CHAPTER

4

# Pressure Design

## 4.1 METHODS FOR INTERNAL PRESSURE DESIGN

The ASME B31.3 Code provides four basic methods for the design of components for internal pressure, as described in para. 302.2:

(a) Components in accordance with listed standards for which pressure ratings are provided in the standard, such as ASME B16.5 for flanges, are considered suitable by ASME B31.3 for the pressure rating specified in the standard. Note that the other methods of pressure design provided in ASME B31.3 can be used to rerate such listed components and/or extend their temperature range.

(b) Some listed standards, such as ASME B16.9 for pipe fittings, state that the fitting has the same pressure rating as matching seamless pipe. ASME B31.3 modifies this slightly by stating that the fittings are accepted to have the same rating as the matching seamless pipe, considering only 87.5% of the wall thickness (removing the 87.5% allowance is presently under consideration). This takes into consideration the typical mill tolerance for pipe. Note that design calculations are not usually performed for these components; design calculations are performed for the straight pipe, and matching fittings are simply selected.

(c) Design equations for some components, such as straight pipe and branch connections, are provided in para. 304 of ASME B31.3. These can be used to determine the required wall thickness with respect to internal pressure of components. Furthermore, some specific branch connection designs are assumed to be acceptable.

(d) Components that are not in accordance with a listed standard and for which design equations are not provided in the Code are treated in para. 304.7.2. This paragraph provides accepted methods, such as burst testing, for determining the pressure capacity of unlisted components.

The equations in the Code provide the minimum thickness required to limit the membrane, and in some cases bending stresses in the piping component, to the appropriate allowable stress. Mechanical and corrosion/erosion allowances must be added to this thickness. Finally, the nominal thickness selected must be such that the minimum thickness that may be provided, per specifications and considering mill tolerance, is at least equal to the required minimum thickness.

The pressure design rules in the Code are based on maximum normal stress, or maximum principal stress, versus maximum shear stress, or von Mises stress intensity. When the rules were developed in the 1940's, it was understood that stress intensity provided a better assessment of yielding, but it was felt that the maximum principal stress theory could generally provide a better measure of pressure capacity in situations where local yielding could simply lead to stress redistribution [Rossheim and Markl (1960)].

Mechanical allowances include physical reductions in wall thickness, such as due to threading and grooving the pipe. Corrosion and erosion allowances are based on the anticipated corrosion and/or erosion over the life of the pipe. This is based on estimates, experience, or literature, such as National Association of Corrosion Engineers (NACE) publications. These allowances are added to the pressure design thickness to determine the minimum required thickness of the pipe or component when it is new.

For threaded components, the nominal thread depth (dimension $h$ of ASME B1.20.1; see Appendix I or equivalent) is used for the mechanical allowance. For machined surfaces or grooves where the tolerance is not specified, the Code requires that a tolerance of 0.5 mm (0.02 in.) on the depth of the cut be assumed.

Mill tolerances are provided in specifications. The most common tolerance on the wall thickness of straight pipe is 12.5%. This means that the wall thickness at any given location around the circumference of the pipe must not be less than 87.5% of the nominal wall thickness. Note that the tolerance on pipe weight is typically tighter, so that the volume of metal and its weight may be present although a thin region would control design for hoop stress due to internal pressure.

The appropriate specification for the pipe must be referred to in order to determine the specified mill tolerance. For example, plate typically has an undertolerance of 0.25 mm (0.01 in.). However, pipe formed from plate does not have this undertolerance; it can be much greater. The pipe specification, which can permit a greater undertolerance, governs for the pipe. The manufacturer of pipe can order plate that is thinner than the nominal wall thickness for manufacturing the pipe, as long as the pipe specification mill tolerances are satisfied. However, the weight tolerance could then govern. For example, the thickness tolerance for A53 pipe is 12.5%, but the weight tolerance is 10%. As a result, the minimum thickness for A53 welded pipe made from plate material would be 10% under thickness because of the weight tolerance.

## 4.2 PRESSURE DESIGN OF STRAIGHT PIPE FOR INTERNAL PRESSURE

Equations for pressure design of straight pipe are provided in para. 304.1. The minimum thickness of the pipe selected, considering manufacturer's minus tolerance, must be at least equal to $t_m$, defined as[1]

$$t_m = t + c \tag{4.1}$$

where

$c$ = sum of the mechanical allowances plus corrosion and erosion allowances
$t$ = pressure design thickness
$t_m$ = minimum required thickness including allowances

For pipe with $t < D/6$, the basic equation for determining pressure design thickness is provided in the Code,[2]

$$t = \frac{PD}{2(SEW + PY)} \tag{4.2}$$

where

$D$ = pipe outside diameter (not nominal diameter)
$E$ = quality factor
$P$ = internal design gage pressure
$S$ = allowable stress value
$W$ = weld joint strength reduction factor per para. 302.3.5(e). (See Section 3.4)
$Y$ = coefficient provided in Table 304.1.1 of the Code and Table 4.1 here

Note that Eq. (4.2) is based on the outside diameter, rather than the inside diameter, which is used in pressure vessel codes. This is for a very good reason: The outside diameter of pipe is independent of wall thickness. That is, an NPS 6 pipe will have an outside diameter of 6.625 in., regardless of the wall

[1] ASME B31.3, Eq. (2).
[2] ASME B31.3, Eq. (3a).

**TABLE 4.1**
**VALUES OF COEFFICIENT $Y$ FOR $t < D/6$**

| Material | Temperature, °C (°F) ≤482 (900 & lower) | 510 (950) | 538 (1000) | 566 (1050) | 593 (1100) | ≥621 (1150 & up) |
|---|---|---|---|---|---|---|
| Ferritic steels | 0.4 | 0.5 | 0.7 | 0.7 | 0.7 | 0.7 |
| Austenitic steels | 0.4 | 0.4 | 0.4 | 0.4 | 0.5 | 0.7 |
| Other ductile metals | 0.4 | 0.4 | 0.4 | 0.4 | 0.4 | 0.4 |
| Cast iron | 0.0 | ... | ... | ... | ... | ... |

thickness. Therefore, the wall thickness can be directly calculated when the outside diameter is used in the equation.

Equation (4.2) is an empirical approximation of the more accurate and complex Lamé equation (ca. 1833). The hoop or circumferential stress is higher toward the inside of the pipe than toward the outside. This stress distribution is illustrated in Fig. 4.1. The Lamé equation, provided below, can be used to calculate the stress as a function of location through the wall thickness. Equation (4.2) is the Boardman equation. Although it has no theoretical basis, it provides a good match to the more accurate and complex Lamé equation for a wide range of diameter-to-thickness ratios. It becomes increasingly conservative for lower $D/t$ ratios (thicker pipe) if $Y$ is held constant.

