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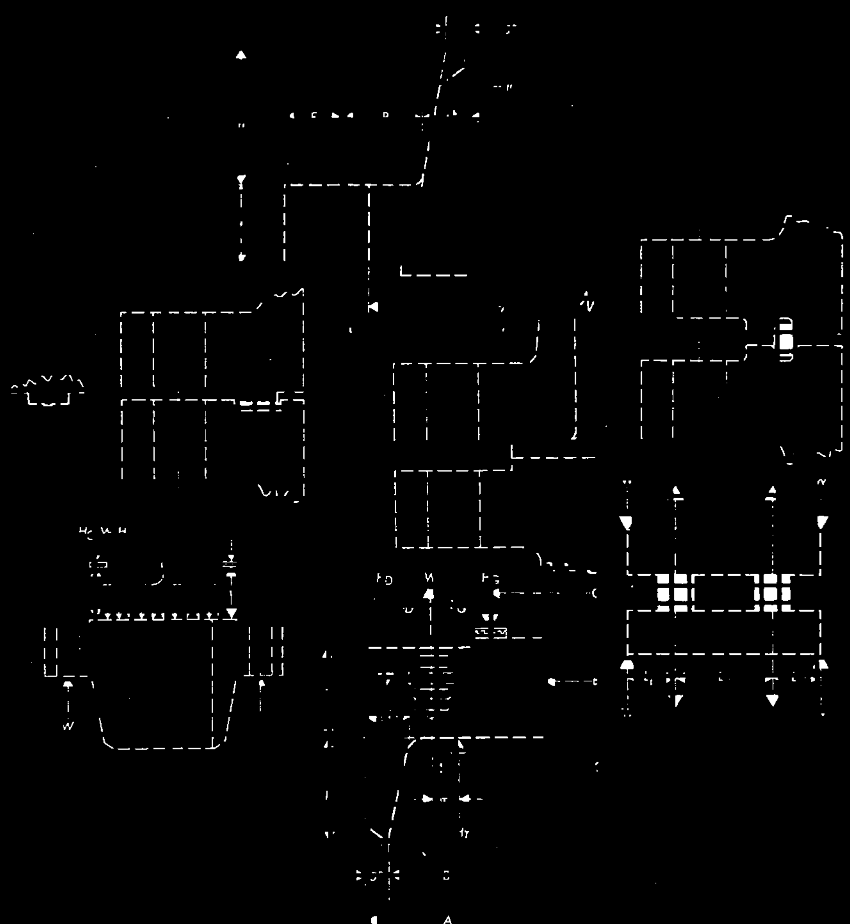
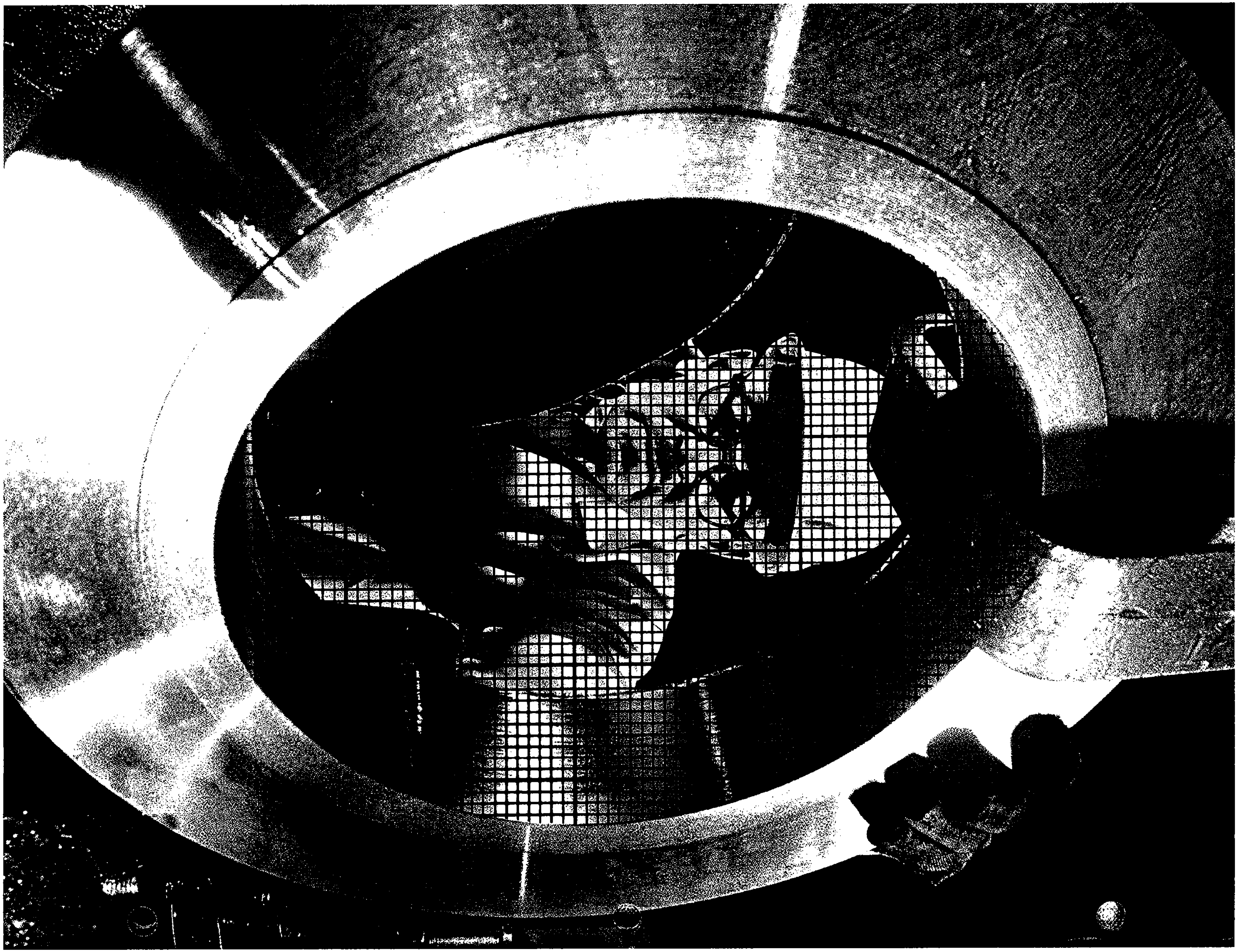


MODERN FLANGE DESIGN

Bulletin 502



Foreword:

American Spiral Pipe Works was organized at the first of the century to produce spiral riveted pipe. This unique product was widely used in water supply systems for city aquaducts, surface mining, and penstocks in this country and overseas. A serious problem developed with this pipe when flanged joints were required. At that time the flanges were made of cast iron and were often broken during shipment and fabrication. J. Hall Taylor and his staff solved this problem by developing processes and equipment to manufacture a superior, virtually unbreakable product—the forged flange. Then under a new name, Taylor Forge and Pipe Works, the company pressed on to become a leader in flange manufacturing.

They did not stop with that, but topped their achievement by developing analytical flange design formulas for industry and Code use. Professor E. O. Waters of Yale University spearheaded this work and joined Mr. Taylor to co-author the technical papers which led into the Taylor Forge publication *Modern Flange Design.* Industry immediately accepted this convenient system of flange design and used it worldwide.

This up-dated version of *Modern Flange Design* carries on the tradition with a skillful blend of data, graphics, and text. We sincerely hope that this issue will prove to be the most useful of all.

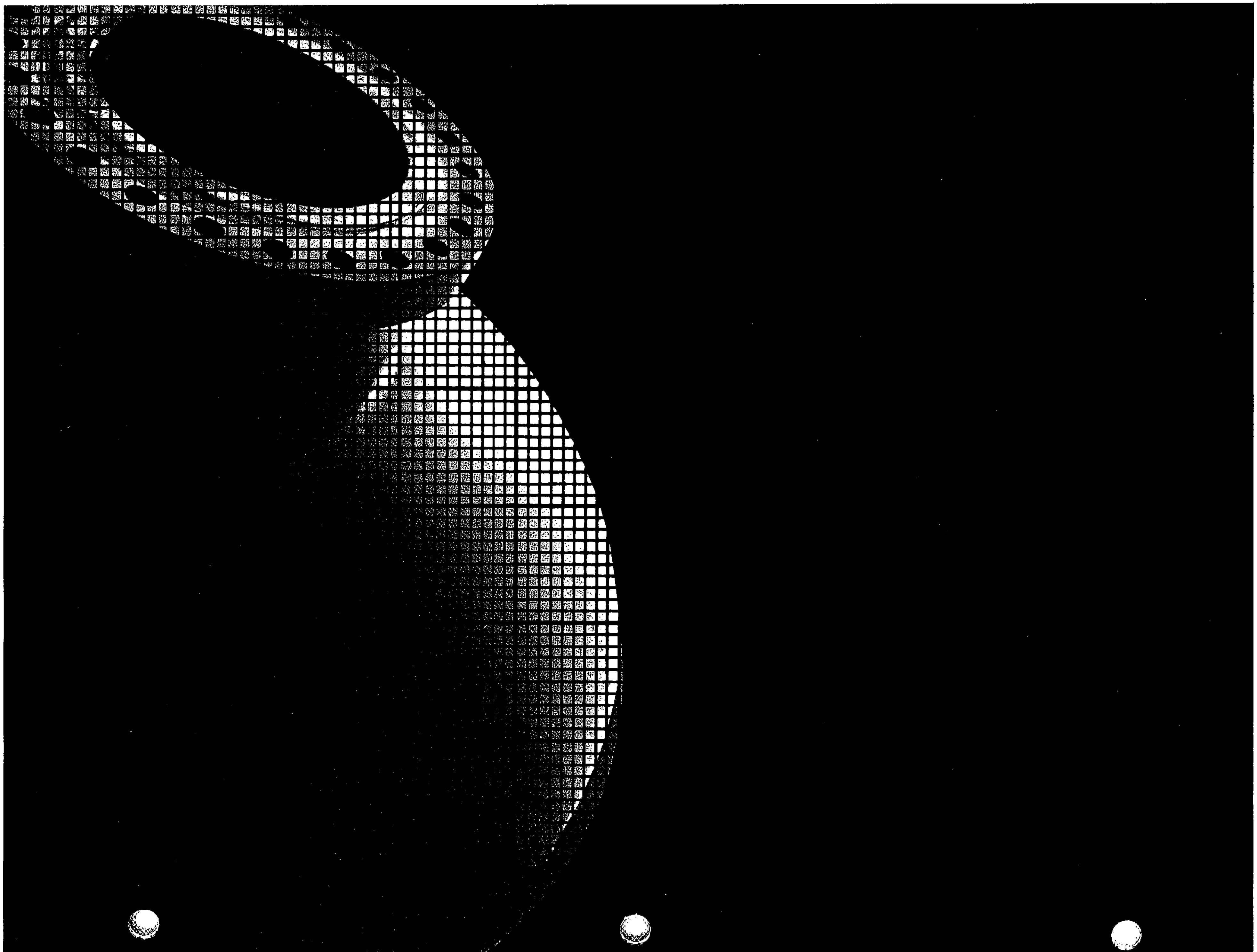


Table of Contents

Introduction

Bulletin 502 was first published in 1938 by Taylor Forge and Pipe Works as *Modern Flange Design.* It has since become a standard reference manual for the design of bolted flanges—now called Part A Flanges by the Code.① These are defined as having gaskets wholly within the circle enclosed by the bolt holes and no point of contact beyond this circle.[1] Part A rules limit the analysis to consideration of the flange moment that results from:

- bolt load
- gasket load
- face pressure load
- hydrostatic end load

The analytical aspect of this bulletin covering Part A Flanges is based on paper number FSP-59-4 "Formulas for Stresses in Bolted Flanged Connections".[2] This paper considered the effect of tapered hubs and built on an earlier work by E. O. Waters and J. Hall Taylor "The Strength of Pipe Flanges".[3] These two papers still form the technical base of the Code rules for flange design. Later, the authors published the derivation of the flange formulas in a booklet titled, "Development of General Formulas for Bolted Flanges.[4]"

This seventh edition covers code revisions through the winter '78 addenda, includes revised design formulas for reverse flanges, has a reference-only paragraph with design sheets for full face designs, adds a bibliography and a computer program listing for F, V, f, F_L, V_L, shape factors for hubbed flanges.

The PVRC Subcommittee on Bolted Flanged Connections and the ASME Subgroup on Openings cooperatively developed design methods for flat face flanges in metal-to-metal contact. Rules for analyzing *identical* pairs of such flanges were first published in Mandatory Appendix II of the 1971 Edition of the Code. These rules give a method of analysis that satisfies all conditions of equilibrium, provides for compatibility of rotation and translation between elements (hub and flange), and accounts for the radial effect of pressure acting on the pipe, hub and flange. The flanges designed by these rules were identified as "Part B" to distinguish them from "Part A" flanges.

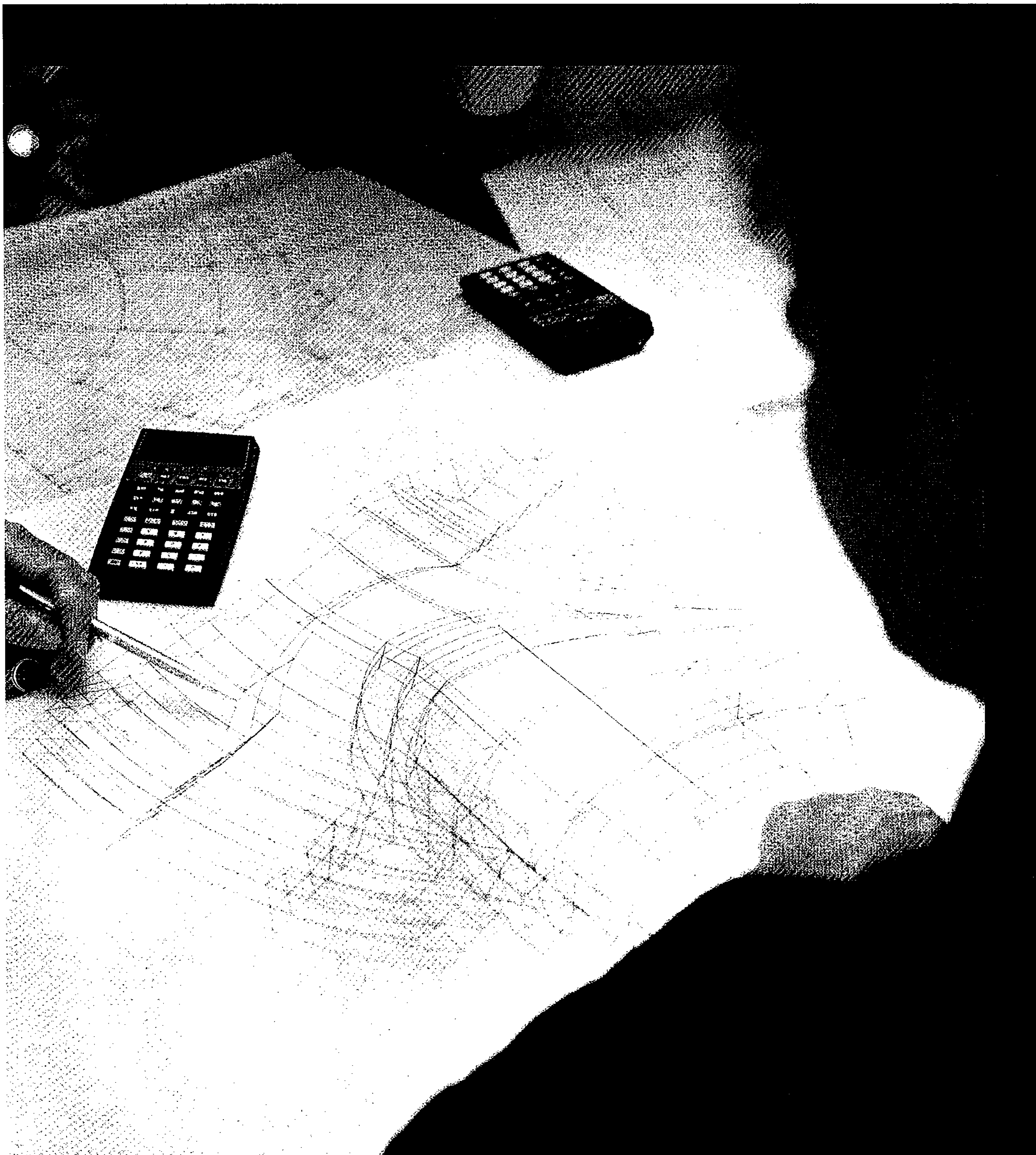
The theory used as the basis for the rules in the Code for identical Part B flange pairs was extended on a consistent basis to also cover nonidentical flanges. Because such an analysis is laborious, a simpler method suitable for analyzing *both* identical and nonidentical pairs was developed and published as ASME Code Case 1828: "A Simplified Method for Analyzing Flat Face Flanges with Metal-To-Metal Contact Outside the Bolt Circle." The Case was approved by Council January 9, 1978, after the rules for Part B flanges had been transferred from Mandatory Appendix II to Nonmandatory Appendix Y in the Summer '77 Code Addenda.

The simpler rules of Code Case 1828 resulted from designing for tangential contact between the flanges at their outside diameter. Also many of the assumptions and charts that apply equally to Part A or Part B flanges were used.

It is beyond the scope of this Bulletin to cover the design of flat face flanges in metal-to-metal contact, however, the work leading up to the Case is fully documented in the open technical literature. For further information refer to bibliography items 6 through 12.

PART A FLANGES: GASKET COMPLETELY INWARD FROM BOLT HOLES — NO OTHER CONTACT.

① The use of "Code" in this manual will specifically mean ASME Code Section VIII, Division 1.



Sidestep Designing

Sidestep Designing

Any flange design work costs time and effort, and leads one to ask, "Is there a way I can sidestep designing?"

The answer for **Part A Flanges** is yes—two ways. *One way* is to consider that the Code recognizes the proper use of existing flange dimensional standards. These are fully prepared and ready to be specified within the limits of their scope and rating. The Code incorporates ANSI B16.5 "Steel Pipe Flanges . . .",[13] API 605 "Large Diameter Carbon Steel Flanges",[14] and ANSI B16.1 "Cast Iron Flanges . . .[15]." Not included, but designed per Code rules are Industry Standards Classes 75, 175, and 350,[5] and MSS-SP44 "Pipe Line Flanges."[15] Note that the B16.5 ratings above ambient do not apply to the large diameter MSS-SP44 flanges as they are intended for use with thin-wall high-yield strength line pipe. These should be checked for Code applications; sizes over 36″ in Classes 300 and greater were designed using higher than Code allowed stress levels.

Flanges greater than 24″ size that are required to match valves often must use ANSI B16.1 cast iron dimensions. This may mean adjustments in material, bolting, and facing. All such changes affect the rating, and require Code Inspector approval. He will often ask for supporting calculations.

The *other way* to sidestep the task of designing is to call on TAYLOR FORGE for help. When design calculations must be done, and no standard is found to satisfy all requirements, we will bid a custom designed flange. This work is part of the quote and includes price and delivery, but full dimensional details are held pending an order. In the case where only a design calculation is required, this can be furnished for a nominal charge that depends on the complexity of the problem and engineering/computer time required.

For any Part A design calculation, the following information is essential:

- type of flange
- shell or pipe dimensions
- design pressure and temperature
- flange material

and this information is very helpful:

- governing code
- media contained
- shell or pipe material
- bolt and gasket material
- facing style—if specified
- corrosion allowance—if required

If the design mates existing equipment, give complete details of facing, drilling, and O.D.

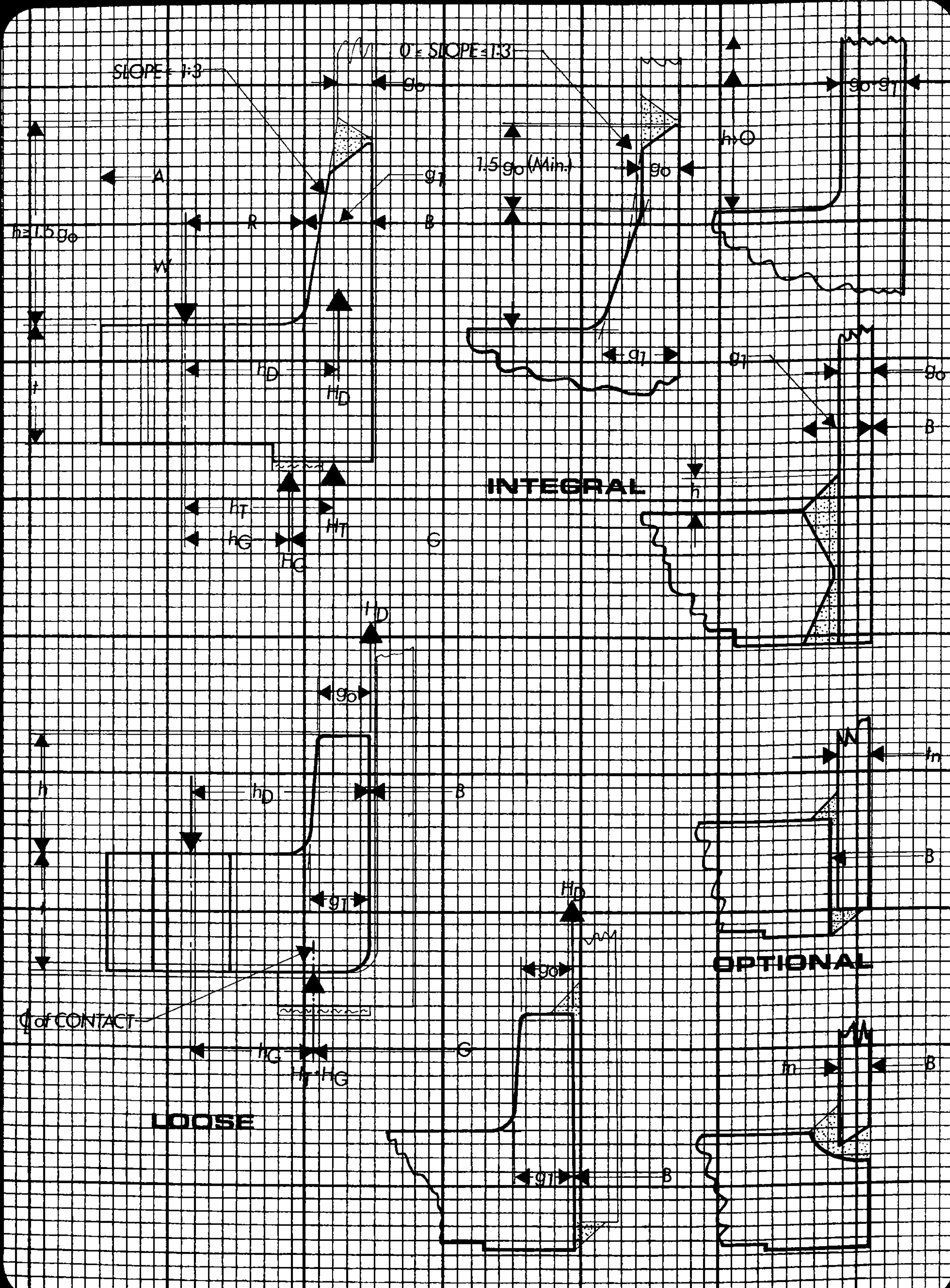
OPTIMUM INFORMATION INPUT IS A TRUE ECONOMY — IT SAVES QUESTIONS, GUESSES, AND GOOFS.

