APPENDIX 2
RULES FOR BOLTED FLANGE CONNECTIONS WITH RING TYPE GASKETS

GENERAL

2-1 SCOPE

(a) The rules in Appendix 2 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, and are to be used in conjunction with the applicable requirements in Subsections A, B, and C of this Division. These rules are not to be used for the determination of the thickness of supported or unsupported tubesheets integral with a bolting flange as illustrated in Fig. UW-13.2 sketches (h) through (l) or Fig. UW-13.3 sketch (c). Appendix S provides discussion on Design Considerations for Bolted Flanged Connections.

These rules provide only for hydrostatic end loads and gasket seating. The flange design methods outlined in 2-4 through 2-8 are applicable to circular flanges under internal pressure. Modifications of these methods are outlined in 2-9 and 2-10 for the design of split and noncircular flanges. See 2-11 for flanges with ring type gaskets subject to external pressure, 2-12 for flanges with nut-stops, and 2-13 for reverse flanges. Proper allowance shall be made if connections are subject to external loads other than external pressure.

(b) The design of a flange involves the selection of the gasket (material, type, and dimensions), flange facing, bolting, hub proportions, flange width, and flange thickness. See Note 1, 2-5(c)(1). Flange dimensions shall be such that the stresses in the flange, calculated in accordance with 2-7, do not exceed the allowable flange stresses specified in 2-8. All calculations shall be made on dimensions in the corroded condition.

(c) It is recommended that bolted flange connections conforming to the standards listed in UG-44 be used for connections to external piping. These standards may be used for other bolted flange connections within the limits of size in the standards and the pressure–temperature ratings permitted in UG-44. The ratings in these standards are based on the hub dimensions given or on the minimum specified thickness of flanged fittings of integral construction. Flanges fabricated from rings may be used in place of the hub flanges in these standards provided that their strength, calculated by the rules in this Appendix, is not less than that calculated for the corresponding size of hub flange.

(d) Except as otherwise provided in (c) above, bolted flange connections for unfired pressure vessels shall satisfy the requirements in this Appendix.

(e) The rules of this Appendix should not be construed to prohibit the use of other types of flanged connections provided they are designed in accordance with good engineering practice and method of design is acceptable to the Inspector. Some examples of flanged connections which might fall in this category are as follows:

(1) flanged covers as shown in Fig. 1-6;

(2) bolted flanges using full-face gaskets;

(3) flanges using means other than bolting to restrain the flange assembly against pressure and other applied loads.

2-2 MATERIALS A02

(a) Materials used in the construction of bolted flange connections shall comply with the requirements given in UG-4 through UG-14.

(b) Flanges made from ferritic steel and designed in accordance with this Appendix shall be full-annealed, normalized, normalized and tempered, or quenched and tempered when the thickness of the flange section exceeds 3 in. (76 mm).

(c) Material on which welding is to be performed shall be proved 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 this Division.

(d) Fabricated hubbed flanges shall be in accordance with the following.

(1) Hubbed flanges may be machined from a hot rolled or forged billet or forged bar. The axis of the finished flange shall be parallel to the long axis of the original billet or bar. (This is not intended to imply that the axis of the finished flange and the original billet must be concentric.)

(2) Hubbed flanges [except as permitted in (1) above] shall not be machined from plate or bar stock material unless the material has been formed into a ring, and further provided that:

(a) in a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange. (This is not intended to imply that the original plate surface be present in the finished flange.)

(b) the joints in the ring are welded butt joints that conform to the requirements of this Division. Thickness to be used to determine postweld heat treatment and radiography requirements shall be the lesser of

$$t \text{ or } \frac{(A - B)}{2}$$

where these symbols are as defined in 2-3.

01 *(c)* the back of the flange and the outer surface of the hub are examined by either the magnetic particle method as per Appendix 6 or the liquid penetrant method as per Appendix 8.

(e) Bolts, studs, nuts, and washers shall comply with the requirements in this Division. It is recommended that bolts and studs have a nominal diameter of not less than ½ in. If bolts or studs smaller than ½ in. are used, ferrous bolting material shall be of alloy steel. Precautions shall be taken to avoid over-stressing small-diameter bolts.

2-3 NOTATION

The symbols described below are used in the formulas for the design of flanges (see also Fig. 2-4):

A = outside diameter of flange or, where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots, in. (mm)

A_b = cross-sectional area of the bolts using the root diameter of the thread or least diameter of unthreaded position, if less, sq in. (mm^2)

A_m = total required cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2}, sq in. (mm^2)

A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions, sq in. (mm^2)

$= W_{m1} / S_b$

A_{m2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating, sq in. (mm^2)

$= W_{m2} / S_a$

B = inside diameter of flange, in. (mm). When B is less than $20g_1$, it will be optional for the designer to substitute B_1 for B in the formula for longitudinal stress S_H.

$B_1 = B + g_1$, in. (mm), for loose type flanges and for integral type flanges that have calculated values h / h_o and g_1 / g_o which would indicate an f value of less than 1.0, although the minimum value of f permitted is 1.0.

$B_1 = B + g_o$, in. (mm), for integral type flanges when f is equal to or greater than one

b = effective gasket or joint-contact-surface seating width, in. (mm) [see Note 1, 2-5(c)(1)]

b_o = basic gasket seating width, in. (mm) (from Table 2-5.2)

C = bolt-circle diameter, in. (mm)

c = basic dimension used for the minimum sizing of welds, in. (mm), equal to t_n or t_x, whichever is less

d = factor, in.^{3} (mm^3)

$$d = \frac{U}{V} h_o g_o^2 \text{ for integral type flanges}$$

$$d = \frac{U}{V_L} h_o g_o^2 \text{ for loose type flanges}$$

e = factor, in.$^{-1}$ (mm^{-1})

$$e = \frac{F}{h_o} \text{ for integral type flanges}$$

$$e = \frac{F_L}{h_o} \text{ for loose type flanges}$$

F = factor for integral type flanges (from Fig. 2-7.2)

F_L = factor for loose type flanges (from Fig. 2-7.4)

f = hub stress correction factor for integral flanges from Fig. 2-7.6 (When greater than one, this is the ratio of the stress in the small end of hub to the stress in the large end.) (For values below limit of figure, use $f = 1$.)

G = diameter, in. (mm), at location of gasket load reaction. Except as noted in sketch (1) of Fig. 2-4, G is defined as follows (see Table 2-5.2):

When $b_o \leq 1/4$ in. (6 mm), G = mean diameter of gasket contact face, in. (mm)

When $b_o > 1/4$ in. (6 mm), G = outside diameter of gasket contact face less $2b$, in. (mm)

g_o = thickness of hub at small end, in. (mm)

g_1 = thickness of hub at back of flange, in. (mm)

H = total hydrostatic end force, lb (kN)

$$= 0.785G^2P$$

H_D = hydrostatic end force on area inside of flange, lb (kN)

$$= 0.785B^2P$$

H_G = gasket load (difference between flange design bolt load and total hydrostatic end force), lb (kN)

$$= W - H$$

H_p = total joint-contact surface compression load, lb (kN)

$$= 2b \times 3.14\ GmP$$

H_T = difference between total hydrostatic end force and the hydrostatic end force on area inside of flange, lb (kN)

$$= H - H_D$$

h = hub length, in. (mm)

h_D = radial distance from the bolt circle, to the circle on which H_D acts, as prescribed in Table 2-6, in. (mm)

h_G = radial distance from gasket load reaction to the bolt circle, in. (mm)

$$= (C - G)/2$$

h_o = factor, in. (mm)

$$= \sqrt{Bg_o}$$

h_T = radial distance from the bolt circle to the circle on which H_T acts as prescribed in Table 2-6, in. (mm)

K = ratio of outside diameter of flange to inside diameter of flange

$$= A/B$$

L = factor

$$= \frac{te + 1}{T} + \frac{t^3}{d}$$

M_D = component of moment due to H_D, in.-lb (kN•m)

$$= H_D h_D$$

M_G = component of moment due to H_G, in.-lb (kN•m)

$$= H_G h_G$$

M_0 = total moment acting upon the flange, for the operating conditions or gasket seating as may apply, in.-lb (kN•m) (see 2-6)

M_T = component of moment due to H_T, in.-lb (kN•m)

$$= H_T h_T$$

m = gasket factor, obtain from Table 2-5.1 [see Note 1, 2-5(c)(1)]

N = width, in. (mm), used to determine the basic gasket seating with b_o, based upon the possible contact width of the gasket (see Table 2-5.2)

P = internal design pressure (see UG-21), psi (kPa). For flanges subject to external design pressure, see 2-11.

