3
Flange Design

Introduction 140
Procedure 3-1: Design of Flanges 148
Procedure 3-2: Design of Spherically Dished Covers 165
Procedure 3-3: Design of Blind Flanges with Openings 167
Procedure 3-4: Bolt Torque Required for Sealing Flanges 169
Procedure 3-5: Design of Studding Outlets 172
Procedure 3-6: Reinforcement for Studding Outlets 175
Procedure 3-7: Studding Flanges 176
Procedure 3-8: Design of Elliptical, Internal Manways 181
Procedure 3-9: Through Nozzles 182
References 183

Introduction

Standard flanges should be used wherever possible. The cost of designing and fabricating a custom flange is expensive and should be a last resort. If a custom flange is required, and there is no alternative, then the procedures included in this chapter can be utilized for designing a custom flange. The ASME Code accepts the standard pressure-temperature ratings of ASME B16.5 for flanges, classes 150 to 2500, ½" to 24". For larger diameter flanges use ASME B16.47.

Flange Standards

1. ASME/ANSI B16.5 (1/2" to 24" Class 150 to 2500)
2. ASME/ANSI B16.47
   - Series A: Replaced the Old API 605 (26" to 60", Class 75 to Class 900)
   - Series B: Based on MSS SP 44 Steel Pipeline flanges designed to ASME Section VIII Large flanges (26" to 60", Class 150 to Class 900)
3. TEMA Flanges, Class R
4. ASME/ANSI B16.1 Cast Iron Pipe Flanges
5. ASME/ANSI B16.24 Bronze Flanges and Flanged Fittings
6. AWWA C207 Steel Pipe Flanges for Water Works Services
7. BS 3293 British Standards – Specification for Carbon Steel Pipe Flanges (over 24" nominal size)
8. API Specification 6A (2,000 to 20,000 psi for wellhead equipment)
9. MSS SP-51 Corrosion Resistant Cast Flanges and Flanged Fittings
10. MSS SP-65 High Pressure Chemical Industry Flanges

Flange Design

In general, special designs as outlined in this procedure are done for large or high-pressure designs. Flanges in this category will be governed by one of two conditions:

1. Gasket seating force, W_{m2}
2. Hydrostatic end force, H

For high-pressure flanges, typically the hydrostatic end force, H, will govern. For low-pressure flanges, the gasket seating force will govern. Therefore the strategy for approaching the design of these flanges will vary. The strategy is as follows:

- *For low-pressure flanges*
  a. Minimize the gasket width to reduce the force necessary to seat the gasket.
  b. Use a larger number of smaller diameter bolts to minimize the bolt circle diameter and thus reduce the moment arm which governs the flange thickness.
  c. Utilize hubless flanges (either lap joint or plate flanges) to minimize the cost of forgings.
- *For high-pressure flanges*
  High-pressure flanges require a large bolt area to counteract the large hydrostatic end force. Large bolts, in turn, increase the bolt circle with a corresponding increase in the moment arm. Thicker flanges and large hubs are necessary to distribute the bolt loads. Seek a balance between the quantity and size of bolts, bolt spacing, and bolt circle diameter.

Design Strategy

Step 1: Determine the number and size of bolts required. As a rule of thumb, start with a number of bolts equal to the nominal size of the bore in inches, rounded to the nearest multiple of four. First, calculate W_{m1} or W_{m2}. A_m is equal to the larger of W_{m1} or W_{m2} divided by S_a. The quantity of bolts required is:

$$n = A_m/R_a$$

To find the size of bolt for a given quantity:

$$R_a = A_m/n$$

With these two equations a variety of combinations can be determined.

Step 2: Determine the bolt circle diameter for the selected bolt size.

$$C = B + 2g_1 + 2R$$

The flange O.D. may now be established.

$$A = C + 2E$$

Step 3: Check the minimum bolt spacing (not an ASME requirement). Compare with the value of B_s in Table 3-3.

$$B_s = C/n$$

Note: Dimensions R_a, R, E, and B_s are from Table 3-3.

Step 4: After all of the preliminary dimensions and details are selected, proceed with the detailed analysis of the flange by calculating the balance of forces, moments, and stresses in the appropriate design form.

Bolt Loads in Bolted Flange Connections Bolted flange connections are designed to balance the gasket reaction with the bolt loads. Since these two loads are applied at different locations on the flange, a moment is developed in the flange. The flange geometry is proportioned to accommodate this internal moment. Flange design is the process of determining the load applied at the gasket due to seating or operating conditions and then finding the corresponding bolt load to counteract that load.

The ASME Code defines three separate distinct bolt loads used in the design of flanges. The three separate bolt loads are as follows;

1. W Flange design bolt load for operating or gasket seating as may apply
2. W_{m1} Required bolt load for operating condition
3. W_{m2} Required bolt load for gasket seating

The values of W_{m1} and W_{m2} are used to determine the bolt area required for the operating and gasket seating conditions respectively. Once the bolt area required is determined, and the actual bolt size and quantity selected, then the actual bolt load W, can be determined. Thus W is defined as the "Design Bolt Load" since it is not based on the theoretical required loads, but on the actual quantity and sizes of bolts used.

Ordinarily, the bolt area is selected to correspond closely with the minimum required. If excess bolting is provided, as is common practice with low pressure flanges, some recognition of this excess bolting should be made in the flange design to guard against excessive flange stress being developed when the bolts are tightened and to provide reasonable protection against abuse due to over tightening.

Factor "m" Gasket factor m is a dimensionless constant, also referred to as the maintenance factor. It has been found that the pressure on gasket faces required to avoid leakage must bear a minimum ratio, "m", to the hydrostatic pressure expected to be confined. The value m depends on the type of gasket material and the initial pressure to which the gasket is installed. It also depends on the type of flange facing used, but design methods usually account for this by choosing an effective gasket width. The term 'm' is actually a "pressure ratio". The ASME Code has used various pressure ratios, defined as the follows;

m = effective ratio @ mean gasket diameter
r = contact pressure ratio @ outside gasket diameter
m_a = absolute pressure ratio @ inside gasket diameter

Actual pressure ratios can be defined by the following equation;

$$m = \left(W - P^*\pi/4^*G^2\right)/\left(P^*\pi/4^*\left(OD^2 - ID^2\right)\right)$$

Equations for r or m_a can be accomplished by substituting OD or ID for G in the above equation. Relationships between the terms can be defined as follows;

$$m_a < m + 1/2$$

$$m_a = r + 1$$

The terms m, r and m_a are based on the same fundamental assumption; that some multiple of the confined *unit pressure* must be kept as a *unit pressure* on the gasket surfaces. This assumption is probably valid for small pressure ratios and soft gaskets with fluid-like characteristics. In this manner the pressure ratio is purported to be an indicator of leakage prediction. However it is a poor indicator of leakage prediction for hard or metal gaskets where the gasket material does not flow.

When the gasket width is increased, the m value is smaller than required for a narrow gasket from the equation. This would seem to indicate a dimensional constant involving load per linear inch of gasket, ratioed to the pressure.

The equation for m above was the basis for the development of the values given in the ASME Code. We do not calculate the value of m but utilize the values from the table.

Gasket Facing and Selection

The gasket facing and type correspond to the service conditions, fluid or gas handled, pressure, temperature, thermal shock, cyclic operation, and the gasket selection. The greater the hazard, the more care that should be invested in the decisions regarding gasket selection and facing details.

Facings which confine the gasket, such as male and female, tongue and groove, and ring joint offer greater security against blowouts. Male and female and tongue and groove have the disadvantage that mating flanges are not alike. These facings, which confine the gasket, are known as enclosed gaskets and are required for certain services, such as TEMA Class "R."

For tongue and groove flanges, the tongue is more likely to be damaged than the groove; therefore, from a maintenance standpoint, there is an advantage in placing the tongue on the part which can be transported for servicing, i.e., blind flanges, manway heads, etc. If the assembly of these joints is horizontal then there will be less difficulty if the groove is placed in the lower side of the joint. The gasket width should be made equal to the width of the tongue. Gaskets for these joints are typically metal or metal jacketed.

Table 3-1
Types of gaskets and surface finish

Type of Gasket	Surface Finish	Notes
1 Ring Type (Flat)	Concentric Serrated or stock finish	Thickness of gasket should be at least 3 times the depth of the grooves. Previously most commonly known as "compressed asbestos". Typical thickness' are 1/16" and 1/8".
2 Solid Metal a. Flat b. Profile c. Profile with Filler	 Concentric Serrated Very Smooth Smooth	Brute force seals. Gaskets made of flat metal. Relatively thin compared to width. Metal selection dependent on corrosion and temperature.
3 Spiral Wound (Note 1) a. Inner Ring Only (Style RIR) b. Outer Ring Only (Style CG) c. Inner and Outer Ring (Style CGI) d. No Inner or Outer Ring (Style R)	Smooth or Serrated; 125 to 250 AARH	Consists of preformed "V" shaped strip of metal which is wound into a spiral form with a compressible filler.
4 Metal Jacketed a. Flat Metal jacketed b. Corrugated Metal Jacketed	Very Smooth	Many variations. Consists of soft, compressible filler, either partially or wholly enclosed in a metal jacket.
5 Corrugated Metal a. With Filler b. Without Filler	Smooth	Line contact seals consisting of thin metal, corrugated or embossed. Can be coated with filler.
6 Elastomers	Concentric Serrated	Rubber, Fiber, Teflon, etc.
7 Ring Type Joint (RTJ) a. Hex b. Oval	Very Smooth	Metal gaskets sealed by line of contact and wedging action
8 Special a. Delta Ring b. Lens Ring c. Double Cone d. Bridgman e. O-Ring (Metal)	Very Smooth	Metal gasket sealed by line of contact and/or wedging action.

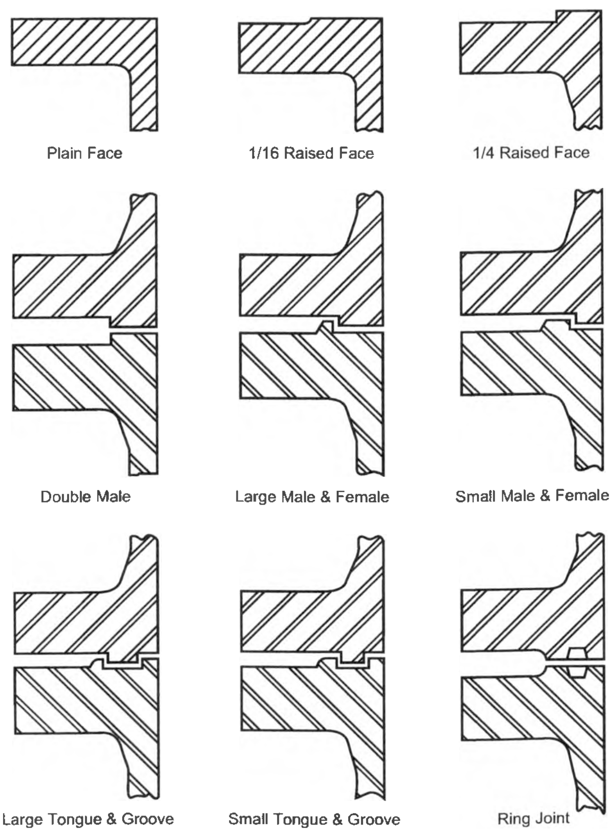
Notes:

1. ASME B16.20 requires solid metal inner rings for following;
   a. 900#: 24" & larger
   b. 1500#: 12" & larger
   c. 2500#: 4" & larger
   d. Vacuum service
   e. Surface finish > 125 AARH

Table 3-2
Recommendations for gasket/facing selection

Service	Design Conditions: Pressure Rating	Design Conditions: Temperature Range, F	Facing	Gasket Type
Oil or Hydrogen	150 to 600	1000 & Below	RF	Corrugated, Double Jacketed or Spiral Wound
		Above 1000	RTJ	Oval Ring
	900 and Higher	Any	RTJ	Oval Ring
Steam and Boiler Feedwater	900 or Lower	1000 & Below	RF	Corrugated, Double Jacketed or Spiral Wound
	1500	Any	RTJ	Oval Ring
Air	300 or Lower	750 & Below	RF	1/16" Flexible Graphite
		>750–1000	RF	Corrugated, Double Jacketed or Spiral Wound
		>1000–1200	RF	Spiral Wound
Water	150	250 & Below	RF	1/16" Flexible Graphite
		>250	RF	Corrugated, Double Jacketed or Spiral Wound
	300-900	Any	RF	Corrugated, Double Jacketed or Spiral Wound
	1500	Any	RTJ	Oval Ring
Fluid Catalyst	300 or Lower	1000 & Below	RF	Corrugated, Double Jacketed or Spiral Wound
		>1000–1200	RF	Spiral Wound
Toxic Fluids including Acids & Caustics	300 or Lower	750 & Lower	RF	Corrugated, Double Jacketed or Spiral Wound
Refrigerants and Refrigerated Hydrocarbons	Any	Below (–) 50	RF	Spiral Wound, Teflon Filled
		(–) 50 to 400	RF	Corrugated, Double Jacketed or Spiral Wound

Types of Contact Faces for Flanges



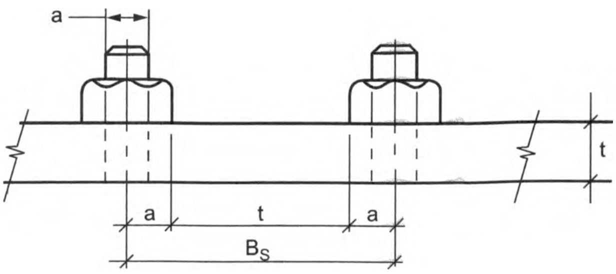
Gasket Contact Surface Finishes

The following is a list of typical surface finishes for flange faces;

1. Stock Finish: 250 to 500 AARH: This finish is produced with a continuous spiral groove generated with a round nosed tool. This finish is suitable for all ordinary service conditions and is the most widely used gasket surface finish
2. Spiral Serrated: 125 to 250 AARH: This finish is produced with a continuous spiral groove but utilizes a 90° included angle "V" tool. The groove is 1/64" deep and the feed is 1/32" for all sizes.
3. Concentric Serrated: 125 to 250 AARH: This finish is produced with concentric grooves using the same tools and parameters as the "spiral serrated" finish.
4. Smooth Finish: 63 to 125 AARH: This finish can be generated with a variety of tooling, but shows no tool marks apparent to the naked eye.
5. Cold Water Finish: 32 to 63 AARH: This finish is very smooth and has the appearance of a ground finish. It is mirrorlike in appearance.
6. Flat Face: Stock or Serrated
7. 1/16" Raised Face: Stock or Serrated
8. ¼" Raised Face: Stock or Serrated

9. Male & Female: Smooth
10. Tongue & Groove: Smooth
11. Side Wall of Ring Joint: 63 AARH

Bolt Spacing – Maximum and Minimum



Maximum Bolt Spacing: B_s Max $= 2a + t$
Minimum Bolt Spacing: B_s Min $= 2$ to $2.5 \times a$ (See Note 2)
Actual Bolt Spacing: $B_s = (\pi C)/n$
C = Bolt Circle Diameter, in
n = Quantity of Bolts (Multiple of 4)
a = Bolt/Stud Diameter, in
t = Flange Thickness, in

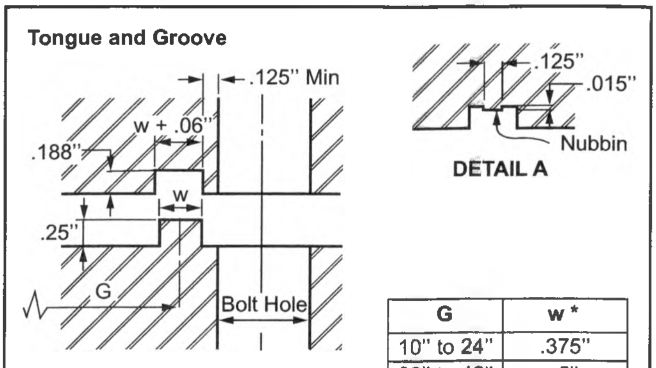
If bolt spacing exceeds B_s Max, multiply m_o and m_G by;

$\sqrt{B_s/(2a + t)}$

Notes

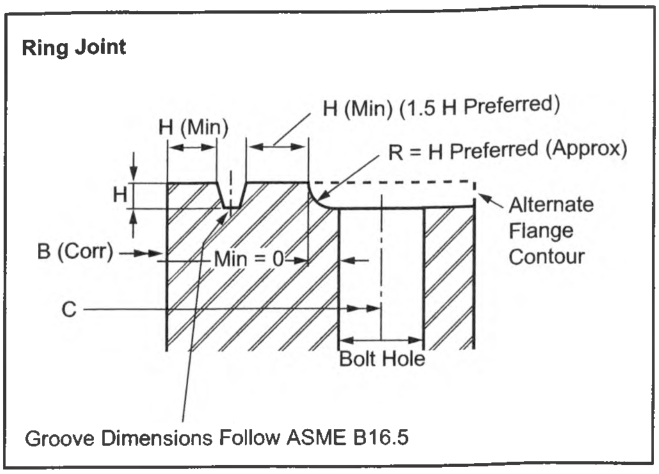
1. The requirements for bolt spacing is no longer an ASME Code requirement. However it is still good design practice.
2. B_s Min is based on wrench clearances. See Table 3-3.

Dimensions of Flange Faces



G	w*
10" to 24"	.375"
26" to 48"	.5"
48" to 72"	.625"

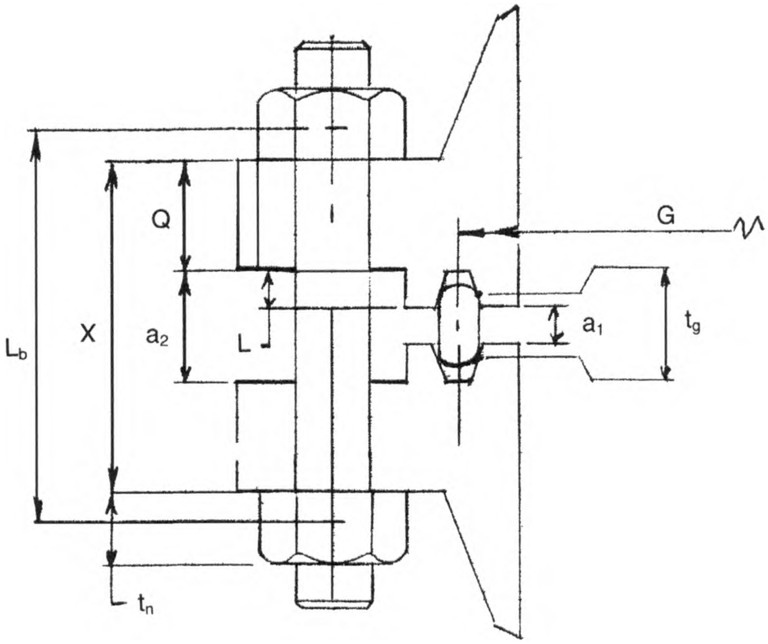
*Suggested Minimum



Notes

1. The procedures as outlined herein are based on Taylor Forge Bulletin No. 502, 7th Edition, entitled "Modern Flange Design." The forms and tables have been duplicated here for the user's convenience. The design forms are fast and accurate and are accepted throughout the industry. For additional information regarding flange design, please consult this excellent bulletin.
2. Flange calculations are done either as "integral" or "loose." A third classification, "optional," refers to flanges which do not fall into either of the foregoing categories and thus can be designed as either integral or loose. Definitions and examples of these categories are:
   - *Integral*—Hub and flange are one continuous structure either by manufacture or by full penetration welding. Some examples are:
     a. Welding neck flanges.
     b. Long weld neck flanges.
     c. Ring flanges attached with full penetration welds. Use design form "Type 1: Weld Neck Flange Design (Integral)," or "Type 3: Ring Flange Design."
   - *Loose*—Neither flange nor pipe has any attachment or is non-integral. It is assumed for purposes of analysis, that the hubs (if used) act independent of the pipe. Examples are:
     a. Slip-on flanges.
     b. Socket weld flanges.
     c. Lap joint flanges.
     d. Screwed flanges.
     e. Ring flanges attached without full penetration welds.

