining the reduced MAOP for the line. That table is produced here for convenience as Fig. 11.2.

It is also possible that, in the process of determining what to do to any pipeline, some intermediate pressure may be desirable and allowed. For this reason the codes give a procedure and formula to calculate what that revised pressure might be. That process is discussed in Paragraph 451.7 in B31.4, Paragraph 1151.7 in B31.11, and Appendix L in B31.8. The formula is given here for convenience. An interesting note is that the length L involved in the discussion of repair of corrosion is a factor in this calculation. Note that while there are presentation differences (i.e., decimal 0.67 versus fraction 2/3) in constants, the formulas give essentially the same results to the desired accuracy required by the codes. That formula is

$$P_l = 1.1P_i\left[\frac{1-0.67d/t_n}{1-0.67d\Big/\left(t_n\sqrt{G^2+1}\right)}\right]$$

where $G = 0.893L\sqrt{Dt_n}$; if $G > 4$, then

$$P_l = 1.1P_t\left(1-\frac{d}{t_n}\right)$$

but in no case can P_l exceed P_i

Table 854.1(c) Location Class

Original [Note (1)]		Current		
Location Class	Number of Buildings	Location Class	Number of Buildings	Maximum Allowable Operating Pressure (MAOP)
1 Division 1	0–10	1	11–25	Previous MAOP but not greater than 80% SMYS
1 Division 2	0–10	1	11–25	Previous MAOP but not greater than 72% SMYS
1	0–10	2	26–45	0.800 × test pressure but not greater than 72% SMYS
1	0–10	2	46–65	0.667 × test pressure but not greater than 60% SMYS
1	0–10	3	66+	0.667 × test pressure but not greater than 60% SMYS
1	0–10	4	[Note (2)]	0.555 × test pressure but not greater than 50% SMYS
2	11–45	2	46–65	Previous MAOP but not greater than 60% SMYS
2	11–45	3	66+	0.667 × test pressure but not greater than 60% SMYS
2	11–45	4	[Note (2)]	0.555 × test pressure but not greater than 50% SMYS
3	46+	4	[Note (2)]	0.555 × test pressure but not greater than 50% SMYS

NOTES:

(1) At time of design and construction.

(2) Multistory buildings become prevalent.

Figure 11.2 Code Table 854.1(c).

L = length, as calculated above
d = depth of corrosion
t_n = nominal wall
P_l = new lower pressure
P_i = initial or previous MAOP
D = pipe diameter

It may be that the company wants to increase the pressure, abandon a line, recommission an abandoned line, or convert a line from liquid to gas. Those requirements are set out in Paragraphs that are 55 or above, that is, B31.4 455 and above, B31.8 855 and above, and B31.11 1155 and above. Since those involve administration and some of the previously discussed technical issues, they are not discussed here.

Corrosion Control

Corrosion may occur in all piping. As the reader will recall, in the design phase the designer is urged to add an allowance in all the codes that covers mechanical allowances and includes corrosion allowances. Both by the pipeline codes and CFR 49, Parts 186 to 199, pipelines have more stringent requirements.

They are essentially required to install corrosion control technology, i.e., some corrosion control coating and cathodic protection. The designer is given an option to provide proof that cathodic protection may not be required because of the conditions of the environment and materials used. However, the designer is then required to test to see if that is really the case and to take remedial action. The test procedure is also required on existing lines that may or may not have been designed and installed under the current codes.

Once again, as in operation and maintenance, the codes recognize that they cannot write to cover every possible environment that the pipeline and its materials may encounter. They codes do set out requirements in a more generic form. They also provide other sources of information which may help in making the decisions as to type and technology of corrosion control. Those sources are listed in Table 11.5 as well as which codes refer to them.

Some general comments apply. Because of their buried nature, the pipelines cause concern with both external and internal corrosion control. The external corrosion control is the one where cathodic protection is invoked in addition to any coating invocation. For internal corrosion control it is the coating. It might be pointed out that B31.8 includes Chapter IX, Sour Gas Pipelines.

This chapter, like the high-pressure chapter in B31.3, follows each major portion of the code and relates whether the base code applies

TABLE 11.5 Sources of Corrosion Control Data and Code Referencing them

Source	Codes referenced
Nace RP-02-75	B31.8
Nace RP-02-77	B31.8, B31.4
EPRI El-3106	B31.8
B31-G	B31.8
Nace, the corrosion data survey	B31.8
Nace RP-01-69	B31.4, B31.11
Nace RP-06-75	B31.4, B31.11
NFPA 70	B31.4
API RP 500C	B31.4
Nace RP -01-75	B31.4
Nace RP-01-77	B31.11
Nace MR0175	B31.8, Chapter IX

and, if not, what is excepted. And in some cases paragraphs are added to cover particular concerns about sour gas.

Sour gas has a recognizable portion of H_2S (hydrogen sulfide) mixed in with the gas. This is a poisonous combination so restrictions are greater. The major problem is that it is highly corrosive, so requirements of that chapter are included in this discussion of corrosion control.

As mentioned, there needs to be greater caution about all the issues because of the more highly corrosive nature. One of the results of this concern along with the poisonous aspect is a separate procedure to ensure public safety in case of a leak.

The first issue is to determine the concentration of the gaseous mixture. Suitable standards for that are listed in Paragraph B850.1. They are ASTM D 2385, GPA C-1, or GPA Publication 2265. Once that determination is made, the lines can be separated into two groups. The groups are lines with 100 ppm, or lines with 500 ppm. From that a radius of exposure can be calculated by using equations given in B31.8. Those equations are given in both U.S. Customary System (USCS) and SI (metric) units, one of the few times that occurs in this code. They are seen in Table 11.6.

Note in using the formula that M is the molal fraction of hydrogen sulfide in the gaseous mixture; Q is in standard cubic feet for the USCS and in standard cubic meters for the metric. And ROE is in feet or meters, respectively; Q is the amount that would escape in a 24-h period.

TABLE 11.6 Radius of Exposure Equations

Event	100 ppm USCS	100 ppm metric	500 ppm USCS	500 ppm metric
Equation	$ROE = (1.589MQ)^{0.6258}$	$ROE = (8.404MQ)^{0.6258}$	$ROE = (0.456MQ)^{0.6258}$	$ROE = (2.404MQ)^{0.6258}$

The issue revolves around "fixed facilities"—fencing, locking valves, and plugging ports—to provide security in those areas. The radius of exposure is used to determine what is within that radius if the radius is over 50 ft (15.2 m). The requirements are spelled out in Paragraph B855.1 and are summarized here.

- 100 ppm if *ROE* includes any part of public area except a public road
- 500 ppm if *ROE* includes any part of a public road
- 100 ppm if *ROE* is greater than 3000 ft (915 m)

When any of the above criteria are met, then additional control is required beyond the prescribed Poison Gas signs. Included is a written contingency plan that is given to emergency response authorities.

This extra requirement for corrosion control for sour gas is established. The rest of the discussion is pointed to the regular corrosion control requirements. They are essentially the same for all three codes.

Since the coatings are applied to the pipes at a coating facility, in the majority of cases extra precautions are taken not to damage the external or internal coating while delivering, handling, or lowering into the ditch at the site. As mentioned, the choice of the specific coating is left to the operating company. However, it must be inspected by an electric holiday detector just prior to lowering into the ditch. If any holidays are found, they shall be repaired and reinspected.

Cathodic protection may be provided by a galvanic anode or an impressed current anode system. The system shall be electrically isolated at all interconnections with foreign systems. Provisions must be made to prevent damage from lightning or ground fault currents, except where impractical provision shall be made with test lead to check the system. These leads shall be attached in a manner to make them usable for the expected life of the monitoring of these system. The testing system shall be tested on a frequency which will ensure with reasonable accuracy the degree of protection that is being delivered.

Code B31.8 contains Paragraphs 864 and 865 which discuss additional concerns for arctic and high-temperature situations. The arctic conditions place empasis on frozen ground, permafrost, and similar conditions which, with improper protection, might lead to local thaw and resulting problems. For many of these considerations, the Alaskan Pipeline has significant portions placed aboveground to forestall such problems. High temperature is defined as above 150°F. At these types of temperatures, corrosion may be faster than normal.

As might be expected, records of location and results of the tests are required. The location records are for the lifetime of the service. Test results must be at least until subsequent tests supercede the test on record.

Offshore Piping

Offshore piping has many different requirements. Codes B31.4 and B31.8 have chapters devoted to that type of piping (Chapter VIII in B31.8 and Chapter IX in B31.4). Both codes state that the chapters, insofar as possible, use parallel numbering systems for the paragraphs to the base code. The base code is, the chapters just before and up to the chapters for offshore piping. These chapters specifically point out that those base code requirements, unless modified, in the offshore chapter apply.

Both codes have sets of definitions that complement and add to the normal definitions of piping. One of the key definitions is that of *offshore*. It is the area beyond the line of ordinary high water along the portion of the coast that is in direct contact with the open seas and beyond the line marking the seward limit of inland coastal waters. These lines include the risers to the platforms. Tankers or barge loading hoses are not considered part of the offshore pipeline system.

One of the primary concerns of offshore piping is pipe collapse that may occur by excessive external pressure. This relates to any buckle that may occur as a result of this pressure and includes considerations for mitigating that possibility. This is often done by using what are called *buckle arresters*. One technique that could be used is the external pressure section and the design of stiffening rings, as outlined in Section VIII.

The major differences in the design of offshore pipelines can be reduced to the fact that there are loads that the chapters express in detail. Each of the codes lists specific loads. It is incumbent on the reader to refer to the chapter of the particular code for details. The loads are summarized here for convenience:

Waves	Platform motion
Current	Temperature
Marine soils	Pressure
Wind	Water depth
Ice	Support settlements
Seismic activity	Accidental loads

In addition, Code B31.8 gives a nice checklist of possible modes of failure which should be considered. These make a good reminder list to use in any design problem, even if the design is not being done for a Code B31.8 problem. That list is as follows:

Excessive yielding	Propagating fracture
Buckling	Corrorison
Fatigue failure	Collapse

TABLE A402.3.5(a)
DESIGN FACTORS FOR OFFSHORE PIPELINE SYSTEMS

Location	Hoop Stress F_1	Longitudinal Stress F_2	Combined Stress F_3
Pipeline	0.72	0.80	0.90
Riser and Platform Piping [Note (1)]	0.60	0.80	0.90

GENERAL NOTE: In the setting of design factors, due consideration has been given to, and allowance has been made for, the underthickness tolerance and maximum allowable depth of imperfections provided for in the specifications approved by the Code.

NOTE:
(1) Platform piping does not include production facility piping on a platform; see definitions para. A400.2.

Figure 11.3 Code Table A402.3.5(a) from B31.4.

Ductile fracture	Impact from such things as anchors, boats, trawl boards
Brittle fracture	
Loss of in-place stability	

One of the important considerations and requirements is whether the design for the pipe itself is different from that in the base code. First there are different design factors, which are shown in Fig. 11.3, taken from Code B31.4, but the same numerical values appear in Code B31.8 (Fig. 11.4), except that F_1 for B31.8 and riser and platform piping is 0.50, rather than the 0.60 shown in the B31.4 chart. This affects the t calculated in the hoop stress equation.

Table A842.22
Design Factors for Offshore Pipelines, Platform Piping, and Pipeline Risers

Location	F_1 Hoop Stress	F_2 Longitudinal Stress	F_3 Combined Stress
Pipeline	0.72	0.80	0.90
Platform piping and risers	0.50	0.80	0.90 [Note (1)]

NOTE:
(1) The wall thickness used in the calculation of combined stress for platform piping and risers shall be based upon specified minimum wall thickness, including manufacturing, corrosion, and erosion allowances.

Figure 11.4 Design factors from B31.8 (note differences from Fig. 11.3 in hoop stress and notes).

However, there is a significant difference between the two. Note 1 in Code B31.8 reads as follows: "The wall thickness used in the calculation of combined stress for platform piping and risers shall be based upon specified minimum wall thickness, including manufacturing, corrosion and erosion allowance." This reflects the conservatism inherent in B31.8.

There are three equations. Two are the hoop stress and longitudinal stress, which is common. The third is the combined stress. The designer has the option of using Tresca or Von Mises formulas. Both are given in the books.

Hoop stress. For pipelines and risers, the tensile hoop stress due to the difference between internal and external pressures shall not exceed the values given below:

NOTE: Sign convention is such that tension is positive and compression is negative.

$$S_h \leq F_1 S T$$

$$S_h = (P_i - P_e)\frac{D}{2t}$$

where D = nominal outside diameter of pipe, in.
F_1 = hoop stress design factor from Table A842.22
P_e = external pressure, psi
P_i = internal design pressure, psi
S = specified minimum yield strength, psi
S_h = hoop stress, psi
T = temperature derating factor from Table 841.116A—Note in B31.4 T is always 1
t = nominal wall thickness, in.

Longitudinal stress. For pipelines and risers the longitudinal stress shall not exceed values found from

$$|S_L| \leq F_2 S$$

where A = cross sectional area of pipe material, in.^{2}
F_a = axial force, lbs
F_2 = longitudinal stress design factor from Table A842.22
M_i = in-plane bending moment, in.-lb
M_o = out-plane bending moment, in.-lb
S = specified minimum yield strength, psi
S_L = maximum longitudinal stress, psi (positive tensile or negative compressive)
= $S_a + S_b$ or $S_a - S_b$, whichever results in the larger stress value

S_a = axial stress, psi (positive tensile or negative compressive)
= F_a/A
S_b = resultant bending stress, psi = $[(i_i M_i)^2 + (i_o M_o)^2]^{1/2}/z$
i_i = in-plane stress intensification factor from Appendix E
i_o = out-plane stress intensification factor from Appendix E
z = section modulus of pipe, in.^{3}
| | = absolute value

Combined stress. For pipelines and risers the combined stress shall not exceed the value given by the maximum shear stress equation (Tresca combined stress):

$$2\left[\left(\frac{S_L - S_h}{2}\right)^2 + S_t^{\,2}\right]^{1/2} \le F_3 S$$

where A = cross-sectional area of pipe material, in.^{2}
F_a = axial force, lbs
F_3 = combined stress design factor from Table A842.22
M_i = in-plane bending moment, in.-lb
M_o = out-plane bending moment, in.-lb
M_t = torsional moment, in.-lb
S = specified minimum yield strength, psi
S_L = maximum longitudinal stress, psi (positive tensile or negative compressive)
= $S_a + S_b$ or $S_a - S_b$, whichever results in the larger stress value
S_a = axial stress, psi (positive tensile or negative compressive)
= F_a/A
S_b = resultant bending stress, psi
= $[(i_i M_i)^2 + (i_o M_o)^2]^{1/2}/z$
S_h = hoop stress, psi
S_t = torsional stress, psi = $M_t/2Z$
i_i = in-plane stress intensification factor from Appendix E
i_o = out-plane stress intensification factor from Appendix E
z = section modulus of pipe, in.^{3}

Alternatively, the Maximum Distortional Energy Theory (Von Mises combined stress) may be used for limiting combined stress values. Accordingly, the combined stress should not exceed values given by

$$(S_h^2 - S_L S_h + S_L^{\,2} + 3S_t^2)^{1/2} \le F_3 S$$

In general, the pipeline would lay on the bottom. Under certain circumstances the subsea floor soil conditions should be checked for hazards that may affect the construction or operation. The weight of the pipeline may have to be set so that it will not float. This may be accomplished by a coating of a heavy substance such as cement. Alternatively, an anchoring system may be employed to accomplish this. Consideration should be given to scouring due to wave or current action in making those determinations.

Where storms are considered, the return interval of the storm shall be placed at five times the design life or 100 years, whichever is less. Note that this is usually the reciprocal of the probability of a storm of the magnitude occurring in any one year. Thus for a 100-year storm cycle, it may be considered as a storm that has a historical probability of 0.01 of occurring in any given year.