R = radial distance from bolt circle to point of intersection of hub and back of flange, in. (mm). For integral and hub flanges,

$$R = \frac{C - B}{2} - g_1$$

S_a = allowable bolt stress at atmospheric temperature, psi (kPa) (see UG-23)

S_b = allowable bolt stress at design temperature, psi (kPa) (see UG-23)

S_f = allowable design stress for material of flange at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply, psi (kPa) (see UG-23)

S_n = 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, psi (kPa) (see UG-23)

S_H = calculated longitudinal stress in hub, psi (kPa)

S_R = calculated radial stress in flange, psi (kPa)
S_T = calculated tangential stress in flange, psi (kPa)
T = factor involving K (from Fig. 2-7.1)
t = flange thickness, in. (mm)
t_n = nominal thickness of shell or nozzle wall to which flange or lap is attached, in. (mm)
t_x = two times the thickness g_0, when the design is calculated as an integral flange, in. (mm), or two times the thickness, in. (mm), of shell nozzle wall required for internal pressure, when the design is calculated as a loose flange, but not less than $\frac{1}{4}$ in. (6 mm)
U = factor involving K (from Fig. 2-7.1)
V = factor for integral type flanges (from Fig. 2-7.3)
V_L = factor for loose type flanges (from Fig. 2-7.5)
W = flange design bolt load, for the operating conditions or gasket seating, as may apply, lb (kN) [see 2-5(e)]
W_{m1} = minimum required bolt load for the operating conditions, lb (kN) [see 2-5(c)]. For flange pairs used to contain a tubesheet for a floating head for a U-tube type of heat exchangers, or for any other similar design, W_{m1} shall be the larger of the values as individually calculated for each flange, and that value shall be used for both flanges.
W_{m2} = minimum required bolt load for gasket seating, lb (kN) [see 2-5(c)]
w = width, in. (mm), used to determine the basic gasket seating width b_0, based upon the contact width between the flange facing and the gasket (see Table 2-5.2)
Y = factor involving K (from Fig. 2-7.1)
y = gasket or joint-contact-surface unit seating load, psi (kPa) [see Note 1, 2-5(c)]
Z = factor involving K (from Fig. 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 mechanical strength equivalent of integral attachment. See Fig. 2-4 sketches (1), (1a), (2), (2a), (3), (3a), (4), and (4a) for typical loose type flanges and the location of the loads and moments. Welds and other details of construction shall satisfy the dimensional requirements given in Fig. 2-4 sketches (1), (1a), (2), (2a), (3), (3a), (4), and (4a).

(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 arc or gas 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. In welded construction, the nozzle neck, vessel, or pipe wall is considered to act as a hub. See Fig. 2-4 sketches (5), (6), (6a), (6b), and (7) for typical integral type flanges and the location of the loads and moments. Welds and other details of construction shall satisfy the dimensional requirements given in Fig. 2-4 sketches (5), (6), (6a), (6b), and (7).

(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:

$$g_0 = \tfrac{5}{8} \text{ in. (16 mm)}$$

$$B/g_0 = 300$$

$$P = 300 \text{ psi (2 070 kPa)}$$

$$\text{operating temperature} = 700°\text{F (371°C)}$$

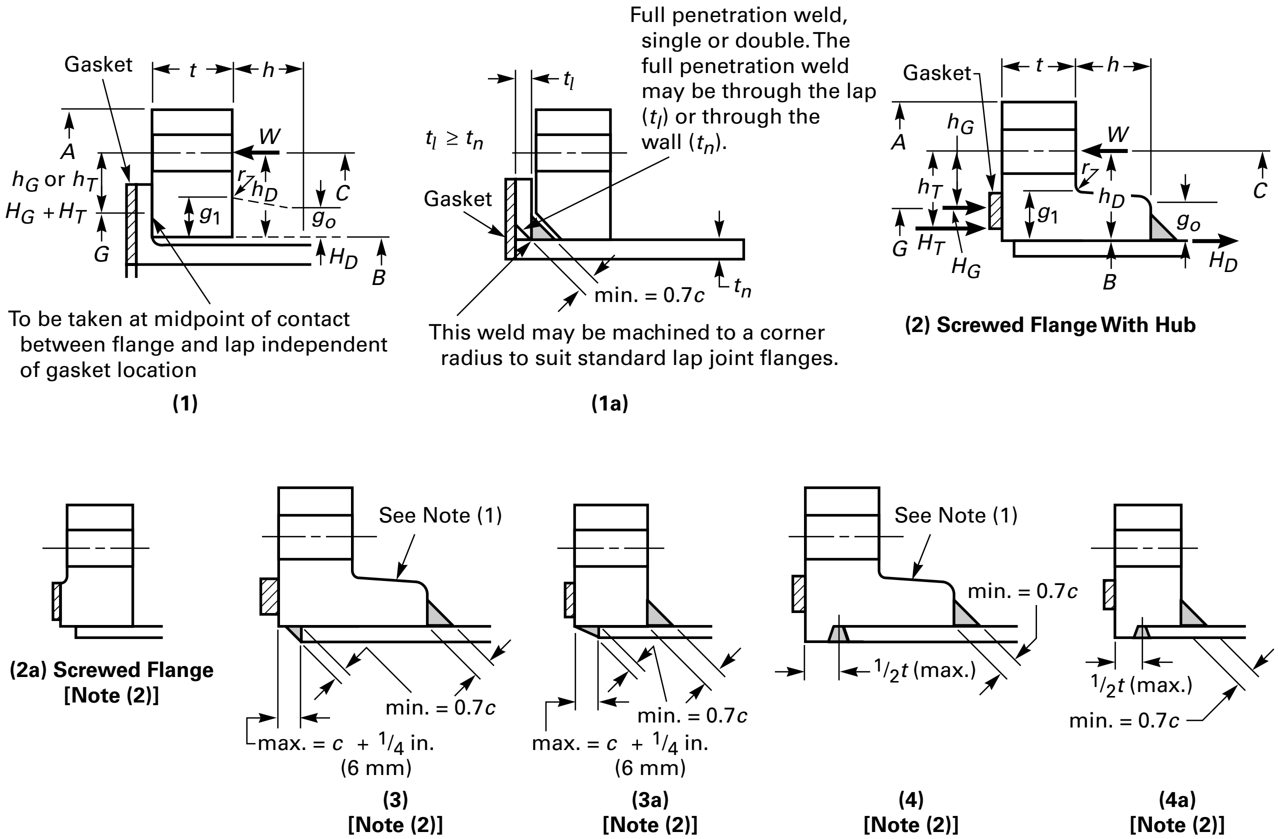
See Fig. 2-4 sketches (8), (8a), (9), (9a), (10), (10a), and (11) for typical optional type flanges. Welds and other details of construction shall satisfy the dimensional requirements given in Fig. 2-4 sketches (8), (8a), (9), (9a), (10), (10a), and (11).

2-5 BOLT LOADS

(a) General Requirements

(1) In the design of a bolted flange connection, calculations shall be made for each of the two design conditions of operating and gasket seating, and the more severe shall control.

(2) In the design of flange pairs used to contain a tubesheet of a heat exchanger or any similar design where the flanges and/or gaskets may not be the same, loads must be determined for the most severe condition of operating and/or gasket seating loads applied to each side at the same time. This most severe condition may be gasket seating on one flange with operating



NOTES (Loose Type Flanges):
(1) For hub tapers 6 deg or less, use $g_o = g_1$.
(2) Loading and dimensions for sketches (2a), (3), (3a), (4), and (4a) not shown are the same as for sketch (2).

Loose Type Flanges

FIG. 2-4 TYPES OF FLANGES

on the other, gasket seating on each flange at the same time, or operating on each flange at the same time. Although no specific rules are given for the design of the flange pairs, after the loads for the most severe conditions are determined, calculations shall be made for each flange following the rules of Appendix 2.

(b) Design Conditions

(1) Operating Conditions. The conditions required to resist the hydrostatic end force of the design pressure tending to part the joint, and to maintain on the gasket or joint-contact surface 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 or contact area to be kept tight under pressure, per Formula (1) in (c)(1) below, and determines one of the two requirements for the amount of the bolting A_{m1}. This load is also used for the design of the flange, per Formula (3) in (d) below.