FLANGE DIMENSIONS FOR RTJ GASKET



Flange Size & Rating: ____________

L = ____________

Q = ____________

Bolting:

Qty, n = ____________

Size: ____________

Length: ____________

t_n = ____________

d = ____________

d_m = ____________

$A_b = .25 (\pi d^2 n)$

Gasket:

w = ____________

h = ____________

Dimensions:

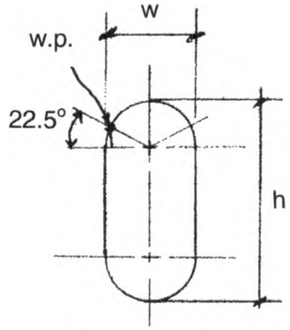
a_1 = ____________

$a_2 = a_1 + 2L$

$X = 2Q + a_2$

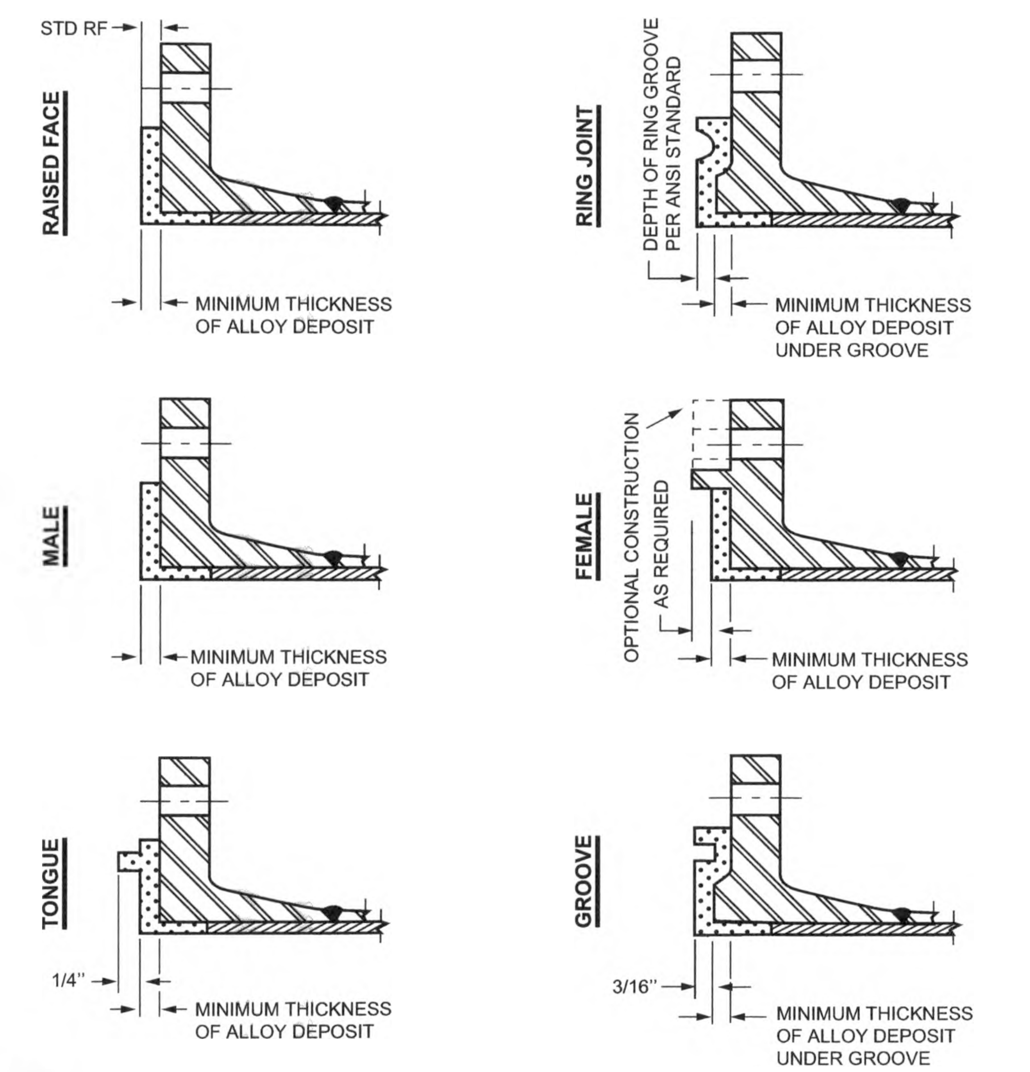
$L_b = X + t_n$

$t_g = (h - w) + 2 (\mathrm{Sin}\ 22.5) (.5\ w)$



RTJ Gasket

WELD OVERLAY DETAILS FOR FLANGE FACINGS



Notes:

1. Raised face is not included in minimum flange thickness for 400#, 600#, 900#, 1500# and 2500#. Therefore the raised face can be removed and replaced with weld overlay. ASME B16.5 - 1996, Para 6.3.13 states; "*with ring joint groove or female face, the minimum flange thickness shall be first provided and then sufficient metal added so that the bottom of the groove is in the same plane as the flange edge of a full thickness flange*". Therefore the weld overlay must be assumed as not contributing to the flange or, as an option, the flange must be checked as a solid alloy flange.

2. The raised face thickness is included in the minimum flange thickness for 150# and 300#. Therefore the raised face may not be removed without rerating the flange. Weld overlay must be added above the elevation of the raised face.

Notes *(Cont)*

Use design form "Type 2: Slip-On Flange Design (Loose)," or "Type 3: Ring Flange Design."

3. Hubs have no minimum limit for h and g_o, but values of $g_o < 1.5\,t_n$ and $h < g_o$ are not recommended. For slip-on flanges as a first trial, use $g_1 = 2$ times pipe wall thickness.
4. The values of T, Z, Y, and U in Table 3-6 have been computed based on Poisson's ratio of 0.3.

Table 3-3
Dimensional data for bolts and flanges

Bolt Size	Standard Thread: No. of Threads	Standard Thread: Root Area, R_a, In^2	8-Thread Series: No. of Threads	8-Thread Series: Root Area, R_a, In^2	Bolt Spacing: Minimum, B_S	Bolt Spacing: Preferred	Minimum Radial Distance, R	Edge Distance, E	Nut Dimension, width across flats. W_{AF}	Max Fillet Radius at base of hub, r	Nut Stop, R_1
1/2"	13	0.126	N.A.		1.25	3	0.813	0.625	0.875	0.25	0.5
5/8"	11	0.202			1.5	3	0.938	0.75	1.063	0.313	0.594
3/4"	10	0.302			1.75	3	1.125	0.813	1.25	0.375	0.688
7/8"	9	0.419			2.0625	3	1.25	0.938	1.438	0.375	0.781
1"	8	0.551	8	0.551	2.25	3	1.375	1.063	1.63	0.438	0.875
1-1/8"	7	0.693	8	0.728	2.5	3	1.5	1.125	1.813	0.438	0.969
1-1/4"	7	0.89	8	0.929	2.8125	3	1.75	1.25	2	0.563	1.063
1-3/8"	6	1.054	8	1.155	3.0625	3.25	1.875	1.375	2.188	0.563	1.156
1-1/2"	6	1.294	8	1.405	3.25	3.25	2	1.5	2.375	0.625	1.25
1-5/8"	5.5	1.515	8	1.68	3.5	3.5	2.125	1.625	2.563	0.625	1.344
1-3/4"	5	1.744	8	1.98	3.75	3.75	2.25	1.75	2.75	0.625	1.438
1-7/8"	5	2.049	8	2.304	4	4	2.375	1.875	2.938	0.625	1.531
2"	4.5	2.3	8	2.652	4	4.25	2.5	2	3.125	0.688	1.625
2-1/4"	4.5	3.02	8	3.423	4.5	4.75	2.75	2.25	3.5	0.688	1.813
2-1/2"	4	3.715	8	4.292	5	5.25	3.063	2.375	3.875	0.813	2
2-3/4"	4	4.618	8	5.259	5.5	5.75	3.375	2.625	4.25	0.875	2.188
3"	4	5.621	8	6.324	6	6.25	3.625	2.875	4.625	0.938	2.375
3-1/4"	N.A.		8	7.49	6.5	8.125	3.88	3.25	5	1	2.563
3-1/2"			8	8.75	7	8.75	4.125	3.5	5.375	1.063	2.75
3-3/4"			8	10.11	7.5	9.375	4.375	3.75	5.75	1.125	2.938
4"			8	11.57	8	10	4.625	4	6.125	1.25	3.125

Notes:

1. All dimensions are in inches except as shown.
2. Nut Stop dimension, R_1, refers to a modified dimension, R, such that the hub prevents the nut from spinning without use of a wrench.

Notes *(Cont)*

5. B is the I.D. of the flange and not the pipe I.D. For small-diameter flanges when B is less than 20 g_1, it is optional for the designer to substitute B_1 for B in Code formula for longitudinal hub stress, S_H. (See [1, Para. 2-3 of Section VIII, Division 1].)

6. In general, bolts should always be used in multiples of 4. For large-diameter flanges, use many smaller bolts on a tight bolt circle to reduce the flange thickness. Larger bolts require a large bolt circle, which greatly increases flange thickness.
7. If the bolt holes are slotted to allow for swing-away bolting, substitute the diameter of the circle tangent to the inner edges of the slots for dimension A and follow the appropriate design procedures.
8. Square and oval flanges with circular bores should be treated as "inscribed" circular flanges. Use a bolt circle passing through the center of the outermost bolt holes. The same applies for noncircular openings; however, the bolt spacing becomes more critical. The spacing factor can be less than required for circular flanges since the metal available in the corners tends to spread the bolt load and even out the moment.
9. Design flanges to withstand both pressure and external loads, use "equivalent" pressure P_e as follows:

$$P_e = \frac{16M}{\pi G^3} + \frac{4F}{\pi G^2} + P$$

where M = bending moment, in.-lb
F = radial load, lb

10. Hubs: Minimum 3:1 taper. Ideal taper is 4:1. This allows the hub stresses to dissipate over a longer distance.
11. Maximum bolt spacing is 2a + t. For minimum bolt spacing see Table 3–3
12. Gasket width check:

$$N_{min} = (A_b\, S_a)/(2\pi y G)$$

13. Unit stress on a gasket: This value should not exceed twice the gasket yield point when the bolts are stressed to their nominal value (20,000 psi for alloy bolts)

$$S_g = (A_b\, S_a)/.785\left[(d_o - .125)^2 - d_i^2\right]$$

14. Only hubless flanges can be machined from plate. See UW-13(f) and Appendix 2-2.
15. The design of flanges includes the following;
    a. Flange type, integral or loose
    b. Flange configuration, lap joint, weld neck, slip on, etc.
    c. Gasket selection (type, material, dimensions)
    d. Flange facing type (RF, FF, M&F, T&G, ring joint)
    e. Flange face finish (smooth, rough, serrated)
    f. Bolting
    g. Hub proportions
    h. Flange width
16. In welded construction the nozzle neck comprised of the vessel or pipe wall to which it is attached, is considered to act as the hub.
17. The yield point of gasket material is related to the "m" value. It is not a "yield stress" in the conventional sense.
18. The flange and bolt design temperature may be reduced by 10% if the flange is not insulated.
19. Unless otherwise ordered, the manufacturer can supply either the spiral or concentric grooves if a "serrated" finish is specified.

Procedure 3-1: Design of Flanges [1–4]

Notation

A = flange O.D., in.
A_b = cross-sectional area of bolts, in.^{2}
A_m = total required cross-sectional area of bolts, in.^{2}
a = nominal bolt diameter, in.
B = flange I.D., in.
B_1 = flange I.D., in.
B_s = bolt spacing, in.
b = effective gasket width, in.
b_o = gasket seating width, in.
C = bolt circle diameter, in.
d = hub shape factor
d_1 = bolt hole diameter, in.
E, h_D, h_G, h_T, R = radial distances, in.
e = hub shape factor
F = hub shape factor for integral-type flanges
F_L = hub shape factor for loose-type flanges

f = hub stress correction factor for integral flanges
G = diameter at gasket load reaction, in.
g_o = thickness of hub at small end, in.
g_1 = thickness of hub at back of flange, in.
H = hydrostatic end force, lb
H_D = hydrostatic end force on area inside of flange, lb
H_G = gasket load, operating, lb
H_p = total joint-contact surface compression load, lb
H_T = pressure force on flange face, lb
h = hub length, in.
h_o = hub factor
M_D = moment due to H_D, in.-lb
M_G = moment due to H_G, in.-lb
M_o = total moment on flange, operating, in.-lb
M'_o = total moment on flange, seating
M_T = moment due to H_T, in.-lb
m = gasket factor
m_o = unit load, operating, lb
m_g = unit load, gasket seating, lb
N = width of gasket, in.
n = number of bolts
v = Poisson's ratio, 0.3 for steel
P = design pressure, psi
S_a = allowable stress, bolt, at ambient temperature, psi
S_b = allowable stress, bolt, at design temperature, psi
S_{fa} = allowable stress, flange, at ambient temperature, psi
S_{fo} = allowable stress, flange, at design temperature, psi
S_H = longitudinal hub stress, psi
S_R = radial stress in flange, psi
S_T = tangential stress in flange, psi
T, U, Y, Z = K-factors (see Table 3-6)
T_r, U_r, Y_r = K-factors for reverse flanges
t = flange thickness, in.
t_n = pipe wall thickness, in.
V = hub shape factor for integral flanges
V_L = hub shape factor for loose flanges
W = flange design bolt load, lb
W_{m1} = required bolt load, operating, lb
W_{m2} = required bolt load, gasket seating, lb
w = width of raised face or gasket contact width, in. (See Table 3-5)
y = gasket design seating stress, psi

Formulas

$$h_D = \frac{C - \text{dia. } H_D}{2}$$

$$h_T = \frac{C - \text{dia. } H_T}{2}$$

$$h_G = \frac{C - G}{2}$$

$$h_o = \sqrt{B\ g_o}$$

$$H_D = \frac{\pi B^2 P}{4}$$

$$H_T = H - H_D$$

$$H_G = \text{operating} = W_{ml} - H$$
$$\text{gasket seating} = W$$

$$H = \frac{\pi G^2 P}{4}$$

$$m_o = \frac{M_o}{B}$$

$$m_G = \frac{M_G}{B}$$

$$M_D = H_D h_D$$

$$M_T = H_T h_T$$

$$M_G = W h_G$$

$$E = \frac{A - C}{2}$$

$$K = \frac{A}{B}$$

$$T = \frac{(1 - v^2)(K^2 - 1)\,U}{(1 - v) + (1 + v)\,K^2}$$

$$Z = \frac{K^2 + 1}{K^2 - 1}$$

$$Y = (1 - v^2)U$$

$$U = \frac{K^2(1 + 4.6052\,(1 + v/1 - v)\log_{10} K) - 1}{1.0472\,(K^2 - 1)\,(K - 1)(1 + v)}$$

B_1 = loose flanges = $B + g_1$

= integral flanges, $f < 1 = B + g_1$

= integral flanges, $f \geq 1 = B + g_o$

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

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

$$= \text{reverse flanges} = \frac{U_r h_o g_o^2}{V}$$

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

$$= \text{integral flanges} = \frac{F}{h_o}$$

G = (if $b_o \leq 0.25$ in.) mean diameter of gasket face

= (if $b_o > 0.25$ in.) O.D. of gasket contact face $- 2b$

Stress Formula Factors

$\alpha = t\,e + 1$

$\beta = 1.333\,t\,e + 1$

$$\delta = \frac{t^3}{d}$$

$$\gamma = \frac{\alpha}{T} \text{ or } \frac{\alpha}{T_r} \text{ for reverse flanges}$$

$\lambda = \gamma + \delta$

$$\alpha_R = \frac{1}{K^2}\left[1 + \frac{3(K + 1)(1 - v)}{\pi Y}\right]$$

For factors, F, V, F_L, and V_L, see Table 2-7.1 of the ASME Code [1].

TYPE 1: WELD NECK FLANGE DESIGN (INTEGRAL)

1	DESIGN CONDITIONS				
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5	MOMENT CALCULATIONS				
Load	×	Lever Arm		=	Moment
		Operating			
$H_D = \pi B^2P/4$		$h_D = R + 0.5g_1$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(R + g_1 + h_G)$		$M_T = H_Th_T$	
				M_o	
		Seating			
$H_G = W$		$h_G = 0.5(C - G)$		M_o'	

6	K AND HUB FACTORS		
$K = A/B$		h/h_o	
T		F	
Z		V	
Y		f	
U		$e = F/h_o$	
g_1/g_o		$d = \frac{U}{V} h_o g_o^2$	
$h_o = \sqrt{Bg_o}$			

7	STRESS FORMULA FACTORS		
t			
$\alpha = te + 1$			
$\beta = 4/3\ te + 1$			
$\gamma = \alpha/T$			
$\delta = t^3/d$			
$\lambda = \gamma + \delta$			
$m_o = M_o/B$			
$m_G = M_o'/B$			

If bolt spacing exceeds 2a + t, multiply m_o and m_G in above equation by: $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$

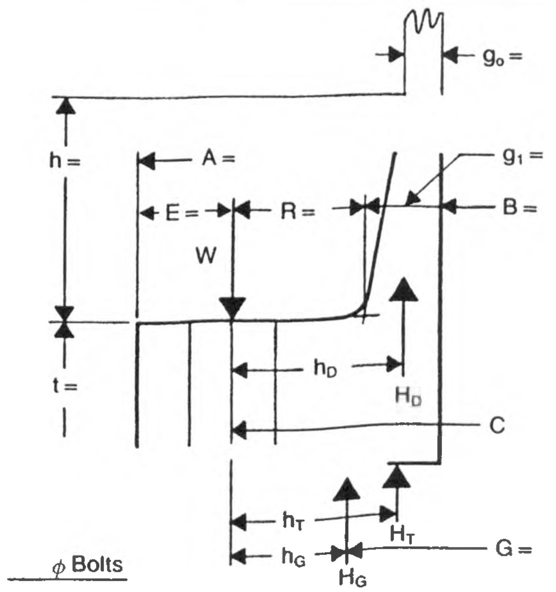


Figure 3-1. Dimensional data and forces for a weld neck flange (integral).