You may sidestep **Code Case 1828 Flange Design** calculation by using our design/computer services.

The calculation work for Code Case 1828 requires much more effort than Part A because the flanges behave differently. One flange is influenced strongly by the other since they are in contact outside the bolt circle; accordingly, calculated stresses are meaningful only when the interaction is considered in the analysis. As a result, when we are requested to bid on a custom job and perform calculations, it is essential that information on both flanges of the pair be provided. We require a sketch of the assembly showing all dimensions which are known or must be held. Include flange types, shell or pipe dimensions for each flange, material, type and size of gasket, design pressure and temperature. If the flanges are to be tested separately, for example with a blind cover, this information should be provided to check adequacy of the flanges for this situation.

CODE CASE 1828 FLANGE PAIRS ARE DESIGNED FOR OD TANGENTIAL CONTACT.

Part A Designs



Part A Designs

Three types of flanges designed by the rules of Part A are defined and illustrated by the Code in Paragraph UA-48 and Figure UA-48. See page 7, Fig. 1—Flange Types. The first type is

INTEGRAL

Integral which means the hub and flange are one continuous structure, either as manufactured from the original material or made so by full penetration welds. Examples are shown in Figure 1a thru 1d with loads and dimensions needed for designing. Examine hub details carefully: g_o is defined in the Code as the hub thickness at the small end, but for calculation purposes it equals the wall thickness of the attached pipe. Then the hub length h extends to the point where the slope of the hub meets the OD of the pipe. Thus, g_o in the design formulas may be different from g_o as defined, and h shorter or longer than the hub length as manufactured.

Note that B = Flange ID = Pipe ID
Use Design Sheet A, page 35.

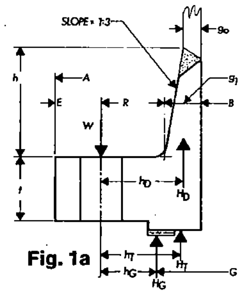


Fig. 1a

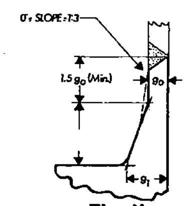


Fig. 1b

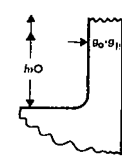


Fig. 1c

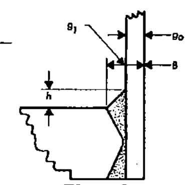


Fig. 1d

LOOSE

Loose types either have no attachment to the pipe as in Figure 1e, a lap joint, or have a non-integral structure as shown in Figures 1f and 1g for slip-on flanges. Threaded and socket-weld are also classed as loose flanges. Lap joints and threaded flanges transfer loads to the pipe at or near the face, their hubs act independently of the pipe. The hubs of slip-ons and socket-welds actually interact with the pipe but this is disregarded in the analysis. See par. UA-52 in the code. The hubs have no minimum limit on h or g_o but values of $g_o < 1.5\, t_n$ and $h < g_o$ are not recommended. If the hub is too small to meet these limits, it is best to design per Figure 1d as an integral type. (Hub thickness $= g_1 + t_n$ at the base, and t_n at the top.)

Note that B = ID of flange—*not* pipe ID.
Use Sheet B or C, pages 36 and 37.

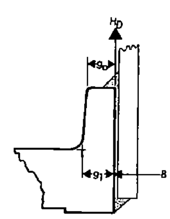


Fig. 1f

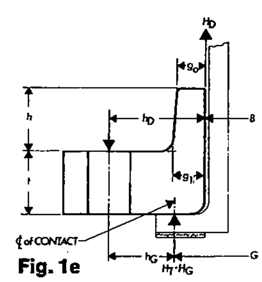


Fig. 1e

OPTIONAL

Optional types are constructed in such a way that flange and pipe act as a unit—similar to integral flanges. Optionals would normally be designed the same way as integral flanges but when:

$$g_o \le 5/8''$$
$$B/g_o \le 300$$
$$p \le 300 \text{ psi}$$
$$\text{Temperature} \le 700°\text{F}$$

The designer may choose the simpler approach and design as a loose type, and thus the term "optional flange." This classification is a convenience not a necessity, because a flange calculation is done either as integral or loose.

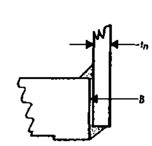


Fig. 1g

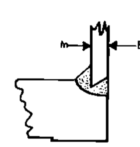
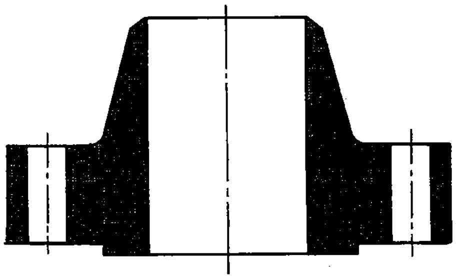


Fig. 1h

For those who are beginning designers, example 1, page 12 and the following instructions point the way for

WELDING NECK FLANGES



Welding Neck Flange calculation. This is a typical design problem; it shows the formulas and the sequence of their use. It lists the most frequently specified material and facing: ASTM A105 carbon steel flange, ASTM A 193 Grade B7 bolts, 1/16″ thick asbestos composition gasket, and a raised face. The first step is to choose material for flange, bolts, and gasket.[17] Then select facing type, facing diameter, and gasket width. Locate the gasket where it will not interfere with bore or bolts. If this data is not stipulated in the job specification, look for general guidance on material capability, facing options, and gasket width in the standards mentioned earlier.[13,14,15]

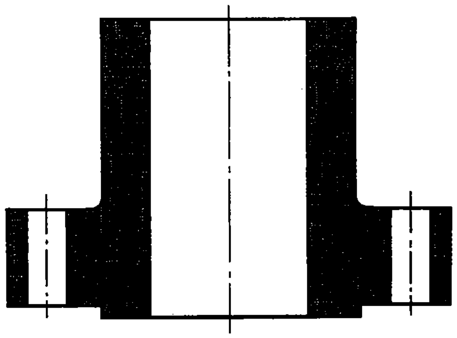
Next compute the loads, required bolting area, and complete block 4.

From Table 1, page 28, choose bolt size, in multiples of four, in number approximately equal to the flange bore. For starting hub dimensions setting $g_1 \approx 2g_o$ is suggested, R&E per the bolt size, and h based on 3:1 minimum hub slope as required by the Code. It is good practice to draw the flange section full scale, check the bolt spacing and proportion the hub so that $f \leq 1$, and make adjustments before calculating the moments in block 5.

Fill out block 6 using Table 2, page 29 and the F, V, and f graphs. Block 7 follows after trial t is chosen. Use an "educated" guess from a similar design or

$$t\,(\text{trial}) = 0.72\sqrt{\frac{\text{MY}}{\text{Sfo}}}$$

Calculate the stresses in Block 8 and 9. If calculated stresses are greater than those allowed, increase thickness or adjust hub size until the stresses are satisfactory.

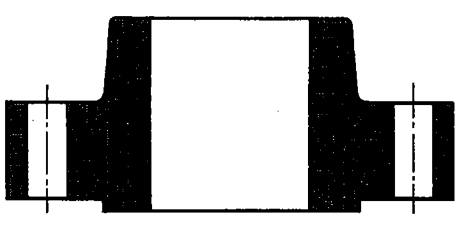


Welding Necks are designed the same way as welding neck flanges but the hub is always straight. This causes the following terms to be constant for each design:

$$g_0 = g_1$$
$$g_0/g_1 = 1$$
$$f = 1$$
$$F = 0.9089$$
$$V = 0.5501$$

Use Design Sheet A, page 35

SLIP-ON FLANGES



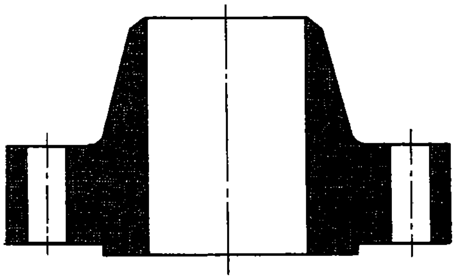
Slip-on Flanges also follow the same procedures as welding neck flanges except f is omitted, F_L and V_L are used instead of F and V. The hub may be straight or tapered and trial $g_1 \approx$ 2x pipe wall thickness. The flange ID is not equal to the pipe OD; allowance is needed to compensate for the nominal pipe OD and irregularities.

Use Design Sheet B, page 36

For those who are beginning designers, example 1, page 12 and the following instructions point the way for

WELDING NECK FLANGES

Welding Neck Flange calculation. This is a typical design problem; it shows the formulas and the sequence of their use. It lists the most frequently specified material and facing: ASTM A105 carbon steel flange, ASTM A 193 Grade B7 bolts, 1/16″ thick asbestos composition gasket, and a raised face. The first step is to choose material for flange, bolts, and gasket.[17] Then select facing type, facing diameter, and gasket width. Locate the gasket where it will not interfere with bore or bolts. If this data is not stipulated in the job specification, look for general guidance on material capability, facing options, and gasket width in the standards mentioned earlier.[13,14,15]



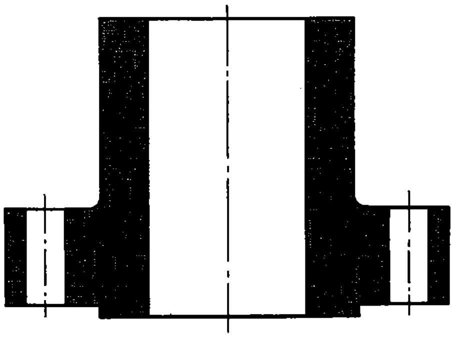
Next compute the loads, required bolting area, and complete block 4.

From Table 1, page 28, choose bolt size, in multiples of four, in number approximately equal to the flange bore. For starting hub dimensions setting $g_1 \approx 2g_o$ is suggested, R&E per the bolt size, and h based on 3:1 minimum hub slope as required by the Code. It is good practice to draw the flange section full scale, check the bolt spacing and proportion the hub so that $f \leq 1$, and make adjustments before calculating the moments in block 5.

Fill out block 6 using Table 2, page 29 and the F, V, and f graphs. Block 7 follows after trial t is chosen. Use an "educated" guess from a similar design or

$$t\text{(trial)} = 0.72\sqrt{\frac{MY}{Sfo}}$$

Calculate the stresses in Block 8 and 9. If calculated stresses are greater than those allowed, increase thickness or adjust hub size until the stresses are satisfactory.



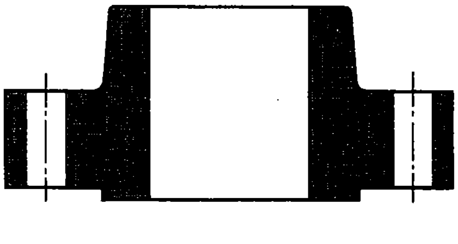
Welding Necks are designed the same way as welding neck flanges but the hub is always straight. This causes the following terms to be constant for each design:

$$g_o = g_1$$
$$g_o/g_1 = 1$$
$$f = 1$$
$$F = 0.9089$$
$$V = 0.5501$$

Use Design Sheet A, page 35

SLIP-ON FLANGES

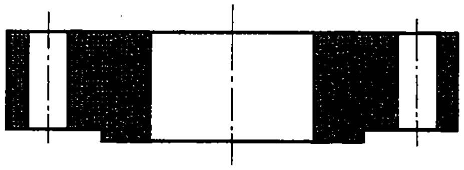
Slip-on Flanges also follow the same procedures as welding neck flanges except f is omitted, F_L and V_L are used instead of F and V. The hub may be straight or tapered and trial $g_1 \approx$ 2x pipe wall thickness. The flange ID is not equal to the pipe OD; allowance is needed to compensate for the nominal pipe OD and irregularities.



Use Design Sheet B, page 36

RING FLANGES

Ring Flanges in this manual are hubless. They may be loose type or slip-on designed as loose type. Use all the data and procedures of blocks 1 through 5, as in a welding neck flange. In block 6, only K and Y apply. Since there is no hub or hub factors, the tangential stress is the only one to be calculated using blocks 8 & 9.



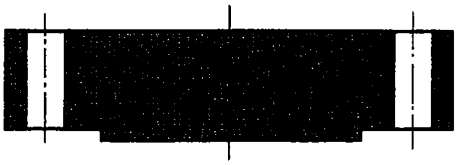
The calculated flange thickness is the greater of

$$t=\sqrt{\frac{MoY}{S_{fo}B}} \quad \text{or} \quad t=\sqrt{\frac{M_G Y}{S_{fa}B}}$$

Use Design Sheet C, page 37

BLIND FLANGES

Blind Flanges in this manual are circular and limited to those shown in the Code in Figure UG-34 sketches (j) and (k). Use the load data given in blocks 1 through 5 and dimensions—except thickness—from the matching flange.



1) Operating conditions $t = G\sqrt{\frac{0.3P}{Sfo} + \frac{1.9\ W_{m1}\ h_G}{Sfo\ G^3}}$

2) Gasket seating conditions $t = G\ \sqrt{\frac{1.9\ Wh_G}{Sfa\ G^3}}$

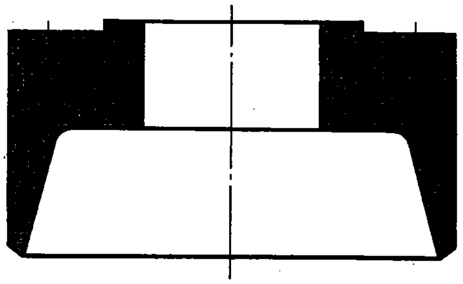
Use the greater thickness.

When the blind has a grooved facing, t under the groove must at least equal that required for gasket seating.

Use Sheet D, page 37

REVERSE FLANGES

Reverse Flanges have the hub at the O.D. as shown in Figure 2. For operating conditions H_T and h_D are negative, and h_T may be positive or negative. Add the moments algebraically, then use the absolute value $|M_o|$ in all subsequent calculations. Use B' to calculate K, and A to find h_o.

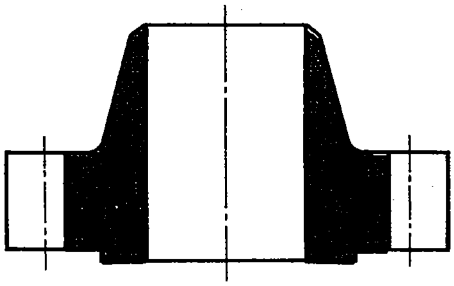


A fourth stress equation S_T, tangential stress at B', has been added for both design conditions. This will often be the controlling stress.

Use sheet E page 38.

SLOTTED FLANGES

Slotted Flanges — The effect of bolt holes on flange design is not considered in Part A rules, but some flanges require T-bolts with radially slotted holes. This allows the bolts to swing out of the way of a cover for quick opening. The slots destroy the continuity of the outer boundary and the interrupted fibers can no longer contribute to the ring action. Substitute the diameter of the circle tangent to the inner edges of the slots for A, and follow the appropriate standard design procedure. See Figure 3.



Use Design Sheet A page 35

UNUSUAL SHAPES

Unusual Shapes — Bolted connections must sometimes be fit into tight spaces where the usual geometry cannot be applied. In this case, flanges may be made square or oval with circular bores and are treated as inscribed circular flanges. Bolt loads and moments, as well as stresses, are calculated as for other flanges, using a bolt circle passing through the centers of the outermost bolt holes. Similar assumptions are in order for oval-shaped and two-bolt flanges departing not too greatly from the circular. For both types, the bolt spacing should be checked and an allowance made for maldistribution of the moment. The spacing factor can be less than is required for circular flanges since the metal available in the corners tends to spread the bolt load and even out the moment.

Special care is needed when the effective (highly stressed) part of the hub is interrupted. This may occur with channel flanges on heat exchangers where large nozzle openings are located in a portion of the hub. Here is a case where the moment is introduced uniformly but it cannot be absorbed evenly. Increased moment or local reinforcement can help compensate for this condition.

The Code rules for flange design provides an alternate method of calculating hubs stresses for small diameter flanges. When the inside diameter is less than 20 times the hub thickness (20 g_1), B_1 may be used for B in the formula for longitudinal hub stress, S_H. Calculate B_1 as follows:

$B_1 = B + g_1$ for loose hubbed flanges and integral flanges where the point corresponding to g_1/g_0 and h/h_o falls below the $f = 1$ line. See the values of f chart, page 34.

$B_1 = B + g_0$ for integral flanges where $f \geq 1$

This option is not shown on sheets A & B — it is quoted here for reference. See the Code paragraph UA-47.

EXAMPLE 1 — WELDING NECK FLANGE DESIGN

1 DESIGN CONDITIONS			2 GASKET	FACE	3 FROM Fig. UA-49
Design Pressure, P	400		33"ID X 36"OD X 1/16" THK. ASB. COMP.	34 1/2" OD X 1/16" RF	N = 0.750"
Design Temperature	500° F				b = 0.306"
Flange Material	A105				G = 33.888"
Bolting Material	A193-B7				y = 3700
Corrosion Allowance	0				m = 2.750

4 LOAD AND BOLT CALCULATIONS

Allowable Stresses					
Flange	Design Temp., S_{fo}	17,500	$W_{m2} = b\pi G y$ = 120609	A_m = greater of W_{m2}/S_a or W_{m1}/S_b =	17.299
Flange	Atm. Temp., S_{fa}	17,500	$H_p = 2b\pi GmP$ = 71713	A_b =	19.836
Bolting	Design Temp., S_b	25,000	$H = G^2\pi P/4$ = 360771	$W = .5(A_m + A_b)S_a$ =	464192
Bolting	Atm. Temp., S_a	25,000	$W_{m1} = H_p + H$ = 432484		

CONDITION	LOAD	X LEVER ARM	= MOMENT
5 Operating	$H_D = \pi B^2P/4$ = 321699	$h_D = R + .5g_1$ = 1.938"	$M_D = H_D h_D$ = 623292
	$H_G = W_{m1} - H$ = 71713	$h_G = .5(C - G)$ = 1.556"	$M_G = H_G h_G$ = 111599
	$H_T = H - H_D$ = 39072	$h_T = .5(R + g_1 + h_G)$ = 2.028"	$M_T = H_T h_T$ = 79242
			M_o = 814133
Seating	$H_G = W$ = 464192	$h_G = .5(C - G)$ = 1.556"	M_G = 722371

8 Allowable Stress	STRESS CALCULATION—*Operating*	
1.5 S_{fo}	Long. Hub, $S_H = fm_o/\lambda g_1^2$ =	22865
S_{fo}	Radial Flg., $S_R = \beta m_o/\lambda t^2$ =	10982
S_{fo}	Tang. Flg., $S_T = m_oY/t^2 - ZS_R$ =	6800
S_{fo}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$ =	16923

9 Allowable Stress	STRESS CALCULATION— seating	
1.5 S_{fa}	Long. Hub, $S_H = fm_G/\lambda g_1^2$ =	20288
S_{fa}	Radial Flg., $S_R = \beta m_G/\lambda t^2$ =	9744
S_{fa}	Tang. Flg., $S_T = m_G Y/t^2 - ZS_R$ =	6033
S_{fa}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$ =	15016

