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EVALUATION OF THE BOLTING AND FLANGES OF ANSI B16.5 FLANGED JOINTS — ASME PART A DESIGN RULES

E. C. Rodabaugh
S. E. Moore

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TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

FOREWORD

The work reported here was performed at the Oak Ridge National Laboratory and at Battelle-Columbus Laboratories under Union Carbide Corporation, Nuclear Division, Subcontract No. 2913 in support of the ORNL Design Criteria for Piping and Nozzles Program. This program is sponsored by the Division of Reactor Safety Research, Office of Nuclear Regulatory Research, U. S. Nuclear Regulatory Commission; E. K. Lynn of the Metallurgy and Materials Branch is the cognizant RSR engineer, and S. E. Moore, Reactor Division, ORNL, is the program manager.

Activities under the Design-Criteria program are coordinated with other safety-related piping and pressure-vessel research through the Design Division of the Pressure Vessel Research Committee (PVRC) of the Welding Research Council and through the ASME Boiler and Pressure Vessel Code Committee as part of a cooperative effort with industry to confirm and/or improve the adequacy of design criteria and analytical methods used to assure the safe design of nuclear power plants.

The study of ANSI B16.5 flanged piping joints, conducted using the ASME Part-A design rules described in this report, is a portion of a three-part study of flanged joints for the evaluation of design rules in current codes and standards. Results from these studies will be used by appropriate ASME Code groups in drafting new and improved design rules.

Other reports in this series are:

1. E. C. Rodabaugh, F. M. O'Hara, Jr., and S. E. Moore, FLANGE: A Computer Program for the Analysis of Flanged Joints with Ring-Type Gaskets, ORNL-5055 (January, 1976).

2. E. C. Rodabaugh and S. E. Moore, Flanged Joints with Contact Outside the Bolt Circle -- ASME Part B Design Rules, ORNL/Sub/2913, Report No. 1 (May, 1976).

1. INTRODUCTION

The bolted flanged joint represents perhaps the most complicated structure for which specific design rules are given in the Code(1)*. For most structures (pressure boundaries) covered by the Code, the stress is proportional to the pressure, and "failure" occurs when the pressure boundary is penetrated by a crack. For flanged joints, however, the major stresses occur due to tightening the bolts and, in general, are changed only slightly by pressure. The criterion defining failure is usually excessive leakage at the gasket. Such leakage can often by stopped by additional tightening of the bolts; i.e., the problem is solved by imposing still higher stresses on the bolts and flanges.

Perhaps because of the unusual characteristics of a flanged joint, the basic philosophy of the Code, in which maximum stress limits are imposed, is somewhat difficult to intrepret when applied to a bolted flanged joint. This is particularly true for piping where ANSI B16.5 flanges are ordinarily used.

A review of the NB-3600 rules for Class 1 piping design, as they existed in the 1971 edition of the Code, was made in 1972 by one of the authors. At that time, several major problems in the Code rules were pointed out concerning flanged joints. One of these involved subarticles NC-3600 and ND-3600 which, by reference to NB-3647, in effect, required a fatigue analysis of the bolting of flanged joints in Class 2 and 3 piping. This has been corrected by copying the text of NB-3647 into NC-3647 and into ND-3647, but leaving out the sentence "These minimum bolt areas may have to be increased to meet requirements of NB-3230." Another problem involved the use of the equation contained in NB-3647.1(a) for calculating the equivalent pressure, P_{eq}. It was noted that using P_{eq} for checking the capacity of B16.5 flanged joints for pipe bending-moment capacity would give very low allowable moments. It was recommended, as a partial solution, that P rather than P_{eq} be used in calculating H_p. This was incorporated into the Code as Case 1677.

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* Code, in this report, refers specifically to the ASME Boiler and Pressure Vessel Code, Section III, Nuclear Power Plant Components, Reference (1).

At the present time, a number of ambiguities and unnecesary restrictions on the use of B16.5 flanged joints in the Code rules still remain. This report was prepared to investigate the use of B16.5 flanged joints in Code Applications and to make recommendations for appropriate modifications of the Code rules. Toward this end, calculations were made for a representative sampling of B16.5 flanges as described in Chapter 3, Specific Flanged Joint Calculations".

Calculations were made using the Code procedures and also with the more basic theoretical analysis of Reference (3), and the computer program FLANGE which was developed to implement that analysis.

Both the Code procedures and the theoretical analysis of Reference (3) are based on simplified analytical methods. Available test data on flanged joints are presented and compared with results from FLANGE calculations in Appendix A. These comparisons indicate that the calculation methods used in the text of the report are conservative, but not necessarily accurate.

One objective of this report is to develop relatively simple but conservative methods of evaluating the adequacy of piping products used in Code Class-1, Class-2, and Class-3 piping systems. The rather extensive and detailed evaluations of ANSI B16.5 flanged joints contained in the text of this report provide the basis for the development of relatively simple equations for ANSI B16.5 flanged joints developed in Appendix B. These simple equations from Appendix B are then included in Appendix C in the form of recommended revisions to the Code.

Nomenclature

A_b = Bolt cross sectional area (all bolts)

A_b' = Bolt cross sectional area of a single bolt

A_m = Design bolt area = larger of W_{m1}/S_b or W_{m2}/S_a, Code definition

a_1, a_2, a_3, a_4 = Bolt change factors, see Eq. (9) and Tables 5 and 6.*

B = Flange inside diameter

b = Effective gasket seating width, Code definition

C = Bolt circle diameter

D_o = Pipe outside diameter

D_m = Mean (midwall) pipe diameter

d = Bolt diameter

E = Modulus of elasticity

E_1 = Modulus of elasticity at bolt preload temperature

E_2 = Modulus of elasticity at operating temperatures

F = Axial force in attached pipe (see Table 1)

F_b = Axial force in bolt

G = Effective gasket diameter, Code definition

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* For users of the computer program FLANGE:

a_1 = (W1-W2B)/A_b for $P = P_{eq} = \overline{P}$

a_2 = (W1-W2C)A_b for $P = \overline{P}$

a_3 ≈ (W1-W2A)/A_b for $(T_b - T_f) = 50$ F

a_4 = (W1-W2D)/A_b for $(T_{hp} - T_f) = 100$ F

Nomenclature (Continued)

G_o = Gasket outside diameter

G_i = Gasket inside diameter

G_m = Gasket mean diameter = $(G_o + G_i)/2$

G_e = Effective gasket diameter

g_o = Attached pipe wall thickness, welding neck flanges

H_p = $2b\,\pi GmP$, Code definition

h = Tapered hub length, welding neck flanges

h_G = $(C - G')/2$, where $G' = G$, G_m or G_e as indicated in text

ℓ = Effective bolt length, taken herein as $2t + d + 1/8''$

M_{pb} = The symbol M for pipe moment is used only in Table 1, taken from the Code. Elsewhere, it is identified as M_{pb}. This is to distinguish it from the moments applied by couples to flange (e.g., bolt-load/gasket reaction couple).

M = Moment applied by couples to flange e.g., bolt load/gasket reaction couple).

M_o = Total moment acting upon flange, Code definition

M_{gs} = Total moment acting upon flange, gasket seating conditions, Code definition

m = Gasket factor, Code definition

N_d = Design cyclic life as obtained by Code procedure

P = Internal pressure

$\overline{P}$ = Primary rating pressure per ANSI B16.5 (equal to class designation)

P_{eq} = Pressure equivalent of pipe bending moment, NB-3647 definition, $= 16\, M_{pb}/\pi G^3$

P'_{eq} = Pressure equivalent of pipe bending moment as used in FLANGE, $= 4S_{pb}g_o/D_o$

P_L = Leakage pressure

Nomenclature (Continued)

P_b, Q	=	Stress categories, Code definitions
S	=	Stress
S_{all}	=	Allowable flange stress
S_a	=	Allowable bolt stress at atmospheric temperature, Code definition
S_b	=	Allowable bolt stress at design temperature, Code definition
S_{bb}	=	Bending stress in bolts
S_{b1}	=	Initial or preload bolt stress
S_{b2}	=	Bolt stress after application of loads
S_{bc}	=	Critical bolt stress
S_{bs}	=	Shear bolt stress
S_H, S_R, S_T	=	Flange stresses, Code definition
S_m	=	Allowable stress intensity, Code definition, Class 1 components
S_{pb}	=	Maximum stress in attached pipe due to moment, M_{pb}
S_n	=	Range of primary-plus-secondary stress
S_p	=	Range of primary-plus-secondary-plus-peak stress
$\overline{S}_b$	=	Stress intensity in a bolt, including bending stress
S_a'	=	Stress amplitude (Used in conjunction with Code fatigue curves)
T	=	Torque applied in tightening bolts
T_b	=	Temperature of bolts
T_f	=	Temperature of flange ring
T_{hp}	=	Temperature of flange hub and pipe
t	=	Flange ring thickness and blind flange thickness
W	=	Total bolt load
W_{m1}	=	Design bolt load, operating conditions, Code definition

Nomenclature (Continued)

W_{m2} = Design bolt load, gasket seating conditions, Code definition

y = Gasket seating load, Code definition

Θ = Rotation in radians

Θ_1 = Rotation of flange ring of one flange in joint

Θ_2 = Rotation of flange ring of other flange in joint

Θ_m = Rotation of flange ring due to moment

Θ_p = Rotation of flange ring due to pressure

Θ_T = Rotation of flange ring due to thermal gradient $(T_{hp} - T_f)$

2. CODE REQUIREMENTS FOR FLANGED JOINTS

In the Code, NB-3640 (Pressure Design) requires that the product satisfy the stated criteria for the design pressure at the design temperature. Under NB-3650, the product must satisfy the stated criteria for all loads; i.e., pressures, moments and thermal gradients. The rules for flanged joints in the Code do not conform to this basic organization; one of the objectives of the recommendations of this report is to provide rules under NB-3640 and NB-3650 that parallel those for all other products.

NB-3640 contains two paragraphs which are relevant to the subject of B16.5 flanged joints: These are NB-3647 and NB-3649. NB-3649 states that:

> "Other piping products manufactured in accordance with the standards listed in Table NB-3691-1 shall be considered suitable for use provided the design is consistent with the design philosophy of this subsection."

There are three significant points with respect to the above quote.

(1) ANSI B16.5 is included in Table NB-3691-1.

(2) "This subsection", by Code definition, is the entire NB-portion of the Code.

(3) NB-3612.1 provides an additional restriction in stating that the pressure-temperature ratings of the standards shall not be exceeded.

Accordingly, NB-3649 implies that B16.5 flanges meet the requirements of NB-3640 (pressure design) provided the design pressure/temperature is below the rated pressure/temperature combination given in B16.5. However, it is not clear what is meant by "the design is consistent with the design philosophy of this subsection". As written, the Code requires that the design of B16.5 flanged joints be consistent with the NB subsection design philosophy. A seemingly appropriate way to check for "consistency" is to see if B16.5 flanged joints conform to the requirements of NB-3647, "Pressure Design of Flanged Joints and Blanks"; in particular, NB-3647.1, "Flanged Joints". NB-3647 is shown herein as Table 1. NB-3647.1(b) gives methods to establish minimum bolt areas, but then states "These minimum bolt areas may have to be increased to meet requirements of NB-3230. NB-3230 is shown herein as Table 2.

A major portion of this report is directed towards the evaluation of B16.5 flanged joints and their pressure-temperature ratings to see if they are consistent with the requirements of NB-3647.1. The detailed interpretation of the rules is covered in the text. At this time, we only point out that:

(1) NB-3647.1 is a part of NB-3640 on Pressure Design, yet it gives explicit rules for moments applied to the flanged joint by the attached piping. Moment loads, for all other products, are considered explicitly <u>only</u> under NB-3650 on Analysis of Piping Products.

(2) The moment to be used in NB-3647.1 apparently does not include all of the significant sources of moment; e.g., relief valve thrust, anchor movements or misalignment that is "pulled-out" during make-up of the joint.

(3) The explicit inclusion of axial force loads in NB-3647.1 is different from requirements for other piping products. This type of load is not explicitly considered for other piping products and, if considered, would presumably be done under NB-3650.

(4) NB-3647 does not cover blind flanges.

The Code requirements for what is needed to show that a B16.5 flanged joint meets NB-3650 appear to be quite clear. First,

TABLE 1, NB-3647 FROM ASME III

NB-3647 Pressure Design of Flanged Joints and Blanks

NB-3647.1 Flanged Joints. This paragraph presents an acceptable method for calculating pressure stresses in flanged joints but should not be used for calculating maximum service stresses which are required to meet NB-3230. The designer may use an alternative procedure provided that it includes the modifications of this paragraph for the loads and stress limits for Class 2 piping. Flanged joints shall be analyzed and the stresses evaluated by using the methods given in Appendix XI, revised in accordance with (a) through (d) below.

(a) The design pressure used for the calculation of loads in a flanged joint by the equations in XI-3220 and XI-3230, shall be replaced by a flange design pressure, $P_{FD} = P + P_{eq}$, where P is the design pressure as defined in NB-3112.1 and P_{eq} is an equivalent pressure to account for the moments and forces acting on the flanged joint due to weight and thermal expansion of the piping. The equivalent pressure, P_{eq} shall be determined by the equation

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

where:

M = bending moment applied to the joint due to weight and thermal expansion of the piping, in. lb

F = axial force applied to the joint due to weight and thermal expansion of the piping, in. lb

G = diameter at location of effective gasket load reaction as defined in XI-3130, in.

(b) Equations (3) and (4) in XI-3220 shall be used to establish minimum bolt area required using allowable stress values equal to S_m as given in Table I-1.3. These minimum bolt areas may have to be increased to meet requirements of NB-3230.

(c) Equation (6) in XI-3240 for longitudinal hub stress shall be revised to include the primary axial membrane stress as follows

$$S_H = \frac{fM_0}{Lg_1{}^2B} + \frac{PB}{4g_0}$$

where:

P = design pressure as defined in NB-3112.1, psi. Other terms are defined in XI-3130.

(d) The allowable stress limits shall be

S_H not greater than 1.5 S_m
S_R not greater than 1.5 S_m and
S_T not greater than 1.5 S_m

TABLE 2. NB-3230 FROM ASME III

NB-3230 STRESS LIMITS FOR BOLTS

NB-3231 Design Conditions

(a) The number and cross sectional area of bolts required to resist the design pressure shall be determined in accordance with the procedures of Appendix E, using the larger of the bolt loads given by the equations of Appendix E as a design mechanical load. The allowable bolt design stresses shall be the values given in Table I-1.3, for bolting materials.

(b) When sealing is effected by a seal weld instead of a gasket, the gasket factor, *m*, and the minimum design seating stress, *y*, may be taken as zero.

(c) When gaskets are used for preservice testing only, the design is satisfactory if the above requirements are satisfied for $m = y = 0$, and the requirements of NB-3232 are satisfied when the appropriate *m* and *y* factors are used for the test gasket.

NB-3232 Normal Conditions

Actual service stresses in bolts, such as those produced by the combination of preload, pressure, and differential thermal expansion may be higher than the values given in Table I-1.3.

NB-3232.1 Average Stress. The maximum value of service stress, averaged across the bolt cross section and neglecting stress concentrations, shall not exceed two times the stress values of Table I-1.3.

NB-3232.2 Maximum Stress. The maximum value of service stress except as restricted by NB-3223.3, at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations shall not exceed 3 times the stress values of Table I-1.3. Stress intensity, rather than maximum stress, shall be limited to this value when the bolts are tightened by methods other than heaters, stretchers, or other means which minimize residual torsion.

NB-3232.3 Fatigue Analysis of Bolts. Unless the components on which they are installed meet all the conditions of NB-3222.4(d) and thus require no fatigue analysis, the suitability of bolts for cyclic operation shall be determined in accordance with the procedures of (a) through (e) below.

(a) Bolting Having Less Than 100,000 psi Tensile Strength. Bolts made of materials which have specified minimum tensile strengths of less than 100,000 psi shall be evaluated for cyclic operation by the methods of NB-3222.4(e), using the applicable design fatigue curve of Fig. I-9.4 and an appropriate fatigue strength reduction factor (NB-3232.3(c)).

(b) High Strength Alloy Steel Bolting. High strength alloy steel bolts and studs may be evaluated for cyclic operation by the methods of NB-3222.4(e) using the design fatigue curve of Fig. I-9.4 provided:

(1) The maximum value of the service stress (NB-3232.2) at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentration shall not exceed 2.7 S_m, if the higher of the two fatigue design curves given in Fig. I-9.4 is used. The 2 S_m limit for direct tension is unchanged.

(2) Threads shall be of a Vee-type having a minimum thread root radius no smaller than 0.003 in.

(3) Fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060.

(c) Fatigue Strength Reduction Factor (NB-3213.17). Unless it can be shown by analysis or tests that a lower value is appropriate, the fatigue strength reduction factor used in the fatigue evaluation of threaded members shall not be less than 4.0. However, when applying the rules of NB-3232.3(b) for high strength alloy steel bolts, the value used shall not be less than 4.0.

(d) Effect of Elastic Modulus. Multiply S_{alt} (as determined in NB-3216.1 or NB-3216.2) by the ratio of the modulus of elasticity given on the design fatigue curve to the value of the modulus of elasticity used in the analysis. Enter the applicable design fatigue curve at this value on the ordinate axis and find the corresponding number of cycles on the axis of abscissas. If the operational cycle being considered is the only one which produces significant fluctuating stresses, this is the allowable number of cycles.

(e) Cumulative Damage. The bolts shall be acceptable for the specified cyclic application of loads and thermal stresses provided the cumulative usage factor, *U*, as determined in NB-3222.4(e)(5) does not exceed 1.0.

NB-3233 Upset Conditions

The stress limits for Normal Conditions (NB-3232) apply.

NB-3234 Emergency Conditions

The stress limits of NB-3232.1 and NB-3232.2 apply.

NB-3235 Faulted Conditions

If the Design Specifications specify any Faulted Conditions (NB-3113.4), the rules contained in Appendix F may be used in evaluating these Faulted Conditions, independently of all other Design and Operating Conditions.

we note that stress indices for flanged joints (or flanges) are not given in Table NB-3683.2-1. Accordingly, the flanged joint must* be analyzed according to the detailed requirements of NB-3200. The analysis of the bolting is already covered by NB-3647.1 through its reference to NB-3230. This leaves the flanges to be analyzed; this aspect is discussed in this report in Chapter 5, "Flange Stresses".

3. SPECIFIC FLANGED JOINT CALCULATIONS

There are a large number of variables included in B16.5 flanges; i.e., pressure class, size, type, inside diameter (pipe wall thickness for integral flanges), gasket dimensions and materials. While a complete coverage of all these variables was not deemed feasible, calculations for a fairly extensive sampling have been made, and the results are presented herein. The sampling should be sufficient to give the designer a good feel for important aspects of B16.5 flanged joints and, with some restrictions, is deemed sufficient to provide the equivalent of stress indices for such joints. The parameters covered are listed below.

Classes, Sizes, Types

Pressure Class: all, i.e., 150, 300, 600, 900, 1500 and 2500 lb.

Sizes: 4", 8", 16" and 24" in all but the 2500 lb. For 2500 lb (maximum size covered is 12") the sizes are 4", 8", and 12".

Types:

(a) Welding neck flange joined to another welding neck

(b) Welding neck flange joined to straight hub flange

(c) Welding neck flange joined to a blind flange

---

* In principle, the Code user could develop stress indices or conduct an experimental analysis.

Pressure-Temperature Ratings

The ratings given in the 1968 edition of B16.5 are used in this report. The primary rating pressure, $\overline{P}$, is:

$$\overline{P} = \text{class designation (psi)} \qquad (1)$$

For example, $\overline{P}$ = 600 psi for the 600 class. The temperature at which $\overline{P}$ applies depends upon the flange material. Ratings at other temperatures are given in B16.5-1968.

Dimensions

Most dimensions of B16.5 flanged joints are given in B16.5; however, a few are not. These were selected as follows.

Inside Diameter (Wall Thickness): The wall thickness of the pipe attached to the welding neck flange was established by the equation:

$$g_o = \frac{\overline{P}D_o}{2S} + 0.05 \qquad (2)$$

where $\overline{P}$ = primary rating pressure, $\overline{P}$ (psi) is equal to the class designation; e.g., $\overline{P}$ = 150 psi for the 150 lb class

D_o = pipe outside diameter

S = allowable stress, taken as 8750 psi. The value of S = 8750 psi is approximately equal to the allowable stress of flange materials at the primary rating temperature given in B16.5. It is also the "representative allowable stress" used in proportioning ratings in B16.5 to ASME Code allowable stresses.

