Consumers Energy

A CMS Energy Company

Palisades Nuclear Plant
27780 Blue Star Memorial Highway
Covert, MI 49043

Tel: 616 764 2276
Fax: 616 764 3265

Nathan L. Haskell
Director, Licensing and
Performance Assessment

May 1, 2000

U.S. Nuclear Regulatory Commission
ATTN: Document Control Desk
Washington, DC 20555-0001

DOCKET 50-255 - LICENSE DPR-20 - PALISADES PLANT
ANALYSIS OF BOLTED CONNECTION WITH BORIC ACID LEAK, SPENT FUEL POOL HEAT EXCHANGER E-53A AND E-53B - SUBMITTED IN ACCORDANCE WITH APPROVED ASME CODE RELIEF REQUEST PR-06

In accordance with the provisions of the American Society of Mechanical Engineers, Boiler and Pressure Vessel Code, Section XI (1989 Edition), IWB-3144(b), the attached "Analysis of Bolted Connection with Boric Acid Leak, Spent Fuel Pool Heat Exchanger E-53A and E-53B," is hereby submitted. Authorization to utilize this approach to bolted connection leakage was granted via Palisades Plant Relief Request PR-06, approved by NRC on August 26, 1999.

This analysis addresses noted leakage at the shell-to-head bolted connections on Spent Fuel Pool Heat Exchangers E-53A and E-53B. The analysis, performed in accordance with IWB-3142.4, concludes that sufficient bolt stress bearing area exists, and will exist, until repair scheduled to be completed prior to the end of the 2001 refueling outage.

SUMMARY OF COMMITMENTS

This letter contains no new commitments and no revisions to existing commitments.

Nathan L. Haskell
Director, Licensing and Performance Assessment

CC Administrator, Region III, USNRC
Project Manager, NRR, USNRC
NRC Resident Inspector - Palisades

Attachment

A047

ATTACHMENT

CONSUMERS ENERGY COMPANY
PALISADES PLANT
DOCKET 50-255

May 1, 2000

ANALYSIS OF BOLTED CONNECTION WITH BORIC ACID LEAK, SPENT FUEL POOL HEAT EXCHANGER E-53A AND E-53B

19 Pages

ENGINEERING ANALYSIS COVER SHEET

Title Analysis of Bolted Connection with Boric Acid Leak, Spent Fuel Pool Heat Exchanger E-53A and E-53B

INITIATION AND REVIEW

Calculation Status	Preliminary ☐	Pending ☐	Final ☒	Superseded ☐

Rev #	Description of Revision	Initiated By	Initiated Date	Init Appd By	Review Method: Alt Calc	Review Method: Detail Rev'w	Review Method: Qual Test	Technically Reviewed By	Technically Reviewed Date	Rev'r Appd By	S/DR Appd
0	Original Issue	GESchneider	12/20/99	[illegible]		✓		[illegible]	12/28/99	[illegible]	

	Analysis Detail	Reference/Comment
1	**1.0 OBJECTIVE:**	
2–7	This EA is written to evaluate leakage from E-53A and E-53B through the bolted connections at each shell to heat exchanger bolted head. It will document the acceptability of the connection in accordance with ASME Section XI, 1989 Edition and Engineering Manual EM-09-13, Relief Request Number 6 and recommend inspection and repair actions to assure continued compliance with regulatory requirements.	

Analysis Detail	Reference/Comment
2.0 ANALYSIS INPUT	
2.1 Vendor Drawing M-9, Sheet 1, Rev. 6 (M9-SH1-6), Fuel Pool Cooling Heat Exchanger E-53A.	
2.2 Vendor Drawing M-9, Sheet 2, Rev. 6 (M9-SH2-6), Fuel Pool Cooling Heat Exchanger E-53B.	
2.3 ASME Section XI, 1989 Edition	
2.4 ASME Section III, Appendix XI, Rules for Bolted Flange Connections	
2.5 Material Specs - Bolts and Nuts are SA-193 Grade B7 and SA-194, Grade 2H (available in DCC Library)	
2.6 ASTM 307, Table 2 (available in DCC Library)	
2.7 10CFR50.55a	
2.8 SER dated August 26, 1999, "Approval for Third 10-Year Interval Inservice Inspection Program Request for Relief for the Palisades Plant"	
2.9 Design Conditions: 1. Temperature Tube Side - 220°F 2. Pressure Tube Side - 125 psig 3. No. of bolts - 32 4. Diameter - 3/4" - 10 Stud Bolts	
2.10 Technical Specifications 3.8.5 - When spent fuel which has decayed less than one year is placed in the tilt pit storage racks, the bulk water temperature in the tilt pit storage area must be monitored continuously to assure that the water temperature does not exceed 150°F. Monitoring will continue for 24 hours after any addition of fuel to the main pool or the tilt pit or when a failure of the spent fuel pool cooling system occurs.	
2.11 Operating Requirements Manual - None	
2.12 FSAR Chapter 9.4, Spent Fuel Pool Cooling System and Chapter 9.11, Fuel Handling and Storage Systems	
2.13 Design Basis Document 2.07, Spent Fuel Pool Cooling System	
2.14 Engineering Analysis EA-C-PAL-98-1748-02, "Evaluation of Allowable Leakage Rate from the Spent Fuel Pool Cooling System"	
2.15 Specification Change SC-97-031, "Generic Stainless Steel Bolting Replacement"	
2.16 Engineering Assistance Request EAR-99-0211, "Use of SS Bolting for SFP Heat Exchangers"	
2.17 Industrial Process Engineers, Exchanger Specification Sheet (C/F 0429/1103)	
2.18 Industrial Process Engineers, Form U-1, Manufacturers' Data Report for Unfired Pressure Vessels (C/F 0429/104)	
2.19 Condition Report C-PAL-99-183, "SFP Heat Exchanger Labeling and P&ID M221, Sheet 2 are Incorrect"	
2.20 Administrative Procedure 3.02, "Action Item Report	
2.21 Administrative Procedure 3.03, "Corrective Action Process"	
2.22 Administrative Procedure 5.01, "Processing Work Requests/Work Orders"	
2.23 Administrative Procedure 9.28, "Engineering Assistance Request"	
2.24 Permanent Maintenance Procedure MSM-M-48, "Standard Torque Tables"	
2.25 NMAC, Boric Acid Corrosion Guidebook, EPRI TR-102748, dated April 1995	

