FLANGE DESIGN – Appendix 2 of ASME Section VIII, Division 1

The rules in Mandatory Appendix 2 of ASME Section VIII, Division 1 apply specifically to the design of bolted flange connections with gaskets that are entirely within the circle enclosed by the bolt holes and with no contact outside this circle. These rules provide only for hydrostatic end loads and gasket seating. The design involves the selection of the gasket (material, type and dimensions), flange facing, bolting, hub proportions, flange width and flange thickness. Flange dimensions shall be such that the stresses in the flange do not exceed the allowable flange stresses. Flanges shall also mee the rigidity requirements. All calculations shall be made on dimensions in the corroded condition.

It is recommended that bolted flange connections conforming to the standards listed below be used for connections to external piping. These standards may also be used for other bolted flange connections in the pressure vessel. Except for the flanges that conform to one of the standards listed in the Table below, all other flanges shall satisfy the requirements in Appendix 2.

Table 1: Flange and Pipe Fitting Standards Acceptable for use in ASME Section VIII, Div. 1 Construction

ASME B16.1	Gray Iron Pipe Flanges and Flanged Fittings, Classes 25, 125 and 250
ASME B16.5	Pipe Flanges and Flanged Fittings, NPS 1/2 through NPS 24
ASME B16.9	Factory-made Wrought Buttwelding Fittings
ASME B16.11	Forged Fittings, Socket-Welding and Threaded
ASME B16.20	Metallic Gaskets for Pipe Flanges
ASME B16.24	Cast Copper Alloy Pipe Flanges, Flanged Fittings and Valves, Classes 150, 300, 600, 900, 1500 and 2500
ASME B16.42	Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300
ASME B16.47	Large Diameter Steel Flanges, NPS 26 through NPS 60

A forged nozzle may use the ASME B16.5/ B16.47 pressure-temperature ratings for the flange material being used, provided all of the following are met:

a) For ASME B16.5 applications, the forged nozzle flange shall meet all dimensional requirements of a flanged fitting given in ASME B16.5 with the exception of the inside diameter. The inside diameter of the forged nozzle shall not exceed the inside diameter of the same size lap joint flange given in ASME B16.5. For ASME B16.47 applications, the inside diameter shall not exceed the weld hub diameter A given in the ASME B16.47 tables.

b) For ASME B16.5 applications, the outside diameter of the forged nozzle neck shall be at least equal to the hub diameter of the same size and class ASME B16.5 lap joint flange. For ASME B16.47 applications, the outside diameter of the hub shall at least equal the X diameter given in ASME B16.47 tables. Large hub diameters shall be limited to nut stop diameter dimensions.

MATERIAL REQUIREMENTS

- Flanges made from ferritic steel shall be full-annealed, normalized, normalized and tempered, or quenched and tempered when thickness of the flange, t, exceeds 3 in.
- Materials on which welding is to be performed shall be of good weldable quality. Satisfactory qualification of the welding procedure under Section IX is considered as proof.

- Welding shall not be performed on steel that has a carbon content greater than 0.35%. All welding on flange connections shall comply with the requirements for postweld heat treatment given in ASME Section VIII, Division 1.
- Flanges with hubs that are machined from plate or bar material shall not be machined unless the material has been formed into a ring and the following additional conditions are met:
    - In a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange. However, the original plate surfaces need not be present in the finished flange.
    - The joints in the ring are welded butt joints. Thickness to be used to determine postweld heat treatment and radiography requirements shall be $\min\left[t, \frac{(A-B)}{2}\right]$

      In the expression above,
      t = Flange thickness
      A = Outside diameter of flange
      B = Inside diameter of flange
    - The back of flange and the outer surface of the hub are examined by either magnetic particle method or liquid penetrant method.
    - It is recommended that bolts and studs have a nominal diameter of less than ½ in. If bolts or studs smaller than ½ in. are used, ferrous bolting material shall be of alloy steel.

CIRCULAR FLANGE TYPES

There are three types of flanges:

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 mechanical strength equivalent of integral attachment.

2) INTEGRAL TYPE FLANGES

   This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel or pipe wall, butt welded thereto or attached by other forms of welding of such a nature that the flange and nozzle neck, vessel or pipe wall is considered to be the equivalent of an integral structure.

3) OPTIONAL TYPE FLANGES

   This type covers designs where the attachment of the flange to the nozzle neck, vessel, or pipe wall is such that the assembly is considered to act as a unit, which shall be calculated as an integral flange, except that for simplicity the designer may calculate the construction as a loose-type flange, provided none of the following values is exceeded:

   $g_o = 5/8$ in. (g_o is thickness of hub at small end)
   $B/g_o = 300$ (B is inside diameter of flange)
   $P = 300$ psi (P is internal design pressure)
   Operating temperature = 700°F

BOLT LOADS

GENERAL REQUIREMENTS

Calculations shall be made for each of the two design conditions of operating and gasket seating, and the more sever shall control.