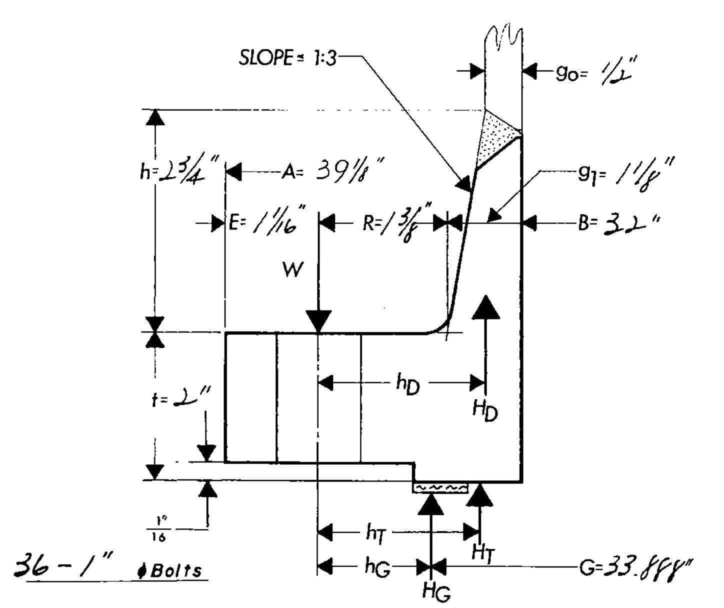
6 K AND HUB FACTORS

$K = A/B$	= 1.223	h/h_o	= 0.688
T	= 1.830	F	= 0.777
Z	= 5.041	V	= 0.162
Y	= 9.773	f	= 1.000
U	= 10.740	$e = F/h_o$	= 0.194
g_1/g_o	= 2.250	$d = \frac{U}{V} h_o g_o^2$	= 66.48
$h_o = \sqrt{Bg_o}$	= 4.000		

7 STRESS FORMULA FACTORS

t	= 2.000"
$\alpha = te + 1$	= 1.388
$\beta = 4/3\ te + 1$	= 1.518
$\gamma = \alpha/T$	= 0.759
$\delta = t^3/d$	= 0.120
$\lambda = \gamma + \delta$	= 0.879
$m_o = M_o/B$ = operating	25442
$m_G = M_G/B$ = seating	22574

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

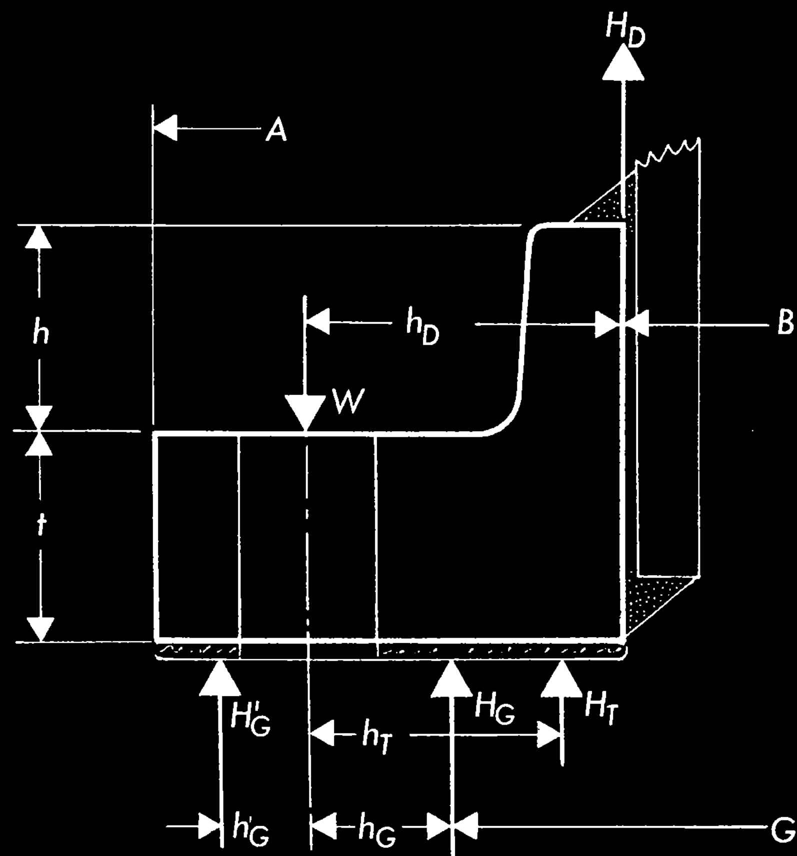


G+W Taylor-Bonney Division

Computed DLS Date 7-17-79

Checked ____ Number ____

Full Face Design



Flanges With Full-Face Soft Gaskets have no design rules included in the Code. They are assigned to the general category of "good engineering practice," or they are to be accepted at the discretion of the Code Inspector. In lieu of official rules, D. B. Rossheim proposed a design method as long ago as 1943. Taylor Forge adapted his work to the Code pattern and published it as **Engineering Department Bulletin No. 45.** This has not been made part of the Code but has served as a conservative guideline with apparent success. This design approach assumes:

- Rigidity at the bolt circle before pressurizing
- Inner flange edge is unrestrained
- Uniform gasket pressure on the annular surfaces on each side of the bolt circle.
- Ring effect and bolt holes cause no significant loss of strength
- A counter moment between the bolt circle and the flange OD

This analysis is published for reference only—as a suggested "Good Practice." We invite your comments.

The design procedures are sequenced much the same as in Example 1, page 12. Example 2, page 14, illustrates this by listing calculations for a 24 inch Class 125 Light Weight Slip-On flange.

Use Sheet F, page 39

EXAMPLE 2

SLIP ON FLANGE DESIGN—FLAT FACED

1 DESIGN CONDITIONS		
Design Pressure, P		75
Design Temperature		300
Flange Material		A181 60
Bolting Material		A307B
Corrosion Allowance		0
Allowable Stresses — Flange	Design Temp., S_{fo}	15000
	Atm. Temp., S_{fa}	15000
Allowable Stresses — Bolting	Design Temp., S_b	7000
	Atm. Temp., S_a	7000

2 GASKET

FULL FACE RUBBER
75A SHORE DUROMETER

3	
$G = C - 2h_G =$	26.85"
$b = (C - B)/4 =$	1.375"
$y =$	200
$m =$	1

4 LOAD AND BOLT CALCULATIONS

$W_{m2} = b\pi Gy + H'_{GY} =$	23196	A_m = greater of W_{m2}/S_a or W_{m1}/S_b =	13.76
$H_p = 2b\pi GmP =$	17400	$A_b =$	17.80
$H'_p = {}^{hG}/_{h'G}\, H_p =$	36420	$W = .5(A_m + A_b)\, S_a =$	110460
$H = G^2\pi P/4 =$	42466	$H'_{GY} = {}^{hG}/_{h'G}\, b\pi GY =$	48555
$W_{m1} = H + H_p + H'_p =$	96286		

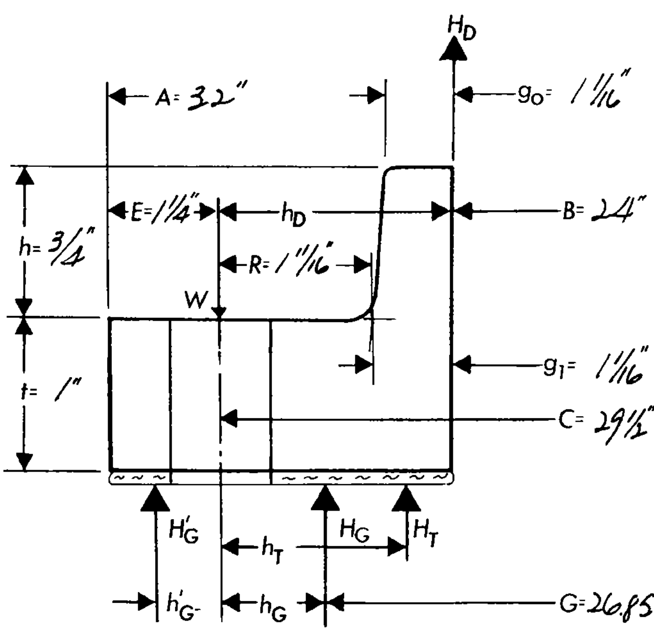
CONDITION	LOAD	X	LEVER ARM	=	MOMENT	
5 *Operating*	$H_D = \pi B^2P/4 =$ 33929		$h_D = R + g_1 =$ 2.75"		$M_D = H_D h_D =$	93305
	$H_T = H - H_D =$ 8537		$h_T = .5(R + g_1 + h_G) =$ 2.037"		$M_T = H_T h_T =$	17390
					$M_o =$	110695
LEVER ARMS	$h_G = \frac{(C - B)(2B + C)}{6(B + C)} =$ 1.325"		$h'_G = \frac{(A - C)(2A + C)}{6(C + A)} =$ .633"			
REVERSE MOMENT	$H_G = W - H =$ 67994		$h''_G = \frac{h_G h'_G}{h_G + h'_G} =$ .428"		$M_G = H_G h_G'' =$	29101

8 Allowable Stress	STRESS CALCULATION—*Operating* Conditions (use M)	
$1.5\, S_{fo}$	Long. Hub, $S_H = m_o/\lambda g_1^2 =$	1995
S_{fo}	Radial Flg., $S_R = \beta m_o/\lambda t^2 =$	5236
S_{fo}	Tang. Flg., $S_T = m_o Y/t^2 - ZS_R =$	13176
S_{fo}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$ =	7586
S_{fo}	RADIAL STRESS AT BOLT CIRCLE $S_{RAD} = \frac{6 M_G}{t^2(\pi C - nd_1)} =$	2679

6 K AND HUB FACTORS			
$K = A/B$	= 1.333	h/h_o	= .149
T	= 1.78	F_L	= 5
Z	= 3.57	V_L	= 40
Y	= 6.91	$e = \frac{F_L}{h_o}$	= .992
U	= 7.59		
g_1/g_o	= 1.00	$d = \frac{U}{V_L} h_o g_o^2 =$	1.08
$h_o = \sqrt{Bg_o}$	= 5.04		

7 STRESS FORMULA FACTORS		
$t =$		= 1.00"
α	$= te + 1$	= 1.992
β	$= 4/3\, te + 1$	= 2.324
γ	$= \alpha/T$	= 1.120
δ	$= t^3/d$	= .927
λ	$= \gamma + \delta$	= 2.047
$m_o = M_o/B$		= 4612

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

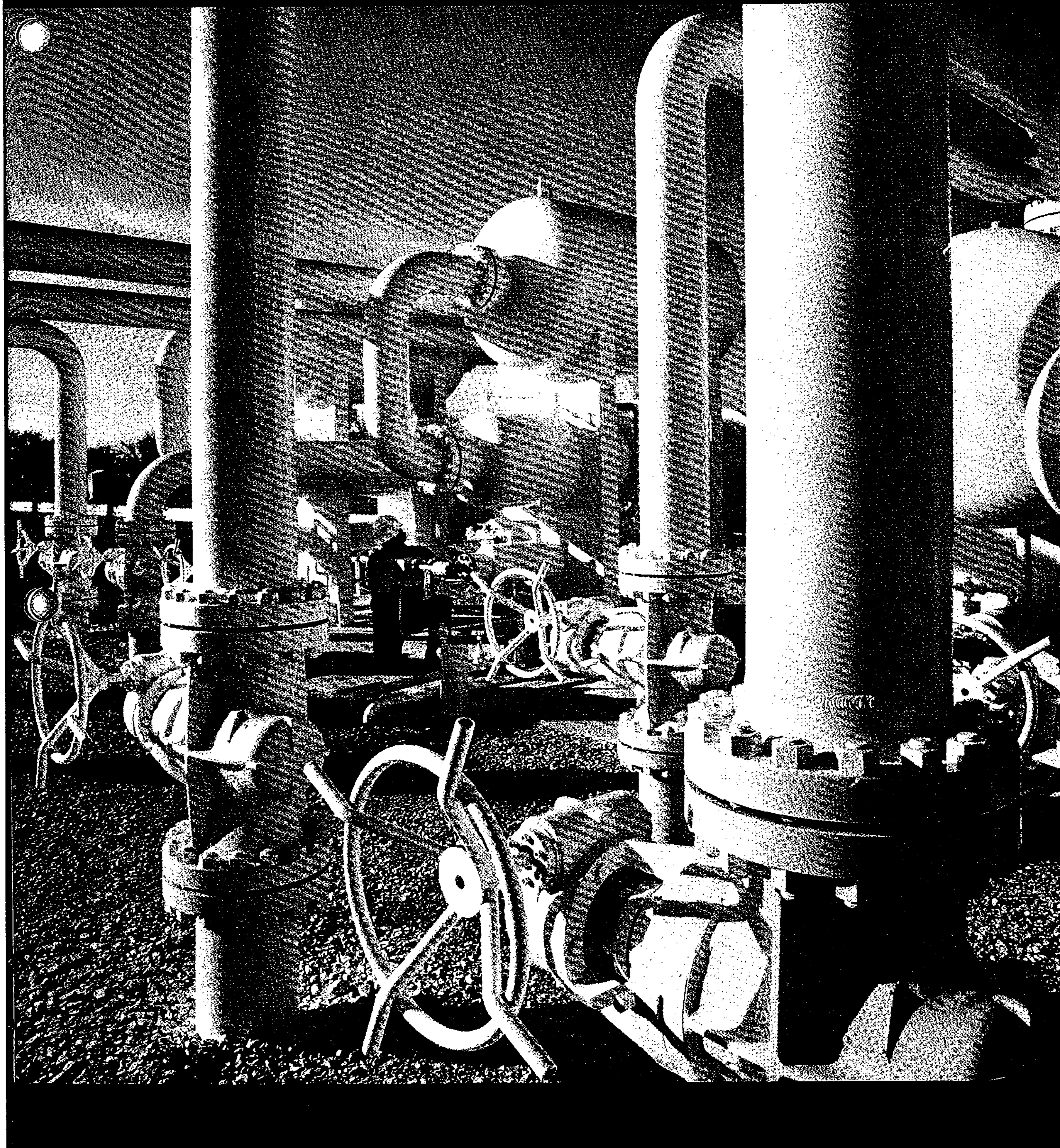


20-1 1/4" ϕ Bolts

G+W Taylor-Bonney Division — BONNEY FORGE

Computed DLS	Date 7-17-79
Checked	Number

Face and Gasket



Face and Gasket

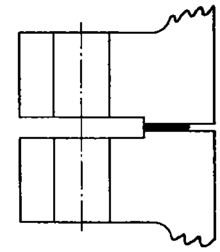
Figures to the right picture a variety of flange facing styles used in industry. Specific recommendations for facing/gasket combinations are not within the scope of this manual, but pressure, temperature, thermal shock, cyclic operation, and the fluid handled should be considered. The various flange standards give valuable suggestions in this area, along with dimensional data.

Two general categories are shown. The first seals by bolt force squeezing the gasket and this includes raised face, tongue and groove, ring type joint, male and female, and lap joint. The tongue face can be made with a small nubbin for extra gasket grip without bolt load penalty. The nubbin style also reduces *b* and is used with metal clad gaskets.

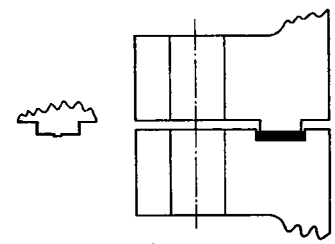
Gasket materials used in this first category include the various sheet stocks of rubber, cork, and asbestos compositions, metallic-elastomer mixes as the spiral wound, and solid metal rings made as flat washers or ring-type-joint forms.

The other category is called self-energizing or pressure actuated. This means that initial sealing is achieved by the gasket/facing geometry without significant bolt load, and the gasket seating force increases with pressure. O-rings made of elastomer materials are used most often in this style seal. Calculations for flanges using such gaskets often omit gasket loads and gasket seating stresses, W_{M2}, H_p, and W, as being negligibly small.

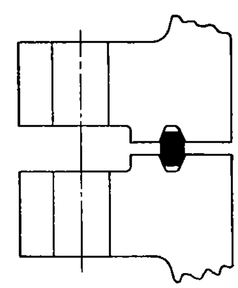
Metallic self-energizing seals, not shown, are delta ring, double cone, lens joint, metallic O-rings, API RX and BX ring joints.[19] All of these require very fine surface finishes, tight tolerances, and meticulous care in assembling.



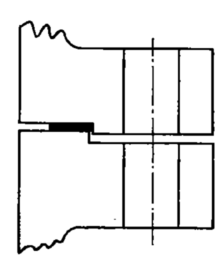
RAISED FACE



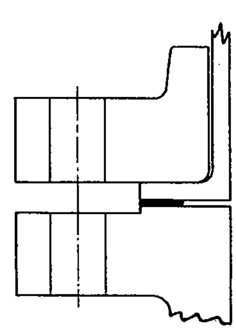
TONGUE & GROOVE



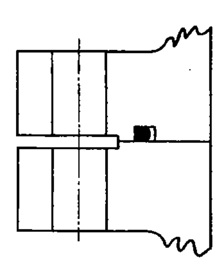
RING TYPE JOINT



MALE & FEMALE



LAP JOINT



O RING

FACTORS m,y,b

The problem of specifying acceptable gasket factors has been a long-standing concern of the ASME Boiler and Pressure Committees.

m = factor needed to hold a seal under internal pressure, dimensionless.

y = unit seating load, psi

b = effective gasket width, in.

In 1941, Rossheim and Markl[20] reviewed values then in use, made changes, presented m and y tables for commonly used gasket materials, and suggested rules for finding b. It was known that m increased as the initial bolt load was given lower values — when the intitial pressure was zero m was very high. Therefore, the authors decided that y could be the m value required for a bolt load that would seal at zero internal pressure, and proposed the empirical formula $y = 180\ (2M - 1)^2$. Although Rossheim and Markl made no claim other than they hoped to stimulate research, their m and y values were adopted by the Code.

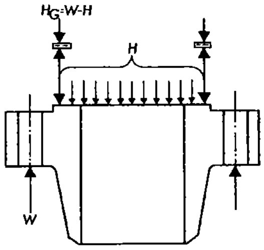


Fig. 4

Gasket tests show leakage usually happens well before the internal pressure has relieved the gasket reaction.

Suppose the ratio of the compressive stress of the gasket to the internal pressure of the fluid trying to escape is designated m' at the instant when leakage starts, then for a very poor gasket m' would be a large number. For good quality commercial gaskets it might range between 3 and 1.5. For a theoretically perfect case a leak would not start until the fluid pressure just exceeded the gasket pressure, with $m' = 1$. Assuming that reliable values of m' are available, the equilibrium conditions of Figure 4, can be expressed: bolt load = (fluid pressure) X (area subjected to fluid pressure) + m' (fluid pressure) X (area subjected to gasket pressure). In formula form this is $W_{m1} = \pi/4\, G^2 P + \pi\, 2b\, Gmp$ where W_{m1} is the bolt load (operating). The first term on the right side is the total hydrostatic pressure load acting on the effective gasket diameter G, and the second is the gasket reaction over an annular area $2b$ wide on the same G. Sealing theory requires that the load $2b\pi Gmp$ be resisted by the gasket when the internal pressure equals P. But the gasket also resists the total load, W_{m1}. When pressure P is applied, load $\pi/4\, G^2P$ is removed and $2b\pi GmP$ remains as the theoretical load required to hold a tight joint at P.

Factor m is derived from m' and has an added safety margin. Neither m or y values have theoretical standing and those now in use are based on practical experience and some formal experimentation. They have a direct effect on flange design and have been discussed for years without reaching fixed values that could be made mandatory. Many variables are involved, and much time is required to make a single test.[18]

At the present time, leakage criteria is getting a "hard look" and research programs are under study by the Pressure Vessel Research Committee to determine if m and y values can be set up in relation to specified leak rates. For example, a joint that held for one minute without the escape of one drop of water was considered "tight". This equals a leak rate of 10^{-3} cc. per second and would not be allowed in many industries. Possibly a leakage of 10^{-4} cc. per sec., corresponding to 6 drops per hour, would be acceptable for liquids, while 10^{-5} would be necessary for heavy gas and 10^{-7} for lethal substances. This concept may lead to multiple listings of gasket factors for a given material—with a different level of m and y values assigned to each leak rate and contained fluid.

Note: Code m, y and b values do *not* pertain to full face gaskets.