Analysis Detail	Reference/Comment
3.0 ASSUMPTIONS	
3.1 Major Assumptions - None	
3.2 Minor Assumptions Bolt preload is based on requirements associated with ASME Section III, 1989 Edition, Appendix XI for the "Design Condition" and "Gasket Seating Condition." The calculations of ASME Boiler and Pressure Vessel Code, 1965 Edition would have been applicable during construction of the Spent Fuel Pool Heat Exchangers. Use of ASME Boiler and Pressure Vessel Code (B&PV), 1965 Edition, Article I-12, "Stress in Bolting" would result in a higher bolt preload value for the "Gasket Seating Condition" then would be calculated using the 1989 Edition. Use of the later Code is satisfactory since calculations are experienced based and use of the ASME Section III, 1989 edition incorporates experience gained since publication of ASME B&PV, 1965.	

Analysis Detail	Reference/Comment
ANALYSIS	
NOTE: Terms and symbols used in this analysis are defined in the ASME Section III attachments to this EA.	
Spent fuel pool components have a history of leakage. Examples include cooling pumps P-51A and P-51B, check valve CK-SFP930 and the fuel pool liner. In order to avoid adverse degradation at bolted connections, specification change SC-87-031 was approved and allows the replacement of carbon steel bolting with stainless steel. While this replacement has occurred for some components, it has not occurred for heat exchangers E-53A or E-53B.	
The heat exchangers are mounted in series and it is not possible to isolate one unit at a time for maintenance. However, it may be possible to replace bolting by removing one fastener at a time, performing cleaning and inspection and replacing, if required.	
In accordance with MSM-M-48, "Standard Torque Tables," ASTM A-193 Grade B7 bolts installed in HC class piping should be installed to 30 - 60 KSI fastener stress. Greater than 60 KSI may be used with Engineering concurrence.	
Using Appendix XI, equation 1 of ASME Code, Section III, the required bolt load for the "Design Condition" in pounds force is calculated to be 182,415 lbf. Using this value, the required total bolt area is calculated to be 6.081 square inches when bolts are stressed to 30 ksi in accordance with MSM-M-48. This area decreases to 3.04 square inches when bolts are stressed to 60 ksi as allowed by MSM-M-48. The equation and variables are shown below:	
$W_{m1} = H + H_P = 0.785G^2P + (2b \times 3.14GmP) = 182{,}415$ lbf	
Where variables obtained from Section XI and Vendor Drawing M-9, Sheet 1 and 2 are:	
G = 26 inches = diameter at location of gasket load reaction. P = 125 psig = design pressure b = 1.625 inches = effective gasket seating width. m = 3.50 = gasket factor (Table XI-3221.1-1)	

Analysis Detail	Reference/Comment
Using Appendix XI, equation 2 of ASME Code, Section III, the required bolt load for the "Gasket Seating Condition" in pounds force is calculated to be 862,323 lbf. Using this value, the required total bolt area is calculated to be 28.744 square inches when bolts are stressed to 30 ksi in accordance with MSM-M-48. This area decreases to 14.372 square inches when bolts are stressed to 60 ksi as allowed by MSM-M-48. The equation and variables are shown below:	
W_{m2} = 3.14bGy = 862, 323 lbf	
A = 862,323/ 30,000 = 28.744 square inches	
Where variables obtained from Section XI and Vendor Drawing M-9, Sheet 1 and 2 are:	
G = 26 inches = diameter at location of gasket load reaction. b = 1.625 inches = effective gasket seating width. y = 6500 lbf/sq.in. = gasket seating load (Table XI-3221.1-1).	
From ASTM A-307, Table 2, the stress area for 3/4" - 10 bolts is 0.334 square inches. Using this information, the total area of the 32 bolts is 10.688 square inches. This area is approximately 26% less than the area required to satisfy the "Gasket Seating Condition" when bolting is stressed to 60 ksi. In accordance with Appendix XI, page 383, note 1, this condition creates the possibility of leakage from the shell heads without affecting safety since the "Design Condition" is satisfied.	
Leakage is, most likely, due to bolt stress relaxation. A review of maintenance history could not find any evidence that these bolts have ever been replaced or retorqued. Therefore, it is assumed that these bolts and nuts are in excess of 30 years old, and there has been no documented effort to stop leakage.	Comment 1
The Spent Fuel Pool is filled with water borated to approximately 2500 ppm. This fluid has been leaking from the heat exchanger shell head flanges and contacting the bottom 5 to 7 bolts. Visual inspection performed during August 1999, discovered evidence of surface corrosion on the bottom 3 to 5 fastencers on each heat exchanger shell head flange. As distance from the bottom and along the bolt circle increased, evidence of corrosion decreased rapidly, and there was no substantial evidence of corrosion past the bottom 3 to 5 fasteners.	Comment 3

Analysis Detail	Reference/Comment
Fluid flow is to low for measurement, and contact with bolting materials is due to the growth of boric acid accumulations. Leakage is occurring from the Spent Fuel Pool System which is generally operated between 90 and 120°F. Room components, including the heat exchanger are also operated in this temperature range. According to the NMAC Manual, page 4-7, "If the leaking borated water is at room temperature, and it is leaking onto room temperature surfaces, the corrosion rates are relatively low. Reported corrosion rates for this situation range from 0.002 in/yr for 2500 ppm boron at 70°F to 0.007 in/yr for 2000 ppm boron at 104°F. Generally, higher temperatures tend to increase corrosion rates. However, in this case, temperatures are not sufficiently higher to cause a measurable increase in corrosion rates. Based on this analysis, it is concluded that sufficient bolt stress bearing area exists and will exist until scheduled repairs during the 2003 refueling outage. Degradation is estimated to occur at a maximum rate of 0.002 inches per year. At this rate, the bottom 2-3 bolts will be degraded by more than 5% of their cross sectional area before repairs are made. This would exceed ASME Section XI limits, however at this level of degradation sufficient bolt area would remain to maintain the pressure boundary. As a precaution, the operability of heat exchangers E-53A and E-53B should be reevaluated at the conclusion of the 2001 refueling outage. To avoid the affects of multiple failed bolts, the heat exchanger shell flanges are reinspected daily during operator and other plant staff rounds. While observations are not specifically recorded, unacceptable levels of leakage would result in corrective actions per Administrative Procedure 3.03 or 5.01.	

