\[ h = \text{hub length} \]
\[ H_D = \text{hydrostatic end force on area inside of flange} \]
\[ h_D = \text{radial distance from the bolt circle, to the circle on which } H_D \text{ acts, as prescribed in Table 2-6} \]
\[ H_G = \text{gasket load (difference between flange design bolt load and total hydrostatic end force)} \]
\[ h_G = \text{radial distance from gasket load reaction to the bolt circle} \]
\[ h_o = \text{factor} \]
\[ H_p = \text{total joint-contact surface compression load} \]
\[ H_T = \text{difference between total hydrostatic end force and the hydrostatic end force on area inside of flange} \]
\[ h_T = \text{radial distance from the bolt circle to the circle on which } H_T \text{ acts as prescribed in Table 2-6} \]
\[ K = \text{ratio of outside diameter of flange to inside diameter of flange} \]
\[ L = \text{factor} \]
\[ m = \text{gasket factor, obtain from Table 2-5.1 [see Note in 2-5(c)(1)]} \]
\[ M_D = \text{component of moment due to } H_D \]
\[ M_G = \text{component of moment due to } H_G \]
\[ M_o = \text{total moment acting upon the flange, for the operating conditions or gasket seating as may apply (see 2-6)} \]
\[ M_T = \text{component of moment due to } H_T \]
\[ N = \text{width used to determine the basic gasket seating with } b_o, \text{ based upon the possible contact width of the gasket (see Table 2-5.2)} \]
\[ P = \text{internal design pressure (see UG-21). For flanges subject to external design pressure, see 2-11.} \]
\[ R = \frac{C - B}{2} - g_o \]
\[ R = \text{radial distance from bolt circle to point of intersection of hub and back of flange. For integral and hub flanges,} \]
\[ S_a = \text{allowable bolt stress at atmospheric temperature (see UG-23)} \]
\[ S_b = \text{allowable bolt stress at design temperature (see UG-23)} \]
\[ S_T = \text{allowable design stress for material of flange at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (see UG-23)} \]
\[ S_H = \text{calculated longitudinal stress in hub} \]
\[ S_n = \text{allowable design stress for material of nozzle neck, vessel or pipe wall, at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (see UG-23)} \]
\[ S_R = \text{calculated radial stress in flange} \]
\[ S_T = \text{calculated tangential stress in flange} \]
\[ T = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ t = \text{flange thickness} \]
\[ t_n = \text{nominal thickness of shell or nozzle wall to which flange or lap is attached} \]
\[ t_s = \text{two times the thickness } g_o, \text{ when the design is calculated as an integral flange or two times the thickness of shell nozzle wall required for internal pressure, when the design is calculated as a loose flange, but not less than } \frac{1}{4} \text{ in. (6 mm)} \]
\[ U = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ V = \text{factor for integral type flanges (from Figure 2-7.3)} \]
\[ W_L = \text{factor for loose type flanges (from Figure 2-7.5)} \]
\[ W = \text{flange design bolt load, for the operating conditions or gasket seating, as may apply (see 2-5(e))} \]
\[ w = \text{width used to determine the basic gasket seating width } b_o, \text{ based upon the contact width between the flange facing and the gasket (see Table 2-5.2)} \]
\[ W_{m1} = \text{minimum required bolt load for the operating conditions (see 2-5(c)). For flange pairs used to contain a tubesheet for a floating head or a U-tube type of heat exchangers, or for any other similar design, } W_{m1} \text{ shall be the larger of the values as individually calculated for each flange, and that value shall be used for both flanges.} \]
\[ W_{m2} = \text{minimum required bolt load for gasket seating (see 2-5(c)). For flange pairs used to contain a tubesheet for a floating head or U-tube type of heat exchanger, or for any other similar design where the flanges or gaskets are not the same, } W_{m2} \text{ shall be the larger of the values calculated for each flange and that value shall be used for both flanges.} \]
\[ Y = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ y = \text{gasket or joint-contact-surface unit seating load, [see Note 1, 2-5(c)]} \]
\[ Z = \text{factor involving } K \text{ (from Figure 2-7.1)} \]

### 2-4 CIRCULAR FLANGE TYPES

(a) For purposes of computation, there are three types:

1. **Loose Type Flanges.** This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall, and designs where the method of attachment is not considered to give the
vessel, or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed $0.8 S_n$. The shearing stress shall be calculated on the basis of $W_{m1}$ or $W_{m2}$ as defined in 2-3, whichever is greater. Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

2-9 SPLIT LOOSE FLANGES

Loose flanges split across a diameter and designed under the rules given in this Appendix may be used under the following provisions.