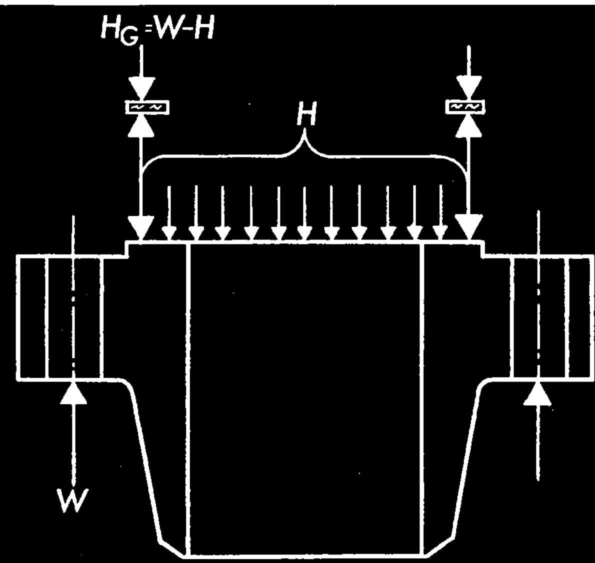


Fig. 4

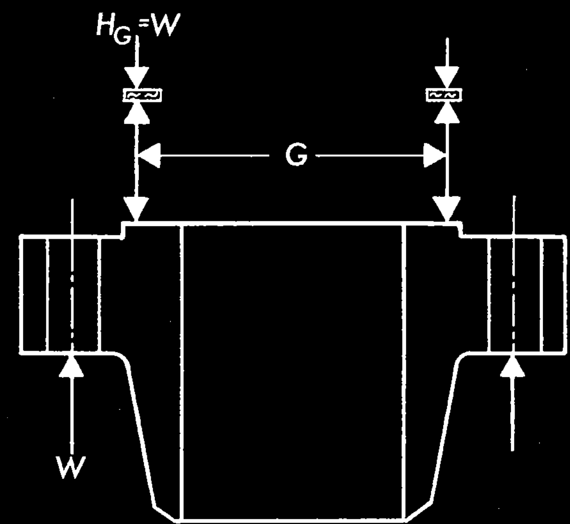


Fig. 5

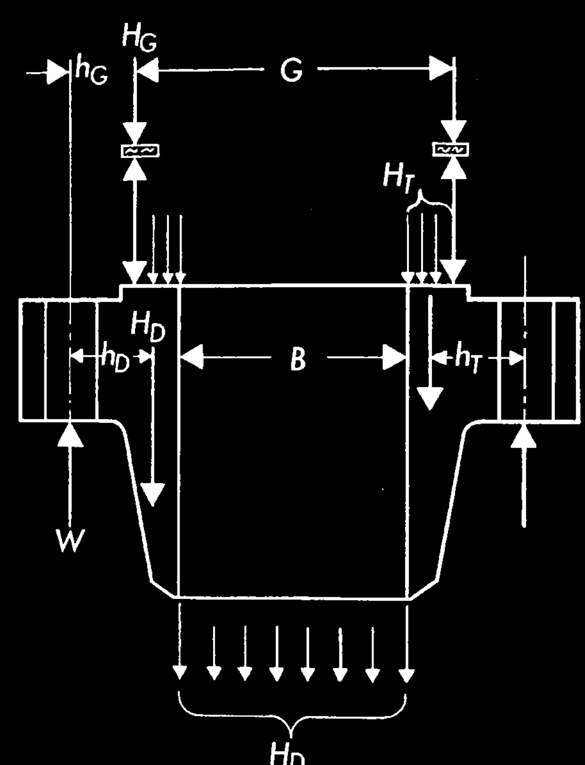


Fig. 6

The Code requires the analysis of two distinct load systems. The first is gasket seating — when there is no internal pressure. To achieve a seal, all facing surface irregularities must be filled with gasket material. This is done with direct force by bolting the flanges. The required seating load is represented by: $W_{m2} = b\pi Gy$ and is illustrated in Figure 5 where the bolt load (seating) is balanced by the gasket reaction.

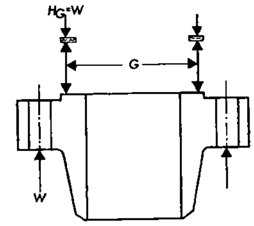


Fig. 5

The other load system relates to internal pressure and has four forces:

H_D — the hydrostatic end force
H_T — pressure force on the flange face
H_p — the total gasket load required to maintain seal
W — the bolt load

The first three forces work to separate the flange pair. They are balanced by the fourth, W, which holds the assembly together. This is illustrated in Figure 6.

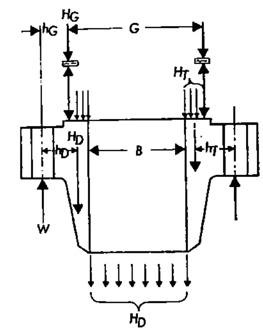


Fig. 6

The hydrostatic end force H_D comes to the flange from the closed end of the pipe system to which it is welded. The end force reaches the flange through the hub, and pulls on the ring portion of the flange mid-hub at its large end if it is a tapered hub.

The fluid pressure force H_T acts directly on the face of the flange where it is exposed. For a gasket covering the entire raised face in Figure 6, H_T would equal zero, but as a conservative allowance, leakage is assumed to be possible as far as G. The H_T force acts on a circle half-way between B & G.

The gasket load, $H_p = 2b\pi GmP$ where gasket factor m relates the required gasket stress *at* design pressure P *to* that design pressure. For example, an m factor of 3 means that the residual gasket stress at P must be at least $3P$ for the joint to be tight.

In order to calculate the moment acting on a flange, the forces are multiplied by the appropriate lever arms which are measured from the point of force application to the bolt circle. The forces and the lever arms indicated in Figure 6 are for an integral flange.

The gasket reaction load is generally assumed to decrease as internal pressure is applied. The actual change is affected by flange rotation, bolt stretch, and the gasket's ability to resist and recover from compression. Bolt loads frequently change too, for the same reason. They must be retightened, especially when gasket relaxation or creep occurs.

All of these variables may be explained by the following illustration. Let a pair of flanges be represented by two steel bars placed side by side as in Figure 7, page 20; they are separated by blocks, as shown, which represent the gasket, and forces W at the ends of the bars represent the bolt loads, which are balanced by opposite reactions at the blocks. Under the action of the "bolt loads" the bars are pulled together at their ends, the amount of deflection can be calculated by formulas in engineering handbooks. Consider now two cases:

HARD GASKETS

Hard Gaskets. The blocks are completely rigid, and the bars deflect. Before internal pressure is applied, the bolt load balances the gasket reaction, and pulls the ends of the bars towards each other so that their separation is reduced as the bolts are tightened. For a cantilever, the ends of each bar would deflect a distance proportional to $Wl_1^3/3$, but due to slight rotation as the bars pivot about the blocks, this is increased to $(Wl_1^3/3)\ (l + 1.5\, l_2/l_1)$. Therefore, in the initial bolt-up, the nuts are wrenched down until the rod ends are brought closer together by twice this amount, proportional to $(Wl_1^3/3)\ (2 + 3l_2/l_1)$. Now, when pressure is applied to the bars between the blocks, equal in amount to Pl_2, and tending to spread the bars apart, the "bolt load" W may be kept constant but the "gasket load" on the blocks is reduced; this may continue until, in the case of a real flange joint, leakage occurs. In any event, under the most favorable conditions P may be increased until, in our illustrative example, the pressure on the blocks is completely relieved, and $Pl_2 = 2W$. At this point, calculation of the deflection of the ends of each bar indicates that it is more than before, the total reduction of space between the bars now being proportional to $(Wl_1^3/3)\ (2 + 3l_2/l_1 + 0.5\ (l_2/l_1)^2)$. In other words, in order to maintain full bolt load while internal pressure is applied, the effective bolt lengths must be shortened by wrenching down the nuts still further.

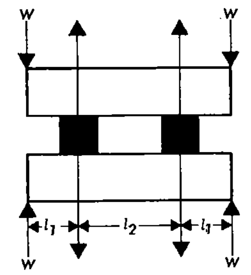


Fig. 7

SOFT GASKETS

Soft Gaskets. The bars are inflexible, and remain straight and parallel during all changes of loading. During bolt-up the blocks are compressed a small distance and the rod ends are pulled together the same amount; the "bolt load" $= W$. Then when pressure Pl_2 is applied between the blocks, the "gasket load" can only be relieved by letting the blocks return to their original thickness; this means that the bars must be separated by the same amount, from end to end, and the bolt lengths *increased* to avoid stressing the bolt beyond the original value of W.

These illustrations represent two extremes, between which an endless variety of conditions can occur.

Bolts

AREA

Area. The minimum total bolt area required (A_m) equals the larger of W_{m1}/S_b or W_{m2}/S_a where S_b is the allowable bolt stress at operating temperature and S_a at ambient. Use A_m to select bolt size and number. This is easily done for most flange designs, but thin flanges at low pressure may require excess bolt area because:

1) Bolt sizes <1/2″ are easily overstressed, avoid using them when possible.
2) The number of bolts should be specified in multiples of four to guide fit-up and alignment.
3) Bolts must be close enough to hold the seal.

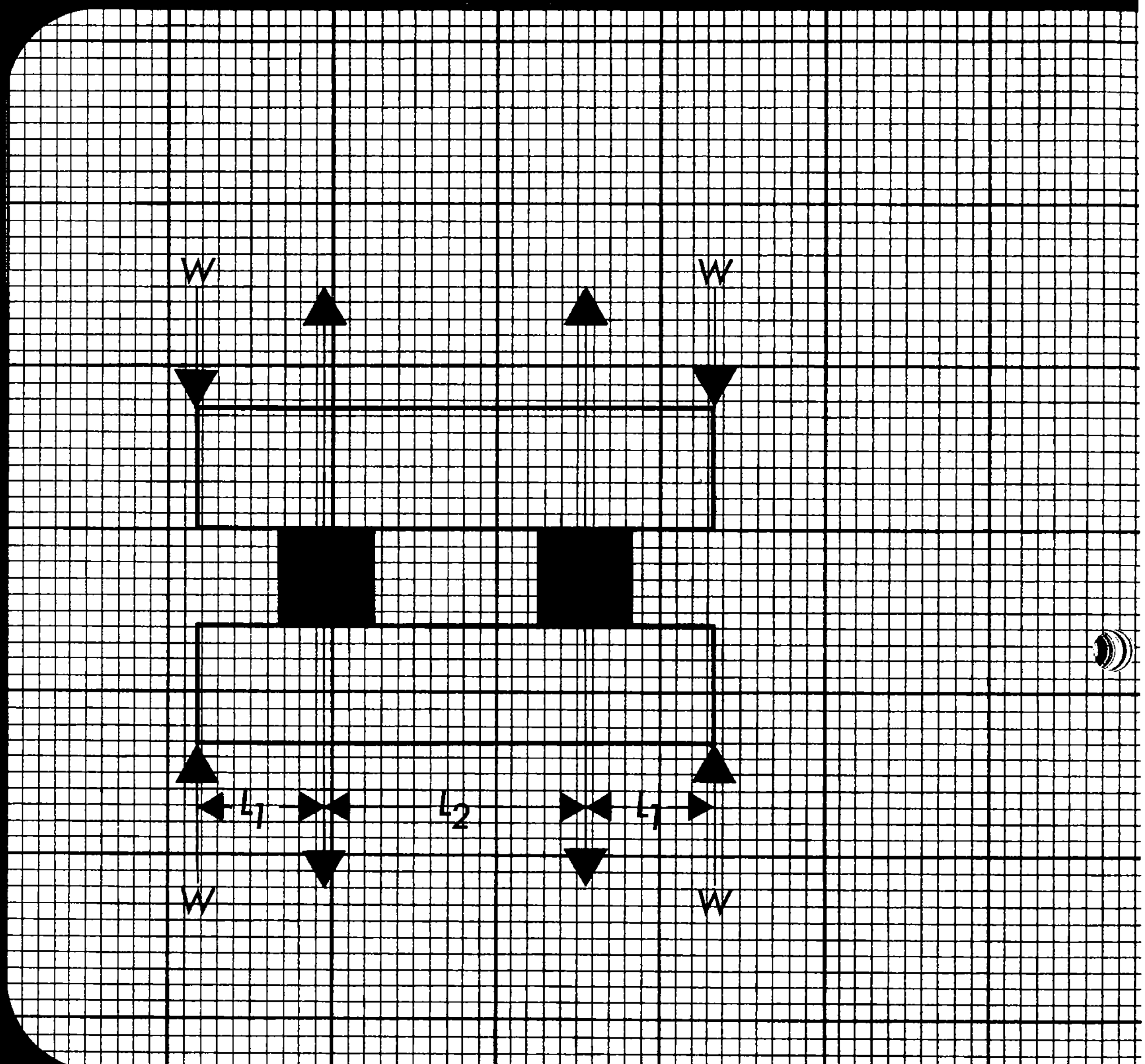
SPACING

Spacing. When a very few bolts will satisfy the Code but give bolt spacing so wide that leakage may occur, adjust the design by increasing the flange thickness, increasing the number of bolts, or both. This will help the flange seal the gasket—even between bolts. The best bolting practice is to combine smallest practical size with minimum spacing to carry the load.

In 1950, Irving Roberts completed a detailed study of the interaction of bolts, gasket and flanges.[21] Investigating the gasket factor m' and the relation of bolt spacing to leakage, he recognized that gasket elasticity distributes the load between bolts. He also derived a bolt spacing curve in terms of flange thickness, indicating a 5 percent loss of gasket pressure half way between bolts. Mr. Robert's work leads to a formula for "normal" bolt spacing when a workable gasket-to-flange compression ratio S is used. Normal bolt spacing $= tS^{1/4}$, where $S =$ compression of gasket/compression of flange, or E_f (gasket thickness) (flange area)/E_g (flange thickness) (gasket area).

$E_f, E_g =$ modulus of elasticity of flange and gasket, respectively. E_g is the uncertain factor in this formula, but gasket manufacturers or miscellaneous material tables might be data sources.

The formula for maximum bolt spacing, also recognizes the importance of gasket compression by using m. Maximum bolt spacing $= 2a + 6t / (m + 0.5)$ It assumes proper load distribution when bolt spacing $= 2a + t$. Minimum bolt spacing is controlled by the requirements of nut and wrench clearance—see Table 1, page 28. If special nuts and wrenches are used to permit very close spacing, the flange section may be seriously drilled away, and the weakening effect *will not be exposed* by Part A analysis, discussed pp. 9-16.



SUCCESSFUL FLANGE JOINTS OFTEN STRESSED ABOVE CODE ALLOWANCE: EXPERIENCE

Stress

FLANGES

Example 1, Page 12 shows the stresses computed for Part A flanges. Each stress must be equal or less than the Code allowable except for the longitudinal hub stress which may reach 1.5 x the allowable. This is permitted because S_H is a bending stress. The formula gives a maximum value which only exists on the inside and outside surfaces of the hub, and decreases to zero at a point half way between. If a slight overstress in the hub causes yielding, the load shifts to the ring portion of the flange. The ring is also subjected primarily to bending and thus able to absorb the additional load so that a new equilibrium within safe limits is established. When localized yielding extends to the point where the flange and hub cease to act elastically as an integral structure, the flange takes a permanent set or "dishes" and the joint usually cannot be disassembled and remade in a satisfactory manner. This is likely to occur when an integral flange has a thin straight hub because bending stress and direct pressure stresses must be carried at the same time. If there is cyclic loading, fatigue, or operation at high temperature in the creep range, with the stress above the yield point over a considerable portion of the hub, the use of $S_H = 1.5$ allowable should be reviewed.

It is reasonable to expect that stress concentrations higher in magnitude could be withstood before the flange ceases to act elastically but at this time methods for setting this critical level accurately are very laborious. These limitations apply to ductile materials having elongations of 15% or more (in 2″) and yield points of 50% to 70% of the ultimate tensile strength. High strength materials may have yield points very close to the ultimate tensile strength or no observable yield point, and with such materials the higher hub stress should only be used after careful consideration.

The existence of localized stresses, stress concentrations, and discontinuity stresses of a relatively high order in all pressure equipment is well known. Typical examples are stress concentrations around nozzle and branch connections, stiffener rings or flat head attachments. The Code Committees and the Pressure Vessel Research Committee are currently studying such effects in an effort to arrive at a better method of evaluating safe limits for design. It is not likely that any simple answer will be found. The character of the stress and the area over which it exists must be evaluated in addition to the capacity of the adjacent structure to carry increased load. The Code accounts for localized stresses by using compensating factors in the design formulas for stress.

In flange design, the most obvious example of stress concentration occurs with hubbed flanges at the corner where the outside surface of the hub meets the back face of the flange. Integral flanges provide for this with a fillet radius that is required to be 0.25 g_0 minimum — but never less than 0.188 inches.

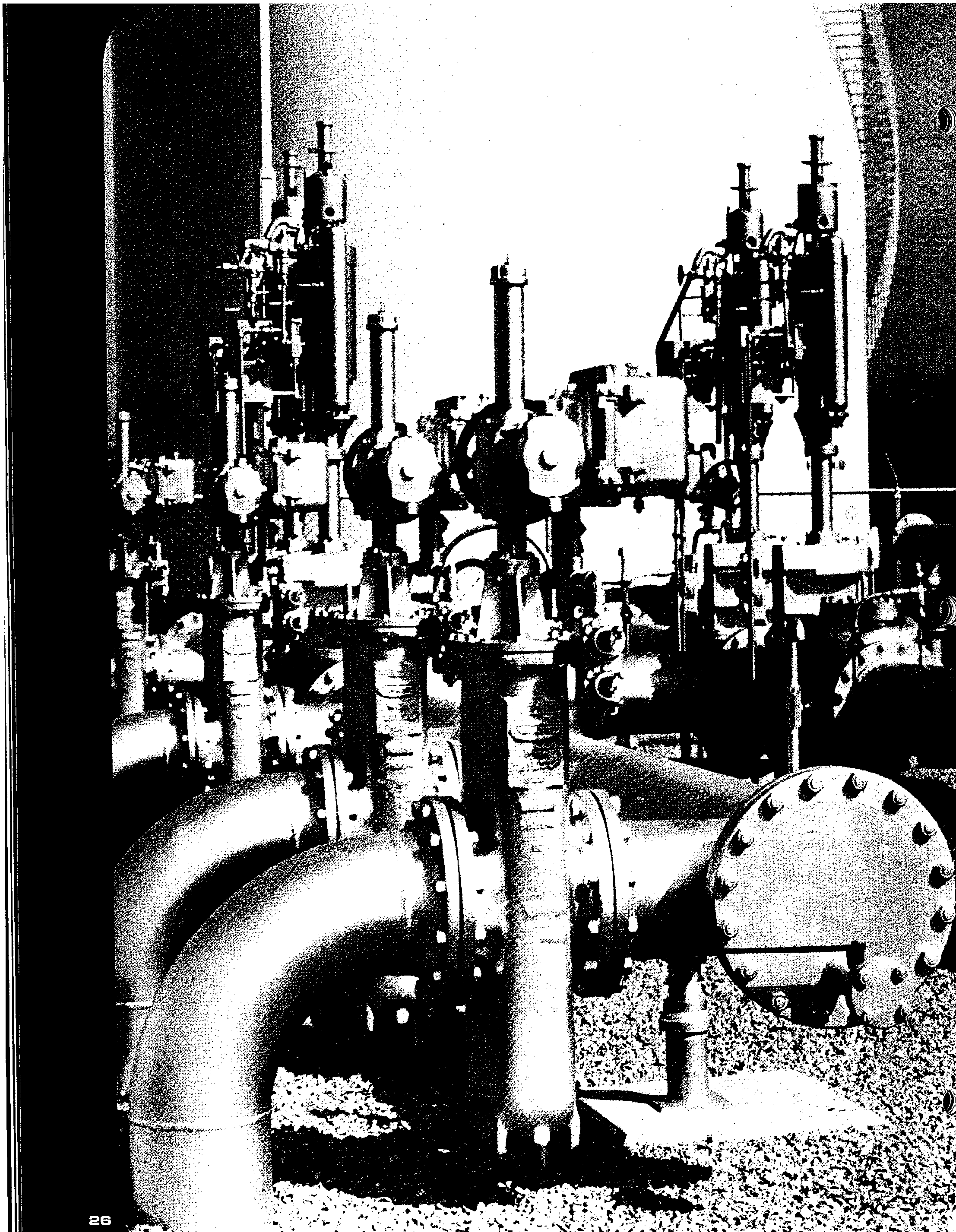
BOLTS

The allowable bolt stresses listed in the Code may seem to be too conservative, but they were very carefully selected by the Committees. Particular reference was made to stresses in flange bolting. It was recognized that bolts are often stressed above the tabulated values, and that relaxation occurs in many service applications due to creep in the gasket, flange or bolt material.

Appendix S of the Code distinguishes between the design value of bolt stress and that which might actually be needed for both design and test conditions. The initial tightening of the bolts prior to hydrostatic test may be the most severe load that they will receive. It is sometimes thought that bolts stressed 1½ times the allowable will pass the test, but this is not necessarily true. As internal pressure is applied, flange rotation and gasket properties may actually lower bolt load. To offset this, the bolts must be tightened as internal pressure is added, or there will be leakage. After the test, the flange will tend to relax to its original flatness, the gasket will recover a large part of its original thickness, and the twice tightened bolts may be loaded beyond the design value. Bolt capacity for higher initial loading is desirable and experience bears this out.

Code allowable bolt streses also account for severe torsion and bending during service. The actual tension achieved in each bolt is never exactly uniform whether by hand wrenching with plain or torque wrenches, power driven impact wrenches, or hydraulic bolt tensioners. Tests show that manual tightening develops bolt stresses as high as 45,000/ $\sqrt{a}$ psi. This indicates why a potential problem exists when bolts ½″ and less are specified.

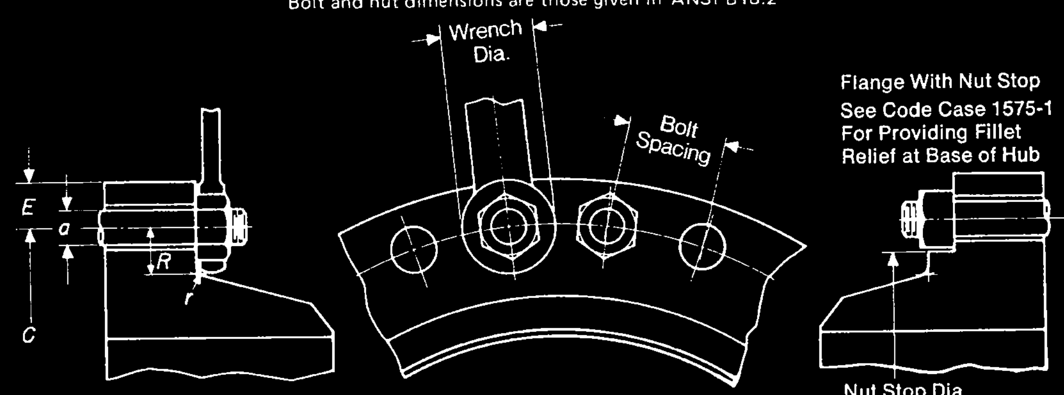
Allowable bolt stress is related to bolt temperature and in uninsulated piping the bolts are cooler than the contained medium. It is suggested that the higher stresses permitted at lower temperature be avoided unless reliable test data is available and specifications permit their use. See ANSI B31.3 Paragraph 301.3.1.