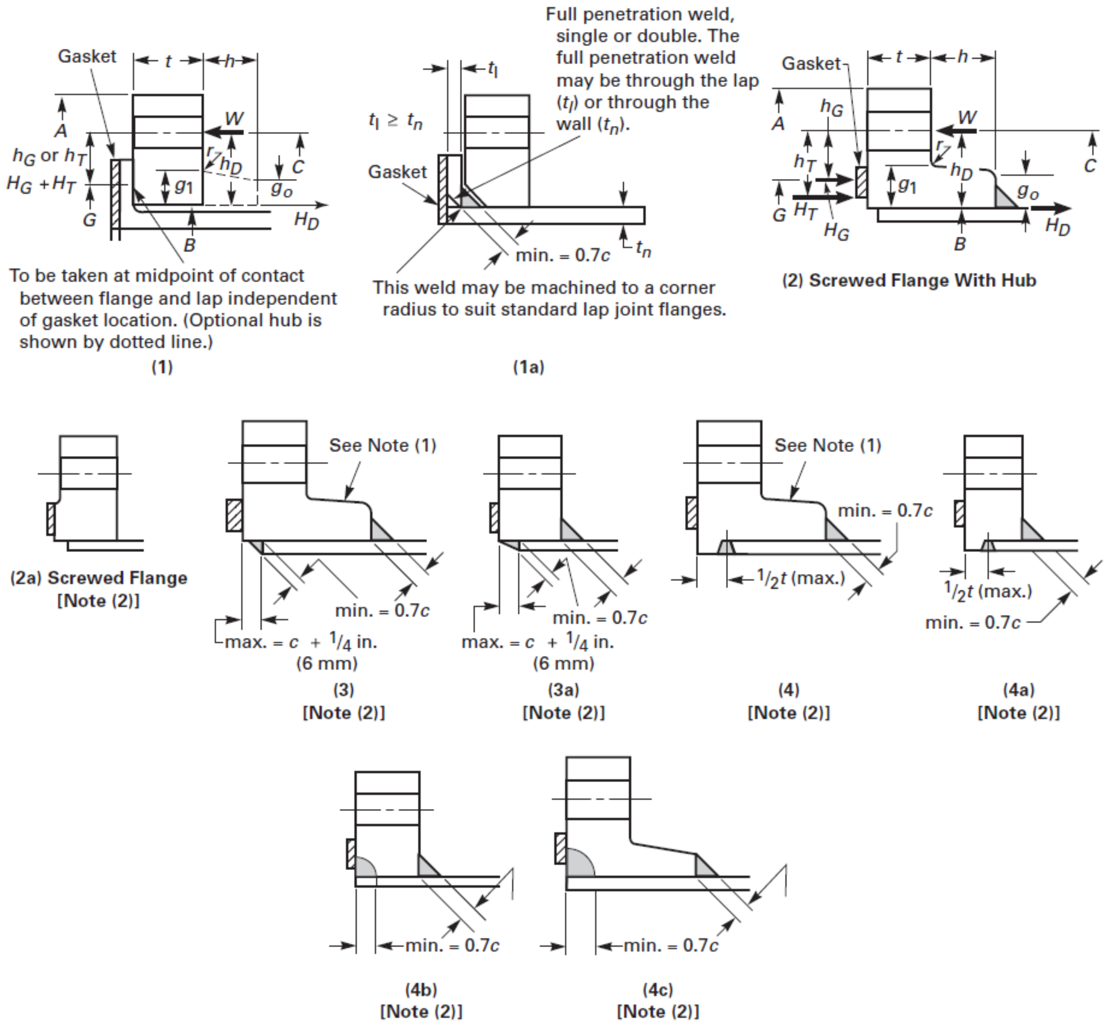


Figure 1: Loose Type Flanges
(*From ASME Section VIII, Division 1, Appendix 2)*

In the design of flange pairs used to contain a tubesheet of a heat exchanger, loads must be determined for the most severe condition of operating and /or gasket seating loads applied to each side at the same time. Then calculations shall be made for each flange.

DESIGN CONDITIONS

Operating condition refers to the conditions required to resist the hydrostatic end force of the design pressure tending to part the joint, and to maintain on the gasket sufficient compression to assure a tight joint, all at the design temperature. The minimum load is a function of the design pressure, the gasket material and the effective gasket area to be kept tight under pressure and determines one of the two requirements for the amount of bolting A_{m1}. This load is also used for the design of the flange.

Gasket seating conditions exist when the gasket is seated by applying an initial load with the bolts when assembling the joint at atmospheric pressure and temperature. The minimum initial load considered to be adequate for proper seating is a function of the gasket material and the effective gasket area to be seated and determines the other of the two requirements for the amount of bolting A_{m2}. For the design of flange, this load is modified to take account of the operating conditions when these govern the amount of bolting required A_m as well as the amount of bolting actually provided A_b.

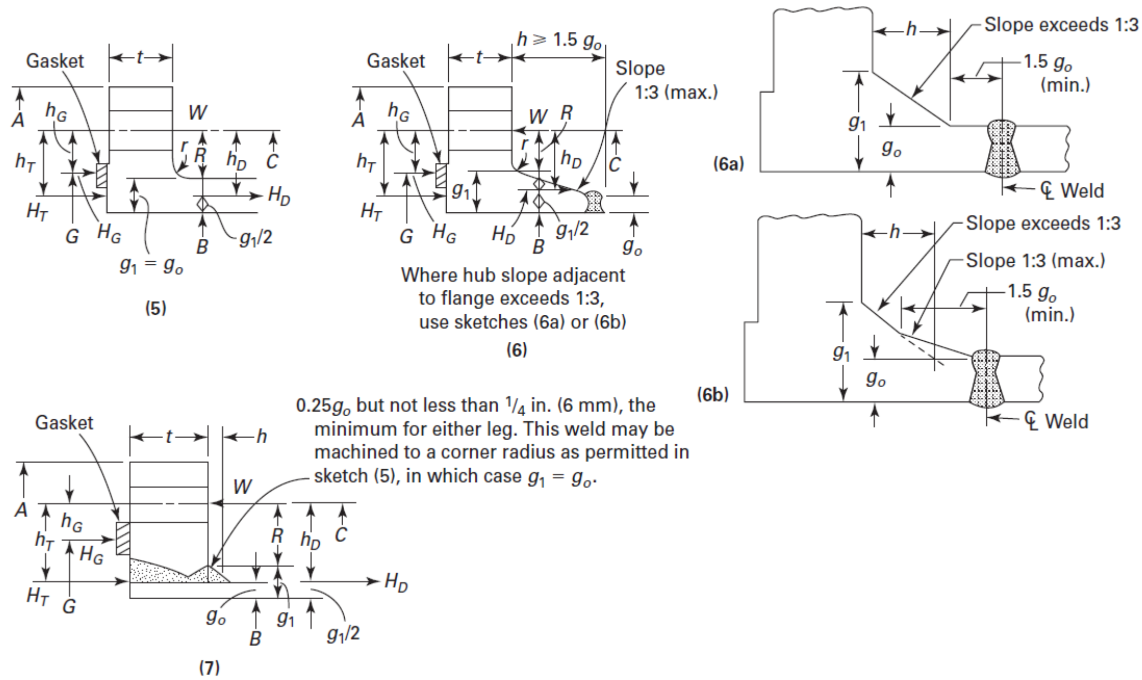


Figure 2: Integral Type Flanges

(*From ASME Section VIII, Division 1, Appendix 2*)