The remaining requirements are similar enough to the base code that they are not specifically discussed.

Pipeline Integrity

This is the new book supplement to B31.8, designated B31.8 S. This book addresses the methodology of making an integrity assessment plan and the elements of such a plan. As such, it is not necessarily something to discuss in a simplified presentation. Experienced personnel and knowledge of the pipeline operation are inherently required. This is an emerging field in all pressure technology. It is very similar to risk-based inspection and quality control. The major emphasis or goal of any such system is to check the critical portions of the subject frequently, and the less critical less frequently.

The purpose of the plans is to develop a disciplined and rational approach to accomplishing that goal. Any reader interested in going on to the next step might find reading that document valuable.

The book has a flowchart and element diagram in Fig. 11.5 that gives a very good idea of what should happen in any such plan. It is reproduced here so readers may determine whether this is something they need to check.

The program also sets out a method to determine the potential impact area. This is somewhat similar to the *ROE* for sour gas but is not mathematically the same. The formula given has a constant of 0.69. A note farther down points out that the constant is valid only for natural gas. The derivation of the equation is given with the complete equations in the footnotes. This equation is somewhat similar to the equation given in which an equivalent to 1 lb of TNT was calculated for the potential damage due to pneumatic testing. It is not given here because it requires the knowledge of several factors unique to the gas

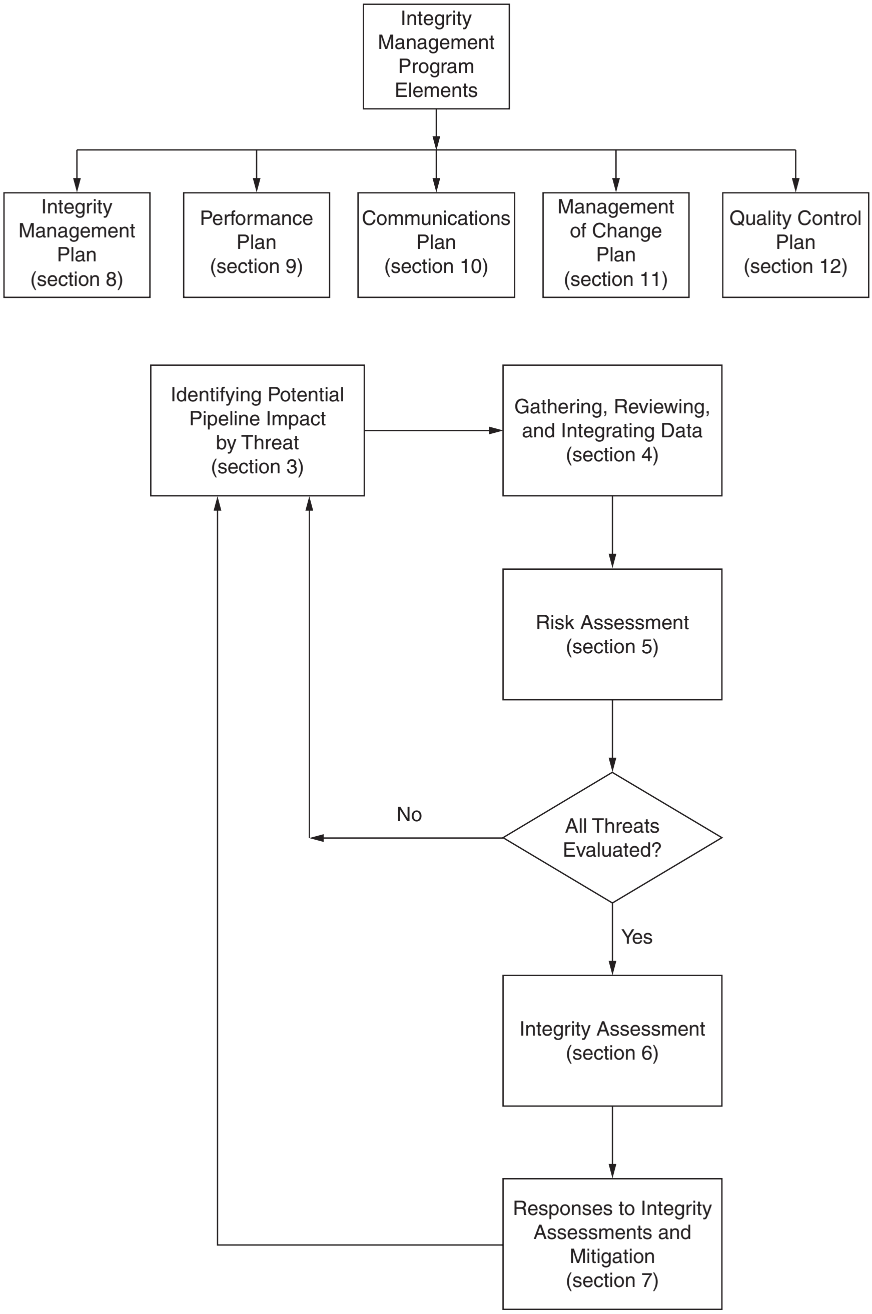


Figure 11.5 Integrity management flowchart.

in question. One thanks the committee for simplifying most of the variable to the 0.69 factor for natural gas. The formula given for natural gas is

$$r = 0.69d\sqrt{P}$$

where r = impact circle, ft
P = pipe segment's MAOP
d = diameter of pipe, in.

There are many other elements to B31.8 S. These tend to require, as the book points out, the realm of experienced and knowledgeable personnel. Therefore they are not considered fodder for this book.

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Chapter

12

Special Considerations for Other Pipe Systems

Pressure Relief

Actually the subject of pressure relief applies to all piping systems, even to all pressure containment systems. Readers only need to look closely at the water heater serving their needs to notice that even there is a need for a pressure relief valve. It is simply a requirement for any prudently designed and constructed pressure containment system to have a means of releasing a runaway pressure surge.

The piping codes are no exception to that need. There may be a considerable difference in the detail which these codes approach. But none fail to state something about them. The codes do not necessarily tell you how to design the pressure relief device. That falls very much into the proprietary area. Each manufacturer tends to solve the various problems in a different way.

The two most basic ways to resolve the overpressure situation are via a pressure relief valve and via a rupture disk. Often in a piping system, there are controls that tend to shut down the system when an operation begins to go awry. The pressure relief system is the last resort. No matter the sophistication of the systems, they can fail. A final basically mechanical system is what may save the disaster.

As mentioned, each of the codes handles the issue differently, and Table 12.1 cites the paragraphs addressing this issue.

Code B31.3 in 322.6.3 invokes very completely BPVC, Section VIII, Division 1, and paragraphs UG 125(c), UG 126 through UG 128, UG-132 through UG 136. They exclude UG 135(e) and UG 136(c). There are

TABLE 12.1 Paragraphs Regarding Pressure Relief

Code	Paragraphs relating to pressure relief
B31.1	107.8, 122 and all subparagraphs
B31.3	301.2.2, B322.6.3, A, M, and K versions
B31.4	434.20.6, 437.1.4, 452
B31.5	501.2.5, 501.5.5, 507
B31.8	803.331, 843.41–43, 845 and all subparagraphs
B31.9	901.2.3, 922.1.1
B31.11	1134.20.6, 1137.1.4, 1152.2–3

prescriptions as to what certain definitions in that pressure vessel code mean in this piping code. It is a fact that terminology changes in meaning, but not intent, between the various codes.

Code B31.8 refers to Section VIII, but does not go into paragraph detail. Code B31.8 does address it with somewhat different requirements. These are all outlined in paragraph 845 subparagraphs. Many of the constraints are to establish the line's MAOP. This varies with type of line and with the testing pressure and location factor of that line segment. It addresses the situation where a lower-pressure line supplied by a higher-pressure line needs a pressure-regulating device. And it differentiates between a main line and a high- or low-pressure distribution line.

All pipeline codes, in keeping with their operation and maintenance proclivity, also stipulate that all such protection systems be tested. These tests are to be performed at a frequency that will ensure the system will adequately provide the intended safety.

Codes B31.4 and B31.11 have similar but less stringent requirements for the safety relief systems. They do not get very specific as to what and where. They do refer to the plans and specifications as the source of information in this regard.

Code B31.9 requires that if a pressure-relieving system cannot be installed, the piping shall be designed to the highest pressure that it might receive. Code B31.5 in Paragraph 501.5.5 mentions that the system shall be designed for the reaction forces that occur with discharge of fluid.

This leads to two points. First, the alert reader may have noted and wondered that there is no previous discussion of B31.1. Second, Paragraph 501.5.5 may point to the more crucial design point to any pressure relief system. When a relief valve blows, there is a reaction that imposes moments and forces on the pipeline. It is similar to the recoil on one's shoulder from discharging a shotgun. The installation and the piping system must be designed for that force.

It is for this reason that B31.1 includes the nonmandatory appendix regarding installation of pressure relief valves. There are a significantly large number of ways that a safety valve installation may be designed. Figure 12.1 shows some of the more common ways.

To calculate those forces is a rather involved process. It can add moments to the piping and mounting system that did not exist before. This says it is a dynamic problem, not a static problem. The system described in Appendix II is specifically designed for steam lines, which are the lines that most often occur in power plants. It is beyond the scope of a simplified presentation. However, it can be stipulated by experience in many piping applications that it works and gives a safe system.

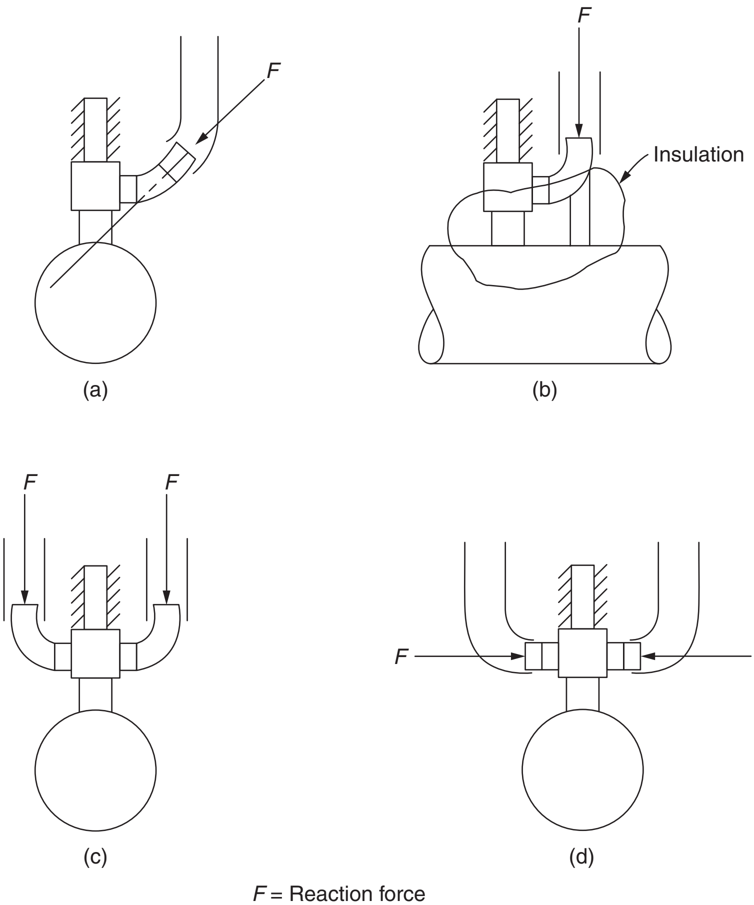

Figure 12.1 Some acceptable safety valve installations.

However, another problem often occurs in these situations that these codes do not address explicitly. It is the problem of flow-induced vibration. The best explanation of this phenomenon is to think of blowing across the top of an open soda bottle, even though as time goes on, another analogy may have to be substituted since bottles seem to be going the way of the tin can. At any rate, when one blows across the bottle top, a sound, or vibration, is set up.

The pressure relief valve installation is hopefully closed most of, if not all, the time, and it is rather like a bottle stuck to the pipe with the fluid blowing past the opening setting up a similar vibration. This does not necessarily cause a problem. However, if the size of the chamber and the velocity of the flowing fluid are not considered adequately, a vibration can be set up that is in resonance with the natural frequency of the chamber created by the valve.

If this resonance is set up, the pressure in that chamber can begin to oscillate and bump or exceed the set pressure of the valve, which causes it to chatter open and a loss of steam occurs. Over a long time, the valve seat may incur damage and leak more or less constantly. This most often occurs in systems where the capacity is changed. There have been reported cases of utilities that have retrofitted properly designed installation fittings and reduced the operating costs by thousands of dollars.

Most new systems have the problem designed away at first construction, thereby eliminating the problem before it occurs. One problem is that the designer-operator must know and address the issue. There is much literature that addresses the problem and many empirical studies to suggest “design solutions.” Solutions are available. At this time, the best resources might be the fitting manufacturers of installation hardware and the manufacturers of the relief valves.

To relate back to B31.1, the code requires that some pressure protection be installed and in its paragraphs gives requirements for the same. The base code refers to the nonmandatory appendix as a source of installation guidance.

Expansion Joints

Expansion joints are used quite often and have a set of unique problems in piping systems. These problems are in reality offset by the piping problems that they resolve. The Expansion Joint Manufacturers Association (EJMA) publishes a resource on the design of expansion joints. Code B31.3 has established an Appendix X, Metallic Bellows Expansion Joints.

As stated in the general paragraph of the appendix, it does not specify design details that are the responsibility of the expansion joint manufacturer. That manufacturer's responsibilities, in Paragraph X302, include

- All piping and components within the end connections of the assembly they supply
- Specifying any need for supports or restraints external to their assembly
- Determining the design conditions for all their supplied components which do not come in contact with the fluid

These are based on the information given to them by the piping designer who is responsible for giving the expansion joint manufacturer the following:

- Static design conditions, including a design temperature for the metal if other than that of the fluid
- Cyclic design conditions, including transient conditions separately
- Any other loads such as wind and seismic
- Properties of the fluid

Basically, the bellows type of expansions will be designed in accordance with EJMA standards unless otherwise permitted or required in the appendix. Any other design shall be qualified as required by B31.3, Paragraph 304.7.2. There are factors of safety specified. Those are not less than 2.25 on squirm design. And the factor of safety on ultimate rupture shall not be less than 3.0.

For those not totally familiar with the term *squirm*, it is a term that relates to the way an expansion joint leaves the straight, or design, line or shape. Some might relate to the toy Slinky.

The appendix has a design fatigue curve for bellows of austenitic stainless steel. This is based on EJMA nomenclature and is derived from ASME paper, with adjustments for empirical testing to make it reflect the best knowledge.

In Paragraph X302.1.3, the fatigue analysis requirements are set out. Whenever a new fabrication process or a new material is contemplated, fatigue testing is required. That test procedure is spelled out in subparagraph (c) of that paragraph. Subparagraph (d) requires that a minimum of five tests each for reinforced and unreinforced be carried out.

The examination and tests required are set out in Paragraphs X302.2.2 and 302.2.3. The *leak test pressure* is defined to include the ratio of the modulus of elasticity at test temperature to that at design temperature rather than the ratio of the stresses. There is also a stability pressure based on column instability. There then is a reference to the EJMA standards.

Aluminum Flanges

Aluminum is used quite frequently in certain, especially cryogenic, industries. Aluminum is not a recognized material in the flange standard B16.5. In B31.3, the designer was left with the design process as outlined in paragraphs 304.5.1(b) and 304.5.2(b).

The committee decided that this full design process was not needed. They included, in their book Appendix L. This basically is the only place where the pressure-temperature ratings of aluminum flanges are defined in any event.

This is limited to certain design flanges. Those are class 150, class 300, and class 600. In addition, it is limited to the aluminum alloys as follows:

- 3003-H112
- 6061-T6 rating for weld neck lapped and blind
- 6061-T6 rating for slip-on and socket welding flanges

The user may use the dimensions and tolerances specified in the dimensional tables for those classes of flanges in B16.5. He or she then may use them for the pressure ratings in Appendix L.