(2) Gasket Seating. The conditions existing when the gasket or joint-contact surface is seated by applying an initial load with the bolts when assembling the joint, at atmospheric temperature and pressure. The minimum initial load considered to be adequate for proper seating is a function of the gasket material, and the effective gasket or contact area to be seated, per Formula (2) in (c)(2) below, and determines the other of the two requirements for the amount of bolting A_{m2}. For the design of the flange, this load is modified per Formula (4) in (d) below to take account of the operating conditions, when these govern the amount of bolting

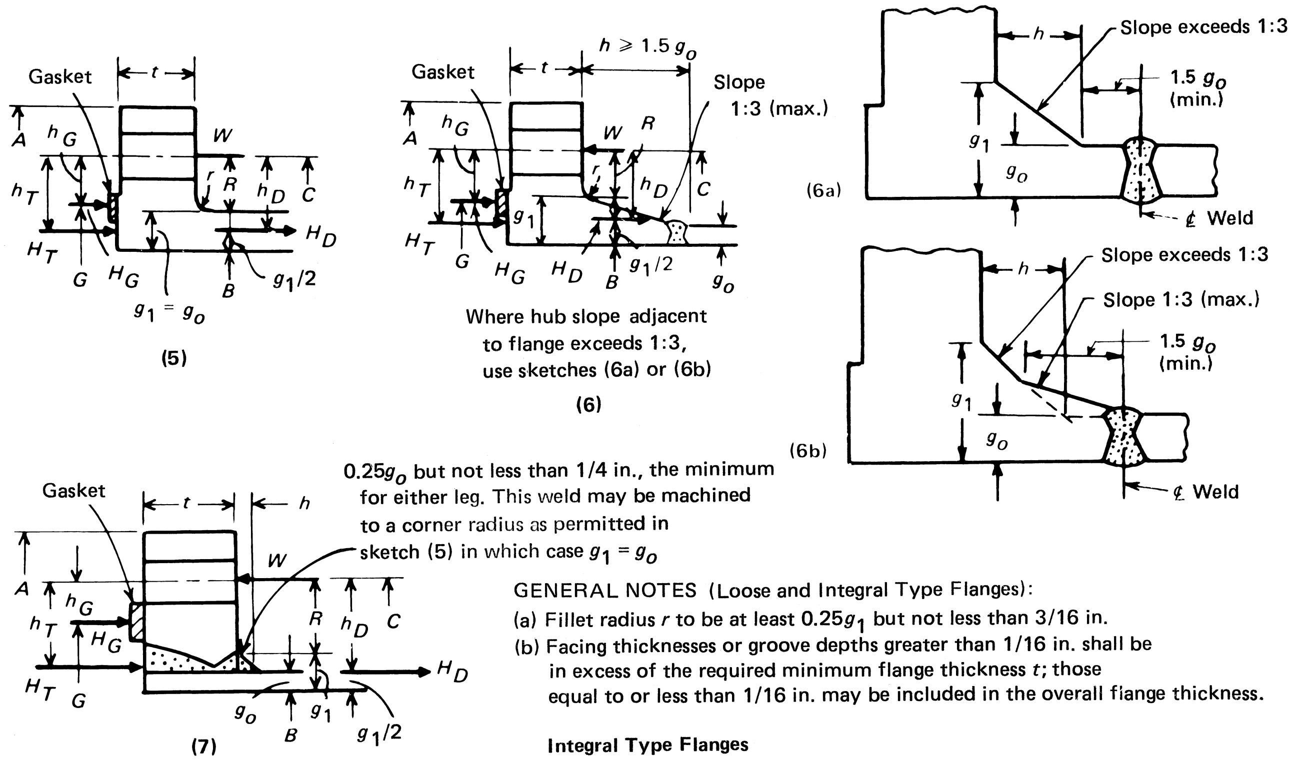


FIG. 2-4 TYPES OF FLANGES (CONT'D)

required A_m, as well as the amount of bolting actually provided A_b.

(c) Required Bolt Loads. The flange bolt loads used in calculating the required cross-sectional area of bolts shall be determined as follows.

(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 or joint-contact surface a compression load H_p, which experience has shown to be sufficient to assure 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. See Note 1.)

NOTE 1: Tables 2-5.1 and 2-5.2 give a list of many commonly used gasket materials and contact facings, with suggested values of m, b, and y that have proved satisfactory in actual service. These values are suggested only and are not mandatory. Values that are too low may result in leakage at the joint without affecting the safety of the design. The primary proof that the values are adequate is the hydrostatic test.

The required bolt load for the operating conditions W_{m1} is determined in accordance with Formula (1).

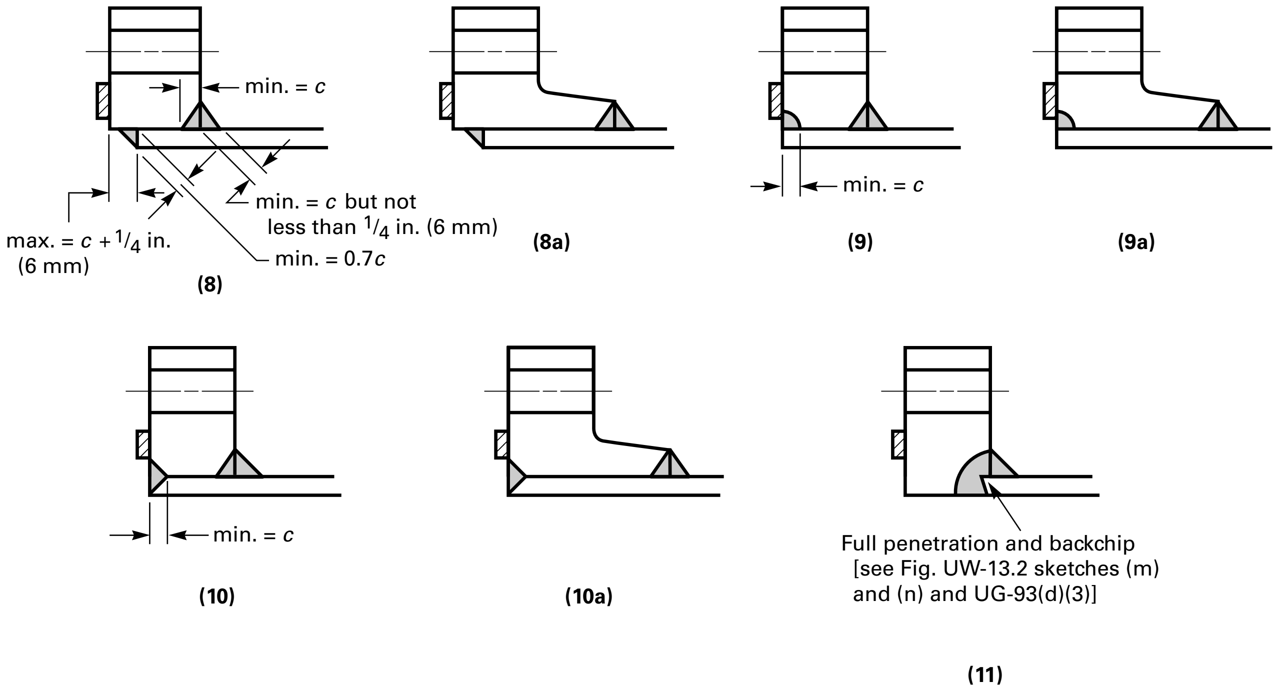
$$W_{m1} = H + H_p = 0.785G^2P + (2b \times 3.14GmP) \qquad (1)^{**}$$

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

$$W_{m2} = 3.14bGy \qquad (2)^{**}$$

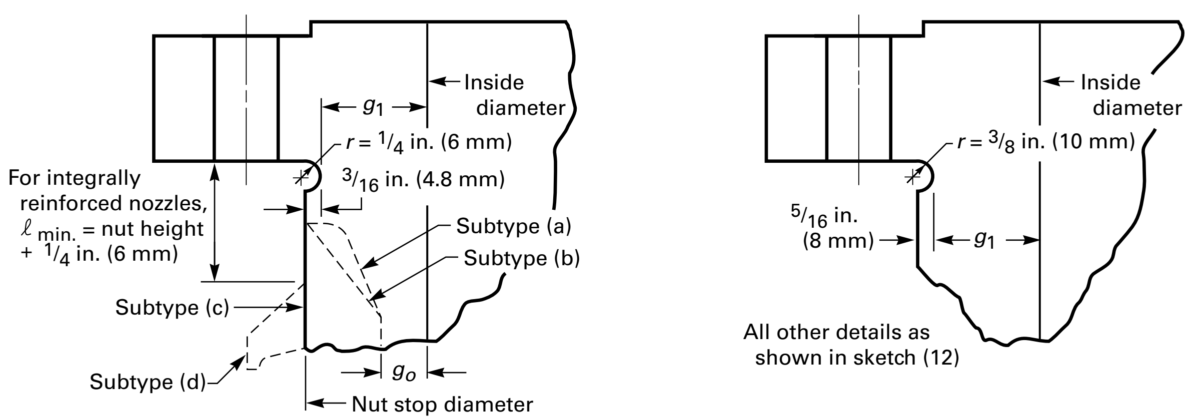
For flange pairs used to contain a tubesheet for a floating head for a U-tube type of heat exchanger, or for any other similar design, and where the flanges and/or gaskets are not the same, W_{m2} shall be the larger of the values obtained from Formula (2) as

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.