Technical Data

Bolt Size a	Coarse Thread Series		8 Thread Series	Nut Dimensions		Minimum			Maximum Fillet Radius r	Wrench Diameter
	Threads per Inch	Root Area Sq. In.	Root Area Sq. In.	Across Flats	Across Corners	Bolt Spacing	Radial Distance R	Edge Distance E		
1/2	13	.126	- -	7/8	.969	1 1/4	13/16	5/8	1/4	1 1/2
5/8	11	.202	- -	1 1/16	1.175	1 1/2	15/16	3/4	5/16	1 3/4
3/4	10	.302	- -	1 1/4	1.383	1 3/4	1 1/8	13/16	3/8	2 1/16
7/8	9	.419	- -	1 7/16	1.589	2 1/16	1 1/4	15/16	3/8	2 3/8
1	8	.551	.551	1 5/8	1.796	2 1/4	1 3/8	1 1/16	7/16	2 5/8
1 1/8	7	.693	.728	1 13/16	2.002	2 1/2	1 1/2	1 1/8	7/16	2 7/8
1 1/4	7	.890	.929	2	2.209	2 13/16	1 3/4	1 1/4	9/16	3 1/4
1 3/8	6	1.054	1.155	2 3/16	2.416	3 1/16	1 7/8	1 3/8	9/16	3 1/2
1 1/2	6	1.294	1.405	2 3/8	2.622	3 1/4	2	1 1/2	5/8	3 3/4
1 5/8	5 1/2	1.515	1.680	2 9/16	2.828	3 1/2	2 1/8	1 5/8	5/8	4
1 3/4	5	1.744	1.980	2 3/4	3.035	3 3/4	2 1/4	1 3/4	5/8	4 1/4
1 7/8	5	2.049	2.304	2 15/16	3.242	4	2 3/8	1 7/8	5/8	4 1/2
2	4 1/2	2.300	2.652	3 1/8	3.449	4 1/4	2 1/2	2	11/16	4 3/4
2 1/4	4 1/2	3.020	3.423	3 1/2	3.862	4 3/4	2 3/4	2 1/4	11/16	5 1/4
2 1/2	4	3.715	4.292	3 7/8	4.275	5 1/4	3 1/16	2 3/8	13/16	5 7/8
2 3/4	4	4.618	5.259	4 1/4	4.688	5 3/4	3 3/8	2 5/8	7/8	6 1/2
3	4	5.621	6.324	4 5/8	5.102	6 1/4	3 5/8	2 7/8	15/16	7

Bolt and nut dimensions are those given in ANSI B18.2



$K = A/B$

The values of T, Z, Y, and U in Table 2 have been computed using Poisson's ratio = 0.3 and cover $1.00 \leq K \leq 5.00$. The increments of K are chosen to provide calculation accuracy that is consistent with formulas of the Code. Values of T, Z, Y, and U are listed with three of four significant figures, and the difference between successive tabulations is not greater than 1% in the range $1.100 \leq K \leq 5.00$. Linear interpolation is adequate for values of Z, Y, and U when $1.000 \leq K \leq 1.020$ where T is almost constant.

When K ranges less than 1.020, the values of Z, Y, and U change rapidly and the Interpolation Formulas should be used as follows:

$$Z = \frac{1}{\varepsilon} + \frac{1}{2} + \frac{\varepsilon}{4}$$

$$U = 1.0493 \left[\frac{2}{\varepsilon} + 1.3\left(1 - \frac{\varepsilon}{6}\right)\right]$$

T as below; $Y = (1 - \nu^2)\, U$

where $\varepsilon = K - 1$; $\nu = 0.3$

When K is beyond the scope of Table 2, or a material with a different Poisson's ratio is required, calculate T, Z, Y and U according to the following formulas. These are equal to those in the Code Figure $UA - 51.1$ when Poisson's ratio $U = 0.3$.

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

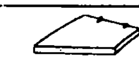
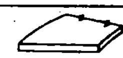
$$Z = \frac{K^2+1}{K^2-1}\ ;\ Y = (1-\nu^2)\, U$$

$$U = \frac{K^2\left(1 + 4.6052\, \frac{1+\nu}{1-\nu}\, \log_{10} K\right) - 1}{1.04720\,(K^2-1)\,(K-1)\,(1+\nu)}$$

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	1.031	1.90	32.76	62.85	69.06	1.061	1.89	16.91	32.55	35.78
1.002	1.91	500.50	956.16	1050.72	1.032	1.90	31.76	60.92	66.94	1.062	1.89	16.64	32.04	35.21
1.003	1.91	333.83	637.85	700.93	1.033	1.90	30.81	59.11	64.95	1.063	1.89	16.40	31.55	34.68
1.004	1.91	250.50	478.71	526.05	1.034	1.90	29.92	57.41	63.08	1.064	1.89	16.15	31.08	34.17
1.005	1.91	200.50	383.22	421.12	1.035	1.90	29.08	55.80	61.32	1.065	1.89	15.90	30.61	33.65
1.006	1.91	167.17	319.56	351.16	1.036	1.90	28.29	54.29	59.66	1.066	1.89	15.67	30.17	33.17
1.007	1.91	143.36	274.09	301.20	1.037	1.90	27.54	52.85	58.08	1.067	1.89	15.45	29.74	32.69
1.008	1.91	125.50	239.95	263.75	1.038	1.90	26.83	51.50	56.59	1.068	1.89	15.22	29.32	32.22
1.009	1.91	111.61	213.40	234.42	1.039	1.90	26.15	50.21	55.17	1.069	1.89	15.02	28.91	31.79
1.010	1.91	100.50	192.19	211.19	1.040	1.90	25.51	48.97	53.82	1.070	1.89	14.80	28.51	31.34
1.011	1.91	91.41	174.83	192.13	1.041	1.90	24.90	47.81	53.10	1.071	1.89	14.61	28.13	30.92
1.012	1.91	83.84	160.38	176.25	1.042	1.90	24.32	46.71	51.33	1.072	1.89	14.41	27.76	30.51
1.013	1.91	77.43	148.06	162.81	1.043	1.90	23.77	45.64	50.15	1.073	1.89	14.22	27.39	30.11
1.014	1.91	71.93	137.69	151.30	1.044	1.90	23.23	44.64	49.05	1.074	1.88	14.04	27.04	29.72
1.015	1.91	67.17	128.61	141.33	1.045	1.90	22.74	43.69	48.02	1.075	1.88	13.85	26.69	29.34
1.016	1.90	63.00	120.56	132.49	1.046	1.90	22.05	42.75	46.99	1.076	1.88	13.68	26.36	28.98
1.017	1.90	59.33	111.98	124.81	1.047	1.90	21.79	41.87	46.03	1.077	1.88	13.56	26.03	28.69
1.018	1.90	56.06	107.36	118.00	1.048	1.90	21.35	41.02	45.09	1.078	1.88	13.35	25.72	28.27
1.019	1.90	53.14	101.72	111.78	1.049	1.90	20.92	40.21	44.21	1.079	1.88	13.18	25.40	27.92
1.020	1.90	50.51	96.73	106.30	1.050	1.89	20.51	39.43	43.34	1.080	1.88	13.02	25.10	27.59
1.021	1.90	48.12	92.21	101.33	1.051	1.89	20.12	38.68	42.51	1.081	1.88	12.87	24.81	27.27
1.022	1.90	45.96	88.04	96.75	1.052	1.89	19.74	37.96	41.73	1.082	1.88	12.72	24.52	26.95
1.023	1.90	43.98	84.30	92.64	1.053	1.89	19.38	37.27	40.96	1.083	1.88	12.57	24.24	26.65
1.024	1.90	42.17	80.81	88.81	1.054	1.89	19.03	36.60	40.23	1.084	1.88	12.43	24.00	26.34
1.025	1.90	40.51	77.61	85.29	1.055	1.89	18.69	35.96	39.64	1.085	1.88	12.29	23.69	26.05
1.026	1.90	38.97	74.70	82.09	1.056	1.89	18.38	35.34	38.84	1.086	1.88	12.15	23.44	25.77
1.027	1.90	37.54	71.97	79.08	1.057	1.89	18.06	34.74	38.19	1.087	1.88	12.02	23.18	25.48
1.028	1.90	36.22	69.43	76.30	1.058	1.89	17.76	34.17	37.56	1.088	1.88	11.89	22.93	25.20
1.029	1.90	34.99	67.11	73.75	1.059	1.89	17.47	33.62	36.95	1.089	1.88	11.76	22.68	24.93
1.030	1.90	33.84	64.91	71.33	1.060	1.89	17.18	33.04	36.34	1.090	1.88	11.63	22.44	24.66
1.091	1.88	11.52	22.22	24.41	1.151	1.86	7.16	13.86	15.23	1.242	1.82	4.69	9.08	9.98
1.092	1.88	11.40	21.99	24.16	1.152	1.86	7.11	13.77	15.14	1.244	1.82	4.65	9.02	9.91
1.093	1.88	11.28	21.76	23.91	1.153	1.86	7.07	13.69	15.05	1.246	1.82	4.62	8.95	9.84
1.094	1.88	11.16	21.54	23.67	1.154	1.86	7.03	13.61	14.96	1.248	1.82	4.59	8.89	9.77
1.095	1.88	11.05	21.32	23.44	1.155	1.86	6.99	13.54	14.87	1.250	1.82	4.56	8.83	9.70
1.096	1.88	10.94	21.11	23.20	1.156	1.86	6.95	13.45	14.78	1.252	1.82	4.52	8.77	9.64
1.097	1.88	10.83	20.91	22.97	1.157	1.86	6.91	13.37	14.70	1.254	1.82	4.49	8.71	9.57
1.098	1.88	10.73	20.71	22.75	1.158	1.86	6.87	13.30	14.61	1.256	1.82	4.46	8.65	9.51
1.099	1.88	10.62	20.51	22.39	1.159	1.86	6.83	13.22	14.53	1.258	1.81	4.43	8.59	9.44
1.100	1.88	10.52	20.31	22.18	1.160	1.86	6.79	13.15	14.45	1.260	1.81	4.40	8.53	9.38
1.101	1.88	10.43	20.15	22.12	1.161	1.85	6.75	13.07	14.36	1.263	1.81	4.36	8.45	9.28
1.102	1.88	10.33	19.94	21.92	1.162	1.85	6.71	13.00	14.28	1.266	1.81	4.32	8.37	9.19
1.103	1.88	10.23	19.76	21.72	1.163	1.85	6.67	12.92	14.20	1.269	1.81	4.28	8.29	9.11
1.104	1.88	10.14	19.58	21.52	1.164	1.85	6.64	12.85	14.12	1.272	1.81	4.24	8.21	9.02
1.105	1.88	10.05	19.38	21.30	1.165	1.85	6.60	12.78	14.04	1.275	1.81	4.20	8.13	8.93
1.106	1.88	9.96	19.33	21.14	1.166	1.85	6.56	12.71	13.97	1.278	1.81	4.16	8.05	8.85
1.107	1.87	9.87	19.07	20.96	1.167	1.85	6.53	12.64	13.89	1.281	1.81	4.12	7.98	8.77
1.108	1.87	9.78	18.90	20.77	1.168	1.85	6.49	12.58	13.82	1.284	1.80	4.08	7.91	8.69
1.109	1.87	9.70	18.74	20.59	1.169	1.85	6.46	12.51	13.74	1.287	1.80	4.05	7.84	8.61
1.110	1.87	9.62	18.55	20.38	1.170	1.85	6.42	12.43	13.66	1.290	1.80	4.01	7.77	8.53
1.111	1.87	9.54	18.42	20.25	1.171	1.85	6.39	12.38	13.60	1.293	1.80	3.98	7.70	8.46
1.112	1.87	9.46	18.27	20.08	1.172	1.85	6.35	12.31	13.53	1.296	1.80	3.94	7.63	8.39
1.113	1.87	9.38	18.13	19.91	1.173	1.85	6.32	12.25	13.46	1.299	1.80	3.91	7.57	8.31
1.114	1.87	9.30	17.97	19.75	1.174	1.85	6.29	12.18	13.39	1.302	1.80	3.88	7.50	8.24
1.115	1.87	9.22	17.81	19.55	1.175	1.85	6.25	12.10	13.30	1.305	1.80	3.84	7.44	8.18
1.116	1.87	9.15	17.68	19.43	1.176	1.85	6.22	12.06	13.25	1.308	1.79	3.81	7.38	8.11
1.117	1.87	9.07	17.54	19.27	1.177	1.85	6.19	12.00	13.18	1.311	1.79	3.78	7.32	8.05
1.118	1.87	9.00	17.40	19.12	1.178	1.85	6.16	11.93	13.11	1.314	1.79	3.75	7.26	7.98
1.119	1.87	8.94	17.27	18.98	1.179	1.85	6.13	11.87	13.05	1.317	1.79	3.72	7.20	7.92
1.120	1.87	8.86	17.13	18.80	1.180	1.85	6.10	11.79	12.96	1.320	1.79	3.69	7.14	7.85
1.121	1.87	8.79	17.00	18.68	1.182	1.85	6.04	11.70	12.86	1.323	1.79	3.67	7.09	7.79
1.122	1.87	8.72	16.87	18.54	1.184	1.85	5.98	11.58	12.73	1.326	1.79	3.64	7.03	7.73
1.123	1.87	8.66	16.74	18.40	1.186	1.85	5.92	11.47	12.61	1.329	1.78	3.61	6.98	7.67
1.124	1.87	8.59	16.62	18.26	1.188	1.85	5.86	11.36	12.49	1.332	1.78	3.58	6.92	7.61
1.125	1.87	8.53	16.49	18.11	1.190	1.84	5.81	11.26	12.37	1.335	1.78	3.56	6.87	7.55
1.126	1.87	8.47	16.37	17.99	1.192	1.84	5.75	11.15	12.25	1.338	1.78	3.53	6.82	7.50
1.127	1.87	8.40	16.25	17.86	1.194	1.84	5.70	11.05	12.14	1.341	1.78	3.51	6.77	7.44
1.128	1.87	8.34	16.14	17.73	1.196	1.84	5.65	10.95	12.03	1.344	1.78	3.48	6.72	7.39
1.129	1.87	8.28	16.02	17.60	1.198	1.84	5.60	10.85	11.92	1.347	1.78	3.46	6.68	7.33
1.130	1.87	8.22	15.91	17.48	1.200	1.84	5.55	10.75	11.81	1.350	1.78	3.43	6.63	7.28

K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U
1.131	1.87	8.16	15.79	17.35	1.202	1.84	5.50	10.65	11.71	1.354	1.77	3.40	6.57	7.21
1.132	1.87	8.11	15.68	17.24	1.204	1.84	5.45	10.56	11.61	1.358	1.77	3.37	6.50	7.14
1.133	1.86	8.05	15.57	17.11	1.206	1.84	5.40	10.47	11.51	1.362	1.77	3.34	6.44	7.08
1.134	1.86	7.99	15.46	16.99	1.208	1.84	5.35	10.38	11.41	1.366	1.77	3.31	6.38	7.01
1.135	1.86	7.94	15.36	16.90	1.210	1.84	5.31	10.30	11.32	1.370	1.77	3.28	6.32	6.95
1.136	1.86	7.88	15.26	16.77	1.212	1.83	5.27	10.21	11.22	1.374	1.77	3.25	6.27	6.89
1.137	1.86	7.83	15.15	16.65	1.214	1.83	5.22	10.12	11.12	1.378	1.76	3.22	6.21	6.82
1.138	1.86	7.78	15.05	16.54	1.216	1.83	5.18	10.04	11.03	1.382	1.76	3.20	6.16	6.77
1.139	1.86	7.73	14.95	16.43	1.218	1.83	5.14	9.96	10.94	1.386	1.76	3.17	6.11	6.72
1.140	1.86	7.68	14.86	16.35	1.220	1.83	5.10	9.89	10.87	1.390	1.76	3.15	6.06	6.66
1.141	1.86	7.62	14.76	16.22	1.222	1.83	5.05	9.80	10.77	1.394	1.76	3.12	6.01	6.60
1.142	1.86	7.57	14.66	16.11	1.224	1.83	5.01	9.72	10.68	1.398	1.75	3.10	5.96	6.55
1.143	1.86	7.53	14.57	16.01	1.226	1.83	4.98	9.65	10.60	1.402	1.75	3.07	5.92	6.49
1.144	1.86	7.48	14.48	15.91	1.228	1.83	4.94	9.57	10.52	1.406	1.75	3.05	5.87	6.44
1.145	1.86	7.43	14.39	15.83	1.230	1.83	4.90	9.50	10.44	1.410	1.75	3.02	5.82	6.39
1.146	1.86	7.38	14.29	15.71	1.232	1.83	4.86	9.43	10.36	1.414	1.75	3.00	5.77	6.34
1.147	1.86	7.34	14.20	15.61	1.234	1.83	4.83	9.36	10.28	1.418	1.75	2.98	5.72	6.29
1.148	1.86	7.29	14.12	15.51	1.236	1.82	4.79	9.29	10.20	1.422	1.75	2.96	5.68	6.25
1.149	1.86	7.25	14.03	15.42	1.238	1.82	4.76	9.22	10.13	1.426	1.74	2.94	5.64	6.20
1.150	1.86	7.20	13.95	15.34	1.240	1.82	4.72	9.15	10.05	1.430	1.74	2.91	5.60	6.15
1.434	1.74	2.89	5.56	6.10	1.85	1.56	1.83	3.33	3.65	2.92	1.22	1.27	1.92	2.11
1.438	1.74	2.87	5.52	6.05	1.86	1.56	1.81	3.30	3.62	2.95	1.22	1.26	1.90	2.09
1.442	1.74	2.85	5.48	6.01	1.87	1.56	1.80	3.27	3.59	2.98	1.21	1.25	1.88	2.07
1.446	1.74	2.83	5.44	5.97	1.88	1.55	1.79	3.24	3.56	3.02	1.20	1.25	1.86	2.04
1.450	1.73	2.81	5.40	5.93	1.89	1.55	1.78	3.22	3.54	3.06	1.19	1.24	1.83	2.01
1.454	1.73	2.80	5.36	5.89	1.90	1.54	1.77	3.19	3.51	3.10	1.18	1.23	1.81	1.99
1.458	1.73	2.78	5.32	5.85	1.91	1.54	1.75	3.17	3.48	3.14	1.17	1.23	1.79	1.97
1.462	1.73	2.76	5.28	5.80	1.92	1.54	1.74	3.14	3.45	3.18	1.16	1.22	1.77	1.94
1.466	1.73	2.74	5.24	5.76	1.93	1.53	1.73	3.12	3.43	3.22	1.16	1.21	1.75	1.92
1.470	1.72	2.72	5.20	5.71	1.94	1.53	1.72	3.09	3.40	3.26	1.15	1.21	1.73	1.90
1.475	1.72	2.70	5.16	5.66	1.95	1.53	1.71	3.07	3.38	3.30	1.14	1.20	1.71	1.88
1.480	1.72	2.68	5.12	5.61	1.96	1.52	1.70	3.05	3.35	3.34	1.13	1.20	1.69	1.86
1.485	1.72	2.66	5.08	5.57	1.97	1.52	1.69	3.03	3.33	3.38	1.12	1.19	1.67	1.84
1.490	1.72	2.64	5.04	5.53	1.98	1.51	1.68	3.01	3.30	3.42	1.11	1.19	1.66	1.82
1.495	1.71	2.62	5.00	5.49	1.99	1.51	1.68	2.98	3.28	3.46	1.11	1.18	1.64	1.80
1.500	1.71	2.60	4.96	5.45	2.00	1.51	1.67	2.96	3.26	3.50	1.10	1.18	1.62	1.78
1.505	1.71	2.58	4.92	5.41	2.01	1.50	1.66	2.94	3.23	3.54	1.09	1.17	1.61	1.76
1.510	1.71	2.56	4.88	5.37	2.02	1.50	1.65	2.92	3.21	3.58	1.08	1.17	1.59	1.75
1.515	1.71	2.54	4.84	5.33	2.04	1.49	1.63	2.88	3.17	3.62	1.07	1.16	1.57	1.73
1.520	1.70	2.53	4.80	5.29	2.06	1.48	1.62	2.85	3.13	3.66	1.07	1.16	1.56	1.71
1.525	1.70	2.51	4.77	5.25	2.08	1.48	1.60	2.81	3.09	3.70	1.06	1.16	1.55	1.70
1.530	1.70	2.49	4.74	5.21	2.10	1.47	1.59	2.78	3.05	3.74	1.05	1.15	1.53	1.68
1.535	1.70	2.47	4.70	5.17	2.12	1.46	1.57	2.74	3.01	3.78	1.05	1.15	1.52	1.67
1.540	1.69	2.46	4.66	5.13	2.14	1.46	1.56	2.71	2.97	3.82	1.04	1.15	1.50	1.65
1.545	1.69	2.44	4.63	5.09	2.16	1.45	1.55	2.67	2.94	3.86	1.03	1.14	1.49	1.64
1.55	1.69	2.43	4.60	5.05	2.18	1.44	1.53	2.64	2.90	3.90	1.03	1.14	1.48	1.62
1.56	1.69	2.40	4.54	4.99	2.20	1.44	1.52	2.61	2.87	3.94	1.02	1.14	1.46	1.61
1.57	1.68	2.37	4.48	4.92	2.22	1.43	1.51	2.58	2.84	3.98	1.01	1.13	1.45	1.60
1.58	1.68	2.34	4.42	4.86	2.24	1.42	1.50	2.56	2.81	4.00	1.009	1.13	1.45	1.59
1.59	1.67	2.31	4.36	4.79	2.26	1.41	1.49	2.53	2.78	4.05	1.002	1.13	1.43	1.57
1.60	1.67	2.28	4.31	4.73	2.28	1.41	1.48	2.50	2.75	4.10	.996	1.13	1.42	1.56
1.61	1.66	2.26	4.25	4.67	2.30	1.40	1.47	2.48	2.72	4.15	.989	1.12	1.40	1.54
1.62	1.65	2.23	4.20	4.61	2.32	1.40	1.46	2.45	2.69	4.20	.982	1.12	1.39	1.53
1.63	1.65	2.21	4.15	4.56	2.34	1.39	1.45	2.43	2.67	4.25	.975	1.12	1.38	1.51
1.64	1.65	2.18	4.10	4.50	2.36	1.38	1.44	2.40	2.64	4.30	.968	1.11	1.36	1.50
1.65	1.65	2.16	4.05	4.45	2.38	1.38	1.43	2.38	2.61	4.35	.962	1.11	1.35	1.48
1.66	1.64	2.14	4.01	4.40	2.40	1.37	1.42	2.36	2.59	4.40	.955	1.11	1.34	1.47
1.67	1.64	2.12	3.96	4.35	2.42	1.36	1.41	2.33	2.56	4.45	.948	1.11	1.33	1.46
1.68	1.63	2.10	3.92	4.30	2.44	1.36	1.40	2.31	2.54	4.50	.941	1.10	1.31	1.44
1.69	1.63	2.08	3.87	4.26	2.46	1.35	1.40	2.29	2.52	4.55	.934	1.10	1.30	1.43
1.70	1.63	2.06	3.83	4.21	2.48	1.35	1.39	2.27	2.50	4.60	.928	1.10	1.29	1.42
1.71	1.62	2.04	3.79	4.17	2.50	1.34	1.38	2.25	2.47	4.65	.921	1.10	1.28	1.41
1.72	1.62	2.02	3.75	4.12	2.53	1.33	1.37	2.22	2.44	4.70	.914	1.09	1.27	1.39
1.73	1.61	2.00	3.72	4.08	2.56	1.32	1.36	2.19	2.41	4.75	.908	1.09	1.26	1.38
1.74	1.61	1.99	3.68	4.04	2.59	1.31	1.35	2.17	2.38	4.80	.900	1.09	1.25	1.37
1.75	1.60	1.97	3.64	4.00	2.62	1.30	1.34	2.14	2.35	4.85	.893	1.09	1.24	1.36
1.76	1.60	1.95	3.61	3.96	2.65	1.30	1.33	2.12	2.32	4.90	.887	1.09	1.23	1.35
1.77	1.60	1.94	3.57	3.93	2.68	1.29	1.32	2.09	2.30	4.95	.880	1.08	1.22	1.34
1.78	1.59	1.92	3.54	3.89	2.71	1.28	1.31	2.07	2.27	5.00	.873	1.08	1.21	1.33
1.79	1.59	1.91	3.51	3.85	2.74	1.27	1.31	2.04	2.25					
1.80	1.58	1.89	3.47	3.82	2.77	1.26	1.30	2.02	2.22					
1.81	1.58	1.88	3.44	3.78	2.80	1.26	1.29	2.00	2.20					
1.82	1.58	1.86	3.41	3.75	2.83	1.25	1.28	1.98	2.17					
1.83	1.57	1.85	3.38	3.72	2.86	1.24	1.28	1.96	2.15					
1.84	1.57	1.84	3.35	3.69	2.89	1.23	1.27	1.94	2.13					