The calculated value of g_o also establishes the flange inside diameter bore, B, and the hub thickness, g_1; i.e., $B = D_o - 2g_o$ and $g_1 = (X - B)/2$, where X = hub diameter at juncture with ring; X is given in B16.5. The tapered length of the hub was set equal to $Y - C - 1.5g_o$, where Y - C is the total hub length; Y and C are given in B16.5.

For straight hub flanges, the thickness g_1 was assumed to extend through the hub and pipe. This forms a reasonably good model of a flange cast integrally into a valve or fitting body.

Gasket Dimensions: The gasket outside diameter is equal to the raised face diameter (given in B16.5) and the inside diameter is equal to the pipe outside diameter. Gasket thickness is 1/16".

Bolt Dimensions: The bolt areas are based on the root diameter of the thread. Threads are Coarse-thread series in sizes 1" and smaller; 8-pitch-thread series in sizes 1-1/8" and larger. Effective bolt length is taken as 2t + d + 1/8", where t = flange ring thickness, d = bolt diameter.

Calculations were made for bolt and flange stresses in accordance with Code procedures (Code Appendix XI)*. These Code procedures lead to minimum bolt areas and design loads which, in part, are proportioned to the design pressure. The stresses due to the design loads were then calculated. Calculations were also made using the more basic theoretical analysis described in Reference (3). This analysis, because it considers the flanges, bolts, and gasket as a statically redundant structure, uses the modulus of elasticity of the materials of the flanges, bolts, and gaskets. A modulus of 3×10^7 psi was assumed for these materials.

The Code calculations require the use of m and y factors which depend upon the gasket material. The values used in the calculations were m = 2.75, y = 3700; corresponding to the Code values for 1/16" thick asbestos gasket material.

---

* Appendix XI does not cover blind flanges. The procedure of NC-3325 and Figure NC-3325.1(e) was used for code calculations of stresses in blind flanges.

4. BOLT AREAS AND BOLT STRESSES

Required Bolt Areas Per NB-3647.1(b)

NB-3647.1(b) states that "Equations (3) and (4) in XI-3220 shall be used to establish minimum bolt area required using allowable stress values equal to S_m as given in Table I-1.3.

Equations (3) and (4) of XI-3220 are:

For operating conditions:

$$W_{m1} = (\pi/4)\ G^2P + 2b\pi GmP \tag{3}$$

For gasket seating:

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

The required minimum bolt area, A_m, is the larger of W_{m1}/S_b or W_{m2}/S_a, where $W_{m2} = \pi bGy$. The values given by equations (3) and (4) will, of course, depend upon the gasket diameters and the Code m and y gasket factors.

Table 3 shows values of W_{m1} and W_{m2} for a commonly used gasket. The required minimum bolt areas shown in Table 3 are based on S_m = 35000 psi for SA-193 Code B7 bolt material at 100 F. The pressures are the rated pressures at 100 F. It can be seen that the actual bolt areas are larger than the required minimum. Because the rated pressures decrease more with increasing temperature than the decrease in the allowable bolt stress with increasing temperature, it follows that the bolt area is adequate for all temperatures.

Table 3 is for pressure loading only. When a bending moment is imposed on the joint by the attached pipe, NB-3647.1(a) requires* that the design pressure be obtained by:

$$P_{FD} = P + \frac{16\ M_{pb}}{\pi G^3} \tag{5}$$

---

* The direct axial force term is not included because its use is not consistent with the present Code analysis of other piping products.

TABLE 3. COMPARISON OF MINIMUM REQUIRED AND ACTUAL BOLT AREAS, PRESSURE LOADING ONLY, PRESSURE IS RATED PRESSURE AT 100 F

Class	Size	P, psi	W_{m1}, 1000 lb	W_{m2}, 1000 lb	A_b, in^2 min.	A_b, in^2 actual
150	4	275	15.2	20.9	0.60	1.616
	8		37.9	40.7	1.16	2.416
	16		101.0	81.3	2.88	8.816
	24		206.0	138.0	5.89	18.580
300	4	720	39.7	20.9	1.13	2.416
	8		99.2	40.7	2.83	5.028
	16		264.0	81.3	7.55	18.580
	24		540.0	138.0	15.43	33.72
400	4	960	52.9	20.9	1.51	3.352
	8		132.2	40.7	3.78	6.612
	16		352.0	81.3	10.06	23.100
	24		720.0	138.0	20.57	47.520
600	4	1440	79.4	20.9	2.27	3.352
	8		199.0	40.7	5.67	8.736
	16		529.0	81.3	15.10	28.100
	24		1080.0	138.0	30.86	55.296
900	4	2160	119.0	20.9	3.40	5.824
	8		298.0	40.7	8.50	13.860
	16		793.0	81.3	22.65	33.600
	24		1620.0	138.0	46.29	85.840
1500	4	3600	198.0	20.9	5.67	7.432
	8		496.0	40.7	14.17	20.160
	16		1322.0	81.3	37.76	68.672
	24		2700.0	138.0	77.15	139.980
2500	4	6000	331.0	20.9	9.45	11.24
	8		827.0	40.7	23.62	31.824
	12		1510.0	62.1	43.15	63.108

The question arises: what bending stress can be carried by the flanged joint as limited by the bolting of B16.5 flanged joints? Noting that $M_{pb} = S_{pb} Z_p$, and using the approximation that $Z_p = (\pi/4) D_o^2 g_o$, Equation (5) becomes:

$$P_{FD} = P + \frac{4}{G}\left(\frac{D_o}{G}\right)^2 g_o S_{pb} \qquad (6)$$

Substituting P_{FD} from Equation (6) for P in the first term* of Equation (3) and solving for S_{pb} gives:

$$S_{pb} = \frac{S_m A_b - \left[(\pi/4) G^2 P + 2b\pi GmP\right]}{\pi D_o^2 g_o/G} \qquad (7)$$

Table 4 shows values of S_{pb} calculated by Equation (7). The calculated values of S_{pb} are based on S_m = 26,700 psi for SA-193 Grade B7 bolt material at 700 F. Two sets of values for S_{pb} are shown in Table 4:

(1) S'_{pb}, with the attached pipe wall thickness obtained by the equation: $g_o = \overline{P}D_o/17500 + 0.05$. This gives a pipe with about the same pressure rating as the flanges.

(2) S''_{pb} with the attached pipe wall thickness equal to the larger of std. wt. wall thickness or $\overline{P}D_o/17500 + 0.05$. This is significant, particularly for the 150 class, because most piping is std. wt. or heavier wall thickness.

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* Code Case 1677 states that in calculating $H_p = 2b\pi GmP$, the design pressure (not P_{FD}) shall be used.

TABLE 4. ALLOWABLE BENDING STRESS, S'_{pb} or S''_{pb}, IN PIPE ATTACHED TO FLANGED JOINTS, P = RATED PRESSURE AT 700 F, NB-3647.1 RULES BOLT STRESS CRITERIA

Class	Size	P, psi	g_o	S'_{pb}(psi) for g_o	S''_{pb}(psi) for larger of g_o or std. wt. wall
150	4	110	.0886	36400	13600
	8		.1239	16900	6500
	16		.1871	22950	11450
	24		.2557	23500	16100
300	4	470	.1271	26400	14200
	8		.1979	14900	9100
	16		.3243	22000	19000
	24		.4614	17300	17300
400	4	635	.1529	31000	20000
	8		.2471	15300	11700
	16		.4157	20300	20300
	24		.5986	19300	19300
600	4	940	.2043	16000	13800
	8		.3457	12700	12700
	16		.5986	14900	14900
	24		.8729	12900	12900
900	4	1410	.2814	24100	24100
	8		.4936	15100	15100
	16		.8729	9600	9600
	24		1.2843	13900	13900
1500	4	2350	.4357	13800	13800
	8		.7893	11500	11500
	16		1.4214	15000	15000
	24		2.1071	13650	13650
2500	4	3920	.6929	10550	10550
	8		1.2821	10250	10250
	12		1.8614	10400	10400

Pipe made of SA-106 Grade B material has an allowable stress intensity S_m = 16,800 psi at 700 F. The limit on bending stress due to moment loads in straight pipe is given by Equation (12) of NB-3653.6 as $3\ S_m$ = 50,400 psi. It can be seen in Table 4 that the calculated (by NB-3647.1 rules) maximums of S''_{pb} for the flanged joints ranges from 13 to 48 percent of that for straight pipe.

Bolting Requirements of NB-3230

NB-3647.1(b) states that the minimum bolt areas (as established by Equations (3) and (4) of XI-3220) "may have to be increased to meet requirements of NB-3230". The contents of NB-3230 is included herein as Table 2.

NB-3232.1 Average Stress

The maximum value of "service stress", averaged across the bolt cross-section and neglecting stress concentrations, must not exceed two times the tabulated allowable stress value. Service stress is defined as stresses "such as those produced by the combination of preload, pressure, and differential thermal expansion".

To examine the effect of this average stress limit on B16.5 flanged joints, it is necessary to determine the magnitude of preload used in tightening the bolts in B16.5 flanged joints. The preload average bolt stress, S_{b1}, should be such that when pressure, differential thermal expansion, and pipe loads are applied to the flanged joint in operation, the joint will not leak excessively.

It has been shown by Roberts [(4)] that the theoretical leakage* pressure of a flanged joint with a non-pressure-sealing** gasket is given by the axial forces equilibrium equation:

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* Leakage is defined as the gross type of leakage that occurs when the load on the gasket is reduced to zero. Slow, diffusion-type leakage may and generally will occur at lower pressures.

** The types of gaskets covered by Equation (8) includes all of those shown in Table XI-3221.1, except the "ring joint". The "ring joint" may have some pressure-sealing capability. The elastomeric O-ring has substantial pressure-sealing capability.

$$P_L = A_b S_{b2}/(\pi G_o^2/4) \tag{8}$$

where P_L = leakage pressure

S_{b2} = bolt stress after application of loads

G_o = gasket outside contact diameter

It is important to note that S_{b2} is the bolt stress after application of the loads; not S_{b1}, the initial bolt stress applied in preloading. Accordingly, it is necessary to establish a relationship between S_{b1} (which, in principle, is a controlled value) and S_{b2}, which is a function of S_{b1} and the applied loads, and the change in modulus of elasticity of the bolts, flanges, and gasket. The computer program FLANGE was used to determine the change in bolt stress. The results can be expressed by the following Equation (9).

$$S_{b1} - S_{b2} = S_{b1}(1 - E_2/E_1) + a_1 P'_{eq}/\overline{P} + a_2 P/\overline{P} + a_3 \frac{E_2}{E_1}(T_b - T_f)/50 + a_4 (T_{hp} - T_f)/100 \tag{9}$$

where a_1, a_2, a_3, a_4 = coefficients given in Table 5 and 6

S_{b1} = initial bolt stress

S_{b2} = bolt stress after application of loads

E_1 (E_2) = modulus of elasticity of flanges, bolts and gaskets at initial bolt up temperature (operating temperature)

P'_{eq} = pressure equivalent of bending moment applied by attached pipe, = $4S_{pb}g_o/D_o$

S_{pb} = maximum stress in attached pipe due to bending moment

g_o = attached pipe wall thickness

$\overline{P}$ = primary rating pressure (class designation); e.g., for 300 class, P = 300 psi

P = internal pressure

T_b = temperature of bolts

T_f = temperature of flange ring
T_{hp} = temperature of flange hub and pipe.

The values of a_1 and a_2 are given in Table 5; values of a_3 and a_4 are given in Table 6. The values of a_1, a_2, a_3, and a_4 serve as the equivalent of stress indices in Equation (9).

With Equation (9) and the a-values shown in Tables 5 and 6, the value of S_{b2} can be obtained for any combination and magnitude of the loads. In order to arrive at an estimate of a suitable preload average bolt stress, we assume the set of conditions shown in Table 7. The conditions shown in Table 7 are more-or-less representative of conditions that are applied to B16.5 flanged joints, although items (c) and (d) would seldom occur at the same time.

The combination of loads shown in Table 7 include a bending moment applied by the attached pipe to the flanged joint. The axial forces equilibrium Equation (8) must be extended to include the axial force on the tension side of the applied moment. An accurate way to include this force is not known. For the present purpose, we make the conservative assumption that the maximum tensile stress due to the pipe-applied moment (which exists only at one point on the pipe circumference) acts around the complete circumference of the pipe. Accordingly, the axial load which must be added to the axial pressure load is $\pi D_m g_o S_{pb}$, where D_m is the mean diameter of attached pipe, g_o is the attached pipe wall thickness and S_{pb} is the bending stress in the attached pipe. The resulting equation for the critical bolt stress S_{bc}, corresponding to the theoretical leakage pressure is:

$$S_{bc} = \left(\pi G_o^2 P/4 + \pi D_m g_o S_{pb}\right) /A_b \tag{10}$$

TABLE 5. FACTORS a_1 AND a_2 FOR USE IN EQUATION (9)

Class		a_1			a_2		
$\overline{P}$	Size	WN	SH	Blind	WN	SH	Blind
150	4	500	400	200	600	500	700
	8	1,100	900	600	1,700	1,300	2,400
	16	1,100	800	700	1,700	1,200	4,200
	24	1,500	1,100	900	2,500	1,800	5,500
300	4	400	300	200	600	500	700
	8	800	600	500	1,400	1,000	1,800
	16	800	500	500	1,500	1,000	2,800
	24	1,100	800	800	2,100	1,500	4,200
400	4	1,100	800	800	2,100	1,500	4,200
	8	400	1,100	900	2,600	1,900	3,200
	16	900	600	600	1,600	1,100	2,800
	24	1,100	800	800	2,000	1,400	4,000
600	4	500	400	300	700	500	900
	8	800	500	400	1,200	800	1,600
	16	900	600	600	1,700	1,100	2,800
	24	1,300	900	900	2,300	1,500	3,700
900	4	500	300	300	600	400	700
	8	700	400	400	900	500	1,500
	16	1,100	700	700	1,700	1,000	3,000
	24	900	600	600	1,600	1,000	2,600
1500	4	500	400	300	600	400	700
	8	600	400	300	700	400	900
	16	600	400	400	800	400	1,300
	24	700	500	500	1,100	700	1,600
2500	4	400	300	200	300	200	400
	8	400	400	200	300	200	500
	12	400	300	200	300	200	500

WN = Welding neck flange mated to welding neck flange
SH = Straight hub flange mated to welding neck flange
Blind = Blind flange mated to welding neck flange

TABLE 6. FACTORS a_3 AND a_4 FOR USE IN EQUATION (9)

Class	Size	a_3			a_4		
		WN	SH	Blind	WN	SH	Blind
150	4	3,400	3,500	3,400	6,400	6,700	3,300
	8	4,100	4,300	4,200	10,900	11,100	5,600
	16	1,700	1,900	2,100	8,700	8,900	5,400
	24	1,800	2,100	2,300	8,700	9,200	5,400
300	4	4,400	4,600	4,700	5,900	6,000	3,100
	8	3,600	3,900	4,100	8,000	8,200	4,500
	16	2,000	2,300	2,700	6,900	7,200	4,600
	24	1,600	2,000	2,200	6,800	7,200	4,700
400	4	4,500	4,600	4,700	4,900	5,000	2,600
	8	4,000	4,300	4,500	6,900	7,100	3,900
	16	2,200	2,600	2,900	6,300	6,600	4,100
	24	1,600	1,900	2,200	5,600	6,000	3,900
600	4	4,500	4,600	4,900	4,900	4,900	2,700
	8	4,200	4,400	4,800	6,200	6,300	3,600
	16	2,500	2,800	3,300	6,000	6,100	4,000
	24	2,400	2,900	3,300	5,900	6,400	4,000
900	4	3,800	3,800	4,400	3,400	3,500	2,000
	8	3,200	3,400	4,100	4,600	4,600	2,900
	16	3,200	3,400	4,100	5,000	4,800	3,800
	24	3,200	3,500	4,300	5,300	5,500	3,600
1500	4	4,600	4,700	5,300	2,700	2,800	1,600
	8	5,300	5,400	6,100	3,500	3,600	2,000
	16	4,100	4,300	5,300	3,800	3,800	2,500
	24	4,200	4,300	5,400	,900	4,000	2,500
2500	4	5,600	5,600	6,300	1,700	1,700	900
	8	6,000	6,100	6,800	2,100	2,200	1,200
	12	6,300	6,300	7,100	2,300	2,300	1,300

WN = Welding neck flange mated to welding neck flange
SH = Straight hub flange mated to welding neck flange
Blind = Blind flange mated to welding neck flange.

TABLE 7. REPRESENTATIVE SET OF LOADINGS USED FOR ESTIMATING A SUITABLE PRELOAD AVERAGE BOLT STRESS

(a) Internal pressure equal to the primary rating pressure; $P = \overline{P}$

(b) Decrease in modulus of elasticity from 30,000,000 to 23,000,000 psi, $E_2/E_1 = 0.767$

(c) Bolt temperature 50 F higher than flange ring temperature, $(T_b - T_f) = 50$ F

(d) Pipe and hub temperature 100 F higher than flange ring temperature, $(T_{hp} - T_f) = 100$ F

(e) Pipe moment load that produces a bending stress, S_{pb}, of 8,750 psi in the attached pipe.

Values of S_{bc}, calculated by Equation (10) with $P = \overline{P}$ and $S_{pb} = 8{,}750$ psi, are shown in Table 8. The initial bolt stress, S_{b1}, must be high enough so that S_{b2} is equal to S_{bc}. Equation (9) gives:

$$S_{b2} = S_{b1}(E_2/E_1) - a_1 P'_{eq}/\overline{P} - a_2\ P/\overline{P} - a_3(E_2/E_1)(T_b - T_f)/50$$

$$-a_4(T_{hp} - T_f)/100 \tag{11}$$

Equating S_{b2} to S_{bc} and solving for S_{b1} gives:

$$S_{b1} = \frac{S_{bc} + a_1 P'_{eq}/P + a_2\ P/\overline{P} + a_3(E_2/E_1)(T_b - T_f)/50 + a_4(T_{hp} - T_f)/100}{E_2/E_1} \tag{12}$$

For the conditions assumed in Table 7, $P'_{eq}/\overline{P} = 2.243$, $P/\overline{P} = 1.000$, $E_2/E_1 = 0.767$, $(T_b - T_f)/50 = 1.000$ and $(T_{hp} - T_f)/100 = 1.000$. Accordingly:

$$S_{b1} = \frac{S_{bc} + 2.243a_1 + a_2 + 0.767a_3 + a_4}{0.767} \tag{13}$$

Values of a_1, a_2, a_3, and a_4 are given in Tables 5 and 6. Using these values for welding neck-to-welding neck joints (WN) in Equation (13) along with the values of S_{bc}, gives the values of $(S_{b1})_{min}$ shown in Table 8.

TABLE 8. INITIAL BOLT STRESSES

Class	Size	S_{bc} (1)	S_{b1} (2) Minimum	S_b (3) ASME	$(S_{b1})_p$ (4) Petrie
150	4	9400	26300	12900	56900
	8	17500	46600	16900	52000
	16	13800	36300	9200	45000
	24	13700	38600	7400	40300
300	4	10100	27200	8700	52000
	8	14400	37000	8200	48100
	16	11900	30800	5900	40300
	24	14000	34800	6700	36700
400	4	9000	28600	6600	48100
	8	14000	35800	8300	45000
	16	12400	31300	6400	38500
	24	13000	31700	6300	34100
600	4	12600	29700	9900	48100
	8	15100	35900	9500	42500
	16	14800	34400	7800	36700
	24	16400	38200	8100	32900
900	4	10300	23800	8500	42500
	8	13700	30300	8900	38500
	16	18000	38600	9800	35300
	24	15400	35000	7900	28500
1500	4	12600	26800	11100	40300
	8	15000	32100	10300	35300
	16	14200	30300	8900	28500
	24	15300	32700	8000	24100
2500	4	13100	26500	12300	36700
	8	15100	30000	10800	31800
	12	15900	31500	10000	27200

(1) S_{bc} = critical bolt stress, Equation (10).
(2) S_{b1} = minimum preload bolt stress calculated by Equation (13).
(3) S_b = larger of W_{m1}/A_b or W_{m2}/A_b, where W_m, and W_{m2} are calculated by Equations (3) and (4).
(4) $(S_{b1})_p = 45000/\sqrt{d}$, d = bolt diameter in inches.

Table 8 also shows for comparison, the minimum values of S_b as obtained by Equations (3) and (4). Obviously, these values used as initial bolt stresses are entirely inadequate. The last column in Table 8, headed $(S_{b1})_p$ is of interest in that the initial bolt stress used in assembling B16.5 flanged joints is seldom controlled; the pipe fitter simply tightens the bolts to what he considers to be an appropriate amount. (He may further tighten the bolts if a leak is found in a hydrostatic test or subsequent operation). Petrie[5] indicates that the "appropriate amount" normally applied by a pipe fitter is approximately given by the equation:

$$(S_{b1})_p = \frac{45,000}{\sqrt{d}} \text{ psi} \tag{14}$$

where d = bolt diameter, inches. For most joints, $(S_{b1})_p$ is adequate to prevent leakage as judged by comparison with $(S_{b1})_{min}$ in Table 8.