GENERAL NOTES: (Optional Type Flanges):
(a) Optional type flanges may be calculated as either loose or integral type. See 2-4.
(b) Loadings and dimensions not shown in sketches (8), (8a), (9), (9a), (10), and (10a) are the same as shown in sketch (2) when the flange is calculated as a loose type flange and as shown in sketch (7) when the flange is calculated as an integral type flange.
(c) The groove and fillet welds between the flange back face and the shell given in sketch (8) also apply to sketches (8a), (9), (9a), (10), and (10a).

Optional Type Flanges



(12) For Flanged Nozzles 18 in. (457 mm) and Smaller Nominal Size

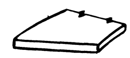
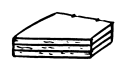
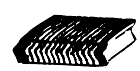
(12a) For Flanged Nozzles Over 18 in. (457 mm) Nominal Size

GENERAL NOTES: (Flanges With Nut Stops):
For subtypes (a) and (b), g_o is the thickness of the hub at the small end. For subtypes (c) and (d), $g_o = g_1$.

Flanges With Nut Stops

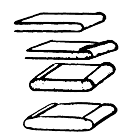
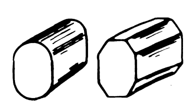
FIG. 2-4 TYPES OF FLANGES (CONT'D)

TABLE 2-5.1
GASKET MATERIALS AND CONTACT FACINGS[1]
Gasket Factors m for Operating Conditions and Minimum Design Seating Stress y

Gasket Material	Gasket Factor m	Min. Design Seating Stress y, psi (MPa)	Sketches	Facing Sketch and Column in Table 2-5.2
Self-energizing types (0 rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0 (0)	. . .	. . .
Elastomers without fabric or high percent of asbestos fiber:				(1a),(1b),(1c),(1d), (4),(5); Column II
Below 75A Shore Durometer	0.50	0 (0)		
75A or higher Shore Durometer	1.00	200 (1.4)		
Asbestos with suitable binder for operating conditions:				(1a),(1b),(1c),(1d), (4),(5); Column II
$\frac{1}{8}$ in. (3.2 mm) thick	2.00	1,600 (11)		
$\frac{1}{16}$ in. (1.6 mm) thick	2.75	3,700 (26)		
$\frac{1}{32}$ in. (0.8 mm) thick	3.50	6,500 (45)		
Elastomers with cotton fabric insertion	1.25	400 (2.8)		(1a),(1b),(1c),(1d), (4),(5); Column II
Elastomers with asbestos fabric insertion (with or without wire reinforcement):				(1a),(1b),(1c),(1d), (4),(5); Column II
3-ply	2.25	2,200 (15)		
2-ply	2.50	2,900 (20)		
1-ply	2.75	3,700 (26)		
Vegetable fiber	1.75	1,100 (7.6)		(1a),(1b),(1c),(1d), (4)(5); Column II
Spiral-wound metal, asbestos filled:				(1a),(1b); Column II
Carbon	2.50	10,000 (69)		
Stainless, Monel, and nickel-base alloys	3.00	10,000 (69)		
Corrugated metal, asbestos inserted, or corrugated metal, jacketed asbestos filled:			 	(1a),(1b); Column II
Soft aluminum	2.50	2,900 (20)		
Soft copper or brass	2.75	3,700 (26)		
Iron or soft steel	3.00	4,500 (31)		
Monel or 4%–6% chrome	3.25	5,500 (38)		
Stainless steels and nickel-base alloys	3.50	6,500 (45)		

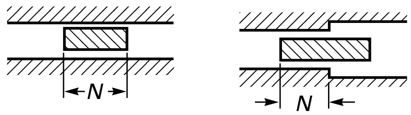
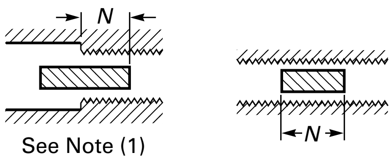
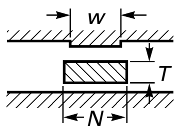
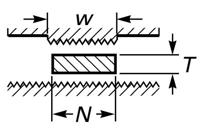
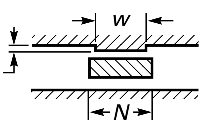
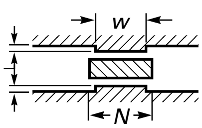
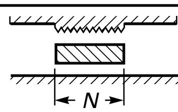
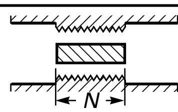
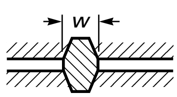
(continued)

TABLE 2-5.1
GASKET MATERIALS AND CONTACT FACINGS[1] (CONT'D)
Gasket Factors m for Operating Conditions and Minimum Design Seating Stress y

Gasket Material	Gasket Factor m	Min. Design Seating Stress y, psi (MPa)	Sketches	Facing Sketch and Column in Table 2-5.2
Corrugated metal:				
Soft aluminum	2.75	3,700 (26)		
Soft copper or brass	3.00	4,500 (31)		
Iron or soft steel	3.25	5,500 (38)		(1a),(1b),(1c),(1d); Column II
Monel or 4%–6% chrome	3.50	6,500 (45)		
Stainless steels and nickel-base alloys	3.75	7,600 (52)		
Flat metal, jacketed asbestos filled:				
Soft aluminum	3.25	5,500 (38)		
Soft copper or brass	3.50	6,500 (45)		
Iron or soft steel	3.75	7,600 (52)		(1a),(1b),(1c),[2] (1d)[2];(2)[2]; Column II
Monel	3.50	8,000 (55)		
4%–6% chrome	3.75	9,000 (62)		
Stainless steels and nickel-base alloys	3.75	9,000 (62)		
Grooved metal:				
Soft aluminum	3.25	5,500 (38)		
Soft copper or brass	3.50	6,500 (45)		
Iron or soft metal	3.75	7,600 (52)		(1a),(1b),(1c),(1d), (2),(3); Column II
Monel or 4%–6% chrome	3.75	9,000 (62)		
Stainless steels and nickel-base alloys	4.25	10,100 (70)		
Solid flat metal:				
Soft aluminum	4.00	8,800 (61)		
Soft copper or brass	4.75	13,000 (90)		(1a),(1b),(1c),(1d),
Iron or soft steel	5.50	18,000 (124)		(2),(3),(4),(5);
Monel or 4%–6% chrome	6.00	21,800 (150)		Column I
Stainless steels and nickel-base alloys	6.50	26,000 (179)		
Ring joint:				
Iron or soft steel	5.50	18,000 (124)		
Monel or 4%–6% chrome	6.00	21,800 (150)		(6); Column I
Stainless steels and nickel-base alloys	6.50	26,000 (179)		

NOTES:
(1) This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b given in Table 2-5.2. The design values and other details given in this Table are suggested only and are not mandatory.
(2) The surface of a gasket having a lap should not be against the nubbin.

TABLE 2-5.2
EFFECTIVE GASKET WIDTH[2]

Facing Sketch (Exaggerated)	Basic Gasket Seating Width b_o Column I	Basic Gasket Seating Width b_o Column II
(1a) 	$\frac{N}{2}$	$\frac{N}{2}$
(1b)  See Note (1)		
(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) $w \le N$		
(2) $^1/_{64}$ in. nubbin  $w \le N/2$	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
(3) $^1/_{64}$ in. 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) 	$\frac{w}{8}$	. . .

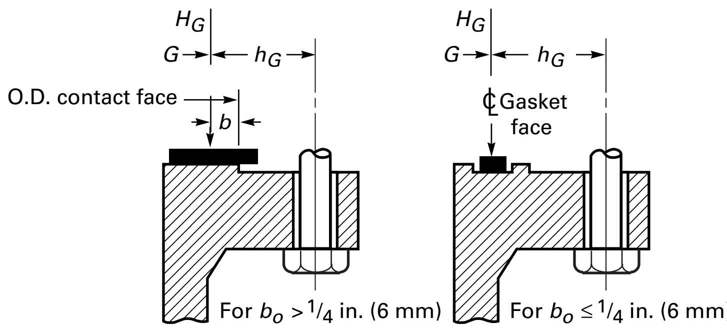
(continued)

TABLE 2-5.2
EFFECTIVE GASKET WIDTH[2] (CONT'D)

Effective Gasket Seating Width, b

$b = b_o$, when $b_o \leq 1/4$ in. (6 mm); $b = 0.5\sqrt{b_o}$, when $b_o > 1/4$ in. (6 mm)

Location of Gasket Load Reaction



NOTES:
(1) Where serrations do not exceed 1/64 in. (0.4 mm) depth and 1/32 in. (0.8 mm) width spacing, sketches (1b) and (1d) shall be used.
(2) The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

TABLE 2-6
MOMENT ARMS FOR FLANGE LOADS UNDER OPERATING CONDITIONS

	h_D	h_T	h_G
Integral type flanges [see Fig. 2–4 sketches (5), (6), (6a), (6b), and (7)]; and optional type flanges calculated as integral type [see Fig. 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 Fig. 2-4 sketches (2), (2a), (3), (3a), (4), and (4a)]; and optional type flanges calculated as loose type [see Fig. 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 Fig. 2-4 sketches (1) and (1a)]	$\frac{C - B}{2}$	$\frac{C - G}{2}$	$\frac{C - G}{2}$

individually calculated for each flange and gasket, and that value shall be used for both flanges.