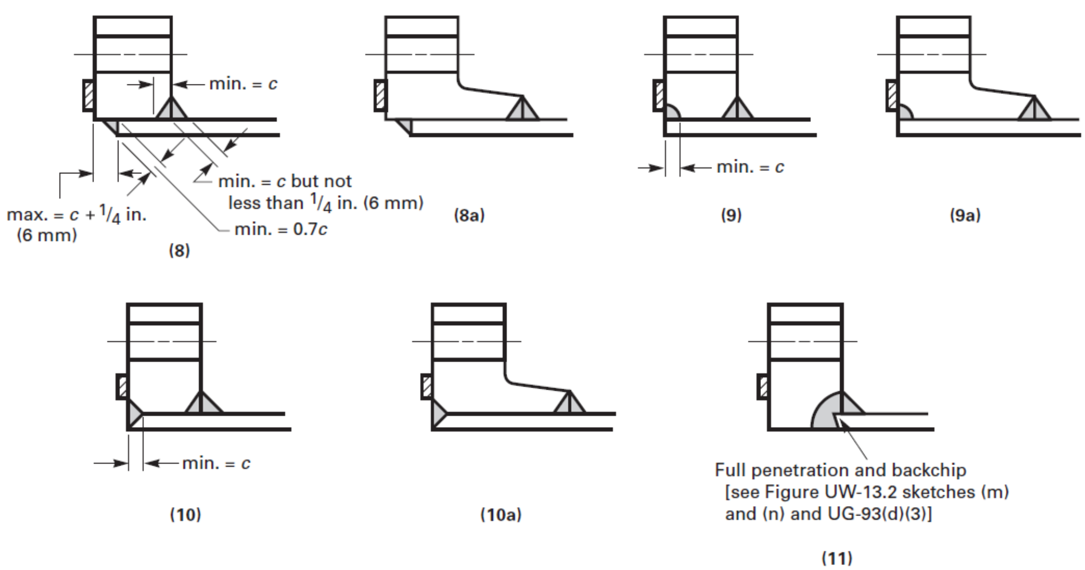


Figure 3: Optional Type Flanges

(*From ASME Section VIII, Division 1, Appendix 2*)

<u>REQUIRED BOLT LOADS</u>

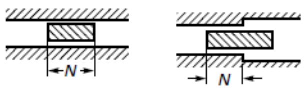
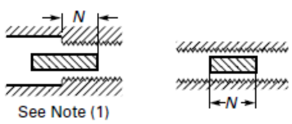
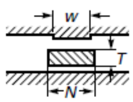
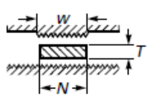
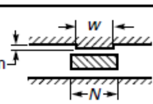
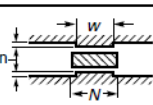
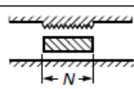
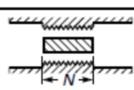
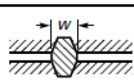
1) The required bolt load for the operating conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the maximum allowable working pressure on the area bounded by the diameter of gasket

reaction and, in addition, to maintain on the gasket a compression load H_p, which experience has shown to be sufficient to ensure a tight joint. This compression load is expressed as a multiple *m* of the internal pressure. Its value is a function of the gasket material and construction.

$$W_{m1} = H + H_p = 0.785^2 P + (2b \text{ x } 3.14GmP)$$

Where, b = Effective gasket width
= b_o when $b_o \le 1/4$ in.
= $C_b\sqrt{b_o}$ when $b_o > 1/4$ in.
C_b = Conversion factor (0.5 for US Customary units and 2.5 for SI units)
b_o = Basic gasket seating width (see Table 2)

Table 2: Effective Gasket Seating Width (*From ASME Section VIII, Division 1, Appendix 2*)

Facing Sketch (Exaggerated)	Basic Gasket Seating Width, b_o	
	Column I	Column II
(1a) 	$\frac{N}{2}$	$\frac{N}{2}$
(1b)  See Note (1)	$\frac{N}{2}$	$\frac{N}{2}$
(1c)  $w \le N$	$\frac{w+T}{2};\left(\frac{w+N}{4}\text{max}\right)$	$\frac{w+T}{2};\left(\frac{w+N}{4}\text{max}\right)$
(1d)  See Note (1) $w \le N$	$\frac{w+T}{2};\left(\frac{w+N}{4}\text{max}\right)$	$\frac{w+T}{2};\left(\frac{w+N}{4}\text{max}\right)$
(2)  1/64 in. (0.4 mm) nubbin $w \le N/2$	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
(3)  1/64 in. (0.4 mm) nubbin $w \le N/2$	$\frac{N}{4}$	$\frac{3N}{8}$
(4)  See Note (1)	$\frac{3N}{8}$	$\frac{7N}{16}$
(5)  See Note (1)	$\frac{N}{4}$	$\frac{3N}{8}$
(6) 	$\frac{w}{8}$	...

Note: Column I is for solid flat metal and ring type joints, and Column II is for all other type of joints.

N = Width used to determine the basic gasket seating width
t_G = Gasket thickness
w = Width used to determine the basic gasket seating width
G = Diameter at location of gasket load reaction
= Mean diameter of gasket when $b_o \le 1/4$ in

= Outside diameter of gasket – 2b when $b_o > 1/4$ in

m = Gasket factor

2) Before a tight joint can be obtained, it is necessary to seat the gasket properly by applying a minimum initial load (under atmospheric temperature conditions without the presence of internal pressure) which is a function of gasket material and the effective gasket area to be seated. The minimum initial bolt load required for this purpose W_{m2} shall be determined as follows:

$$W_{m2} = 3.14bGy$$

Where, y = Gasket seating load

The need for providing sufficient bolt load to seat the gasket will prevail on many low-pressure designs and with facings and materials that require a high seating load and where bolt load computed for operating conditions is insufficient to seat the joint. Accordingly, it is necessary to furnish bolting and to pretighten the bolts to provide a bolt load sufficient to satisfy both of these requirements, each one being individually investigated. When W_{m2} governs, flange proportions will be a function of the bolting instead of internal pressure.