8	STRESS CALCULATIONS				
Allowable Stress	Operating		Allowable Stress	Seating	
1.5 S_{fo}	Longitudinal hub, $S_H = fm_o/\lambda g_1^2$		1.5 S_{fa}	Longitudinal hub, $S_H = fm_G/\lambda g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda t^2$		S_{fa}	Radial flange, $S_R = \beta m_G/\lambda t^2$	
S_{fo}	Tangential flange, $S_T = m_oY/t^2 - ZS_R$		S_{fa}	Tangential flange, $S_T = m_GY/t^2 - ZS_R$	
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$		S_{fa}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	

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TYPE 2: SLIP-ON FLANGE DESIGN (LOOSE)

1		DESIGN CONDITIONS			
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5		MOMENT CALCULATIONS			
Load		× Lever Arm		= Moment	
		Operating			
$H_D = \pi B^2P/4$		$h_D = R + g_1$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(R + g_1 + h_G)$		$M_T = H_Th_T$	
				M_o	
		Seating			
$H_G = W$		$h_G = 0.5(C - G)$		M_o'	

6	K AND HUB FACTORS		
$K = A/B$		h/h_o	
T		F_L	
Z		V_L	
Y			
U		$e = \frac{F_L}{h_o}$	
g_1/g_o		$d = \frac{U}{V} h_og_o^2$	
$h_o = \sqrt{Bg_o}$			

7	STRESS FORMULA FACTORS	
t		
$\alpha = te + 1$		
$\beta = 4/3\ te + 1$		
$\gamma = \alpha/T$		
$\delta = t^3/d$		
$\lambda = \gamma + \delta$		
$m_o = M_o/B$		
$m_G = M_o'/B$		

If bolt spacing exceeds 2a + t, multiply m_o and m_g in above equation by: $\sqrt{\dfrac{\text{Bolt spacing}}{2a + t}}$

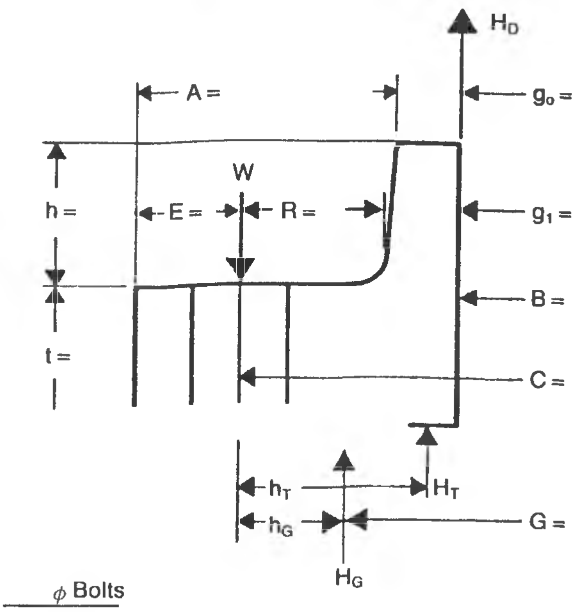

Figure 3-2. Dimensional data and forces for a slip-on flange (loose).

8		STRESS CALCULATIONS			
Allowable Stress	Operating		Allowable Stress	Seating	
1.5 S_{fo}	Longitudinal hub, $S_H = m_o/\lambda g_1^2$		1.5 S_{fa}	Longitudinal hub, $S_H = m_G/\lambda g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda t^2$		S_{fa}	Radial flange, $S_R = \beta m_G/\lambda t^2$	
S_{fo}	Tangential flange, $S_T = m_oY/t^2 - ZS_R$		S_{fa}	Tangential flange, $S_T = m_GY/t^2 - ZS_R$	
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$		S_{fa}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	

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TYPE 3: RING FLANGE DESIGN

1 — DESIGN CONDITIONS

		Allowable Stresses			
Design pressure, P		Flange		Bolting	
Design temperature		Design temp., S_{fo}		Design temp., S_b	
Flange material		Atm. temp., S_{fa}		Atm. temp., S_a	
Bolting material					
Corrosion allowance					

2 — GASKET AND FACING DETAILS

Gasket		Facing	

3

N	
b	
G	
y	
m	

4 — LOAD AND BOLT CALCULATIONS

$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
$H_P = 2b\pi GmP$			
$H = G^2\pi P/4$		A_b	
$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	

5 — MOMENT CALCULATIONS

Load		× Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$		$h_D = 0.5(C - B)$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(h_D + h_G)$		$M_T = H_Th_T$	
				M_o	
Seating					
$H_G = W$		$h_G = 0.5(C - G)$		M_o'	

6 — SHAPE CONSTANTS

K = A/B		Y	

If bolt spacing exceeds 2a + t, multiply M_o and M_o' in above equation by: $\sqrt{\dfrac{\text{Bolt spacing}}{2a + t}}$

7 — FLANGE THICKNESS REQUIRED

t = greater of

Operating	Seating
$t = \sqrt{\dfrac{M_oY}{S_{fo}B}}$	$t = \sqrt{\dfrac{M_o'Y}{S_{fa}B}}$

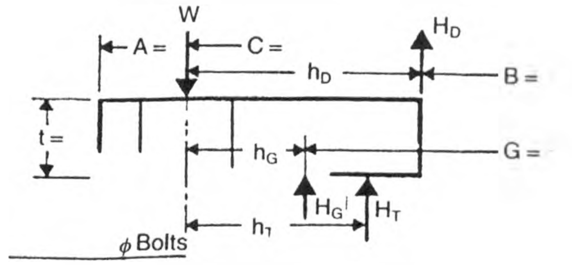

Figure 3-3. Dimensional data and forces for a ring flange.

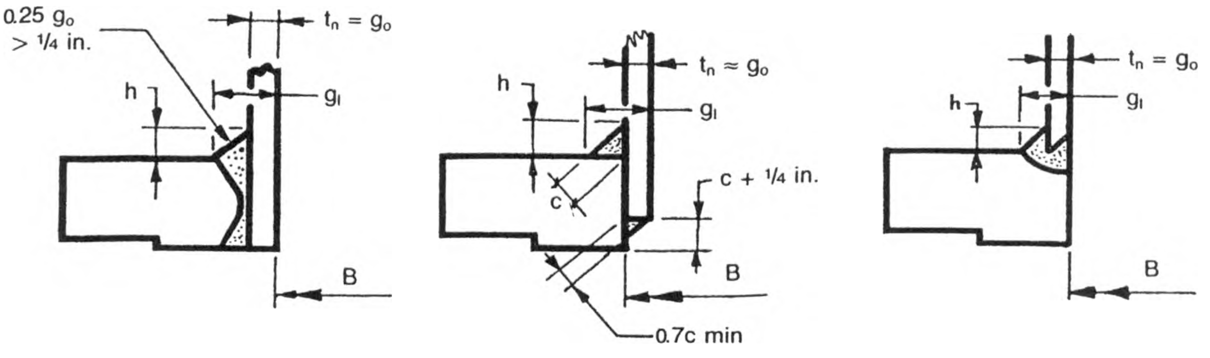

Figure 3-4. Various attachments of ring flanges. (All other dimensions and loadings per Figure 3-2.)

8 NOTES

If $g_o < 1.5t_n$ and $h < g_o$, design as integral. If $g_o > 1.5t_n$ and $h > g_o$, design as loose.
If $g_o \le 5/8$ in., $B/g_o \le 300$, $P \le 300$ psi and design temp. < 700°, design as integral or loose.
c = lesser of t_n or $\begin{Bmatrix}\text{loose: } 2t_m \\ \text{integral: } 2g_o\end{Bmatrix}$ but not less than 1/4 in.

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TYPE 4: REVERSE FLANGE DESIGN

1	DESIGN CONDITIONS				
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5	MOMENT CALCULATIONS				
Load		× Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$		$h_D = 0.5(C + g_1 - 2g_o - B)$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(C - (B + G)/2)$		$M_T = H_Th_T$	
Add moments algebraically, then use the absolute value $\lvert M_o \rvert$ in all subsequent calculations.				$\lvert M_o \rvert$	
Seating					
$H_G = W$		$h_G = 0.5(C - G)$		M_o'	

6	K AND HUB FACTORS		
$K = A/B'$		h/h_o	
T		F	
Z		V	
Y		f	
U		$e = F/h_o$	
g_1/g_o		$d = \dfrac{U_R}{V} h_o g_o^2$	
$h_o = \sqrt{Ag_o}$			
$Y_R = \alpha_R Y$		$U_R = \alpha_R U$	
$\alpha_R = \dfrac{1}{K^2}\left[1 + \dfrac{3(K+1)(1-\nu)}{\pi Y}\right]$			
$T_R = \dfrac{(Z+\nu)}{(Z-\nu)}\alpha_R T$			

7	STRESS FORMULA FACTORS		
t		$\delta = t^3/d$	
$\alpha = te + 1$		$\lambda = \gamma + \delta$	
$\beta = 4/3\ te + 1$		$m_o = M_o/B'$	
$\gamma = \alpha/T_R$		$m_G = M_o'/B'$	

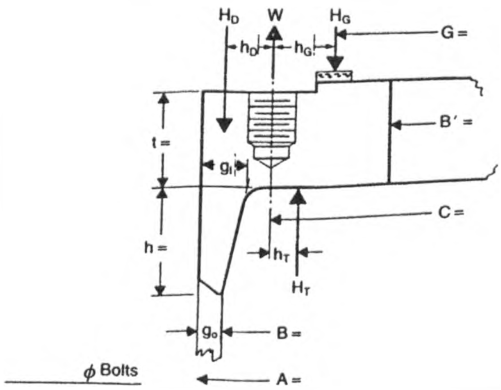

Figure 3-5. Dimensional data and forces for a reverse flange.

8	STRESS CALCULATIONS				
Allowable Stress	Operating		Allowable Stress	Seating	
1.5 S_{fo}	Longitudinal hub, $S_H = fm_o/\lambda g_1^2$		1.5 S_{fa}	Longitudinal hub, $S_H = fm_G/\lambda g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda t^2$		S_{fa}	Radial flange, $S_R = \beta m_G/\lambda t^2$	
S_{fo}	Tangential flange, $S_T = m_oY_R/t^2 - ZS_R(0.67te + 1)/\beta$		S_{fa}	Tangential flange, $S_T = m_GY_R/t^2 - ZS_R(0.67te + 1)/\beta$	
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$		S_{fa}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	
S_{fo}	Tangential flange $S_T(A_TB') = \dfrac{m_o}{t^2}\left[Y - \dfrac{2k^2\left(1 + \frac{2}{3}te\right)}{(k^2-1)\lambda}\right]$		S_{fa}	Tangential flange $S_T(A_TB') = \dfrac{m_G}{t^2}\left[Y - \dfrac{2k^2\left(1 + \frac{2}{3}te\right)}{(k^2-1)\lambda}\right]$	

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TYPE 5: SLIP-ON FLANGE, FLAT FACE, FULL GASKET

1 DESIGN CONDITIONS

		Allowable Stresses			
Design pressure, P		Flange		Bolting	
Design temperature					
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2 GASKET AND FACING DETAILS

Gasket		Facing	

3

$G = C - 2h_G$	
$b = (C - B)/4$	
y	
m	

4 LOAD AND BOLT CALCULATIONS

$W_{m2} = b\pi Gy + H'_{GY}$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
$H_P = 2b\pi GmP$			
$H'_P = (h_G/h'_G)H_P$		A_b	
$H = G^2\pi P/4$		$W = 0.5(A_m + A_b)S_a$	
$W_{m1} = H + H_P + H'_P$		$H'_{GY} = (h_G/h'_G)b\pi Gy$	

5 MOMENT CALCULATIONS

Load		× Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$		$h_D = R + g_1$		$M_D = H_Dh_D$	
$H_T = H - H_D$		$h_T = 0.5(R + g_1 + h_G)$		$M_T = H_Th_T$	
				M_o	
Lever Arms					
$h_G = \dfrac{(C - B)(2B + C)}{6(B + C)}$		$h'_G = \dfrac{(A - C)(2A + C)}{6(C + A)}$			
Reverse Moment					
$H_G = W - H$		$h''_G = \dfrac{h_Gh'_G}{h_G + h'_G}$		$M_G = H_Gh''_G$	

6 K AND HUB FACTORS

$K = A/B$		h/h_o	
T		F_L	
Z		V_L	
Y		$e = \dfrac{F_L}{h_o}$	
U			
g_1/g_o		$d = \dfrac{U}{V_L}h_og_o^2$	
$h_o = \sqrt{Bg_o}$			

7 STRESS FORMULA FACTORS

t		$\delta = t^3/d$	
$\alpha = te + 1$		$\lambda = \gamma + \delta$	
$\beta = 4/3\ te + 1$		$m_o = M_o/B$	
$\gamma = \alpha/T$			

If bolt spacing exceeds 2a + t, multiply m_o in above equation by: $\sqrt{\dfrac{\text{Bolt spacing}}{2a + t}}$

8 STRESS CALCULATIONS

Allowable Stress	Operating	
1.5 S_{fo}	Longitudinal hub, $S_H = m_o/\lambda g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda t^2$	
S_{fo}	Tangential flange, $S_T = m_oY/t^2 - ZS_R$	
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	
S_{fo}	Radial stress at bolt circle $S_{RAD} = \dfrac{6M_G}{t^2(\pi C - nd_1)}$	

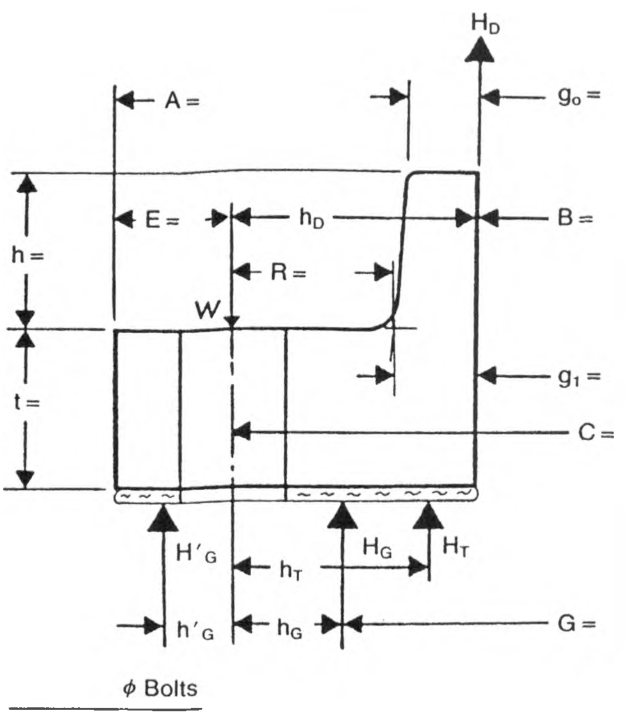

Figure 3-6. Dimensional data and forces for a slip-on flange, flat face, full gasket.

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Table 3-4
Gasket materials and contact facings[1]
Gasket factors (m) for operating conditions and minimum design stress (y)

Gasket Material		Gasket Factor m	Min. Design Seating Stress y	Sketches and Notes	Use Facing Sketch (Refer to Table 3-5)	Use Column III (Refer to Table 3-5)
Self-energizing types: O rings, metallic, elastomer or other gasket types considered as self-sealing		0	0			
Elastomers without fabric or a high percentage of asbestos fiber: Below 75A Shore Durometer		0.50	0		(1a), (1b), (1e), (1d), (4), (5)	II
Elastomers without fabric or a high percentage of asbestos fiber: 75A or higher Shore Durometer		1.00	200		(1a), (1b), (1e), (1d), (4), (5)	II
Asbestos with a suitable binder for the operating conditions	1/8 thick	2.00	1,600		(1a), (1b), (1e), (1d), (4), (5)	II
	1/16 thick	2.75	3,700			
	1/32 thick	3.50	6,500			
Elastomers with cotton fabric insertion		1.25	400		(1a), (1b), (1e), (1d), (4), (5)	II
Elastomers with asbestos fabric insertion, with or without wire reinforcement	3-ply	2.25	2,200		(1a), (1b), (1e), (1d), (4), (5)	II
	2-ply	2.50	2,900			
	1-ply	2.75	3,700			
Vegetable fiber		1.75	1,100		(1a), (1b), (1e), (1d), (4), (5)	II
Spiral-wound metal, asbestos filled	Carbon	2.50	10,000			II
	Stainless or Monel	3.00	10,000			
Corrugated metal, asbestos inserted or corrugated metal, jacketed asbestos filled	Soft aluminum	2.50	2,900			II
	Soft copper or brass	2.75	3,700			
	Iron or soft steel	3.00	4,500			
	Monel or 4%–6% chrome	3.25	5,500			
	Stainless steels	3.50	6,500			
Corrugated metal	Soft aluminum	2.75	3,700		(1a), (1b), (1c), (1d)	II
	Soft copper or brass	3.00	4,500			
	Iron or soft steel	3.25	5,500			
	Monel or 4%–6% chrome	3.50	6,500			
	Stainless steels	3.75	7,600			
Flat metal jacketed asbestos filled	Soft aluminum	3.25	5,500		(1a), (1b), (1c), (1d), (2)[2]	II
	Soft copper or brass	3.50	6,500			
	Iron or soft steel	3.75	7,600			
	Monel or	3.50	8,000			
	4%–6% chrome	3.75	9,000			
	Stainless steels	3.75	9,000			
Grooved metal	Soft aluminum	3.25	5,500		(1a), (1b), (1c), (1d), (2), (3)	II
	Soft copper or brass	3.50	6,500			
	Iron or soft steel	3.75	7,600			
	Monel or 4%–6% chrome	3.75	9,000			
	Stainless steels	4.25	10,100			
Solid flat metal	Soft aluminum	4.00	8,800		(1a), (1b), (1c), (1d), (2), (3), (4), (5)	I
	Soft copper or brass	4.75	13,000			
	Iron or soft steel	5.50	18,000			
	Monel or 4%–6% chrome	6.00	21,800			
	Stainless steels	6.50	26,000			
Ring joint	Iron or soft steel	5.50	18,000		(6)	I
	Monel or 4%–6% chrome	6.00	21,800			
	Stainless steels	6.50	26,000			

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 3-5. 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.

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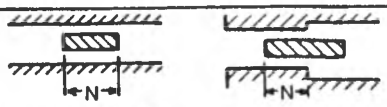
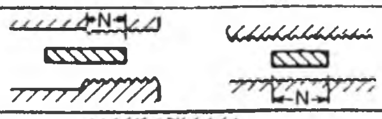
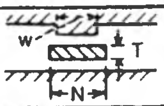
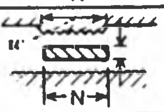
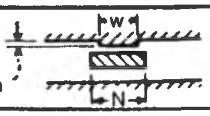
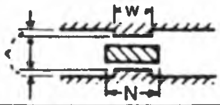
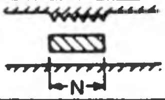
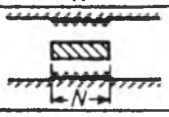
Table 3-5
Effective gasket width

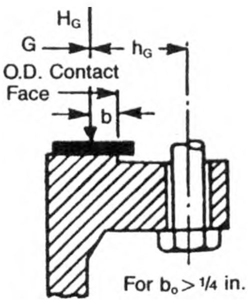
Facing Sketch (Exaggerated)		Basic Gasket Seating Width, b_o Column I	Column II
(1a)		$\frac{N}{2}$	$\frac{N}{2}$
(1b)*			
(1c)	 $w \leq N$	$\frac{w+T}{2}; \left(\frac{w+N}{4} \text{ max}\right)$	$\frac{w-T}{2}; \left(\frac{w-N}{4} \text{ max}\right)$
(1d)	 $w \leq N$		
(2) 1/64 in. Nubbin	 $w \leq \frac{N}{2}$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

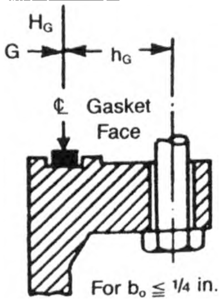
Effective Gasket Seating Width, b

$b = b_o$, when $b_o \leq 1/4$ in.

$b = \frac{\sqrt{b_o}}{2}$, when $b_o > 1/4$ in.

Location of Gasket Load Reaction





Note: The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes

* Where serrations do not exceed 1/64-in. depth and 1/32-in. width spacing, sketches (1b) and (1d) shall be used.