Table 8 is <u>not</u> intended to give specific recommended values of the initial bolt stresses that should be used in assembly of B16.5 flanged joints. Rather, it is intended to show that an initial bolt stress of about 40,000 psi is usually needed and applied. Based on an initial bolt stress of 40,000 psi, we may then answer the question:

> What bolting materials can be used in B16.5 flanged joints so that the criterion of NB-3232.1, that the average bolt service stress shall be less than two times S_m, is met?

In so far as bolt-up conditions are concerned, the answer is obtained from Table I-1.3, "Design Stress Intensity Values, S_m, for Bolting Materials for Class 1 Components". The value of S_m for almost all bolt materials listed are greater than 20,000 psi at 100 F; hence, an average bolt service stress of 40,000 psi is acceptable. Exceptions include the annealed austenitic stainless steel materials such as SA-193 Grade B8.

The answer to the question for operating conditions is more complex and one must know the time history of P'_{eq}, P, T_b, T_{hp} and T_f. However, knowing that history, the answer can be obtained* from Equation (9). One notes that P'_{eq} and P will produce a decrease in bolt stress; this may be such that the average bolt stress at operating temperature is less than 2 S_m at the operating temperature. However, the quantities $(T_b - T_f)$ and $(T_{hp} - T_f)$ can be either positive or negative depending upon whether the pipeline is being heated up or cooled down. As an example of how average bolt stresses could increase, let us consider the case where the flowing fluid is rapidly decreasing in temperature, and that $(T_b - T_f) = 40$ F and $(T_{hp} - T_f) = -120$ F, with $P'_{eq} = P = 0$ and $E_2/E_1 = 1.00$. The flanged joint consists of a pair of 24" - 300 class flanges. From Equation (9) and Table 6:

$$S_{b1} - S_{b2} = 1{,}600\text{x}\ \frac{40}{50} + 6{,}800\text{x}\ \frac{-120}{100} = -6{,}880 \text{ psi.}$$

If $S_{b1} = 40{,}000$, then $S_{b2} = 40{,}000 + 6{,}880 = 46{,}880$ psi.

If the bolt material were SA-193 Grade B7 (S_m = 26,700 psi at 700 F) the average bolt service stress could still meet the criterion of $S_{b2} < 2\ S_m$; i.e., 46,880 is less than 2 x 26,700.

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* The a-values shown in Table 5 and 6 are for a sampling of B16.5 flanged joints. However, the computer program FLANGE will give similar information for any flanged joint of geometrical shape covered by the program.

Now that we have established the point that $S_{b1} = 40,000$ is a representative initial bolt stress for B16.5 flanged joints, it is of interest to use Equations (9) and (10) to determine the attached pipe bending moment capacity of the joints when the bolts are initially tightened to 40,000 psi and when the assumed loadings are the same as those in Equation (5); i.e., pressure and pipe moment, M_{pb}. For these loadings, Equation (11) becomes:

$$S_{b2} = 40,000 - a_1 P'_{eq}/\overline{P} - a_2 P/\overline{P} \tag{15}$$

Noting that $P'_{eq} = 4\, S_{pb} g_o/D_o$, and that leakage occurs when $S_{b2} = S_{bc}$ as defined by Equation (10), we obtain the equation:

$$(\pi G_o^2\ P/4 + \pi D_m g_o S_{pb})/A_b = \\ 40,000 - a_1\ (4S_{pb} g_o D_o)/\overline{P} - a_2 P/\overline{P} \tag{16}$$

Solving Equation (16) for S_{pb} gives:

$$S_{pb} = \frac{40,000 - (\pi/4)\ G_o^2\ P/A_b - a_2 P/\overline{P}}{g_o(\pi D_m/A_b + 4\ a_1/D_o\overline{P})} \tag{17}$$

For comparison with S_{pb} as obtained by Equation (7), we use the same pressure; i.e., the rated pressure for carbon steel flange material at 700 F. The results are shown in Table 9.

It can be observed in Table 9 that for the 150 class flanges, the two methods of calculating allowable pipe bending stresses in attached pipe give about the same answer. This is deemed to be purely fortuitous. The two methods are entirely different, as can be seen by comparing Equation (7) with Equation (17). For the higher pressure

TABLE 9. ALLOWABLE STRESSES, S'_{pb} and S''_{pb}, IN PIPE ATTACHED[1] TO FLANGED JOINTS, P = RATED PRESSURE AT 700 F. BOLT STRESS CRITERIA (NB-3647.1 DATA ARE FROM TABLE 4)

Class	Size	P, psi	g_o	S'_{pb} FLANGE Eq. 7	S'_{pb} NB-3647.1 Eq. 17	S''_{pb} FLANGE Eq. 7	S''_{pb} NB-3647.1 Eq. 17
150	4	110	.0886	36700	36400	13700	13600
	8		.1239	19400	16900	7500	6500
	16		.1871	25300	22950	12600	11450
	24		.2557	23900	23500	16300	16100
300	4	470	.1271	38000	26400	20400	14200
	8		.1979	22900	14900	14100	9100
	16		.3243	28700	22000	24800	19000
	24		.4614	22100	17300	22100	17300
400	4	635	.1529	31100	31000	20100	20000
	8		.2471	24900	15300	19100	11700
	16		.4157	27000	20300	27000	20300
	24		.5986	24200	19300	24200	19300
600	4	940	.2043	31300	16000	27000	13800
	8		.3457	23000	12700	23000	12700
	16		.5986	22600	14900	22600	14900
	24		.8729	18100	12900	18100	12900
900	4	1410	.2814	40800	24100	40800	24100
	8		.4936	27200	15100	27200	15100
	16		.8729	17400	9600	17400	9600
	24		1.2843	21800	13900	21800	13900
1500	4	2350	.4357	33700	13800	33700	13800
	8		.7893	25700	11500	25700	11500
	16		1.4214	27100	15000	27100	15000
	24		2.1071	23800	13650	23800	13650
2500	4	39[illegible]	.6929	34700	10550	34700	10550
	8		1.2821	27900	10250	27900	10250
	12		1.8714	25800	10400	25800	10400

(1) S'_{pb} for pipe of wall thickness g_o

S''_{pb} for pipe of wall thickness equal to the larger of g_o or std. wt.

classes, Equation (17), which is used in program FLANGE, predicts significantly higher allowable pipe bending stresses than Equation (7); up to 3.3 times as much for the 4" - 2,500 lb flanged joint. However, Equation (17) does not include thermal gradient effects; these, if included, would decrease the values of allowable pipe bending stress and the results from Equation (7) might not be overconservative.

NB-3232.2 Maximum Stress

The maximum value of service stress at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations must not exceed three times the tabulated allowable stress intensity values. Stress intensity, rather than maximum stress, shall be limited to these values when the bolts are tightened by methods other than heaters, stretchers, or other means that minimize residual torsion.

To examine the effect of this maximum stress intensity limit on B16.5 flanged joints, it is necessary to establish

(a) The magnitude of the bending stresses in the bolts

(b) The stress intensity in the bolts - since B16.5 flanges are usually tightened with a wrench that does produce residual torsion in the bolts.

An engineering evaluation of the bending stresses in the bolts can be obtained by assuming that the nuts on the bolt remain parallel with the flange ring, as illustrated by Figure 1. The bending stress in the bolts, S_{bb}, is given by:

$$S_{bb} = \frac{Ed}{2\ell} (\theta_1 + \theta_2) \qquad (18)$$

where E = modulus of elasticity of bolt material

d = bolt diameter

ℓ = effective bolt length

θ_1 = rotation of flange ring of one flange

θ_2 = rotation of flange ring of the other flange.

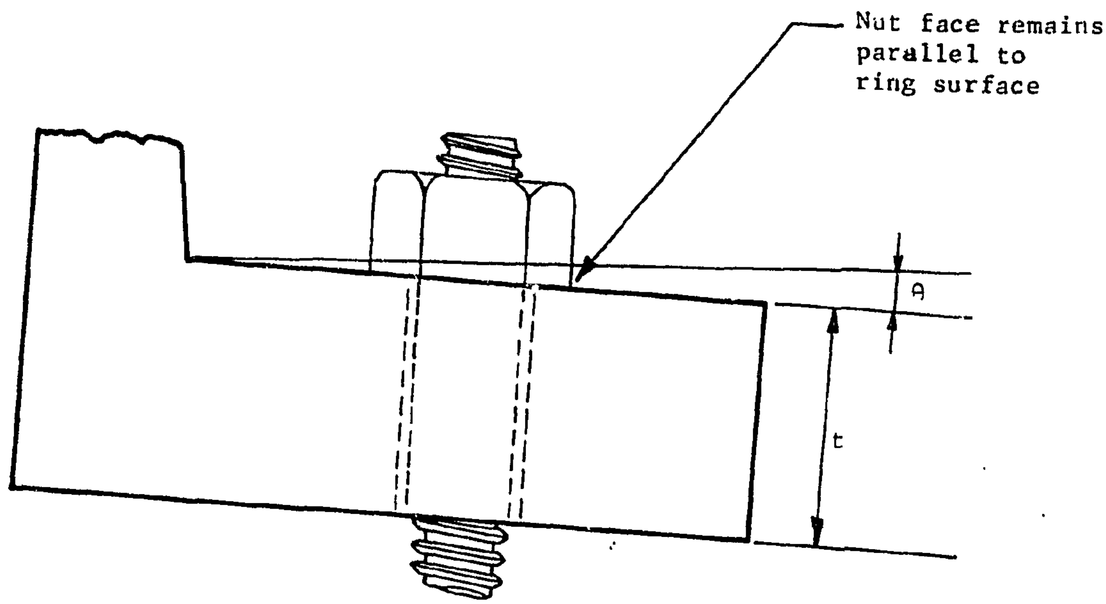

FIGURE 1 . ASSUMPTION USED IN CALCULATING BOLT BENDING STRESS

Values of θ, calculated using the computer program FLANGE, are shown in Table 10. As an example of their use, consider an 8" - 900 class flanged joint, welding neck to straight hub. For an initial bolt stress of 40,000 psi, $M = S_b A_b h_G = 40,000 \times 13.86 \times (15.5 - 8.625)/2 = 1,905,750$ in. lb. For this joint, d = 1.375", ℓ = 6.5", and taking E as 3×10^7 and using values of θ_M from Table 10, we obtain

$$S_{bb} = \frac{3 \times 10^7 \times 1.375}{2 \times 6.5} (1.554 + 1.078) \times 10^{-3} = 8,350 \text{ psi.}$$

Table 11 shows bolt bending stresses calculated by equation (18) for all of the flanged joints covered by this report.

An adequate estimate of the torsional stress in the bolts can be obtained as follows. For well-lubricated bolts and nuts of the type used with B16.5 flanged joints, the relationship between tightening torque and axial force in the bolt is:

$$T = kdF_b = k\ d\ S_{b1} A_b' \tag{19}$$

where

T	=	torque, in.-lb
k	=	~0.2
d	=	bolt diameter, inch
F_b	=	axial force in bolt, lb, = $S_{b1}A_b'$
S_{b1}	=	initial (preload) bolt stress
A_b'	=	bolt area of a single bolt.

TABLE 10. ROTATIONS(1) OF B16.5 FLANGES

		Welding Neck			Straight Hub			Blind	
		θ_M	θ_p	θ_T	θ_M	θ_p	θ_T	θ_M	θ_p
Class	Size	$\times 10^3$	$\times 10^5$	$\times 10^3$	$\times 10^3$	$\times 10^5$	$\times 10^3$	$\times 10^3$	$\times 10^5$
150	4	2.104	2.434	0.910	1.704	1.365	0.903	2.868	8.439
	8	1.898	6.407	1.342	1.430	3.038	1.265	2.622	32.25
	16	4.454	13.66	1.746	2.484	3.123	1.383	8.910	90.15
	24	6.038	22.54	2.054	3.202	5.318	1.621	8.568	141.8
300	4	1.298	2.381	0.726	1.000	0.780	0.710	2.028	6.724
	8	2.012	6.919	1.020	1.385	2.056	0.949	2.780	19.81
	16	4.098	13.73	1.336	1.982	1.988	1.047	6.171	44.50
	24	5.872	22.73	1.609	2.288	3.173	1.177	7.633	82.82
400	4	1.457	2.629	0.665	1.137	0.827	0.659	2.114	6.736
	8	1.963	7.189	0.904	1.389	2.173	0.866	2.380	17.20
	16	4.029	14.55	1.225	2.020	2.146	0.996	5.596	43.24
	24	6.543	23.78	1.480	2.666	3.439	1.122	8.285	85.05
600	4	1.110	2.151	0.610	0.882	0.365	0.597	1.972	7.395
	8	1.579	6.280	0.785	1.096	1.215	0.739	2.316	15.33
	16	3.128	12.55	1.028	1.680	1.491	0.862	4.751	36.00
	24	4.227	19.36	1.161	1.781	2.277	0.947	4.648	52.69
900	4	1.340	1.453	0.524	1.079	0.304	0.513	2.602	6.625
	8	1.554	3.917	0.674	1.078	0.403	0.610	3.344	14.18
	16	2.109	9.073	0.863	1.284	0.205	0.741	3.360	29.45
	24	2.586	11.19	0.855	1.424	0.773	0.751	3.540	28.39
1500	4	0.963	0.611	0.420	0.781	0.013	0.416	1.925	5.981
	8	0.924	1.415	0.469	0.749	0.011	0.464	1.577	7.673
	16	1.384	3.164	0.548	0.936	0.016	0.509	2.455	11.27
	24	1.674	5.298	0.578	1.146	0.499	0.557	2.415	14.23
2500	4	0.582	0.018	0.277	0.531	0.024	0.280	1.282	3.251
	8	0.567	0.018	0.317	0.509	0.026	0.319	1.167	4.483
	12	0.550	0.017	0.324	0.497	0.026	0.326	1.064	4.501

(1) θ_M = rotation due to moment applied to flange ring of 40,000 $A_b h_G$, radians

θ_p = rotation due to pressure equal to $\overline{P}$, radians

θ_T = rotation due to thermal gradient $(T_{hp} - T_f)$ = 100 F, radians

TABLE 11. BOLT BENDING STRESSES,(1) S_{bb}, AT INITIAL BOLT STRESS OF 40,000 PSI

Class	Size	d, in.	ℓ, in.	Ed/ℓ psi x 10^{-6}	Bolt Bending Stresses (psi) WN-WN(2)	WN-SH(2)	WN-Blind(2)
150	4	0.625	2.625	7.143	15,030	13,600	17,760
	8	0.750	3.125	7.200	13,660	11,980	16,270
	16	1.000	4.000	7.500	33,400	26,020	50,110
	24	1.25	5.125	7.317	44,180	33,800	53,440
300	4	0.75	3.375	6.667	8,650	7,660	11,090
	8	0.875	4.250	6.176	12,430	10,490	14,800
	16	1.25	5.875	6.383	26,160	19,400	32,770
	24	1.50	7.125	6.316	37,090	25,770	42,650
400	4	0.875	3.750	7.000	10,200	9,080	12,500
	8	1.000	4.875	6.154	12,080	10,310	13,360
	16	1.375	6.500	6.346	25,570	19,190	30,540
	24	1.75	7.875	6.667	43,620	30,700	49,430
600	4	0.875	4.000	6.562	7,280	6,540	10,110
	8	1.125	5.625	6.000	9,470	8,030	11,690
	16	1.5	7.625	5.902	18,460	14,190	23,250
	24	1.875	10.000	5.625	23,780	16,900	24,960
900	4	1.125	4.750	7.105	9,520	8,590	14,000
	8	1.375	6.500	6.346	9,860	8,350	15,540
	16	1.625	8.750	5.571	11,750	9,450	15,230
	24	2.5	13.625	5.505	14,230	11,040	16,860
1500	4	1.25	5.625	6.667	6,420	5,810	9,630
	8	1.625	9.000	5.417	5,000	4,530	6,770
	16	2.5	14.125	5.310	7,350	5,890	10,190
	24	3.5	19.625	5.350	8,960	7,540	10,940
2500	4	1.5	7.625	5.902	3,430	3,280	5,500
	8	2.000	12.125	4.948	2,810	2,660	4,290
	12	2.75	17.375	4.748	2,610	2,490	3,830

(1) Stresses calculated by equation (18) using values of θ_1 and θ_2 from Table 10, and with $M = S_b A_b h_G$; P and $(T_{hp} - T_f)$ equal to zero.

(2) WN - WN = welding neck mated to welding neck flange
WN - SH = welding neck mated to straight hub flange
WN - Blind = welding neck mated to blind flange.

The constant k represents frictional resistance between threads plus frictional resistance between nut and flange. Approximately one-half of the torque is resisted by thread friction; the other half by nut/flange friction.[6] Accordingly, the bolt is subjected to a torsional load of $T_s = T/2$. The shear stress in the bolt, S_{b2}, is then:

$$S_{b2} = \frac{1}{2} \, kdS_{b1} \times \frac{\pi d^2}{4} \times \frac{16}{\pi d^3} = 2 \, k \, S_{b1} \qquad (20)$$

The stress intensity, including the bending stress, is then given by:

$$\bar{S}_b = \left[(S_{b1} + S_{bb})^2 + 4 \times (2kS_{b1})^2 \right]^{1/2} \qquad (21)$$

According to NB-3232.2, the value of $\bar{S}_b$ must be less than $3S_m$. From equating $\bar{S}_b$ to $3S_m$ we obtain:

$$\frac{S_{bb}}{S_{b1}} = \left[(3S_m/S_{b1})^2 - 16k^2 \right]^{1/2} - 1 \qquad (22)$$

For SA-193 Grade B7, bolt material at 100 F, S_m = 35,000 for $d \leq 2.5''$; S_m = 31,600 psi for $d > 2.5''$. With an average initial bolt stress S_{b1}=40,000 psi, Equation (22) gives:

(S_{bb}/S_{b1}) maximum = 1.500 for $d \leq 2.5''$

(S_{bb}/S_{b1}) maximum = 1.231 for $d \geq 2.5''$

Using the results of Table 11 (i.e., S_{bb}), it is apparent that for an initial bolt stress of S_{b1} = 40,000 psi, the ratios of S_{bb}/S_{b1} are well below the ratios permitted by NB-3232.2 and that NB-3232.2 is met for preload bolting conditions.

To check whether the NB-3232.2 criterion that the maximum bolt stress must be less than $3S_m$ is met for operating conditions requires a knowledge of the time history of P'_{eq}, P, T_b, T_{hp}, and T_f. Once these four quantities are known, the average bolt stress S_b can be obtained by Equation (11), the bolt bending stress S_{bb} can be obtained by Equation 11 and Equation (18), and the shear stress S_{b2} can be obtained by Equation (20). These three values can then be used in Equation (21) to determine whether the maximum bolt stress is less than $3S_m$, as required by NB-3232.2.

As an example, consider an 8" - 900 class flanged joint, welding neck-to-straight hub. The initial bolt stress is 40,000 psi. The assumed operating condition to be checked is:

$$\begin{aligned}
P &= 1{,}410 \text{ psi} \\
\overline{P} &= 900 \text{ psi} \\
P_{eq} &= 2{,}700 \text{ psi} \\
T_b &= 600 \text{ F } (S_m \text{ for bolts} = 28{,}400 \text{ psi}) \\
T_{hp} &= 700 \text{ F} \\
T_f &= 650 \text{ F} \\
E_2/E_1 &= 0.90
\end{aligned}$$

Equation (11) and the a-values from Tables 5 and 6 give:

$$\begin{aligned}
S_{b2} &= 40{,}000 \times 0.90 - 400 \times \frac{2700}{900} - 500 \times \frac{1410}{900} - 3400 \times 0.90 \times \frac{-50}{50} \\
&\quad - 4600 \times \frac{50}{100} \\
&= 36000 - 1200 - 783 + 3060 - 2300 \\
&= 34{,}780 \text{ psi}
\end{aligned}$$

We note that the average stress of $S_{b2} = 34{,}780$ psi meets NB-3232.1; i.e., 34,780 is less than $2\ S_m = 2 \times 28{,}400 = 56{,}800$ psi. Using the values from Table 10 for the assumed 8" - 900 class welding neck-to-straight hub flanged joint and the assumed operating conditions:

$$\begin{aligned}
\theta_M &= (1.554 + 1.078) \times 10^{-3} \text{ at } S_{b1} = 40{,}000 \text{ psi} \\
&= 2.632 \times 10^{-3} \times \frac{34780}{40000} = 2.289 \times 10^{-3} \text{ at } S_{b2} = 34{,}780 \text{ psi}^*
\end{aligned}$$

$$\theta_p = (3.917 + 0.403) \times 10^{-5} \times \frac{1410}{900} = 6.768 \times 10^{-5}$$

$$\theta_T = (0.674 + 0.610 \times 10^{-3} \times \frac{50}{100} = 0.642 \times 10^{-3}$$

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* The procedure used here is approximately correct, but, more precisely, θ_M is proportional to the flange moment rather than bolt load. The computer program FLANGE gives the change in moment so that the more precise results can be used.