There is both a metric (SI) set of tables and a USCS set of tables. And as a note, since the newest edition of B16.5 has been metricated, the conversion has been completed. If at some time in the future B16.5 includes those materials in the allowed materials tables, it is presumed that this appendix would be withdrawn.

There are several other appendices that may be of interest to the reader but that are specialized enough to not warrant any extensive discussion in this book. Note that B31.1, B31.3, and B31.8 are the most prolific appendix codes.

Before the mention of any appendices that may be of interest to the reader, a general comment might be made about the books that apply.

In the early chapters of the book, each code has a set of definitions that the committees deem to be important. Quite often they have the same definitions for the same terms, but in no case is it a complete duplication as to which terms are defined. Perusing these definitions can be quite informative regarding what is important to the particular service that the code addresses.

It is also quite common for the codes to have both a foreword and an introduction. Many times those few pages have duplicate subjects of address. Often they afford an insight into the history of the particular code.

There is one sentence that is found in this area that is universal and one that users of the code should be aware of at all times as they progress through the process of a particular piping project: "The designer is cautioned that the Code is not a design handbook; it does not do away with the need for the designer or for competent engineering judgment."

The user of a code would do well to read the foreword and introduction on some frequency basis. It is quite appropriate for all to be reminded of what the code and we the readers are here for.

In many of the codes, there are sketches intended to guide the reader insofar as what the boundaries may be for a particular code. Interestingly the ones in B31.8 are in Appendix Q. They attempt to define an offshore line, a transmission line, and a distribution system.

Other appendices are recommended as a good resource:

- Many of the codes have appendices which define the properties or allowable stresses. These are not mentioned further.
- Codes B31.1 and B31.3 both have appendices concerning nomenclature; B31.8 has one in process. These are quite handy in finding where a particular symbol is used in the code. In B31.1 it is Appendix G, and in B31.3 it is Appendix J.
- Code B31.3 has Appendix F, Precautionary Considerations. This appendix is quite helpful and is referenced in many other standards. It states concerns that may not be needed in the base code but could quite often be used in engineering judgment, as mentioned above.
- Code B31.3 has Appendix M as a guide to classifying fluid service. It will be remembered that the code has three classifications of fluid service: D, ordinary, and M.
- Code B31.8 has an appendix describing the testing of welders for very low (<20 percent of SMYS) hoop stress. It is Appendix G.
- Code B31.8, Appendix H, is a flattening test for pipe.
- Code B31.8, Appendix N, is the recommended practice for testing pipelines in place.

And with the enumeration of the appendices, the book is finished.

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Appendix

A

Area Replacement Drawings

The drawings in this appendix illustrate many of the differences in the various codes. As mentioned in the text, Code B31.9 has a unique way of establishing the requirement. The charts in B31.9 are relatively self-explanatory. Code B31.9 uses limited materials, and when those materials are used within the scope of the chart, nothing much more need be done. Then of course if something needs to be done, that code defers to Code B31.1.

Of the other codes, each has two basic setups. The first is the welded branch connection. All the sketches are for the fundamental solution of adding a pad of material and welding that pad according to the rules and in general accordance with the drawings. Note that there are no prohibitions in the codes for another method of achieving the base requirement of having enough reinforcement metal in the correct place. The options are discussed in the main text. Note that the buried pipe codes lay out specific rules as to when and where each method of reinforcement can be used. These, too, are discussed in the main text.

The second basic setup is the extruded header, in which the transition for the attachment from the header to the branch is produced using the header. This is analogous to the way many tees are made. The fundamental process is to pull the lip, which is where the branch will be attached to the header, to create the weld preparation. This is usually done by placing a forming tool in a hole and then pulling the lip up. The hole in the header is usually considerably smaller than the ID of the intended branch, which allows the material for the lip to protrude after the forming process and an appropriate weld preparation to be made.

Careful examination of the drawings leads to the conclusion that less material is required in this type of arrangement. That is due to the fact that the transition from a horizontal header to a vertical branch is essentially a smooth curve. This would not have the stress risers one can expect from any sort of groove weld. There are, however, restrictions on the radius that is formed by this process.

One could also note that the ratio of the branch diameter to the header diameter has some upper limit. This is related to the fact that the transverse length of the material—that side which is at 90° to the axis of the header pipe—would have to reach to above the top of the header. The larger the branch, the harder that is to achieve.

In general, for technical reasons this type of construction is limited to a 90° intersection. In the recent years, there has been some success with extruding an intersection at some angle less than 90°, but the process has not been developed to the point that is commonplace.

This is often used in manifold situations where there are several branches on a specific length of header pipe. This situation often occurs in the pipeline industry. The extruded header is generally the method of choice as much for economic reasons of construction costs as for any other reason.

It can be shown that area replacement and in some cases its near cousin, the pressure area method, are very simple processes for the design process. Many tests have shown that in general it is very conservative, regarding the use of metal and construction costs, compared to many of the other methods used today. The branch outlet fitting and all its variations were invented and developed in the 1930s and have been successfully used in many applications, eliminating the relatively simple calculations required by the area replacement.

The reader is cautioned that, as one moves from code to code, there are some subtle but important differences in the exact method of calculation. One company uses an entirely different program for each code to cover that diversity.

FIGURE NUMBERS FOR
AREA REPLACEMENT BY CODE
(also shown on each of the following figures)

B31.1

1. B31.1 Fig. 104.3.1(D)
2. B31.1 Fig. 104.3.1 (G)

B31.3

1. B31.3 Fig. 304.3.3
2. B31.3 Fig. 304.3.4

B31.4

1. B31.4 Fig. 404.3.1(b)(3)
2. B31.4 Fig. 404.3.1(d)(2)

B31.5

1. B31.5 Fig. 504.3.1-A
2. B31.5 Fig. 504.3.1-B
3. B31.5 Fig. 504.3.1-C

B31.8

1. B31.8 App. F Fig. F1
2. B31.8 App. F Fig. F2
3. B31.8 App. F Fig. F3
4. B31.8 App. F Fig. F4
5. B31.8 App. F Fig. F5

B31.9

1. B31.9 Fig. 904.3.3A
2. B31.9 Fig. 904.3.3B
3. B31.9 Table 904.3.3

B31.11

1. B31.11 Fig. 1104.3.1(b)(3)
2. B31.11 Fig. 1104.3.1(d)(2)

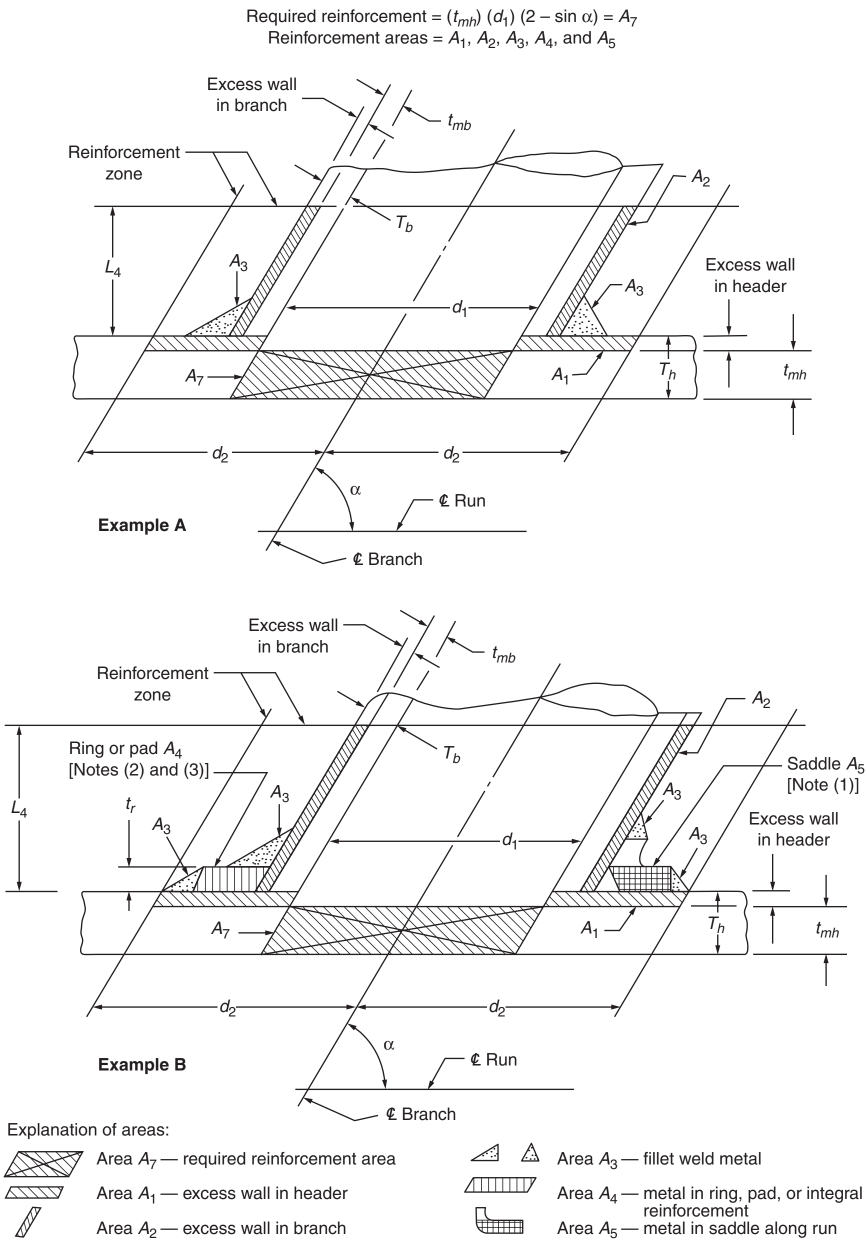
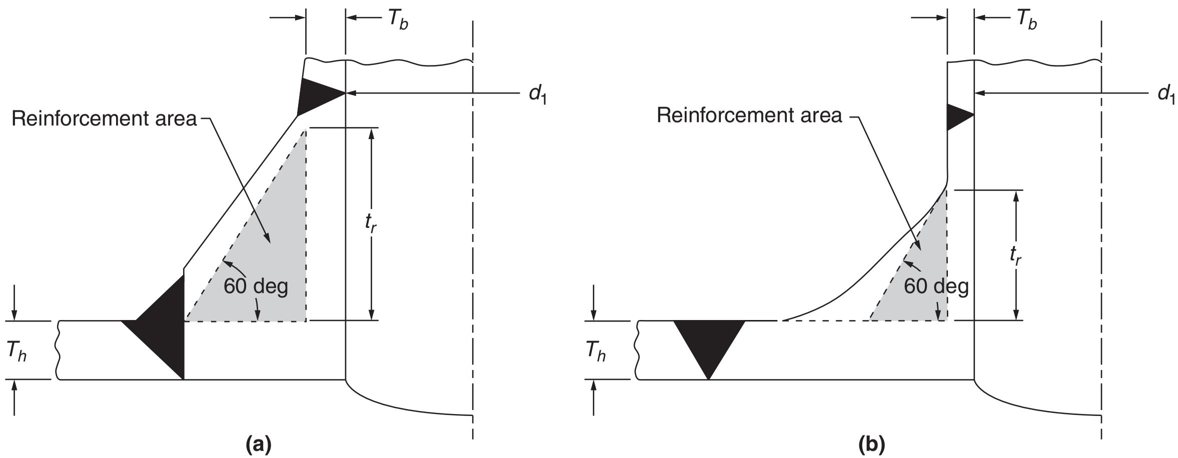


Figure A.1 Code Figure B31.1 Fig. 104.3.1(D).

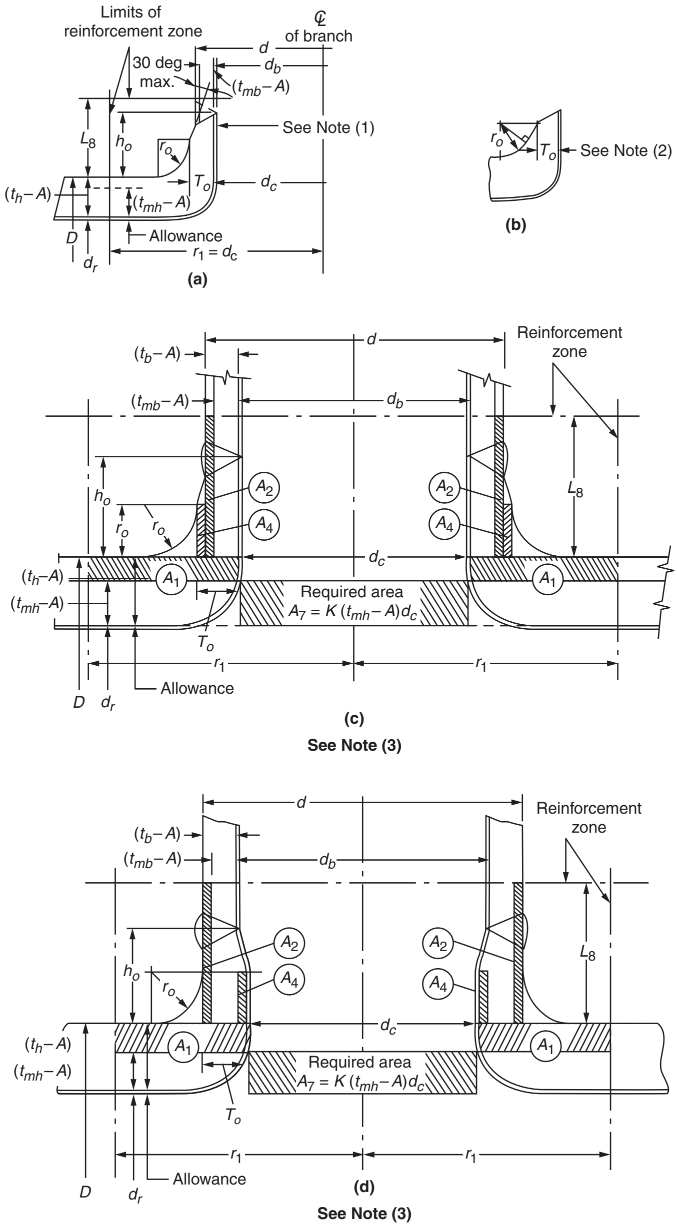


Example C

NOTES:

(1) Reinforcement saddles are limited to use on 90° branches (Example B).
(2) When a ring or pad is added as reinforcement (Example B), the value of reinforcement area may be taken in the same manner in which excess header metal is considered, provided the weld completely fuses the branch pipe, header pipe, and ring or pad. Typical acceptable methods of welding which meet the above requirement are shown in Fig. 127.4.8(D), sketches (c) and (d).
(3) Width to height of rings and pads shall be reasonably proportioned, preferably on a ratio as close to 4:1 as the available horizontal space within the limits of the reinforcing zone along the run and the outside diameter of the branch will permit, but in no case may the ratio be less than 1:1.