The Lamé equation for hoop stress on the inside surface of pipe follows. Note that for internal pressure, the stress is higher on the inside than the outside. This is because strain in the longitudinal direction of the pipe must be constant through the thickness, so that any longitudinal strain caused by the compressive radial stress (due to Poisson effects and considering that the radial stress on the inside surface is equal to the surface traction of internal pressure) must be offset by a corresponding increase in hoop tensile stress to cause an offsetting Poisson effect on longitudinal strain. The Lamé equation is

$$\sigma_h = P\,\frac{0.5(D/t)^2 - (D/t) + 1}{(D/t) - 1} \tag{4.3}$$

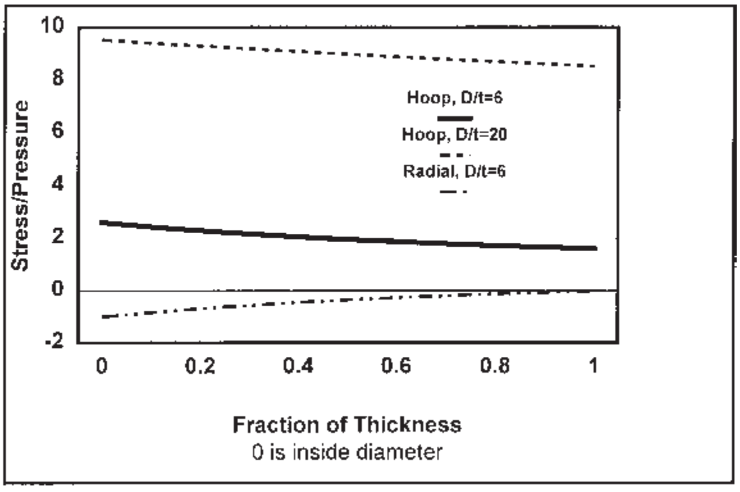


**FIG. 4.1**
**STRESS DISTRIBUTION THROUGH PIPE WALL THICKNESS DUE TO INTERNAL PRESSURE**

where

$\sigma_h$ = hoop stress

The empirical Boardman representation of this simply bases the calculation of pressure stress on some intermediate diameter, between the inside and outside diameters of the pipe, as

$$\sigma_h = P\,\frac{D-2Yt}{2t} \tag{4.4}$$

where

$$Y = 0.4 \tag{4.5}$$

Simple rearrangement of the above equation and substitution of $SE$ for $\sigma_h$ leads to Eq. (3a) of the code [Eq. (4.2) here]. Further, inside-diameter-based formulas add 0.6 times the thickness to the inside radius of the pipe rather than subtract 0.4 times the thickness from the outside radius. Thus, the inside-diameter-based formula in the pressure vessels codes and Eqs. (3a) and (3b) of ASME B31.3, the Piping Code, are consistent. The additional consideration in Eq. (3b) of ASME B31.3 is the addition of the allowances (internal corrosion increases the inside diameter in the corroded condition). With this additional consideration, Eq. (3b) of ASME B31.3 based on inside diameter provides the same required thickness as Eq. (3a) based on outside diameter.

A comparison of hoop stress calculated using the Lamé equation versus the Boardman equation (4.2) is provided in Fig. 4.2. Remarkably, the deviation of the Boardman equation from the Lamé equation is less than 1% for $D/t$ ratios greater than 5.1. Thus, the Boardman equation can be directly substituted for the more complex Lamé equation.

For thicker wall pipe, ASME B31.3 provides the following equation for the calculation of the $Y$ factor in the definition of $Y$ in para. 304.1.1. Use of this equation to calculate $Y$ results in Eq. (4.2), matching the Lamé equation for heavy wall pipe as well:

$$Y = \frac{d+2c}{D+d+2c} \tag{4.6}$$

The factor $Y$ depends on temperature. At elevated temperatures, when creep effects become significant, creep leads to a more even distribution of stress across the pipe wall thickness. Thus, the factor $Y$ increases, leading to a decrease in the calculated required wall thickness (for a constant allowable stress).

Three additional equations were formerly provided by the Code, but two were removed to be consistent with ASME B31.1 and simplify the Code. They may continue to be used. The first of the removed equations is

$$t = \frac{PD}{2SE} \tag{4.7}$$

This equation is the simple Barlow equation, which is based on the outside diameter and is always conservative. It may be used, because it is always more conservative than the Boardman equation, which is based on a smaller diameter (except when $Y = 0$). The second removed equation is

$$t = \frac{D}{2}\left(1-\sqrt{\frac{SE-P}{SE+P}}\right) \tag{4.8}$$

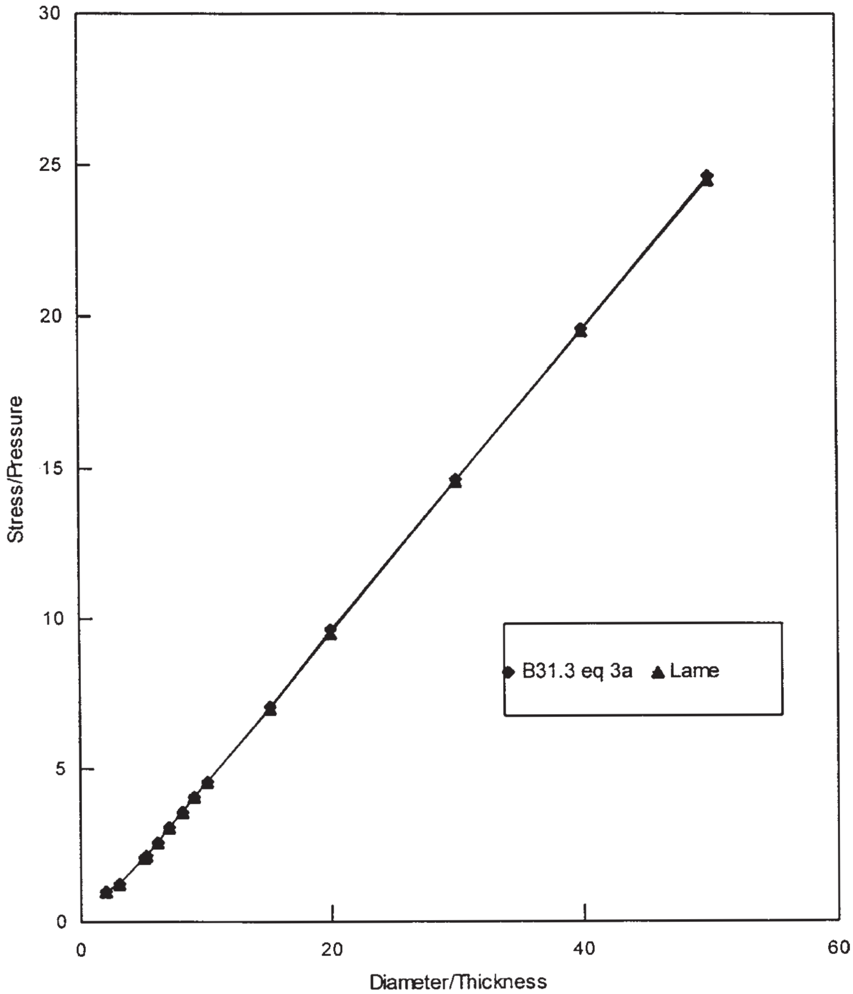


**FIG. 4.2**
**COMPARISON OF LAMÉ AND BOARDMAN EQUATIONS**

This equation is the Lamé equation rearranged to calculate thickness. Although it is not specifically included, it could be used, in accord with para. 300(c)3. However, it should not make a significant difference in the calculated wall thickness.

The following optional equation remains in ASME B31.3[3]:

$$t = \frac{P(d+2c)}{2[SE - P(1-Y)]} \tag{4.9}$$

where

$d$ = inside diameter

[3] ASME B31.3, Eq. (3b).

Equation (4.9) is the same as (4.2), but with $(d + 2c + 2t)$ substituted for $D$ and rearranged to keep thickness on the left side.

### Insert 4.1 Sample Wall Thickness Calculation

What is the required thickness of NPS 2 threaded A53 Grade B seamless pipe for the following conditions?:

- Design pressure = 150 psi
- Design temperature = 500°F
- Corrosion allowance ($CA$) = 1/16 in.
- $SE$ = 18,900 psi
- $W$ = 1.0
- $D$ = 2.375 in.
- $t = 150(2.375)/[2(18{,}900 + 0.4 \times 150)] = 0.0094$ in.
- $c = CA$ + thread depth = 0.0625 + 0.07 in.
- $t_m = 0.0094 + 0.0625 + 0.07 = 0.14$ in.

The minimum nominal pipe thickness, considering mill tolerance, is

$$\overline{T}_{\text{min.}} = \frac{t_m}{0.875} = 0.16 \text{ in.} \tag{4.10}$$

Schedule 80, XS pipe, with a nominal wall thickness of 0.218 in. is acceptable.