3) Bolt loads for flanges using gaskets of self-energizing type differ from those shown above.

a. The required bolt load for the operating conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the MAWP on the area bounded by the outside diameter of the gasket. H_p is to be considered as 0 for all self-energizing gaskets except certain seal configurations which generate axial loads which may be considered.

b. $W_{m2} = 0$.

Self-energizing gaskets may be considered to require an inconsequential amount of bolting force to produce a seal. Bolting, however, must be pretightened to provide a bolt load sufficient to withstand the hydrostatic end force H.

TOTAL REQUIRED AND ACTUAL BOLT AREAS

The total cross-sectional area of bolts A_m required for both the operating conditions and gasket seating is the greater of the values for A_{m1} and A_{m2}.

$A_{m1} = W_{m1}/S_b$; S_b = Allowable bolt stress at atmospheric temperature

$A_{m2} = W_{m2}/S_a$; S_a = Allowable bolt stress at design temperature

A selection of bolts to be used shall be made such that the actual cross-sectional area of bolts A_b will not be less than A_m. For vessels in lethal service or when specified by the user, the maximum bolt spacing shall not exceed the value given below:

$$B_{smax} = 2a + \frac{6t}{m+0.5}; \qquad a = \text{Nominal bolt diameter}$$

FLANGE DESIGN BOLT LOAD

The bolt loads used in the design of the flange shall be the values obtained for the operating conditions and for gasket seating.

For operating conditions, $W = W_{m1}$.

For gasket seating, $W = \frac{(A_m+A_b)S_a}{2}$.

In addition to the minimum requirements for safety, the bolt load for gasket seating provides a margin against abuse of the flange from overbolting. Since the margin against such abuse is needed primarily for the initial, bolting-up operation which is done at atmospheric temperature and before application of internal pressure, the flange design is required to satisfy this loading only under such conditions.

FLANGE MOMENTS

The moment of a load acting on the flange is the product of the load and its moment arm. The moment arm is determined by the relative position of the bolt circle with respect to that of the load producing the moment. It is recommended that the value of h_G[(C-G)/2] be kept to a minimum to reduce flange rotation at the sealing surface.

For operating condition:

H_D	=	$0.785 \text{ x } B^2 \text{x } P$	Hydrostatic end force on area inside of flange
H	=	$0.785 \text{ x } G^2 \text{x } P$	Total hydrostatic end force
H_P	=	$2b \text{ x } 3.14GmP$	Total joint-contact surface compression load
H_G	=	$W_{m1} - H$	Gasket load for the operating condition

The moment arms for the flange loads shown above are as given in the table below:

Table 3: Moment Arms for Flange Loads Under Operating Conditions
(*From ASME Section VIII, Division 1, Appendix 2*)

	h_D	h_T	h_G
Integral-type flanges [see Figure 2-4, sketches (5), (6), (6a), (6b), and (7)] and optional type flanges calculated as integral type [see Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]	$R + 0.5g_1$	$\frac{R + g_1 + h_G}{2}$	$\frac{C-G}{2}$
Loose type, except lap-joint flanges [see Figure 2-4, sketches (2), (2a), (3), (3a), (4), and (4a)]; and optional type flanges calculated as loose type [see Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]	$\frac{C-B}{2}$	$\frac{h_D + h_G}{2}$	$\frac{C-G}{2}$
Lap-type flanges [see Figure 2-4, sketches (1) and (1a)]	$\frac{C-B}{2}$	$\frac{C-G}{2}$	$\frac{C-G}{2}$

$$M_D = H_D \text{ x } h_D$$

$$M_T = H_T \text{ x } h_T$$

$$M_G = H_G \text{ x } h_G$$

$$M_O = M_D + M_T + M_G$$

For gasket seating:

$$M_O = W\frac{(C-G)}{2}$$

For vessels in lethal service or when specified by the user, the bolt spacing correction shall be applied in calculation of flange stresses. When bolt spacing exceeds 2a + t, multiply M_O by the bolt spacing correction factor B_{SC} for calculating flange stress.

$$B_{SC} = \sqrt{\frac{B_S}{2a+t}}$$

B_S is the actual bolt spacing.

CALCULATION OF FLANGE STRESSES

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

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

$S_H = \frac{f \text{ x } M_O}{L \text{ x } g_1^2 \text{ x } B}$ Longitudinal hub stress; $L = \frac{t\,e+1}{T} + \frac{t^3}{d}$

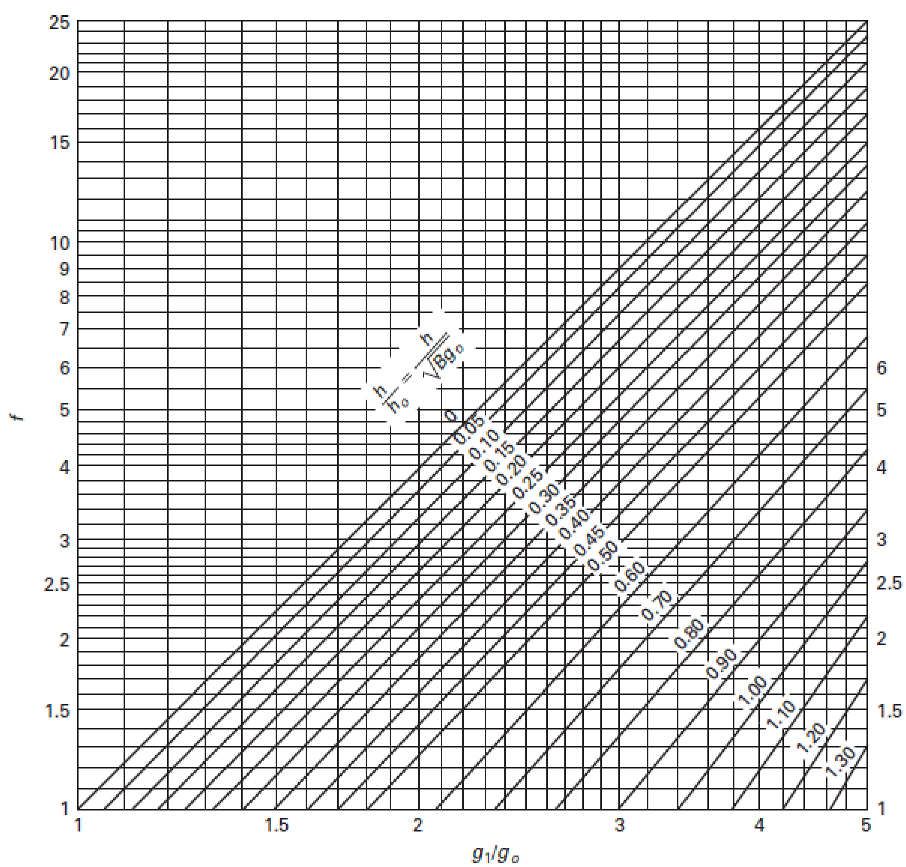
e = F/h_o for integral type flanges; F_L/h_o for loose type flanges