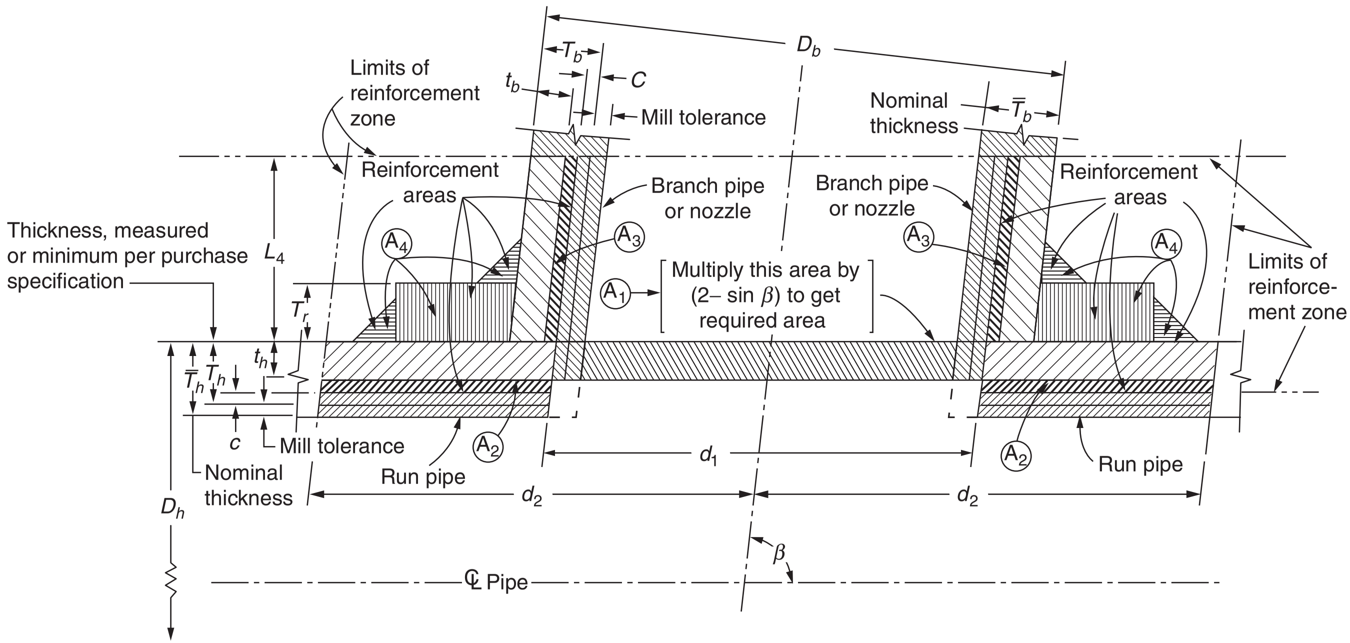
Figure A.1 (*Continued*)



NOTES:

(1) Taper bore inside diameter (if required) to match branch pipe 1:3 maximum taper
(2) Sketch to show method of establishing T_o when the taper encroaches on the crotch radius
(3) Sketch is drawn for condition where $k = 1.00$.

Figure A.2 Code Figure B31.1 Fig. 104.3.1(G).



GENERAL NOTE: This figure illustrates the nomenclature of para. 304.3.3. It does not indicate complete welding details or a preferred method of construction. For typical weld details, see Fig. 328.5.4D.

Figure A.3 Code Figure B31.3 Fig. 304.3.3, branch connection nomenclature.

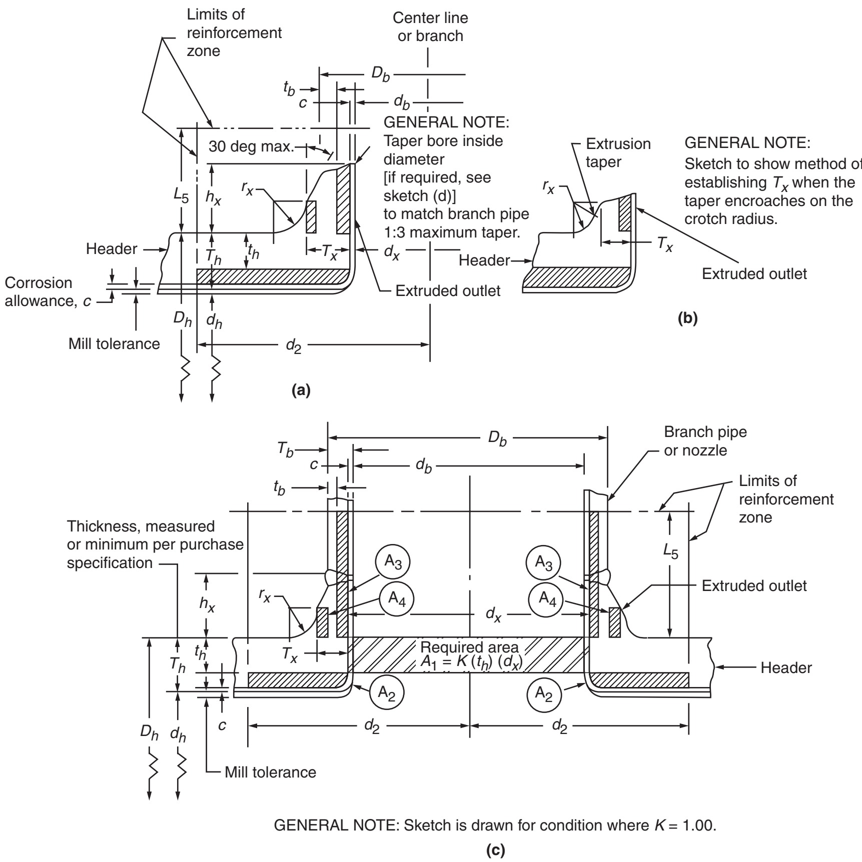
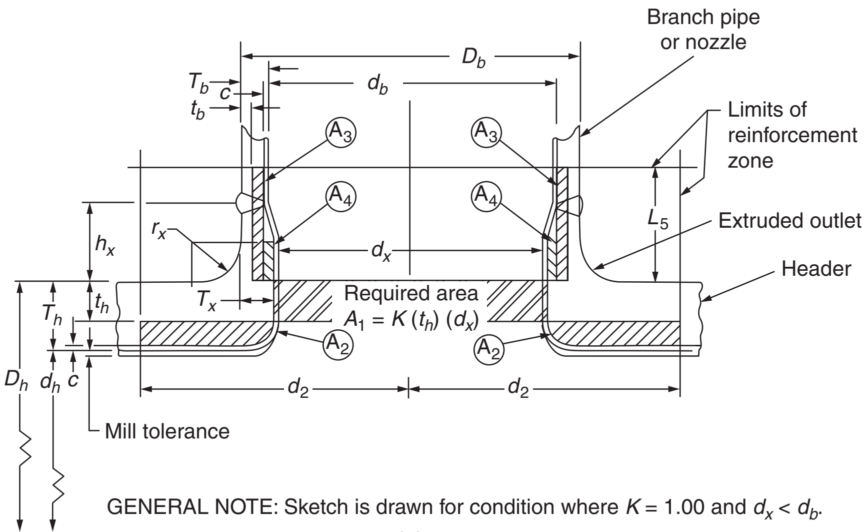


Figure A.4 Code Figure B31.3 Fig. 304.3.4.



GENERAL NOTE: Sketch is drawn for condition where $K = 1.00$ and $d_x < d_b$.

(d)

Figure A.4 (*Continued*)

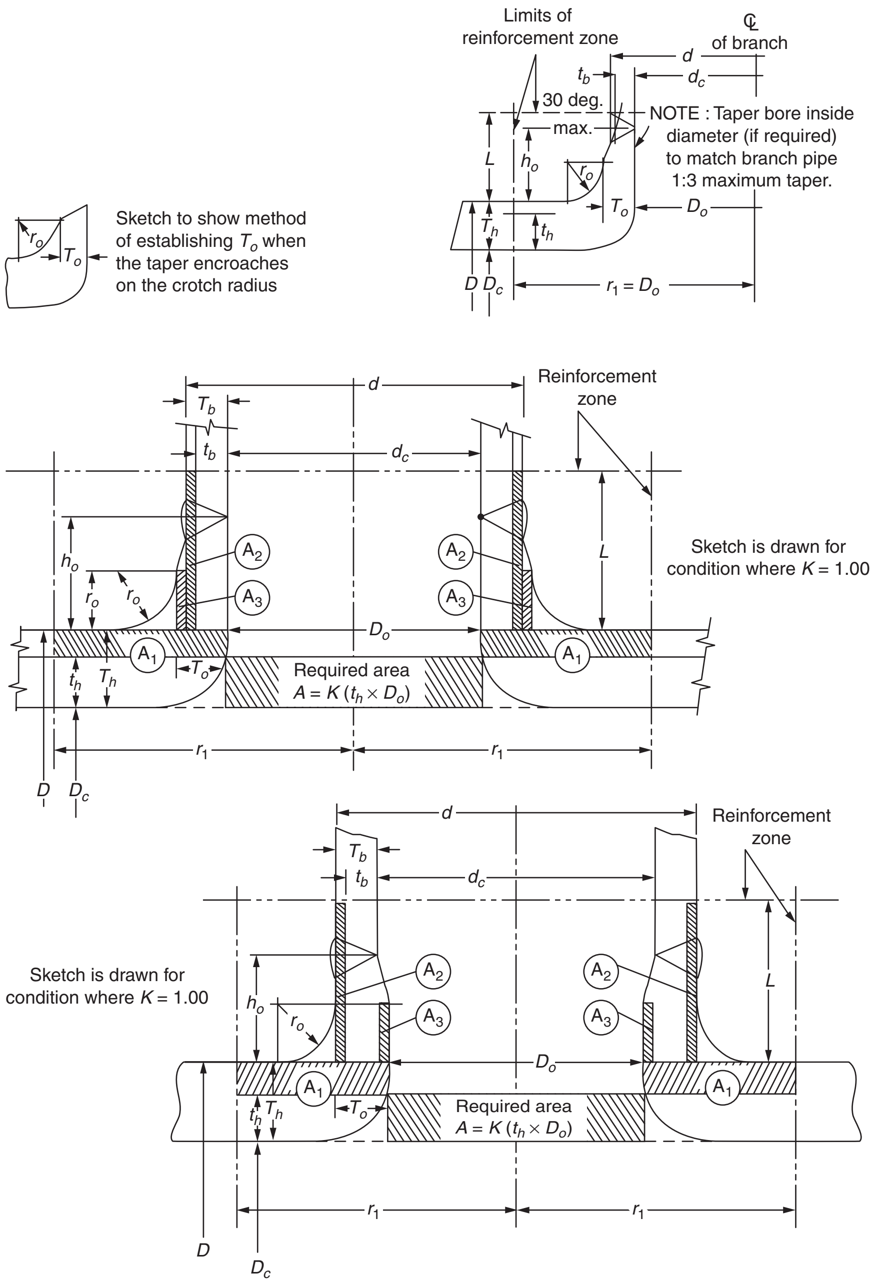
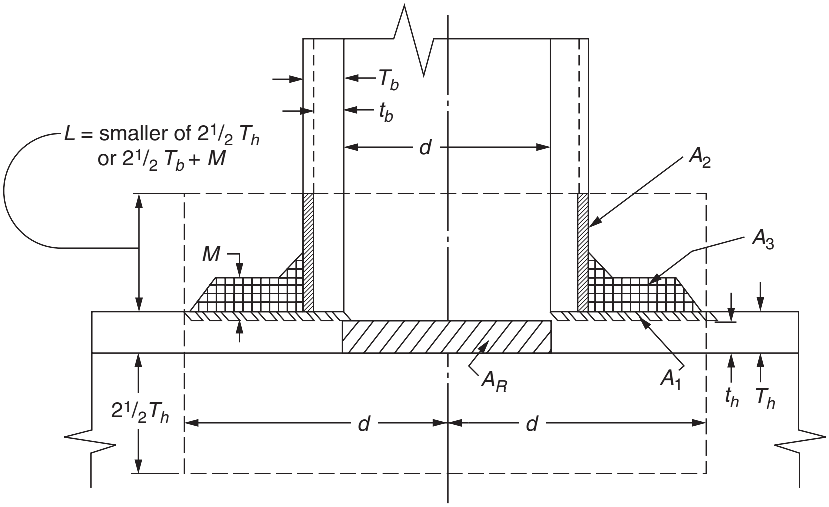
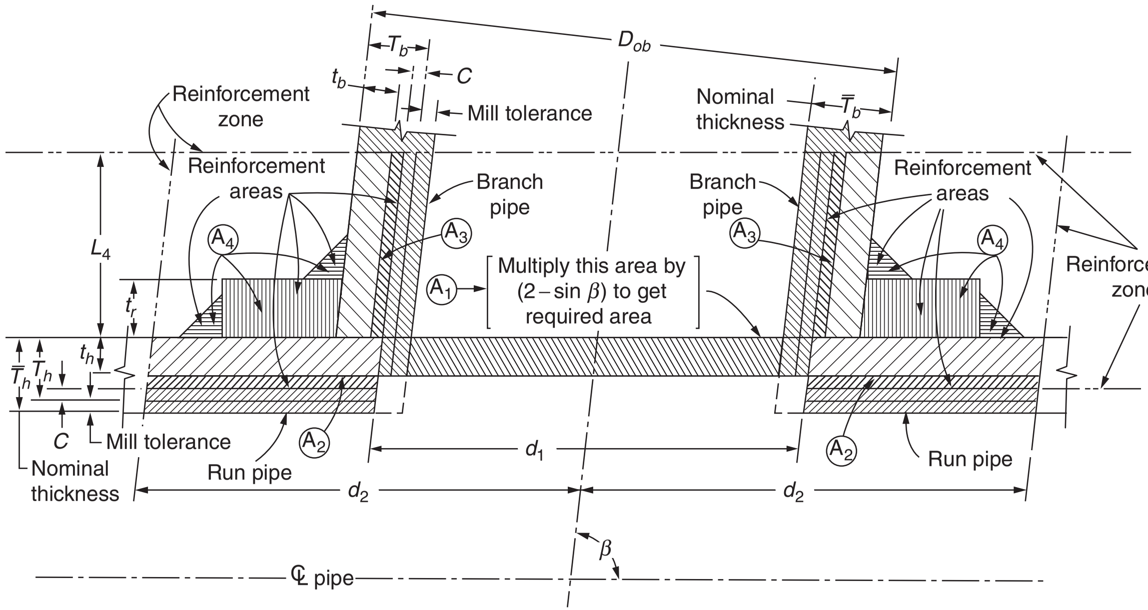


Figure A.5 Code Figure B31.4 Fig. 404.3.1(b)(3).