Accordingly, the flange rotation at the operating conditions is 2.999×10^{-3} radians.

The flange bending stress from equation (18) with $E = 3 \times 10^7$ psi, $d = 1.375''$ and $\ell = 6.5''$ is

$$S_{bb} = \frac{Ed}{2\ell} (\theta) = \frac{6.346 \times 10^6}{2} \times 2.999 \times 10^{-3} = 9{,}520 \text{ psi}$$

From Equation (20), the initial shear stress is $2 \times 0.2 \times 40{,}000 =$ 16,000 psi. This shear stress will decrease by the ratio E_2/E_1; assumed for this example to be 0.90. Accordingly, the maximum stress intensity at the assumed operating conditions is:

Average bolt stress = 34,780 psi

Bending bolt stress = 9,520 psi

Shear bolt stress = 14,400 psi

$$\text{Maximum stress intensity} = \left[(44{,}300)^2 + 4 \times 14{,}400^2\right]^{1/2}$$

$$= 52{,}800 \text{ psi.}$$

Because 53,090 psi is less than $3S_m$ at 600 F (85,200 psi in this example) the criterion of NB-3232.2 is satisfied.

NB-3232.3 Fatigue Analysis of Bolts

NB-3232.3 requires* that changes in bolt stresses due to the postulated history of operating conditions must be determined and evaluated. In the following discussion of fatigue analysis, a "fatigue strength reduction factor" of 4.0 will be used to account for the "notch" at the root of the bolt threads.

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* The fatigue analysis is not required if the conditions of NB-3222.4(d) are met. It is difficult for the authors to see the relevance of NB-3222.4(d) to bolted flanged joints. First, the bolt tightening /untightening cycle is completely ignored; this is probably the major significant cycle involved. Second, the definition of "adjacent points" does not seem to have any meaning because a major temperature gradient to be considered is that between the flanges and bolts.

One of the postulated cycles may consist of bolting and unbolting the flanged joint a postulated number of times during the service life. If the average initial bolt stress is 40,000 psi, obtained by tightening with a wrench, then the peak stress range, S_p, due to tightening/untightening, is obtained from equation (21) using k = 0.2, S_{bl} = 40,000. Equation (21), multiplied by 4.0 for the notch at the root of the bolt threads, gives:

$$S_p = \left[(40{,}000 + S_{bb})^2 + 1.024 \times 10^9\right]^{1/2} \times 4.0 \qquad (23)$$

Values of S_{bb} are given in Table 11 for an initial bolt load of 40,000 psi. Using these in Equation (23) gives the peak stress ranges shown in Table 12. From Code Figure I-9.4, with $S_a' = S_p/2$ and using the curve labeled "Max. nominal stress = $3.0S_m$", we obtain the permissible cycles, N_D, shown in Table 12. It can be seen in Table 12 that the number of tightening/untightening cycles, N_D, is large even for the worst flange (225 cycles for the 24" - 400 class, welding neck-to-blind flanged joint). The values of N_D shown in Table 12 are far in excess of the number of tightening/untightening cycles normally encountered in B16.5 flanged joints.

In operation, cyclic bolt stresses would be superimposed on the initial bolt stresses. The complete fatigue analysis would require that the history of P_{eq}', P, T_b, T_{np} and T_f be known. Knowing that history, the bolt stress variations can be obtained from the data presented herein, for the flanged joints covered herein, or more generally, by use of the computer program FLANGE.

The fatigue analysis procedure can be illustrated by continuing the example of the 8" - 900 class flanged joint and the assumed set of operating conditions. As developed previously, the operating conditions give a maximum stress intensity in the bolts of 53,090 psi. The peak stress intensity is 4 x 53090 = 212,360 psi. The peak stress intensity at the 40,000 psi preload condition is 232,000 psi (see Table 12). Assuming cycles between the operating conditions and preload

TABLE 12. BOLT FATIGUE ANALYSIS, STRESS RANGE AND ALLOWABLE DESIGN CYCLES, N_D, FOR BOLT TIGHTENING/UNTIGHTENING CYCLE

		WN-WN		WN-SH		WN-Blind	
Class	Size	Stress, ksi(1)	Cycles, N_D(2)	Stress, ksi(1)	Cycles, N_D(2)	Stress, ksi(1)	Cycles, N_D(2)
150	4	255	455	250	471	264	428
	8	250	472	244	492	259	443
	16	320	305	293	356	293	356
	24	360	247	322	301	395	210
300	4	233	534	230	547	241	503
	8	246	485	239	511	254	458
	16	294	354	270	411	318	308
	24	334	282	293	356	355	253
400	4	238	514	234	530	246	485
	8	245	489	238	514	249	475
	16	292	358	269	414	310	322
	24	358	250	310	322	380	225
600	4	228	555	226	564	238	514
	8	236	522	231	542	243	496
	16	267	420	252	465	284	376
	24	285	374	261	437	290	362
900	4	236	522	233	534	251	468
	8	237	518	232	538	256	452
	16	243	496	236	522	255	455
	24	252	465	241	503	261	437
1500	4	226	564	224	573	236	522
	8	221	587	219	596	227	559
	16	229	551	224	573	238	514
	24	234	530	229	551	241	503
2500	4	216	611	215	616	223	577
	8	214	621	213	626	219	596
	12	213	626	213	626	217	606

(1) Stress Range from Equation (23)
(2) Allowable Design Cycles from Code Figure I-9.4, Curve labeled "Max. nominal stress = $3.0\ S_m$

conditions then gives a peak stress intensity range for those cycles of 232,000 - 212,000 = 20,000 psi. Entering Code Figure I-9.4, with S_a' = 20,000/2 we find the design cycles to be 55,000. Recalling that the example conditions correspond to going from shutdown to operating and back to shutdown conditions, it is apparent that exceeding the Code fatigue limitations for any credible operating history is unlikely. This result was probably obvious to many readers who recognize that in a flanged joint with adequately high preload bolt stresses, the variations in bolt stresses due to subsequently applied loadings are relatively small.

Summary of Results on Bolt Areas and Bolt Stresses

Calculations were made on a representative sampling of B16.5 flanged joints with:

(a) Rated pressures of ANSI B16.5 - 1968 for carbon steel flange material

(b) Gasket O.D. = raised face O.D.
Gasket I.D. = pipe O.D.
m = 2.75, y = 3,700

(c) Bolt material: SA-193 Grade B7

From the results of these calculations, the following conclusions can be drawn.

(1) The bolting area is sufficient to meet the requirement of NB-3647.1(b) which states that "Equations (3) and (4) of XI-3220 shall be used to establish minimum bolt area...".

(2) The limitation on bending stresses imposed by the use of P_{eq} is such that the allowable pipe bending stress in attached pipe ranges from 13 to 48 percent (average of 28 percent) of that permitted by Equation (12) in NB-3653.6. The values cited are specifically for SA106 Grade B pipe at 700 F; Table 4, S''_{pb}, values calculated according to the rules of NB-3647.1.

(3) The limitation on bending stresses by a seemingly more rational analysis is such that the allowable pipe bending stress in attached pipe ranges from 15 to 81 percent (average of 45 percent) of that permitted by Equation (12) in NB-3653.6. The values cited are specifically for SA 106 Grade B pipe at 700 F, Table 9, S''_{pb}, values calculated using program FLANGE.

(4) Calculations indicate that a preload average bolt stress of around 40,000 psi is appropriate for B16.5 flanged joints. This initial bolt stress is permitted by the Code.

(5) With a preload average bolt stress of 40,000 psi, the Code limit of $3S_m$ for combined average, bending and torsional stress is met.

(6) With a preload average bolt stress of 40,000 psi, the fatigue limits of the Code will be satisfied for any ordinarily anticipated loading history.

5. FLANGE STRESSES

Flange Stress Limits of NB-3647

Flange stresses are to be calculated in accordance with Article XI-3000 of the Code, with the following modifications.

(a) The "design pressure", P, shall be replaced by the "flange design pressure", P_{eq},
where $P_{eq} = (16M_{pb}/\pi G^3 + 4\ F/\pi G) + P$

(b) The equation for longitudinal hub stress is:

$$S_H = \frac{fM_o}{Lg_1^2} + \frac{PB}{4g_o}$$

The second term on the right is an addition to the equation for S_H as given in Article XI-3000.

(c) The allowable stress limits are:

S_H, S_R and S_T not greater than 1.5 S_m.

The stress limits in Article XI-3000 essentially are:

S_R and S_T < S

S_H < 1.5S

$(S_H + S_R)/2<S$

$(S_T + S_R)/2<S$

Stresses of Rated Pressures, Zero Pipe Bending Moment

Table 13 shows the values of maximum flange stresses calculated according to the rules of NB-3647 for a pressure equal to the rated pressure at 100 F and with $S_a = S_b = 35{,}000$ psi; the value of S_m at 100 F for SA 193 Grade B7 bolt material. The locations and directions of the calculated stresses are shown in Figure 2. The bending moment in the attached pipe is assumed to be zero. If the flanges are made of SA-105 material, then $S_m = 23{,}300$ psi at 100 F and the maximum allowable flange stress is 1.5 x 23,300 = 34,950 psi. It can be noted in Table 13 that the following flanges have maximum stresses exceeding 34,950 psi:

Welding Neck: 24", 300, 400 and 600 classes.
Blind: 24" - 400, 16" - 900, 4" - 1,500 and 4" - 2,500.

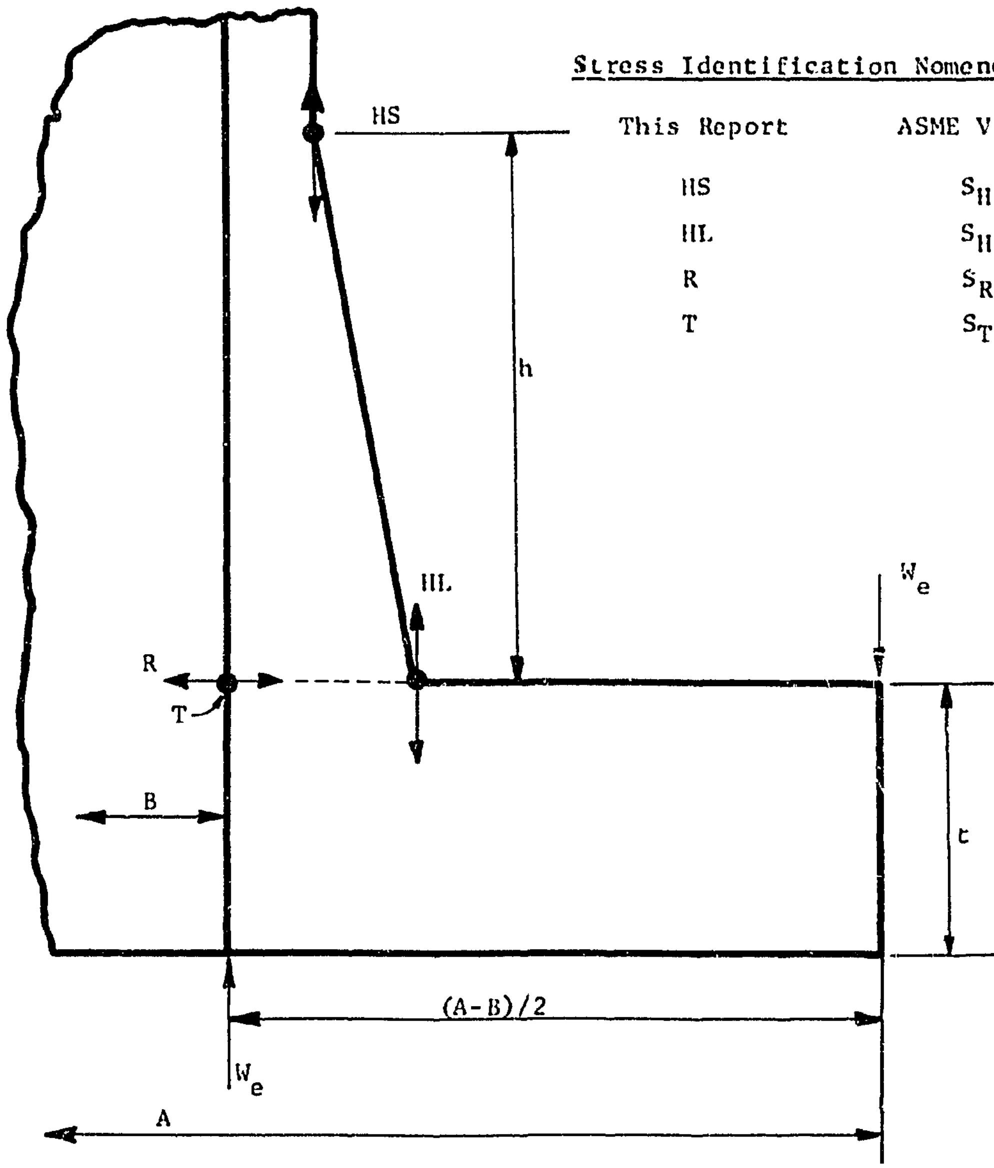
Table 14 shows the calculated values of maximum flange stresses in accordance with Appendix II of Section VIII - Div. 1. The calculated stresses in Table 14 are different than those in Table 13 for three reasons.

(1) In Table 13, the maximum stress is usually $S_H + PB/4g_o$, whereas in Table 14, the controlling stress is usually $(S_H + S_R)/2$.

TABLE 13. MAXIMUM CALCULATED FLANGE STRESSES, NB-3647 RULES, PRESSURE IS RATED PRESSURE AT 100 F, $S_a = S_b =$ 35,000 PSI

Class (Pressure)	Size	Welding Neck			Straight Hub			Blind[3]	
		Max. Stress	Stress[1] Ident.	Control[2] Condition	Max. Stress	Stress[1] Ident.	Control[2] Condition	Max. Stress	Control[2] Condition
150	4	20600	HL	GS	24200	HL	GS	13900	GS
(275)	8	15000	HL	OP	15500	HL	GS	11300	OP
	16	18400	R	GS	20300	HL	GS	21200	OP
	24	17700	HL	OP	20900	HL	GS	22500	OP
300	4	17000	HL	OP	19100	HL	GS	14900	GS
(720)	8	21500	HL	OP	21500	HL	GS	18400	OP
	16	24400	HS	GS	23200	HL	GS	26000	OP
	24	42600	HS	GS	23700	HL	GS	33500	OP
400	4	19100	HL	OP	22600	HL	GS	16900	GS
(960)	8	23000	HL	OP	24500	HL	GS	18500	OP
	16	26900	HS	OP	25700	HL	GS	28000	OP
	24	50000	HS	GS	28700	HL	GS	37500	OP
600	4	23600	HL	OP	22500	HL	GS	22700	OP
(1440)	8	23300	HL	OP	22600	HL	GS	23100	OP
	16	27800	HS	OP	26600	HL	GS	32900	OP
	24	47300	HS	OP	25300	HL	GS	33900	OP
900	4	27100	HL	OP	29700	HL	GS	31500	GS
(2160)	8	27000	R	GS	29800	R	GS	34100	OP
	16	26000	HL	OP	27600	R	GS	37900	OP
	24	30500	HS	OP	25900	HL	GS	31400	OP
1500	4	29000	HL	OP	29000	HL	OP	35300	OP
(3600)	8	25700	HL	OP	24600	HL	OP	27000	OP
	16	22600	R	GS	26600	R	GS	30400	OP
	24	24600	HS	OP	28500	HL	GS	29800	OP
2500	4	30600	HL	OP	29700	HL	OP	36900	OP
(6000)	8	25900	HL	OP	24400	HL	OP	28800	OP
	12	24000	HL	OP	22200	HL	OP	25100	OP

(1) Stress location/directions are indicated in Figure 2. Stresses are in psi.
(2) GS indicates gasket seating conditions.
OP indicates operating conditions
(3) Maximum stress is in center of blind flange.



Stress Identification Nomenclature

This Report	ASME VIII-1
HS	S_H
HL	S_H
R	S_R
T	S_T

FIGURE 2. IDENTIFICATION OF STRESSES CALCULATED BY CODE PROCEDURE

TABLE 14: CONTROLLING CALCULATED FLANGE STRESSES, SECTION VIII - DIV. 1 RULES, PRESSURE IS RATED PRESSURE AT 100 F, $S_a = S_b = 25{,}000$ PSI

Class (Pressure)	Size	Welding Neck Control Stress, psi	Welding Neck Stress(1) Iden.	Straight Hub Control Stress, psi	Straight Hub Stress(1) Iden.	Blind Max Stress, psi	Controlling Moment (2)
150 (275)	4	12400	HR	14300	HR	11000	GS
	8	8600	HR	9600	HR	11300	GS
	16	14400	R	15600	HR	21200	GS
	24	12200	HR	13900	HR	22500	GS
300 (720)	4	10800	HR	12800	HR	13800	GS
	8	12000	HR	14500	HR	18400	GS
	16	17800	HR	18400	HR	26000	GS
	24	25200	HR	19000	HR	33500	GS
400 (960)	4	12300	HT	14700	HR	15200	GS
	8	12400	HT	15300	HR	18500	OP
	16	18900	HR	19700	HR	28000	GS
	24	29300	HR	22500	HR	37500	GS
600 (1440)	4	15100	HR	17600	HR	22700	OP
	8	13800	HR	17100	HR	23100	OP
	16	19300	HR	21600	HR	33900	GS
	24	26800	HT	19800	HR	33900	OP
900 (2160)	4	19700	HR	22800	HR	29700	GS
	8	22200	R	24500	R	34100	GS
	16	21800	R	24500	R	37900	OP
	24	19100	HR	20300	HR	31400	GS
1500 (3600)	4	21300	HR	25300	HR	35300	OP
	8	16500	HR	19500	HR	27000	OP
	16	18400	R	21700	R	30400	GS
	24	16200	HT	20900	HR	29800	GS
2500 (6000)	4	22600	HT	24600	HR	36900	OP
	8	17800	HR	19800	HR	28800	OP
	12	15500	HR	17100	HR	25100	OP

(1) HR = $(S_H + S_R)/2$, R = S_R, HT = $(S_H + S_T)/2$

(2) Controlling moment for welding neck and straight hub. GS = gasket seating, OP = operating. For blind flanges, the operating moment controls for all but the 4" - 150 class.

(2) The allowable bolt stress in Table 13 is 35,000 psi, whereas in Table 14 the allowable bolt stress is 25,000 psi. Where gasket seating conditions control, the higher allowable bolt stress leads to a higher flange stress.

(3) In Table 14, stresses due to internal pressure are completely ignored; i.e., Section VIII - Div. 1 procedure.

If the flanges are made of SA-105 material, then the allowable stress at 100 F is 17,500 psi. It can be seen in Table 14 that all of the flanges that were unacceptable in Table 13 are also unacceptable in Table 14. Indeed, Table 14 shows that the following number of flanges are either acceptable or unacceptable.

	Number Acceptable	Number Unacceptable
Welding Neck	13	14
Straight Hub	10	17
Blind	4	23

Table 15 shows the calculated values of maximum flange stresses for a pressure equal to the rated pressure at 700 F and with S_a = 35,000 psi, S_b = 26,800 psi; the values of S_m for SA-193 Grade B7 bolt material at 100 F and 700 F, respectively. If the flanges are made of SA-105 material, then S_m = 23,300 psi at 100 F and S_m = 17,300 psi at 700 F. The maximum allowable flange stress is 1.5 x 23,300 = 34,950 psi at gasket seating conditions and 1.5 x 17,300 x 25,950 psi at operating conditions. Table 15 shows only the stress for the condition (gasket seating or operating) which is the higher. It can be seen that the 24" - 300, 600, and 900 class welding neck flanges have stresses which exceed 34,950 psi for the gasket seating condition. While not shown in Table 15, these are the only three flanges which are not acceptable by Code criteria for the 700 F ratings.