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Table 3-6
Table of Coefficients

K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U
1.001	1.91	1000.50	1911.16	2100.18	1.046	1.90	22.05	42.75	46.99	1.091	1.88	11.52	22.22	24.41	1.136	1.86	7.88	15.26	16.77
1.002	1.91	500.50	956.16	1050.72	1.047	1.90	21.79	41.87	46.03	1.092	1.88	11.40	21.99	24.16	1.137	1.86	7.83	15.15	16.65
1.003	1.91	333.83	637.85	700.93	1.048	1.90	21.35	41.02	45.09	1.093	1.88	11.28	21.76	23.91	1.138	1.86	7.78	15.05	16.54
1.004	1.91	250.50	478.71	526.05	1.049	1.90	20.92	40.21	44.21	1.094	1.88	11.16	21.54	23.67	1.139	1.86	7.73	14.95	16.43
1.005	1.91	200.50	383.22	421.12	1.050	1.89	20.51	39.43	43.34	1.095	1.88	11.05	21.32	23.44	1.140	1.86	7.68	14.86	16.35
1.006	1.91	167.17	319.56	351.16	1.051	1.89	20.12	38.68	42.51	1.096	1.88	10.94	21.11	23.20	1.141	1.86	7.62	14.76	16.22
1.007	1.91	143.36	274.09	301.20	1.052	1.89	19.74	37.96	41.73	1.097	1.88	10.83	20.91	22.97	1.142	1.86	7.57	14.66	16.11
1.008	1.91	125.50	239.95	263.75	1.053	1.89	19.38	37.27	40.96	1.098	1.88	10.73	20.71	22.75	1.143	1.86	7.53	14.57	16.01
1.009	1.91	111.61	213.40	234.42	1.054	1.89	19.03	36.60	40.23	1.099	1.88	10.62	20.51	22.39	1.144	1.86	7.48	14.48	15.91
1.010	1.91	100.50	192.19	211.19	1.055	1.89	18.69	35.96	39.64	1.100	1.88	10.52	20.31	22.18	1.145	1.86	7.43	14.39	15.83
1.011	1.91	91.41	174.83	192.13	1.056	1.89	18.38	35.34	38.84	1.101	1.88	10.43	20.15	22.12	1.146	1.86	7.38	14.29	15.71
1.012	1.91	83.84	160.38	176.25	1.057	1.89	18.06	34.74	38.19	1.102	1.88	10.33	19.94	21.92	1.147	1.86	7.34	14.20	15.61
1.013	1.91	77.43	148.06	162.81	1.058	1.89	17.76	34.17	37.56	1.103	1.88	10.23	19.76	21.72	1.148	1.86	7.29	14.12	15.51
1.014	1.91	71.93	137.69	151.30	1.059	1.89	17.47	33.62	36.95	1.104	1.88	10.14	19.58	21.52	1.149	1.86	7.25	14.03	15.42
1.015	1.91	67.17	128.61	141.33	1.060	1.89	17.18	33.04	36.34	1.105	1.88	10.05	19.38	21.30	1.150	1.86	7.20	13.95	15.34
1.016	1.90	63.00	120.56	132.49	1.061	1.89	16.91	32.55	35.78	1.106	1.88	9.96	19.33	21.14	1.151	1.86	7.16	13.86	15.23
1.017	1.90	59.33	111.98	124.81	1.062	1.89	16.64	32.04	35.21	1.107	1.87	9.87	19.07	20.96	1.152	1.86	7.11	13.77	15.14
1.018	1.90	56.06	107.36	118.00	1.063	1.89	16.40	31.55	34.68	1.108	1.87	9.78	18.90	20.77	1.153	1.86	7.07	13.69	15.05
1.019	1.90	53.14	101.72	111.78	1.064	1.89	16.15	31.08	34.17	1.109	1.87	9.70	18.74	20.59	1.154	1.86	7.03	13.61	14.96
1.020	1.90	50.51	96.73	106.30	1.065	1.89	15.90	30.61	33.65	1.110	1.87	9.62	18.55	20.38	1.155	1.86	6.99	13.54	14.87
1.021	1.90	48.12	92.21	101.33	1.066	1.89	15.67	30.17	33.17	1.111	1.87	9.54	18.42	20.25	1.156	1.86	6.95	13.45	14.78
1.022	1.90	45.96	88.04	96.75	1.067	1.89	15.45	29.74	32.69	1.112	1.87	9.46	18.27	20.08	1.157	1.86	6.91	13.37	14.70
1.023	1.90	43.98	84.30	92.64	1.068	1.89	15.22	29.32	32.22	1.113	1.87	9.38	18.13	19.91	1.158	1.86	6.87	13.30	14.61
1.024	1.90	42.17	80.81	88.81	1.069	1.89	15.02	28.91	31.79	1.114	1.87	9.30	17.97	19.75	1.159	1.86	6.83	13.22	14.53
1.025	1.90	40.51	77.61	85.29	1.070	1.89	14.80	28.51	31.34	1.115	1.87	9.22	17.81	19.55	1.160	1.86	6.79	13.15	14.45
1.026	1.90	38.97	74.70	82.09	1.071	1.89	14.61	28.13	30.92	1.116	1.87	9.15	17.68	19.43	1.161	1.85	6.75	13.07	14.36
1.027	1.90	37.54	71.97	79.08	1.072	1.89	14.41	27.76	30.51	1.117	1.87	9.07	17.54	19.27	1.162	1.85	6.71	13.00	14.28
1.028	1.90	36.22	69.43	76.30	1.073	1.89	14.22	27.39	30.11	1.118	1.87	9.00	17.40	19.12	1.163	1.85	6.67	12.92	14.20
1.029	1.90	34.99	67.11	73.75	1.074	1.88	14.04	27.04	29.72	1.119	1.87	8.94	17.27	18.98	1.164	1.85	6.64	12.85	14.12
1.030	1.90	33.84	64.91	71.33	1.075	1.88	13.85	26.69	29.34	1.120	1.87	8.86	17.13	18.80	1.165	1.85	6.60	12.78	14.04
1.031	1.90	32.76	62.85	69.06	1.076	1.88	13.68	26.36	28.98	1.121	1.87	8.79	17.00	18.68	1.166	1.85	6.56	12.71	13.97
1.032	1.90	31.76	60.92	66.94	1.077	1.88	13.56	26.03	28.69	1.122	1.87	8.72	16.87	18.54	1.167	1.85	6.53	12.64	13.89
1.033	1.90	30.81	59.11	64.95	1.078	1.88	13.35	25.72	28.27	1.123	1.87	8.66	16.74	18.40	1.168	1.85	6.49	12.58	13.82

1.034	1.90	29.92	57.41	63.08	1.079	1.88	13.18	25.40	27.92	1.124	1.87	8.59	16.62	18.26	1.169	1.85	6.46	12.51	13.74
1.035	1.90	29.08	55.80	61.32	1.080	1.88	13.02	25.10	27.59	1.125	1.87	8.53	16.49	18.11	1.170	1.85	6.42	12.43	13.66
1.036	1.90	28.29	54.29	59.66	1.081	1.88	12.87	24.81	27.27	1.126	1.87	8.47	16.37	17.99	1.171	1.85	6.39	12.38	13.60
1.037	1.90	27.54	52.85	58.08	1.082	1.88	12.72	24.52	26.95	1.127	1.87	8.40	16.25	17.86	1.172	1.85	6.35	12.31	13.53
1.038	1.90	26.83	51.50	56.59	1.083	1.88	12.57	24.24	26.65	1.128	1.87	8.34	16.14	17.73	1.173	1.85	6.32	12.25	13.46
1.039	1.90	26.15	50.21	55.17	1.084	1.88	12.43	24.00	26.34	1.129	1.87	8.28	16.02	17.60	1.174	1.85	6.29	12.18	13.39
1.040	1.90	25.51	48.97	53.82	1.085	1.88	12.29	23.69	26.05	1.130	1.87	8.22	15.91	17.48	1.175	1.85	6.25	12.10	13.30
1.041	1.90	24.90	47.81	53.10	1.086	1.88	12.15	23.44	25.77	1.131	1.87	8.16	15.79	17.35	1.176	1.85	6.22	12.06	13.25
1.042	1.90	24.32	46.71	51.33	1.087	1.88	12.02	23.18	25.48	1.132	1.87	8.11	15.68	17.24	1.177	1.85	6.19	12.00	13.18
1.043	1.90	23.77	45.64	50.15	1.088	1.88	11.89	22.93	25.20	1.133	1.86	8.05	15.57	17.11	1.178	1.85	6.16	11.93	13.11
1.044	1.90	23.23	44.64	49.05	1.089	1.88	11.76	22.68	24.93	1.134	1.86	7.99	15.46	16.99	1.179	1.85	6.13	11.87	13.05
1.045	1.90	22.74	43.69	48.02	1.090	1.88	11.63	22.44	24.66	1.135	1.86	7.94	15.36	16.90	1.180	1.85	6.10	11.79	12.96
1.182	1.85	6.04	11.70	12.86	1.278	1.81	4.16	8.05	8.85	1.434	1.74	2.89	5.56	6.10	1.75	1.60	1.97	3.64	4.00
1.184	1.85	5.98	11.58	12.73	1.281	1.81	4.12	7.98	8.77	1.438	1.74	2.87	5.52	6.05	1.76	1.60	1.95	3.61	3.96
1.186	1.85	5.92	11.47	12.61	1.284	1.80	4.08	7.91	8.69	1.442	1.74	2.85	5.48	6.01	1.77	1.60	1.94	3.57	3.93
1.188	1.85	5.86	11.36	12.49	1.287	1.80	4.05	7.84	8.61	1.446	1.74	2.83	5.44	5.97	1.78	1.59	1.92	3.54	3.89
1.190	1.84	5.81	11.26	12.37	1.290	1.80	4.01	7.77	8.53	1.450	1.73	2.81	5.40	5.93	1.79	1.59	1.91	3.51	3.85
1.192	1.84	5.75	11.15	12.25	1.293	1.80	3.98	7.70	8.46	1.454	1.73	2.80	5.36	5.89	1.80	1.58	1.89	3.47	3.82
1.194	1.84	5.70	11.05	12.14	1.296	1.80	3.94	7.63	8.39	1.458	1.73	2.78	5.32	5.85	1.81	1.58	1.88	3.44	3.78
1.196	1.84	5.65	10.95	12.03	1.299	1.80	3.91	7.57	8.31	1.462	1.73	2.76	5.28	5.80	1.82	1.58	1.86	3.41	3.75
1.198	1.84	5.60	10.85	11.92	1.302	1.80	3.88	7.50	8.24	1.466	1.73	2.74	5.24	5.76	1.83	1.57	1.85	3.38	3.72
1.200	1.84	5.55	10.75	11.81	1.305	1.80	3.84	7.44	8.18	1.470	1.72	2.72	5.20	5.71	1.84	1.57	1.84	3.35	3.69
1.202	1.84	5.50	10.65	11.71	1.308	1.79	3.81	7.38	8.11	1.475	1.72	2.70	5.16	5.66	1.85	1.56	1.83	3.33	3.65
1.204	1.84	5.45	10.56	11.61	1.311	1.79	3.78	7.32	8.05	1.480	1.72	2.68	5.12	5.61	1.86	1.56	1.81	3.30	3.62
1.206	1.84	5.40	10.47	11.51	1.314	1.79	3.75	7.26	7.98	1.485	1.72	2.66	5.08	5.57	1.87	1.56	1.80	3.27	3.59
1.208	1.84	5.35	10.38	11.41	1.317	1.79	3.72	7.20	7.92	1.490	1.72	2.64	5.04	5.53	1.88	1.55	1.79	3.24	3.56
1.210	1.84	5.31	10.30	11.32	1.320	1.79	3.69	7.14	7.85	1.495	1.71	2.62	5.00	5.49	1.89	1.55	1.78	3.22	3.54
1.212	1.83	5.27	10.21	11.22	1.323	1.79	3.67	7.09	7.79	1.500	1.71	2.60	4.96	5.45	1.90	1.54	1.77	3.19	3.51
1.214	1.83	5.22	10.12	11.12	1.326	1.79	3.64	7.03	7.73	1.505	1.71	2.58	4.92	5.41	1.91	1.54	1.75	3.17	3.48
1.216	1.83	5.18	10.04	11.03	1.329	1.78	3.61	6.98	7.67	1.510	1.71	2.56	4.88	5.37	1.92	1.54	1.74	3.14	3.45
1.218	1.83	5.14	9.96	10.94	1.332	1.78	3.58	6.92	7.61	1.515	1.71	2.54	4.84	5.33	1.93	1.53	1.73	3.12	3.43
1.220	1.83	5.10	9.89	10.87	1.335	1.78	3.56	6.87	7.55	1.520	1.70	2.53	4.80	5.29	1.94	1.53	1.72	3.09	3.40
1.222	1.83	5.05	9.80	10.77	1.338	1.78	3.53	6.82	7.50	1.525	1.70	2.51	4.77	5.25	1.95	1.53	1.71	3.07	3.38
1.224	1.83	5.01	9.72	10.68	1.341	1.78	3.51	6.77	7.44	1.530	1.70	2.49	4.74	5.21	1.96	1.52	1.70	3.05	3.35
1.226	1.83	4.98	9.65	10.60	1.344	1.78	3.48	6.72	7.39	1.535	1.70	2.47	4.70	5.17	1.97	1.52	1.69	3.03	3.33
1.228	1.83	4.94	9.57	10.52	1.347	1.78	3.46	6.68	7.33	1.540	1.69	2.46	4.66	5.13	1.98	1.51	1.68	3.01	3.30
1.230	1.83	4.90	9.50	10.44	1.350	1.78	3.43	6.63	7.28	1.545	1.69	2.44	4.63	5.09	1.99	1.51	1.68	2.98	3.28
1.232	1.83	4.86	9.43	10.36	1.354	1.77	3.40	6.57	7.21	1.55	1.69	2.43	4.60	5.05	2.00	1.51	1.67	2.96	3.26
1.234	1.83	4.83	9.36	10.28	1.358	1.77	3.37	6.50	7.14	1.56	1.69	2.40	4.54	4.99	2.01	1.50	1.66	2.94	3.23

(*Continued*)

Table 3-6
Table of Coefficients—cont'd

K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U
1.236	1.82	4.79	9.29	10.20	1.362	1.77	3.34	6.44	7.08	1.57	1.68	2.37	4.48	4.92	2.02	1.50	1.65	2.92	3.21
1.238	1.82	4.76	9.22	10.13	1.366	1.77	3.31	6.38	7.01	1.58	1.68	2.34	4.42	4.86	2.04	1.49	1.63	2.88	3.17
1.240	1.82	4.72	9.15	10.05	1.370	1.77	3.28	6.32	6.95	1.59	1.67	2.31	4.36	4.79	2.06	1.48	1.62	2.85	3.13
1.242	1.82	4.69	9.08	9.98	1.374	1.77	3.25	6.27	6.89	1.60	1.67	2.28	4.31	4.73	2.08	1.48	1.60	2.81	3.09
1.244	1.82	4.65	9.02	9.91	1.378	1.76	3.22	6.21	6.82	1.61	1.66	2.26	4.25	4.67	2.10	1.47	1.59	2.78	3.05
1.246	1.82	4.62	8.95	9.84	1.382	1.76	3.20	6.16	6.77	1.62	1.65	2.23	4.20	4.61	2.12	1.46	1.57	2.74	3.01
1.248	1.82	4.59	8.89	9.77	1.386	1.76	3.17	6.11	6.72	1.63	1.65	2.21	4.15	4.56	2.14	1.46	1.56	2.71	2.97
1.250	1.82	4.56	8.83	9.70	1.390	1.76	3.15	6.06	6.66	1.64	1.65	2.18	4.10	4.50	2.16	1.45	1.55	2.67	2.94
1.252	1.82	4.52	8.77	9.64	1.394	1.76	3.12	6.01	6.60	1.65	1.65	2.16	4.05	4.45	2.18	1.44	1.53	2.64	2.90
1.254	1.82	4.49	8.71	9.57	1.398	1.75	3.10	5.96	6.55	1.66	1.64	2.14	4.01	4.40	2.20	1.44	1.52	2.61	2.87
1.256	1.82	4.46	8.65	9.51	1.402	1.75	3.07	5.92	6.49	1.67	1.64	2.12	3.96	4.35	2.22	1.43	1.51	2.58	2.84
1.258	1.81	4.43	8.59	9.44	1.406	1.75	3.05	5.87	6.44	1.68	1.63	2.10	3.92	4.30	2.24	1.42	1.50	2.56	2.81
1.260	1.81	4.40	8.53	9.38	1.410	1.75	3.02	5.82	6.39	1.69	1.63	2.08	3.87	4.26	2.26	1.41	1.49	2.53	2.78
1.263	1.81	4.36	8.45	9.28	1.414	1.75	3.00	5.77	6.34	1.70	1.63	2.06	3.83	4.21	2.28	1.41	1.48	2.50	2.75
1.266	1.81	4.32	8.37	9.19	1.418	1.75	2.98	5.72	6.29	1.71	1.62	2.04	3.79	4.17	2.30	1.40	1.47	2.48	2.72
1.269	1.81	4.28	8.29	9.11	1.422	1.75	2.96	5.68	6.25	1.72	1.62	2.02	3.75	4.12	2.32	1.40	1.46	2.45	2.69
1.272	1.81	4.24	8.21	9.02	1.426	1.74	2.94	5.64	6.20	1.73	1.61	2.00	3.72	4.08	2.34	1.39	1.45	2.43	2.67
1.275	1.81	4.20	8.13	8.93	1.430	1.74	2.91	5.60	6.15	1.74	1.61	1.99	3.68	4.04	2.36	1.38	1.44	2.40	2.64
2.38	1.38	1.43	2.38	2.61	2.83	1.25	1.28	1.98	2.17	3.46	1.11	1.18	1.64	1.80	4.15	0.989	1.12	1.40	1.54
2.40	1.37	1.42	2.36	2.59	2.86	1.24	1.28	1.96	2.15	3.50	1.10	1.18	1.62	1.78	4.20	0.982	1.12	1.39	1.53
2.42	1.36	1.41	2.33	2.56	2.89	1.23	1.27	1.94	2.13	3.54	1.09	1.17	1.61	1.76	4.25	0.975	1.12	1.38	1.51
2.44	1.36	1.40	2.31	2.54	2.92	1.22	1.27	1.92	2.11	3.58	1.08	1.17	1.59	1.75	4.30	0.968	1.11	1.36	1.50
2.46	1.35	1.40	2.29	2.52						3.62	1.07	1.16	1.57	1.73					
					2.95	1.22	1.26	1.90	2.09						4.35	0.962	1.11	1.35	1.48
2.48	1.35	1.39	2.27	2.50	2.98	1.21	1.25	1.88	2.07	3.66	1.07	1.16	1.56	1.71	4.40	0.955	1.11	1.34	1.47
2.50	1.34	1.38	2.25	2.47	3.02	1.20	1.25	1.86	2.04	3.70	1.06	1.16	1.55	1.70	4.45	0.948	1.11	1.33	1.46
2.53	1.33	1.37	2.22	2.44	3.06	1.19	1.24	1.83	2.01	3.74	1.05	1.15	1.53	1.68	4.50	0.941	1.10	1.31	1.44
2.56	1.32	1.36	2.19	2.41						3.78	1.05	1.15	1.52	1.67	4.55	0.934	1.10	1.30	1.43
					3.10	1.18	1.23	1.81	1.99						4.60	0.928	1.10	1.29	1.42
2.59	1.31	1.35	2.17	2.38	3.14	1.17	1.23	1.79	1.97	3.82	1.04	1.15	1.50	1.65	4.65	0.921	1.10	1.28	1.41
2.62	1.30	1.34	2.14	2.35	3.18	1.16	1.22	1.77	1.94	3.86	1.03	1.14	1.49	1.64	4.70	0.914	1.09	1.27	1.39
2.65	1.30	1.33	2.12	2.32	3.22	1.16	1.21	1.75	1.92	3.90	1.03	1.14	1.48	1.62	4.75	0.908	1.09	1.26	1.38
2.68	1.29	1.32	2.09	2.30	3.26	1.15	1.21	1.73	1.90	3.94	1.02	1.14	1.46	1.61	4.80	0.900	1.09	1.25	1.37
2.71	1.28	1.31	2.07	2.27	3.30	1.14	1.20	1.71	1.88	3.98	1.01	1.13	1.45	1.60	4.85	0.893	1.09	1.24	1.36
2.74	1.27	1.31	2.04	2.25	3.34	1.13	1.20	1.69	1.86	4.00	1.009	1.13	1.45	1.59	4.90	0.887	1.09	1.23	1.35
2.77	1.26	1.30	2.02	2.22	3.38	1.12	1.19	1.67	1.84	4.05	1.002	1.13	1.43	1.57	4.95	0.880	1.08	1.22	1.34
2.80	1.26	1.29	2.00	2.20	3.42	1.11	1.19	1.66	1.82	4.10	0.996	1.13	1.42	1.56	5.00	0.873	1.08	1.21	1.33