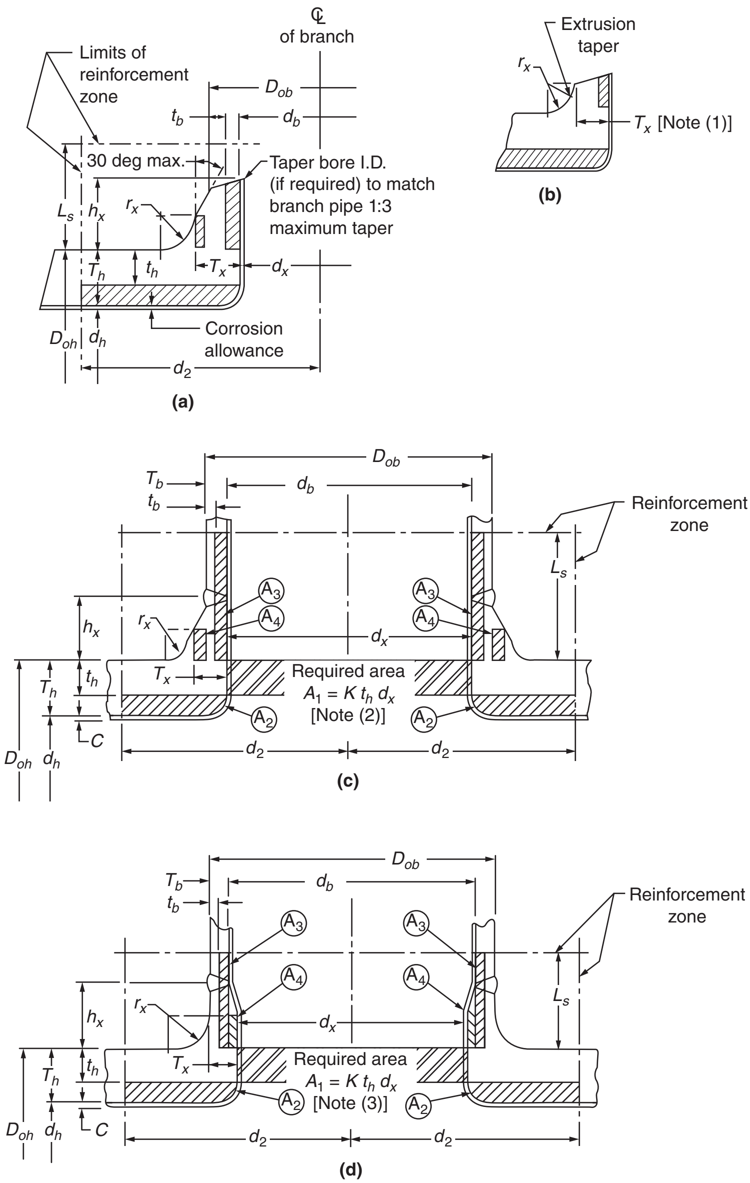
"Area of reinforcement" enclosed by ——–—– lines
Reinforcement area required $A_R = dt_h$
Area available as reinforcement = $A_1 + A_2 + A_3$
$A_1 = (T_h - t_h)d$
$A_2 = 2\ (T_b - t_b)L$
A_3 = summation of area of all added reinforcement, including weld areas that lie within the "area of reinforcement"
$A_1 + A_2 + A_3$ must be equal to or greater than A_R
where
T_h = nominal wall thickness of header
T_b = nominal wall thickness of branch
t_b = design branch wall thickness required by para, 404.1.2
t_h = design header wall thickness required by para, 404.1.2
d = length of the finished opening in the header wall (measured parallel to the axis of the header)
M = actual (by measurement) or nominal thickness of added reinforcement

Figure A.5 Code Figure B31.4 Fig. 404.3.1(d)(2) (*Continued*)



GENERAL NOTE: This figure is merely to illustrate the notation of para. 504.3.1(f) and does not indicate complete welding details, or a preferred method of construction. For typical weld details, see Fig. 527.4.6-D.

Figure A6-A Code Figure B31.5 Fig. 504.3.1-A, reinforcement of branch connections.



GENERAL NOTE: This figure is merely to illustrate the notations of para. 504.3.1(g) and does not indicate complete welding details, or a preferred method of construction.

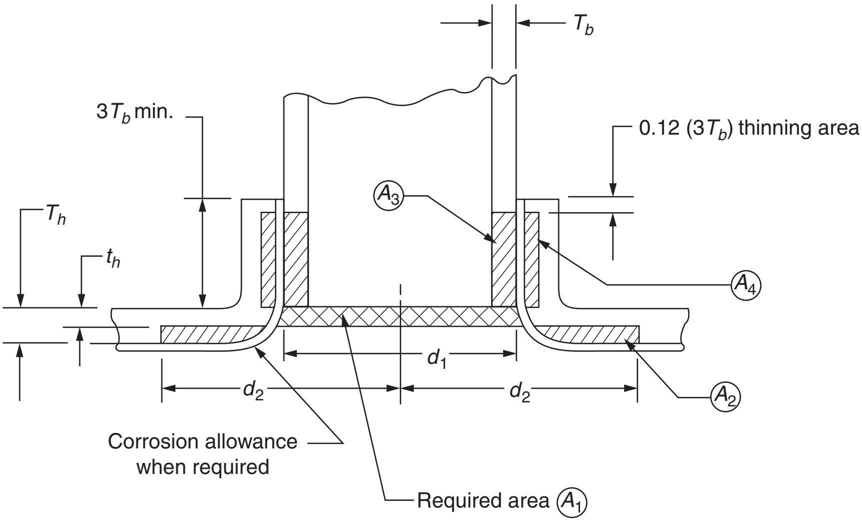
NOTES:

(1) Sketch to show method of establishing T_x when the taper encroaches on the crotch radius.

(2) Sketch is drawn for condition where $K = 1.00$.

(3) Sketch is drawn for condition where $K = 1.00$ and $d_x < d_b$.

Figure A6-B Code Figure B31.5 Fig. 504.3.1-B (*Continued*)

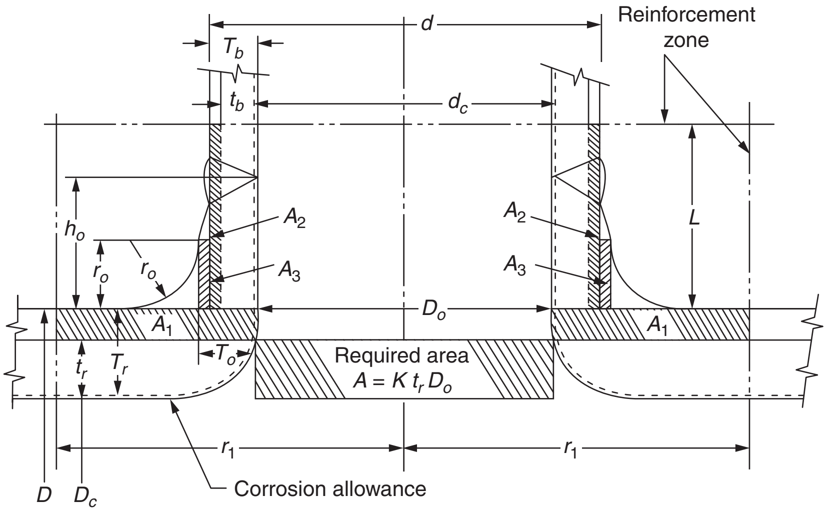


A_1 = required area, sq in. (sq mm) = $t_h d_1$
A_2 = area lying within the reinforcement zone resulting from any excess thickness available in the header wall
A_3 = area lying within the reinforcement zone resulting from any excess thickness in the branch tube wall
A_4 = area lying within the reinforcement zone resulting from any excess thickness available in the extruded lip

$A_2 + A_3 + A_4 \geq A_1$

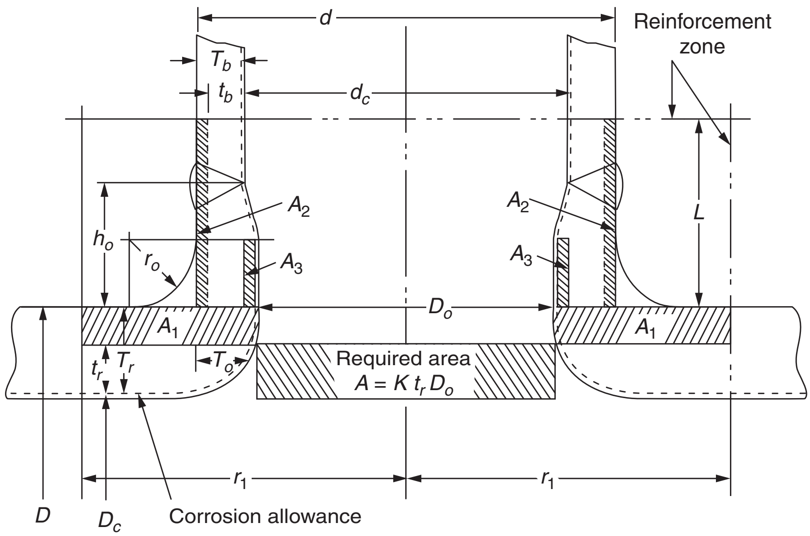
T = actual thickness of tube wall
b = branch
d_1 = opening size in header tube
d_2 = d_1 = reinforcement zone
h = header
t = pressure design thickness

Figure A6-C Code Figure B31.5 Fig. 504.3.1-C (*Continued*)



GENERAL NOTE: Sketch is drawn for condition where $K = 1.00$.

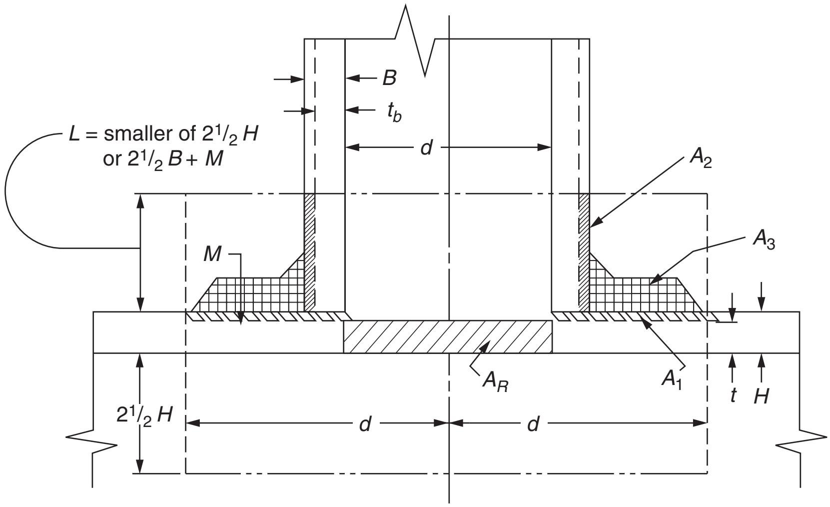
F-3



GENERAL NOTE: Sketch is drawn for condition where $K = 1.00$.

F-4

Figure A.7 Code Figure B31.8.



Area of reinforcement enclosed by ——— -- ——— lines.
Reinforcement area required $A_R = dt$
Area available as reinforcement = $A_1 + A_2 + A_3$

$A_1 = (H - t)(d)$ (if negative, use zero for value of A_1)
$A_2 = 2(B - t_b)L$
A_3 = summation of area of all added reinforcement, including weld areas that lie within the area of reinforcement
$A_1 + A_2 + A_3$ must be equal to or greater than A_R
where

B = nominal wall thickness of branch
H = nominal wall thickness of header
M = actual (by measurement) or nominal thickness of added reinforcement
d = the greater of the length of the finished opening in the header wall measured parallel to the axis of the run or the inside diameter of the branch connection
t = required nominal wall thickness of the header (under the appropriate section of the Code)
t_b = required nominal wall thickness of the branch (under the appropriate section of the Code)

F-5

Figure A.7 (*Continued*)

F1 EXTRUDED HEADERS

Definitions and limitations applicable to Figs. F1 through F4 are as follows:

D = outside diameter of run
D_c = corroded internal diameter of run
D_o = corroded internal diameter of extruded outlet measured at the level of the outside surface of run
L = height of the reinforcement zone
$= 0.7\sqrt{\mathrm{d}T_0}$
T_b = actual thickness of branch wall, not including corrosion allowance
T_r = actual thickness of the run wall, not including the corrosion allowance
T_o = corroded finished thickness of extruded outlet measured at a height equal to r_o above the outside surface of the run
d = outside diameter of branch pipe
d_c = corroded internal diameter of branch pipe
h_o = height of the extruded lip. This must be equal to or greater than r_o, except as shown in limitation (b) of r_o below.
r_l = half width of reinforcement zone (equal to D_o)
r_o = radius of curvature of external contoured portion of outlet measured in the plane containing the axes of the run and branch. This is subject to the following limitations:

(*a*) *Minimum Radius*. This dimension shall not be less than $0.05d$, except that on branch diameters larger than 30 in., it need not exceed 1.50 in.

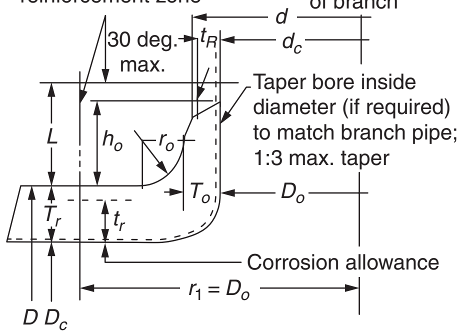
(*b*) *Maximum Radius*. For outlet pipe sizes NPS 6 and larger, this dimension shall not exceed $0.10d + 0.50$ in. For outlet pipe sizes less than NPS 8, this dimension shall not be greater than 1.25 in.

(*c*) When the external contour contains more than one radius, the radius on any arc sector of approximately 45 deg shall meet the requirements of (a) and (b) above.

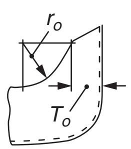
(*d*) Machining shall not be employed to meet the above requirements.

t_b = required thickness of branch pipe according to the steel pipe design formula of Para. 841.11, but not including any thickness for corrosion
t_r = required thickness of the run according to the steel pipe design formula Para. 841.11, but not including any allowance for corrosion or under-thickness tolerance

[1]See para. 831.6



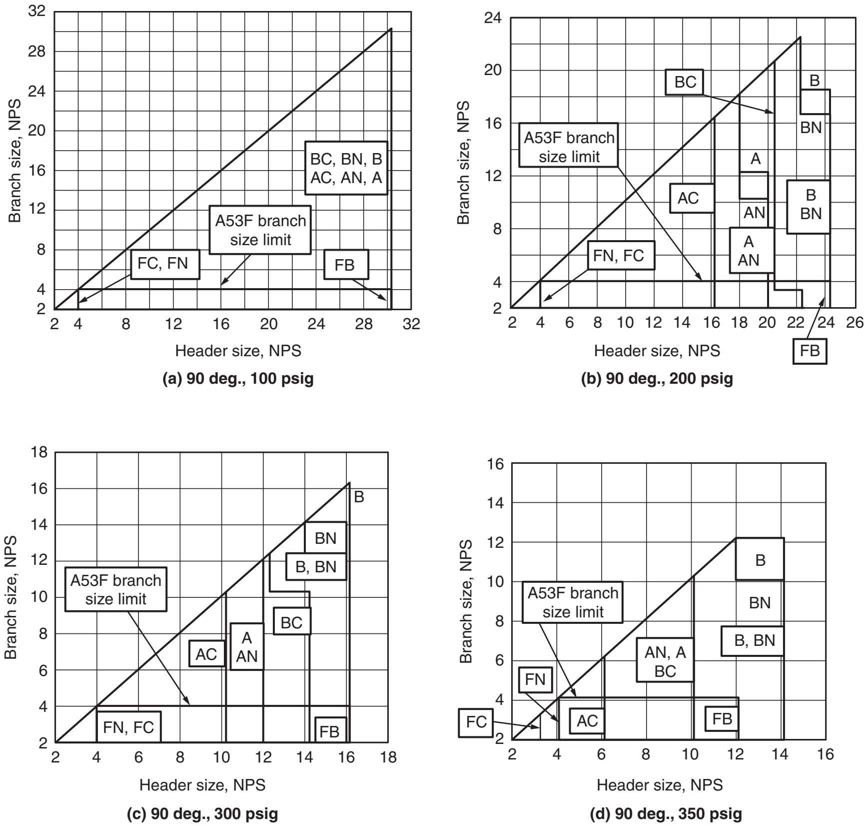
F-1



GENERAL NOTE: Sketch to show method of establishing T_o when the taper encroaches on the crotch radius.