(a) When the flange consists of a single split flange or flange ring, it shall be designed as if it were a solid flange (without splits), using 200% of the total moment $M_o$ as defined in 2-6.

(b) When the flange consists of two split rings each ring shall be designed as if it were a solid flange (without splits), using 75% of the total moment $M_o$ as defined in 2-6. The pair of rings shall be assembled so that the splits in one ring shall be 90 deg from the splits in the other ring.

(c) The splits should preferably be midway between bolt holes.

2-10 NONCIRCULAR SHAPED FLANGES WITH CIRCULAR BORE

The outside diameter $A$ for a noncircular flange with a circular bore shall be taken as the diameter of the largest circle, concentric with the bore, inscribed entirely within the outside edges of the flange. Bolt loads and moments, as well as stresses, are then calculated as for circular flanges, using a bolt circle drawn through the centers of the outermost bolt holes.

2-11 FLANGES SUBJECT TO EXTERNAL PRESSURES

(a) The design of flanges for external pressure only [see UG-99(f)] shall be based on the equations given in 2-7 for internal pressure except that for operating conditions:

$$
M_o = h_p(h_D - h_G) + h_s(h_T - h_G)
$$

For gasket seating,

$$
M_o = Wh_G
$$
2-7  CALCUATION OF FLANGE STRESSES

The stresses in the flange shall be determined for both the operating conditions and gasket seating condition, whichever controls, in accordance with the following equations:

(a) for integral type flanges [Figure 2-4, sketches (5), (6), (6a), (6b), and (7)], for optional type flanges calculated as integral type [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)], and for loose type flanges with a hub which is considered [Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c)];

- Longitudinal hub stress
  \[ S_H = \frac{fM_0}{L_0 Z B} \]  

- Radial flange stress
  \[ S_R = \frac{(1.333\pi + 1)M_0}{L_1^2 B} \]  

- Tangential flange stress
  \[ S_T = \frac{Y M_0}{Z^2 B} - Z S_R \]  

(b) for loose type flanges without hubs and loose type flanges with hubs which the designer chooses to calculate without considering the hub [Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c)] and optional type flanges calculated as loose type [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]:

- Longitudinal hub stress \( S_H \) not greater than the smaller of \( 1.5S_f \) or \( 1.5S_n \) for optional type flanges designed as integral [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)], also integral type [Figure 2-4, sketch (7)] where the neck material constitutes the hub of the flange;

- Longitudinal hub stress \( S_H \) not greater than the smaller of \( 1.5S_f \) or \( 2.5S_n \) for integral type flanges with hub welded to the neck, pipe or vessel wall [Figure 2-4, sketches (6), (6a), and (6b)].

(b) For hub flanges attached as shown in Figure 2-4, sketches (2), (2a), (3), (3a), (4), (4a), (4b), and (4c), the nozzle neck, vessel or pipe wall shall not be considered to have any value as a hub.

(c) In the case of loose type flanges with laps, as shown in Figure 2-4, sketches (1) and (1a), where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed 0.8 \( S_n \) for the material of the lap, as defined in 2-3. In the case of welded flanges, shown in Figure 2-4, sketches (3), (3a), (4), (4a), (4b), (4c), (7), (8), (8a), (9), (9a), (10), and (10a) where the nozzle neck, vessel, or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed 0.8 \( S_n \). The shearing stress shall be calculated on the basis of \( W_m \) as defined in 2-3, whichever is greater. Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

2-8  ALLOWABLE FLANGE DESIGN STRESSES

(a) The flange stresses calculated by the equations in 2-7 shall not exceed the following values:

- Longitudinal hub stress \( S_H \) not greater than \( S_f \) for cast iron and, except as otherwise limited by (a) and (b) below, not greater than 1.5\( S_f \) for materials other than cast iron;

\( S_H \) not greater than the smaller of \( 1.5S_f \) or \( 1.5S_n \) for optional type flanges designed as integral [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)], also integral type [Figure 2-4, sketch (7)] where the neck material constitutes the hub of the flange;

(b) Longitudinal hub stress \( S_H \) not greater than the smaller of \( 1.5S_f \) or \( 2.5S_n \) for integral type flanges with hub welded to the neck, pipe or vessel wall [Figure 2-4, sketches (6), (6a), and (6b)].

(b) For hub flanges attached as shown in Figure 2-4, sketches (2), (2a), (3), (3a), (4), (4a), (4b), and (4c), the nozzle neck, vessel or pipe wall shall not be considered to have any value as a hub.