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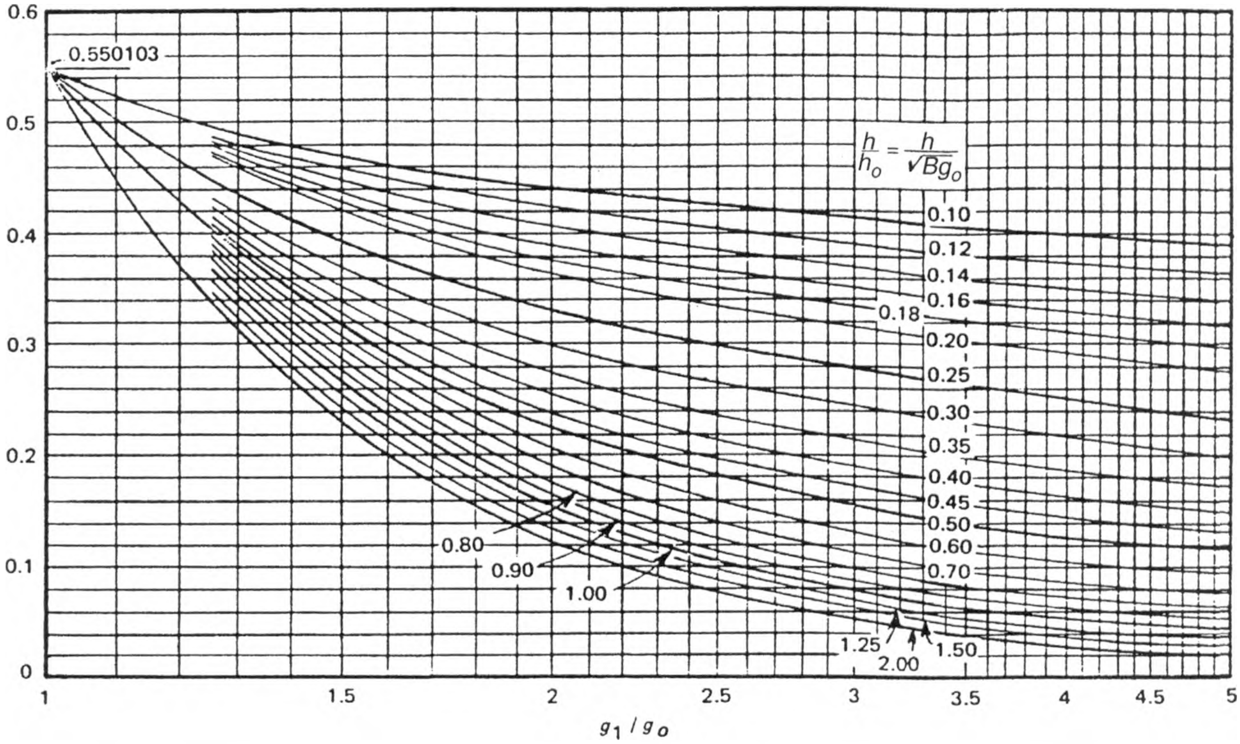


Figure 3-7. Values of V (integral flange factors). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.3.)*

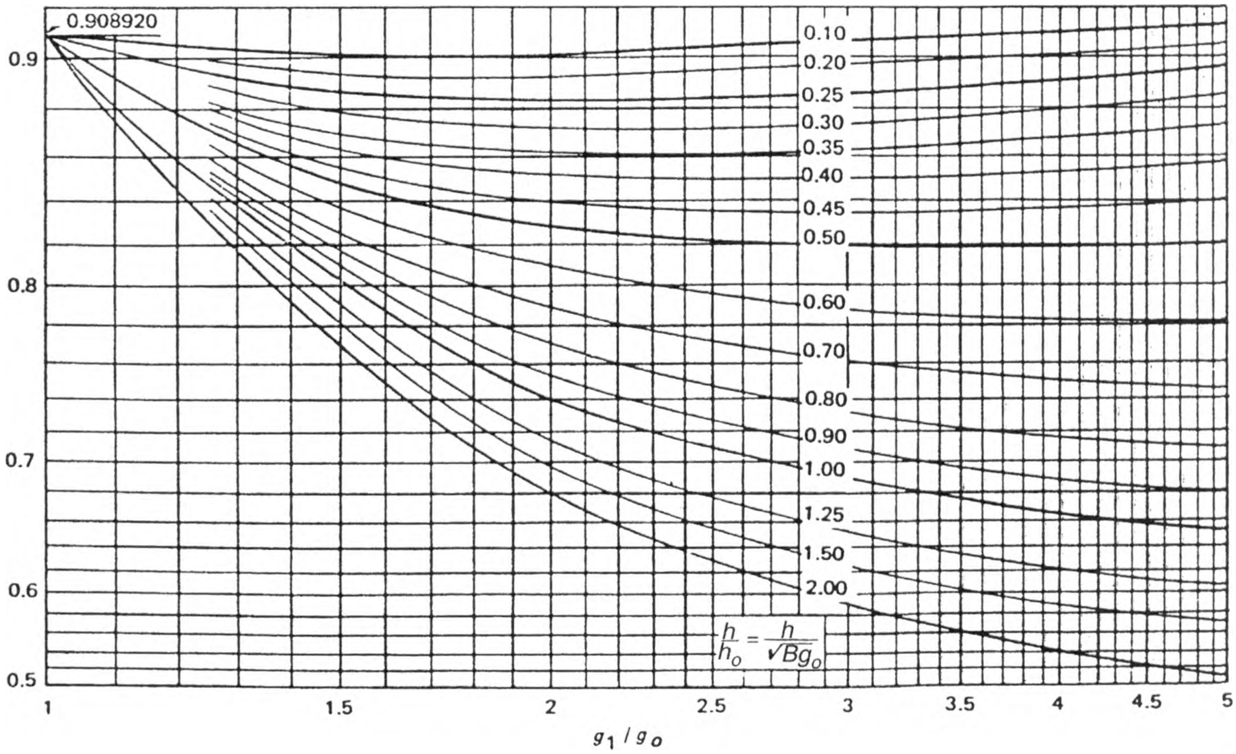


Figure 3-8. Values of F (integral flange factors). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.2.)*

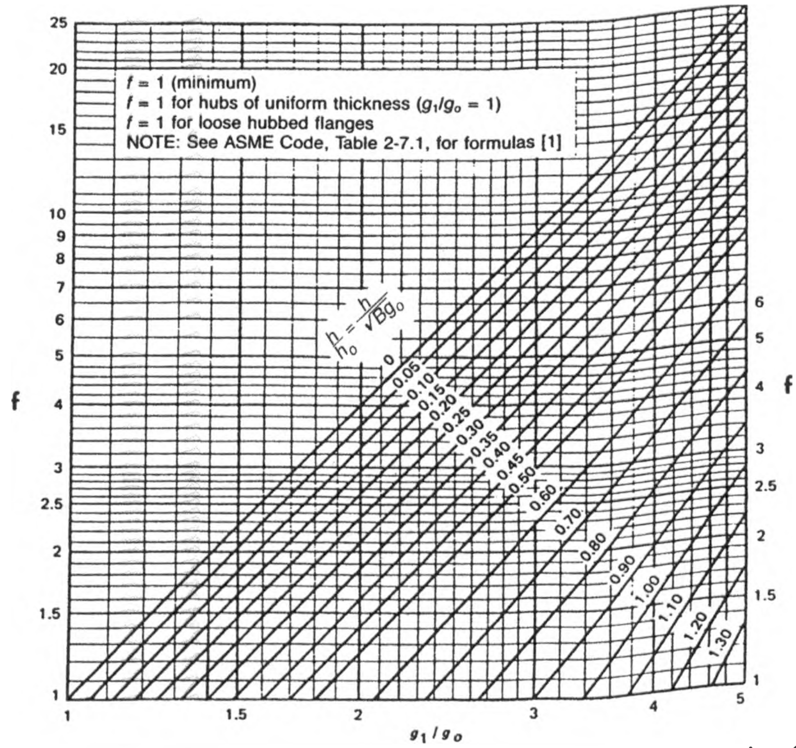


Figure 3-9. Values of f (hub stress correction factor). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.6.)*

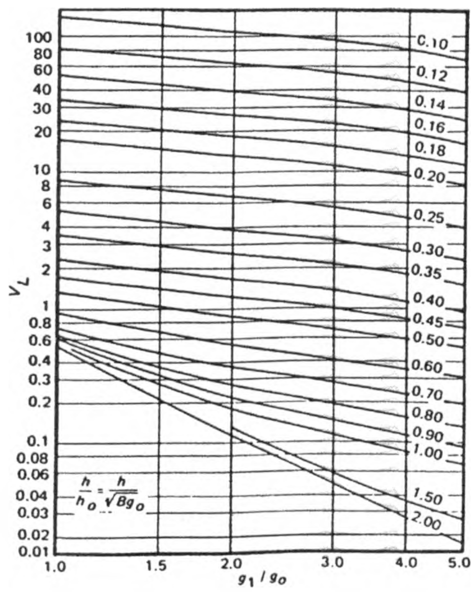


Figure 3-10. Values of V_L (loose hub flange factors). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.5.)*

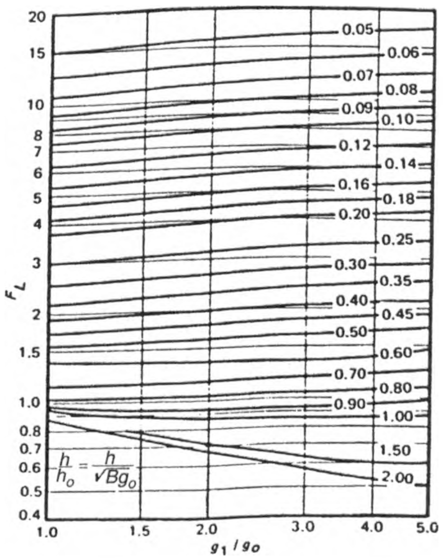


Figure 3-11. Values of F_L (loose hub flange factors). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.4.)*

Table 3-7
Number and Size of Bolts for Flanged Joints

Primary Service Pressure Rating	Bolting	Flange Facing	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12	14	16	18	20	24
			Nominal Pipe Size																			
150 Pound	Number		4	4	4	4	4	4	4	4	8	8	8	8	8	12	12	12	16	16	20	20
	Diameter		½	½	½	½	½	⅝	⅝	⅝	⅝	⅝	¾	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1¼
	Length of Stud Bolts	1/16" RF	2¼	2¼	2½	2½	2¾	3	3¼	3½	3½	3½	3¾	3¾	4	4½	4½	5	5¼	5¾	6	6¾
		RTJ			3	3	3¼	3½	3¾	4	4	4	4¼	4¼	4½	5	5	5½	5¾	6¼	6½	7¼
	Length of Mach, Bolts	1/16" RF	1¾	2	2	2¼	2¼	2¾	3	3	3	3	3¼	3¼	3½	3¾	4	4¼	4½	4¾	5¼	5¾
300 Pound	Number		4	4	4	4	4	8	8	8	8	8	8	12	12	16	16	20	20	24	24	24
	Diameter		½	⅝	⅝	⅝	¾	⅝	¾	¾	¾	¾	¾	¾	⅞	1	1⅛	1⅛	1¼	1¼	1¼	1½
	Length of Stud Bolts	1/16" RF	2½	2¾	3	3	3½	3¼	3¾	4	4¼	4¼	4½	4¾	5¼	6	6½	6¾	7¼	7½	8	9
		RTJ	3	3¼	3½	3½	4	4	4½	4¾	5	5	5¼	5½	6	6¾	7¼	7½	8	8¼	8¾	10
	Length of Mach, Bolts	1/16" RF	2	2½	2½	2¾	3	3	3¼	3½	3¾	3¾	4	4¼	4¾	5¼	5¾	6	6½	6¾	7	7¾
400 Pound	Number		4	4	4	4	4	8	8	8	8	8	8	12	12	16	16	20	20	24	24	24
	Diameter		½	⅝	⅝	⅝	¾	⅝	¾	¾	⅞	⅞	⅞	⅞	1	1⅛	1¼	1¼	1⅜	1⅜	1½	1¾
	Length of Stud Bolts	¼" RF	3	3¼	3½	3¾	4	4	4½	4¾	5¼	5¼	5½	5¾	6½	7¼	7¾	8	8½	8¾	9½	10½
		RTJ	3	3¼	3½	3¾	4	4¼	4¾	5	5½	5½	5¾	6	6¾	7½	8	8¼	8¾	9	9¾	11
		M&F, T&G	2¾	3	3¼	3½	3¾	3¾	4¼	4½	5	5	5¼	5½	6¼	7	7½	7¾	8¼	8½	9¼	10¼
600 Pound	Number		4	4	4	4	4	8	8	8	8	8	8	12	12	16	20	20	20	20	24	24
	Diameter		½	⅝	⅝	⅝	¾	⅝	¾	¾	⅞	⅞	1	1	1⅛	1¼	1¼	1⅜	1½	1⅝	1⅝	1⅞
	Length of Stud Bolts	¼" RF	3	3¼	3½	3¾	4	4	4½	4¾	5¼	5½	6¼	6½	7½	8¼	8½	9	9¾	10½	11¼	12¾
		RTJ	3	3¼	3½	3¾	4	4¼	4¾	5	5½	5¾	6½	6¾	7¾	8½	8¾	9¼	10	10¾	11½	13¼
		M&F, T&G	2¾	3	3¼	3½	3¾	3¾	4¼	4½	5	5¼	6	6¼	7¼	8	8¼	8¾	9½	10¼	11	12½
900 Pound	Number		4	4	4	4	4	8	8	8		8	8	12	12	16	20	20	20	20	20	20
	Diameter		¾	¾	⅞	⅞	1	⅞	1	⅞		1⅛	1¼	1⅛	1⅜	1⅜	1⅜	1½	1⅝	1⅞	2	2½
	Length of Stud Bolts	¼" RF	4	4¼	4¾	4¾	5¼	5½	6	5½		6½	7¼	7½	8½	9	9¾	10½	11	12¾	13½	17
		RTJ	4	4¼	4¾	4¾	5¼	5¾	6¼	5¾		6¾	7½	7½	8¾	9¼	10	11	11½	13¼	14	17¾
		M&F, T&G	3¾	4	4½	4½	5	5¼	5¾	5¼		6¼	7	7¼	8¼	8¾	9½	10¼	10¾	12¼	13¼	16¾
1500 Pound	Number		4	4	4	4	4	8	8	8		8	8	12	12	12	16	16	16	16	16	16
	Diameter		¾	¾	⅞	⅞	1	⅞	1	1⅛		1¼	1½	1⅜	1⅝	1⅞	2	2¼	2½	2¾	3	3½
	Length of Stud Bolts	¼" RF	4	4¼	4¾	4¾	5¼	5½	6	6¾		7½	9½	10	11¼	13¼	14¾	16	17½	19¼	21	24
		RTJ	4	4¼	4¾	4¾	5¼	5¾	6¼	7		7¾	9¾	10¼	11¾	13½	15¼	16¾	18½	20¼	22¼	25½
		M&F, T&G	3¾	4	4½	4½	5	5¼	5¾	6½		7¼	9¼	9¾	11	13	14½	15¾	17¼	19	20¾	23¾
2500 Pound	Number		4	4	4	4	4	8	8	8		8	8	8	12	12	12					
	Diameter		¾	¾	⅞	1	1⅛	1	1⅛	1¼		1½	1¾	2	2	2½	2¾					
	Length of Stud Bolts	¼" RF	4¾	4¾	5¼	5¾	6½	6¾	7½	8½		9¾	11½	13½	15	19	21					
		RTJ	4¾	4¾	5¼	6	6¾	7	7¾	8¾		10¼	12¼	14	15½	20	22					
		M&F, T&G	4½	4½	5	5½	6¼	6½	7¼	8¼		9½	11¼	13¼	14¾	18¾	20¾					

DERIVATION OF FLANGE MAXIMUM ALLOWABLE PRESSURE

	1. Calculate Moments M_1 through M_5 as follows:	@ Design Temperature	@ Ambient Temperature
INTEGRAL TYPE	$M_1 = (\text{Lesser of } 1.5\, S_{fo} \text{ or } 2.5\, S_{fa})\dfrac{\lambda g_1^2 B}{f}$		
	$M_2 = \dfrac{S_{fo}\lambda B t^2}{1.33te + 1}$		
	$M_3 = \dfrac{S_{fo}\lambda B t^2}{Y\lambda - Z(1.33te + 1)}$		
	$M_4 = \dfrac{2S_{fo}\lambda B t^2 g_1^2}{ft^2 + (1.33te + 1)g_1^2}$		
	$M_5 = \dfrac{2S_{fo}\lambda B t^2 g_1^2}{ft^2 + Y\lambda g_1^2 - Z(1.33te + 1)g_1^2}$		
	M_{MAX} = Lesser of M_1 thru M_5		
LOOSE TYPE	1. Calculate the Maximum Allowable Moment	@ Design Temperature	@ Ambient Temperature
	$M_{MAX} = \dfrac{S_{fo} t^2 B}{Y}$		

2.

$$A_{m(MAX)} = \frac{2M_{MAX}(@\text{Ambient Temperature})}{h_G S_a} - A_b$$

Note : If $A_{m2} > A_{m\,(MAX)}$, then the gasket width, seating stress, or bolting is insufficient.

3. Determine the Maximum Allowable Pressure set by the Maximum Allowable Moment: (*Operating Condition*)

$$\frac{M_{MAX}(@\text{Design Temperature})}{0.785B^2h_D + 6.28bGmh_G + 0.785(G^2 - B^2)h_T}$$

4. Determine the Maximum Allowable Pressure set by $A_{m(MAX)}$: (*Gasket Seating*)

$$\frac{S_b A_{m(MAX)}}{6.28bGm + 0.785G^2}$$

5. The Maximum Allowable Pressure = the lesser of 3. or 4.

Note that this pressure includes any static head applicable for the case under consideration.

Maximum Allowable Pressure = ______________________________

MAWP is based on corroded condition at design temperature.
When M_{MAX} is governed by M_2: Check integral type flange for new & uncorroded condition.
MAP (cold & corroded) is based on corroded condition @ ambient temperature.
MAP (new & cold) is based on new condition @ ambient temperature.