F-2

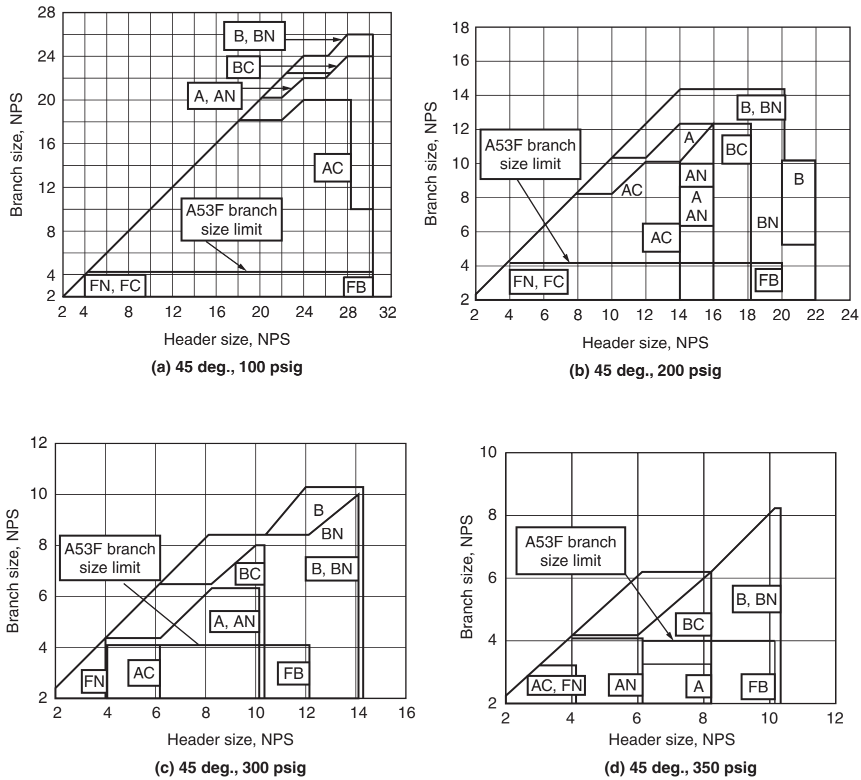
Figure A.7 (*Continued*)



GENERAL NOTES:

(a) See Table 904.3.3 for instructions on the use of this figure.
(b) A 12.5% mill tolerance and a 1/32 in. corrosion allowance have been used in the calculations for this figure.
(c) The pipe size limit for this Code is NPS 30. The sketches end at that size, but allowable unreinforced branches may extend to larger sizes for some materials and pressures.
(d) A53 Type F, butt weld pipe, is limited to NPS 4 as it is not available above that size.
(e) This figure is based on the rules of B31.1.

Figure A.8 (*a*) Code Figure B31.9 Fig. 904.3.3A.



GENERAL NOTES:

(a) See Table 904.3.3 for instructions on the use of this figure.
(b) A 12.5% mill tolerance and a 1/32 in. corrosion allowance have been used in the calculations for this figure.
(c) The pipe size limit for this Code is NPS 30. The sketches end at that size, but allowable unreinforced branches may extend to larger sizes for some materials and pressures.
(d) A53 Type F, butt weld pipe, is limited to NPS 4 as it is not available above that size.
(e) This figure is based on the rules of B31.1.

Figure A.8 Code Figure B31.9 Fig. 904.3.3B (*Continued*)

TABLE 904.3.3
HEADER AND BRANCH MATERIALS FOR STANDARD WALL PIPE[1]

	Header				Branch			
Symbol	Material	Type	S_h	E_h	Material	Type	S_b	E_b
B	A53B, A106B	SML	15,000	1.00	A53B, A106B	SML	15,000	1.00
BN	A53B, A135B (Header seam not cut by branch)	ERW	15,000	1.00	A53B, A135B	ERW	15,000	0.85
BC	A53B, A135B (Header seam cut by branch)	ERW	15,000	0.85	A53B, A135B	ERW	15,000	0.85
A	A53A, A106A	SML	12,000	1.00	A53A, A106A	SML	12,000	1.00
AN	A53A, A135A (Header seam not cut by branch)	ERW	12,000	1.00	A53A, A135A	ERW	12,000	0.85
AC	A53A, A135A (Header seam cut by branch)	ERW	12,000	0.85	A53A, A135A	ERW	12,000	0.85
FB	A53B, A135B	ERW	15,000	1.00	A53 Type F	BW	11,250	0.60
FN	A53 Type F	BW	11,250	1.00	A53 Type F	BW	11,250	0.60
FC	A53 Type F	BW	11,250	0.60	A53 Type F	BW	11,250	0.60

NOTE:

(1) This Table is to be used in conjunction with Figs. 904.3.3A and B. Select the header and branch materials from the Table. Using the appropriate symbol for the materials and the appropriate sketch for the pressure and branch intersection angle, determine if the combination of header and branch size to be used can be made without reinforcement. The sizes below and to the left of the tagged line within each sketch do not need reinforcement.

Figure A.8 Code Figure B31.9 Table 904.3.3 (*Continued*)

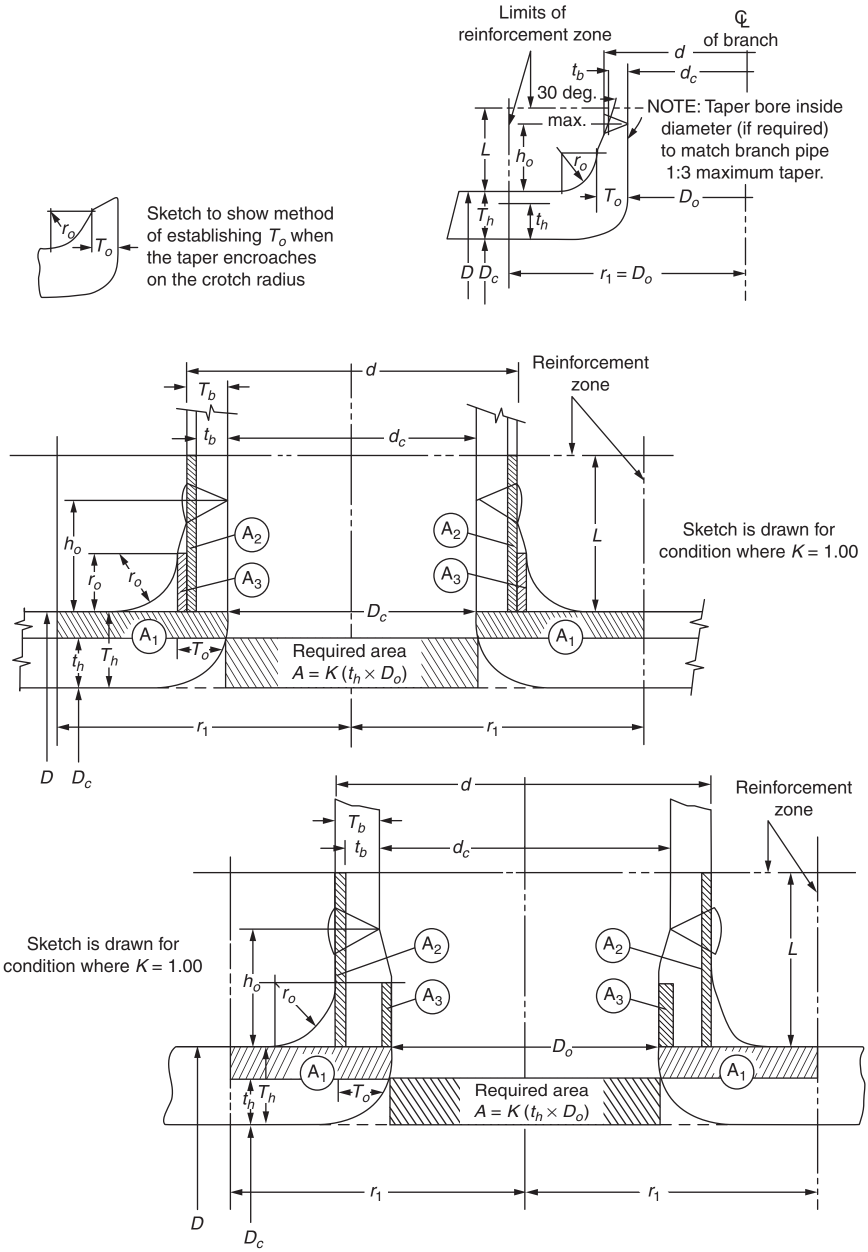
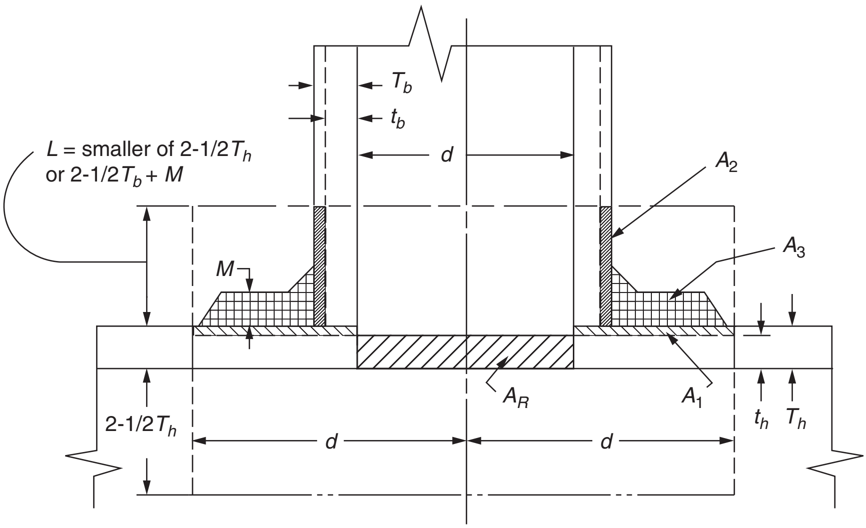


Figure A.9 Code Figure B31.11 Fig. 1104.3.1(b)(3).



"Area of reinforcement" enclosed by ——— - - ——— lines

Reinforcement area required $A_R = dt_h$

Area available as reinforcement = $A_1 + A_2 + A_3$

$A_1 = (T_h - t_h)d$

$A_2 = 2(T_b - t_b)L$

A_3 = summation of area of all added reinforcement, including weld areas that lie within the "area of reinforcement"

$A_1 + A_2 + A_3$ must be equal to or greater than A_R

where

- T_h = nominal wall thickness of header
- T_b = nominal wall thickness of branch
- t_b = design branch wall thickness required by para. 404.1.2
- t_h = design header wall thickness required by para. 404.1.2
- d = length of the finished opening in the header wall (measured parallel to the axis of the header)
- M = actual (by measurement) or nominal thickness of added reinforcement

Figure A.9 Code Figure B31.11 Fig. 1104.3.1(d)(2) (*Continued*)

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Appendix

B

Pipe Charts

Seamless and Welded Steel Pipe

Nominal Pipe Size	NOMINAL WALL THICKNESS FOR Sched. 5	Sched. 10	Sched. 20	Sched. 30	Standard*	Sched. 40	Sched. 60	Extra Strong †	Sched. 80	Sched. 100	Sched. 120	Sched. 140	Sched. 160	XX Strong
1/8		0.049			0.068	0.068		0.095	0.095					
1/4		0.065			0.088	0.088		0.119	0.119					
3/8		0.065			0.091	0.091		0.126	0.126					
1/2	.065	0.083			0.109	0.109		0.147	0.147				0.188	0.294
3/4	.065	.083			0.113	0.113		0.154	0.154				0.219	0.308
1	.065	.109			0.133	0.133		0.179	0.179				0.250	0.358
1¼	.065	.109			0.140	0.140		0.191	0.191				0.250	0.382
1½	.065	.109			0.145	0.145		0.200	0.200				0.281	0.400
2	.065	.109			0.154	0.154		0.218	0.218				0.344	0.436
2½	.083	.120			0.203	0.203		0.276	0.276				0.375	0.552
3	.083	.120			0.216	0.216		0.300	0.300				0.438	0.600
3½	.083	.120			0.226	0.226		0.318	0.318					0.636
4	.083	.120			0.237	0.237		0.337	0.337		0.438		0.531	0.674
5	.109	.134			0.258	0.258		0.375	0.375		0.500		0.625	0.750
6	.109	.134			0.280	0.280		0.432	0.432		0.562		0.719	0.864
8	.109	.148	0.250	0.277	0.322	0.322	0.406	0.500	0.500	0.594	0.719	0.812	0.906	0.875
10	.134	.165	0.250	0.307	0.365	0.365	0.500	0.500	0.594	0.719	0.844	1.000	1.125	1.000
12	.156	.180	0.250	0.330	0.375	0.406	0.562	0.500	0.688	0.844	1.000	1.125	1.312	1.000
14 O.D.	.156	.250	0.312	0.375	0.375	0.438	0.594	0.500	0.750	0.938	1.094	1.250	1.406	
16 O.D.	.165	.250	0.312	0.375	0.375	0.500	0.656	0.500	0.844	1.031	1.219	1.438	1.594	
18 O.D.	.165	.250	0.312	0.438	0.375	0.562	0.750	0.500	0.937	1.156	1.375	1.562	1.781	
20 O.D.	.188	.250	0.375	0.500	0.375	0.594	0.812	0.500	1.031	1.281	1.500	1.750	1.969	
22 O.D.	.188	.250	0.375	0.500	0.375		0.875	0.500	1.125	1.375	1.625	1.875	2.125	
24 O.D.	.218	.250	0.375	0.562	0.375	0.688	0.969	0.500	1.219	1.531	1.812	2.062	2.344	
26 O.D.		.312	0.500		0.375			0.500						
30 O.D.	.250	.312	0.500	0.625	0.375			0.500						
34 O.D.		.312	0.500	0.625	0.375	0.688		0.500						
36 O.D.		.312	0.500	0.625	0.375	0.750		0.500						
42 O.D.					0.375			0.500						

All dimensions are given in inches and are in accordance with ANSI B36.10 or ANSI B36.19 as applicable.
The decimal thickness listed for the pipe sizes represent their normal or average wall dimensions. The actual thickness may be as much as 12.5% under the normal thickness because of mill tolerance.

*For nominal sizes through 10", Standard Weight thickness are identical to Schedule 40 thicknesses.

†For nominal sizes through 8", Extra Strong thicknesses are identical to Schedule 80 thicknesses.

BORE CHART (I.D. OF PIPE)

Pipe Size	O.D.	Sched. 5	Sched. 10	Sched. 20	Sched. 30	Sched. 40	Sched. 60	Sched. 80	Sched. 100	Sched. 120	Sched. 140	Sched. 160	Std.	XS	XXS
1/8"	.405		.307			.269		.215					.269	.215	
1/4"	.540		.410			.364		.302					.364	.302	
3/8"	.675		.545			.493		.423					.493	.423	
1/2"	.840	.710	.674			.622		.546				.464	.622	.546	.252
3/4"	1.050	.920	.884			.824		.742				.612	.824	.742	.434
1"	1.315	1.185	1.097			1.049		.957				.815	1.049	.957	.599
1¼"	1.660	1.530	1.442			1.380		1.278				1.160	1.380	1.278	.896
1½"	1.900	1.770	1.682			1.610		1.500				1.338	1.610	1.500	1.100
2"	2.375	2.245	2.157			2.067		1.939				1.689	2.067	1.939	1.503
2½"	2.875	2.709	2.635			2.469		2.323				2.125	2.469	2.323	1.771
3"	3.500	3.334	3.260			3.068		2.900				2.624	3.068	2.900	2.300
3½"	4.000	3.834	3.760			3.548		3.364					3.548	3.364	2.728
4"	4.500	4.334	4.260			4.026		3.826		3.624		3.438	4.026	3.826	3.152
5"	5.563	5.345	5.295			5.047		4.813		4.583		4.313	5.047	4.813	4.063
6"	6.625	6.407	6.357			6.065		5.761		5.501		5.187	6.065	5.761	4.897
8"	8.625	8.407	8.329	8.125	8.071	7.981	7.813	7.625	7.437	7.187	7.001	6.813	7.981	7.625	6.875
10"	10.750	10.482	10.420	10.250	10.136	10.020	9.750	9.562	9.312	9.062	8.750	8.500	10.020	9.750	8.750
12"	12.750	12.438	12.390	12.250	12.090	11.938	11.626	11.374	11.062	10.750	10.500	10.126	12.000	11.750	10.750
14"	14.000	13.688	13.500	13.376	13.250	13.124	12.812	12.500	12.124	11.812	11.500	11.188	13.250	13.000	
16"	16.000	15.670	15.500	15.376	15.250	15.000	14.688	14.312	13.938	13.562	13.124	12.812	15.250	15.000	
18"	18.000	17.670	17.500	17.376	17.124	16.876	16.500	16.126	15.688	15.250	14.876	14.438	17.250	17.000	
20"	20.000	19.624	19.500	19.250	19.000	18.812	18.376	17.938	17.438	17.000	16.500	16.062	19.250	19.000	
24"	24.000	23.564	23.500	23.250	22.876	22.624	22.062	21.564	20.938	20.376	19.876	19.312	23.250	23.000	
30"	30.000	29.500	23.376	29.000	28.750								29.250	29.000	

Above data was compiled from existing standards and should be used as a guide.