TABLE 15: MAXIMUM CALCULATED FLANGE STRESSES, NB-3647 RULES, PRESSURE IS RATED PRESSURE AT 700 F, S_a = 35,000 PSI, S_b = 26,800 PSI

		Welding Neck			Straight Hub			Blind (3)	
Class	Size	Max. Stress	Stress (1) Iden.	Control (2) Condition	Max. Stress	Stress (1) Iden.	Control (2) Condition	Max. Stress	Control (3) Condition
150	4	20600	HL	GS	24200	HL	GS	13900	GS
(110)	8	13600	HL	GS	15500	HL	GS	8100	GS
	16	17500	R	GS	19300	HL	GS	16800	GS
	24	16200	HL	GS	19200	HL	GS	11900	GS
300	4	14800	HL	GS	18200	HL	GS	14200	GS
(470)	8	16200	HL	GS	20400	HL	GS	13600	GS
	16	23300	HS	GS	22200	HL	GS	20800	GS
	24	40600	HS	GS	22600	HL	GS	21800	OP
400	4	17300	HL	GS	21700	HL	GS	16200	GS
(635)	8	17100	HL	GS	22300	HL	GS	13600	GS
	16	25700	HS	GS	24700	HL	GS	21400	GS
	24	47900	HS	GS	27500	HL	GS	24800	OP
600	4	17600	HL	GS	21200	HL	GS	19600	GS
(940)	8	16300	HL	GS	21300	HL	GS	17100	GS
	16	23500	HS	GS	25200	HL	GS	24200	GS
	24	43900	HS	GS	24000	HL	GS	22100	OP
900	4	23400	HL	GS	28100	HL	GS	29700	GS
(1410)	8	19100	HL	GS	28100	R	GS	29600	GS
	16	23200	R	GS	25900	R	GS	24900	GS
	24	26100	HS	GS	24500	HL	GS	22400	GS
1500	4	21400	HL	GS	26500	HL	GS	30300	GS
(2350)	8	18200	HL	GS	22400	HL	GS	21500	GS
	16	21400	R	GS	25200	R	GS	26900	GS
	24	19100	HS	GS	27000	HL	GS	24000	GS
2500	4	22400	HL	GS	25300	HL	GS	31400	GS
(3920)	8	18500	HL	GS	21000	HL	GS	23900	GS
	12	17300	HL	GS	19500	HL	GS	21000	GS

(1) Stress location/directions are indicated in Figure 2.
(2) GS indicates gasket seating conditions
OP indicates operating conditions
(3) Maximum stress is in center of blind flange.

Allowable Pressures, Zero Pipe Bending Moment

It is pertinent to calculate the allowable pressure for the flanges with conditions as used in Tables 13, 14, and 15; i.e., under the following conditions.

(a) 100 F, $S_a = S_b$ = 35,000 psi, max. flange stress = 34,950 psi

(b) 100 F, $S_a = S_a = S_b$ = 25,000 psi; controlling flange stress = 17,500 psi

(c) 700 F, S_a = 35,000 psi, S_b = 26,800 psi, max. flange stress = 25,950 psi.

Calculated maximum allowable pressures are shown in Table 16. It is apparent that the "pressure ratings" are erratic and illogical. Note that the 24" - 400 class welding neck flange has a zero maximum pressure rating. The 24" - 600 class welding neck flange has a lower pressure rating than the 24" - 150 class welding neck flange. The 24" - 400 class straight hub flange has a lower pressure rating than the 24" - 150 class straight hub flange.

The reason for these illogical pressure ratings is due to the Code method of calculating loads under gasket seating conditions using the equation:

$$W = \frac{(A_m + A_b)\ S_a}{2} \tag{24}$$

The value of A_m is the larger of

$$W_{m1} = \left[(\pi/4)\ G^2 + 2\pi bGm\right] P/S_b \tag{25}$$

and

$$W_{m2} = \pi bGy/S_a \tag{26}$$

TABLE 16. MAXIMUM ALLOWABLE PRESSURES (PSI)

		Welding Neck			Straight Hub			Blind		
		100 F	100 F	700 F	100 F	100 F	700 F	100 F	100 F	700 F
Class	Size	NB-3647	VIII-1	NB-3647	NB-3647	VIII-1	NB-3647	NB-3647	VIII-1	NB-3647
150	4	760	680	560	830	590	620	1160	580	860
	8	640	590	480	760	530	560	850	420	630
	16	610	450	450	770	420	560	450	220	340
	24	540	470	400	650	420	480	430	210	320
300	4	1480	890	1100	1730	1120	1280	1830	910	1350
	8	1170	1050	870	1320	870	980	1360	680	1010
	16	1050	0+	780	1550	860	1150	970	490	720
	24	630	0+	470	1330	570*	990	750	380	560
400	4	1760	1610	1310	1980	1330	1470	2210	1110	1640
	8	1470	1350	1090	1570	1100	1170	1820	910	1350
	16	1250	0+	930	1740	670*	1290	1200	600	890
	24	0+	0+	0+	1490	390*	1110	890	450	660
600	4	1630	1670	1210	2360	1430	1750	2220	1110	1650
	8	2160	1820	1600	2320	1470	1720	2170	1090	1610
	16	1810	1130*	1340	2230	810*	1650	1530	770	1140
	24	450*	0+	790	2140	1060*	1590	1490	740	1100
900	4	2780	1630*	2070	2980	1050*	2210	2540	620*	1890
	8	3060	0+	2270	3410	820*	2530	2220	650*	1650
	16	2910	1520*	2160	3080	1120*	2290	1990	1000	1480
	24	2480	1730*	1840	3280	1480*	2430	2400	1200	1780
1500	4	4340	2960	3220	4340	2040*	3220	3560	1130*	2640
	8	4890	3820	3630	5110	3230	3800	4660	2330	3460
	16	5650	3190*	4200	5810	2000*	4310	4140	2070	3070
	24	5110	3960	3790	4780	2250*	3550	4230	2120	3140
2500	4	6850	4640	5090	7050	4200*	5230	5680	1720*	4210
	8	8100	5890	6010	8590	5300	6380	7270	3640	5400
	12	8740	6780	6490	9450	6120	7020	8340	4180	6200

\+ Stress at gasket seating conditions exceeds allowable flange stress.
* Maximum allowable pressure controlled by gasket seating conditions.
No symbol; maximum allowable pressure controlled by operating conditions.

If W_{m1} is larger than W_{m2} (it almost always is for Table 16 calculations), the flange moment for seating conditions is:

$$M_{gs} = Wh_G = \left\{ \left[(\pi/4)G^2 + 2\pi bGm \right] P/S_b + A_b \right\} \frac{S_a h_G}{2} \qquad (27)$$

The flange stresses are proportional to M_{gs}; i.e., $S_f = K\,M_{gs}$, where K is the proportionality constant. If $S_f^{'} = K^{'} M_{gs}$, where $S_f^{'}$ = maximum (or controlling for VIII-1) flange stress and S_{all} = allowable flange stress, then:

$$K^{'} M_{gs} = S_{all} = K^{'} \left\{ \left[(\pi/4)G^2 + 2\pi bGm \right] P/S_b + A_b \right\} \frac{S_a h_G}{2} \qquad (28)$$

Solving equation (28) for P gives:

$$P = \left[\frac{2S_{all}}{S_a h_G K^{'}} - A_b \right] \frac{S_b}{(\pi/4)G^2 + 2\pi bGm} \qquad (29)$$

As an example, for the 24" - 600 class welding neck flange with S_{all} = 34,950 psi and $S_a = S_b$ = 35,000 psi. For this flange, A_b = 55.296 in.^{2}, h_G = 3.3257", $K^{'}$ = 0.009232 in.$^{-3}$ and $[(\pi/4)G^2 + 2\pi bGm]$ = 750.49 in.^{2}. With this specific data, Equation (29) gives:

$$P = \left[\frac{69{,}900}{35{,}000 \times 3.3257 \times 9.232 \times 10^{-3}} - 55.296 \right] \frac{35{,}000}{750.49}$$

$$= 455 \text{ psi}$$

For the 24" - 400 class welding neck flange, Equation (29) gives P = 5.18 psi. However, for this particular flange, W_{m2} is larger than W_{m1}. Accordingly,

$$S_f' = K' \left(\frac{\pi b G y}{S_a} + A_b \right) S_a h_G / 2 \tag{30}$$

For the 24" - 400 class welding neck flange; $K' = 0.01484$ in.$^{-3}$, πbGy = 138,040 lb, A_b = 47.52 in.^{2} and h_G = 2.8257 in. With S_a = 35,000 psi,

$$S_f' = 0.01484\ (138{,}040 + 35{,}000 \times 47.52) \times 2.8257/2$$

$$= 37{,}770 \text{ psi.}$$

Because 37,770 is greater than S_{all} = 34,950 psi, the Code evaluation is that the 24" 400 class welding neck flange is not acceptable for <u>any</u> pressure. It is interesting to note that even if y = 0, the flange stress by Equation (30) is 34,871 psi hence, the Code evaluation indicates that this particular flange, made of material such that S_m = 23,300 psi or less, is not acceptable for any pressure using any gasket.

<u>Allowable Pipe Bending Stresses at Rated Pressure</u>

Allowable pipe bending stresses as limited by the <u>bolting</u> are shown in Table 4 and Table 9. These are based on internal pressure equal to the rated pressure at 700 F. However, NB-3647.1 appears to be written so that another limit is imposed on pipe bending stresses by the limits on allowable flange stresses. Note that NB-3647.1(a) states that P_{FD} is to be used in XI-3220 and XI-3230. Because XI-3230 is on "Flange Moments", the flange stresses will obviously depend upon the magnitude of $P_{eq} = 16\ M_{pb}/\pi G^3$ and M_{pb}, the bending moment in the attached pipe.

The allowable pipe bending stresses, as limited by the flange stress, are shown in the first two columns of Table 17. Also shown, for comparison, are the allowable pipe bending stresses as limited by the bolt stress; taken from Table 9.

Table 17 data are for flanges made of SA-105 material (S_m = 17,300 psi at 700 F) and bolts made of SA-193 Grade B7 (S_m = 26,800 psi at 700 F).

It may be seen in Table 17 that the 24" - 400 and 24" - 600 class welding neck flanges have zero allowable pipe bending stress; actually, the results give negative allowable pipe bending stresses because the rated pressure is higher than the maximum allowable pressure (see Table 16). The 24" - 300 class allowable pipe bending stress is essentially zero; in Table 16, the allowable pressure is shown as equal to the rated pressure, but before round off the allowable pressure was calculated as 0.64 psi higher than the rated pressure.

It can be seen in Table 17 that the NB-3647 rules place severe restrictions on permissible bending stresses in the attached pipe. The large sizes of B16.5 flanges, while probably less capable of carrying high bending stresses in the attached pipe than the smaller sizes, have never-the-less been used for many years in piping systems where significant bending moment loads were imposed on the flanged joints.

Basic Theory Flange Stresses

In the preceding section on "Flange Stress Limits of NB-3647", we have followed the Code procedure for calculating flange stresses. In so doing, we have arrived at some seemingly contradictory results: B16.5 flanges which have a long and satisfactory use history are shown to not meet the Code criteria and, in one extreme case (the 24" - 400 class) the flanges will not meet Code criteria for any pressure with any gasket; even with zero pipe bending moment. The Code restrictions on allowable pipe bending stresses appear to be highly questionable for large size flanges.

TABLE 17. ALLOWABLE BENDING STRESSES IN PIPE ATTACHED TO FLANGED JOINT, P = RATED PRESSURE AT 700 F. PIPE WALL THICKNESS = g_o.

		Flange Stress(1)		Bolt Stress(2)	
Class (P)	Size	Welding Neck	Straight Hub	NB-3647	Program FLANGE
150 (110)	4	28700	25000	36400	36700
	8	21500	19700	16900	19400
	16	21400	19700	22950	25300
	24	17700	15800	23500	23900
300 (470)	4	32600	29300	26400	38000
	8	16600	15000	14900	22900
	16	11500	18000	22000	28700
	24	19	13000	17300	22100
400 (635)	4	28400	25000	31000	31100
	8	15000	12600	15300	24900
	16	8000	13500	20300	27000
	24	0	9100	19300	24200
600 (940)	4	7600	18700	16000	31300
	8	16100	13600	12700	23000
	16	7700	10300	14900	22600
	24	0	8700	12900	18100
900 (1410)	4	14200	13500	24100	40800
	8	14700	14000	15100	27200
	16	10100	8700	9600	17400
	24	5000	9600	13900	21800
1500 (2350)	4	11800	9500	13800	33700
	8	12900	11300	11500	25700
	16	16200	12400	15000	27100
	24	10900	6900	13650	23800
2500 (3920)	4	9400	9100	10550	34700
	8	12000	11700	10250	27900
	12	13000	12800	10400	25800

(1) Allowable pipe bending stress, psi, as limited by flange stress criterion.
(2) Allowable pipe bending stress, psi, as limited by bolt stress criterion. From Table 9.

The observation that B16.5 flanges do not necessarily meet the Code criteria has been brought to the attention of the ASME Boiler Code Committee many times; the standard response has been that B16.5 flanges are accepted by the Code on the basis of successful operating history. However, it is pertinent to look in a more realistic way at calculated stresses in flanges and their significance. For this purpose, we will present in the following some flange stress calculations obtained using the computer program FLANGE.

In the Code procedure, a number of somewhat arbitrary aspects are involved in the design procedure; e.g., the effective gasket width, b_o; the effective gasket diameter, G; the gasket factors, m and y; and the flange stress limits. While all of these have a measure of engineering significance, and provide a good method of designing non-standard flanges, the total design procedure of the Code is not necessarily a good way to evaluate the adequacy of B16.5 flanged joints.

Flange Stresses at Preload Bolt Stress of 40,000 psi

In the basic theory approach, we start with the assumption that the bolts are initially tightened to give an average bolt stress of 40,000 psi; an assumption whose basis is partially indicated in Table 8. The maximum calculated stresses are shown in Table 18. The locations and directions of the calculated stresses are shown in Figure 3. It should be noted that these results do not involve the gasket factors m and y.

For welding neck and straight hub flanges, the moment applied to the flange ring is simply $M = 40{,}000\ A_b h_g$, where $h_G = (C - G_m)/2$, $G_M = (G_o + G_i)/2$ and C is the bolt circle diameter. There is potentially a major conservatism in this value of h_G; as the flange rotates the effective gasket diameter will tend to shift closer to G_o and the effective line of bolt load application will tend to shift inward. Some test data relevant to these effects are included in Appendix A.

TABLE 18. BASIC THEORY FLANGE STRESSES (PSI) AT PRELOAD BOLT STRESS OF 40,000 PSI

		Welding Neck		Straight Hub		Blind[2]	
Class	Size	Max. Stress	Stress Iden.[1]	Max. Stress	Stress Iden.[1]	Code	FLANGE
150	4	39360	LL	44520	LL	23190	19680
	8	24390	LL	27380	LL	12550	12640
	16	36400	RH	39640	LL	30400	30040
	24	37350	LL	44720	LL	22490	25850
300	4	27160	LL	31910	LL	23220	18340
	8	28000	LL	33940	LL	21050	18530
	16	40920	CSO	41210	LL	35340	32080
	24	66200	LS	42080	LL	34070	33040
400	4	32610	LL	38590	LL	26610	21030
	8	30020	LL	36980	LL	20790	18300
	16	42480	CSO	44640	LL	35590	32310
	24	79020	LS	51610	LL	40290	39130
600	4	27960	LL	32330	LL	28350	20930
	8	26700	LL	33170	LL	25080	20490
	16	39370	RH	42640	RH	37770	32420
	24	66290	LS	40910	LL	31020	29000
900	4	40780	RH	44530	LL	45370	31390
	8	43280	RH	45470	RH	44400	32630
	16	38810	RH	41080	RH	35950	30370
	24	36830	LS	40870	LL	35050	29780
1500	4	36760	RH	38850	RH	41860	28030
	8	28660	LL	33440	LL	30720	22340
	16	39920	RH	42960	RH	41960	30730
	24	34980	RH	43700	LL	37380	28780
2500	4	34610	RH	35850	LL	41880	25380
	8	30090	RH	31090	RH	33420	22130
	12	27390	RH	29510	LL	30320	20380

(1) Stress location/directions are indicated in Figure 3. For LL and LS, surface is not indicated because stresses are equal and opposite sign (pure bending).
(2) Maximum stress is in center of flange.

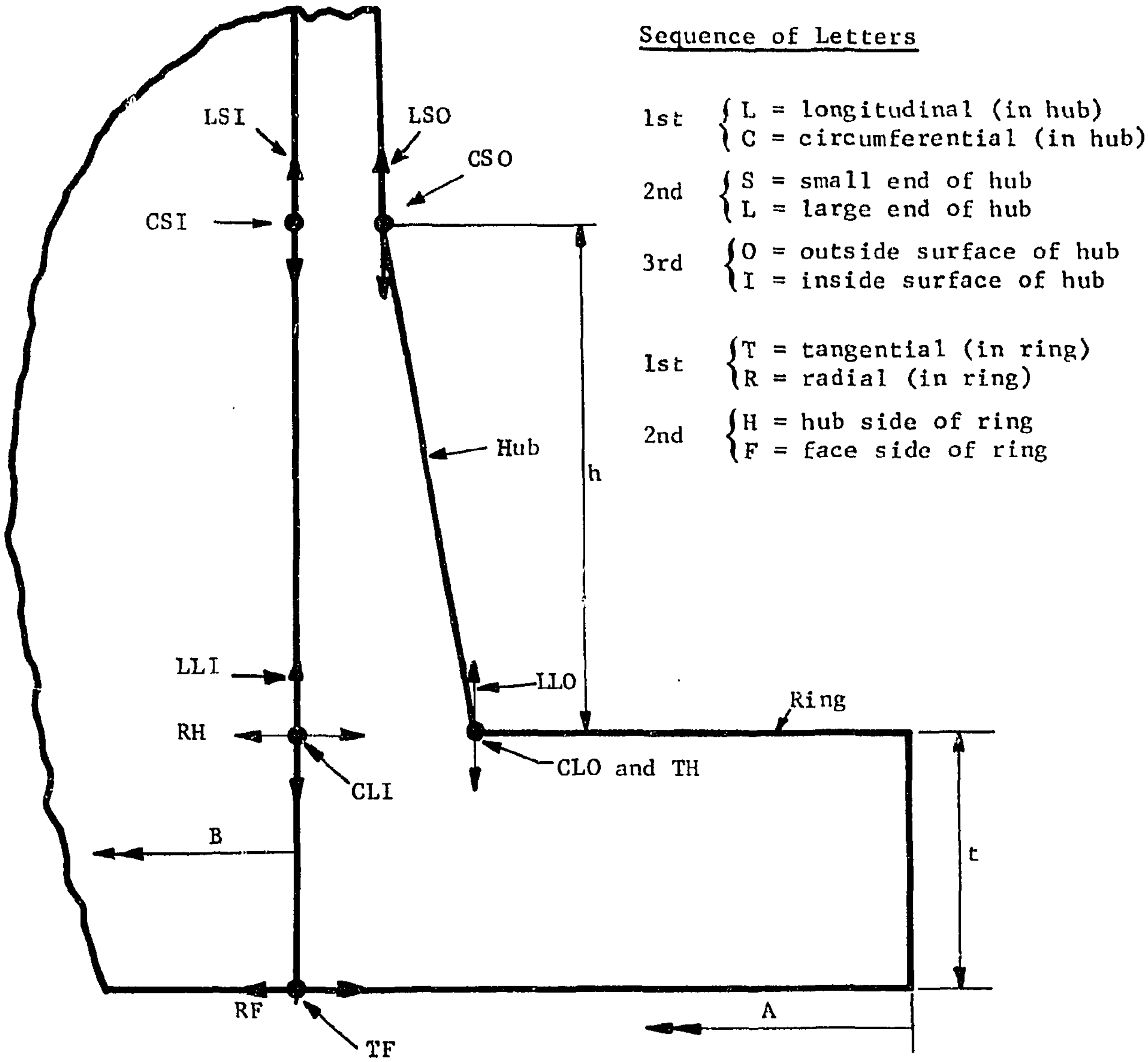


FIGURE 3. IDENTIFICATION OF STRESSES FROM COMPUTER PROGRAM FLANGE

For blind flanges, Table 18 shows stresses identified as "Code" and "FLANGE". The "Code" stresses were calculated using the equation given in NC-3325.2(b) with $P = 0$ and $W = 40,000\ A_b$:

$$S = \left(\frac{G}{t}\right)^2 \frac{1.78 \times 40,000 \times A_b \times h_G}{G^3} \tag{31}$$

Equation (31) car almost* be derived by assuming that a blind flange is a flat circular plate of outside diameter equal to G, that the metal outside of G is ignored, and that the plate is loaded by an edge moment of $M/\pi G$.