Procedure 3-2: Design of Spherically Dished Covers

1 DESIGN CONDITIONS

		Allowable Stresses			
		Flange		Bolting	
Design pressure, P					
Design temperature					
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Amb. temp., S_{fa}		Amb. temp., S_a	

2 GASKET AND FACING DETAILS

Gasket		Facing	

3 / **4 LOAD AND BOLT CALCULATIONS**

3		4			
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5 GASKET WIDTH CHECK

$N_{min} = A_bS_a/2y\pi G$	

6 MOMENT CALCULATIONS

Load		× Lever Arm		= Moment	
		Operating			
$H_D = \pi B^2P/4$		$h_D = 0.5(C - B)$		$M_D = H_Dh_D$	
$H_G = H_P$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(h_D + h_G)$		$M_T = H_Th_T$	
$H_r = H_D/\tan \beta_1$		h_r		$M_r = H_rh_r$	
		β Calculation			
$\beta_1 = \arcsin \dfrac{B}{2L + t}$		$M_o = M_D + M_G + M_T \pm M_r$ *Note:* M_r is (+) if ℄ of head is below the center of gravity; (−) if above.			
		Seating			
$H_G = W$		h_G		$M_o' = Wh_G$	

7 FLANGE AND HEAD THICKNESS CALCULATION

Head Thickness Required	
$t = \dfrac{5PL}{6S}$	
Flange Thickness Required	
$F = \dfrac{PB\sqrt{4L^2 - B^2}}{8S_{fo}(A - B)}$	
$J = \dfrac{M}{S_{fo}B}\left(\dfrac{A + B}{A - B}\right)$ where $M = M_o$ or M_o', whichever is greater	
$T = F + \sqrt{F^2 + J}$	

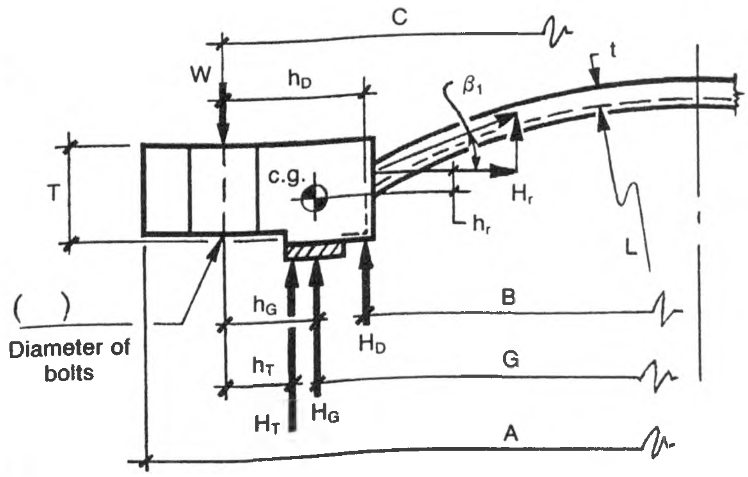


Figure 3-12. Dimensional data and forces for a spherically dished cover.

SPHERICALLY DISHED COVER

1	DESIGN CONDITIONS				
Design pressure, P	150 PSIG	Allowable Stresses (PSI)			
Design temperature	400 deg F	Flange		Bolting	
Flange material	SA-285-C	Design temp., S_{fo}	13,750	Design temp., S_b	25,000
Bolting material	SA-193-B7	Amb.temp., S_{fa}	13,750	Amb.temp., S_a	25,000

2	GASKET AND FACING DETAILS		
Gasket	.125 Thk DJAF	Facing	RF

3		4	LOAD AND BOLT CALCULATIONS		
N	1.00	$W_{m2} = b\pi Gy$	265,939	A_m = greater of W_{m2}/S_a or W_{m1}/S_b	15.50
b	.353	$H_P = 2b\pi GmP$	53,187		
G	53.29	$H = G^2\pi P/4$	334,558	A_b	20.38
y	4600 PSI	$W_{m1} = H_P + H$	387,745	$W = 0.5(A_m + A_b)S_a$	448,500
m	3.00				

5	GASKET WIDTH CHECK
$N_{min} = A_b S_a / 2y\pi G$	.33

6	MOMENT CALCULATIONS				
Load		x Lever Arm		= Moment	
Operating					
$H_D = \pi B^2 P/4$	321,628	$h_D = 0.5(C-B)$	2.50	$M_D = H_D h_D$	804,070
$H_G = H_P$	53,187	$h_G = 0.5(C-G)$	1.98	$M_G = H_G h_G$	105,310
$H_T = H - H_D$	12,930	$h_T = 0.5(h_D + h_G)$	2.24	$M_T = H_T h_T$	28,963
$H_r = H_D/\tan\beta_1$	547,325	h_r	1.375	$M_r = H_r h_r$	(−)752,572
β Calculation					
$\beta_1 = \arcsin \dfrac{B}{2L+t}$	30.44 deg	$M_o = M_D + M_G + M_T \pm M_r$ *Note:* M_r is (+) if ℄ of head is below the center of gravity; (−) if above.		185,771	
Seating					
$H_G = W$	448,500	h_G	1.98	$M_o' = Wh_G$	888,030

7 FLANGE AND HEAD THICKNESS CALCULATION	
Head Thickness Required	
$t = \dfrac{5PL}{6S}$	464+.125=.589 Use .625
Flange Thickness Required	
$F = \dfrac{PB\sqrt{4L^2-B^2}}{8S_{fo}(A-B)}$	.64
$J = \dfrac{M}{S_{fo}B}\left(\dfrac{A+B}{A-B}\right)$ where $M = M_o$ or M_o', whichever is greater	14.475
$T = F + \sqrt{F^2+J}$	4.498 Use 4.50

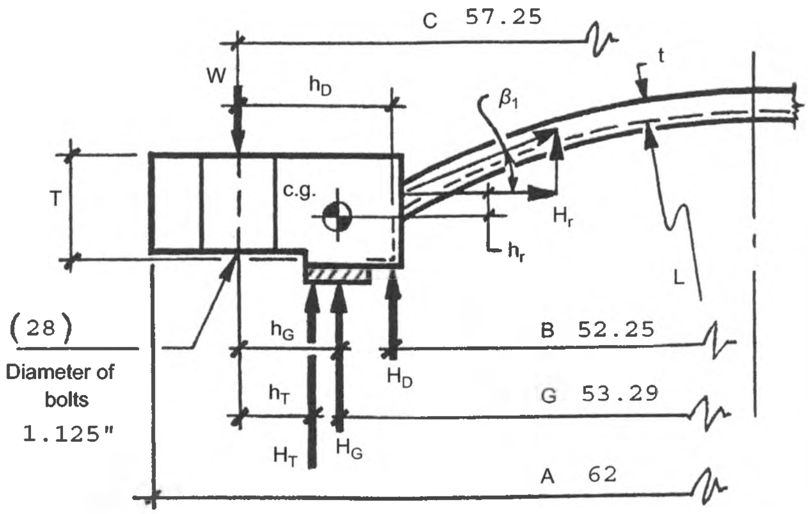


Figure 3-13. Dimensional and forces for a spherically dished cover.

Procedure 3-3: Design of Blind Flanges with Openings [1,4]

1 DESIGN CONDITIONS

		Allowable Stresses			
		Flange		Bolting	
Design pressure, P					
Design temperature					
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2 GASKET AND FACING DETAILS

Gasket		Facing	

3 / **4 LOAD AND BOLT CALCULATIONS**

N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G (see below)		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m				$h_G = 0.5(C - G)$	

5 THICKNESS AND REINFORCEMENT CALCULATIONS

Dimension, G

If $b_o \le 0.25$ in., G = mean gasket diameter	
If $b_o > 0.25$ in., G = lesser of raised face diameter or gasket O.D. − 2b	

Thickness Required

Operating, t_o [1, UG-34(c)(2)] (See Note 1)		Seating, t_G	
$t_o = G\sqrt{\frac{0.3P}{S_{fo}} + \frac{1.9W_{m1}h_G}{S_{fo}G^3}}$		$t_G = G\sqrt{\frac{1.9Wh_G}{S_{fa}G^3}}$	

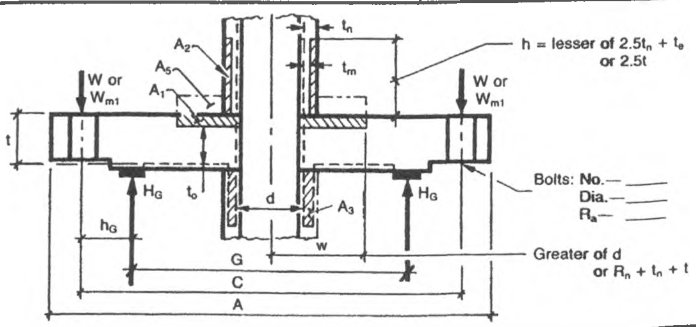


Figure 3-14. Dimensional data and forces for a blind flange.

Reinforcement

$t_m = \frac{PR_n}{SE - 0.6P}$		$A_3 = 2t_nh$	
		A_4 = area of welds	
$A_r = 0.5dt_o$		$A_5 = t_e$(O.D. pad − O.D. nozzle)	
$A_1 = (t - t_o)(2w - d)$		$\Sigma A = A_1$ through A_5	
$A_2 = 2h(t_n - t_m)$		$\Sigma A > A_r$	

Notes

1. Reinforcement is only required for operating conditions not bolt up.
2. Options in lieu of calculating reinforcement:
   Option 1—No additional reinforcement is required if flange thickness is greater than 1.414 t_o.
   Option 2—If opening exceeds one-half the nominal flange diameter, the flange may be computed as an optional-type reducing flange.
   Option 3—No additional reinforcement is required if t_o is calculated substituting 0.6 for 0.3 in the equation for t_o (doubling of c value).

BLIND FLANGE WITH OPENING

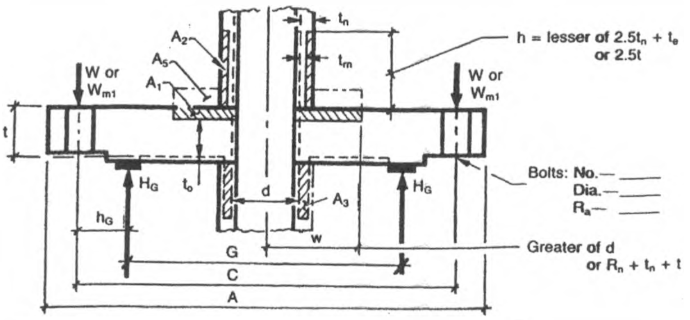
1	DESIGN CONDITIONS				
Design pressure, P	293 PSIG	Allowable Stresses			
Design temperature	500 deg F	Flange		Bolting	
Flange material	SA-105	Design temp., S_{fo}	17,500	Design temp., S_b	25,000
Bolting material	SA-193-B7	Atm. temp., S_{fa}	17,500	Atm. temp., S_a	25,000
Corrosion allowance	.118				

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N	1.315	$W_{m2} = b\pi Gy$	185,670	A_m = greater of W_{m2}/S_a or W_{m1}/S_b	17.55
b	.405	$H_P = 2b\pi GmP$	80,867		
G (see below)	39.44	$H = G^2\pi P/4$	357,957	A_b	84.26
y	3700	$W_{m1} = H_P + H$	438,824	$W = 0.5(A_m + A_b)S_a$	1,280,162
m	2.75			$h_G = 0.5(C - G)$	3.28

5	THICKNESS AND REINFORCEMENT CALCULATIONS
Dimension, G	
If $b_o \leq 0.25$ in., G = mean gasket diameter	40.25-2(.405)=39.44
If $b_o > 0.25$ in., G = lesser of raised face diameter or gasket O.D. − 2b	

Thickness Required			
Operating, t_o [1, UG-34(c)(2)] (See Note 1)		Seating, t_G	
$t_o = G\sqrt{\frac{0.3P}{S_{fo}} + \frac{1.9W_{m1}h_G}{S_{fo}G^3}}$	3.426	$t_G = G\sqrt{\frac{1.9Wh_G}{S_{fa}G^3}}$	3.40



A = 50 t = 4.12
C = 46
G = 39.44

Figure 3-15. Dimensional data and forces for a blind flange.

(32)2"dia studs
Ra = 2.625

Reinforcement			
$t_m = \frac{PR_n}{SE - 0.6P}$	.083	$A_3 = 2t_nh$	0
		A_4 = area of welds	0
$A_r = 0.5dt_o$	12.71	$A_5 = t_e$(O.D. pad − O.D. nozzle)	5.99
$A_1 = (t - t_o)(2w - d)$	5.31	$\Sigma A = A_1$ through A_5	13.01
$A_2 = 2h(t_n - t_m)$	1.71	$\Sigma A > A_r$	OK

Alternate Reinforcement check
1. t > 1.414 X 3.426 = 4.84 No Good!
2. Calculate t using C = .6 = 4.42" No Good!
Area of pad required = Ar-A1-A2 = 5.62 sq in
Use 16" OD X .813" Thk
A5 = (16-8.625) .813 = 5.99 Sq in

Procedure 3-4: Bolt Torque Required for Sealing Flanges [5–7]

Notation

A_b = cross-sectional area of bolts, in.^{2}
A_g = actual joint-contact area of gasket, in.^{2}
b = effective gasket seating width, in.
d = root diameter of threads, in.
d_m = pitch diameter of threads, in.
G = diameter at location of gasket load reaction, in.
M = external bending moment, in.-lb
m = gasket factor
N = gasket width, in.
n = number of bolts
E_b = modulus of elasticity of bolting material at temperature, psi
E_g = modulus of elasticity of gasket material at temperature, psi
P = internal pressure, psi
P_e = equivalent pressure including external loads, psi
P_r = radial load, lb
P_T = test pressure, psi
F = restoring force of gasket (decreasing compression force) from initial bolting strain, lb
F_{bo} = initial tightening force, lb
L_b = effective length of bolt, mid nut to mid nut, in.
W = total tightening force, lb
W_{ml} = $H + H_p$ = required bolt load, operating, lb
W_{m2} = required bolt load, gasket seating, lb
y = gasket unit seating load, psi
H = total hydrostatic end force, lb
H_P = total joint-contact surface compression load, lb
T = initial tightening torque required, ft-lb
t_g = thickness of gasket, in.
t_n = thickness of nut, in.
K = total friction factor between bolt/nut and nut/flange face
w = width of ring joint gasket, in.

Note: See Procedure 3-1 for values of G, N, m, b, and y.

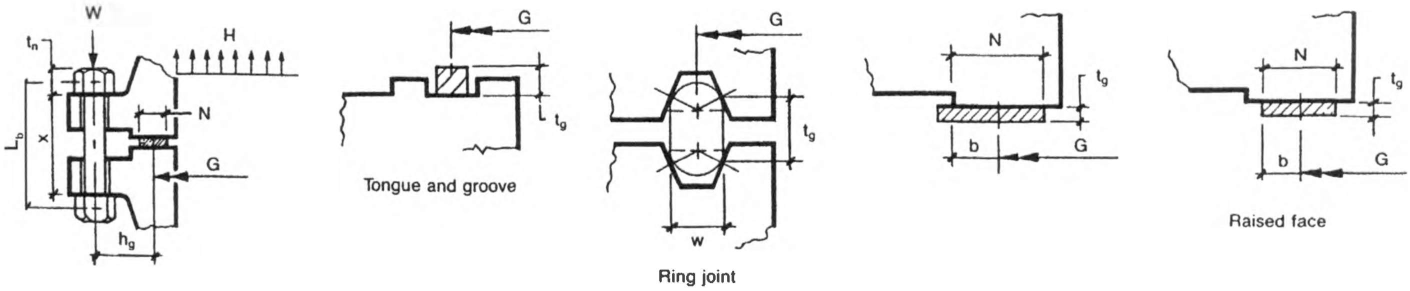


Figure 3-16. Flange and joint details.

Table 3-8
Bolting Dimensional Data

Size	¾ in.	⅞ in.	1 in.	1⅛ in.	1¼ in.	1⅜ in.	1½ in.	1⅝ in.
d	0.6273	0.7387	0.8466	0.9716	1.0966	1.2216	1.3466	1.4716
d_m	0.6850	0.8028	0.9188	1.0438	1.1688	1.2938	1.4188	1.5438
t_n	0.7344	0.8594	0.9844	1.1094	1.2188	1.3438	1.4688	1.5938
Size	1¾ in.	1⅞ in.	2 in.	2¼ in.	2½ in.	2¾ in.	3 in.	3¼ in.
d	1.5966	1.7216	1.8466	2.0966	2.3466	2.5966	2.8466	3.0966
d_m	1.6688	1.7938	1.9188	2.1688	2.4188	2.6688	2.9188	3.1688
t_n	1.7188	1.8438	1.9688	2.2031	2.4531	2.7031	2.9531	3.1875

Note: 3/4 and 7/8 in. bolts are UNC series threads. All others are 8 series threads. All dimensions are from ANSI B18.2.

Table 3-9
Modulus of Elasticity, E_b, 10^6 psi

Material	Temperature, °F								
	70°	200°	300°	400°	500°	600°	700°	800°	900°
Carbon steel A-307-B	27.9	27.7	27.4	27.0	26.4	25.7	24.8	23.4	18.5
Low alloy A-193-B7, B16, B7M	29.9	29.5	29.0	28.6	28.0	27.4	26.6	25.7	24.5
Straight chrome A-193-B6, B6X	29.2	28.7	28.3	27.7	27.0	26.0	24.8	23.1	22.1
Stainless A-193-B8 series	28.3	27.7	27.1	26.6	26.1	25.4	24.8	24.1	23.4

Note: Values per ASME Code, Section II.

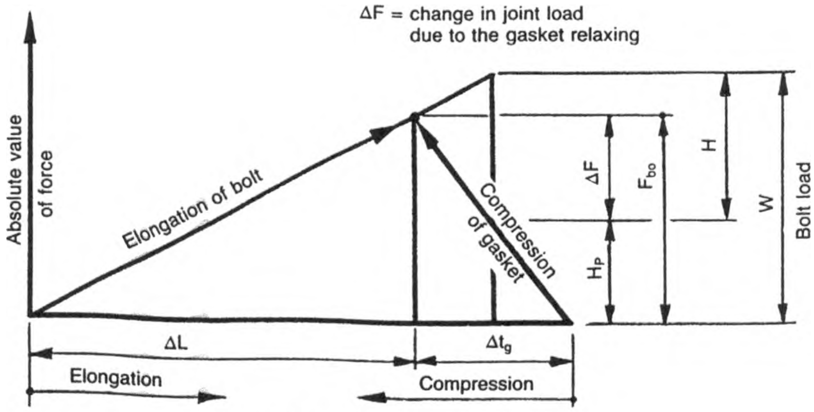


Figure 3-17. Typical joint diagram.

Design Data		Gasket Data		Bolting Data	
Flange size		Type		Nominal size	
Design pressure, P		Diameter of raised face		Quantity n	
Test pressure, P_T		O.D., I.D.		d	
		N or w		d_m	
Moment, M		y		E_b	
Radial load, Pr		m		$Ab = \frac{\pi d^2 n}{4}$	
		Eg			
Friction factor, K		tg		$\ell_b = x + t_n$	
Design temperature		b			
		G			

Modulus of Elasticity of Gasket Material, E_g

- Ring joint and flat metal: Select values from ASME Section II, or Appendix K of this book.
- Comp asb = 70 ksi
- Rubber = 10 ksi
- Grafoil = 35 ksi
- Teflon = 24 ksi
- Spiral wound = 569 ksi

Friction Factor, K

- Lubricated = 0.075–0.15
- Nonlubricated = 0.15–0.25

Calculations

- *Equivalent pressure, P_e, psi.*

$$P_e = \frac{16M}{\pi G^3} + \frac{4P_r}{\pi G^2} + P$$

- *Hydrostatic end force, H, lb.*

$$H = \frac{\pi G^2 P_e}{4}$$

- *Total joint-contact-surface compression load, H_p, lb.*

$$H_P = 2b\pi GmP_e$$

- *Minimum required bolt load for gasket seating, W_{m2}, lb.*

$$W_{m2} = \pi bGy$$

- *Actual joint area contact for gasket, A_g, in.^{2}*

$$A_g = 2\pi bG$$

- *Decreasing compression force in gasket, ΔF, lb.*

$$\Delta F = \frac{H}{1 + \frac{A_b E_b t_g}{A_g E_g L_b}}$$

- *Initial required tightening force (tension), F_{bo}, lb.*

$$F_{bo} = H_p + \Delta F$$

- *Total tightening force required to seal joint, W, lb.*

$$W = \text{greater of } F_{bo} \text{ or } W_{m2}$$

- *Required torque, T, ft-lb.*

$$T = \frac{KWd_m}{12n}$$

Notes

1. Bolted joints in high-pressure systems require an initial preload to prevent the joint from leaking. The loads which tend to open the joint are:
   a. Internal pressure.
   b. Thermal bending moment.
   c. Dead load bending moment.
2. Either stud tensioners or torque wrenches are used for prestressing bolts to the required stress for gasket seating. Stud tensioners are by far the most accurate. Stud tension achieved by torquing the nut is affected by many variables and may vary from 10% to 100% of calculated values. The following are the major variables affecting tension achieved by torquing:
   a. Class of fit of nut and stud.
   b. Burrs.
   c. Lubrication.
   d. Grit, chips, and dirt in threads of bolts or nuts.
   e. Nicks.
   f. The relative condition of the seating surface on the flange against which the nut is rotated.
3. Adequate lubrication should be used. Nonlubricated bolting has an efficiency of about 50% of a well-lubricated bolt. For standard applications, a heavy graphite and oil mixture works well. For high temperature service (500°F to 1000°F), a high temperature thread compound may be used.
4. The stiffness of the bolt is only 1/3 to 1/5 that of the joint. Thus, for an equal change in deformation, the change of the load in the bolt must be only 1/3 to 1/5 of the change in the load of the joint.
5. Joints almost always relax after they have first been tightened. Relaxation of 10% to 20% of the initial preload is not uncommon. Thus an additional preload of quantity F is required to compensate for this "relaxing" of the joint.