### Insert 4.2 Basic Stress Calculations for Cylinders Under Pressure

The average (through-thickness) circumferential and longitudinal (axial) stresses in a cylinder due to internal pressure can be calculated from equilibrium considerations. The circumferential stresses can be calculated from a longitudinal section, as shown in Fig. 4.3. The forces acting on that section must equilibrate, or, per Newton's law, the parts on either side of the section will start accelerating away from each other.

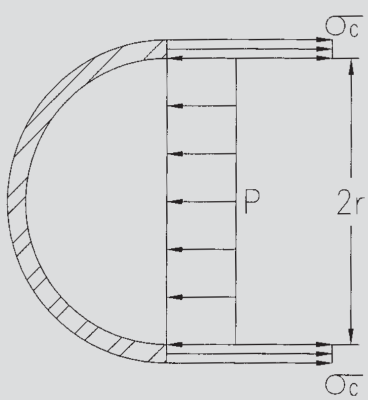


**FIG. 4.3**
**EQUILIBRIUM AT A CIRCUMFERENTIAL CUT**

The pressure force acting on the section is $2rP$, where $r$ is the inside radius and $P$ is the internal pressure. The circumferential force in the pipe wall resisting the pressure force is $2t\sigma_c$, where $t$ is the thickness (times two, because there are two sides), and $\sigma_c$ is the average circumferential stress. These must be equal, and solving for $\sigma_c$, one arrives at the following equation:

$$\sigma_c = \frac{Pr}{t} \tag{4.11}$$

The longitudinal stress can be determined by making a girth cut (as a guillotine cut) on the pipe, as shown in Fig. 4.4. The pressure force acting on the section is $\pi r^2 P$ and the longitudinal force in the pipe wall resisting this pressure force is $2\pi rt\sigma_\ell$. Note that, to be more precise, the mean radius of the pipe should be used to calculate the area of the pipe wall, but using the inside radius is generally close enough and conservative. The longitudinal stress can be calculated by equating these two forces and solving for $\sigma_\ell$:

$$\sigma_\ell = \frac{Pr}{2t} \tag{4.12}$$

Thus, the longitudinal stress in a cylinder due to internal pressure is about one-half of the circumferential stress. This is quite convenient in the design of piping, because the wall thickness is determined based on pressure design. This leaves at least one-half of the strength in the longitudinal direction available for supporting the pipe weight.

A common example of the fact that the stress in the circumferential direction is twice that in the longitudinal direction can be found when cooking a hot dog. A hot dog has a pressure-containing skin. When the internal temperature reaches the point where the fluids contained inside begin to vaporize, the hot dog skin is pressurized. When the skin is overpressurized and fails, the split is always longitudinal, transverse to the direction of highest stress, the circumferential direction. Hot dogs, at least in the experience of this author, never experience guillotine failures during cooking.

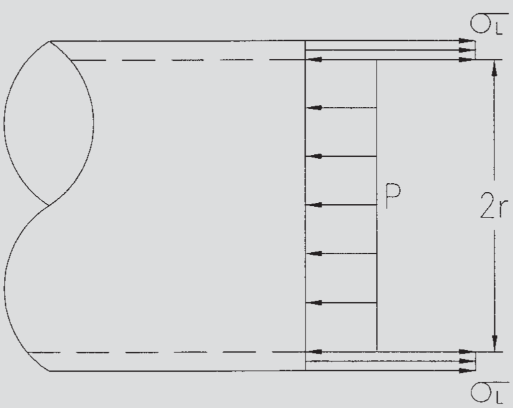


**FIG. 4.4**
**EQUILIBRIUM AT A LONGITUDINAL CUT**

## 4.3 PRESSURE DESIGN OF STRAIGHT PIPE UNDER EXTERNAL PRESSURE

For straight pipe under external pressure, there is a membrane stress check in accordance with Eq. (3a) [or (3b)] of ASME B31.3 [the equation for internal pressure; Eq. (4.2) or Eq. (4.9) here], as well as a buckling check in accordance with the external pressure design rules of ASME BPVC, Section VIII, Division 1.

Flanges, heads, and stiffeners that comply with ASME BPVC, Section VIII, Division 1, para. UG-29 are considered stiffeners. The length between stiffeners is the length between such components. The buckling pressure is a function of geometrical parameters and material properties.

Buckling pressure calculations in ASME BPVC, Section VIII, Division 1 require first the calculation of a parameter *A*, which is a function of geometry, and then a parameter *B*, which depends on *A* and a material property curve. The charts that provide the parameter *B* account for plasticity between the proportional limit of the stress–strain curve and the 0.2% offset yield stress. The chart for determination of *A* is provided in Fig. 4.5. A typical chart for *B* is provided in Fig. 4.6.

Two equations are provided for calculating the maximum permissible external pressure. The first uses the parameter *B*,

$$p = \frac{4B}{3D/t} \tag{4.13}$$

where

$B$ = parameter from material curves in ASME BPVC, Section II, Part D, Subpart 3
$D$ = inside diameter (note that this differs from the ASME B31.3 definition of *D*, which is outside diameter; this is the Pressure Vessel Code definition of *D*)
$p$ = allowable external pressure
$t$ = pressure design thickness

The second equation is for elastic buckling, and must be used when the value of *A* falls to the left of the material property curves that provide *B*. This equation is

$$p = \frac{2AE}{3D/t} \tag{4.14}$$

where

$A$ = parameter from geometry curves in ASME BPVC Section II, Part D, Subpart 3, Fig. G (see Fig. 4.5 here)

The second equation is based on elastic buckling, so the elastic modulus is used. A chart of *B* could be used, with the linear elastic portion of the curve extended to lower values of *B*, but this would unnecessarily enlarge the charts. The charts provided in ASME B31.5 have this form, with the elastic lines extended.

The procedures of ASME BPVC Section VIII include consideration of the allowable out-of-roundness in pressure vessels, and use a design margin of three. Although pipe is not generally required to comply with the same out-of-roundness tolerance as is required for pressure vessels, this has historically been ignored, and has not led to any apparent problems.

The basis for the approach of ASME BPVC Section VIII is provided in Bergman (1960), Holt (1960), Saunders and Windenburg (1960), Windenburg and Trilling (1960), and Windenburg (1960).

A new buckling evaluation procedure, provided in Code Case 2286, is more relevant to piping, because it permits consideration of combined loads, including external pressure, axial load, and gross bending