Compiled from B36.10 & B36.19 by WFI, a division of Bonney Forge

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Appendix C

Sample Test Report

The purpose of this appendix is to show one way a “proof of design” test might be summarized in a report. Several of the standards that allow proof testing as a means to establish the pressure rating of the covered product require making the documentation available to users for their consideration. Few at present require any formal reporting. There is little guidance for the user or the manufacturer regarding what should be included in such reports. As users read any of the testing requirements, they could develop their own criteria as to what is important. And at a minimum the criteria should include:

1. What was tested
2. When it was tested
3. The materials involved
4. The results of that test

Consideration could be given to the manner in which the manufacturer controls the requirement that the tested product be representative of the production. That might also include geometric symmetry of the extended sizes from the tested size. As the modem analytical techniques grow in acceptance, some discussion of those might be appropriate.

Some test documentation constituted a simple statement to the effect that such and such a product was tested and then the conclusion that the test was successful. Such documentation essentially fails the test of allowing the reviewers the capability of judging for themselves the adequacy of the particular product for the system in which they intend to use it. It goes without saying that not all tests are successful.

Further, a specific intended use might demand more rigorous consideration than a specific test. The user/reviewer is cautioned that a set of documentation that would not allow evaluation on that basis would have little value.

The report in this appendix was taken from *The Engineering Data Book* of WFI Inc. of Houston, Texas (see Fig. C.1). It is a sample of the way that company presents its summary report of its test data. It is not intended to prescribe how one should present or what should be presented. It is offered as an example of how one leading company does present the data. Note that the actual test data including charts and graphs are available for consideration as prescribed. Certain administrative details such as signatures and page numbers have been omitted for clarity.

PROJECT NUMBER 5556-01
TEST NUMBER 01 RESULTS

On November 21, 1978, WFI International Inc. sponsored a test on a 20″ NPS × 12″ NPS (Std. Wt.) Buttweld Vesselet, to prove the adequacy of design of the integrally reinforced branch connection fitting. The Vesselet's design is in accordance with WFI's Engineered Proprietary Design Data.

The rules and guidelines of Section VIII, Division 1, Part UG-101 of the ASME Boiler and Pressure Code 1977 Edition and WFI's specification 1008-1, Revision 0, dated November 13, 1978, were followed.

The yield and burst pressures were compared to the test procedure of Section 8 of ANSI B16.9-1971 Edition and WFI's specification 2008-1, Revision 0, dated November 13, 1978.

The test was conducted by Southwestern Laboratories and was witnessed by Hartford Steam Boiler Inspection and Insurance Company.

As demonstrated by the full scale internal pressure proof test, the Vesselet restores the run and branch pipes to their original pressure retaining yield and burst strength, satisfying the code requirement of 100% strength replacement for the design of the integrally reinforced branch connection.

The actual pressure achieved at burst for this fitting was 3,200 psig, 162.5% of the pressure required by ANSI B16.9, 1971 Edition.

BURST TEST PRESSURE CALCULATIONS

I. PURPOSE

The purpose of these calculations is to determine the required yield and burst pressure of the test samples of this program. The calculated pressures will be based on the component with lowest calculated yield and/or burst pressure.

II. CALCULATIONS

The following calculations are based on the rules and guidelines of Section VIII, Division 1, Part UG-101 of the ASME Boiler and Pressure Code 1977 Edition per WFI's specification 1008-1, Revision 0, dated November 13, 1978.

a) Yield Pressure

$$P = \frac{2St}{D}$$

where P = computed pressure at the yield point of pipe that the fitting marking identifies, psig
S = minimum specified yield strength of the pipe
t = nominal pipe wall thickness, inches. For the purpose of this formula, t is defined as $87\frac{1}{2}$ percent of the nominal thickness of the pipe for which the fitting is recommended for use.
D = specified outside diameter of pipe

b) Burst Pressure

$$P = \frac{2St}{D}$$

where P = computed burst pressure of the pipe that the fitting marking identifies, psig
S = minimum specified tensile strength of the pipe
t = nominal pipe wall thickness, inches. For the purpose of this formula, t is defined as $87\frac{1}{2}$ percent of the nominal thickness of the pipe for which the fitting is recommended for use.
D = specified outside diameter of pipe

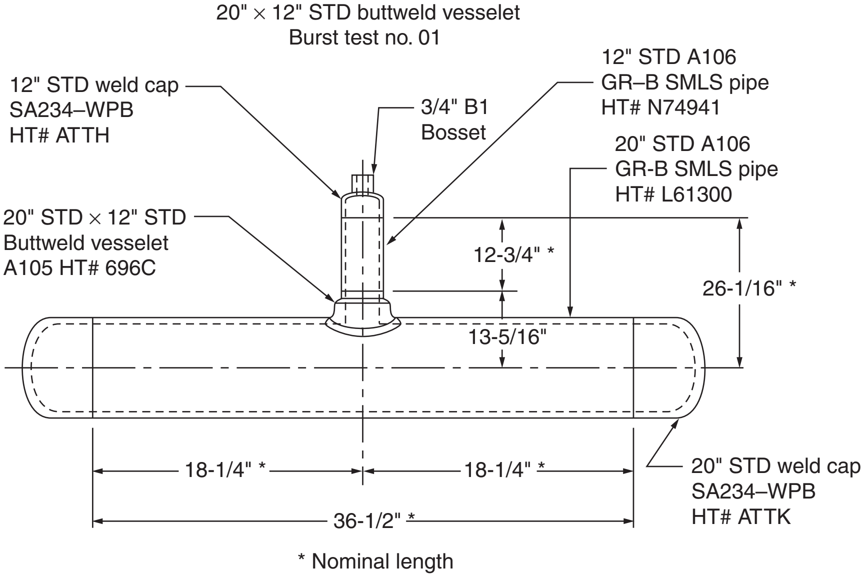

Figure C.1 Assembly fabrication drawing.

III. HEAT CODE DESCRIPTION OF THE FITTINGS AND PIPE

20″ × 12″ (Std Wt) Buttweld Vesselet	No. 696C
20″ (Std Wt) Buttweld Cap	No. ATTK
20″ (Std Wt) Pipe	No. L61300
12″ (Std Wt) Buttweld Cap	No. ATTH
12″ (Std Wt) Pipe	No. N74941
3/4″ B1 Bosset	Not Available

IV. DESCRIPTION OF PIPE AND SPECIFIED MINIMUM PROPERTIES

Material	O.D. in.	Nominal wall in.	Min. spec yld. strg. psi	Min. spec. ten. strg. psi
A106-B SMLS	20.000	0.375	35,000	60,000
A106-B SMLS	12.750	0.375	35,000	60,000

V. DESCRIPTION OF FITTING AND SPECIFIED MINIMUM PROPERTIES

Fitting description	Material	Min. spec yld. strg. psi	Min. spec. ten. strg. psi
20″ × 12″ (Std Wt) Buttweld Vesselet	ASTMA105	36,000	70,000

VI. ACTUAL PHYSICAL PROPERTIES OF THE TESTED PIPE AND FITTING

Fitting description	Heat code	Actual yld. strg. psi	Actual ten. strg. psi
20″ × 12″ Vesselet	696C	43,500	77,500
20″ Pipe	L61300	46,700	70,500
12″ Pipe	N74941	44,400	73,600

VII. CALCULATED YIELD PRESSURE

Component that yielded first	Calculated yield pressure	Actual pressure at yield	% of B16.9 yield
20″ Pipe, Gage #6	1148 psig	1600 psig	139.3%

VIII. CALCULATED BURST PRESSURE

Component that burst	Burst presssure	Pressure at burst	% of B16.9 burst pressure
20″ Pipe	1969 psig	3200 psig	162.5%

IX. SUMMARY

The test as described above was conducted by the rules and guidelines of Section VIII, Division 1-1977, Part UG-101. The test has shown that the WFI design philosophy of integrally reinforced branch connection fitting proved successful. These results satisfy the code requirement of 100% strength replacement for the design on integrally reinforced branch connections.

A comparison of the test data to the calculated yields and burst pressure using the procedure of ANSI B16.9 1971 Edition further reinforce the statement that the WFI design procedures for these type fittings are safe and conservative.

X. STRAIN GAGE DATA

Stress distribution test data in accordance with Section VIII, Division 1, Part UG-101, Paragraph (n) is available upon Request.

Engineering Approval

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BURST TEST NUMBER 01

Review of Design Proof Test (Section 9)
ANSI B16.9, 1978 Edition

I. PURPOSE

The purpose of this review is to qualify sizes for Burst Test Number 01 in accordance with Section 9, Subsections 9.3.1, 9.3.2, and 9.3.3 of 1978 Edition of ANSI B16.9.

II. FITTING QUALIFICATIONS (BASED ON THE BRANCH)

Size and Limiting Wall Thickness	Standard Schedules
6″ NPS (0.097″ W) through (0.585 W)	Sch 5S through Sch 120
8″ NPS (0.127″ W) through (0.761 W)	Sch 10S through Sch 120
10″ NPS (0.158″ W) through (0.949 W)	Sch 10S through Sch 120
12″ NPS (0.188″ W) through (1.125 W)	Sch 20 through Sch 140
14″ NPS (0.206″ W) through (1.235 W)	Sch 10 through Sch 120
16″ NPS (0.235″ W) through (1.412 W)	Sch 10 through Sch 120
18″ NPS (0.265″ W) through (1.588 W)	Sch 20 through Sch 140
20″ NPS (0.294″ W) through (1.765 W)	Sch 20 through Sch 140
22″ NPS (0.324″ W) through (1.941 W)	Sch 20 through Sch 140
24″ NPS (0.353″ W) through (2.118 W)	Sch 20 through Sch 140

III. SUMMARY

Burst Test Number 01 was conducted in strict accordance with the rules and guidelines of ANSI B16.9, 1971 Edition. Subsequent Editions (1978, 1986 and 1993) have changed the procedural and computational rules to alternative increase and/or decrease of the calculated target pressures. This change also affects the methodology of coordinating test materials. Those changes made apparent changes in the conclusions drawn by the test results. In this test, the test specimen returned the run pipe to its original retaining strength.

WFI has analyzed this test through the use of finite element techniques and statistical analysis. The results of this analysis show that if the test procedures and material coordinating methods dictated by the current codes were used, the results of the test would be equally as positive as those promulgated under the 1971 Edition rules. That analysis in conjunction with the extensive years of successful application make these test results valid.

Engineering Approval

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Review of Design Proof Test (Annex B)
MSS-SP-97, 1987 Edition

I. PURPOSE

The purpose of this review is to determine if Burst Test Number 01 meets the requirements of ANNEX B of MSS-SP-97.

II. CALCULATIONS

The following calculation is based on ANNEX B of MSS-SP-97, 1987 Edition.

The actual test pressure prior to rupture must at least equal the computed bursting strength of pipe as determined by the following formula:

$$P = \frac{2St}{D}$$

where P = computed bursting pressure of pipe that the fitting marking identifies, psig
S = actual tensile strength of the run pipe material that the fitting marking identifies, psig
t = nominal run pipe wall thickness, inches
D = specified outside diameter of the run pipe, inches

III. COMPUTED BURSTING PRESSURE OF PIPE, psig

Actual tensile strg.	O.D. in.	Nominal wall in.	P 100% (psig)	P 105% (psig)
70,500	20.000	0.375	2,643.75	2,775.94

IV. SUMMARY

For MSS-SP-97, 1987 Edition, the test assembly must withstand at least 100 percent of the calculated pressure. As an alternate, if the pipe fails before the assembly or if sufficient pressure to burst the assembly cannot be attained, the test pressure is acceptable if it is at least 105 percent of the proof test pressure.

V. CONCLUSION

The actual pressure achieved at burst for this fitting was 3,200 psig, 115.3 percent of the pressure required by MSS-SP-97, 1987 Edition.

Although this fitting is not directly covered by the rules and guidelines of MSS-SP-97, it does however show an analytical review of the test data to that of the controlling codes and standards.

The results show the fitting design meets and/or exceeds the requirements as set forth in ANNEX B of MSS-SP-97, 1987 Edition, and further reinforce the statement that WFI's patented Vesselet is a safe and conservative design.

Engineering Approval

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Appendix D

Stress

As mentioned in the text, Lamé developed a precise formula for the stresses in a cylinder. That formula can be written in many ways. Code B31.3 in previous editions published that formula in the format that fit the code requirements as formula 3c in Paragraph 304.1.2 of those earlier editions. It is also published as formula 3b, the Barlow equation.

The design task force decided that those formulas were redundant and simplified things by publishing only the present formula. The designer wishing to use the more rigorous Lamé equation has the right to do so. The Barlow formula will give a slightly thicker wall than the current formula which is perfectly acceptable under the rules of the code. So in effect, the permission need not have been expressed as it was implicit already.

For reference the equation published in the code as 3c is

$$t = \frac{D}{2}\left(1 - \sqrt{\frac{SE - P}{SE + P}}\right)$$

Formula 3b was

$$t = \frac{PD}{2SE}$$

where t = calculated thickness required before corrosion or mechanical allowances (in)
P = pressure, psig
D = outside diameter (in)
S = allowable stress, psi
E = weld joint efficiency factor (dimensionless)

Noted that as long as the units are consistent, that formula will work in all measurement systems. Formula 3b is quite useful in making decisions between two or more alternatives. Its usefulness lies in its simplicity. As noted, it begins to become more inaccurate as the walls of the pipe get thicker, and this book does not recommend its use without consideration of the necessity for accuracy in the computation.

Perhaps an understanding of the derivation and assumptions underlying the Barlow equation would be helpful in making the decision of when it is acceptable to use this formula. One who has studied the product and dimensional standards will note that it is the formula most often used in those standards.

First, look at the end view of the pipe. As is standard in these sorts of derivations, take the dimension into the screen as a unit distance so that everything in that dimension has a value of 1. As we can see from Fig. D.1, the pressure basically is constant all around the ID. This means that at any diameter the forces would be the same.

Assume we redraw the pipe as in Fig. D.2 where we replace the bottom half with two forces acting on the thickness as the forces that are holding the two halves together. The force trying to pull it apart is equal to the

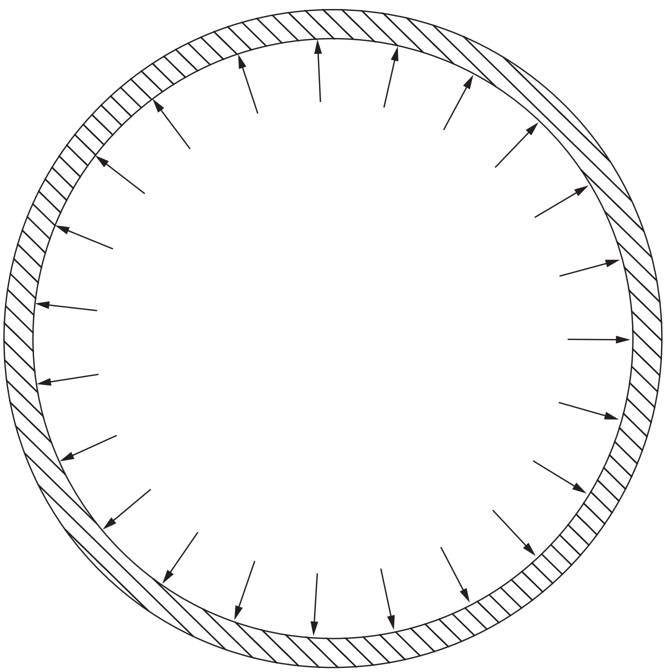

Figure D.1 Full pipe cross section showing internal pressure acting uniformly around circumference.