d = $(U/V)\ h_o g_o^2$ for integral type flanges; $(U/V_L)\ h_o g_o^2$ for loose type flanges

$S_R = \frac{[(1.33 \text{ x } t \text{ x } e)+1] \text{ x } M_O}{L \text{ x } t^2 \text{ x } B}$ Radial flange stress

$S_T = \frac{Y \text{ x } M_O}{t^2 \text{ x } B} - Z \text{ x } S_R$ Tangential flange stress

Values of *f* (Hub Stress Correction Factor)



f = 1 (minimum)
= 1 for hubs of uniform thickness (g_1/g_o = 1)
= 1 for loose hubbed flanges

GENERAL NOTE:

Figure 4: Values of f (Hub Stress Correction Factor)
(*From ASME Section VIII, Division 1, Appendix 2*)

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

S_H	=	0	Longitudinal hub stress
S_R	=	0	Radial flange stress
S_T	=	$\frac{Y \text{ x } M_O}{t^2 \text{ x } B}$	Tangential flange stress

(1) $A = (g_1/g_o) - 1$
(2) $C = 43.68(h/h_o)^4$
(3) $C_1 = 1/3 + A/12$
(4) $C_2 = 5/42 + 17A/336$
(5) $C_3 = 1/210 + A/360$
(6) $C_4 = 11/360 + 59A/5040 + (1 + 3A)/C$
(7) $C_5 = 1/90 + 5A/1008 - (1 + A)^3/C$
(8) $C_6 = 1/120 + 17A/5040 + 1/C$
(9) $C_7 = 215/2772 + 51A/1232 + (60/7 + 225A/14 + 75A^2/7 + 5A^3/2)/C$
(10) $C_8 = 31/6930 + 128A/45{,}045 + (6/7 + 15A/7 + 12A^2/7 + 5A^3/11)/C$
(11) $C_9 = 533/30{,}240 + 653A/73{,}920 + (1/2 + 33A/14 + 39A^2/28 + 25A^3/84)/C$
(12) $C_{10} = 29/3780 + 3A/704 - (1/2 + 33A/14 + 81A^2/28 + 13A^3/12)/C$
(13) $C_{11} = 31/6048 + 1763A/665{,}280 + (1/2 + 6A/7 + 15A^2/28 + 5A^3/42)/C$
(14) $C_{12} = 1/2925 + 71A/300{,}300 + (8/35 + 18A/35 + 156A^2/385 + 6A^3/55)/C$
(15) $C_{13} = 761/831{,}600 + 937A/1{,}663{,}200 + (1/35 + 6A/35 + 11A^2/70 + 3A^3/70)/C$
(16) $C_{14} = 197/415{,}800 + 103A/332{,}640 - (1/35 + 6A/35 + 17A^2/70 + A^3/10)/C$
(17) $C_{15} = 233/831{,}600 + 97A/554{,}400 + (1/35 + 3A/35 + A^2/14 + 2A^3/105)/C$
(18) $C_{16} = C_1\ C_7C_{12} + C_2\ C_8C_3 + C_3C_8C_2 - (C_3{}^2C_7 + C_8{}^2C_1 + C_2{}^2C_{12})$
(19) $C_{17} = [C_4C_7C_{12} + C_2C_8C_{13} + C_3C_8C_9 - (C_{13}C_7C_3 + C_8{}^2C_4 + C_{12}C_2C_9)]/C_{16}$
(20) $C_{18} = [C_5C_7C_{12} + C_2C_8C_{14} + C_3C_8C_{10} - (C_{14}C_7C_3 + C_8{}^2C_5 + C_{12}C_2C_{10})]/C_{16}$
(21) $C_{19} = [C_6C_7C_{12} + C_2C_8C_{15} + C_3C_8C_{11} - (C_{15}C_7C_3 + C_8{}^2C_6 + C_{12}C_2C_{11})]/C_{16}$
(22) $C_{20} = [C_1C_9C_{12} + C_4C_8C_3 + C_3C_{13}C_2 - (C_3{}^2C_9 + C_{13}C_8C_1 + C_{12}C_4C_2)]/C_{16}$
(23) $C_{21} = [C_1C_{10}C_{12} + C_5C_8C_3 + C_3C_{14}C_2 - (C_3{}^2C_{10} + C_{14}C_8C_1 + C_{12}C_5C_2)]/C_{16}$
(24) $C_{22} = [C_1C_{11}C_{12} + C_6C_8C_3 + C_3C_{15}C_2 - (C_3{}^2C_{11} + C_{15}C_8C_1 + C_{12}C_6C_2)]/C_{16}$
(25) $C_{23} = [C_1C_7C_{13} + C_2C_9C_3 + C_4C_8C_2 - (C_3C_7C_4 + C_8C_9C_1 + C_2{}^2C_{13})]/C_{16}$
(26) $C_{24} = [C_1C_7C_{14} + C_2C_{10}C_3 + C_5C_8C_2 - (C_3C_7C_5 + C_8C_{10}C_1 + C_2{}^2C_{14})]/C_{16}$
(27) $C_{25} = [C_1C_7C_{15} + C_2C_{11}C_3 + C_6C_8C_2 - (C_3C_7C_6 + C_8C_{11}C_1 + C_2{}^2C_{15})]/C_{16}$
(28) $C_{26} = -(C/4)^{1/4}$
(29) $C_{27} = C_{20} - C_{17} - 5/12 + C_{17}C_{26}$
(30) $C_{28} = C_{22} - C_{19} - 1/12 + C_{19}C_{26}$
(31) $C_{29} = -(C/4)^{1/2}$
(32) $C_{30} = -(C/4)^{3/4}$
(33) $C_{31} = 3A/2 - C_{17}C_{30}$
(34) $C_{32} = 1/2 - C_{19}C_{30}$
(35) $C_{33} = 0.5C_{26}C_{32} + C_{28}C_{31}C_{29} - (0.5C_{30}C_{28} + C_{32}C_{27}C_{29})$
(36) $C_{34} = 1/12 + C_{18} - C_{21} - C_{18}C_{26}$
(37) $C_{35} = -C_{18}(C/4)^{3/4}$
(38) $C_{36} = (C_{28}C_{35}C_{29} - C_{32}C_{34}C_{29})/C_{33}$
(39) $C_{37} = [0.5C_{26}C_{35} + C_{34}\ C_{31}C_{29} - (0.5C_{30}C_{34} + C_{35}C_{27}C_{29})]/C_{33}$
(40) $E_1 = C_{17}C_{36} + C_{18} + C_{19}C_{37}$
(41) $E_2 = C_{20}C_{36} + C_{21} + C_{22}C_{37}$
(42) $E_3 = C_{23}C_{36} + C_{24} + C_{25}C_{37}$
(43) $E_4 = 1/4 + C_{37}/12 + C_{36}/4 - E_3/5 - 3E_2/2 - E_1$
(44) $E_5 = E_1(1/2 + A/6) + E_2(1/4 + 11A/84) + E_3(1/70 + A/105)$
(45) $E_6 = E_5 - C_{36}\ (7/120 + A/36 + 3A/C) - 1/40 - A/72 - C_{37}(1/60 + A/120 + 1/C)$