The need for providing sufficient bolt load to seat the gasket or joint-contact surfaces in accordance with Formula (2) will prevail on many low-pressure designs and with facings and materials that require a high seating load, and where the bolt load computed by Formula (1) for the 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 Formula (2) governs, flange proportions will be a function of the bolting instead of internal pressure.

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

(a) The required bolt load for the operating conditions W_m shall be sufficient to resist the hydrostatic end force H exerted by the maximum allowable working pressure 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 must 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.

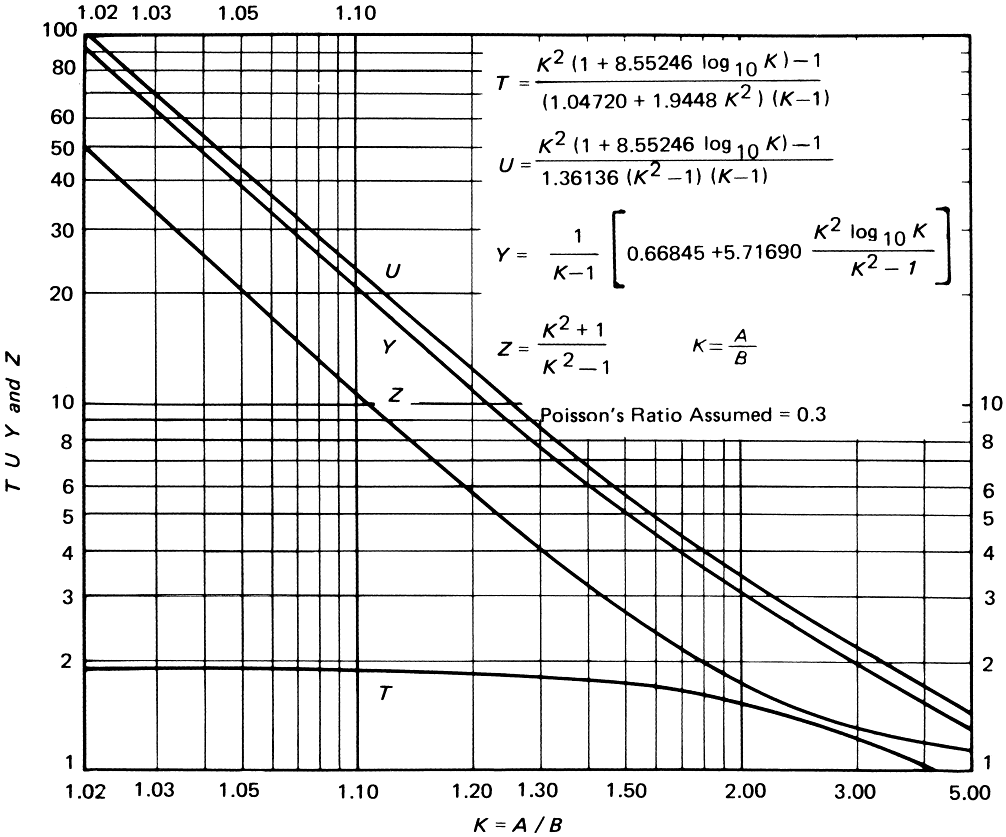


FIG. 2-7.1 VALUES OF T, U, Y, AND Z
(Terms Involving K)

(d) Total Required and Actual Bolt Areas, A_m and A_b. 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} where $A_{m1} = W_{m1}/S_b$ and $A_{m2} = W_{m2}/S_a$. A selection of bolts to be used shall be made such that the actual total cross-sectional area of bolts A_b will not be less than A_m.

(e) Flange Design Bolt Load W. The bolt loads used in the design of the flange shall be the values obtained from Formulas (3) and (4). For operating conditions,

$$W = W_{m1} \qquad (3)$$

For gasket seating,

$$W = \frac{(A_m + A_b)S_a}{2} \qquad (4)^{**}$$

S_a used in Formula (4) shall be not less than that tabulated in the stress tables (see UG-23). In addition to the minimum requirements for safety, Formula (4) 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 (see Note 2).

NOTE 2: Where additional safety against abuse is desired, or where it is necessary that the flange be suitable to withstand the full available bolt load A_bS_a, the flange may be designed on the basis of this latter quantity.

2-6 FLANGE MOMENTS

In the calculation of flange stress, the moment of a load acting on the flange is the product of the load

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

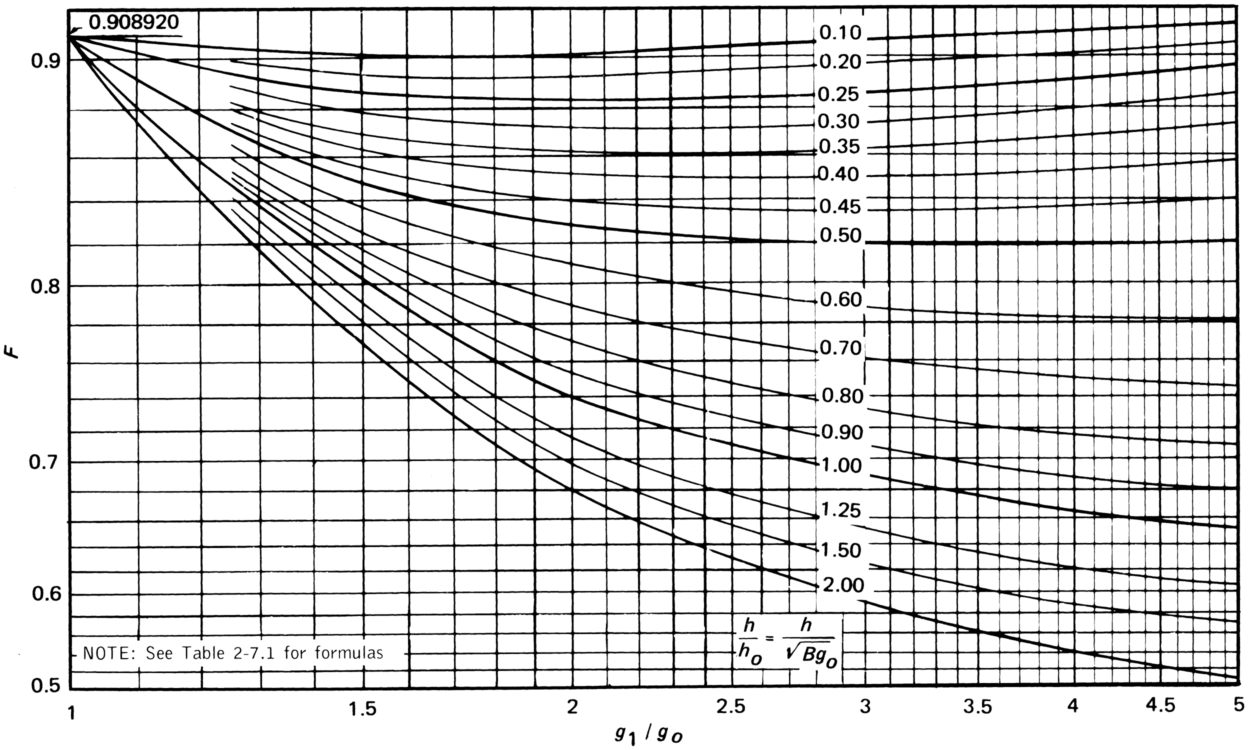


FIG. 2-7.2 VALUES OF F
(Integral Flange Factors)

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 (see Fig. 2-4). No consideration shall be given to any possible reduction in moment arm due to cupping of the flanges or due to inward shifting of the line of action of the bolts as a result thereof.

For the operating conditions, the total flange moment M_o is the sum of the three individual moments M_D, M_T, and M_G, as defined in 2-3 and based on the flange design load of Formula (3) with moment arms as given in Table 2-6.