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* The stress at the center of a flat plate, modeled as described above plus the assumption of simple support along diameter G, and with pressure loading as well as edge moment loading, is theoretically given by

$$S = \frac{3(3+\nu)}{32} P\left(\frac{G}{t}\right)^2 + \frac{6(1-\nu^2)}{\pi}\left(\frac{Wh_G}{Gt^2}\right)$$

With ν = Poisson's ratio taken as 0.25, this gives

$$S = 0.3047\ P\left(\frac{G}{t}\right)^2 + 1.7905\ \frac{Wh_G}{Gt^2} \quad .$$

This is identical to the equation given in NC-3325 except that constants of 0.3 and 1.78 are used.

The FLANGE analysis method[3] treats the blind flange more completely; the circular plate is divided into three parts. Part 1 extends from the center to the gasket diameter, G_m; Part 2 extends from G_m to the bolt circle, C, and Part 3 extends from C to the flange outside diameter, A. Bolt loads are represented by line loads along G_m and C.

The major difference between the two methods is that the FLANGE method takes into account the flange material outside of the gasket diameter. As can be seen in Table 18, this makes a significant difference for the higher pressure classes.

We note that the second highest stressed flange in Table 18 is the 24" - 600 class welding neck. Let us look at the stresses in this flange (rather than the 24" - 400 class, which is not used very much) in more detail.

The complete set of stresses calculated by FLANGE for a bolt stress of 40,000 psi are shown in Table 19 in the column headed g_o = 0.873", h = 2.691". The symbols used to define the stress location and directions are identified in Figure 3. Now, neither g_o nor h are specified in B16.5 and it is of interest to see how the calculated stresses vary as g_o and h are varied. It is apparent from Table 19 that large variations in maximum stresses (a ratio of about 3) occurs within ranges of g_o and h that could be used in conjunction with the stress analysis of a 24" - 600 class welding neck flange. An important conclusion, therefore, is that one cannot state what the magnitude of stresses are in B16.5 welding neck flanges unless one specifically states the values of g_o and h used in the calculations.

A more significant aspect of Table 19 consists of an evaluation of the stresses shown for g_o = 0.438" and h = 2.691". The 24" - 600 class flange has been frequently installed in gas transmission pipe lines, using a pipe with a wall thickness of around 0.438". The question arises: Can the bolts in a 24" - 600 lb welding neck flanged joint (flanges made

TABLE 19. EFFECT OF VARIATION OF PIPE WALL THICKNESS, g_o, AND HUB LENGTH, h, ON STRESSES (PSI) IN A 24" - 600 CLASS WELDING NECK FLANGE LOADED WITH BOLT STRESS OF 40,000 PSI

Stress (1)	g_o = 0.873" h = 2.691"	g_o = 0.438" h = 2.691"	g_o = 1.309" h = 2.691"	g_o = 0.873" h = 4.0"	g_o = 0.873" h = 2.0"
LSO	66,290	103,000	47,370	34,480	88,720
LSI	-66,290	-103,000	-47,370	-34,480	-88,720
CSO	53,670	87,150	36,310	35,700	64,300
CSI	13,900	25,330	7,890	15,010	11,060
LLO	26,980	26,860	24,760	30,990	24,250
LLI	-26,980	-26,860	-24,760	-30,990	-24,250
CLO	20,290	30,590	14,070	16,950	23,450
CLI	4,100	14,470	-800	-1,650	8,900
TH	21,970	30,940	17,310	17,980	25,460
TF	-40,560	-50,340	-34,650	-36,160	-43,930
RH	32,580	28,030	35,560	34,440	30,940
RF	-23,870	-19,530	-26,910	-25,930	-22,280
Membrane Stresses					
MLS	0	0	0	0	0
MCS	33,790	56,240	23,100	25,360	37,680
MLL	0	0	0	0	0
MCL	12,200	22,530	6,640	7,650	16,180
MT	-9,300	-9,700	-8,670	-9,090	-9,240
MR	4,360	4,250	4,330	4,260	4,330

(1) See Figure 3. MLS is the membrane stress in the longitudinal direction at the small-end of the hub. Following symbols are analogous thereto.

of SA-105 material, bolts of SA-193 Grade B7 material) with attached pipe of 0.438" wall thickness be tightened to an average bolt stress of 40,000 psi? Despite the calculated stress of 103,000 psi, the answer is yes. One of the authors has supervised the installation of such flanged joints; no difficulty* was encountered and, insofar as we are aware, these flanged joints are performing satisfactorily some 15 years after installation.

The magnitude of the calculated stress of the small end of the hub is deemed to be an accurate evaluation of the elastic stress in an actual flanged joint, provided h_G is actually as assumed in the analysis. Indeed, the stresses at this location are probably the only place in the flanged joint where, theoretically, the calculated stress should be accurate. However, at high bolt loads, the flange tends to rotate and one would expect a decrease in h_G as the gasket load shifts outward and the bolt load shifts inward. Figure A10 in Appendix A gives some evidence of the inward shift of the bolt load. To illustrate the possible magnitude of these shifts, let us assume that at high bolt loads the effective gasket diameter is (G_o - gasket width/2) = 27.25" - .8125" = 26.4375", rather than at the gasket mean diameter of 25.625". Also assume that the bolt load acts at (C-d) = 33 - 1.875 = 31.125, rather than at C = 33". The value of h_G is then (31.125 - 26.4375)/2 = 2.344" rather than (33 - 25.625)/2 = 3.688". The maximum calculated stress is then 103,000 x 2.344/3.688 = 65,500 psi rather than 103,000 psi.

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* Tightening flanged joint bolts is ordinarily done by step-tightening in a more-or-less regular criss-cross pattern on individual bolts. If both of the flanges in the joint are capable of sustaining the desired initial bolt stress without gross yielding, the bolts can be tightened to the desired stress in 3 or 4 "rounds" of individual bolt tightening. In borderline cases, wherein the flange strength is increasing due to strain-hardening and stress redistribution, it may take 5 or 6 rounds of individual bolt tightening. If the flange strength is inadequate, it will be impossible to reach the desired bolt stress. Continued tightening will simply rotate the flange ring until, eventually, the outer edges of the flange rings contact each other.

In addition to the bolt load and gasket reaction shifts, the characteristics of the stresses and their distribution help to explain the load capacity of the flanged joint. As shown in Table 19, the highest stress is a longitudinal bending stress at the small end of the hub; at the large end of the hub, this stress has decreased to 26,860 psi. The membrane longitudinal stress in the hub is zero. The circumferential stresses in the hub, both membrane and bending, also decrease substantially from the small end of the hub to the large end of the hub. The radial stresses in the flange ring are shown in Table 19 at the inside diameter of the flange; they decrease to zero at the outer edge of the flange ring. The tangential stresses, which are the most significant of the stresses with respect to gross yielding, are relatively low; i.e., + 30,940, - 50,340 psi. Because of the large variation in stresses throughout the flange structure, one would not expect gross yielding to occur when the highest stress reaches the yield strength of the material. Some local yielding can and probably does occur, but the lower-stressed portions of the flanges have the capability of picking up the load*. From a theoretical standpoint, a limit-load or elastic-plastic analysis (taking into account the bolt load and gasket reaction shifts) would be required to evaluate the load capacity of flanged joints.

In summary, Table 18 shows high calculated (elastic) stresses for an applied bolt stress of 40,000 psi. Table 19 shows that, for a welding neck flange, the calculated stresses can vary significantly

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* Some recognition of this aspect is included in Section VIII - Div. 1 in the use of a "controlling stress". The stresses in the hub are permitted to reach 1.5 times the allowable stress, provided there is reserve strength in the ring as shown by lower radial and tangential stresses.

with g_o and h. Operating history with the 24" - 600 class welding neck flanges shows that high calculated maximum stresses do not necessarily indicate that a flanged joint with such high calculated stresses is not capable of withstanding an initial bolt stress of 40,000 psi. The high load capacity can be attributed to bolt load/gasket reaction shifts and to the detailed stress distribution in a flanged joint.

Unfortunately, in the absence of a limit load analysis, it is not possible to quantify the load capacity of a flanged joint. Appendix A includes the results of a number of tests on B16.5 flanged joints in which the maximum calculated stresses were significantly higher than the flange material yield strength. However, there was one test (Table A9, Test No. 3) in which measurable yielding of the flange occurred even though the maximum calculated stress was less than the yield strength of the flange material; a result for which the author has no explanation.

NB-3600, as interpreted by the authors, requires that a flanged joint be analyzed under the rules of NB-3650; just as any other piping product (elbows, tees) purchased to an ANSI standard must be. The analysis of the bolting in flanged joints was discussed earlier. Now we will discuss an analysis of the flanges.

In terms of an NB-3650 analysis of the flanges, it is necessary to categorize the stresses shown in Table 18 so that a fatigue analysis of the bolt tightening/untightening cycle can be performed. Because the stresses are mostly local bending and because the bolt load itself is displacement controlled, it is deemed appropriate to categorize the stresses as primary plus secondary. For flanges made of SA-105 material, the allowable numbers of bolt tightening/untightening cycles, N_D, then are as follows.

Temp, F	S_m, psi	Design Cycles, N_D					
		Welding Neck		Straight Hub		Blind	
		Max	Min	Max	Min.	Max.	Min.
100	23,300	35,000	650	25,000	2,000	10^6	8,500
400	20,600	35,000	350	25,000	2,000	10^6	8,500

The above tabulation is based on stresses shown in Table 18 and a stress concentration factor (K-index) of 2.0 to account for such aspects as the bolt holes and the fillet between the hub and ring. The temperature to use for the tightening/untightening cycle, if operating temperatures get up to 700 F, is taken as 400 F; see footnote 1 to Code Table NB-3222-1.

As was found in the fatigue analysis of bolting, it appears that permissible bolt tightening/untightening cycles are far greater than would be anticipated for any normal B16.5 flanged joint installation.

Stresses at Preload (S_{b1}= 40,000) Plus Operating Conditions

At this point, we will consider only pressure loading and the range of stresses resulting from pressure loading. Table 20 shows how the maximum stresses change due to a pressure equal to the primary rating pressure. The change in stress is due to

(a) The moment on the flange ring changes because of internal pressure and, therefore, the stress from the moment changes.

(b) The stress produced by pressure is added to the stress due to the change in moment by pressure.

The combination of these two effects may either increase or decrease that stress which was a maximum under preload only. However, the maximum stress range and/or maximum stress is not necessarily at the location where stresses are maximum due to preload. Table 21 shows the variations of all stresses in the 24" - 300 class welding neck flange; in this case, the maximum stress at pressure occurs on the small end of the hub, outside surface, but the maximum stress range of 5,550 psi occurs on the inside surface. Table 22 shows the variations in all stresses in the 16" - 1500 class welding neck flange, where the pressure of 5,400 psi represents a hydrostatic test pressure of 1.5

TABLE 20. FLANGE STRESSES WITH PRELOAD, S_{b1} = 40,000 PSI, PLUS PRESSURE EQUAL TO PRIMARY RATING PRESSURE.

Class	Size	Welding Neck Stress Due to: S_{b1}	P	S_{b2} + P	Stress Iden.(1)	Straight Hub Stress Due to: S_{b1}	P	S_{b2} + P	Stress Iden.(1)
150	4	39360	130	39600	LLO	44520	320	44820	LLO
	8	24390	250	24930	LLO	27380	520	27880	LLO
	16	36400	-275	35730	RH	39640	560	39470	LLO
	24	37350	360	37160	LLO	44720	770	44460	LLO
300	4	27160	120	27460	LLO	31910	430	32430	LLO
	8	28000	160	28170	LLO	33940	650	34290	LLO
	16	40920	4380	44810	CSO	41250	690	41110	LLO
	24	66200	2570	67610	LSO	42080	850	41740	LLO
400	4	32610	130	32950	LLO	38590	540	39150	LLO
	8	30020	160	30270	LLO	36980	790	37530	LLO
	16	42480	4820	46810	CSO	44600	850	44540	LLO
	24	79020	3080	80830	LSO	51610	1040	51330	LLO
600	4	27960	3280	31520	LLO	32330	650	33020	LLO
	8	26700	220	27050	LLO	33170	870	33820	LLO
	16	39370	-955	38060	RH	42640	-670	41180	RH
	24	66290	2940	68400	LSO	40910	1180	41070	LLO
900	4	40780	-1000	40060	RH	44530	760	45320	LLO
	8	43280	-1080	42330	RH	45470	-860	44270	RH
	16	33810	-1210	37620	RH	41080	-910	39600	RH
	24	36830	3200	39900	LSO	40870	1220	41490	LLO
1500	4	36760	-1520	35780	RH	38850	-1300	37840	RH
	8	28660	540	29760	LLO	33440	1180	34870	LLO
	16	39920	-1700	38550	RH	42960	-1320	41550	RH
	24	34980	-1880	33370	RH	43700	1440	42220	LLO
2500	4	34610	-1980	33400	RH	35850	1240	37680	LLO
	8	30090	-2060	28910	RH	31090	-1900	29790	RH
	12	27390	-2090	26150	RH	29510	1380	31500	LLO

(1) See Figure 3.

TABLE 21. FLANGE STRESSES IN A 24" - 600 CLASS WELDING NECK FLANGE, S_{b1} = 40,000 PSI, P = 600 PSI.

M, 10^{-6} in.-lb → P, psi →	8.156 0	8.053 0	0 600	8.053 600	Stress Range, psi
(1)					
LSO	66290	65450	2940	68390	2100
LSI	-66290	-65450	4710	-60740	5550
CSO	53670	52990	5090	58080	4410
CSI	13900	13720	5620	19340	5440
LLO	26980	26640	530	27170	190
LLI	-26980	-26640	1690	-24950	2030
CLO	20290	20030	2750	22780	2490
CLI	4100	4050	3100	7150	3050
TH	21970	21690	2270	23960	1990
TF	-40560	-40050	440	-39610	950
RH	32580	32170	-1060	31110	1470
RF	-23870	-23570	-200	-23770	100
MLS	0	0	3825	3825	--
MCS	33790	33360	5360	38710	--
MLL	0	0	1110	1110	--
MCL	12200	12040	2925	14965	--
MT	-9300	-9180	1355	-7825	--
MR	4360	4300	-630	3670	--

(1) See Figure 3. MLS is the membrane stress in the longitudinal direction at the small end of the hub. Following symbols are analogous thereto.

TABLE 22. FLANGE STRESSES IN A 16" - 1500 CLASS WELDING NECK FLANGE, S_{b1} = 40,000 PSI, P = 5,400 PSI.

$M \times 10^{-7}$ in.-lb →	1.442	1.490	0	1.490	Stress Range
P, psi →	*0*	*0*	*5,400*	*5,400*	
(1)					
LSO	14,440	14,920	11,010	25,930	11,490
LSI	-14,440	-14,920	13,980	- 940	13,500
CSO	20,250	20,920	18,820	39,740	19,490
CSI	11,590	11,980	19,710	31,690	20,100
LLO	28,870	29,830	1,710	31,540	2,670
LLI	-28,870	-29,830	6,560	-23,270	5,600
CLO	16,730	17,290	10,880	28,170	11,440
CLI	-5,860	-6,060	12,330	6,270	12,130
TH	20,050	20,720	8,520	29,240	9,190
TF	-36,620	-37,840	6,070	-31,770	4,850
RH	39,920	41,250	-6,120	35,130	4,830
RF	-28,020	-28,950	-4,360	-33,310	5,290
MLS	0	0	12,500	12,500	--
MCS	15,920	16,450	19,270	35,720	--
MLL	0	0	4,140	4,140	--
MCL	5,440	5,620	11,600	17,220	--
MT	-8,290	-8,560	7,300	-1,270	--
MR	5,950	6,150	-5,240	910	--

(1) See Figure 3. MLS is the membrane stress in the longitudinal direction at the small end of the hub. Following symbols are analogous thereto.

times the 100 F rating pressure. In the example, the maximum stress location changes from RH at zero pressure to CSO at the test pressure; the maximum stress range occurs at CSI.

The main conclusion that can be drawn from Tables 20, 21, and 22 is that the stress range due to changes in pressure is small. In Table 20, for example, the largest stress range is 4,330 psi (16" - 400 class WN). The stress range is proportional to the pressure range hence, for a hydrostatic test of 3.6 times the 100 F rating, the stress range for the hydrostatic test does not exceed 3.6 x 4,330 = 15,600 psi at the location of maximum stress due to bolt load. A few spot checks suggest that the stress range due to pressure will not exceed that in the attached pipe as calculated by the equation:

$$S = \frac{PD_m}{2g_o} \qquad (32)$$

Because the stress from Equation (32) is limited to S_m for the pipe material, it is apparent that the number of design cycles of pressure loading will be large. For example, S_m for SA-106 Grade B pipe at 100 F is 20,000 psi; the allowable operating pressure cycles (Code Figure I-9.1, stress amplitude = 10,000 psi) is greater than 10^6.

In doing a complete NB-3650 analysis, stresses due to temperature effects must also be evaluated. The computer program FLANGE evaluates the effects of change in modulus of elasticity of the flanges, bolts, and gaskets, the temperature difference between flange-ring and bolts (which gives a change in bolt load) and temperature difference between the flange-ring and the hub (which gives both a change in bolt load and flange stress).

Table 23 gives an example of stress ranges where it is assumed that E_2/E_1 = 0.80 (for all parts of the flanged joint) and that the average temperature of the hub (T_{hp}) is 50 F lower than the average temperature of the flange ring (T_f); $T_{hp} - T_f$ = -50 F. The combination of change in modulus and the thermal gradient changes the moment from 4.55 x 10^5 in.-lb (preload to S_{b1} = 40,000 psi) to 3.32 x 10^5 in.-lb. This gives a decrease in flange stresses ; we add the stress due to the thermal gradient to obtain the stress at the assumed conditions.

TABLE 23. FLANGE STRESSES IN A 4" - 900 CLASS STRAIGHT HUB FLANGE, S_{b1} = 40,000 psi, P = 0, THERMAL GRADIENT (T_{hp} - T_f) = - 50 F.

M x 10^{-5} in.-lb →	4.55	3.32	0	3.32	
T_{hp} - T_f, F →	0	0	-50	-50	Stress Range, psi
Stress(1)					
LLO	44,530	32,490	-480	32,010	12,520
LLI	-45,530	-32,490	480	-32,010	13,520
CLO	18,580	13,560	-4,180	9,380	9,200
CLI	-8,140	-5,940	4,470	-1,470	6,670
TH	18,080	13,190	-3,780	9,410	8,670
TF	-32,920	-24,020	1,760	-22,260	10,660
RH	42,880	31,290	2,990	34,280	8,600
RF	-31,160	-22,740	-1,390	-24,130	7,030
MLL	0	0	0	0	--
MCL	5,220	3,810	150	3,960	--
MT	-7,420	-5,420	-1,010	-6,430	--
MR	5,860	4,280	800	5,080	--

(1) See Figure 3. MLL is the membrane stress in the longitudinal direction of the hub. Following symbols are analogous thereto.

Table 24 gives an example of stress ranges for the assumed set of operating conditions shown in Table 7, except that $S_{pb} = 0$. It is apparent from Table 24 that, even for this severe set of assumed operating conditions, the stress range is quite small; the design cycles for this loading (with a stress concentration factor of 2.0, SA-105 flange material) is around 40,000 cycles.

The main conclusion from Tables 20 thru 24 is that, for any credible set of operating conditions, B16.5 welding neck and straight hub flanged joints should be acceptable under the analysis requirements of NB-3650.

Calculated stresses in blind flanges are shown in Table 25. The stresses identified as "Code" were calculated using the equation given in NC-3325.2(b) with P = primary rating pressure and W = 40,000 A_b:

$$S = \left(\frac{G}{t}\right)^2 \left[\frac{0.3P + 1.78 \times 40,000 \times A_b \times h_G}{G^3}\right] \tag{32}$$

The stresses identified as "FLANGE" were calculated using the computer program FLANGE. The treatment of blind flanges was described under Equation (31). For pressure loading, the pressure acts on Part 1 only. The calculation method recognizes that the bolt load at pressure is not necessarily the same as the initial bolt load.

It can be seen in Table 25 that, particularly for the higher pressure classes, the FLANGE analysis gives significantly lower stresses than those obtained using the Code Equation (32).

To proceed with the NB-3650 analysis of blind flanges, we must categorize the stresses shown in Table 25. The stress due to pressure is a local bending stress in the center of the flange and it appears appropriate to classify it as P_b; primary membrane (which is zero) plus primary bending. The stresses in Table 25 are for a pressure equal to the primary rating pressure; at the 100 F rating pressure, the stresses would be 2.4 times the values shown. The Code limit on P_b - category stresses is 1.5 S_m. For SA-105 flange material, 1.5 S_m ranges from 34,950 at 100 F to 25,950 at 700 F. It is apparent from Table 25 that the stresses due to pressure are less than the allowable stresses.