(c) In the case of loose type flanges with laps, as shown in Figure 2-4, sketches (1) and (1a), where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed 0.8 \( S_n \) for the material of the lap, as defined in 2-3. In the case of welded flanges, shown in Figure 2-4, sketches (3), (3a), (4), (4a), (4b), (4c), (7), (8), (8a), (9), (9a), (10), and (10a) where the nozzle neck, vessel, or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed 0.8 \( S_n \). The shearing stress shall be calculated on the basis of \( W_m \) as defined in 2-3, whichever is greater. Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

2-9  SPLIT LOOSE FLANGES

Loose flanges split across a diameter and designed under the rules given in this Appendix may be used under the following provisions.

(a) When the flange consists of a single split flange or flange ring, it shall be designed as if it were a solid flange (without splits), using 200% of the total moment \( M_0 \) as defined in 12-4.

(b) When the flange consists of two split rings each ring shall be designed as if it were a solid flange (without splits), using 75% of the total moment \( M_0 \) as defined in 12-4. The pair of rings shall be assembled so that the splits in one ring shall be 90 deg from the splits in the other ring.

(c) The splits should preferably be midway between bolt holes.

(d) It is not a requirement that the flange rigidity rules of 2-14 be applied to split loose flanges.
\[ H_D = \text{hydrostatic end force on area inside of flange} \]
\[ h_D = \text{radial distance from the bolt circle, to the circle on which } H_D \text{ acts, as prescribed in Table 2-6} \]
\[ H_G = \text{gasket load for the operating condition} \]
\[ h_G = \text{radial distance from gasket load reaction to the bolt circle} \]
\[ h_o = \text{factor} \]
\[ = \sqrt{B_d} \]
\[ H_p = \text{total joint-contact surface compression load} \]
\[ = 2b \times 3.14 GmP \]
\[ H_T = \text{difference between total hydrostatic end force and the hydrostatic end force on area inside of flange} \]
\[ = H - H_D \]
\[ h_T = \text{radial distance from the bolt circle to the circle on which } H_T \text{ acts as prescribed in Table 2-6} \]
\[ K = \text{ratio of outside diameter of flange to inside diameter of flange} \]
\[ = A/B \]
\[ L = \text{factor} \]
\[ = \frac{t_e + 1}{T} + \frac{t_e^3}{d} \]
\[ m = \text{gasket factor, obtain from Table 2-5.1 [see Note in 2-5(c)(1)]} \]
\[ M_D = \text{component of moment due to } H_D \]
\[ = H_o h_o \]
\[ M_G = \text{component of moment due to gasket} \]
\[ = H_o h_G \]
\[ M_o = \text{total moment acting upon the flange, for the operating conditions or gasket seating as may apply (see 2-6)} \]
\[ M_T = \text{component of moment due to } H_T \]
\[ = H_T h_T \]
\[ N = \text{width used to determine the basic gasket seating with } b_o, \text{ based upon the possible contact width of the gasket (see Table 2-5.2)} \]
\[ P = \text{internal design pressure (see UG-21). For flanges subject to external design pressure, see 2-11.} \]
\[ R = \text{radial distance from bolt circle to point of intersection of hub and back of flange. For integral and hub flanges,} \]
\[ = C - B - s_1 \]
\[ S_a = \text{allowable design stress for material of nozzle neck, vessel or pipe wall, at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (see UG-23)} \]
\[ S_b = \text{calculated radial stress in flange} \]
\[ S_H = \text{calculated tangential stress in flange} \]
\[ S_{a_0} = \text{allowable bolt stress at atmospheric temperature (see UG-23)} \]
\[ S_{b_0} = \text{allowable bolt stress at design temperature (see UG-23)} \]
\[ S_T = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ t = \text{flange thickness} \]
\[ t_n = \text{nominal thickness of shell or nozzle wall to which flange or lap is attached} \]
\[ t_x = \text{two times the thickness } g_0, \text{ when the design is calculated as an integral flange or two times the thickness of shell nozzle wall required for internal pressure, when the design is calculated as a loose flange, but not less than } \frac{1}{4} \text{ in. (6 mm)} \]
\[ U = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ V = \text{factor for integral type flanges (from Figure 2-7.3)} \]
\[ V_L = \text{factor for loose type flanges (from Figure 2-7.5)} \]
\[ W = \text{flange design bolt load, for the operating conditions or gasket seating, as may apply [see 2-5(e)]} \]
\[ w = \text{width used to determine the basic gasket seating width } b_o, \text{ based upon the contact width between the flange facing and the gasket (see Table 2-5.2)} \]
\[ W_{m1} = \text{minimum required bolt load for the operating conditions [see 2-5(c)]. For flange pairs used to contain a tubesheet for a floating head or a U-tube type of heat exchangers, or for any other similar design, } W_{m1} \text{ shall be the larger of the values as individually calculated for each flange, and that value shall be used for both flanges.} \]
\[ W_{m2} = \text{minimum required bolt load for gasket seating [see 2-5(c)]}. \text{For flange pairs used to contain a tubesheet for a floating head or U-tube type of heat exchanger, or for any other similar design where the flanges or gaskets are not the same, } W_{m2} \text{ shall be the larger of the values calculated for each flange and that value shall be used for both flanges.} \]
\[ Y = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ y = \text{gasket or joint-contact-surface unit seating load, [see Note 1, 2-5(c)]} \]
\[ Z = \text{factor involving } K \text{ (from Figure 2-7.1)} \]