Analysis Detail	Reference/Comment
RECOMMENDATIONS: Work Order 24912387 (E-53A) and 24912388 (E-53B) exist to "Remove, clean and inspect bolting and retorque bolting to specification." Initiate an Engineering Assistance Request in accordance with Administrative Procedure 9.28 to determine bolt stress necessary to meet the "Gasket Seating Condition" load requirement. Initiate an Action Item Request in accordance with Administrative Procedure 3.02 to reevaluate E-53A and E-53B operability if repairs are not completed prior to the conclusion of the 2001 refueling outage. **CONCLUSIONS**: Based on current corroded condition and expected condition at the next refueling outage, it is concluded that E-53A and E-53B shell head connections are acceptable for continued service.	
ATTACHMENTS: **NOTE:** The following attachments define the terms and symbols used in the analysis. 1. ASME B&PV, Section III, 1965 Edition, Article I-12, "Stresses in Bolting" 2. ASME B&PV, Section III, 1989 Edition, Article XI-3000, "Design Requirements"	Comment 2

1965 ASME B&PV Code
EA-C-PAL-99-1240-01

ARTICLE I-12
Stresses in Bolting

I-1200 SCOPE

This Article is given as an aid to the designer and is not to be construed as a final method of analysis. The minimum cross-sectional area and the tentative values for the load in flange bolting may be determined as follows:

I-1210 NOTATION

The notations described below are used in the formulas of this Article.

A = outside diameter of flange or, where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots, inches

A_b = actual total cross-sectional area of bolts at root of thread or section of least diameter under stress, square inches

A_m = total required cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2}, square inches

A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions, square inches, $= W_{m1}/S_b$

A_{m2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating, square inches, $= W_{m2}/S_a$

B = inside diameter of flange, inches

b = effective gasket or joint-contact-surface seating width, inches (see Note 1, I-1220)

$2b$ = effective gasket or joint-contact-surface pressure width, inches (see Note 1, I-1220)

b_o = basic gasket seating width, inches (from Table I-1220)

C = bolt-circle diameter, inches

G = diameter at location of gasket load reaction. G is defined as follows (see Table I-1220): When $b_o \leq 1/4$ inch, G = mean diameter of gasket contact face, inches When $b > 1/4$ inch, G = outside diameter of gasket contact face less $2b$, inches

g_1 = thickness of hub at back of flange, inches

H = total hydrostatic end force, pounds = $0.785G^2P$

H_D = hydrostatic end force on area inside of flange, pounds = $0.785B^2P$

H_G = gasket load (difference between flange design bolt load and total hydrostatic end force), pounds = $W - H$

H_T = difference between total hydrostatic end force and the hydrostatic end force on area inside of flange, pounds, $= H - H_D$

h = hub length, inches

h_D = radial distance from the bolt circle, to the circle on which H_D acts, inches

h_G = radial distance from gasket load reaction to the bolt circle, inches, $= \dfrac{C - G}{2}$

h_T = radial distance from the bolt circle to the circle on which H_T acts, inches

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

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

M_o = total moment acting upon the flange, for the operating conditions or gasket seating, as may apply, inch-pounds (see I-1230)

M_T = component of moment due to H_T, inch-pounds, $= H_T h_T$

m = gasket factor, obtain from Table I-1220(a) (see Note 1 I-1220)

P = design pressure, pounds per square inch

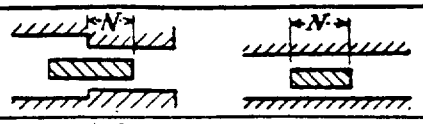
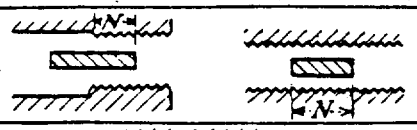
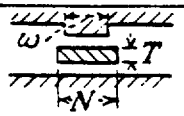
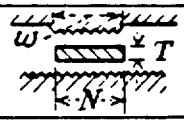
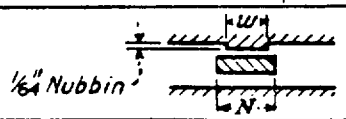
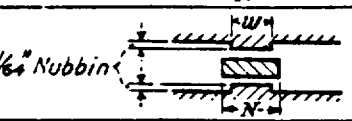
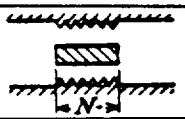
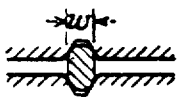
R = radial distance from bolt circle to point of intersection of hub and back of flange, inches (integral and hub flanges), $= \dfrac{C - B}{2} - g_1$

S_a = allowable bolt stress at atmospheric temperature, pounds per square inch (Table N-422)

S_b = allowable bolt stress at design temperature, pounds per square inch (given in Table N-422)