Table 3-10
Bolt torques
Torque required in ft-lb to produce the following bolt stress

Bolt Size	15 ksi	30 ksi	45 ksi	60 ksi
½–13	15	30	45	60
⅝–11	30	60	90	120
¾–10	50	100	150	200
⅞–9	80	160	240	320
1–8	123	245	368	490
1⅛–8	195	390	533	710
1¼–8	273	500	750	1000
1⅜–8	365	680	1020	1360
1½–8	437	800	1200	1600
1⅝–8	600	1100	1650	2200
1¾–8	775	1500	2250	3000
1⅞–8	1050	2000	3000	4000
2–8	1125	2200	3300	4400
2¼–8	–	3180	4770	6360
2½–8	–	4400	6600	8800
2¾–8	–	5920	8880	11840
3–8	–	7720	11580	15440

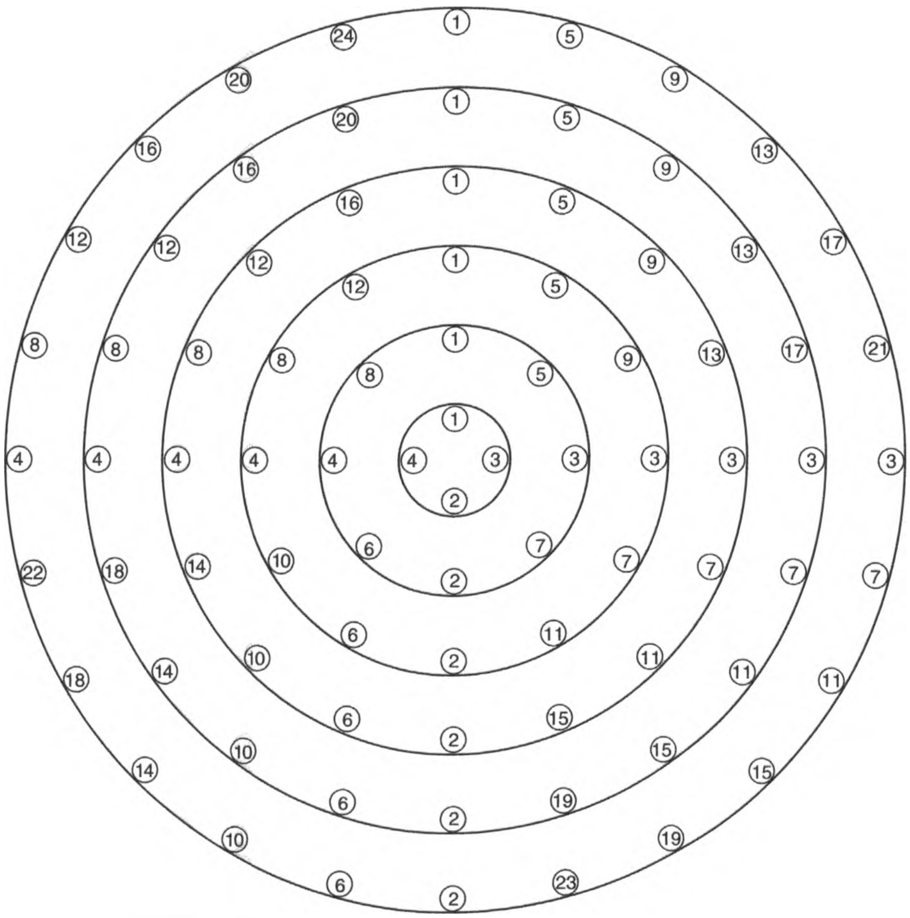


Figure 3-18. Sequence for tightening of flange bolts. *Note:* Bolts should be tightened to 1/3 of the final torque value at a time in the sequence illustrated in the figure. Only on the final pass is the total specified torque realized.

Procedure 3-5: Design of Studding Outlets

The calculations for a studding outlet will vary from standard flange calculations because these flanges are not hubbed or hubless. However many of the loads and moments calculated in a standard flange still apply. The bolt load calculations and gasket calculations are identical.

A studding outlet has other loads that combine with those due to pressure, gasket and bolting loads. The attachment of the shell directly to the studding outlet will cause a moment in the flange ring. This moment acts about the center of gravity of the flange ring and is additive to the other moments.

Another difference is that in a hubbed flange, the hydrostatic end force, H_D would be applied at the centroid of the hub. Since studding outlets are hubless, the load is transferred to the ID of the flange, just as in a spherically dished cover. This load is applied from the centroid of the flange area, and not from the bolt circle.

For studding outlets installed in cylindrical shells, there are two cases to be considered, the circumferential axis case and the longitudinal axis case. Normally, the worst case is the circumferential axis case because the circumferential load from the pressure stress is twice that of the longitudinal case. A check may be made of the longitudinal axis by reducing force T_2 in half and recalculating h_r from the center of shell to centroid of the flange on the longitudinal axis.

For a studding outlet in a sphere or the spherical portion of a head, the loading would be the same in either axis. Therefore the procedure would be identical except that the pressure loading, T_2 would be reduced to .5 PR.

STUDDING OUTLET

1	DESIGN CONDITIONS			
		Allowable Stresses		
		Flange		Bolting
Design pressure, P				
Design temperature				
Flange material		Design temp., S_{fo}		Design temp., S_b
Bolting material		Amb. temp., S_{fa}		Amb. temp., S_a

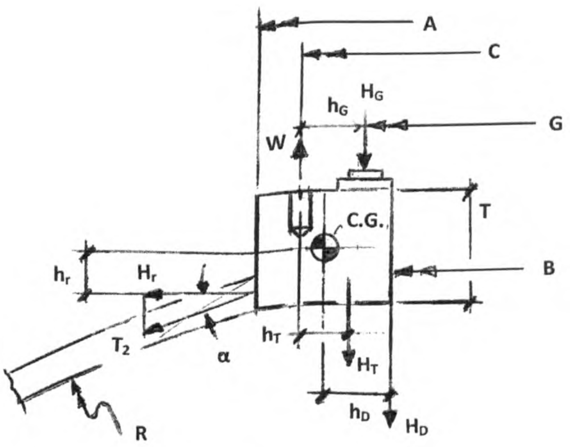
2	GASKET AND FACING DETAILS		
Gasket		Facing	

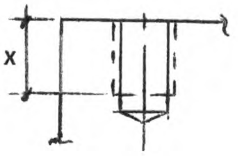
3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = .5(A_m + A_b)S_a$	
m					

5	GASKET WIDTH CHECK
$N_{min} = A_bS_a/2y\pi G$	

6	MOMENT CALCULATIONS				
Load	×	Lever Arm		=	Moment
		Operating			
$H_D = \pi B^2P/4$		$h_D = .5(A - B)$		$M_D = H_Dh_D$	
$H_g = W_{m1} - H$		$h_G = .5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = .5(h_D + h_G)$		$M_T = H_Th_T$	
$H_r = T_2 \cos\alpha$		h_r		$M_r = H_rh_r$	
$\alpha = \arcsin .5\ A/R$		$M_o = M_D + M_G + M_T +/- M_r$ Note: M_r is (+) if C.L. of shell/head is below C.G., (-) if above			
$T_2 = PR$					
		Seating			
$H_G = W$		h_G		$M_o' = Wh_G$	

7 FLANGE	THICKNESS CALCULATION
Thickness required- Gasket seating	
$T_G = \sqrt{\dfrac{M_o'}{S_{fa}B}\left[\dfrac{A+B}{A-B}\right]}$	
Thickness required- Operating	
$F = \dfrac{PB\sqrt{4L^2 - B^2}}{8S_{fo}(A - B)}$	
$J = \dfrac{M}{S_{fo}B}\left(\dfrac{A+B}{A-B}\right)$	
$T_o = F + \sqrt{F^2 + J}$	





X = Minimum depth of threads (UG-44(b))

$X = (.75\ d\ S_b) / S_{fo} > 1.5\ d$

TYPE 4: REVERS FLANGE DESIGN

1 DESIGN CONDITIONS

		Allowable Stresses			
Design pressure, P	400 PSIG				
Design temperature	1,000 deg F	Flange		Bolting	
Flange material	SB-564-800H	Design temp., S_{fo}	14,000	Design temp., S_b	32,300
Bolting material	SB-637	Amb. temp., S_{fa}	16,700	Amb. temp., S_a	37,000

2 GASKET AND FACING DETAILS

Gasket	Spiral Wound	Facing	RF

3 / **4 LOAD AND BOLT CALCULATIONS**

N	1.00	$W_{m2} = b\pi Gy$	222,110	A_m = greater of W_{m2}/S_a or W_{m1}/S_b	6.00
b	.3535	$H_P = 2b\pi GmP$	53,306		
G	20″	$H = G^2\pi P/4$	125,663	A_b	6.612
y	10,000PSI	$W_{m1} = H_P + H$	178,970	$W = .5(A_m + A_b)S_a$	233,322
m	3.0				

5 GASKET WIDTH CHECK

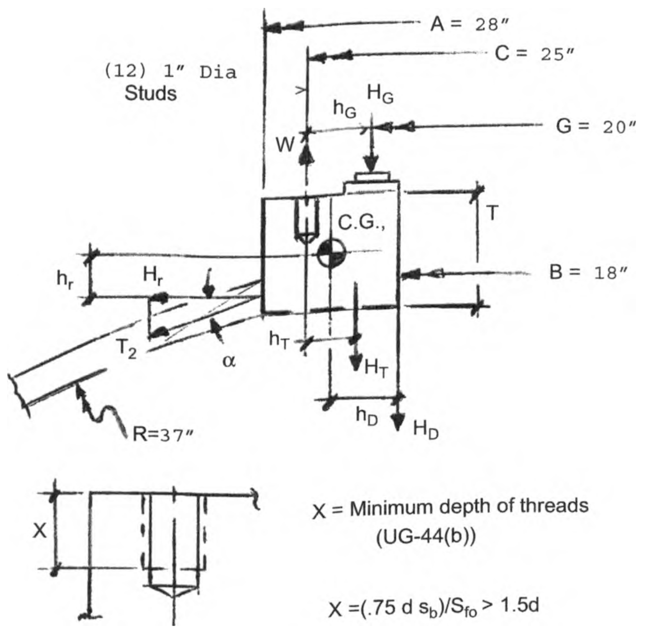
$N_{min} = A_b S_a / 2y\pi G$	.195

6 MOMENT CALCULATIONS

Load		X Lever Arm		= Moment	
Operating					
$H_D = \pi B^2 P/4$	101,787	$h_D = .5(A-B)$	2.50	$M_D = H_D h_D$	254,468
$H_g = W_{m1} - H$	53,306	$h_G = .5(C-G)$	2.50	$M_G = H_G h_G$	133,265
$H_T = H - H_D$	23,876	$h_T = .5(h_D - h_G)$	2.313	$M_T = H_T h_T$	55,213
$H_r = T_2 \cos\alpha$	13,700	h_r	4.20	$M_r = H_r h_r$	57,540
α = arc sin.5A/R	22.23 deg	$M_o = M_D + M_G + M_T +/- M_r$ Note: M_r is(+) if C.L.of shell/head is		500,486	
$T_2 = PR$	14,800	below C.G.,(-) if above			
Seating					
$H_G = W$	233,322	h_G	2.50	$M_o' = Wh_G$	583,305

7 FLANGE THICKNESS CALCULATION

Thickness required-Gasket seating	
$T_{\cdot G} = \sqrt{\dfrac{M_o'}{S_{fa}B}\left[\dfrac{A+B}{A-B}\right]}$	2.98″
Thickness required-Operating	
$F = \dfrac{PB\sqrt{4L^2-B^2}}{8S_{fo}(A-B)}$	.4614
$J = \dfrac{M}{S_{fo}B}\left(\dfrac{A+B}{A-B}\right)$	9.14
$T_o = F + \sqrt{F^2+J}$	3.52″



X = Minimum depth of threads (UG-44(b))

$X = (.75\ d\ s_b)/S_{fo} > 1.5d$

1.73 > 1.5″

OK

Procedure 3-6: Reinforcement for Studding Outlets

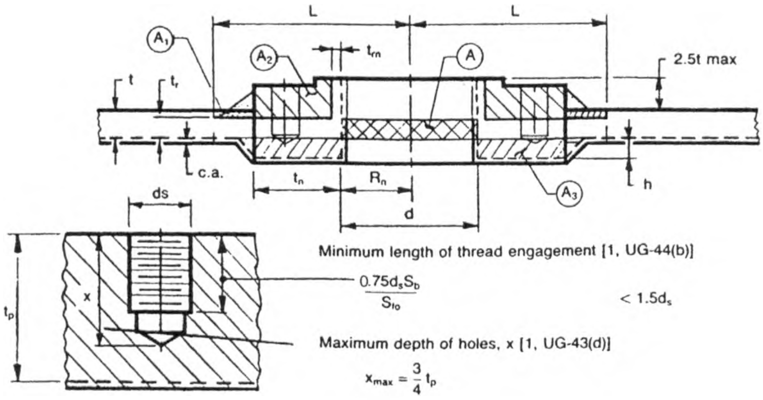


Figure 3-19. Typical studding outlet.

Table 3-11
Tapped Hole Area Loss, S, in.[2]*

d_s	⅝ in.	¾ in.	⅞ in.	1 in.	1⅛ in.	1¼ in.	1⅜ in.	1½ in.	1⅝ in.	1¾ in.	1⅞ in.	2 in.	2 ¼ in.
X	1.11	1.33	1.55	1.77	2.00	2.44	2.66	2.88	3.10	3.32	3.56	3.98	4.44
S	1.28	1.84	2.50	3.28	4.15	5.12	6.20	7.38	8.66	10.05	11.55	13.1	16.6

** Values of x and s are based on the shoulder of the hole at a depth of 1.5d_s and fully threaded.*

Calculation of Area of Reinforcement (Figure 3-19)

$$A = (dt_rF) + S$$

$$L = \text{Greater of } d \text{ or } R_n + t_n + t$$

$$A_1 = 2(L - R_n - t_n)(t - t_r)$$

$$A_2 = 2(t_p - h - t_r)(t_n - t_m)$$

$$A_3 = 2(ht_n)$$

$$A_T = A_1 + A_2 + A_3$$

Notes

1. Check plane which is nearest the longitudinal axis of the vessel and passes through a pair of studded holes.
2. S_b = allowable stress of stud material at design temperature.
   S_{fo} = allowable stress of flange material at design temperature.
3. A_2 as computed ignores raised face.

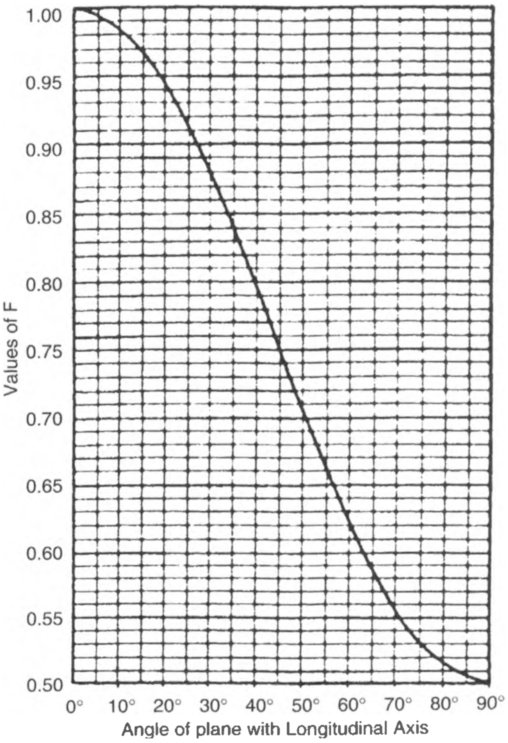


Figure 3-20. Chart for determining the value of F. *(Reprinted by permission from ASME Code, Section VIII, Div. 1, Figure UG-37.)*

Procedure 3-7: Studding Flanges

TYPE 6 : STUDDING FLANGE

1	DESIGN CONDITIONS				
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND DETAILS		
Gasket		Facing	

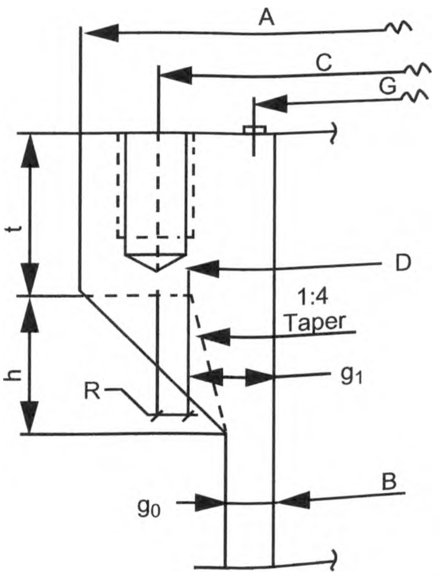
3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = .5(A_m + A_b)S_a$	
m					

5	MOMENT CALCULATIONS				
Load		x Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$		$h_D = R + .5g_1$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = .5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = .5(R + g_1 + h_G)$		$M_T = H_Th_T$	
				M_o	
Seating					
$H_G = W$		$h_G = .5(C - G)$		M_o'	

6	K AND HUB FACTORS		
$K = A/B$		h/h_o	
T		F	
Z		V	
Y		f	
U		$e = F/h_o$	
g_1/g_o		$d = \frac{U}{V} h_o g_o^2$	
$h_o = \sqrt{Bgo}$			

7	STRESS FORMULA FACTORS		
t			
$\alpha = te + 1$			
$\beta = 4/3\ te + 1$			
$\gamma = \alpha/T$			
$\delta = t^3/d$			
$\lambda = \gamma + \delta$			
$m_o = M_o/B$			
$m_G = M_o'/B$			

if bolt spacing exceeds 2a + t, multiply m_o and m_G in above equation by: $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$



8	STRESS CALCULATIONS				
Allowable Stress	Operating		Allowable Stress	Seating	
1.5 S_{fo}	Longitudinal hub, $S_H = fm_o/\lambda g t^2 = g_1^2$		1.5 S_{fa}	Longitudinal hub, $S_H = fm_o = m_G/\lambda g t^2 = g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda g t^2 = t^2$		S_{fa}	Radial flange, $S_R = \beta m_o = m_G/\lambda g t^2 = t^2$	
S_{fo}	Tangential flange, $S_T = m_oY/t^2 - ZS_R$		S_{fa}	Tangential flange, $S_T = m_GY/t^2 - ZS_R$	
S_{fo}	Greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$		S_{fa}	Greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	

Nomenclature

A_b = Total area of studs, in^2
A_m = Area of studs required, in^2
A_r = Area required for one stud, in^2
A_g = Actual gasket area, in^2
B_S = Stud spacing, in
d_S = Diameter of stud, in
f_{a1} = Load between b and r_i, Lbs/in
f_{a2} = Load between r_i and r_o, Lbs/in
f_{a3} = Load between r_o and a, Lbs/in
f_{an} = net load, Lbs/in
f_{aT} = Total load, Lbs/in
G_o, G_i = Gasket OD or ID
L = Minimum length of thread engagement, in
n = Number of studs
R_a = Root area of one stud, in^2
R_f = diameter of raised face, in
S_b = Allowable stress of stud at design temperature, PSI
S_{fo} = Allowable stress of flange material at design temperature, PSI
S_{if} = Hoop stress in flange at flange ID, PSI
S_{ri} = Hoop stress in flange at inner surface of main stud hole, PSI
S_{ro} = Hoop stress in flange at outer surface of main stud hole, PSI
S_{of} = Hoop stress at flange OD, PSI
S'_{if} = Corrected hoop stress at flange ID, PSI
Y_1 = Correction ratio

Establish Flange Dimensions

- Area of studs required, A_m

$$A_m = W_{m1}/S_b$$

- Area required for one stud, A_r

$$A_r = A_m/n$$

- Stud selection
  Qty, n = ________
  Dia, d_S = ________
  Root Area, R_a = ________
- Total area of studs, A_b

$$A_b = nR_a$$

- Minimum length of thread engagement, L
  Greater of ....

$$L = (.75\ d_s\ S_b)/S_{fo} \text{ or } 1.5d_S$$

- Depth of hole, F

$$F = L + .25''\text{Min}$$

- Depth of drill tip, C_O

$$C_O = .288d_S$$

- Determine bolt circle diameter, C
  Larger of ...