Figure 5: Flange Factors
(*From ASME Section VIII, Division 1, Appendix 2*)

In the above equations, for integral flanges, $f = C_{36}/(1 + A)$ and for loose flanges, f = 1. Values of f are shown in graphical form in Figure 4. Various factors used in the equations are shown in Figure 5. Factor A in the Figure is not the outside diameter of the flange, and Factor C is not the bolt circle diameter of the flange.

$$V = \frac{E_4}{\left(\frac{2.73}{C}\right)^{\frac{1}{4}} (1+A)^3}$$

$$V_L = \frac{(1/4)-(C_{24}/5)-(3C_{21}/2)-C_{18}}{\left(\frac{2.73}{C}\right)^{\frac{1}{4}} (1+A)^3}$$

ALLOWABLE FLANGE DESIGN STRESSES

The calculated flange stresses shall satisfy the following acceptance criteria:

1) For cast iron: $S_H \leq S_f$

For materials other than cast iron: $S_H \leq 1.5 \text{ x } S_f$ with following exceptions.

a) For optional flanges designed as integral (sketches 8, 8a, 9, 9a, 10, 10a and 11) and integral type (sketch 7) where the neck material constitutes the hub of the flange: $S_H \leq \min\ (1.5 \text{ x } S_f, 1.5 \text{ x } S_n)$

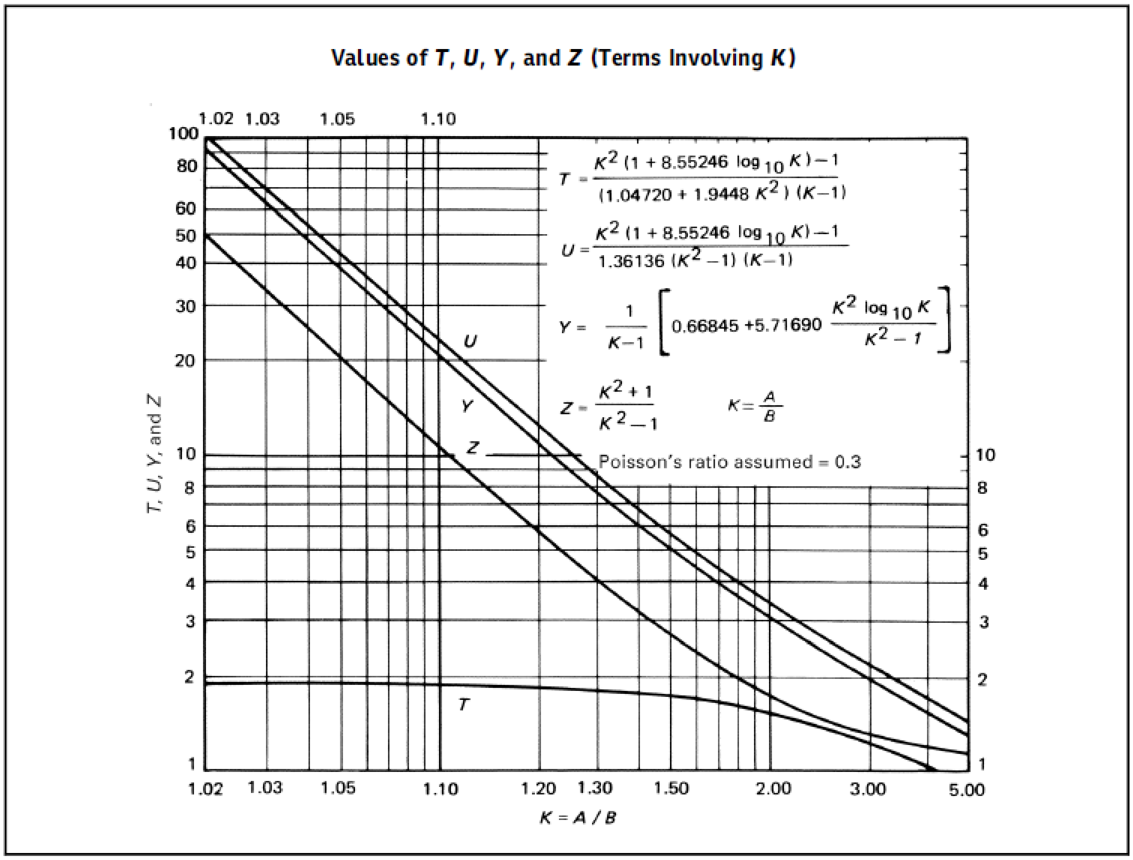


Figure 6: Values of T, U, Y, and Z (Terms Involving K)
(*From ASME Section VIII, Division 1, Appendix 2*)

b) For integral type flanges with hub welded to the neck, pipe or vessel wall (sketches 6, 6a and 6b): $S_H \leq \min\ (1.5 \text{ x } S_f, 2.5 \text{ x } S_n)$