EA·C·PAL·92·1240·01

TABLE I-1220(b) EFFECTIVE GASKET WIDTH

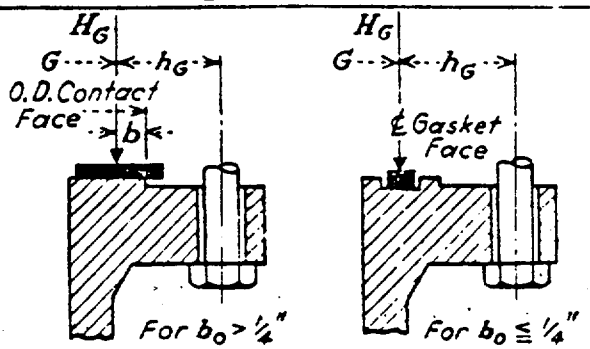
FACING SKETCH Exaggerated		BASIC GASKET SEATING WIDTH b_o COLUMN I	BASIC GASKET SEATING WIDTH b_o COLUMN II
1a		$\frac{N}{2}$	$\frac{N}{2}$
1b*			
1c	 $w \gtreqless N$	$\frac{w+T}{2}; \left(\frac{w+N}{4} \max\right)$	$\frac{w+T}{2}; \left(\frac{w+N}{4} \max\right)$
1d*	 $w \gtreqless N$		
2	 1/64" Nubbin $w \leqq \frac{N}{2}$	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
3	 1/64" Nubbin $w \leqq \frac{N}{2}$	$\frac{w}{2}; \left(\frac{N}{4} \min\right)$	$\frac{w+N}{4}; \left(\frac{3N}{8} \min\right)$
4*		$\frac{3N}{8}$	$\frac{7N}{16}$
5*		$\frac{N}{4}$	$\frac{3N}{8}$
6		$\frac{w}{8}$	

EFFECTIVE GASKET SEATING WIDTH "b"

$b = b_o$, when $b_o \leqq 1/4''$

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

LOCATION OF GASKET LOAD REACTION



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

* Where serrations do not exceed 1/64" depth and 1/32" width spacing, sketches 1b and 1d shall be used

ASME B & PV Code, Section III, 1989 Ed
EA. C. PAL. 99. 1240. 01

ARTICLE XI-3000
DESIGN REQUIREMENTS

XI-3100 GENERAL REQUIREMENTS

XI-3110 SCOPE

(a) The rules of XI-3200 apply to Class RF flanges as defined in XI-3212 while the rules of Appendix L apply to Class FF flanges.

(b) The flange design methods given in XI-3210 through XI-3250 apply to Class RF flanges under internal pressure. The flange design methods for Class RF flanges under external pressure or under both internal and external pressure are given in XI-3260.

~~XI-3120~~ TYPES OF FLANGES

For purposes of computation, there are three types as described in (a), (b), and (c) below.

(a) Loose Type Flanges. This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall and designs where the method of attachment is not considered to give the mechanical strength equivalent to integral attachment. Figure XI-3120-1 sketches (1), (1a), (2), (3), and (4) show typical loose type flanges and the location of the loads and moments; welds and other details of construction shall satisfy the dimensional requirements given in the referenced sketches.

(b) Integral Type Flanges. This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel, or pipe wall, butt welded thereto, or attached by other forms of arc or gas welding of such a nature that the flange and nozzle neck, vessel, or pipe wall is considered to be the equivalent of an integral structure. In welded construction, the nozzle neck, vessel, or pipe wall is considered to act as a hub. Figure XI-3120-1 sketches (5), (6), (6a), and (7) show typical integral type flanges and the location of the loads and moments; welds and other details of construction shall satisfy the dimensional requirements given in the referenced sketches.

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

$$g_0 = \tfrac{5}{8} \text{ in.}$$

$$B/g_0 = 300$$

$$P = 300 \text{ psi}$$

$$\text{Design Temperature} = 700°\text{F}$$

Figure XI-3120-1 sketches (8), (8a), (8b), and (9) show typical optional type flanges. Welds and other details of construction shall satisfy the dimensional requirements given in those sketches.

XI-3130 NOMENCLATURE

The nomenclature defined below and shown in Fig. XI-3120-1 is used in the equations for the design of flanges.

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

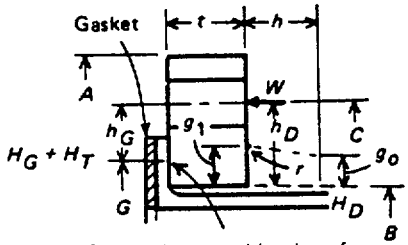
A_b = actual total cross-sectional area of bolts at root of thread or section of least diameter under stress, sq in.

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

A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the Design Conditions, sq in.

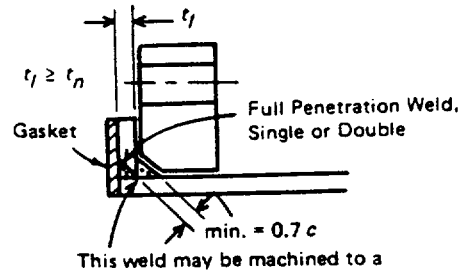
$= W_{m1}/S_b$

Loose Type Flanges



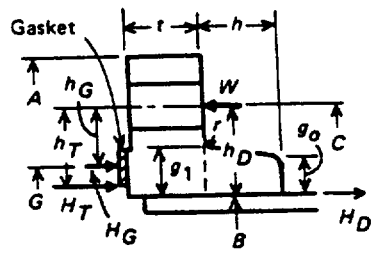
To be taken at mid-point of contact between flange and lap independent of gasket location

(1)



This weld may be machined to a corner radius to suit standard lap joint flanges

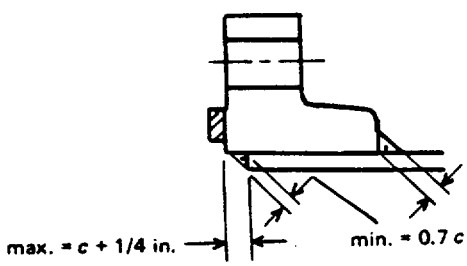
(1a)



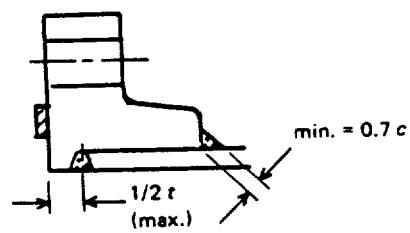
Screwed Flange With or Without Hub

(2)

Loadings and dimensions not shown are the same as for sketch (2).