For gasket seating, the total flange moment M_o is based on the flange design bolt load of Formula (4), which is opposed only by the gasket load, in which case

$$M_o = W \frac{(C - G)}{2} \qquad (5)$$

2-7 CALCULATION 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 formulas:

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

Longitudinal hub stress

$$S_H = \frac{fM_o}{Lg_1{}^2B} \qquad (6)^{**}$$

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

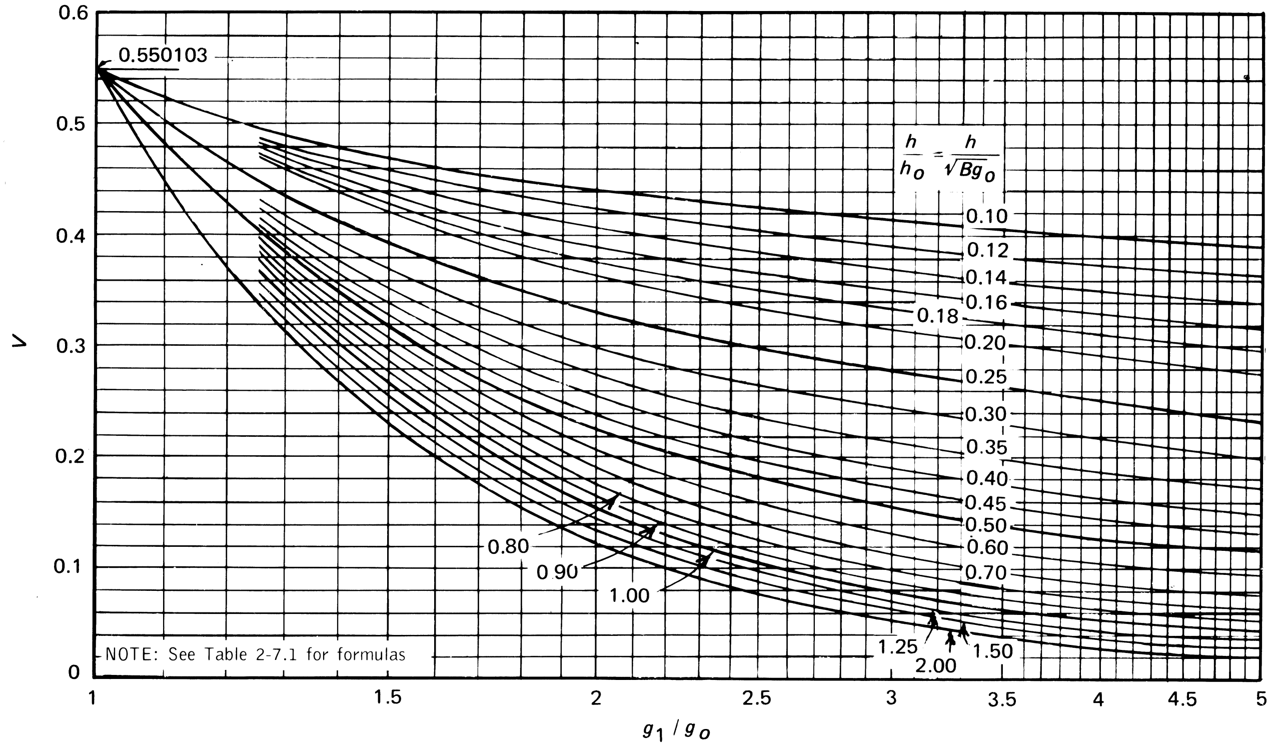


FIG. 2-7.3 VALUES OF V
(Integral Flange Factors)

Radial flange stress

$$S_R = \frac{(1.33te + 1)M_o}{Lt^2B} \tag{7)**}$$

Tangential flange stress

$$S_T = \frac{YM_o}{t^2B} - ZS_R \tag{8)**}$$

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

$$S_T = \frac{YM_o}{t^2B} \tag{9)**}$$

$$S_R = 0 \qquad S_H = 0$$

2-8 ALLOWABLE FLANGE DESIGN STRESSES

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

(1) longitudinal hub stress S_H not greater than S_f for cast iron[1] and, except as otherwise limited by (1)(a) and (1)(b) below, not greater than 1.5 S_f for materials other than cast iron:

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

[1] When the flange material is cast iron, particular care should be taken when tightening the bolts to avoid excessive stress that may break the flange. The longitudinal hub stress has been limited to S_f in order to minimize any cracking of flanges. An attempt should be made to apply no greater torque than is needed to assure tightness during the hydrostatic test.

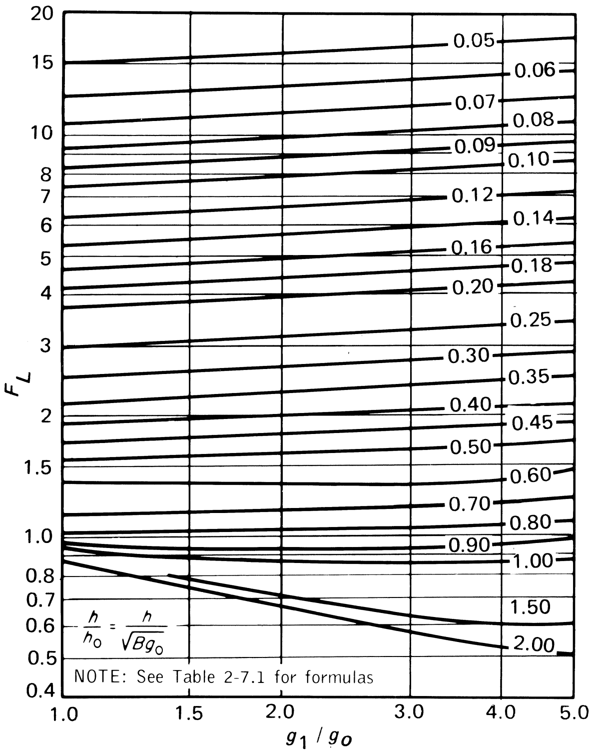


FIG. 2-7.4 VALUES OF F_L
(Loose Hub Flange Factors)

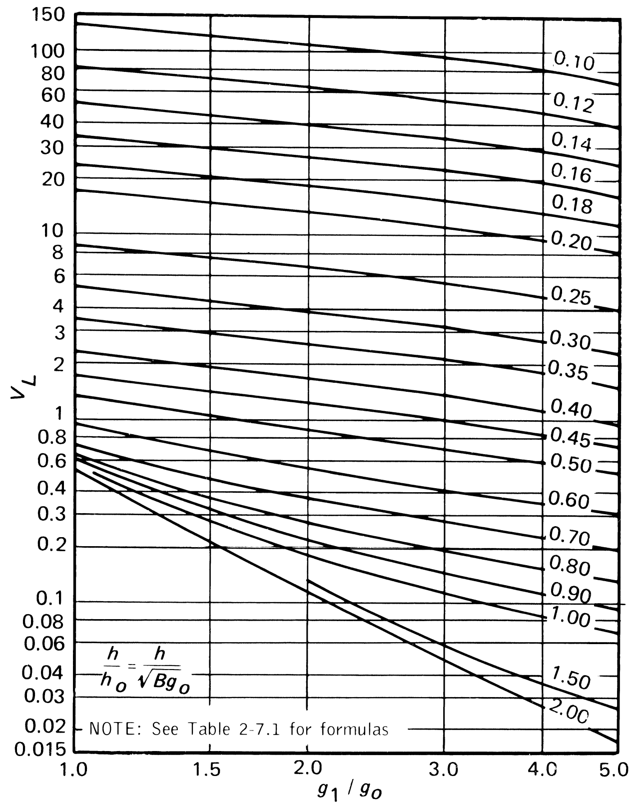


FIG. 2-7.5 VALUES OF V_L
(Loose Hub Flange Factors)

(a) 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 [Fig. 2-4 sketches (8), (8a), (9), (9a), (10), (10a), and (11)], also integral type [Fig. 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 [Fig. 2-4 sketches (6), (6a), and (6b)].

(2) radial flange stress S_R not greater than S_f;

(3) tangential flange stress S_T not greater than S_f;

(4) also $(S_H + S_R)/2$ not greater than S_f and $(S_H + S_T)/2$ not greater than S_f.

(b) For hub flanges attached as shown in Fig. 2-4 sketches (2), (2a), (3), (3a), (4), and (4a), 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 Fig. 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 Fig. 2-4 sketches (3), (3a), (4), (4a), (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_{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[2]

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

[2] Loose flanges of the type shown in Fig. 2-4 sketch (1) are of the split design when it is necessary to install them after heat treatment of a stainless steel vessel, or when for any reason it is desired to have them completely removable from the nozzle neck or vessel.