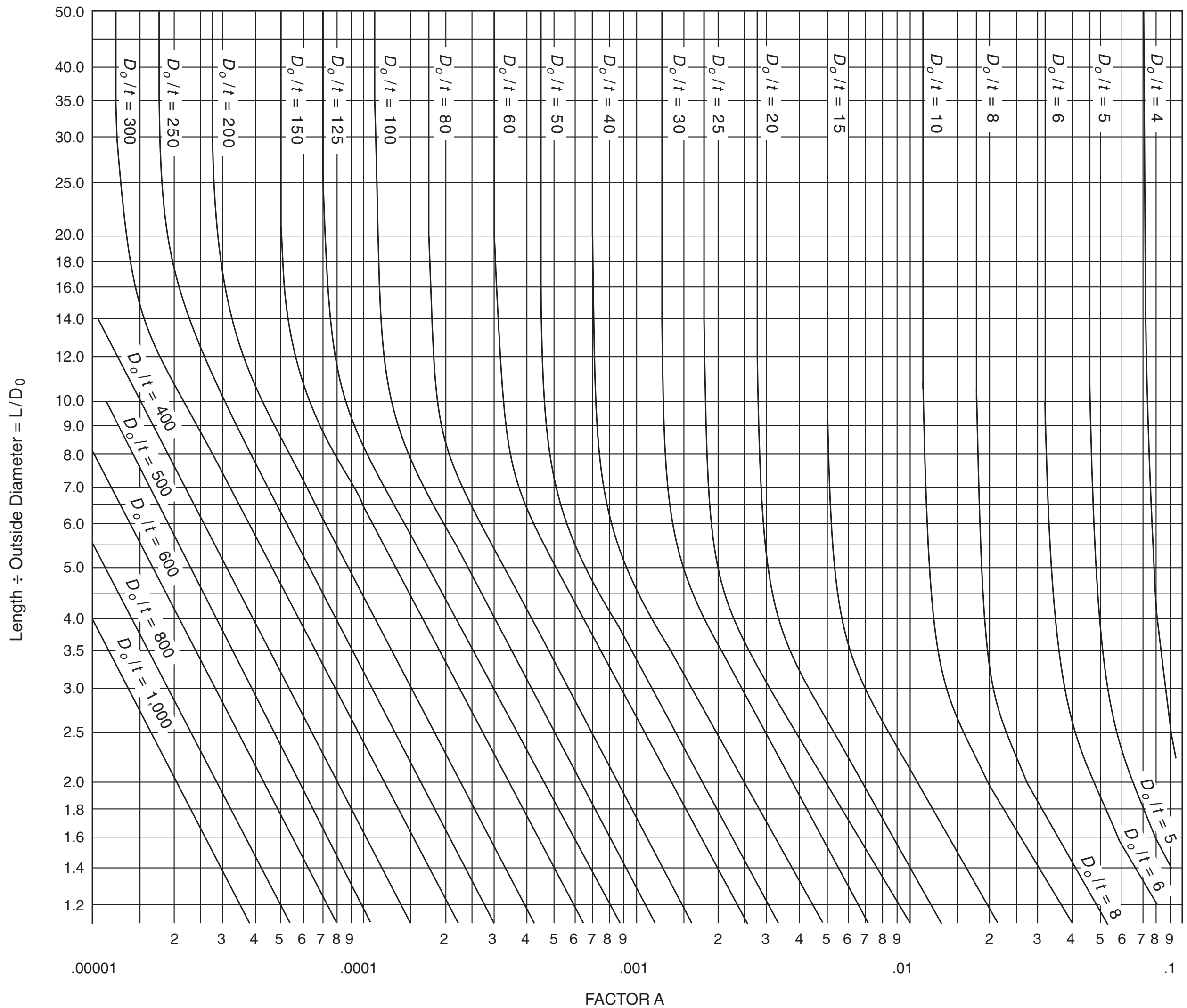


**FIG. 4.5**
**CHART FOR DETERMINING *A* (ASME BPVC, SECTION II, PART D, SUBPART 3, FIG. G).**
**TABLE G CITED IN THE FIGURE IS GIVEN IN ASME BPVC, SECTION II**

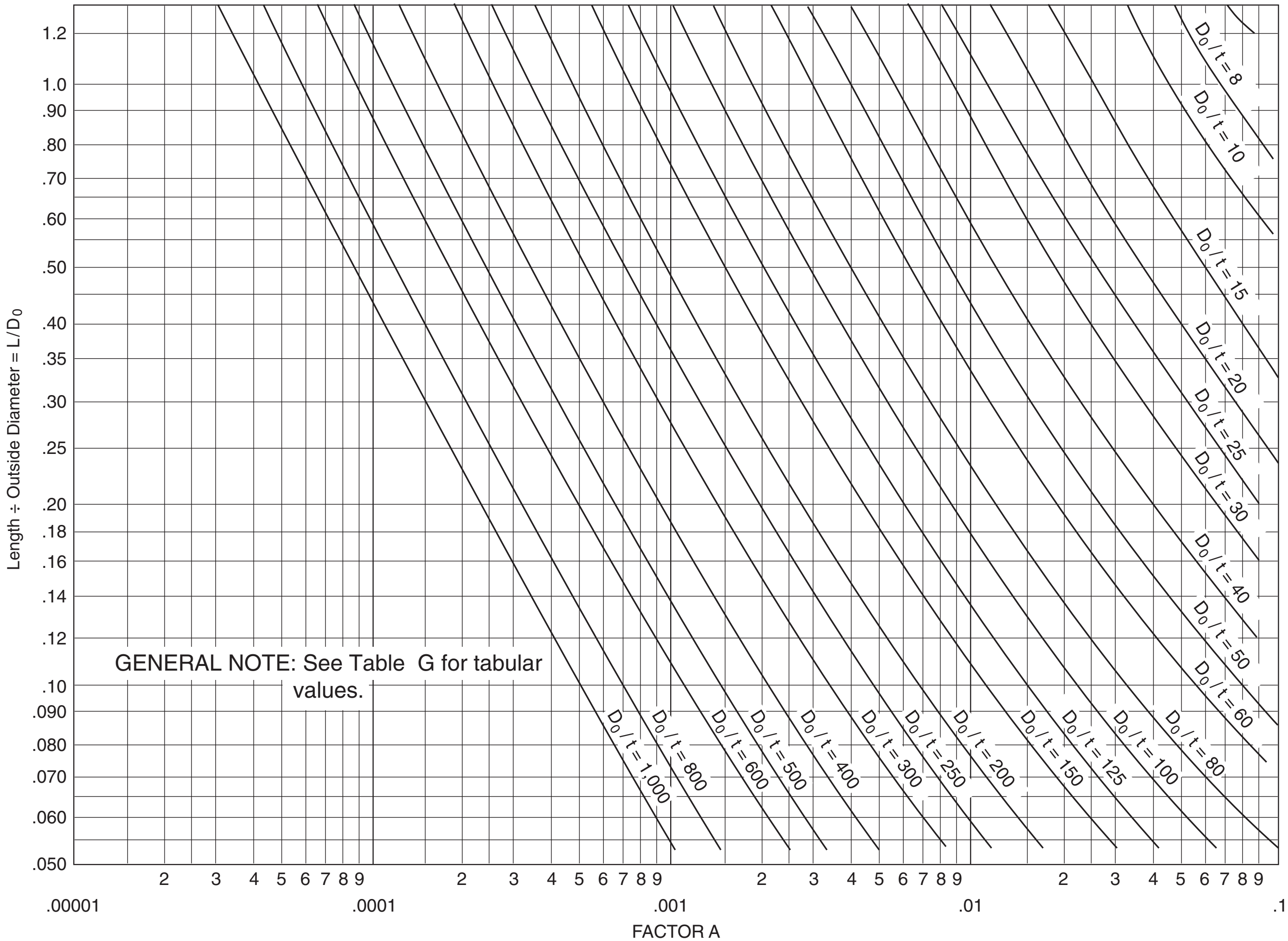


**FIG. 4.5**
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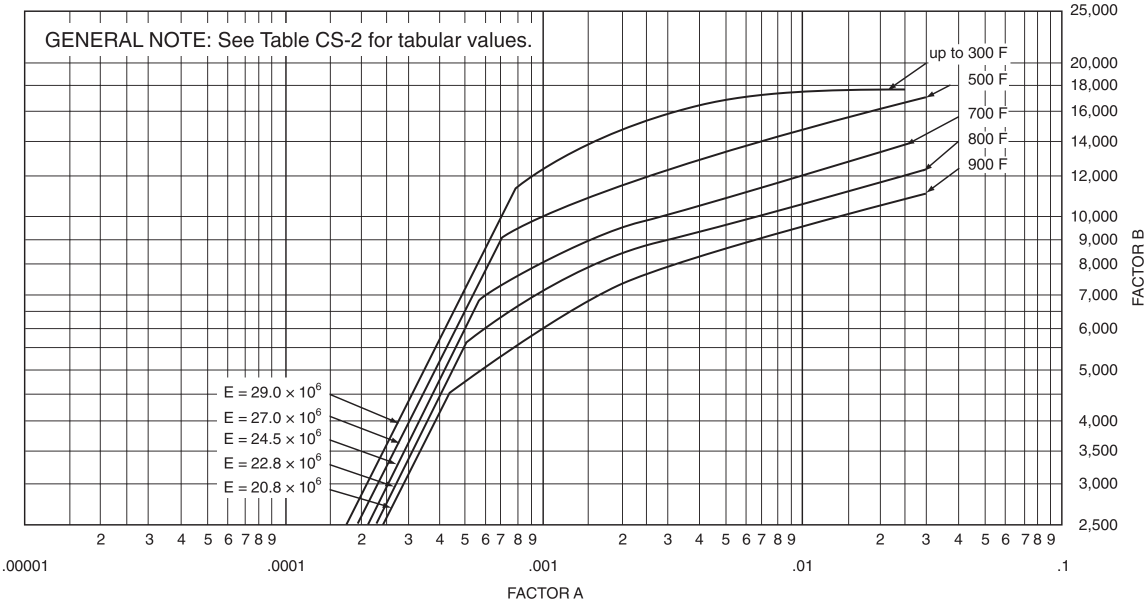



**FIG. 4.6**
**TYPICAL CHART FOR DETERMINING *B* (ASME BPVC, SECTION II, PART D, SUBPART 3, FIG. CS-2). TABLE CS-2 CITED IN THE FIGURE IS GIVEN IN ASME BPVC, SECTION II**