(3)

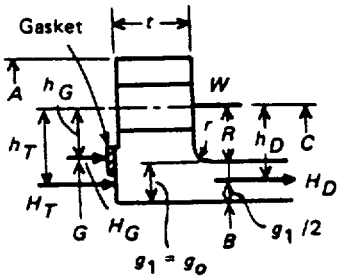


For hub tapers 6 deg. or less, use $g_o = g_1$

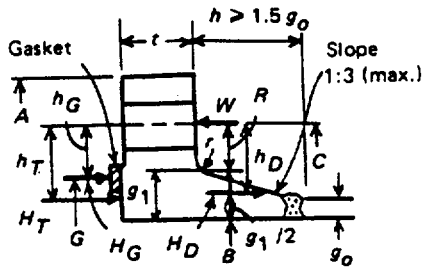
(4)

FIG. XI-3120-1 TYPES OF FLANGES

Integral Type Flanges



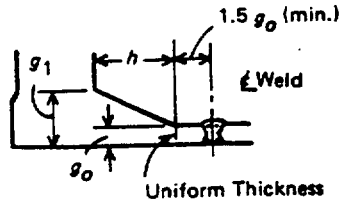
(5)



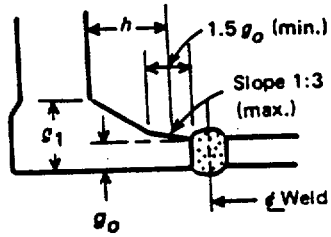
Where hub slope adjacent to flange exceeds 1:3 use sketches (6a) or (6b)

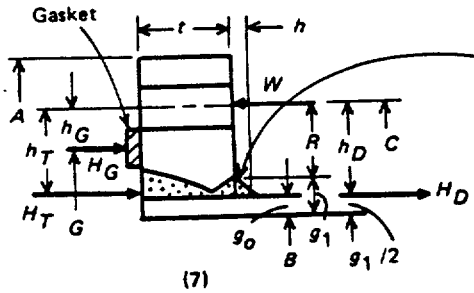
(6)

(6a)



(6b)





(7)

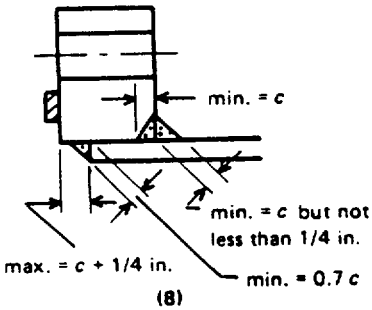
0.25 g_o but not less than 1/4 in., the minimum for either leg. This weld may be machined to a corner radius as permitted in sketch (5) in which case $g_1 = g_o$

GENERAL NOTE: Fillet radius r to be at least 0.25 g_o but not less than $^3/_{16}$ in. Added thickness greater than $^1/_{16}$ in. for raised face, tongue and groove, "O" rings, and ring joint facings shall be in excess of the required minimum flange thickness t; those less than or equal to $^1/_{16}$ in. may be included in the required minimum flange thickness.

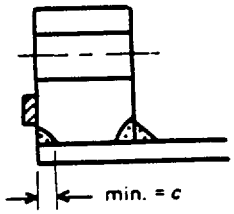
Optional Type Flanges

These may be calculated as either loose or integral type [XI-3120(c)]

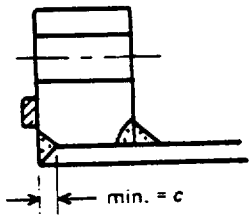
Loading and dimensions not shown are the same as for sketch (2) for loose type flanges or (7) for integral type.



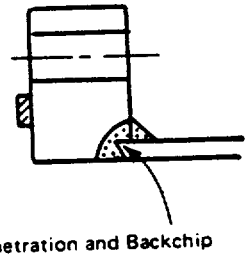
(8)



(8a)



(8b)



(9)

FIG. XI-3120-1 TYPES OF FLANGES (CONT'D)

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

a = shape factor for full face metal-to-metal contact flanges
$= (A + C)/2B_1$

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

$B_1 = B + g_1$, in., for loose type hub flanges and also for integral type flanges when f is less than 1
$= B + g_0$, in., for integral type flanges when f is equal to or greater than 1

b = effective gasket or joint contact surface seating width, in. (footnote 1, XI-3221.1)

b_0 = basic gasket seating width, in. (Table XI-3221.1-2)

C = bolt circle diameter, in.

c = basic dimension used for the minimum sizing of welds, in.; equal to t_n or t_D, whichever is less

D = diameter of bolt hole, in.

d = factor for integral type flanges

$$d = \frac{U}{V} h_0 g_0^2$$

= factor for loose type flanges

$$d = \frac{U}{V_L} h_0 g_0^2$$

E = modulus of elasticity of flange material, corrected for Design Temperature

e = factor for integral type flanges

$$e = F/h_0$$

= factor for loose type flanges

$$e = F_L/h_0$$

F = factor for integral type flanges (Fig. XI-3240-2)

F_L = factor for loose type flanges (Fig. XI-3240-4)

f = hub stress correction factor for integral flanges from Fig. XI-3240-6 (when greater than 1, this is the ratio of the stress in the small end of hub to the stress in the large end; for values below limit of Figure use $f = 1$)

G = diameter at location of gasket load reaction, except as noted in sketch (1) of Fig. XI-3120-1, G is defined as follows for Class RF flanges (see Table XI-3221.1-1):

(a) when $b_0 \leq 1/4$ in., G is the mean diameter of gasket contact face, in.;

(b) when $b_0 > 1/4$ in., G is the outside diameter of gasket contact face less $2b$, in.

= mean diameter of gasket, in., for Class FF flanges

g_0 = thickness of hub at small end, in.