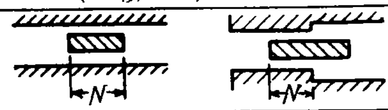
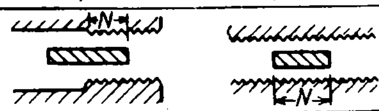
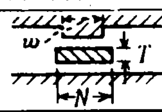
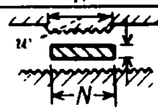
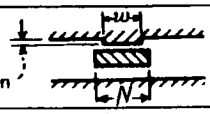
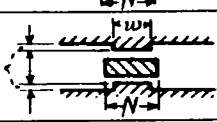
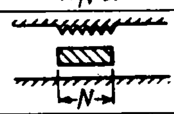
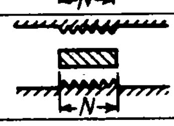
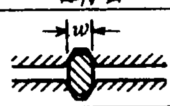
TABLE UA-49.1
GASKET MATERIALS AND CONTACT FACINGS[1]
Gasket Factors (m) for Operating Conditions and Minimum Design Seating Stress (y)

Gasket Material		Gasket Factor m	Min. Design Seating Stress y	Sketches and Notes	Use Facing Sketch (Refer to Table UA-49.2)	Use Column (Refer to Table UA-49.2)
Self-Energizing Types O Rings, Metallic, Elastomer other gasket types considered as self-sealing		0	0	...	...	...
Elastomers without fabric or a high percentage of asbestos fiber: Below 75A Shore Durometer 75A or higher Shore Durometer		 0.50 1.00	 0 200		(1a),(1b), (1c),(1d), (4),(5)	II
Asbestos with a suitable binder for the operating conditions	⅛ thick $\frac{1}{16}$ thick $\frac{1}{32}$ thick	2.00 2.75 3.50	1600 3700 6500		(1a),(1b), (1c),(1d), (4),(5)	II
Elastomers with cotton fabric insertion		1.25	400		(1a),(1b), (1c),(1d), (4),(5)	II
Elastomers with asbestos fabric insertion, with or without wire reinforcement	3-ply	2.25	2200		(1a),(1b), (1c),(1d), (4),(5)	II
	2-ply	2.50	2900			
	1-ply	2.75	3700			
Vegetable fiber		1.75	1100		(1a),(1b), (1c),(1d), (4),(5)	II
Spiral-wound metal, asbestos filled	Carbon Stainless or Monel	2.50 3.00	10,000 10,000		(1a),(1b)	II
Corrugated metal, asbestos inserted or Corrugated metal, jacketed asbestos filled	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4–6% chrome Stainless steels	2.50 2.75 3.00 3.25 3.50	2900 3700 4500 5500 6500		(1a),(1b)	II
Corrugated metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4–6% chrome Stainless steels	2.75 3.00 3.25 3.50 3.75	3700 4500 5500 6500 7600		(1a),(1b), (1c),(1d)	II
Flat metal jacketed asbestos filled	Soft aluminum Soft copper or brass Iron or soft steel Monel 4–6% chrome Stainless steels	3.25 3.50 3.75 3.50 3.75 3.75	5500 6500 7600 8000 9000 9000		(1a),(1b), (1c),[2](1d),[2] (2)[2]	II
Grooved metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4–6% chrome Stainless steels	3.25 3.50 3.75 3.75 4.25	5500 6500 7600 9000 10100		(1a),(1b), (1c),(1d), (2),(3)	II
Solid flat metal	Soft aluminum Soft copper or brass Iron or soft steel Monel or 4–6% chrome Stainless steels	4.00 4.75 5.50 6.00 6.50	8800 13000 18000 21800 26000		(1a),(1b), (1c),(1d), (2),(3), (4),(5)	I
Ring joint	Iron or soft steel Monel or 4–6% chrome Stainless steels	5.50 6.00 6.50	18000 21800 26000		(6)	I

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 UA-49.2. The design values and other details given in this table are suggested only and are not mandatory.
(2) The surface of a gasket having a lap should not be against the nubbin.

*From ASME Code Section VIII Div. 1—Reprinted by permission.

TABLE UA-49.2
EFFECTIVE GASKET WIDTH

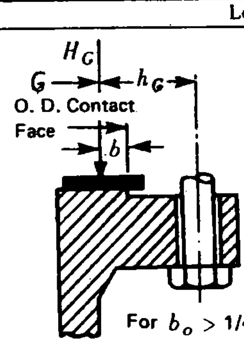
Facing Sketch (Exaggerated)			Basic Gasket Seating Width, b_o	
			Column I	Column II
(1a)			$\frac{N}{2}$	$\frac{N}{2}$
(1b)[1]				
(1c)		$w \gtreqless N$	$\frac{w+T}{2}$; $\left(\frac{w+N}{4}\ max\right)$	$\frac{w+T}{2}$; $\left(\frac{w+N}{4}\ max\right)$
(1d)[1]		$w \gtreqless N$		
(2) 1/64 in. Nubbin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

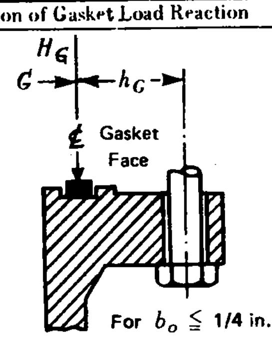
Effective Gasket Seating Width, "b"

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

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

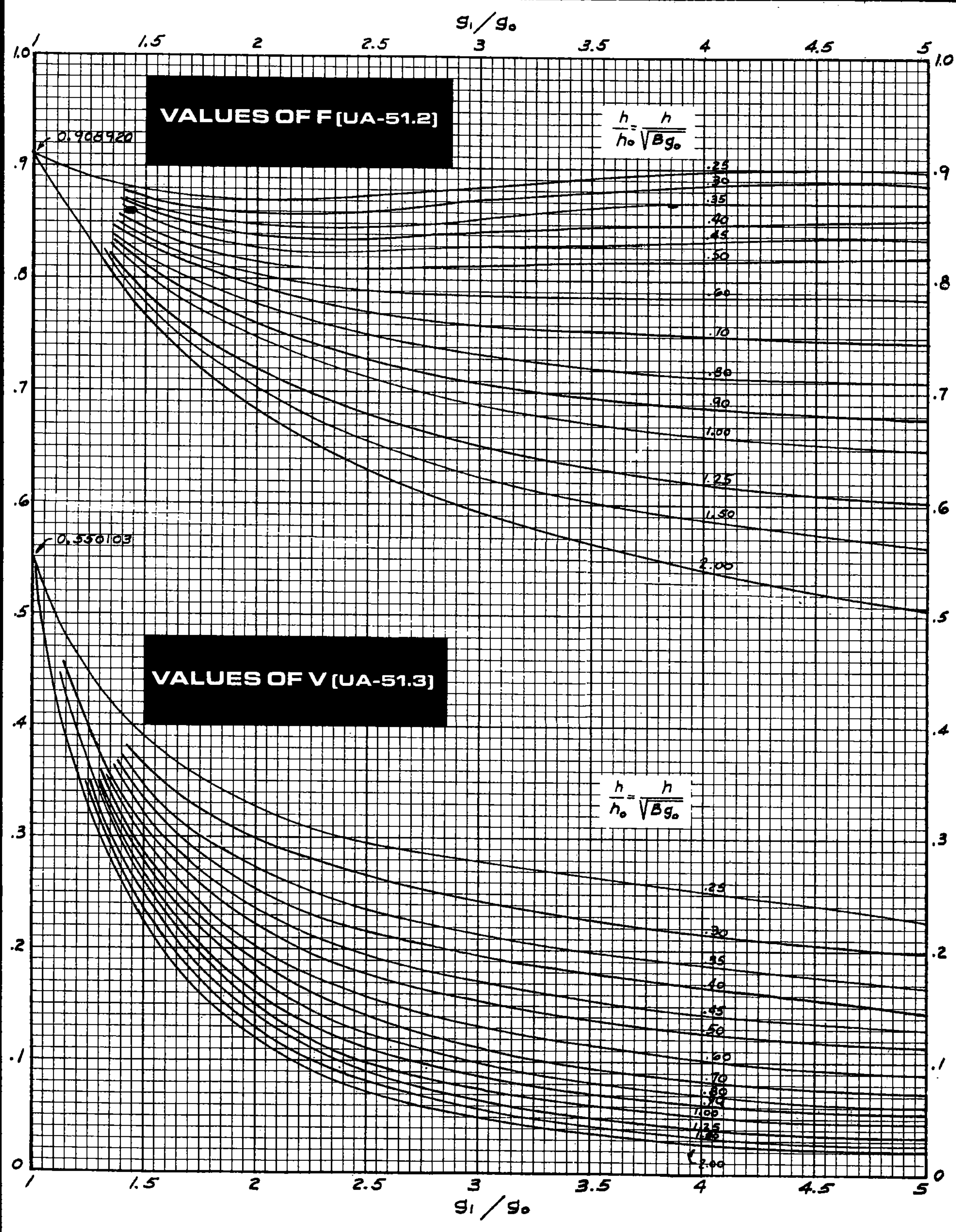
Location of Gasket Load Reaction





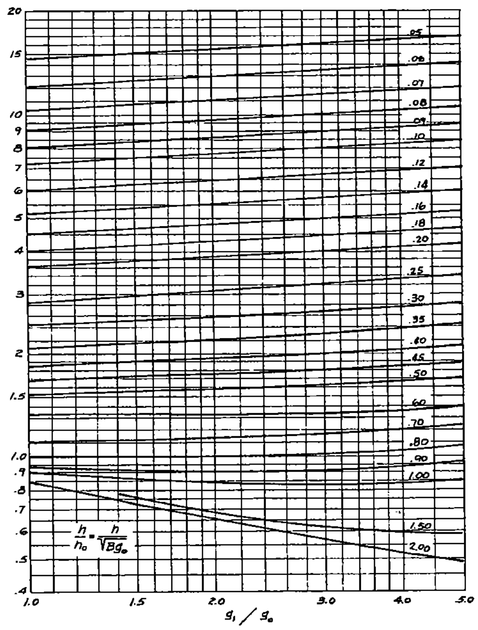
NOTE: The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes

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

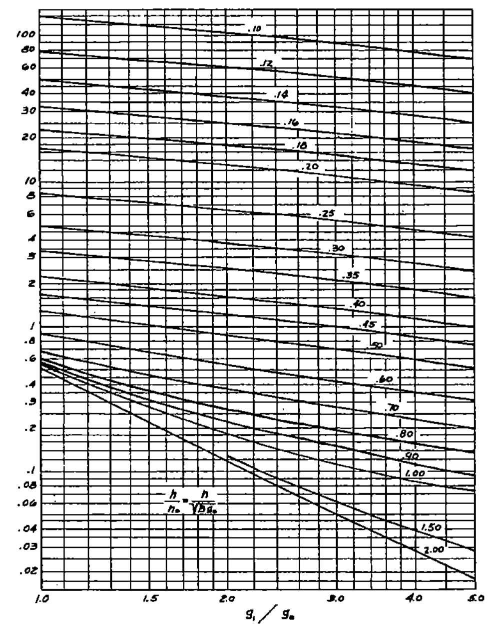


VALUES OF F [UA-51.2]
VALUES OF V [UA-51.3]
$\frac{h}{h_o} = \frac{h}{\sqrt{Bg_o}}$
g_1 / g_o
0.908920
0.550103

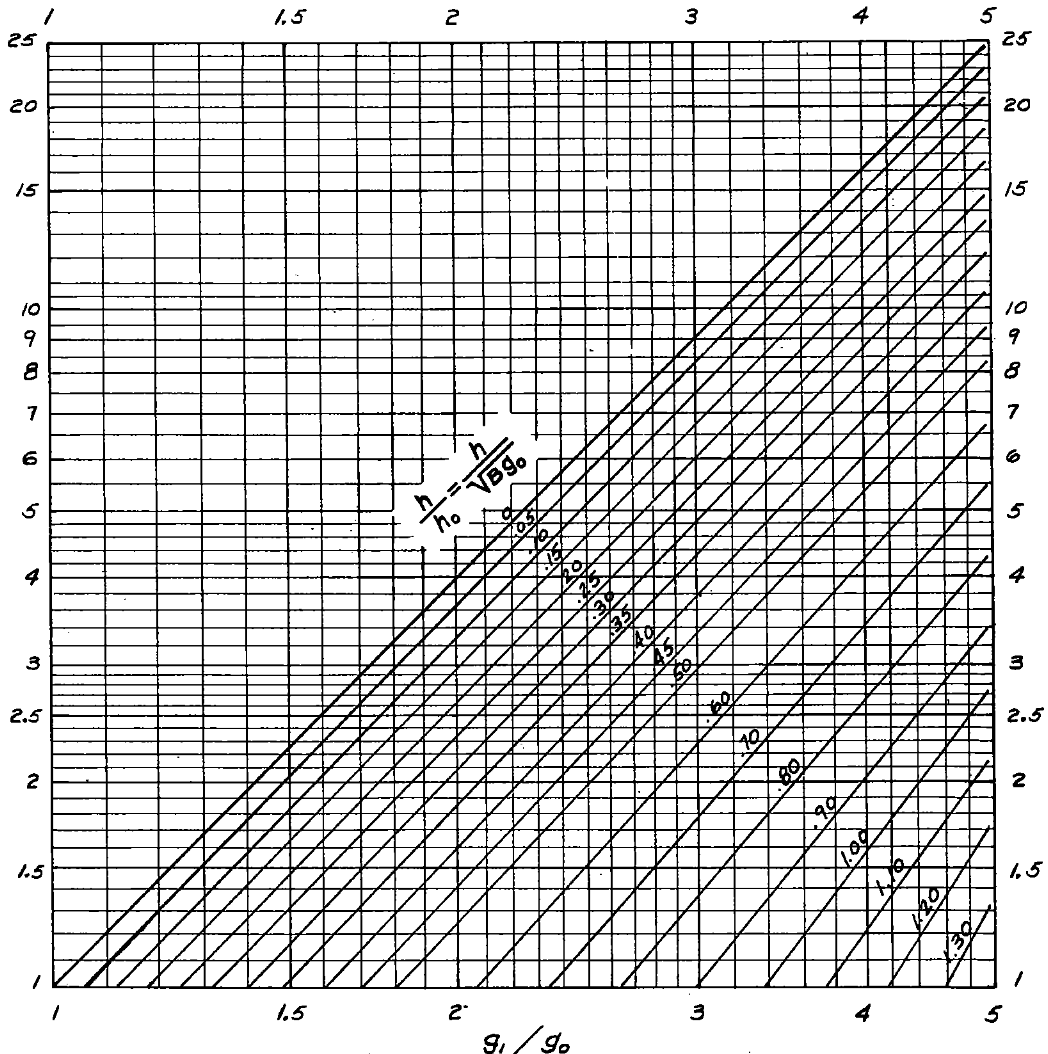
VALUES OF F_L [UA-51.4]



VALUES OF V_L [UA-51.5]



VALUES OF f [UA-51.6]



WELDING NECK FLANGE DESIGN

SHEET A

1 DESIGN CONDITIONS		2 GASKET	FACE	3 FROM Fig. UA-49
Design Pressure, P				$N =$
Design Temperature				$b =$
Flange Material				$G =$
Bolting Material				$y =$
				$m =$

Corrosion Allowance		**4 LOAD AND BOLT CALCULATIONS**	
Allowable Stresses – Flange	Design Temp., S_{fo}	$W_{m2} = b\pi Gy =$	$A_m =$ greater of W_{m2}/S_a or $W_{m1}/S_b =$
Allowable Stresses – Flange	Atm. Temp., S_{fa}	$H_p = 2b\pi GmP =$	$A_b =$
Allowable Stresses – Bolting	Design Temp., S_b	$H = G^2\pi P/4 =$	$W = .5(A_m + A_b)S_a =$
Allowable Stresses – Bolting	Atm. Temp., S_a	$W_{m1} = H_p + H =$	

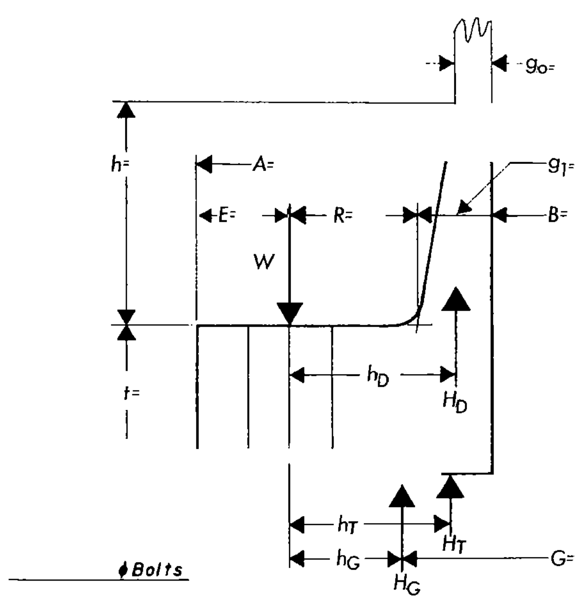
CONDITION	LOAD	X LEVER ARM	= MOMENT
5 *Operating*	$H_D = \pi B^2P/4 =$	$h_D = R + .5g_1 =$	$M_D = H_Dh_D =$
	$H_G = W_{m1} - H =$	$h_G = .5(C - G) =$	$M_G = H_Gh_G =$
	$H_T = H - H_D =$	$h_T = .5(R + g_1 + h_G) =$	$M_T = H_Th_T =$
			$M_o =$
Seating	$H_G = W =$	$h_G = .5(C - G) =$	$M_G =$

8 Allowable Stress	STRESS CALCULATION—*Operating*
$1.5\ S_{fo}$	Long. Hub, $S_H = fm_o/\lambda g_1^2 =$
S_{fo}	Radial Flg., $S_R = \beta m_o/\lambda t^2 =$
S_{fo}	Tang. Flg., $S_T = m_oY/t^2 - ZS_R =$
S_{fo}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T) =$
9 Allowable Stress	**STRESS CALCULATION— seating**
$1.5\ S_{fa}$	Long. Hub, $S_H = fm_G/\lambda g_1^2 =$
S_{fa}	Radial Flg., $S_R = \beta m_G/\lambda t^2 =$
S_{fa}	Tang. Flg., $S_T = m_G Y/t^2 - ZS_R =$
S_{fa}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T) =$