moment. Code Case 2286 also provides a more uniform margin of safety and a higher allowable external pressure. It is not at present explicitly recognized in ASME B31.3, but could be considered, given para. 300(c)3.

## 4.4 PRESSURE DESIGN OF WELDED BRANCH CONNECTIONS

The pressure design of branch connections is based on a rather simple approach, although the resulting design calculations are the most complex of the design-by-formula approaches for pressure design provided in the Code. A branch connection cuts a hole in the run pipe. The metal removed is no longer available to carry the forces due to internal pressure. An area replacement concept is used for those branch connections that do not either comply with listed standards or with certain designs (see Section 4.7). The area of metal removed by cutting the hole, to the extent that it was required for internal pressure, must be replaced by extra metal in a region around the branch connection. This region is within the limits of reinforcement, defined later.

The simplified design approach is limited with respect to the geometries to which it is considered applicable. These limitations are as follows:

- The run pipe diameter-to-thickness ratio is less than 100.
- The branch-to-run diameter ratio is not greater than one.
- The angle $\beta$ (angle between branch and run pipe axes) is at least 45°.
- The axis of the branch intersects the axis of the run pipe.

Where the above limitations are not satisfied, the designer is referred to para. 304.7.2 (see Section 4.15). Alternatives in that paragraph include proof testing and finite-element analysis. Paragraph 304.3.5(e) suggests consideration of integral reinforcement or complete encirclement reinforcement for such branch connections.

The area $A_1$ of metal removed is defined as

$$A_1 = t_h\, d_1(2 - \sin\beta) \tag{4.15}$$

where

$d_1$ = effective length removed from the run pipe at the branch connection
$t_h$ = pressure design thickness of the header
$\beta$ = the smaller angle between axes of branch and run

For a 90 deg. branch connection, $d_1$ is effectively, the largest possible inside diameter of the branch pipe; the inside diameter of the pipe if fully corroded and with the full mill tolerance removed from the inside of the pipe.

Figure 4.7 illustrates the nomenclature and process.

The angle $\beta$ is used in the evaluation because a lateral connection, a branch connection with a $\beta$ other than 90°, creates a larger hole in the run pipe. This larger hole must be considered in $d_1$. For a lateral, $d_1$ is the branch pipe inside diameter, considering mill tolerance and corrosion/erosion allowance, divided by $\sin\beta$. The $(2 - \sin\beta)$ term is used to provide additional reinforcement that is considered to be appropriate because of the geometry of the branch connection.

The pressure design thickness is the pressure design thickness of the run pipe, with one exception. If the run pipe is welded and the branch does not intersect the weld, the weld quality factor E and strength reduction factor W should not be used in calculating the wall thickness. The weld quality factor and strength reduction factor only reduce the allowable stress at the location of the weld.

Only the pressure design thickness is used in calculating the required area since only the pressure design thickness is required to resist internal pressure. Corrosion allowance and mill tolerance at the hole are obviously of no consequence.

The area removed, $A_1$, must be replaced by available area around the opening. This area is available from excess wall thickness that may be available in the branch and run pipes as well as added reinforcement, and the fillet welds that attach the added reinforcement. This metal must be relatively close to the opening of the run pipe to reinforce it. Thus, the metal must be within a certain limited area in order to be considered appropriate reinforcement of the opening.

The limit of reinforcement along the run pipe, taken as a dimension from the centerline of the branch pipe where it intersects the run pipe wall, is $d_2$, defined by

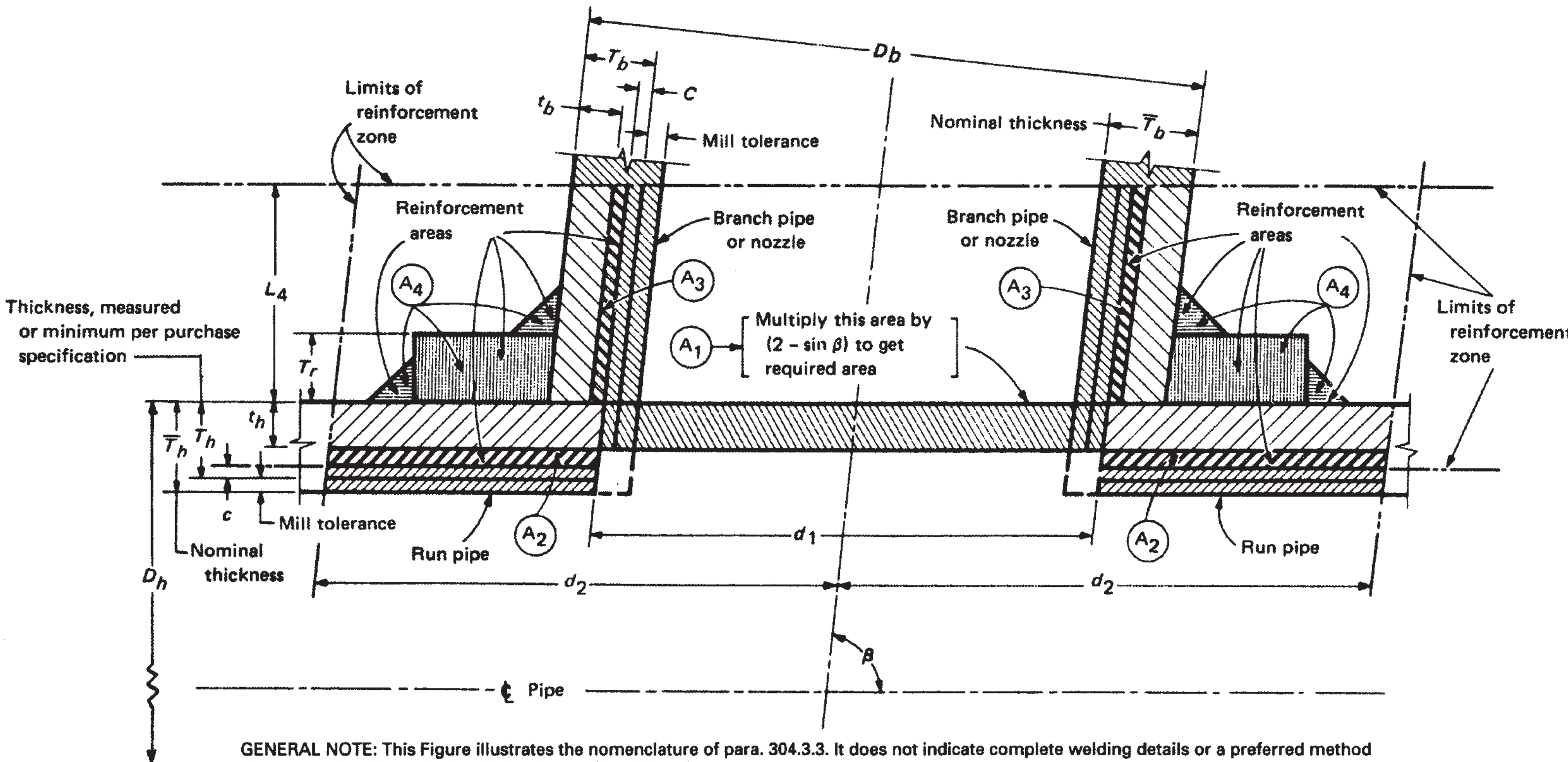
$$\begin{aligned} d_2 &= \text{greater of } [d_1, (T_b - c) + (T_h - c) + d_1/2] \\ &\text{but} \quad d_2 \le D_h \end{aligned} \tag{4.16}$$

where

$D_h$ = outside diameter of header or run pipe
$T_b$ = minimum thickness of branch pipe
$T_h$ = minimum thickness of run pipe
$c$ = allowance (mechanical, corrosion, erosion)