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

H = total hydrostatic end force, lb
$= 0.785G^2P$

H_C = contact force between mating flanges, lb

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

H_G = for Class RF flanges only, gasket load (difference between flange design bolt load and total hydrostatic end force), lb
$= W - H$

H_p = total joint contact surface compression load, lb
$= 2b \times 3.14GmP$ (see Table XI-3221.1-1)

H_T = difference between total hydrostatic end force and the hydrostatic end force on area inside of flange, lb
$= H - H_D$

h = hub length, in.

h_c = radial distance, in., from bolt circle to flange spacer or flange to flange bearing circle, where the slope of the flange is calculated to be zero

$h_{c\,max}$ = radial distance, in., from bolt circle to outer edge of flange or spacer, whichever is less

h_D = radial distance, in., from the bolt circle to the circle on which H_D acts, as prescribed in Table XI-3230-1

h_G = radial distance, in., from gasket load reaction to the bolt circle
$= (C - G)/2$

h_0 = factor equal to $\sqrt{Bg_0}$, in.

h_T = radial distance, in., from the bolt circle to the circle on which H_T acts, as prescribed in Table XI-3230-1

j = ratio factor for hubbed flanges
$= 0.550(g_0/t)\sqrt{B_1/g_0}$

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

k = ratio factor for hubbed flanges
$= 2(Z + 0.3)/(B_1/g_0)$

L = factor equal to $(te + 1)/T + (t^3/d)$

l = calculated strain length of bolt; equal to $2t +$

t_s + ½ nominal diameter of bolt for each threaded end

M_D = component of moment due to H_D, in.-lb
$= H_D h_D$

M_G = component of moment due to H_G, in.-lb
$= H_G h_G$

M_H = moment acting on end of hub, pipe, or shell, at its junction with back face of flange ring, in.-lb

M_0 = total moment acting upon the flange, for the Design Conditions or gasket seating as many apply, in.-lb (XI-3230; this symbol applies to Class RF flanges only)

M_p = moment due to H_D, H_T, and H_G, for Class FF flanges, in.-lb
$= H_D h_D + H_T h_T + H_G h_G$

M_s = total moment on flange ring due to continuity with hub, pipe, or shell, in.-lb
$= M_H + Qt/2$

M_T = component of moment due to H_T, in.-lb
$= H_T h_T$

m = gasket factor obtained from Table XI-3221.1-1 (footnote 1, XI-3221.1)

N = width, in., used to determine the basic gasket seating width b_0, based upon the possible contact width of the gasket (Table XI-3221.1-2)

n = number of bolts

P = Design Pressure, psi (for flanges subject to external pressure see XI-3260 for Class RF flanges)

Q = shear force, lb, between flange ring and end of hub, pipe, or shell positive, as indicated in Fig. L-1312-1 sketch (6)

R = radial distance from bolt circle to point of intersection of hub and back of flange (integral and hub flanges), in.
$= (C - B)/2 - g_1$

r_B = bolt hole flexibility factor (Fig. L-1400-3)

r_E = elasticity factor which is equal to the modulus of elasticity of flange material divided by modulus of elasticity of bolting material, corrected for operating temperature

r_S = prestress factor
$r_S = 1.0 - (S_i/S_b)$

S_a = allowable bolt stress at atmospheric temperature, psi (given in Tables I-7.3, I-8.3, or I-10.3, as applicable)

S_b = allowable bolt stress, psi, at Design Temperature (given in Tables I-7.3, I-8.3, or I-10.3, as applicable)

S_f = allowable design stress, psi, for material of flange at Design Temperature (Design Condition) or atmospheric temperature (gasket seating), as applicable (given in Tables I-7.3, I-8.3, or I-10.3, as applicable)

S_n = allowable design stress, psi, for material of nozzle neck, vessel, or pipe wall at Design Temperature (Design Condition) or atmospheric temperature (gasket seating) as applicable (given in Tables I-7.0, I-8.0, or I-10.0, as applicable)

S_i = calculated bolt prestress, psi, at Design Temperature for Class FF flanges

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 = factor involving K (Fig. XI-3240-1)

t = flange thickness, in.

t_D = two times the thickness g_0, in., when the design is calculated as an integral flange, or two times the thickness, in., of shell or nozzle wall required for internal pressure when the design is calculated as loose flange, but not less than ¼ in.

t_n = nominal thickness, in., of shell or nozzle wall to which flange or lap is attached, less corrosion allowance

t_s = thickness of spacer, in.

U = factor involving K (Fig. XI-3240-1)

V = factor for integral type flanges (Fig. XI-3240-3)

V_L = factor for loose type flanges (Fig. XI-3240-5)

W = flange design bolt load, lb, for the Design Conditions or gasket seating as applicable (XI-3223 for Class RF flanges and L-1200 for Class FF flanges)

W_{m1} = minimum required bolt load, lb, for the Design Conditions (XI-3220)

W_{m2} = minimum required bolt load, lb, for gasket seating (XI-3220)

w = width, in., used to determine the basic gasket seating width b_0, based upon the contact width between the flange facing and the gasket (Table XI-3221.1-2)

Y = factor involving K (Fig. XI-3240-1)

y = gasket or joint contact surface unit seating load, psi (footnote 1, XI-3221.1)

Z = factor involving K (Fig. XI-3240-1)

z = axial separation, in., of identical mating flanges at inside diameter

β = shape factor for full face metal-to-metal contact flanges
$= (C + B_1)/2B_1$

θ_A = slope of flange face at outside diameter, rad

θ_B = slope of flange face at inside diameter, rad

XI-3200 CLASS RF FLANGE DESIGN

XI-3210 GENERAL REQUIREMENTS

XI-3211 Acceptability

The requirements for acceptability of Class RF flange design are given in (a) and (b) below.

(a) The design shall be such that the general design requirements of NC-3100, ND-3100, or NE-3100, as appropriate, and the specific design requirements of this Subarticle are met.

(b) The designs shall be limited to the types of flanges defined in XI-3120.

XI-3212 Definition of Class RF Flanges

Class RF flanges are circular flanges having gaskets which are entirely within the circle enclosed by the bolt holes and which have no contact outside this circle.

XI-3220 BOLT LOADS AND BOLT AREAS

XI-3221 Determination of Bolt Loads

In the design of a bolted flange connection, calculations shall be made for each of the two conditions, namely, Design Loadings and gasket seating loads, and the most severe condition will control. In the design of flange pairs used to contain a tubesheet of a heat exchanger, or any similar design where the flanges and/or gaskets may not be the same, loads shall be determined for the most severe condition of Design Loadings and/or gasket seating loads applied to each side at the same time. This most severe condition may be gasket seating on one flange with Design Loadings on the other, gasket seating on each flange at the same time, or Design Loadings on each flange at the same time.