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

7 STRESS FORMULA FACTORS
$t =$
$\alpha = te + 1 =$
$\beta = 4/3\ te + 1 =$
$\gamma = \alpha/T =$
$\delta = t^3/d =$
$\lambda = \gamma + \delta =$
$m_O = M_O/B =$ operating $=$
$m_G = M_G/B =$ seating $=$

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



G+W Taylor-Bonney Division

TAYLOR FORGE

Computed________ Date________

Checked________ Number ________

SLIP ON or LAP JOINT FLANGE DESIGN

SHEET B

1 DESIGN CONDITIONS				2 GASKET	FACE	3 FROM Fig. UA-49
Design Pressure, P						$N =$
Design Temperature						$b =$
Flange Material						$G =$
Bolting Material						$y =$
Corrosion Allowance						$m =$

Allowable Stresses				4 LOAD AND BOLT CALCULATIONS	
Flange	Design Temp., S_{fo}			$W_{m2} = b\pi Gy =$	$A_m =$ greater of W_{m2}/S_a or $W_{m1}/S_b =$
	Atm. Temp., S_{fa}			$H_p = 2b\pi GmP =$	$A_b =$
Bolting	Design Temp., S_b			$H \doteq G^2\pi P/4 =$	$W = .5(A_m + A_b)S_a =$
	Atm. Temp., S_a			$W_{m1} = H_p + H =$	

CONDITION	LOAD ×	LEVER ARM =	MOMENT
5 *Operating*	$H_D = \pi B^2P/4 =$	$h_D = R + g_1 =$	$M_D = H_Dh_D =$
	$H_G = W_{m1} - H =$	$h_G = .5(C - G) =$	$M_G = H_Gh_G =$
	$H_T = H - H_D =$	$h_T = .5(R + g_1 + h_G) =$	$M_T = H_Th_T =$
			$M_o =$
Seating	$H_G = W =$	$h_G = .5(C - G) =$	$M_o =$

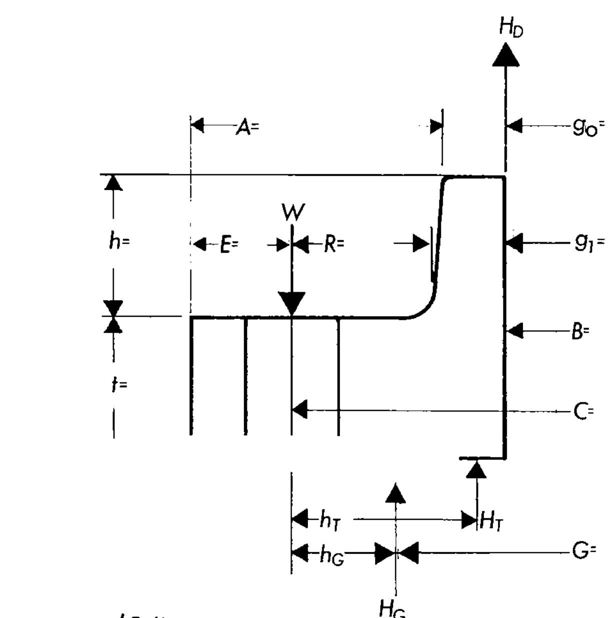
8 Allowable Stress	STRESS CALCULATION—*Operating*	
$1.5\ S_{fo}$	Long. Hub, $S_H = m_o/\lambda g_1^2$	
S_{fo}	Radial Flg., $S_R = \beta m_o/\lambda t^2$	
S_{fo}	Tang. Flg., $S_T = m_o Y/t^2 - ZS_R$	
S_{fo}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	

9 Allowable Stress	STRESS CALCULATION— Seating	
$1.5\ S_{fa}$	Long. Hub, $S_H = m_G/\lambda g_1^2$	
S_{fa}	Radial Flg., $S_R = \beta m_G/\lambda t^2$	
S_{fa}	Tang. Flg., $S_T = m_o Y/t^2 - ZS_R$	
S_{fa}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	

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

7 STRESS FORMULA FACTORS	
t	
$\alpha = te + 1$	
$\beta = 4/3\ te + 1$	
$\gamma = \alpha/T$	
$\delta = t^3/d$	
$\lambda = \gamma + \delta$	
$m_O = M_O/B =$ OPERATING $=$	
$m_G = M_G/B =$ SEATING $=$	

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



ϕ Bolts

G+W Taylor-Bonney Division

TAYLOR FORGE

Computed ________ Date ________

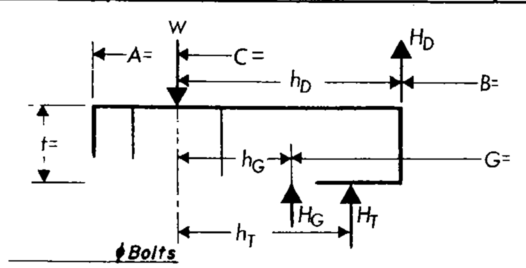
Checked ________ Number ________

RING FLANGE DESIGN — SHEET C

1 DESIGN CONDITIONS				2 GASKET	FACE	3 FROM Fig. UA-49
Design Pressure, P						$N =$
Design Temperature						$b =$
Flange Material						$G =$
Bolting Material						$y =$
Corrosion Allowance						$m =$

Allowable Stresses				**4 LOAD AND BOLT CALCULATIONS**	
Flange	Design Temp., S_{fo}			$W_{m2} = b\pi Gy =$	$A_m =$ greater of W_{m2}/S_a or $W_{m1}/S_b =$
	Atm. Temp., S_{fa}			$H_p = 2b\pi GmP =$	$A_b =$
Bolting	Design Temp., S_b			$H = G^2\pi P/4 =$	$W = .5(A_m + A_b)S_a =$
	Atm. Temp., S_a			$W_{m1} = H_p + H =$	

CONDITION	LOAD	X LEVER ARM	= MOMENT
5 *Operating*	$H_D = \pi B^2P/4 =$	$h_D = .5(C - B) =$	$M_D = H_D h_D =$
	$H_G = W_{m1} - H =$	$h_G = .5(C - G) =$	$M_G = H_G h_G =$
	$H_T = H - H_D =$	$h_T = .5(h_D + h_G) =$	$M_T = H_T h_T =$
			$M_o =$
Seating	$H_G = W =$	$h_G = .5(C - G) =$	$M_G =$



6 SHAPE CONSTANTS $K = A/B$ $Y =$

If bolt spacing exceeds $2a + t$, multiply M_o and M_G in t equations by: $\sqrt{\dfrac{\text{Bolt spacing}}{2a + t}}$

7		
$t =$ GREATER OF	OPERATING	$t = \sqrt{\dfrac{M_o Y}{Sf_o B}}$
	SEATING	$t = \sqrt{\dfrac{M_G Y}{Sf_a B}}$

G+W Taylor-Bonney Division

TAYLOR FORGE

Computed________ Date________

Checked________ Number________

BLIND FLANGE DESIGN — SHEET D

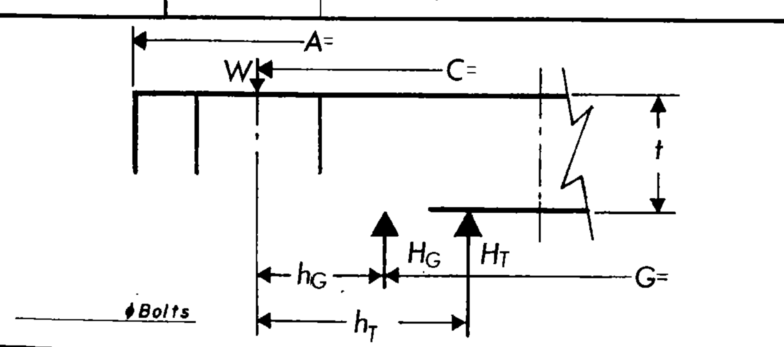
1 DESIGN CONDITIONS				2 GASKET	FACE	3 FROM Fig. UA-49
Design Pressure, P						$N =$
Design Temperature						$b =$
Flange Material						$G =$
Bolting Material						$y =$
Corrosion Allowance						$m =$

Allowable Stresses				**4 LOAD AND BOLT CALCULATIONS**	
Flange	Design Temp., S_{fo}			$W_{m2} = b\pi Gy =$	$A_m =$ greater of W_{m2}/S_a or $W_{m1}/S_b =$
	Atm. Temp., S_{fa}			$H_p = 2b\pi GmP =$	$A_b =$
Bolting	Design Temp., S_b			$H = G^2\pi P/4 =$	$W = .5(A_m + A_b)S_a =$
	Atm. Temp., S_a			$W_{m1} = H_p + H =$	$h_G = .5(C - G) =$

5	Thickness Calculations	
Thickness = Greater of	*Operating*	$t = G\sqrt{\dfrac{0.3\,P}{Sf_o} + \dfrac{1.9\,W_{m1}\,h_G}{Sf_o\,G^3}} =$
	Seating	$t = G\sqrt{\dfrac{1.9\,Wh_G}{Sf_A\,G^3}} =$

NOTES

1. OD, Facing, drilling. Gasket—same as matching flange—if any.
2. No additional thickness is required for facing special design B.F.'s.
3. Unless otherwise specified, faces of blind flanges may be machined only over area required for gasketing.



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TAYLOR FORGE

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REVERSE WELDING NECK FLANGE DESIGN

SHEET E

1 DESIGN CONDITIONS	
Design Pressure, P	
Design Temperature	
Flange Material	
Bolting Material	
Corrosion Allowance	

Allowable Stresses			
Flange	Design Temp., S_{fo}		
	Atm. Temp., S_{fa}		
Bolting	Design Temp., S_b		
	Atm. Temp., S_a		

2 GASKET	FACE	FROM Fig. UA-49
		$N =$
		$b =$
		$G =$
		$y =$
		$m =$

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

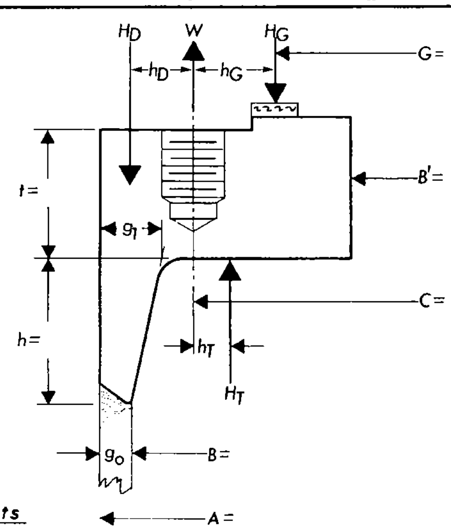
CONDITION	LOAD	X LEVER ARM	= MOMENT
5 *Operating*	$H_D = \pi B^2 P/4 =$	$h_D = .5(C + g_1 - 2g_o - B) =$	$M_D = H_D h_D =$
	$H_G = W_{m1} - H =$	$h_G = .5(C - G) =$	$M_G = H_G h_G =$
	$H_T = H - H_D =$	$h_T = .5(C - \frac{B+G}{2}) =$	$M_T = H_T h_T =$
	ADD MOMENTS ALGEBRAICALLY — THEN USE THE ABSOLUTE VALUE $\lvert M_o \rvert$ IN ALL SUBSEQUENT CALCULATIONS		$\lvert M_o \rvert =$
Seating	$H_G = W =$	$h_G = .5(C - G) =$	$M_G =$

8 Allowable Stress	STRESS CALCULATION—*Operating*
1.5 S_{fo}	Long. Hub, $S_H = fm_o/\lambda g_1^2$
S_{fo}	Radial Flg., $S_R = \beta m_o/\lambda t^2$
S_{fo}	Tang. Flg., $S_T = m_o Y_R/t^2 - ZS_R(0.67te+1)/\beta =$
S_{fo}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T) =$
S_{fo}	Tang. Flg, $S_T (A_T B') = \frac{m_o}{t^2}\left[Y - \frac{2K^2(1 + {}^2/_3 te)}{(K^2-1)\lambda}\right] =$

9 Allowable Stress	STRESS CALCULATION— Seating
1.5 S_{fa}	Long. Hub, $S_H = fm_G/\lambda g_1^2 =$
S_{fa}	Radial Flg., $S_R = \beta m_G/\lambda t^2 =$
S_{fa}	Tang. Flg., $S_T = m_G Y_R/t^2 - ZS_R(0.67te+1)/\beta =$
S_{fa}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T) =$
S_{fa}	Tang. Flg., $S_T (AT B') = \frac{m_G}{t^2}\left[Y - \frac{2K^2(1 + {}^2/_3 te)}{(K^2-1)\lambda}\right] =$

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

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



ϕ Bolts

G+W Taylor-Bonney Division BONNEY FORGE ™

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SLIP ON FLANGE DESIGN—FLAT FACED

SHEET F

1 DESIGN CONDITIONS			
Design Pressure, P			
Design Temperature			
Flange Material			
Bolting Material			
Corrosion Allowance			
Allowable Stresses	Flange	Design Temp., S_{fo}	
		Atm. Temp., S_{fa}	
	Bolting	Design Temp., S_b	
		Atm. Temp., S_a	

2 GASKET

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 $W_{m1}/S_b =$
$H_p = 2b\pi GmP =$	$A_b =$
$H'_p = {}^{h_G}/{}_{h'_G} H_p =$	$W = .5\,(A_m + A_b)\,S_a =$
$H = G^2\pi P/4 =$	$H'_{GY} = {}^{h_G}/{}_{h'_G}\; b\pi GY =$
$W_{m1} = H + H_p + H'P =$	

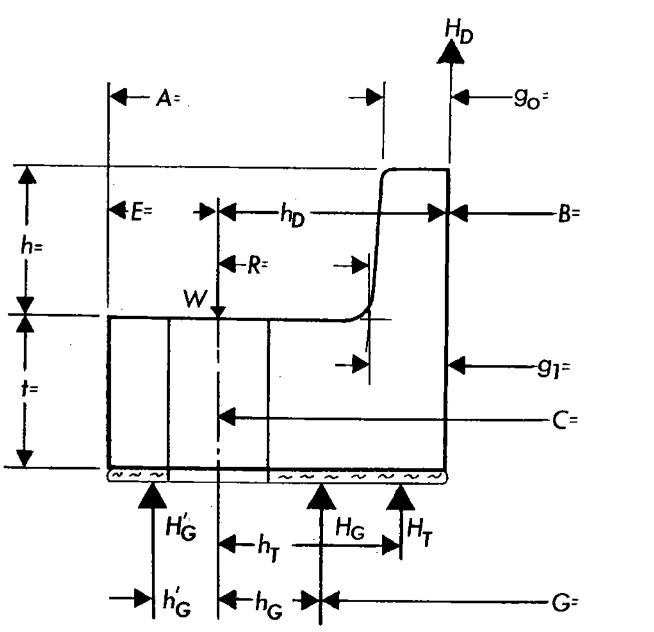
CONDITION	LOAD	X	LEVER ARM	=	MOMENT
5 *Operating*	$H_D = \pi B^2P/4 =$		$h_D = R + g_1 =$		$M_D = H_D h_D =$
	$H_T = H - H_D =$		$h_T = .5(R + g_1 + h_G) =$		$M_T = H_T h_T =$
					$M_o =$
LEVER ARMS	$h_G = \dfrac{(C - B)(2B + C)}{6\,(B + C)} =$		$h'_G = \dfrac{(A - C)\,(2A + C)}{6\,(C + A)} =$		
REVERSE MOMENT	$H_G = W - H =$		$h''_G = \dfrac{h_G\, h'_G}{h_G + h'_G} =$		$M_G = H_G h_G'' =$

8 Allowable Stress	STRESS CALCULATION—*Operating*
$1.5\ S_{fo}$	Long. Hub, $S_H = m_o/\lambda g_1^2 =$
S_{fo}	Radial Flg., $S_R = \beta m_o/\lambda t^2 =$
S_{fo}	Tang. Flg., $S_T = m_o Y/t^2 - ZS_R =$
S_{fo}	greater of $.5(S_H + S_R)$ or $.5\,(S_H + S_T) =$
Sf_o	RADIAL STRESS AT BOLT CIRCLE $S_{RAD} = \dfrac{6\,M_G}{t^2\,(\pi C - nd_1)} =$

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	=	$d = \dfrac{U}{V_L}\, h_o g_o^2 =$	
$h_o = \sqrt{Bg_o}$	=		

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

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



______ ϕ Bolts

G+W Taylor-Bonney Division TAYLOR FORGE

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Part A is a type of flange having the gasket wholly within the circle of the bolt holes and no contact beyond that circle

Part B is a type of flange with metal-to-metal contact outside the bolt circle, identical pairs only

Code Case 1828 Flanges: Type with metal-to-metal contact outside the bolt circle having identical or non-identical pairs, & designed by the simplified rules of cc 1828.

A = outside diameter of flange. For slotted flanges the diameter of the circle tangent to the inner edge of the slots, in.

A_b = actual total cross-sectional area of bolts at root of thread or section of least diameter under stress, in.^{2}

A_m = total required bolt cross-section area, in.^{2}

a = nominal bolt diameter, in.

α = alpha — stress formula factor, $te + 1$

α_R = alpha $_R$ — stress formula factor,
$= \frac{1}{K^2}\left[1 + \frac{3(K+1)(1-\nu)}{\pi Y}\right]$
for reverse flanges

B = inside diameter of flange, in. When B is less than 20 g_1, it is optional for the designer to substitute B_1 for B in the Code formula for longitudinal hub stress S_H

B_1 = $B + g_1$ for loose hubbed flanges and also for integral flanges when $f<1$
$B + g_0$ for integral flanges when $f \geq 1$

b = effective gasket or joint-contact-surface seating width, in.

$2b$ = effective gasket or joint-contact-surface pressure width, in.

b_o = basic gasket seating width, in.

β = Beta — stress formula factor $= 4/3\ te + 1$

C = Bolt-circle diameter, in.

d = hub shape factor, for integral flanges $d = \frac{U\,h_o\,g_o^2}{V}$, for loose flanges $d = \frac{U\,h_o g_o^2}{V_L}$, for reverse flanges $d = \frac{U_R\,h_o\,g_o^2}{V}$

d_1 = Bolt hole dia.

δ = delta — stress formula factor $= t^3/d$

E = radial distance from C to A, in.

e = hub shape factor; for integral flanges $e = F/h_0$, for loose flanges $e = F_L/h_0$

F = hub shape factor for integral flanges

F_L = hub shape factor for loose flanges

f = hub stress-correction factor for integral flanges, when the chart reading gives values < 1 use 1

G = diameter at location of gasket load reaction, in. Except as noted in sketch e of Figure 1, G is defined as follows:

When $b_0 \leq$ ¼ in., G = mean diameter of gasket contact face
When $b_0 >$ ¼ in., G = outside diameter of gasket contact face less $2b$

g_0 = thickness of hub at small end, in.

g_1 = thickness of hub at back of flange, in.

γ = gamma—stress formula factor $= \alpha/T$, α/T_R for reverse flanges.

H = total hydrostatic end force $= \pi/4\ G^2P$, lbs.

H_D = hydrostatic end force on area inside of flange $= \pi/4\ B^2P$, lbs.

H_G = gasket load, operating $= H_p = W_{m1} - H$, lbs.

H_G = gasket load, seating $= W$, lbs.

H_p = total joint-contact-surface compression load $= 2b\pi\ GmP = W_{m1} - H$, lbs.

H_T = difference between total hydrostatic end force and the hydrostatic end for an area inside of flange $= H - H_D$, lbs.

h = hub length, in.

h_D = radial distance from C to the circle on which H_D acts, in.

h_G = radial distance from G to C = $(C - G)/2$, in.

h_o = factor $= \sqrt{Bg_0}$, in.

h_T = radial distance from C to the circle on which H_T acts, in.

K = A/B, A/B′ for reverse flanges

λ = lambda — stress formula factor $= \gamma + \delta$

m_0 = M_0/B — unit load, operating, M_0/B' for reverse flanges, lbs.

m_G = M_G/B — unit load, seating, M_G/B' for reverse flanges, lbs.

M_D = component of moment due to $H_D = H_D h_D$, inch-pounds

M_G = total moment acting on the flange, seating $= W h_G$, inch-pounds

M_G = component of moment due to H_G, operating $= H_G h_G$, inch-pounds

M_0 = total moment acting upon the flange, for operating conditions, inch-pounds.

M_T = component of moment due to $H_T = H_T h_T$

m = gasket factor

N = width, used to determine the basic gasket seating width b_0, based upon the possible contact width of the gasket, in.

n = number of bolts

ν = Nu — Poisson's ratio = 0.3 for steel

P = design pressure, psi

R = radial distance from bolt circle to point of intersection of hub and back of flange for integral and hubbed flanges, in.

= radial distance from bolt circle to bore, ring flange, in.

Sa = allowable bolt stress at atmospheric temperature, psi

Sb = allowable bolt stress at design temp., psi

S_{fa} = allowable design stress for material of flange, nozzle neck, vessel or pipe wall, at atmospheric temperature, psi

S_H = calculated longitudinal stress in hub, psi

S_R = calculated radial stress in flange, psi

S_T = calculated tangential stress in flange, psi

T = K-factor

T_R = K-factor for reverse flanges

t = flange thickness, in.

t_n = pipe wall thickness, in.