\section*{2-4 CIRCULAR FLANGE TYPES}

For purposes of computation, there are three types: (a) Loose Type Flanges. This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall, and designs where the method of attachment is not considered to give the mechanical strength equivalent of integral attachment. See Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c) for typical loose type flanges and the location of the loads and moments. Welds and other details of
When internal pressure occurs only during the required pressure test, the design may be based on external pressure, and auxiliary devices such as clamps may be used during the application of the required test pressure.

The Code as currently written provides minimum requirements for construction and it is recognized to be the responsibility of the designing engineer to determine when the intended service is of a nature that requires supplementary requirements to ensure safety; consequently, the designer should determine when the service warrants that this class of inspection be specified for steel castings of less than 4 in. (100 mm) nominal body thickness.

The coefficients of these formulas include a factor that effectively increases the allowable stress for such construction to 1.55.

The complexity of the work includes factors such as design simplicity versus complexity, the types of materials and welding procedures used, the thickness of materials, the types of nondestructive examinations applied, and whether heat treatments are applied.

The size and complexity of the organization includes factors such as the number of employees, the experience level of employees, the number of Code items produced, and whether the factors defining the complexity of the work cover a wide or narrow range.

Knowing the official rating capacity of a safety valve which is stamped on the valve, it is possible to determine the overall value of $KA$ in either of the following formulas in cases where the value of these individual terms is not known:

\[
KA = \frac{W_{s}}{51.5P} \\
KA = \frac{W_{g} \sqrt{F}}{CP \sqrt{M}}
\]

This value for $KA$ is then substituted in the above formulas to determine the capacity of the safety valve in terms of the new gas or vapor.

Use $E = 1.0$ for Category C and D joints that are not butt welded since stresses in these joints are controlled by the applicable rules for sizing such joints. See Figures UG-34 and UW-13.2

$I = \frac{bt^{3}}{12}$ where $b = 1.0$ for vessels without reinforcements and for vessels with stay plates or stay rods. $I = \frac{pt^{3}}{12}$ for vessels with reinforcements that do not extend around the corners of the vessel [see Figure 13-2(a), sketches (5) and (6)].

For unreinforced vessels of rectangular cross section (13-7 and parts of 13-18), the given moments are defined on a per-unit-width basis. That is, moments have dimensions $[\text{Length} \times \text{Force/Length}] = [\text{Force}]$.


Air or gas is hazardous when used as a testing medium. It is therefore recommended the vessel be tested in such a manner as to ensure personnel safety from a release of the total internal energy of the vessel. See also ASME PCC-2, Article 501, Mandatory Appendix 501-II, "Stressed Energy Calculations for Pneumatic Pressure Test," and Mandatory Appendix 501-III, "Safe Distance Calculations for Pneumatic Pressure Test."

When using high alloys and nonferrous materials either for solid wall or clad or lined vessels, refer to UHA-6, UCL-3, and UNF-4, as appropriate.

See "Stresses in Large Cylindrical Pressure Vessels on Two Saddle Supports," p. 959, Pressure Vessels and Piping: Design and Analysis, A Decade of Progress, Volume Two, published by ASME.

See Transactions ASCE, Volume 98 — 1931 “Design of Large Pipe Lines.”

This construction has the further advantage of not transmitting discharge-pipe strains to the valve. In these types of installation, the back pressure effect will be negligible, and no undue influence upon normal valve operation can result.

A Nonmandatory Appendix Y flange bolted to a rigid foundation may be analyzed as a Class 1 assembly by substituting $2l$ for $l$ in eq. Y-6.1(12) of Y-6.1.