XI-3221.1 Bolt Load for Design Conditions. The required bolt load for the Design Conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the Design Pressure on the area bounded by the diameter of gasket reaction and, in addition, to maintain on the gasket or joint contact surface a compression load H_p which experience has shown to be sufficient to ensure a tight joint. This compression load is expressed as a multiple m of the internal pressure. Its value is a function of the gasket material and construction.[1] The required bolt load for the Design Conditions W_{m1} is determined in accordance with Eq. (1):

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

XI-3221.2 Bolt Load for Gasket Seating Condition. Before a tight joint can be obtained, it is necessary to seat the gasket or joint contact surface properly by applying a minimum initial load under atmospheric temperature conditions without the presence of internal pressure, which is a function of the gasket material and the effective gasket area to be seated. The minimum initial bolt load W_{m2} required for this purpose shall be determined in accordance with Eq. (2):

$$W_{m2} = 3.14bGy \quad (2)$$

The need for providing sufficient bolt load to seat the gasket or joint contact surfaces in accordance with Eq. (2) will prevail on many low pressure designs and with facings and materials that require a high seating load and where the bolt load computed by Eq. (1) for the Design Conditions is insufficient to seat the joint. Accordingly, it is necessary to furnish bolting and to pretighten the bolts to provide a bolt load sufficient to satisfy both of these requirements, each one being individually investigated. When Eq. (2) governs, flange proportions will be a function of the bolting instead of internal pressure.

XI-3221.3 Bolt Load When Self-Energizing Gaskets Are Used. Bolt loads for flanges using gaskets of the self-energizing type differ from those shown above as stipulated in (a) and (b) below.

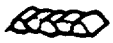
(a) The required bolt load for the Design Conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the Design Pressure on the area bounded by the outside diameter of the gasket. H_p is to be considered as zero for all self-energizing gaskets except certain seal configurations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

(b) $W_{m2} = 0$. Self-energizing gaskets may be considered to require an inconsequential amount of bolting force to produce a seal. Bolting, however, shall be pretightened to provide a bolt load sufficient to withstand the hydrostatic end force H.

[1] Tables XI-3221.1-1 and XI-3221.1-2 give a list of many commonly used gasket materials and contact facings, with suggested values of m, b, and y that have proved satisfactory in actual service. These values are suggested only and are not mandatory. Values that are too low may result in leakage at the joint, without affecting the safety of the design. The primary proof that the values are adequate is the hydrostatic test.

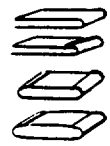
E-NAL-99-1240-01

TABLE XI-3221.1-1
GASKET MATERIALS AND CONTACT FACINGS[1]
Gasket Factors *m* for Operating Conditions and Minimum Design Seating Stress *y*

Gasket Material	Gasket Factor *m*	Min. Design Seating Stress *y*, psi	Sketches	Facing Sketch and Column in Table XI-3221.1-2
Self-energizing types (O rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0	. . .	. . .
Elastomers without fabric or high percent of asbestos fiber:				
Below 75A Shore Durometer	0.50	0		(1a),(1b),(1c),(1d), (4),(5); Column II
75A or higher Shore Durometer	1.00	200		
Asbestos with suitable binder for operating conditions:				
1/8 in. thick	2.00	1600		(1a),(1b),(1c),(1d), (4),(5); Column II
1/16 in. thick	2.75	3700		
1/32 in. thick	3.50	6500		
Elastomers with cotton fabric insertion	1.25	400		(1a),(1b),(1c),(1d), (4),(5); Column II
Elastomers with asbestos fabric insertion (with or without wire reinforcement):				
3-ply	2.25	2200		(1a),(1b),(1c),(1d), (4),(5); Column II
2-ply	2.50	2900		
1-ply	2.75	3700		
Vegetable fiber	1.75	1100		(1a),(1b),(1c),(1d), (4),(5); Column II
Spiral-wound metal, asbestos filled:				
Carbon	2.50	10,000		(1a),(1b); Column II
Stainless or Monel	3.00	10,000		
Corrugated metal, asbestos inserted; or corrugated metal, jacketed asbestos filled:				
Soft aluminum	2.50	2900		
Soft copper or brass	2.75	3700		(1a),(1b); Column II
Iron or soft steel	3.00	4500		
Monel or 4–6% chrome	3.25	5500		
Stainless steels	3.50	6500		

EA-C-PAL-99-1240-01

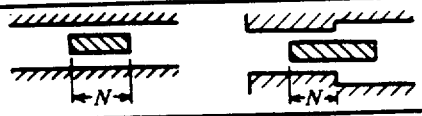
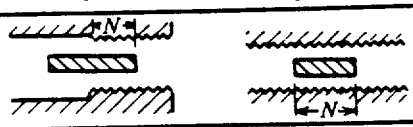
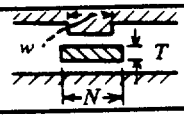
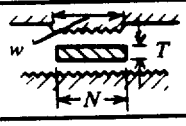
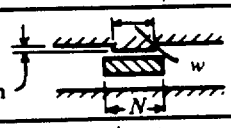
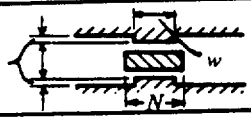
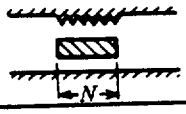
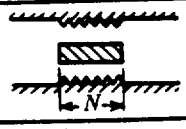
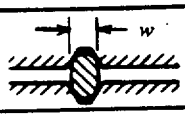
TABLE XI-3221.1-1 (CONT'D)
GASKET MATERIALS AND CONTACT FACINGS[1]
Gasket Factors *m* for Operating Conditions and Minimum Design Seating Stress *y*