The limit of reinforcement along the branch pipe measured from the outside surface of the run pipe is $L_4$, defined as the lesser of $2.5\,(T_h - c)$ and $2.5\,(T_b - c) + T_r$, where $T_r$ is the minimum thickness of reinforcement.

The reinforcement within this zone is required to exceed $A_1$. This reinforcement consists of excess thickness available in the run pipe, $A_2$, excess thickness available in the branch pipe, $A_3$, and additional




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.

**FIG. 4.7**
**ILLUSTRATION OF FABRICATED BRANCH CONNECTION SHOWING NOMENCLATURE (ASME B31.3, FIG. 304.3.3)**

reinforcement, $A_4$. These can be calculated as follows:

$$A_2 = (2d_2 - d_1)(T_h - t_h - c) \tag{4.17}$$

$$A_3 = 2L_4(T_b - t_b - c)/\sin\beta \tag{4.18}$$

The area $A_4$ is the area of properly attached reinforcement and the welds that are within the limits of reinforcement. The Code specifies minimum weld sizes in para. 328.5.4. The designer is directed to assume that the minimum dimensions specified by the Code are provided, unless the welder is specifically directed to make larger welds. The ASME B31.3 Code does not require the designer to specify branch connection weld size, because generally acceptable minimum sizes are specified by the Code. Additionally, the ASME B31.3 Code differs from the Pressure Vessel Code in that strength calculations for load paths through the weld joints are not required.

## 4.5 PRESSURE DESIGN OF EXTRUDED OUTLET HEADER

An extruded outlet header is a branch connection formed by extrusion, using a die or dies to control the radii of the extrusion. Paragraph 304.3.4 provides area replacement rules for such connections; they are applicable for 90-deg. branch connections where the branch pipe centerline intercepts the run pipe centerline, and where there is no additional reinforcement. Figure 4.8 (ASME B31.3, Fig. 304.3.4) shows the geometry of an extruded outlet header. Extruded outlet headers are subject to minimum external contour radius requirements, depending on the diameter of the branch connection.

A similar area replacement calculation is used as described in Section 4.4 for fabricated branch connections, except that the required replacement area is reduced for smaller branch-to-run pipe diameter ratios. The replacement area is from additional metal in the branch pipe, additional metal in the run pipe, and additional metal in the extruded outlet lip.

## 4.6 ADDITIONAL CONSIDERATIONS FOR BRANCH CONNECTIONS UNDER EXTERNAL PRESSURE

Branch connections under external pressure are covered in para. 304.3.6. The same rules described in Sections 4.4 and 4.5 are used. However, only one-half of the area described in Section 4.4, covering welded branch connections, requires replacement. In other words, only one-half of the area $A_1$ requires replacement. Also, the thicknesses used in the calculation are the required thicknesses for the external pressure condition.

## 4.7 BRANCH CONNECTIONS THAT ARE PRESUMED TO BE ACCEPTABLE

Some specific types of branch connections are presumed to be acceptable. This, of course, includes listed fittings (e.g., ASME B16.9 tees and MSS SP-97 branch connection fittings). It also includes the following:

- For branch connections DN 50 (NPS 2) or less that do not exceed one-fourth of the nominal size of the run pipe, threaded or socket welding couplings or half-couplings (Class 2000 or greater) are presumed to provide sufficient reinforcement as long as the minimum thickness of the coupling is at least as thick as the branch pipe.
- Branch connection fittings qualified per para. 304.7.2 are acceptable.

## 4.8 PRESSURE DESIGN OF BENDS AND ELBOWS

Bends were required to have, after bending, a wall thickness at least equal to the minimum required wall thickness for straight pipe in para. 304.2.1. However, this was changed in the 2000 Addendum. The Lorenz equation (ca. 1910) was included; it provides a means of calculating the required wall thickness. Note that the prior requirement that simply stated the thickness should be the same as required for straight pipe was deleted. The new requirement is more conservative for the intrados (inside curves) of bends and less conservative for the extrados (outside curve).