TABLE 24. FLANGE STRESSES IN A 24" - 300 CLASS WELDING NECK FLANGE, S_{b1} = 40,000 PSI, P = 300 psi, E_2/E_1 = 0.767, $(T_b - T_f)$ = 50 F, $(T_{hp} - T_f)$ = 100 F

$M \times 10^{-6}$ in.-lb →	4.299	2.382	0	0	2.382	
P, psi →	0	0	300	0	300	
$T_{hp} - T_f$, F →	0	0	0	100	100	Stress Range, psi
Stress[(1)]						
LSO	66,200	36,670	2,570	10,570	49,810	16,390
LSI	-66,200	-36,670	4,930	-10,570	-42,310	23,890
CSO	60,200	33,350	4,680	4,820	42,850	17,350
CSI	20,480	11,350	5,390	-1,520	15,220	5,260
LLO	29,910	16,570	330	-280	16,620	13,290
LLI	-29,910	-16,570	1,190	280	-15,100	14,810
CLO	18,050	10,000	1,730	-10,580	1,150	16,900
CLI	100	60	1,990	-10,420	-8,370	8,470
TH	20,520	11,370	1,450	6,670	19,490	1,030
TF	-41,690	-23,100	0	-3,560	-26,660	15,030
RH	38,130	21,120	-600	-2,780	17,740	20,390
RF	-29,290	-16,230	0	1,490	-14,740	14,550
MLS	0	0	3,750	0	3,750	--
MCS	40,340	22,350	5,040	1,650	29,040	--
MLL	0	0	760	0	760	--
MCL	9,080	5,030	1,860	-10,500	-3,610	--
MT	-10,590	-5,870	730	1,560	-3,590	--
MR	4,420	2,450	-300	-650	1,500	--

(1) See Figure 3. MLS is the membrane stress in the longitudinal direction at the small end of the hub. Following symbols are analogous thereto.

TABLE 25. BLIND FLANGE STRESSES (PSI) WITH PRELOAD, S_{b1} = 40,000 PSI, PLUS PRESSURE EQUAL TO PRIMARY RATING PRESSURE

Class	Size	FLANGE Stress Due to: S_{b1}	FLANGE Stress Due to: P	FLANGE Stress Due to: S_{b2} + P	CODE Stress Due to: S_{b1}	CODE Stress Due to: P	CODE Stress Due to: S_{b2} + P
150	4	19680	1300	20620	23190	1570	24760
	8	12640	3040	14930	12550	3500	16050
	16	30040	6030	32880	30400	6830	37320
	24	25850	8010	30270	22500	8890	31390
300	4	18340	1440	19450	23220	1770	24990
	8	18530	2850	20530	21050	3350	24400
	16	32080	4830	34700	35340	5570	40910
	24	33040	7210	36790	34070	8260	42330
400	4	21030	1590	22260	26610	1950	28560
	8	18300	2850	20400	20790	3360	24150
	16	32310	5210	35300	35590	6020	41610
	24	39130	8080	43350	40290	9260	49550
600	4	20930	1980	22460	28350	2450	30800
	8	20490	3090	22750	25080	3700	28780
	16	32420	5370	35520	37770	6270	44040
	24	29000	6780	33110	31020	7810	38830
900	4	31390	2160	32990	45370	2700	48070
	8	32630	3490	34940	44400	4250	48650
	16	30370	5900	34030	35930	6910	42860
	24	29780	5260	33130	35050	6200	41250
1500	4	28030	2430	29970	41860	3060	44920
	8	22340	2760	24610	30720	3370	34090
	16	30730	3540	33310	41960	4270	46230
	24	28780	4060	31720	37380	4880	42260
2500	4	25380	2010	27150	41880	2560	44440
	8	22130	2380	24240	33420	2950	36370
	12	20380	2360	22500	30320	2150	32470

The stress due to the bolt load and the sum of stresses due to bolt load and pressure are appropriately categorized, in the authors' opinion, as primary plus secondary stresses. A fatigue analysis leads to the same conclusion reached for welding neck and straight hub flanges; i.e., for any credible set of operating conditions, B16.5 flanges should be acceptable under the analysis requirements of NB-3650.

Effect of Pipe Bending Moment on Flange Stresses

In the preceding we have discussed operating conditions of pressure and temperature effects. We will now discuss the effect of pipe bending moments. In the preceding section on "Flange Stress Limits of NB-3647, Allowable Pipe Bending Stresses at Rated Pressures", we reached the conclusion that several widely used flanges could not take any pipe bending moment, as judged by the NB-3647 criteria.

We will now consider appropriate limits to pipe bending stresses, as limited by flange stresses. We will confine the discussion to welding neck flanges because the limitations will be less severe for straight hub flanges.

First, we note that at the junction of the flange to the pipe, we obtain a stress of S_{pb}. The effect of this load on changing the bolt load is included in the computer program FLANGE; see a_1 in Equation (9) and Table 5. The change in bolt load is theoretically significant but the change in moment acting on the flange ring is small. Because flange stresses are proportional to the moment acting on the flange ring, we can say, to a good first approximation, that the flange stresses are changed, as a result of applying a pipe bending moment, only in the hub region; i.e., the stress range due to pipe bending moment range is simply S_{pb} at the small end of the hub and $S_{pb}(g_o/g_1)$ at the large end of the hub. Considering the pipe bending moment cycle separately, we would only consider the range of S_{pb} at the small end of the hub because the stress range at the large end of the hub is obviously smaller. However, in combining the bolt tightening/untightening cycle, and other operating conditions with the pipe bending moment, the stress at the large end of the hub might be controlling.

As a simple example, let us consider the 24" - 400 lb welding neck flanged joint with S_{pb} (range) limited by the bolts to 24,200 psi (See Table 17, column headed Bolt Stress, Program FLANGE). Let us assume an operating history consisting of a bolt tightening, 100 cycles of S_{pb} = 24,200 psi, and then bolt untightening. Table 18 shows that the stress due to the preload, S_{b1} = 40,000 psi, is equal to 79,020 psi, and is located at the small end of the hub. Accordingly, the usage fraction involves the cycles

Stress Range	Number of Cycles
79,020 + 24,200	1
24,200	99

Assuming the flange material is SA-105 and that S_m = 20,600 psi (at 400 F) for the one major stress cycle, and S_m = 17,300 psi (at 700 F) for the 99 minor stress cycles, then the value of K_e (See NB-3228.3) is 2.35 for the major cycle and 1.00 for the minor cycles. At this point, we must select a suitable stress concentration factor. In general, we are analyzing the flanges and not the girth butt weld between flange and pipe. However, for this example, we will use the K_2-index of 1.8 given in Table NB-3683.2-1 for an "as welded" girth butt weld. The value of S_a' is then (79,020 + 24,200) x 2.35 x 1.8/2 = 218,310 psi for the one major cycle and 24,200 x 1.8/2 = 21,780 psi for the 99 minor cycles. Turning to the Code Figure I-9.1, we find that the usage fraction is 1/80 for the major cycle; 99/60,000 for the minor cycles. The total usage fraction is 0.014 hence, the postulated combination of bolt tightening/untightening with intervening pipe bending stress cycles can be repeated about 70 times in-so-far as flange stress limits are concerned.

There are no test data to check the fatigue life due to bolt tightening/untightening cycles. However, there are some test data given in Appendix A for fatigue under cyclic pipe bending moments. The test data indicate 3,400,000 cycles to failure for an S_{pb} range of 24,200 psi. This, divided by 20 to correspond to the Code factor of safety of 20 on cycles, checks adequately with the Code procedure design life cycles of 60,000.

Summary of Results on Flange Stresses

Calculations were made on a representative sampling of B16.5 flanged joints with:

(a) Rated pressure of ANSI B16.5-1968 for carbon steel flange material; specifically SA-105 for checking Code flange stress criteria

(b) Gasket O.D. = raised face O.D.
Gasket I.D. = pipe O.D.
$m = 2.75$, $y = 3,700$

(c) Bolt material: SA-193 Grade B7

From the results of these calculations, the following conclusions can be drawn.

(1) The number of acceptable flanges with zero pipe bending moment is tabulated below. A total number of 27 flanges of each type were checked.

Code Reference	Temp. F	Number Acceptable			Number Not Acceptable		
		Welding Neck	Straight Hub	Blind	Welding Neck	Straight Hub	Blind
NB-3647.1	100	24	27	23	3	0	4
VIII-Div. 1	100	13	10	4	14	17	23
NB-3647.1	700	24	27	27	3	0	0

(2) The allowable pressures by Code criteria, for those flanges which are not acceptable at the rating pressure, are erratic and in 8 flanged joints the allowable pressure is zero. In view of the satisfactory operating history of B16.5 flanged joints, it is apparent that the Code flange stress criteria are not accurate for evaluating the pressure capacity of B16.5 flanged joints.

(3) Allowable bending stresses in pipe attached to flanged joints, as evaluated by NB-3647.1 flange stress criteria, is zero for 3 flanged joints and very low for most of the flanged joints.

(4) Basic theory calculations of flange stresses with 40,000 psi initial bolt load give high calculated stresses. An explanation of why the flanges are capable of withstanding the bolt loads without gross yielding is presented.

(5) Basic theory calculations indicate that the stresses at points of maximum stress due to bolt load are changed only slightly by internal pressure; they may either decrease or increase.

(6) Application of the NB-3650 type of analysis indicates that, with 40,000 psi initial bolt stress, the fatigue limits of the Code will be satisfied for any ordinarily anticipated loading history.

6. RECOMMENDATIONS

Specific recommended Code changes are included herein as Appendix C. In the following, changes contained in Appendix C are cited, followed by the reason for the recommendation. It is necessary to have the Code at hand to follow the discussion.

(1) Change NB-3651.2 to read:

NB-3651.2 Piping Products for Which Stress Indices Are Not Available. For analysis of flanged joints, see NB-3658. For other piping products for which stress indices are not available, see NB-3680.

Reason: We wish to inform the Code user that, while stress indices are not given for flanged joints, special guidance is given in NB-3658.

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(2) Change NB-3647.1 to read:

NB-3647.1 Flanged Joints.

(a) Flanged joints manufactured in accordance with the standards listed in Table NB-3132-1, as limited by NB-3612.1, shall be considered as meeting the requirements of NB-3640.
(b) Flanged joints not included in Table NB-3132-1 shall be designed in accordance with Article XI - 3000, including the use of the appropriate allowable stress given in Table I-7.

Reason: This subparagraph is written so that it gives requirements for pressure design; analogous to the Code format for all other piping products. The Code user either:

(a) Selects a standard flange with appropriate pressure-temperature ratings, or

(b) Designs the flanges for the design pressure by the Code procedure.

The authors considered deletion of the proposed NB-3647.1(b) because non-standard flanges are rarely used in piping systems. The proposed NB-3647.1(b) is not the same as the present NB-3647.1, even for $P_{eq} = 0$, because no exception is made to Article XI-3000. Article XI-3000 uses stresses for Class 2 or Class 3 components and the stress limits are different (in some cases, more conservative) than the present NB-3647.1(d).

(3) Add a New Paragraph NB-3658 to Read as Follows:

NB-3658 Analysis of Flanged Joints

Flanged joints using flanges, bolting, and gaskets as specified in ANSI B16.5 (1968) and using a bolt material having an S_m - value at 100 F not less than 20,000 psi may be analyzed in accordance with the following rules or in accordance with NB-3200. Other flanged joints shall be analyzed in accordance with NB-3200.

Reason: Analysis requirements for flanged joints, as the Code is presently written, are scattered about and difficult to interpret. The present NB-3647.1, by the last sentence in NB-3647.1(b), invokes operating condition evaluations which, for all other piping products, are handled under NB-3650. The analysis of the flanges (not the bolts or the weld between flanges and pipe) under NB-3650 seems to follow the path:

NB-3651 - No stress indices, go to NB-3680
NB-3680 - go to (d) - Experimental or theoretical analysis
Theoretical analysis - go to NB-3200
Experimental analysis - go to Appendix II

The proposed NB-3658 is intended to gather in one place the rather specialized requirements for the analysis of flanged joints and to give relatively simple and explicit rules for ANSI B16.5 flanged joints.

The first sentence was motivated by:

(a) What few flanged joints are used in Class 1 piping systems are likely to be ANSI B16.5. Accordingly, giving rules for ANSI B16.5 will probably cover most needs. However, they are not mandatory; NB-3200 can be used. The reason for restriction to bolt material with S_m 20,000 psi at 100 F is given in Appendix B.

(b) The data given in this report gives a basis for the proposed analysis rules for ANSI B16.5. However, that basis does not necessarily apply to other standard flanges that are or may later be included in Table NB-3691-1, nor to flanges designed in accordance with Article XI-3000.

NB-3658.1 Design, Normal and Upset Conditions

(a) Bolting. The bolting shall meet the requirements of NB-3232. In addition, the limitations given by Equations (15) and (16) shall be met.

$$M_{fs} \le 3125 \, (S_y/36) C \, A_b \quad , \qquad (15)$$

Reason: The first sentence retains the present Code requirement now contained in NB-3647.1. Specific reference is made to NB-3232 (rather than NB-3230) because it covers normal conditions; NB-3233 simply says: "Use NB-3232 for upset conditions". NB-3234 and NB-3235 are not referred to because the proposed NB-3658.2 and NB-3658.3 cover the emergency and faulted conditions. The quite extensive portion of this report on "Bolt Areas and Bolt Stresses" indicates that the rules of NB-3232 are readily met if the bolts are SA-193 Grade B7. From this standpoint, the first sentence could be deleted and the authors would have no strong objection to doing so. The sentence was left in on the basis that (a) the report evaluations do not cover all conceivable combinations of loadings, and (b) the bolting evaluation is fairly simple to carry out using a computer program such as FLANGE.

The development of Equation (15) is described in Appendix B, where it is identified as Equation (B5). As discussed in Appendix B, Equation (15):

(a) is conservative with respect to available test on joint leakage,

(b) is a good approximation of the moment capability (as limited by joint leakage) as determined by the computer program FLANGE with an assumed bolt prestress of 40,000 psi,

(c) except for the 150 class, gives permissible moments essentially equal to or higher than permitted by the rules in the present NB-3647.1,

(d) tends to discourage the use of the larger sizes of 150 class joints by giving a small permissible moment.

M_{fs} = bending or torsional moment (considered separately) applied to the joint due to weight, thermal expansion of the piping, sustained anchor movements, relief valve steady-state thrust, and other sustained mechanical loads, in.-lb. If cold springing is used, the moment may be reduced to the extent permitted by NB-3672.8.

S_y = yield strength (ksi) of flange material at design temperature (Table I-2.2). The value of $(S_y/36)$ shall not be taken as greater than unity.

C = diameter of bolt circle, in.

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

$$M_{fd} \leq 6250 \, (S_y/36) \, C \, A_b \quad , \qquad (16)$$

where

M_{fd} = bending or torsional moment (considered separately) as defined for M_{fs} but including dynamic loadings, in.-lb.

Reason: The present NB-3647.1 relates P_{eq} to both M and F = axial load. This is the only piping product for which NB-3600 requires consideration of axial loads. Until such time as axial loads are required to be considered for all piping products, it is recommended that F be deleted from consideration for flanged joints.

In the present, NB-3647.1, M is defined as:

M = bending moment applied to the joint due to weight and thermal expansion of the piping, in.-lb.

Several questions arise concerning the definition.

(Q1) Why is relief valve thrust not included? The only application of flanged joints in Class 1 piping known to the authors involves relief valves.

(Q2) What credit, if any, can be taken for "cold springing"

(Q3) Why are torsional moments ignored?

(Q4) Why are dynamic loads (e.g., earthquake) ignored?

(Q5) Why is moment loading due to use of the flanged joint to "pull out" fabrication mismatch ignored? In many commercial piping installations, this may be the major source of moment loading on a flanged joint.

The authors' answers to these questions, and resulting recommendations are:

(A1) The "steady state" relief valve thrust should be included. See also (A4)

(A2) Credit should be given for "cold springing" to the extent provided in NB-3672.8

(A3) As discussed in Appendix B, torsional moments should not be ignored, but they can be considered separately from bending moments and the limit used for bending moments can also be used for torsional moments.

(A4) Presumably the reason dynamic loads were ignored in the definition of M is that they are of short-time duration and flanged-joint leakage would be small during that time. Further, provided these dynamic loads did not break or yield the bolts or flanges, the flanged joint would be "tight" after the dynamic loading stopped. However, there is nothing to guard against a very high dynamic moment which might cause the bolts or flanges to yield or crack. To provide a control for this, Equation (16) is included in the proposed NB-3658.1, to place a limit on static-plus-dynamic moments. The basis of Equation (16) is discussed in Appendix B [identified as Equation (B6)] along with calculated pipe bending stresses resulting from its use. While the equation is quite conservative, it nevertheless permits relatively high dynamic stresses in the attached pipe for all except the 150 class.

(A5) The quality controls for nuclear power plant piping system fabrication should be sufficient to prevent fabrication misalignment pull-out being a major source of moment loading on flanged joints.

(b) Flanges. Flanges of ANSI B16.5 flanged joints meeting the requirements of NB-3612 are not required to be analyzed under NB-3650. However, the pipe-to-flange welds shall meet the requirements of NB-3652, NB-3653, and NB-3654, using appropriate stress indices from Table NB-3683.2-1.

Reason: The first sentence is based on one of the major aspects developed in the report, i.e., that flange stresses, other than at the pipe-to-flange weld area, are due to the bolt prestress load and not due to either pressure or moment loads. This applies even for blind flanges with pressure loading; see Table 25. Accordingly, there is no point in checking the flange stresses under NB-3650 unless we go to the extreme of checking for bolt tightening/untightening cycles. As discussed in the text, any credible number of bolt tightening/untightening cycles gives only a small fatigue usage fraction.

However, as pointed out by the second sentence, the stresses at the welds may vary significantly due to pressure, moments and thermal gradients and hence, should be analyzed just as any other weld in the piping system should be analyzed.

NB-3658.2 Emergency Conditions

(a) The pressure shall not exceed 1.5 times the design pressure.

(b) The limitation given by Equation (17) shall be met.

$$M_{fd} \leq [11250\ A_b - (\pi/16)\ D_f^2\ P_{fd}]\ C\ (S_y/36) \quad , \qquad (17)$$

where

D_f = outside diameter of raised face, in.

P_{fd} = pressure (psi) during the emergency condition concurrent with M_{fd}.

M_{fd}, C, S_y, the limitation on $(S_y/36)$ and A_b are defined in NB-3658.1(a).

(c) Pipe-to-flange welds shall be evaluated by Equation (9) of NB-3652, using a stress limit of 2.25 S_m.

Reason: The pressure limit of 1.5 times the design pressure is analogous to the limit of NB-3655.1. As discussed in Appendix B, the pressure limit is deemed appropriate provided Equation (17) is also invoked. The basis for Equation (17) is discussed in Appendix B, [identified as Equation (B7)] along with numerical examples of its application. The proposed NB-3658.2(c) is the same as the present NB-3655.2.

NB-3658.3 Faulted Conditions

(a) The pressure shall not exceed 2.0 times the design pressure.

(b) The limitation given by Equation (17) of NB-3658.2(b) shall be met, where P_{fd} and M_{fd} are pressures (psi) and moments (in.-lb) occurring concurrently during the faulted condition.

(c) Pipe-to-flange welds shall be evaluated by Equation (9) of NB-3652, using a stress limit of 3.0 S_m.

Reason: The pressure limit of 2.0 times the design pressure is analogous to the limit of F-1360(a). As discussed in Appendix B, the pressure limit is deemed appropriate provided Equation (17) is also invoked. It may be noted that NB-3656, Consideration of Faulted Conditions, refers the Code user to Appendix F of the Code. However, the guidance given by Appendix F for flanged joints is nebulous, at best, and hence, it was deemed appropriate to supply specific rules for flanged joints under faulted conditions. The Code user may use Appendix F, through the pathway of NB-3200 rules.

NB-3658.4 Testing Conditions. Analysis for Testing Conditions is not required.

Reason: At present, NB-3657, "Testing Conditions", refers the Code user to NB-3226. The requirements of NB-3226 are not relevant to flanged-joints. In testing flanged joints, one normally is looking for leakage at the gasket; if the leakage is deemed excessive, the normal next step is to increase the preload on the bolts. There is a control on maximum preload through the requirements in NB-3232.1 that average bolt stress shall not exceed two times the allowable bolt stress. Accordingly, analysis for testing conditions is not deemed to be needed.

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(1) NC-3651, Add:

(c) For analysis of flanged joints, see NC-3658.

(2) NC-3671.1, Delete Period After "NC-3647" and Add:

and NC-3658.