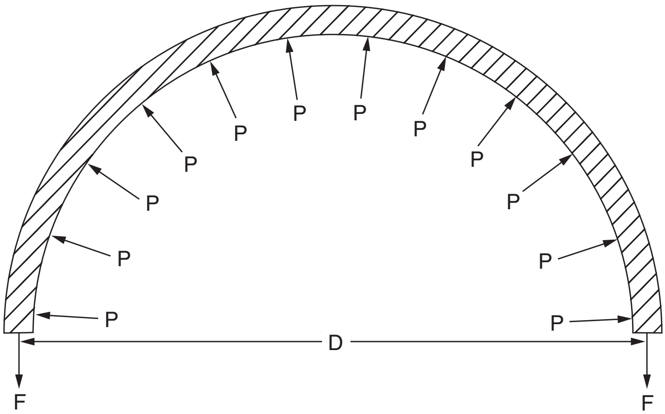

Figure D.2 Half section of pipe showing pressure on any half and the equilibrating forces acting on the walls.

pressure times the diameter. So that can be defined as P times D (times 1 for the unit depth). Since the pipe does not fly apart, the forces representing the bottom half must be equal to the forces going the other way.

Thus $PD = 2F$. Now we are most interested in stresses. Remembering that stress S is defined as F/A and that the depth is 1, we see that F works on it t times the unit depth. Thus, we can rewrite F as St. So we can now write the mathematical expression as $PD = 2St$.

Remembering that E is an efficiency of the thickness, we can just put the E on the same side of the equation as the t. That then becomes $PD = 2SEt$. What we want to do is to solve this equation for t, so it is algebraically manipulated by dividing both sides by $2SE$. The equation that we are looking for appears:

$$\frac{PD}{2SE} = t$$

We have equation 3b.

The main assumption is that t is narrow enough that the stress is essentially equal across the whole thickness. This can be, and has been, shown enough in experiments to be true.

We might ask, then, why the more complicated equation is the current published equation. One fact is that the technical community always wants to be a little more accurate. It was noticed that the Barlow equation did not exactly match the test data being obtained by several investigators. Not surprisingly, the test data diverged more as the walls got thicker. This divergence also grew as the burst pressure approached the ultimate tensile strength.

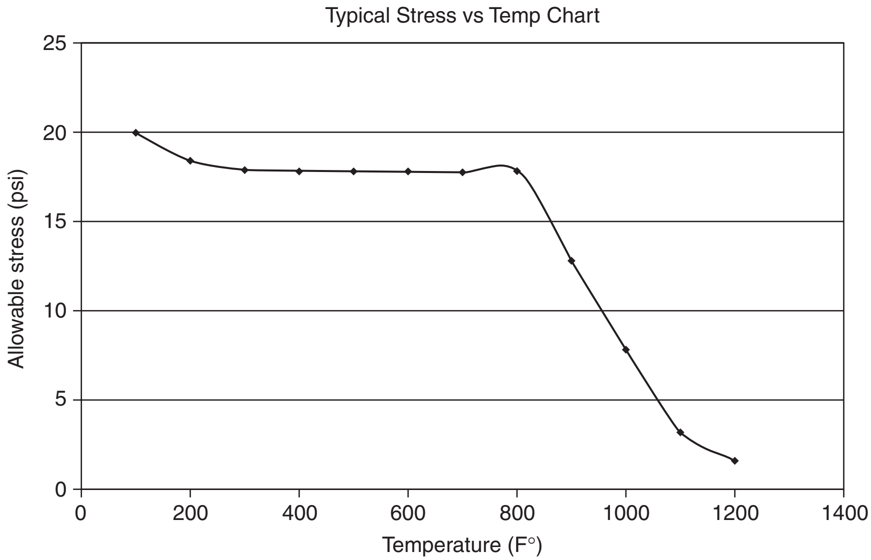

Figure D.3 Typical allowable stress versus temperature showing "Knee."

TABLE 104.1.2(A)
VALUES OF y

Temperature, °F	900 and Below	950	1000	1050	1100	1150	1200	1250 and Above
Temperature, °C	482 and Below	510	538	566	593	621	649	677 and Above
Ferritic steels	0.4	0.5	0.7	0.7	0.7	0.7	0.7	0.7
Austenitic steels	0.4	0.4	0.4	0.4	0.5	0.7	0.7	0.7
Nickel Alloys UNS Nos. N08800, N08810, N08825	0.4	0.4	0.4	0.4	04	0.4	0.5	0.7

GENERAL NOTES:

(a) The value of y may be interpolated between the 50°F (27.8°C) values shown in the Table. For cast iron and nonferrous materials, y equals 0.4.

(b) For pipe with a D_o/t_m ratio less than 6, the value of y for ferritic and austenitic steels designed for temperatures of 900°F (480°C) and below shall be taken as:

$$y = \frac{d}{d + D_o} \tag{5}$$

Figure D.4 y factor chart used in wall thickness calculations. This chart is from B31.1.

There is also a change in the criterion for setting the allowable stresses. That criterion is related to creep, which accelerates as the temperatures get higher. Temperatures and pressures were increasing in order to produce more. So there was a compounding problem.

In the early 1950s, a task force was formed to find a solution to this problem. To say they had data from many investigators is to make a slight understatement. There were at least 31 different equation forms proposed and investigated by this task force. Three men—Winston R. Burrows, R. Michel, and A. W. Rankin—had the goal of improved agreement between the calculated and realized stresses. At the same time, they wanted as much simplicity as possible. The results of their recommendations were published and reviewed and are now part of some code. They are not in the pipeline codes.

The solution was the development of the *y* factor. The *y* factor is different for ferritic and austenitic materials. The factor itself changes as the temperature goes up. This is easily explained by examining the places in the allowable stress charts for the two categories of material and where the creep rules come into play. Many refer to this point as a *knee* in the allowable stress curve. Such a knee can be observed by graphing the allowable stresses and temperature. This is done for a material in Fig. D.3.

Note that this "knee" does not occur at the same place and temperature for each category or grade of material, which is to say that the *y* factor is an improved approximation. The Code B31.1 *y* factor chart is duplicated as Fig. D.4.

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Appendix

E

Bends and Miters

Bends

For many years the code requirement for the wall thickness of bends was simply that the thickness shall be the same as that required for straight pipe of the particular code. Given the general methods of bending that were prevalent, and often are still used today, this usually meant that one needed to start with a wall thicker than needed.

Assuming that one starts with a straight piece of pipe, for the bend there will be different lengths for the different edges of the bend. These edges have names. Figure E.1 shows the net effect. One can see that the extrados will be longer and that the intrados will be shorter than the beginning length, which is the length of the centerline of the straight pipe.

Since there is no new material added by the bending process and no transfer of material from one part of the pipe to the other, the net result is that the extrados gets thinner and the intrados gets thicker. This is a fortunate circumstance as the demands of pressure in the bend were found to need the thicker material at that intrados.

A more unfortunate result is that the extrados gets thinner. This requires starting with thicker wall pipe to meet the same requirement as for straight pipe. But the question became: How much thicker? One result is that two of the codes give a recommendation to the reader of how much thicker one needs to start with, depending on the bend radius required. The shorter the radius desired, the greater the thickness.

It was found that the extrados need not have the same thickness as that of straight pipe. The pressure requirements are not as great at that position. In addition, bending techniques improved to the point where there might be less thinning.

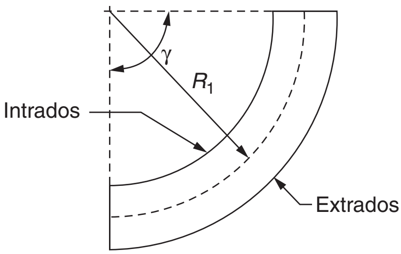

Figure E.1 Diagram showing bend terminology.

It was also found that some techniques did not thicken the intrados enough to guarantee the margins that the codes required. And a need developed to have a quantitative measure for both walls' intrados and extrados.

A mathematical technique to define those required wall thicknesses existed. Code B31.1, Code B31.3, and some of the bending standards have included it in their books. It involves the inclusion of a factor, one each for intrados and extrados, and including that factor in the straight wall thickness equation. Those factors are shown in Table E.1.

To calculate the wall thickness for either the intrados or extrados, use the appropriate factor in the following modified straight wall equation.

$$t = \frac{PD}{2(SE/I + Py)}$$

where t = calculated required wall (note: allowances must be added)
P = pressure
S = allowable stress
E = efficiency factor
I = appropriate intrados or extrados factor
y = factor from table

The formula works when used with all elements in a consistent measuring system.

The method has not been adopted by all codes. And if it had been, the form would have to be slightly different because of the different forms in the codes. However, its technical basis is established, and the qualified designer could use it as a basis for argument in applying the concept to another code.

TABLE E.1 Bend Factor

Factor	Expression	Comment
Intrados	$I = \frac{4(R_1/D) - 1}{4(R_1/D) - 2}$	D = pipe diameter R_1 = bend radius
Extrados	$I = \frac{4(R_1/D) + 1}{4(R_1/D) + 2}$	D = pipe diameter R_1 = bend radius

Miters

One technique for designing miters, where they are allowed, is as follows. Figure E.2 shows the diagram of a miter and labels the symbols. These are expressed in B31.3 and B31.9, where

R_1 = effective radius of miter
T = measured or minimum thickness of miter pipe wall
θ = angle of miter cut
α = angle of change in direction = 2θ
r_2 = mean radius of pipe using nominal wall to calculate
E = efficiency
c = corrosion and mechanical allowances
P_m = maximum internal pressure
D = pipe OD
M = minimum distance from inside crotch to end of miter

There are three equations to utilize in the design process. Equation 3 is only applicable to single miters where the angle θ is greater than 22.5°. Equation 2 is to be used for single miters where the angle θ is not greater than 22.5°. When one wants to use multiple miters, the angle θ must not be greater than 22.5° and one must use equations 1 and 2. The lesser value computed with those two equations is the maximum internal pressure allowed by the code. Then the length M must be calculated and applied to the end sections. Those equations are given in Table E.2.

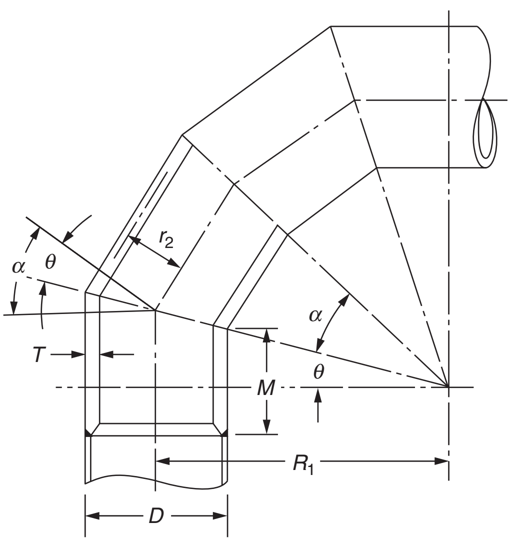

Figure E.2 Diagram showing typical miter and terminology.

TABLE E.2 Equations Utilized in the Design of Miters

Equation number	Equation
1	$P_m = \frac{SE(t-c)}{r_2}\left(\frac{R_1 - r_2}{R_1 - 0.5r_2}\right)$
2	$P_m = \frac{SE(t-c)}{r_2}\left[\frac{T-c}{(t-c)+0.643\tan\theta\sqrt{r_2(T-c)}}\right]$
3	$P_m = \frac{SE(t-c)}{r_2}\left[\frac{T-c}{(t-c)+1.25\tan\theta\sqrt{r_2(T-c)}}\right]$
4	$M = 2.5\sqrt{r_2T}$ or $M = \tan\theta\,(R_1 - r_2)$ whichever is greater

The value of R_1 should meet some minimum for these miters to be in compliance with the code. There are two formulas for that value. The more general formula is found in B31.9 and is given as

$$R_1 = \frac{1}{\tan\theta} + \frac{D}{2}$$

Code B31.3 has a more rigorous requirement, giving the minimum value of R_1 as a function of the thickness. This refinement has the effect of requiring R_1 to be larger for thicker materials. The general formula is the same but substitutes a variable expression A for the 1 in the B31.9 formula. It is

$$R_1 = \frac{A}{\tan\theta} + \frac{D}{2}$$

where A has an empirical value per Table E.3.

TABLE E.3 Empirical Value of *A*

Wall thickness $(T-c)$	A value
≥13 mm	25
$13 < (T-c) < 22$ mm	$2(T-c)$
≥22 mm	$\frac{2(T-c)}{33}+30$
≤0.5	1
$0.5 < (T-c) < 0.88$	$2(T-c)$
≥0.88	$\frac{2(T-c)}{3}+1.17$

Appendix F

Addresses of Organizations Referred to in the B31 Codes

API	American Petroleum Institute Publications and Distribution Section 1220 L Street, NW Washington, DC 20005-4070 202 682-8375 www.api.org
ASCE	The American Society of Civil Engineers 1801 Alexander Bell Drive Reston, Virginia 20191-4400 703 295-6300 or 800 548-2723 www.asce.org
ASME	ASME International Three Park Avenue New York, New York 10016-5990 212 591-8500 or 800 843-2763 www.asme.org
ASME	ASME, Order Department 22 Law Drive Box 2900 Fairfield, New Jersey 07007-2300 973 882-1170 or 800 843-2763
ASNT	American Society for Nondestructive Testing, Inc. P.O. Box 28518 1711 Arlingate Lane Columbus, Ohio 43228-0518 614 274-6003 or 800 222-2768 www.asnt.org
ASQ	American Society for Quality 611 East Wisconsin Avenue Milwaukee, Wisconsin 53202 800 248-1946 www.asq.org
ASTM	American Society for Testing and Materials 100 Barr Harbor Drive West Conshohocken, Pennsylvania 19428-2959 610 832-9500 www.astm.org
AWWA	American Water Works Association 6666 W. Quincy Avenue Denver, Colorado 80235 303 794-7711 or 800 926-7337 www.awwa.org
AWS	American Welding Society 550 NW LeJeune Road Miami, Florida 33126 305 443-9353 or 800 443-9353 www.aws.org
CDA	Copper Development Association, Inc. 260 Madison Avenue, 16th Floor New York, New York 10016 212 251-7200 or 800 232-3282 www.copper.org
CGA	Compressed Gas Association, Inc. 1725 Jefferson Davis Highway, Suite 1004 Arlington, Virginia 22202-4102 703 412-0900 www.cganet.com
CSA	CSA International 178 Rexdale Boulevard Etobicoke (Toronto), Ontario M9W 1R3, Canada 416 747-2620 or 800 463-6727 www.csa-international.org

EJMA Expansion Joint Manufacturers Association
25 North Broadway
Tarrytown, New York 10591
914 332-0040
www.ejma.org

ICBO International Conference of Building Officials
5360 Workman Mill Road
Whittier, California 90601-2298
562 692-4226 or 800 284-4406
www.icbo.org

MSS Manufacturers Standardization Society of the Valve and Fittings Industry, Inc.
127 Park Street, NE
Vienna, Virginia 22180-4602
703 281-6613
www.mss-hq.com

NACE NACE International
1440 S. Creek Drive
Houston, Texas 77084
281 228-6200
www.nace.org

NFPA National Fire Protection Association
1 Batterymarch Park
Quincy, Massachusetts 02269
617 770-3000 or 800 344-3555
www.nfpa.org

PFI Pipe Fabrication Institute
655 32nd Avenue, Suite 201
Lachine, Quebec H8T 3G6
Canada
514 634-3434
www.pfi-institute.org

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