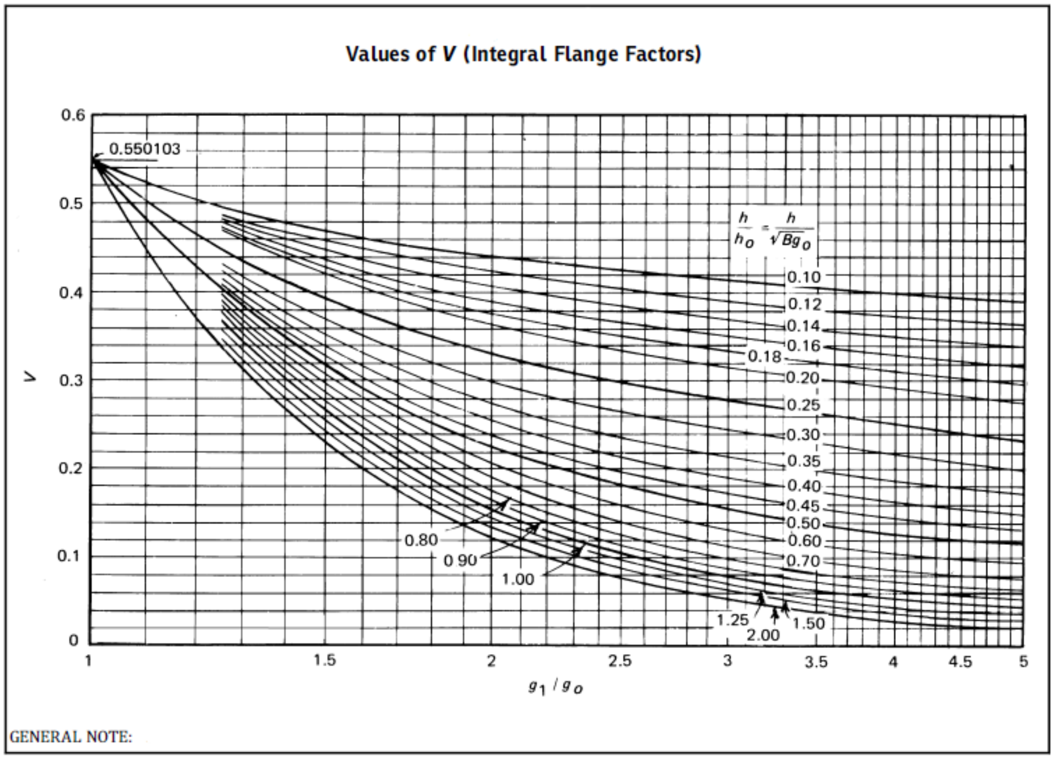


Figure 7: Values of V (Integral Flange Factors)
(*From ASME Section VIII, Division 1, Appendix 2*)

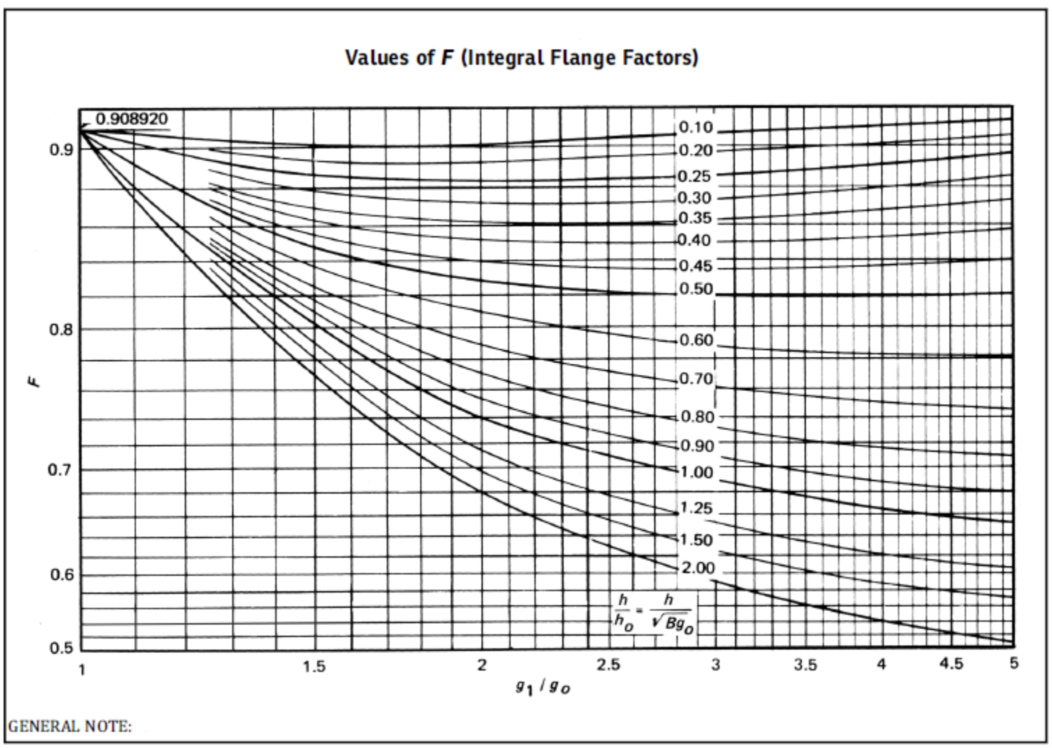
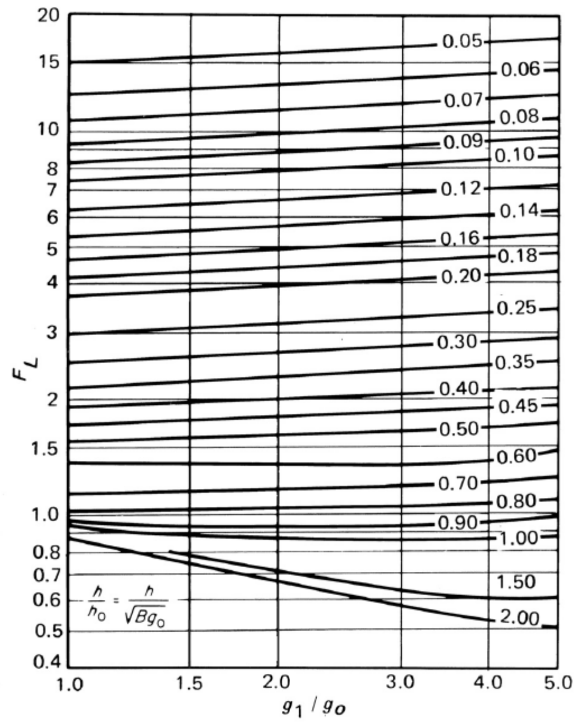