U = K-factor

U_R = K-factor for reverse flanges

V = hub shape factor for integral flanges

V_L = hub shape factor for loose flanges

w = width used to determine the basic gasket seating width b_0, based upon the contact width between the flange facing and the gasket (see Table UA-49.2), in.

W = flange design bolt load for operating or seating conditions, as may apply, lbs.

Wm_1 = required bolt load, operating conditions, lbs.

Wm_2 = min. req'd. bolt load for gasket seating, lbs.

Y = K-factor

Y_R = K-factor for reverse flanges

y = gasket or joint-contact-surface unit seating load, psi

Z = K-factor

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4. Waters, E. O., Rossheim, D. B., Wesstrom, D. B., and Williams, F. S. G., *Development of General Formulas for Bolted Flanges,* Taylor Forge and Pipe Works, Chicago, Illinois, 1949.
5. Taylor Forge Catalog 571.
6. Schneider, R. W., "Flat Face Flanges with Metal-to-Metal Contact Beyond the Bolt Circle," Journal of Engineering for Power, Trans. ASME, Series A, Vol. 90, No. 1, Jan. 1968, pp. 82-88.
7. Waters, E. O. and Schneider, R. W., "Axisymmetric, Non-identical, Flat Face Flanges with Metal-to-Metal Contact Beyond the Bolt Circle," Journal of Engineering for Industry, Trans. ASME, Series B, Vol. 91, No. 3, Aug. 1969, pp. 615-622.
8. Waters, E. O., "Derivation of Code Formulas for Part B Flanges," WRC Bulletin No. 166, Oct. 1971.
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10. Schneider, R. W. and Waters, E. O., "The Background of ASME Code Case 1828: A Simplified Method of Analyzing Part B Flanges," Trans, ASME, Journal of Pressure Vessel Technology, Vol. 100, No. 2, May 1978, pp. 215-219.
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12. *A Simplified Method for Analyzing Flat Face Flanges with Metal-to-Metal Contact Outside the Bolt Circle,* ASME Code Case 1828.
13. *Steel Pipe Flanges and Flanged Fittings,* ANSI Standard B16.5.
14. *Large Diameter Carbon Steel Flanges,* API Standard 605.
15. *Pipe Line Flanges,* MSS-SP44.
16. *Cast Iron Flanges and Flanged Fittings,* ANSI Standard B16.1.
17. Kent, G. R. "Selecting Gaskets for Flanged Joints," Chemical Engineering, May 27, 1978.
18. Raut, H. D., and Leon, G. F. "Report of Gasket Factor Tests," WRC Bulletin 233, December 1977.
19. *Specification for Wellhead Equipment 6A,* American Petroleum Institute.
20. Rossheim, D. B. and Markl, A. R. C. "Gasket-Loading Constants," Mechanical Engineering, September, 1943, pp. 647-648.
21. Roberts, I. "Gaskets and Bolted Joints," Journal of Applied Mechanics, Trans. ASME Vol. 17, June, 1950, pp. 169-179.

Computer Program Listing That Calculates *F*, *V*, *f*, F_L, V_L

This program was written by E. C. Rodabaugh of Battelle Memorial Institute and his permission to reprint is gratefully acknowledged. The program is based on formulas developed in reference (4).

Minor editorial changes have been made to publish the listing in this manual. The program was adapted to run on a 370/145 under OS.

```
C CALCULATE FLANGE FACTORS, F, V, SMALL-F, F-SUB-L, AND V-SUB-L
      DET(D11,D12,D13,D21,D22,D23,D31,D32,D33) =
     1 D11*D22*D33 + D12*D23*D31 + D13*D32*D21
     2 -(D31*D22*D13 + D32*D23*D11 + D33*D12*D21)
   50 READ (5,1,END=100)    G1GO, HHO
    1 FORMAT (2E10.5)
C   G1GO= G1/GO, HHO = H/SQRT(B*GO)
   10 CONTINUE
      A = G1GO -1.
      B = 43.68*HHO**4
C   A = ALPHA,  B = KAPPA
      PRINT 2, G1GO, HHO
    2 FORMAT (9H   G1/GO=, 1P1E10.5,8H   H/HO=, 1E10.5//)
      A2 = A*A
      A3 = A**3
      D11=1./3.+ A/12.
      C11=D11
      D12=5./42. + 17.*A/336.
      C12=D12
      D13=1./210. + A/360.
      C13=D13
      D14=11./360. + 59.*A/5040. + (1.+3.*A)/B
      C14=D14
      D15=1./90. + 5.*A/1008. - (1.+A)**3/B
      C15=D15
      D16=1./120. + 17.*A/5040. + 1./B
      C16=D16
      D21=C12
      C21=D21
      D22=215./2772. + 51.*A/1232. + (60./7. + 225.*A/14.
     1 + 75.*A2/7. + 5.*A3/2.) / B
      C22=D22
      D23=31./6930. + 128.*A/45045. + (6./7. + 15.*A/7. + 12.*A2/7.
     1 + 5.*A3/11.) / B
      C23=D23
      D24=533./30240. + 653.*A/73920. + (1./2. + 33.*A/14.
     1 + 39.*A2/28. + 25.*A3/84.) / B
      C24=D24
      D25=29./3780. + 3.*A/704. - (1./2. + 33.*A/14. + 81.*A2/28.
     1 + 13.*A3/12.) / B
      C25=D25
```

```
      D26=31./6048. + 1763.*A/665280. + (1./2. + 6.*A/7.
     1 + 15.*A2/28. + 5.*A3/42.) / B
      C26=D26
      D31=C13
      C31=D31
      D32=C23
      C32=D32
      C32=D32
      D33=1./2925. + 71.*A/300300. + (8./35. + 18.*A/35.
     1 + 156.*A2 / 385. + 6.*A3/55.) / B
      C33=D33
      D34=761./831600. + 937.*A/1663200. + (1./35. + 6.*A/35.
     1 + 11.*A2/70. + 3.*A3/70.) / B
      C34=D34
      D35=197./415800. + 103.*A/332640. - (1./35. +6.*A/35.
     1 + 17.*A2/70. + A3/10.) / B
      C35=D35
      D36=233./831600. + 97.*A/554400. + (1./35. + 3.*A/35.
     1 + A2/14. + 2.*A3/105.) / B
      C36=D36
      DET1 = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D11=C14
      D21=C24
      D31=C34
      DA1AA0 = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D11=C15
      D21=C25
      D31=C35
      DA1AA1  = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D11=C16
      D21=C26
      D31=C36
      DA1BB0 = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D11=C11
      D21=C21
      D31=C31
      D12=C14
      D22=C24
      D32=C34
      DA2AA0  = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D12=C15
      D22=C25
      D32=C35
      DA2AA1  = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D12=C16
      D22=C26
      D32=C36
      DA2BB0  = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D12=C12
      D22=C22
      D32=C32
      D13=C14
      D23=C24
      D33=C34
      DA3AA0  = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D13=C15
```

```
      D23=C25
      D33=C35
      DA3AA1  = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D13=C16
      D23=C26
      D33=C36
      DA3BB0  = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      A1A0=DA1AA0/DET1
      A1A1=DA1AA1/DET1
      A1B0=DA1BB0/DET1
      A2A0=DA2AA0/DET1
      A2A1=DA2AA1/DET1
      A2B0=DA2BB0/DET1
      A3A0=DA3AA0/DET1
      A3A1=DA3AA1/DET1
      A3B0=DA3BB0/DET1
C ABOVE COMPLETES DETERMINATION OF A1,A2, AND A3 IN TERMS OF AA1,AA0,BB0
      C11=D11 -(B/4.)**.25
      D12=-5./12. - A1A0 + A2A0 - (B/4.)**.25*A1A0
      C12=D12
      D13=-1./12. - A1B0 + A2B0 - (B/4.)**.25*A1B0
      C13=D13
      D21= -(B/4.)**.5
      C21=D21
      D22=1./2.
      C22=D22
      D23= 0.
      C23=D23
      D31= -(B/4.)**.75
      C31=D31
      D32= 3.*A/2. + (B/4.)**.75*A1A0
      C32=D32
      D33=1./2. + (B/4.)**.75*A1B0
      C33=D33
      DET1    = DET(D11,C12,D13,D21,D22,D23,D31,D32,D33)
      A1 = 1./12. + A1A1 - A2A1 + (B/4.)**.25 * A1A1
      A2 = 0.
      A3 = - (B/4.)**.75 * A1A1
      D11 = A1
      D21 = A2
      D31 = A3
      DC5     = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D11 = C11
      D21 = C21
      D31 = C31
      D12 = A1
      D22 = A2
      D32 = A3
      DA0     = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
      D12 = C12
      D22 = C22
      D32 = C32
      D13 = A1
      D23 = A2
      D33 = A3
      DB0     = DET(D11,D12,D13,D21,D22,D23,D31,D32,D33)
```

```
      DC5 = DC5/DET1
      DAO = DAO/DET1
      DBO = DBO/DET1
      PRINT 5, DC5, DAO, DBO
    5 FORMAT (1P3E15.5//)
      A1 = A1AO*DAO + A1A1 + A1BO*DBO
      A2 = A2AO*DAO + A2A1 + A2BO*DBO
      A3 = A3AO*DAO + A3A1 + A3BO*DBO
      THETA = -A1 -3.*A2/2. - A3/5. + DAO/4. + 1./4. + DBO/12.
      P1 = (1./2.+A/6.)*A1 + (1./4.+11.*A/84.)*A2 + (1./70.+A/105.)*A3
     1 -(7./120.+A/36.+3.*A/B)*DAO - (1./40. + A/72.) - (1./60. +
     2 A/120. + 1./B)*DBO
      F = -P1/((B/2.73)**.25 * (1.+A)**3 / B)
      V = THETA / ((2.73/B)**.25 * (1.+A)**3)
      FS = DAO / (1.+A)
C  LOOSE HUB FLANGE, AAO = BBO = 0
      DBO=0.
      DAO=DBO
      A1 = A1AO*DAO + A1A1 + A1BO*DBO
      A2 = A2AO*DAO + A2A1 + A2BO*DBO
      A3 = A3AO*DAO + A3A1 + A3BO*DBO
      THETA = -A1 -3.*A2/2. - A3/5. + DAO/4. + 1./4. + DBO/12.
      P1 = (1./2.+A/6.)*A1 + (1./4.+11.*A/84.)*A2 + (1./70.+A/105.)*A3
     1 -(7./120.+A/36.+3.*A/B)*DAO - (1./40. + A/72.) - (1./60. +
     2 A/120. + 1./B)*DBO
      FL = -P1/((B/2.73)**.25 * (1.+A)**3 / B)
      VL = THETA / ((2.73/B)**.25* (1.+A)**3)
      PRINT 3, F, V, FS, FL, VL
    3 FORMAT (5H    F=, 1P1E10.5,5H    V=, 1E10.5,
     1 11H   SMALL-F=, 1E10.5, /,11H   F-SUB-L=, 1E10.5,
     2 11H   V-SUB-L=, 1E10.5,///)
      GO TO 50
  100 CALL EXIT
      END
```

BOLT TORQUE CHART

	Nominal Dia. in.	Threads Per in.	Root Dia. in.	Root Area in.^{2}	Torque in Foot-Pounds for Various Bolt Stresses: 7500 psi	15000 psi	30000 psi
Machine and Cold-rolled Steel	1/4	20	0.185	0.027	1	2	4
	5/16	18	0.240	0.045	2	4	8
	3/8	16	0.294	0.068	3	6	12
	7/16	14	0.345	0.093	5	10	20
	1/2	13	0.400	0.126	8	15	30
	9/16	12	0.454	0.162	12	23	45
	5/8	11	0.507	0.202	15	30	60
	3/4	10	0.620	0.302	25	50	100
	7/8	9	0.731	0.419	40	80	160
	1	8	0.838	0.551	62	123	245
	1 1/8	7	0.939	0.693	98	195	390
	1 1/4	7	1.064	0.890	137	273	545
	1 3/8	6	1.158	1.054	183	365	730
	1 1/2	6	1.283	1.294	219	437	875
	1 5/8	5 1/2	1.389	1.515	300	600	1200
	1 3/4	5	1.490	1.744	390	775	1550
	1 7/8	5	1.615	2.049	525	1050	2100
	2	4 1/2	1.711	2.300	563	1125	2250
					30000 psi	45000 psi	60000 psi
Alloy Steel	1/4	20	0.185	0.027	4	6	8
	5/16	18	0.240	0.045	8	12	16
	3/8	16	0.294	0.068	12	18	24
	7/16	14	0.345	0.093	20	30	40
	1/2	13	0.400	0.126	30	45	60
	9/16	12	0.454	0.162	45	68	90
	5/8	11	0.507	0.202	60	90	120
	3/4	10	0.620	0.302	100	150	200
	7/8	9	0.731	0.419	160	240	320
	1	8	0.838	0.551	245	368	490
	1 1/8	8	0.963	0.728	355	533	710
	1 1/4	8	1.088	0.929	500	750	1000
	1 3/8	8	1.213	1.155	680	1020	1360
	1 1/2	8	1.338	1.405	800	1200	1600
	1 5/8	8	1.463	1.680	1100	1650	2200
	1 3/4	8	1.588	1.980	1500	2250	3000
	1 7/8	8	1.713	2.304	2000	3000	4000
	2	8	1.838	2.652	2200	3300	4400
	2 1/4	8	2.088	3.423	3180	4700	6360
	2 1/2	8	2.338	4.202	4400	6600	8800
	2 3/4	8	2.588	5.259	5920	8880	11840
	3	8	2.838	6.324	7720	11580	15440

The torque values above are *suggested only* due to the many variables that affect actual results. These loads are based on the use of heavy oil-graphite lubricant and the thread forms shown. Efficiency of non-lubricated bolts may drop 50%. More accurate bolt tension is calculated as follows: Divide total elongation by the bolt length, and multiply the result by E, the modulus of elasticity. Bolt length is taken mid-nut to mid-nut, or an equivalent distance. Careful micrometer measurements are required.

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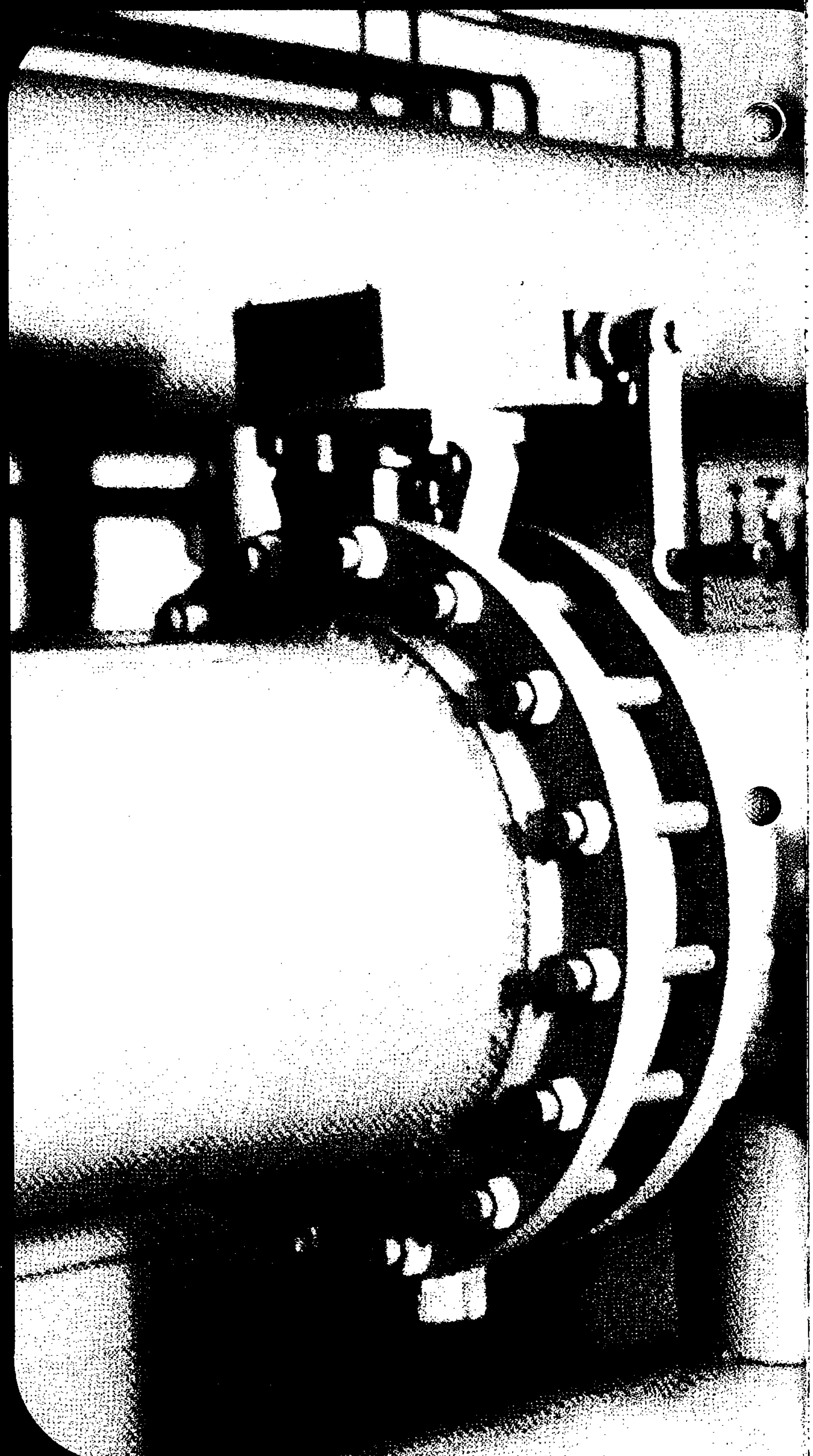

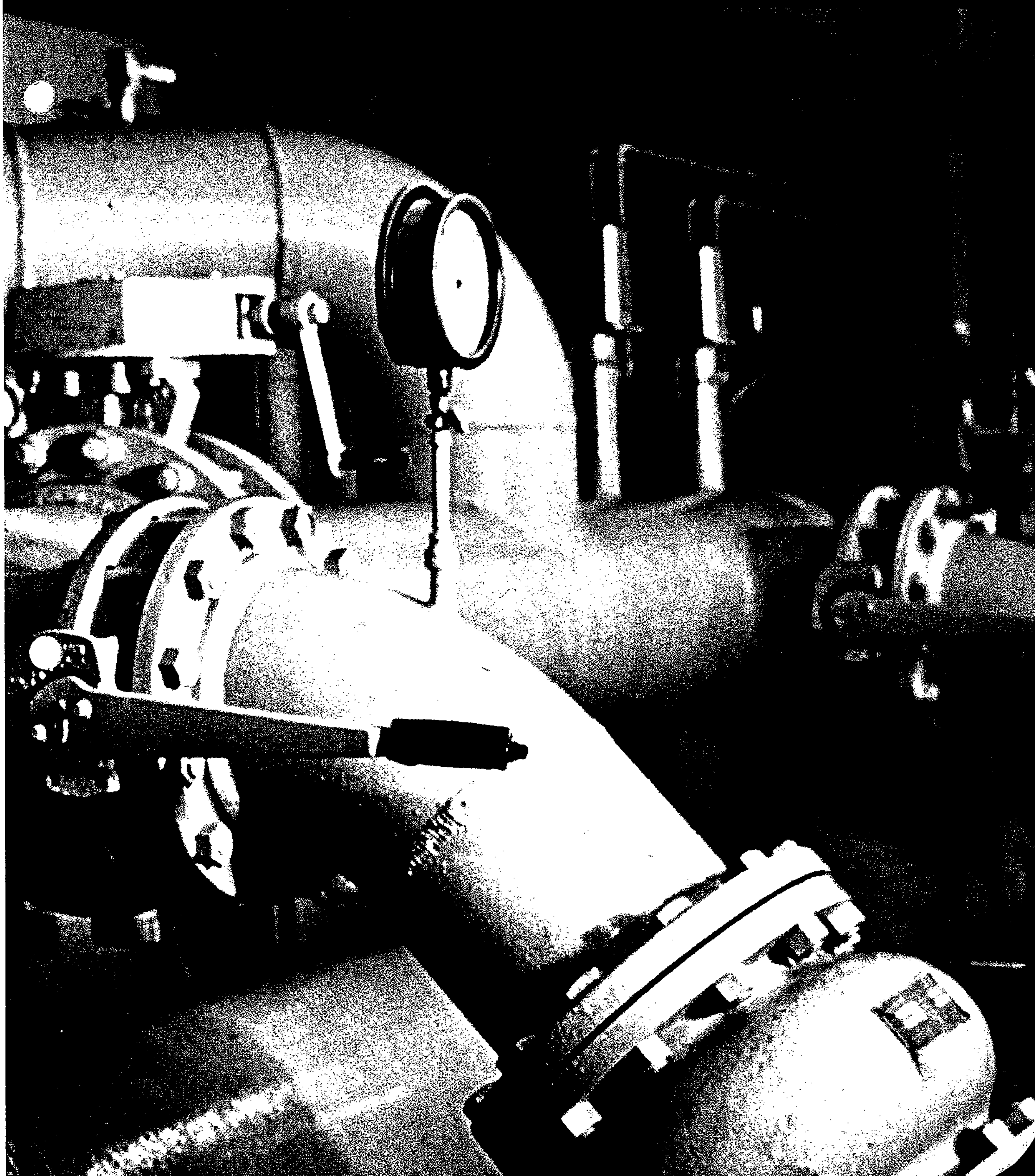

MODERN FLANGE DESIGN
Bulletin 502

TAYLOR
FORGE
TM

G+W Taylor-Bonney Division
P.O. Box 999
Southfield, Michigan 48037

Bulletin 502

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