1. $C = B + d_S + 2''$
2. $C = R_f + d_S + 2''$
3. $C = (B_S\ n)/\pi$
   Use, C = ________

- Hub proportions;
  Assuming a theoretical 1:4 taper for the hub, find corresponding dimensions of h and g_1

$$g_1 = .25h + g_o$$
$$h = 4(g_1 - g_o)$$

Use, h = ________
And g_1 = ________

$$D = B + 2\,g_1$$
$$R = .5(C - D)$$

- Flange OD, A
  Larger of ...

1. $C + 2(d_S - .125'')$
2. $B + 2\,g_o + 2d_S$
   Use A = ________

- Minimum flange thickness, t_{min}

$$t_{min} = F + .5(a - r_o)$$

- Gasket dimensions,
  G_o = ______
  G_i = ______
  N = ______
  $b_o = N / 2$ = ______
  $b = b_o^{1/2} / 2$ = ______
  $G = G_o - 2\,b$ = ________
  R_f = ________
  Use OD of groove for M&F or T&G flange facing
- Gasket area, A_g

$$A_g = (\pi\,G_o{}^2)/4 - (\pi\,G_i^2)/4$$

- Gasket width check, N_{min}

$$N_{min} = (A_h\ S_a)/(2\,\pi\,y\,G)$$

WORKSHEET FOR STUDDING FLANGES

GIVEN		LOADS & STRESSES			
FLANGE OD, A OR A_1		$S_{if} = P[(a^2 + b^2) / (a^2 - b^2)]$		$f_{an} = f_{at} - [(F + .5C_O)f_{a2}/ t]$	
FLANGE ID, B					
BOLT CIRCLE, C		$S_{ri} = [Pb^2/ (a^2 - b^2)][1 + a^2 / r_i^2]$		$Y_1 = f_{at} / f_{an}$	
HYDROSTATIC END FORCE, H					
REQUIRED BOLT LOAD, OPERATING, W_{m1}		$S_{ro} = [Pb^2/ (a^2 - b^2)][1 + a^2 / r_o^2]$		$S'_{if} = Y_1 S_{if}$	
AREA OF GASKET, A_G					
DIAMETER OF STUDS, d_S		$S_{of} = (2Pb^2) / (a^2 - b^2)$		**NOTES:**	
PRESSURE, P				1. If S'_{if} </= Allowable Stress, than the design is OK	
THICKNESS, t		$f_{a1} = .5(r_i - b)(S_{if} + S_{ri})$		2. If S'_{if} > Allowable Stress, then implement a or b below	
DIMENSIONS					
$a = .5(A \text{ or } A_1)$		$f_{a2} = .5(r_o - r_i)(S_{ri} + S_{ro})$		a. Increase flange proportions	
$b = .5B$				b. Add a shrink ring	
$r_i = .5(C - d_S)$		$f_{a3} = .5(a - r_o)(S_{ro} + S_{of})$			
$r_o = .5(C + d_S)$					
$F = L + .25"$		$f_{aT} = f_{a1} + f_{a2} + f_{a3}$			
$C_O = .288d_S$					

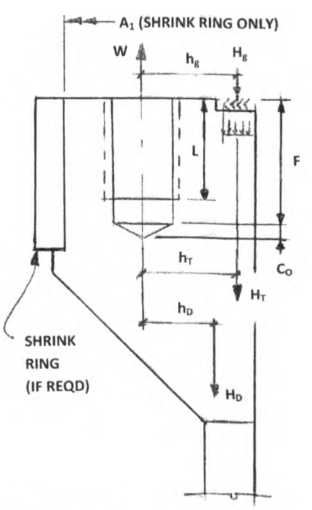


Figure 3.21. Dimensions & Forces

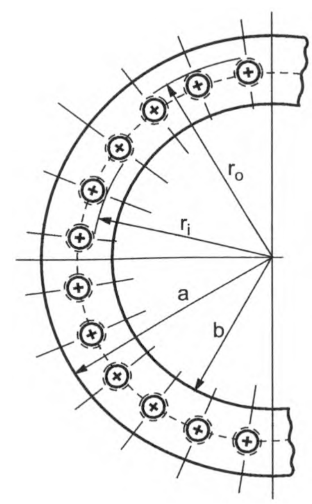


Figure 3.22. Plan View - Dimensions

TYPE 6 : STUDDING FLANGE

1	DESIGN CONDITIONS				
Design pressure, P	1700 PSIG	Allowable Stresses			
Design temperature	810 Deg F	Flange		Bolting	
Flange material	SA-182-F22	Design temp., S_{fo}	22,360	Design temp., S_b	19,260
Bolting material	SA-193-B7	Atm. temp., S_{fa}	25,000	Atm. temp., S_a	23,000
Corrosion allowance	0				

2	GASKET AND FACING DETAILS
Gasket	Facing

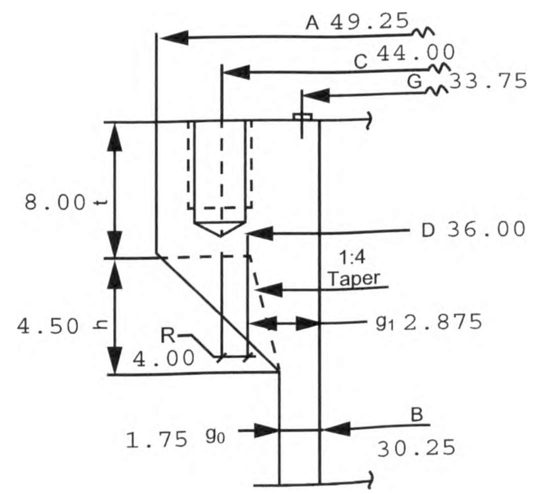
3		4	LOAD AND BOLT CALCULATIONS		
N	2	$W_{m2} = b\pi Gy$	530,143	A_m = greater of W_{m2}/S_a or W_{m1}/S_b	107.04
b	0.5	$H_p = 2b\pi GmP$	540,747		
G	33.75	$H = G^2\pi P/4$	1,520,850	A_b	126.2
y	10,000	$W_{m1} = H_p + H$	2,061,597	$W = .5(A_m + A_b)S_a$	2,682,260
m	3				

5	MOMENT CALCULATIONS				
Load		x Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$	1,221,770	$h_D = R + .5g_1$	5.438	$M_D = H_Dh_D$	6,643,375
$H_G = W_{m1} - H$	540,747	$h_G = 5(C - G)$	5.125	$M_G = H_Gh_G$	2,771,328
$H_T = H - H_D$	299,080	$h_T = .5(R + g_1 + h_G)$	6.00	$M_T = H_Th_T$	1,794,480
				M_O	11,209,183
Seating					
$H_G = W$	2,682,260	$h_G = .5(C - G)$	5.125	M'_O	13,746,583

6	K AND HUB FACTORS		
K = A/B	1.63	h/h_0	.618
T	1.65	F	.822
Z	2.21	V	.270
Y	4.15	f	1.00
U	4.56	$e = F/h_0$	.113
g_1/g_0	1.64	$d = \frac{U}{V}h_0g_0^2$	376
$h_0 = \sqrt{Bg_0}$	7.27		

7	STRESS FORMULA FACTORS
t	8.00
$\alpha = te + 1$	1.90
$\beta = 4/3\ te + 1$	2.21
$\gamma = \alpha/T$	1.15
$\delta = t^3/d$	1.36
$\lambda = \gamma + \delta$	2.51
$m_0 = M_0/B$	370,551
$m_G = M'_0/B$	454,432

if bolt spacing exceeds 2a + t, multiply m_o and m_G in above equation by: $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$



8	STRESS CALCULATION				
Allowable Stress	Operating		Allowable Stress	Seating	
1.5 S_{fo} 33,540	Longitudinal hub, $S_H = fm_o/\lambda g_1^2$	17,860	1.5 S_{fa} 37,500	Longitudinal hub, $S_H = fm_G/\lambda g_1^2$	21,903
S_{fo} 22,360	Radial flange, $S_R = \beta m_o/\lambda t^2$	5,097	S_{fa} 25,000	Radial flange, $S_R = \beta m_G/\lambda t^2$	6,251
S_{fo} 22,360	Tangential flange, $S_T = m_oY/t^2 - ZS_A$	12,763	S_{fa} 25,000	Tangential flange, $S_T = m_GY/t^2 - ZS_R$	15,650
S_{fo} 22,360	Greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	15,311	S_{fa} 25,000	Greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	18,776

WORKSHEET FOR STUDDING FLANGES

GIVEN		LOADS & STRESSES			
FLANGE OD, A OR A_1	49.25	$S_{if} = P\,[\,(a^2 - b^2)\,/\,(a^2 - b^2)\,]$	3760	$f_{an} = f_{at} - [(F + .5\,C_o)\,f_{a2}\,/\,t]$	22,346
FLANGE ID, B	30.25				
BOLT CIRCLE, C	44	$S_{ri} = [P\,b^2/\,(a^2 - b^2)]\,[1 + a^2/\,r_i^2\,]$	2498	$Y_1 = f_{at}\,/\,f_{an}$	1.18
HYDROSTATIC END FORCE, H	1,520,850				
REQUIRED BOLT LOAD, OPERATING, W_{m1}	2,061,597	$S_{ro} = [P\,b^2/\,(a^2 - b^2)]\,[1 + a^2/\,r_o^2\,]$	2172	$S'_{if} = Y_1\,S_{if}$	4421
AREA OF GASKET, A_G	205.77				
DIAMETER OF STUDS, d_S	2.75	$S_{of} = (2\,P\,b^2)\,/\,(\,a^2 - b^2)$	2060	**NOTES:**	
PRESSURE, P	1700			1. If S'_{if} </= Allowable Stress, than the design is OK	
THICKNESS, t	8	$f_{a1} = .5\,(\,r_i - b\,)\,(\,S_{if} + S_{ri}\,)$	17,210	2. If S'_{if} > Allowable Stress, then implement a or b below	
DIMENSIONS					
a = .5 (A or A_1)	24.625	$f_{a2} = .5\,(\,r_o - r_i\,)\,(\,S_{ri} + S_{ro}\,)$	6421	a. Increase flange proportions	
b = .5 B	15.125			b. Add a shrink ring	
$r_i = .5\,(C - d_S)$	20.625	$f_{a3} = .5\,(\,a - r_o\,)\,(\,S_{ro} + S_{of}\,)$	2645		
$r_O = .5\,(C + d_S)$	23.375				
F = L + .25"	4.5	$f_{aT} = f_{a1} + f_{a2} + f_{a3}$	26,276		
$C_O = .288\,d_S$	0.792				

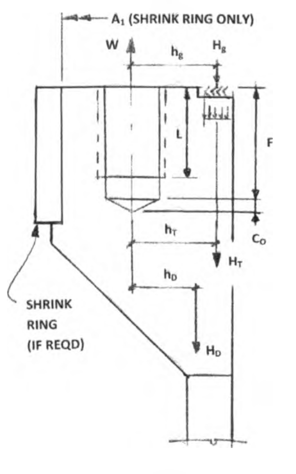


Figure 3.23. Dimensions & Forces

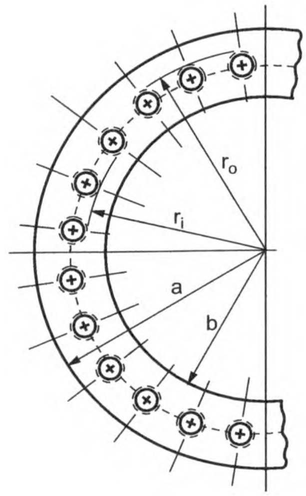
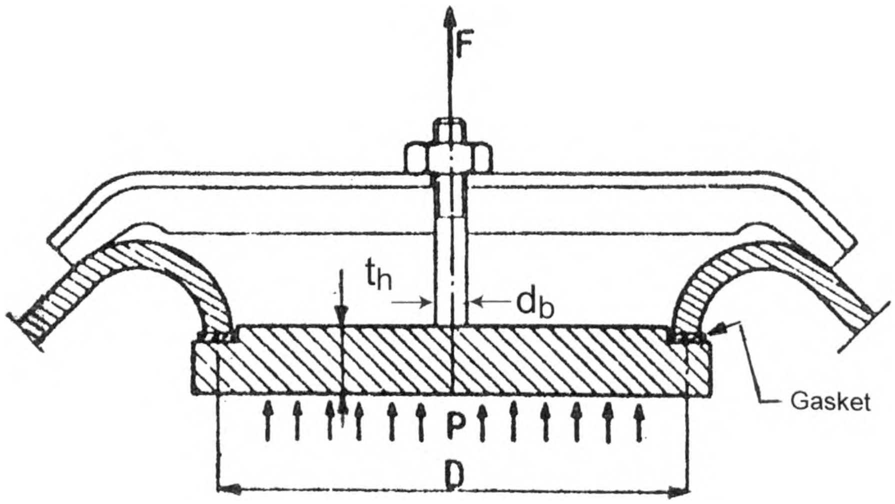


Figure 3.24. Plan View - Dimensions

Procedure 3-8: Design of Elliptical, Internal Manways

DESIGN OF ELLIPTICAL, INTERNAL MANWAYS



NOTATION

C = Attachment Factor

E = Joint Efficiency, Cat A seam only

F = Total force exerted by stud(s), Lbs

P = Design Pressure, PSIG

S = Allowable Stress, head, PSI

S_b = Allowable Stress, bolt, PSI

Z = Factor dependent on D/d ratio

CALCULATIONS

- Maximum force applied from stud(s), F

$$F = .785\ d_b^2\ S_b$$

- Determine attachment factor, C

$$C = \sqrt{0.31 + 0.95\ \frac{F}{PD^2}}$$

- Since head is non-circular, determine factor, Z

$$Z = 3.4 - (2.4d)\ /\ D; < 2.5$$

- Thickness required for head, t_h

$$t_h = d\sqrt{\frac{Z\ C\ P}{S\ E}}$$

EXAMPLE

D = 20" $\quad$ S_b = 25,000 PSI

d = 14" $\quad$ d_b = 1.25"

P = 750 PSIG $\quad$ E = 1.0

S = 20,000 PSI

$$F = .785\ (1.25^2)\ 25{,}000 = 30{,}664 \text{ Lbs}$$

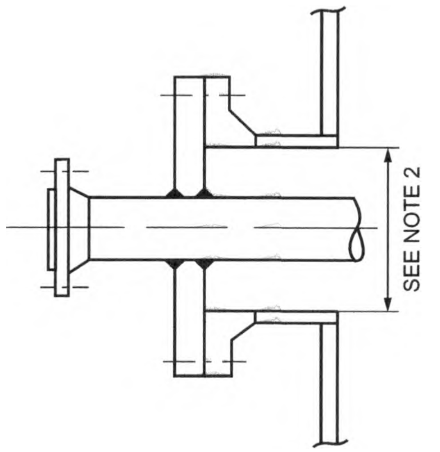
$$C = \sqrt{.31 + .95\ [\ 30{,}664\ /\ (750\ (20^2)\ ]} = .638$$

$$Z = 3.4 - [2.4(14)]\ /\ 20 = 1.72$$

$$t_h = 14\sqrt{\frac{1.72(.638)\ 750}{20{,}000\ (1.0)}} = 2.84"$$

Procedure 3-9: Through Nozzles

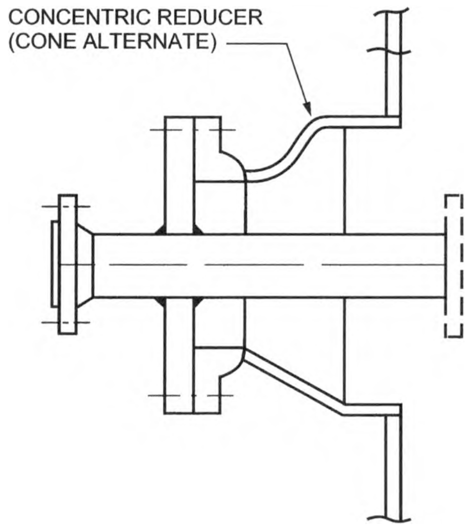
THROUGH NOZZLES



Notes:

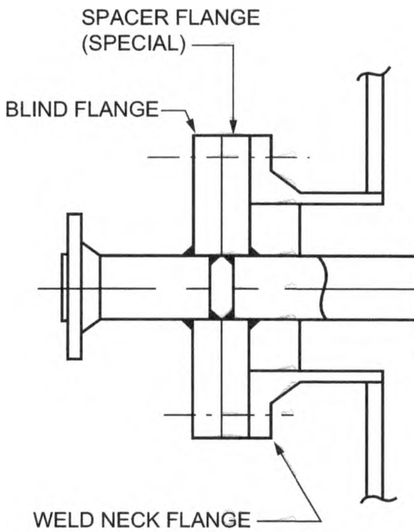
1. Through flanges provide the ability to pull the internal pipe from outside
2. Ensure the ID of flange Is large enough to clear the internal flange
3. Use where extra flexibility is required

TYPE 1

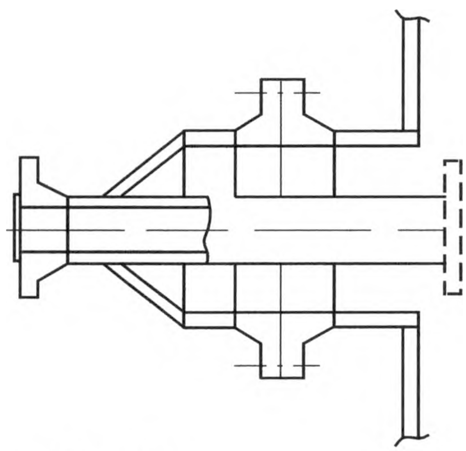


SEE NOTE 3

TYPE 2



TYPE 3



SEE NOTE 3

TYPE 4

References

[1] ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 2010 Edition, American Society of Mechanical Engineers.

[2] Modern Flange Design, 7th Edition, Bulletin 502, Taylor Forge International, Inc.

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[6] Meyer G, Strelow D. Simple Diagrams Aid in Analyzing Forces in Bolted Joints. Assembly Engineering January 1972:28–33.

[7] Horsch R. Solve Complicated Force Problems with Simple Diagrams. Assembly Engineering December 1972:22–4.