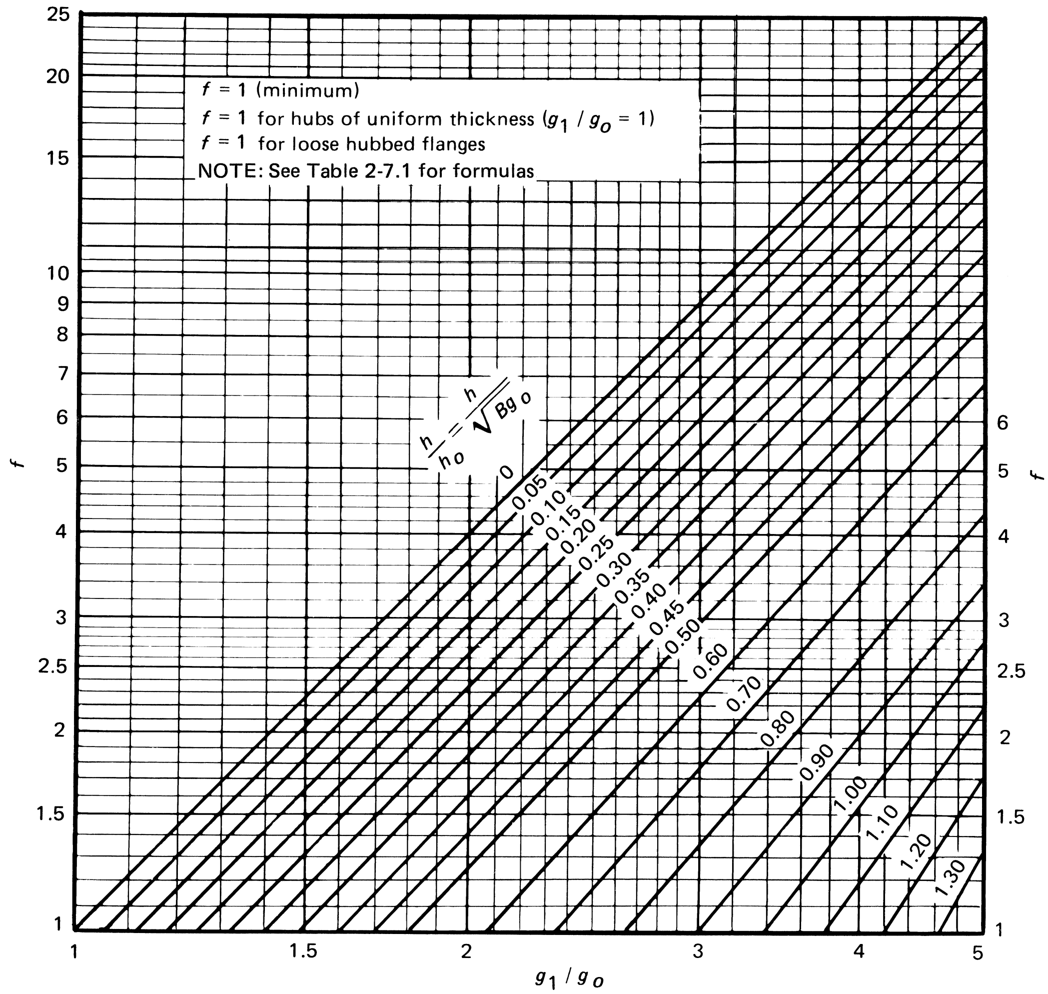


FIG. 2-7.6 VALUES OF f
(Hub Stress Correction Factor)

(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)][3] shall be based on the formulas given in 2-7 for internal pressure except that for operating conditions:

[3] 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.

$$M_o = H_D(h_D - h_G) + H_T(h_T - h_G) \quad (10)^{**}$$

For gasket seating,

$$M_o = Wh_G \quad (11)$$

where

$$W = \frac{A_{m2} + A_b}{2} S_a \quad (11a)$$

$$H_D = 0.785B^2 P_e \quad (11b)$$

$$H_T = H - H_D \quad (11c)$$

$$H = 0.785G^2 P_e \quad (11d)^{**}$$

P_e = external design pressure, psi (kPa)

See 2-3 for definitions of other symbols. S_a used in Formula (11a) shall be not less than that tabulated in the stress tables (see UG-23).

(b) When flanges are subject at different times during operation to external or internal pressure, the design shall satisfy the external pressure design requirements given in (a) above and the internal pressure design requirements given elsewhere in this Appendix.

NOTE: The combined force of external pressure and bolt loading may plastically deform certain gaskets to result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized, consideration should be given to gasket and facing details so that excessive deformation of the gasket will not occur. Joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.

2-12 FLANGES WITH NUT-STOPS

(a) When flanges are designed per this Appendix, or are fabricated to the dimensions of ASME/ANSI B16.5 or other acceptable standards [see UG-44(a)], except that the dimension R is decreased to provide a nut-stop, the fillet radius relief shall be as shown in Fig. 2-4 sketches (12) and (12a) except that:

(1) for flanges designed to this Appendix, the dimension g_1 must be the lesser of $2t$ (t from UG-27) or $4r$, but in no case less than $\frac{1}{2}$ in. (13 mm), where

r = the radius of the undercut

(2) for ASME/ANSI B16.5 or other standard flanges, the dimension of the hub g_o shall be increased as necessary to provide a nut-stop.

2-13 REVERSE FLANGES 01

(a) Flanges with the configuration as indicated in Fig. 2-13.1 shall be designed as integral reverse flanges and those in Fig. 2-13.2 shall be designed as loose ring type reverse flanges. These flanges shall be designed in conformance with the rules in 2-3 through 2-8, but with the modifications as described in the following. Mandatory use of these rules is limited to $K \leq 2$. When $K > 2$, results become increasingly conservative and U-2(g) may be used.

(1) Integral Type Reverse Flange. The shell-to-flange attachment of integral type reverse flanges may be attached as shown in Fig. 2-4 sketches (5) through (11), as well as Fig. UW-13.2 sketches (a) and (b). The requirements of 2-4(a)(3) apply to Fig. 2-4 sketches (8) through (11) as well as Fig. UW-13.2 sketches (a) and (b).

(2) Loose Ring Type Reverse Flange. The shell-to-flange attachment of loose ring type reverse flanges may be attached as shown in Fig. 2-4 sketches (3a), (4a), (8), (9), (10), and (11) as well as Fig. UW-13.2 sketches (c) and (d). When Fig. UW-13.2 sketches (c) and (d) are used, the maximum wall thickness of the shell shall not exceed $\frac{3}{8}$ in. (10 mm), and the maximum design metal temperature shall not exceed 650°F (343°C).

The symbols and definitions in this paragraph pertain specifically to reverse flanges. Except as noted in (b) below, the symbols used in the equations of this paragraph are defined in 2-3.

The formulas for S_H, S_R, and S_{T1} correspond, respectively, to Formulas (6), (7), and (8) in 2-7, in direction, but are located at the flange *outside* diameter. The sole stress at the flange inside diameter is a tangential stress and is given by the formula for S_{T2}.

(b) Notation

B = inside diameter of shell, in. (mm)
B' = inside diameter of reverse flange, in. (mm)
$d_r = U_r h_{or} g_o^2 / V$ **
$e_r = F / h_{or}$
F = factor (use h_{or} for h_o in Fig. 2-7.2)
f = factor (use h_{or} for h_o in Fig. 2-7.6)
H = total hydrostatic end force on attached component, lb (kN)
$= 0.785G^2 P$ **

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

TABLE 2-7.1
FLANGE FACTORS IN FORMULA FORM

Integral Flange

Factor F per Fig. 2-7.2 is then solved by

$$F = -\frac{E_6}{\left(\frac{C}{2.73}\right)^{1/4}\frac{(1+A)^3}{C}}$$

Factor V per Fig. 2-7.3 is then solved by

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

Factor f per Fig. 2-7.6 is then solved by

$$f = C_{36}/(1+A)$$

The values used in the above equations are solved using Eqs. (1) through (45) below based on the values g_1, g_o, h, and h_o as defined by 2-3. When $g_i = g_o$, $F = 0.908920$, $V = 0.550103$, and $f = 1$; thus Eqs. (1) through (45) need not be solved.

Loose Hub Flange

Factor F_L per Fig. 2-7.4 is solved by

$$F_L = -\frac{C_{18}\left(\frac{1}{2}+\frac{A}{6}\right) + C_{21}\left(\frac{1}{4}+\frac{11A}{84}\right) + C_{24}\left(\frac{1}{70}+\frac{A}{105}\right) - \left(\frac{1}{40}+\frac{A}{72}\right)}{\left(\frac{C}{2.73}\right)^{1/4}\frac{(1+A)^3}{C}}$$

Factor V_L per Fig. 2-7.5 is solved by

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

Factor f per Fig. 2-7.6 is set equal to 1.

$$f = 1$$

The values used in the above equations are solved using Eqs. (1) through (5), (7), (9), (10), (12), (14), (16), (18), (20), (23), and (26) below based on the values of g_1, g_o, h, and h_o as defined by 2-3.