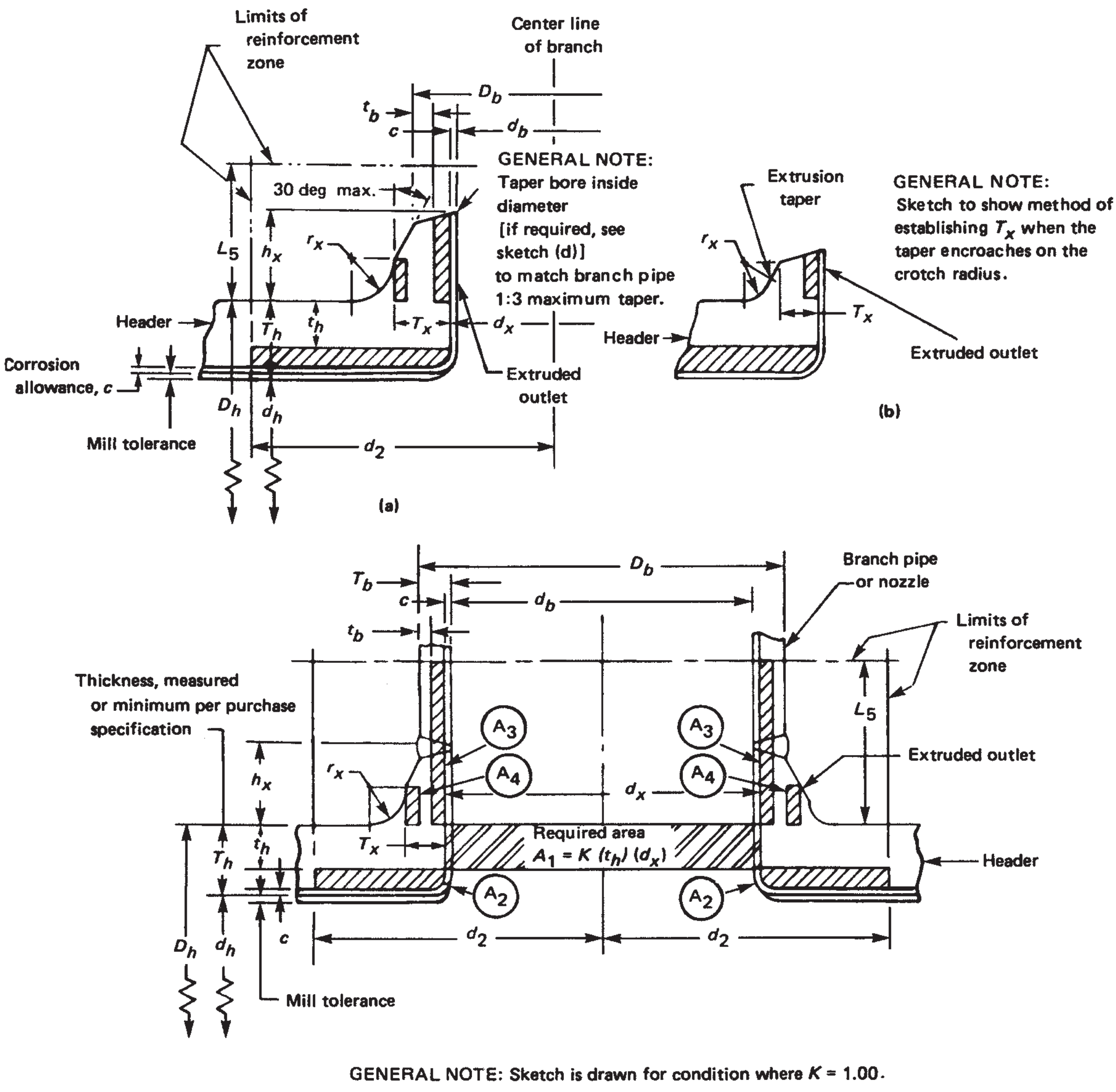



**FIG. 4.8**
**ILLUSTRATION OF EXTRUDED OUTLET FITTING SHOWING NOMENCLATURE (ASME B31.3, FIG. 304.3.4). THIS FIGURE ILLUSTRATES THE NOMENCLATURE OF PARA. 304.3.4. IT DOES NOT INDICATE COMPLETE DETAILS OR A PREFERRED METHOD OF CONSTRUCTION**

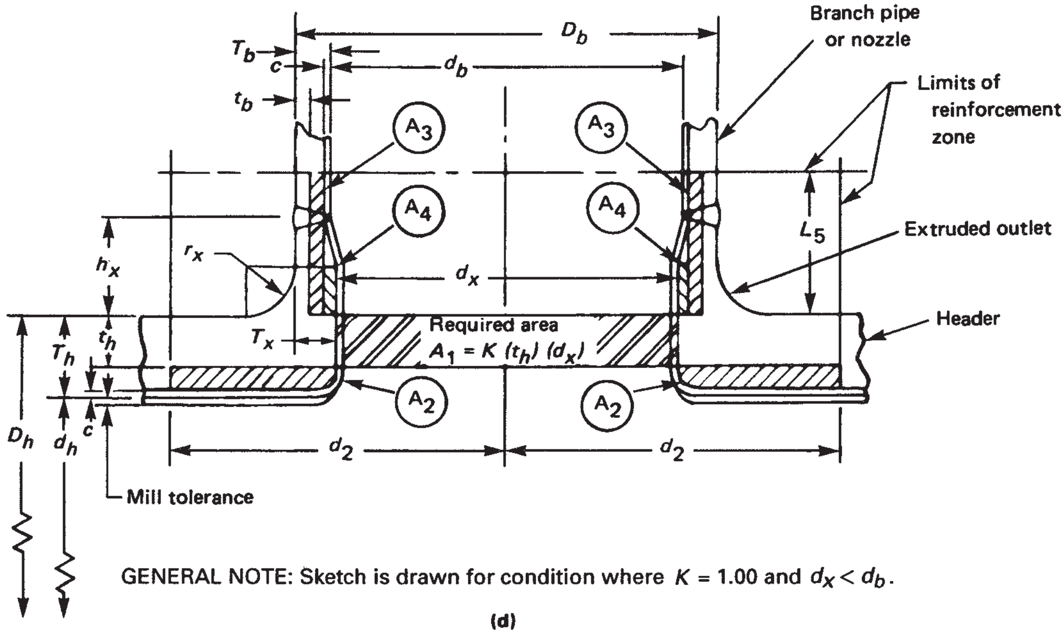



**FIG. 4.8**
**CONTINUED**

The Lorenz equation is basically the equation for a toroid. If the intrados and extrados had the same wall thickness, the inside would be subjected to higher hoop stress than straight pipe and the outside would be subjected to lower hoop stress than straight pipe. A simple way to envision this is that the inside has less metal over the curve and the outside has more metal over the curve. The Lorenz equation for an elbow or bend is given by[4]

$$t = \frac{PD}{2(SEW/I + PY)} \tag{4.19}$$

where the terms are as defined in Section 4.2 for Eq. (4.2), except for $I$, which is a stress index that accounts for the difference in hoop stress due to internal pressure in bends versus straight pipe.

On the inside curve of the bend, the intrados, we have[5]

$$I = \frac{4R_1/D - 1}{4R_1/D - 2} \tag{4.20}$$

where

$R_1$ = radius of bend

[4] ASME B31.3, Eq. (3c).
[5] ASME B31.3, Eq. (3d).

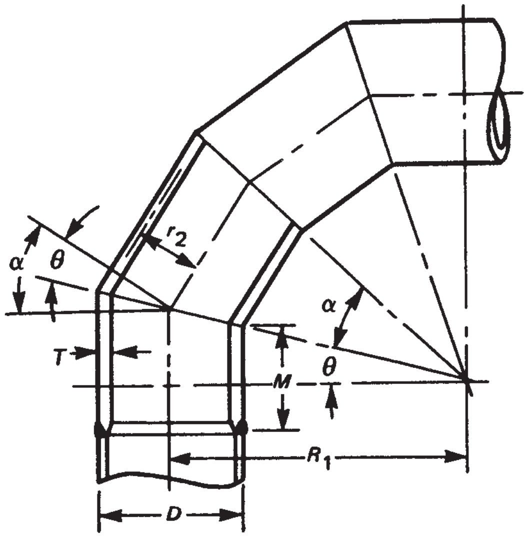



**FIG. 4.9**
**ILLUSTRATION OF MITER BEND SHOWING NOMENCLATURE (ASME B31.3, FIG. 304.2.3)**

On the outside of the bend, or the extrados, we have[6]

$$I = \frac{4R_1/D + 1}{4R_1/D + 2} \tag{4.21}$$

On the side of the elbow, or the crown, $I = 1.0$ (i.e., the hoop stress is the same as in straight pipe).

The thickness variation from the intrados to the extrados is required to be gradual, and the requirements are stated to apply at the midspan of the bend. The thickness at the ends is required to satisfy the required thickness for straight pipe per para. 304.1.

The normal process of making a bend by bending straight pipe produces this type of thickness variation. Part of the reason for providing these new rules is due to the practice of fabricating elbows by forming two "clamshells" out of plate and welding them together. This produces a bend of uniform thickness, and the thickness on the intrados would be too thin if it simply satisfied the required thickness for straight pipe.

Elbows in accordance with listed standards, or qualified by para. 304.7.2, are also permitted.

## 4.9 PRESSURE DESIGN OF MITER BENDS

Miter changes in direction with an angular offset of 3 deg. or less (angle $\alpha$ in Fig. 4.9) do not require design consideration as a miter bend. The required wall thickness for this condition is the same as straight pipe. Design equations for multiple and single miter bends follow.

The design equation for determining the required wall thickness for internal pressure for multiple miter bends is the lesser of the following two equations, 4.22 and 4.23. Note that these equations are only valid

[6] ASME B31.3, Eq. (3e).

when the angle $\theta$ does not exceed 22.5 deg. Equation 4c (4.24 herein) applies for angles $\theta$ greater than 22.5 deg.[7, 8]

$$P_m = \frac{SEW(T-c)}{r_2}\left(\frac{T-c}{(T-c)+0.643\tan\theta\sqrt{r_2(T-c)}}\right) \tag{4.22}$$

$$P_m = \frac{SEW(T-c)}{r_2}\left(\frac{R_1-r_2}{R_1-0.5r_2}\right) \tag{4.23}$$

Single miters with an angle $\theta$ not greater than 22.5 deg. are calculated per equation 4.22. Otherwise, they are calculated per equation 4.24.[9]

$$P_m = \frac{SEW(T-c)}{r_2}\left(\frac{T-c}{(T-c)+1.25\tan\theta\sqrt{r_2(T-c)}}\right) \tag{4.24}$$

The miter wall thickness, $T$, used in Eqs. (4.22)–(4.24) is required to extend a distance at least $M$ from the inside crotch of the miter end welds.