Gasket Material	Gasket Factor *m*	Min. Design Seating Stress *y*, psi	Sketches	Facing Sketch and Column in Table XI-3221.1-2
Corrugated metal:				
Soft aluminum	2.75	3700		(1a),(1b),(1c),(1d), Column II
Soft copper or brass	3.00	4500		
Iron or soft steel	3.25	5500		
Monel or 4–6% chrome	3.50	6500		
Stainless steels	3.75	7600		
Flat metal, jacketed asbestos filled:				
Soft aluminum	3.25	5500		(1a),(1b),(1c),[2] (1d),[2](2)[2]; Column II
Soft copper or brass	3.50	6500		
Iron or soft steel	3.75	7600		
Monel	3.50	8000		
4–6% chrome	3.75	9000		
Stainless steels	3.75	9000		
Grooved metal:				
Soft aluminum	3.25	5500		(1a),(1b),(1c),(1d), (2),(3); Column II
Soft copper or brass	3.50	6500		
Iron or soft steel	3.75	7600		
Monel or 4–6% chrome	3.75	9000		
Stainless steels	4.25	10,100		
Solid flat metal:				
Soft aluminum	4.00	8800		(1a),(1b),(1c),(1d), (2),(3),(4),(5); Column I
Soft copper or brass	4.75	13,000		
Iron or soft steel	5.50	18,000		
Monel or 4–6% chrome	6.00	21,800		
Stainless steels	6.50	26,000		
Ring joint:				
Iron or soft steel	5.50	18,000		(6), Column I
Monel or 4–6% chrome	6.00	21,800		
Stainless steels	6.50	26,000		

NOTES:
(1) This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of *m* and *y* that have generally proved satisfactory in actual service when using effective gasket seating width *b* given in Table XI-3221.1-2. 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.

EA-C-PAL-99-1240-01

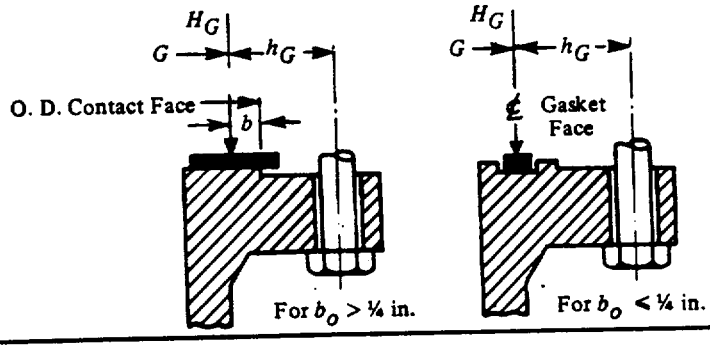
TABLE XI-3221.1-2
EFFECTIVE GASKET WIDTH[2]

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

Effective Gasket Seating Width, b

$$b = b_o, \text{ when } b_o \le \tfrac{1}{4}\text{ in.}; \; b = 0.5\sqrt{b_o}, \text{ when } b_o > \tfrac{1}{4}\text{ in.}$$

Location of Gasket Load Reaction



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

XI-3222 Total Required and Actual Bolt Areas A_m and A_b

The total cross-sectional area of bolts A_m required for both the Design Conditions and gasket seating is the greater of the values for A_{m1} and A_{m2}, where $A_{m1} = W_{m1}/S_b$ and $A_{m2} = W_{m2}/S_a$. A selection of bolts to be used shall be made such that the actual total cross-sectional areas of bolts A_b will not be less than A_m.

XI-3223 Flange Design Bolt Load W

The bolt loads used in the design of the flange shall be the values obtained from Eqs. (3) and (4). For Design Conditions,

$$W = W_{m1} \tag{3}$$

For gasket seating,

$$W = \frac{(A_m + A_b)\,S_a}{2} \tag{4}$$

In addition to the minimum requirements for safety, Eq. (4) provides a margin against abuse of the flange from overbolting. Since margin against such abuse is needed primarily for the initial bolting up operation, which is done at atmospheric temperature and before application of internal pressure, the flange design is required to satisfy this loading only under such conditions.[2]

XI-3230 FLANGE MOMENTS

(a) In the calculation of flange stresses, the moment of a loading acting on the flange is the product of the load and its moment arm. The moment arm is determined by the relative position of the bolt circle with respect to that of the load producing the moment (Fig. XI-3120-1). No consideration shall be given to any possible reduction in moment arm due to cupping of the flanges or due to inward shifting of the line of action of the bolts as a result thereof.

(b) For the Design Conditions, the total flange moment M_0 is the sum of the three individual moments M_D, M_T, and M_G, as defined in XI-3130, and based on the flange design bolt load of Eq. (3) with moment arms as given in Table XI-3230-1.

TABLE XI-3230-1
MOMENT ARMS FOR FLANGE LOADS UNDER OPERATING CONDITIONS

Flange Type	h_D	h_T	h_G
Integral type flanges [see Fig. XI-3120-1 sketches (5), (6), (6a), (6b), (7), (8), (8a), (8b), and (9)]	$R + 0.5g_1$	$\frac{R + g_1 + h_G}{2}$	$\frac{C - G}{2}$
Loose type, except lap joint flanges [Fig. XI-3120-1 sketches (2), (3), and (4)]; and optional type flanges [Fig. XI-3120-1 sketches (8), (8a), (8b), and (9)]	$\frac{C - B}{2}$	$\frac{h_D + h_G}{2}$	$\frac{C - G}{2}$
Lap joint flanges [Fig. XI-3120-1 sketch (1)]	$\frac{C - B}{2}$	$\frac{C - G}{2}$	$\frac{C - G}{2}$

(c) For gasket seating, the total flange moment M_0 is based on the flange design bolt load of Eq. (4), which is opposed only by the gasket load, in which case:

$$M_0 = W\frac{(C - G)}{2} \tag{5}$$

XI-3240 CALCULATION OF FLANGE STRESSES

The stresses in the flange shall be determined for both the Design Conditions and gasket seating, whichever controls, in accordance with the equations in (a) or (b) below.

(a) For integral type flanges and all hub type flanges:

Longitudinal hub stress

$$S_H = fM_0/Lg_1^2B \tag{6}$$

Radial flange stress

$$S_R = [(1.33te + 1)M_0]/Lt^2B \tag{7}$$

Tangential flange stress

$$S_T = (YM_0/t^2B) - ZS_R \tag{8}$$

[2]Where additional safety against abuse is desired, or where it is necessary that the flange be suitable to withstand the full available bolt load $A_b \times S_a$, the flange may be designed on the basis of this latter quantity.