Equations

(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) $C_{14} = 197/415{,}800 + 103A/332{,}640 - (1/35 + 6A/35 + 17A^2/70 + A^3/10)/C$

TABLE 2-7.1
FLANGE FACTORS IN FORMULA FORM (CONT'D)

(17) $C_{15} = 233/831{,}600 + 97A/554{,}400 + (1/35 + 3A/35 + A^2/14 + 2A^3/105)/C$

(18) $C_{16} = C_1C_7C_{12} + C_2C_8C_3 + C_3C_8C_2 - (C_3^2C_7 + C_8^2C_1 + C_2^2C_{12})$

(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) $E_6 = E_5 - C_{36}\,(7/120 + A/36 + 3A/C) - 1/40 - A/72 - C_{37}(1/60 + A/120 + 1/C)$

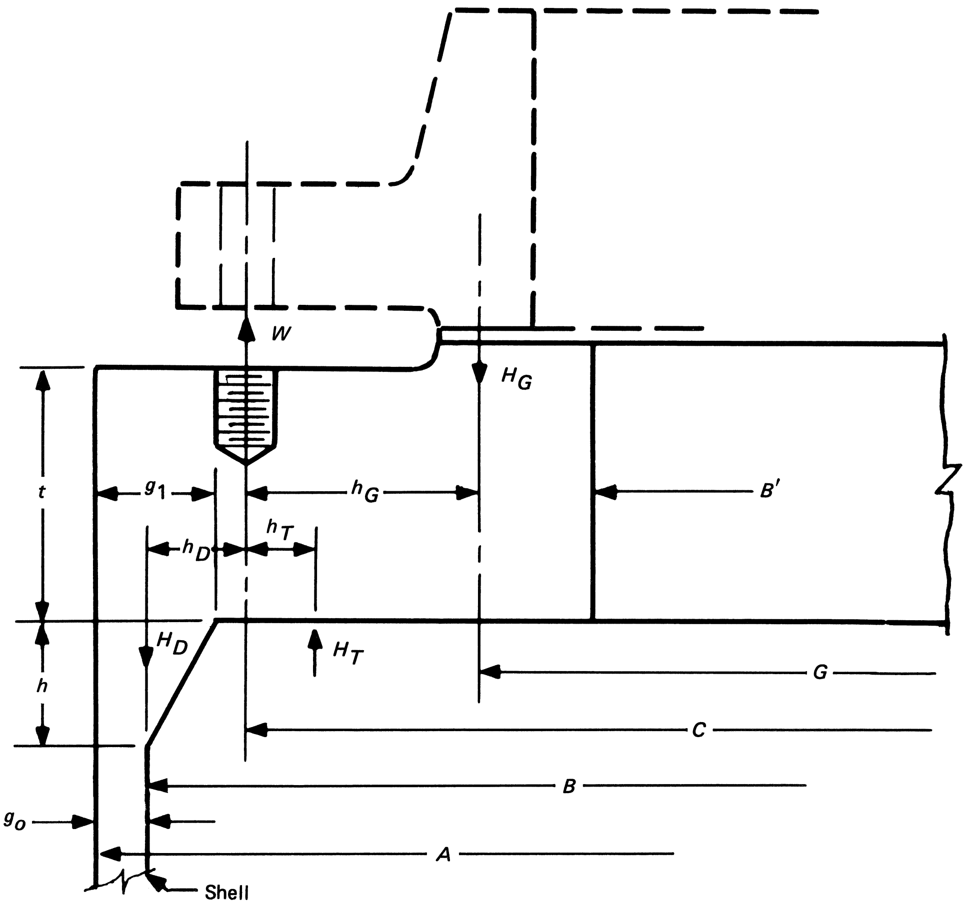


FIG. 2-13.1 REVERSE FLANGE

H_D = hydrostatic end force on area inside of flange, lb (kN)
$= 0.785B^2P$ **

H_T = difference between hydrostatic end force on attached component and hydrostatic end force on area inside of flange, lb (kN)
$= H - H_D$

h_D = radial distance from the bolt circle to the circle on which H_D acts, in. (mm)
$= (C + g_1 - 2g_o - B)/2$ for integral type reverse flanges
$= (C - B)/2$ for loose ring type reverse flanges

h_{or} = factor, in. (mm)

$$= \sqrt{Ag_o}$$

h_T = radial distance from the bolt circle, to the circle on which H_T acts, in. (mm)

$$= \tfrac{1}{2}\left(C - \frac{B + G}{2}\right)$$

K = ratio of outside diameter of flange to inside diameter of flange
$= A/B'$

L_r = factor

$$= \frac{te_r + 1}{T_r} + \frac{t^3}{d_r}$$

M_o = total moment acting on the flange, for the operating conditions or gasket seating as may apply, in.-lb (kN•m)
= algebraic sum of M_D, M_T, and M_G. Values of load H_T and moment arm h_D are negative; value

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

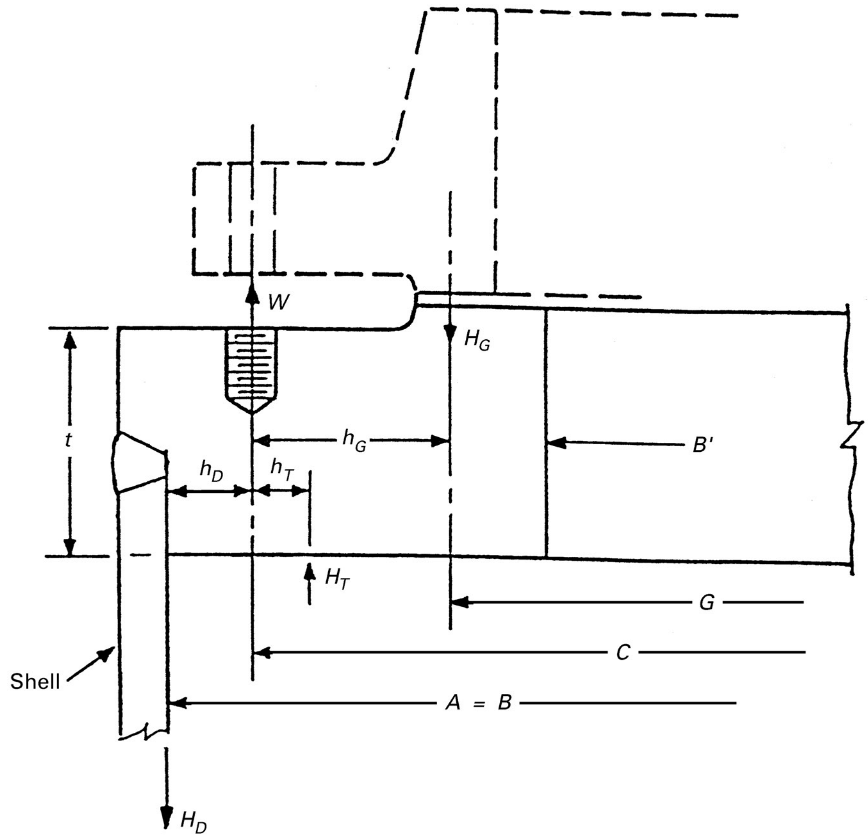


FIG. 2-13.2 LOOSE RING TYPE REVERSE FLANGE

of moment arm h_T may be positive as in Fig. 2-13, or negative. If M_o is negative, use its absolute value in calculating stresses to obtain positive stresses for comparison with allowable stresses.

$$T_r = \left(\frac{Z + 0.3}{Z - 0.3}\right)\alpha_r T$$

$U_r = \alpha_r U$

$V =$ factor (use h_{or} for h_o in Fig. 2-7.3)

$Y_r = \alpha_r Y$

$$\alpha_r = \left[1 + \frac{0.668\ (K + 1)}{Y}\right] / K^2$$

(c) For Integral Type Reverse Flanges

(1) Stresses at the Outside Diameter

$$S_H = fM_o / L_r {g_1}^2 B'$$

$$S_R = (1.33te_r + 1)M_o / L_r t^2 B'$$

$$S_{T1} = (Y_r M_o / t^2 B') - ZS_R(0.67te_r + 1)/(1.33te_r + 1)$$

(2) Stress at Inside Diameter B′

$$S_{T2} = (M_o / t^2 B')\left[Y - \frac{2K^2(1 + \frac{2}{3}te_r)}{(K^2 - 1)L_r}\right]$$

(d) For Loose Ring Type Reverse Flanges

$$S_T = YM_o/t^2B'$$

$$S_R = 0 \qquad S_H = 0$$