In the above equations, the following definitions apply:

$E$ = quality factor
$M$ = greater of $2.5(r_2T)^{0.5}$ and $(\tan\theta)(R_1 - r_2)$
$P_m$ = maximum allowable internal pressure for miter bends
$R_1$ = effective radius of miter bend, defined as the shortest distance from the pipe centerline to the intersection of the planes of adjacent miter joints
$S$ = allowable stress from Table A-1 in Appendix A of ASME B31.3
$T$ = miter pipe wall thickness (measured or minimum per purchase specification)
$W$ = weld joint strength reduction factor per para. 302.3.5(e)
$c$ = sum of mechanical allowances plus corrosion and erosion allowances
$r_2$ = mean radius of pipe using nominal wall
$\theta$ = angle of miter cut
$\alpha$ = angle of change in direction at miter joint
= $2\theta$

## 4.10 PRESSURE DESIGN OF CLOSURES

Closures are covered in para. 304.4.1. Listed components, such as ASME B16.9 pipe caps, can be used for closures. The other two options provided in ASME B31.3 are to design the closure in accordance with ASME BPVC Section VIII, Division 1, or to qualify it as an unlisted component in accordance with para. 304.7.2 (see Section 4.15). Specific references to ASME BPVC, Section VIII, Division 1 paragraphs are provided for ellipsoidal, torispherical, hemispherical, conical, toriconical, and flat heads.

Openings in closures are covered in para. 304.4.2. These requirements are summarized below:

- If the opening is greater than one-half of the inside diameter of the closure (as defined in ASME BPVC Section VIII, Division 1, para. UG-36), it should be designed as a reducer

[7] ASME B31.3, Eq. (4a).
[8] ASME B31.3, Eq. (4b).
[9] ASME B31.3, Eq. (4c).

per para. 304.6 if the closure is dished and as a flange per para. 304.5 if the closure is flat.

- Small openings and connections using branch connection fittings that comply with para. 304.3.2(b) or 304.3.2(c) are considered to be inherently adequately reinforced.
- The required area of reinforcement is determined per the relevant ASME BPVC Section VIII, Division 1 requirements [UG-37(b), UG-38, or UG-39], which depend on the type of closure. For example, only one-half of the area requires replacement for a flat head.
- The available area of reinforcement is calculated per the rules in ASME B31.3, specifically para. 304.3.3 or para. 304.3.4. Note that ASME B31.3 requires that boundaries for a curved closure follow the contour of the closure (versus a chord dimension).
- Rules for multiple openings follow para. 304.3.3 and para. 304.3.4 rules for multiple openings.

## 4.11 PRESSURE DESIGN OF FLANGES

Most flanges are in accordance with listed standards, such as ASME B16.5, and for larger flanges, ASME B16.47. Appendix L is provided in ASME B31.3 to cover aluminum Flanges because they are not included in ASME B16.5. When a custom flange is required, design by analysis is permitted by para. 304.5.1. ASME B31.3 refers to the rules for flange design contained in ASME BPVC Section VIII, Division 1, Appendix 2, but using the allowable stresses and temperature limits of ASME B31.3. For flanges with metal-to-metal contact outside of the bolt circle, the rules of ASME BPVC Section VIII, Division 1, Appendix Y are referenced.

## 4.12 PRESSURE DESIGN OF BLIND FLANGES

Most blind flanges are in accordance with listed standards, such as ASME B16.5. When designing a blind flange, the rules of ASME BPVC Section VIII, Division 1, para. UG-34 apply. The procedure in ASME BPVC Section VIII includes consideration of the moment due to the flange bolts, so the required bolt load for boltup and operation must be determined per the flange design rules of Appendix 2 of ASME BPVC Section VIII, Division 1.

## 4.13 PRESSURE DESIGN OF BLANKS

Blanks are flat plates that get sandwiched between flanges to block flow. A design equation for permanent blanks is provided in para. 304.5.3, as[10]

$$t_m = d_g \sqrt{\frac{3P}{16SEW}} + c \tag{4.25}$$

where

$d_g$ = inside diameter of gasket for raised or flat face flanges, or the gasket pitch diameter for ring joint and fully retained gasketed flanges

Other terms are as defined in Section 4.2.

ASME B16.48, Steel Line Blanks, was added as a listed standard in the 2004 edition. Blanks used for test purposes are not subject to these design rules and are often designed to higher allowable stress levels (e.g., 90% of the specified minimum yield strength).

[10] ASME B31.3, Eq. (15).

## 4.14 PRESSURE DESIGN OF REDUCERS

Most reducers in piping systems are in accordance with listed standards. However, pressure design per para. 304.7.2 and detailed design using rules for conical or toriconical closures are permitted. The rules for closures reference ASME BPVC Section VIII, Division 1, as described above in Section 4.10.

## 4.15 PRESSURE DESIGN OF UNLISTED COMPONENTS

If a component is not in accordance with a listed standard and/or the design rules provided in para. 304 are not applicable, para. 304.7.2 is applicable. This paragraph requires that some calculations be done in accordance with the design criteria provided by the Code and be substantiated by one of several methods. The meat of this paragraph is considered to be the substantiation; the aforementioned calculations are not generally given much consideration. The methods to substantiate the calculations, and thereby the design, include the following:

- Extensive, successful service experience under comparable conditions with similarly proportioned components of the same or like material.
- Experimental stress analysis, such as described in ASME BPVC Section VIII, Division 2, Annex 5.F.
- Proof test in accordance with either ASME B16.9, MSS SP-97, or ASME BPVC Section VIII, Division 1, para. UG-101. Note that of these standards, those of B16.9 and MSS SP-97 are more applicable to piping components and have a margin of safety consistent with other components in the Piping Code (factor of three on burst rather than the factor of four in the Pressure Vessel Code).
- Detailed stress analysis (e.g., finite-element method) with results evaluated as described in ASME BPVC Section VIII, Division 2, Part 5. These are the design-by-analysis rules in the Pressure Vessel Code. Note that the allowable stress from ASME B31.3 is used in the assessment.

Of the above, the methods normally used to qualify new unlisted components are proof testing and detailed stress analysis.

It should be noted that the Code permits interpolation between sizes, wall thicknesses, and pressure classes, and also permits analogies among related materials. Extrapolation is not permitted.

The issue of how to determine that the above has been done in a satisfactory manner has been discussed in detail in B31.3 Section Committee meetings. Earlier editions of the Code required only that proof testing be approved by the Inspector. However, this created concerns that it may be interpreted that the Inspector must witness the proof test, which is not practical when the manufacturer performs proof tests to qualify a line of piping components. Obviously, all the potential future owner's Inspectors could not be gathered for this event. Furthermore, the other methods are of at least equal concern, and their review may be more appropriately done by an engineer rather than an Inspector. As a result of these concerns, the requirement was added that documentation showing compliance with the above must be available for the owner's approval. The review would be by an Inspector or some other qualified individual for the owner.

Although MSS SP-97 and ASME B16.9 provide a clear approach for determining that the rating of a component is equivalent to or better than matching straight pipe, they do not provide clear procedures for determining a rating for a component that may have a unique rating which may differ from matching straight pipe. The procedure generally used here is to establish a pressure-temperature rating by multiplying the proof pressure by the ratio of the allowable stress for the test specimen to the actual tensile strength of the test specimen. An example of this approach is provided by Biersteker et al (1991). In the proposed B31H Standard, this would be reduced by a testing factor depending on the number of tests.

A new standard is under development by ASME that will eventually add to or replace the existing proof test alternatives in para. 304.7.2. This is B31H, *Standard Method to Establish Maximum Allowable Design Pressure for Piping Components*. This Standard provides procedures to determine that a component has a pressure capacity at least as great as a matching straight pipe, or to determine a pressure–temperature rating for a component.