Figure 8: Values of F (Integral Flange Factors)
(*From ASME Section VIII, Division 1, Appendix 2*)



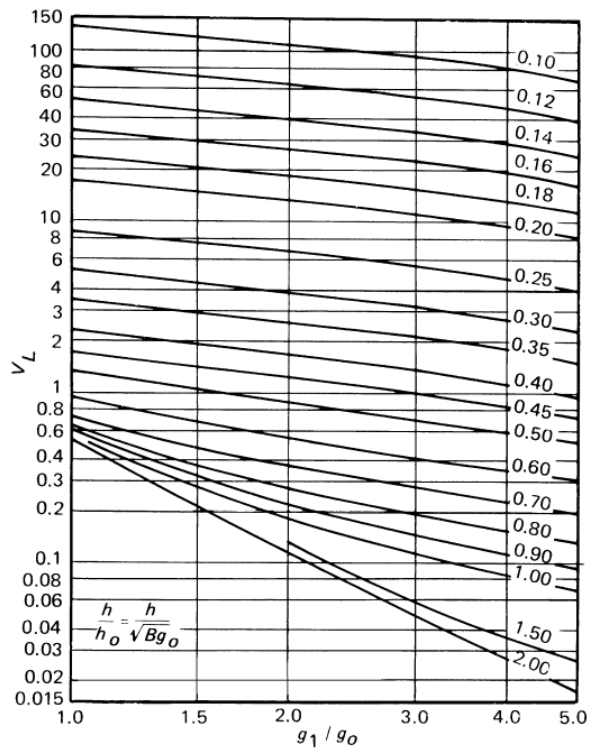


Figure 9: Values of F_L (Loose Hub Flange Factors) **Figure 10: Values of V_L (Loose Hub Flange Factors)**

(*From ASME Section VIII, Division 1, Appendix 2*)

S_n is the 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.

2) Radial flange stress: $S_R \leq S_f$.

3) Tangential flange stress: $S_T \leq S_f$.

4) $(S_H + S_R)/2 \leq S_f$

$(S_H + S_T)/2 \leq S_f$

For hub flanges attached as shown in sketches 2, 3, 4 and 4c of Figure 1, the nozzle neck, vessel or pipe wall shall not be considered to have any value as a hub.

In the case of loose type flanges with laps as shown in sketches 1 and 1a of Figure 1, where the gasket is so located that the lap is subject to shear, the shearing stress shall not exceed 0.8 x S_n for the material of the lap.

In the case of welded flanges, shown in sketches 3, 3a, 4, 4a, 4b, 4c, 7, 8, 8a, 9, 9a, 10 and 10a of Figures 1, 2 and 3 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 x S_n.

The shearing stress shall be calculated on the basis of W_{m1} or W_{m2}, whichever is greater.

FLANGES SUBJECT TO EXTERNAL PRESSURE

The design of flanges for external pressure only be based on the equations for internal pressure except that:

<u>For operating conditions</u>

$$M_O = H_D (h_D - h_G) + H_T (h_T - h_G)$$

For gasket seating

$M_O = W \times H_G$

Where $W = \frac{A_{m2} + A_b}{2} \times S_a$

$H_D = 0.785\ B^2\ P_e$

$H_T = H - H_D$

$H = 0.785\ G^2\ P_e$

P_e = External design pressure

FLANGE RIGIDITY

Flanges that have been designed based on allowable stress limits alone may not be sufficiently rigid to control leakage. The rigidity criteria shown here provide one method of checking flange rigidity. The factors provided here have been proven through extensive user experience for a wide variety of joint design and service conditions. The use of rigidity index does not guarantee a leakage rate within established limits. The use of these factors must be considered as only part of the system of joint design and assembly requirements to ensure leak tightness.

Successful service experience may be used as an alternative to the flange rigidity rules for fluid services that are nonlethal and nonflammable and designed within the temperature range of -20°F to 366°F without exceeding design pressures of 150 psi.

FLANGE RIGIDITY FACTORS

Integral-type flanges and optional type flanges designed as integral-type flanges

$$J = \frac{52.14 \times V \times M_o}{L \times E \times g_o^2 \times K_I \times h_o} \leq 1.0$$

Loose-type flanges with hubs

$$J = \frac{52.14 \times V_L \times M_o}{L \times E \times g_o^2 \times K_L \times h_o} \leq 1.0$$

Loose-type flanges without hubs and optional flanges designed as loose-type flanges

$$J = \frac{109.4 \times M_o}{E \times t^3 \times K_L\ (\ln K)} \leq 1.0$$

Where, E = Modulus of elasticity for the flange material at design temperature (operating condition) or at atmospheric temperature (gasket seating)

J = Rigidity index ≤ 1

K_I = Rigidity factor for integral or optional flange types = 0.3

K_L = Rigidity factor for loose-type flanges = 0.2

Experience has indicated that K_I and K_L provided above are sufficient for most services; other values may be used with the User's agreement.

REFERENCES:

ASME Boiler and Pressure Vessel, Section VIII, Division 1, 2023 Edition