Reason: We wish to inform the Code user that special guidance for flanged joints is given in NC-3658. It may be noted that we jump from NC-3652 to NC-3658. This is to provide consistency in numbering with the NB portion of the Code.

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(3) NC-3673.5(a)

Change equation numbers: (12) to (15) and (13) to (16).

(4) NC-3673.5(b)

Change "Formulas (12) and (13)" to "Equations (15) and (16)".

Reason: Editorial to introduce new equations and their numbers in NC-3658.

---

(5) Change NC-3647.1 to Read:

NC-3647.1 Flanged Joints.

(a) Flanged joints manufactured in accordance with the standards listed in accordance with the standards listed in Table NC-3132-1, as limited by NC-3612.1, shall be considered as meeting the requirements of NC-3640.

(b) Flanged joints not included in Table NC-3132-1 shall be designed in accordance with Article XI - 3000.

Reason: To give requirements for pressure design under NC-3640, "Pressure Design of Piping Products", and eliminate those requirements which should go under NC-3650, "Analysis of Piping Systems".

---

(6) Add a New Paragraph NC-3658 to Read as Follows:

NC-3658 Analysis of Flanged Joints

Flanged joints using flanges, bolting, and gaskets as specified in ANSI B16.5 (1968) and using a bolt material having an S - value at 100 F not less than 20,000 psi may be analyzed in accordance with the following rules or in accordance with NB-3200. Other flanged joints shall be analyzed in accordance with NB-3200. If the NB-3200 analysis is used, allowable stresses from Table I-7 and I-8 shall be used in place of S_m.

Reason: To gather into one place the rather specialized requirements for the analysis of flanged joints and to give relatively simple and explicit rules for ANSI B16.5 flanged joints.

A question arises as to what analysis should be made for flanged joints not conforming to the first sentence. While most flanged joints are B16.5 and SA-193 Grade B7 bolts are commonly used, in Class 2 piping systems there may be other types of joints; particularly in sizes larger than 24". The present rules in NC-3650 are not relevant to any kind of flanged joint. Code case 1744 provides some assistance but it is limited to maximum pressures of 100 psi and maximum temperatures of 200 F. For lack of a better available solution, the last two sentences were included which, in principle, will give guidance for any type of flanged joint.

NC-3658.1 Design, Normal and Upset Conditions

(a) The limitations given by Equations (12) and (13) shall be met.

$$M_{fs} \leq 3125\ (S_y/36)\ C\ A_b \qquad (12)$$

where

M_{fs} = bending or torsional moment (considered separately) applied to the joint due to weight, thermal expansion of the piping, sustained anchor movements, relief valve steady-state thrust and other sustained mechanical loads, in.-lb. If cold springing is used, the moment may be reduced to the extent permitted by NC-3673.5.

S_y = yield strength (ksi) of flange material at design temperature (Table I-2.2). The value of $(S_y/36)$ shall not be taken as greater than unity.

C = diameter of bolt circle, in.

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

$$M_{fd} \leq 6250\ (S_y/36)\ C\ A_b\ , \qquad (13)$$

where

M_{fd} = bending or torsional moment (considered separately) as defined for M_{fs} but including dynamic loadings, in.-lb.

Reason: At present, NC-3600 does not define or use normal conditions, upset conditions, emergency conditions or faulted conditions. However, work by Code committees is in advanced stages in which these terms (or, more likely, their equivalents called "design limits") will be used for both Class 2 and Class 3 components. The recommendations have been written on the assumption that the revised rules will be implemented in the near future.

It may be noted that a requirement analogous to that in NB-3658.1 of "The bolting shall meet the requirements of NB-3232" is not imposed. This is deemed to be consistent with Class-2 design philosophy in which cyclic stresses, other than those due to cyclic moments, are not considered. Otherwise, NC-3658.1 is essentially identical to NB-3658.1 and the same reasons apply. It may be noted that the allowable bolt stresses in Class 2 are lower than in Class 1. However, this is not relevant to the problem. The relevant aspect concerns the assumed preload stress of 40,000 psi and the assumption is deemed to be equally valid for Class 1 and Class 2.

(b) Flanges. Flanges of ANSI B16.5 flanged joints meeting the requirements of NC-3612.1 are not required to be analyzed under NC-3650. However, the pipe-to-flange welds shall meet the requirements of NC-3652, using appropriate stress intensification factors from Figure NC-3673.2(b)-1.

Reason: See those cited under NB-3658.1(b). The second sentence is appropriate for Class 2 piping, which uses stress intensification factors.

NC-3658.2 Emergency Conditions

(a) The pressure shall not exceed 1.5 times the design pressure.

(b) The limitation given by Equation (14) shall be met.

$$M_{fd} \leq [11250\ A_b - (\pi/16)\ D_f^2\ P_{fd}]\ C\ (S_y/36) \quad , \quad (14)$$

where

D_f = outside diameter of raised face, in.

P_{fd} = pressure (psi) during the emergency condition concurrent with M_{fd}.

M_{fd}, C, S_y, the limitation on $(S_y/36)$ and A_b are defined in NC-3658.1(a).

(c) Pipe-to-flange welds shall be evaluated by Equation (9) of NC-3652, using a stress limit of 1.8 S_h.

Reason: The pressure limit in (a) and the stress limit in (c) conform to the general limitations for emergency conditions being established for Class 2 piping. The reason for (b) is discussed under NB-3658.2(b).

NC-3658.3 Faulted Conditions

(a) The pressure shall not exceed 2.0 times the design pressure.
(b) The limitation given by Equation (14) shall be met, where P_{fd} and M_{fd} are pressures (psi) and moments (in.-lb) occurring concurrently during the faulted condition.
(c) Pipe-to-flange welds shall be evaluated by Equation (9) of NC-3652, using a stress limit of 2.4 S_h.

Reason: The pressure limit in (a) and stress limit in (c) conform to the general limitations for faulted conditions being established for Class 2 piping. The reason for (b) is discussed under NB-3658.3(b).

NC-3658.4 Testing Conditions. Analysis for Testing Conditions is not required.

Reason: Analysis for testing conditions is not addressed in the present NC-3600. Testing conditions will be identified in the rules under development but, as presently drafted, analysis for testing conditions is not required.

ND-3600

It is recommended that ND-3600 be changed as shown for NC-3600; changing NC to ND in all places where NC occurs. In ND-3673.5, the present equation numbers (9) and (10) are to be changed to (15) and (16) in two places. Do not change the phrase: "using Equations (9) and (10) (ND-3650)."

Reason: For the purpose of these recommendations, ND-3600 (Class 3 piping) is identical to NC-3600 (Class 2 piping). The only exception is equation numbers which, if the recommendations are followed, will make ND-3600 and NC-3600 identical in this respect.

7. ACKNOWLEDGEMENT

The author wishes to thank the members of Task Group 1 of the Pressure Vessel Research Committee, Subcommittee on Piping, Pumps and Valves, and the members of the ASME Boiler and Pressure Vessel Code Committee, Subgroup on Piping (SGD) (SCIII), for their review and valuable suggestions. The authors wish to express their appreciation to Mr. E. B. Branch, Mr. J. T. McKeon, Mr. F. M. O'Hara, Jr. and Mr. F. E. Vinson for their many valuable comments and suggestions.

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(14) Tube Turns Report No. 2.036, "Effect of Flange Material Yield Strength on the Pressure Capacity of a Flanged Joint", April 25, 1958.

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APPENDIX A

TEST DATA ON FLANGED JOINTS

TABLE OF CONTENTS

INTRODUCTION

The text of this report gives results of calculations using the computer program FLANGE*. The theoretical basis for this program is given in Reference (3). Essentially, it is based on a shell and plate discontinuity analysis and, as applied to actual flanged joints, cannot be expected to be very accurate; particularly with respect to flange stresses. There are larger uncertainties involved in the selection of load application lines; i.e., "effective gasket diameters" and "effective bolt circle diameters". For any but flat metal gaskets, there are major uncertainties concerning the "effective modulus of elasticity" of the gasket.

In this Appendix, available test data are presented which give some indication of the effects of the variables mentioned above. The data are for internal pressure loading and, to a limited extent, for internal pressure combined with a pipe bending moment. No test data are available on thermal gradients in flanged joints.

TEST DATA ON DISPLACEMENTS, STRAINS, AND STRESSES

Because flanged joints are statically redundant structures, their theoretical analysis involves the rotations of the flanges due to the loads (bolt load, pressure, thermal gradients) and the axial strains in the bolts. Three sets of data which can be used to check the validity of the theory as used in the computer program FLANGE are discussed in this section.

---

* All calculations made with FLANGE used the hub-to-ring boundary condition represented by the option IBOND = 2 in FLANGE. This option, in general, gives better correlation with test data than does the hub-to-ring boundary condition used in the Code method (IBOND = 0 option in FLANGE).

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(1) Tests on Heat Exchanger Flanges with Metal Gasket[7]

These data are closest to "ideal" for the purpose of checking the theory because a narrow metal gasket was used; thus reducing uncertainties concerning the "effective gasket diameter" and modulus of elasticity of the gasket material.

(2) Tests on Large Flanges with O-Ring Gasket[8]

While a flanged joint without bolt preload is a statically determinate structure, these tests involved a moderate preload; hence, the interaction effects can be evaluated for low pressures. Further, at higher pressures, flange rotations and flange stresses for the statically-determinant loads can be compared with theoretical predictions. The data include the best available measurements of stresses in a welding neck flange. In the statically redundant portion of the tests, an effective gasket diameter must be selected; this is difficult because the raised face is very wide compared to the radial distance between the center of the raised face and the bolt circle.

(3) Tests on ANSI B16.5 Flanges With Asbestos Gaskets [9]

The leakage aspect of these tests are discussed later under "Leakage Pressure Tests" and details of the gaskets are given thereunder. In this section, the change of bolt stress with pressure is discussed. The major uncertainty in these tests consists of selection of an appropriate effective modulus of elasticity for the gaskets under the test conditions.

<u>Tests on Heat Exchanger Flanged Joint with a Metal Gasket</u>

Reference (7) gives test data for the flanged joint shown in Figure A1. Experimental results consist of:

FLANGE C-1 FLANGE C-2

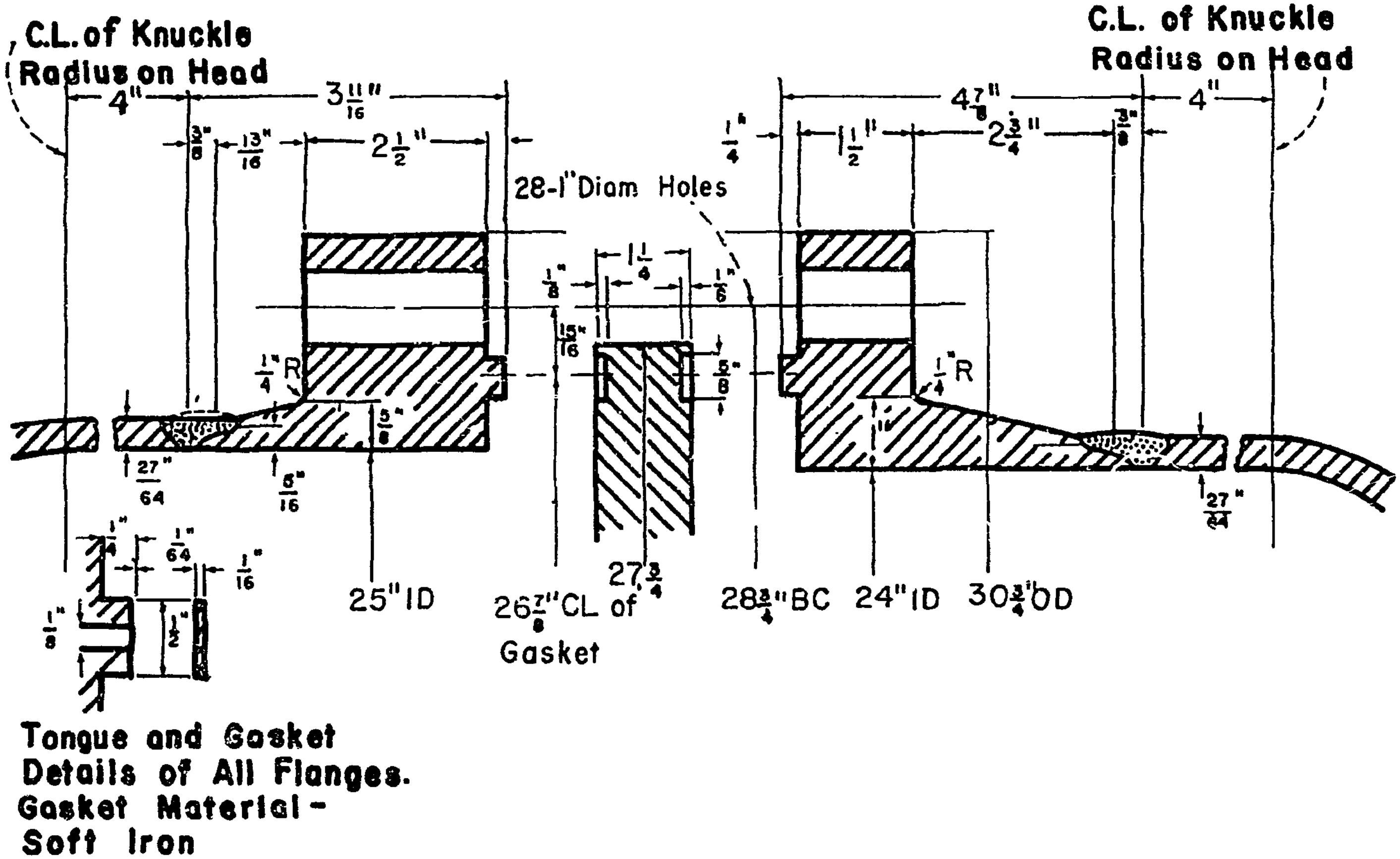

FIGURE A1. DETAILS OF FLANGED JOINT, REFERENCE (7).

(1) Plot of bolt elongation versus internal pressure. Elongation was measured with calipers. Considerable scatter exists in the test data, as might be expected using this method.
(2) Flange rotation due to internal pressure.
(3) Flange rotation due to bolt load.

The printout from the computer program FLANGE for the flanged joint is shown in Table A1. The input data to FLANGE (printed out on the first page of Table A1) is based on:

(a) Dimensions as shown in Figure A1.
(b) Modulus of elasticity = 30×10^6 psi for flanges, bolts, and gasket
(c) Bolt area = 28 (bolts) x 0.419 (root area per bolt) = 11.732 sq. in.
(d) Pressure = 1,000 psi, Moment = $h_G \times A_b \times 40{,}000$ psi = 439,950 in.-lb, W1 = initial bolt load = 40,000 A_b = 469,280 lb.

The somewhat arbitrary aspect of the input consists of appropriate modeling of the plate between the flanges so as to incorporate the elastic response of the plate. This was done by using gasket dimensions of XGO = 27.750" , XGI = 26.000" and G_m = actual mean gasket diameter = 26.875"; gasket thickness (VO) of 1.500". The gasket thickness is equal to the plate thickness, gasket O.D. = plate O.D. and gasket I.D. arbitrarily set at 26" as representing that part of the plate which deforms elastically under load.

Change in Bolt Length with Pressure

The test data are shown in Figure A2, along with the calculated result obtained by the following equation.

$$\delta_b = \frac{(W1-W2)}{1000} \frac{P}{A_b} \frac{\ell}{E} = 5.675 \times 10^{-6} P$$

TABLE A1: OUTPUT FROM COMPUTER PROGRAM FOR HEAT EXCHANGER FLANGED JOINT, FIGURE A1 (SHEET 1 OF 4).

```
FLANGE     FLANGE      FLANGE      PIPE       HUB AT      HUB         BOLT       PRESSURE,
O.D.,A     I.D.,B      THICK.,T   WALL,GO     BASE,G1     LENGTH,H   CIRCLE,C      P
30.75000   25.00000    2.50000    0.42200     0.62500     0.81250    28.75000    1000.000

MOMENT   COEFF. OF    DELTA      MOD. OF MEAN GASKET   ITYPE       IBOND      ICODE      MATE
         THERMAL EXP.            ELASTICITY DIAMETER
4.3590 05 6.0000-06 1.0000D-03 3.0000D 07 2.6880 01      1           2          0          3

FLANGE     FLANGE      FLANGE      PIPE       HUB AT      HUB         BOLT       PRESSURE,
O.D.,A     I.D.,B      THICK.,T   WALL,GO     BASE,G1     LENGTH,H   CIRCLE,C      P
30.75000   24.25000    1.50000    0.42200     1.00000     2.75000    28.75000    1000.000

MOMENT   COEFF. OF    DELTA      MOD. OF MEAN GASKET   ITYPE       IBOND      ICODE      MATE
         THERMAL EXP.            ELASTICITY DIAMETER
4.3590 05 6.0000-06 1.0000D-03 3.0000D 07 2.6880 01      1           2          0          4

   BSIZE            YB              EB              TB              XGO             XGI             AB
 8.75000D-01     3.00000 07      6.00000-06      0.0             2.7750D 01      2.60000 01      1.1732D 01
   VO               YG              EG              TG              FACE            PBE
 1.50000 00      3.00000 07      6.00000-06      0.0             0.0             0.0
   W1               TF              TFP             YF2             YFP2            YB2             YG2
 4.69280 05      0.0             0.0             3.00000 07      3.00000 07      3.00000 07      3.00000 07

   FLANGE JOINT BOLT LOAD CHANGE DUE TO APPLIED LOADS. INTEGER TO INTEGER PAIR

      FLANGE JOINT SIDE ONE (PRIMED QUANTITIES)

QFHG=  7.39020-09  QPHG=  1.58100-06  QTHG=  1.52740-05   XB =  2.50000 01      GO=  4.22000-01       TH  =  2.50000 00
      YH =  3.00000 07          YF2 =  3.00000 07     EF =  6.00000-06

      FLANGE JOINT SIDE TWO (UNPRIMED QUANTITIES)

QFHG=  8.36030-09  QPHG=  1.56720-06  QTHG=  2.04480-05   XB =  2.42500 01      GO=  4.22000-01       TH  =  1.50000 00
      YH =  3.00000 07          YF2 =  3.00000 07     EF =  6.00000-06

        BOLTING

BOLT LENGTH=  6.37500 00  BOLT AREA=  1.17320 01  BOLT CIRCLE=  2.87500 01
      YB =  3.00000 07        YB2 =  3.00000 07     EB =  6.00000-06

        GASKET

 VO  =  1.50000 00  XGO =  2.77500 01  XGI =  2.60000 01
    YG  =  3.00000 07        YG2 =  3.00000 07     EG =  6.00000-06
```

TABLE A1: SHEET 2 OF 4.

```
      LOADINGS

INITIAL BOLT LOAD=  4.6928D 05  BOLT TEMP.=  0.0          FLANGE ONE TEMP.=  0.0          FLANGE TWO TEMP.=  0.0
GASKET TEMP.=  0.0          DELTA=  1.0000D-03   DELTAP=  1.0000D-03  PRESSURE=  1.0000D 03

   RESIDUAL BOLT LOADS AFTER THERMAL-PRESSURE LOADS

AXIAL THERMAL,W2A=  4.6928D 05  MOMENT SHIFT,W2B=  2.5031D 05

TOTAL PRESSURE,W2C=  1.5649D 05   DELTA THERMAL,W2D=  4.6928D 05

COMBINED,W2=  1.5649D 05

W1-W2A=  0.0         W1-W2B=  2.1897D 05  W1-W2C=  3.1279D 05  W1-W2D=  1.0646D 00  W1-W2=   3.1279D 05

W2A/W1=  1.0000D 00  W2B/W1=  5.3339D-01  W2C/W1=  3.3347D-01  W2D/W1=  1.0000D 00  W2/W1=  3.3347D-01

    INITIAL AND RESIDUAL MOMENTS AFTER THERMAL PRESSURE LOADS.

M1=  4.3995D 05  M2A=  4.3995D 05  M2B=  8.1258D 05  M2C=  7.2462D 05  M2D=  4.3995D 05  M2=  7.2462D 05

M2BP=  6.2709D 05  M2CP=  5.3914D 05  M2P=  5.3914D 05
```

TABLE A1: SHEET 3 OF 4.

```
       TAPERED HUB FLANGE
   CALCULATIONS FOR MOMENT LOADING

SLSO=  2.9370D 04  SLSI= -2.9370D 04  SCSO=  2.1282D 04  SCSI=  3.6601D 03
SLLO=  2.6418D 04  SLLI= -2.6418D 04  SCLO=  1.5918D 04  SCLI=  6.7218D 01
 STH=  9.0742D 03   STF= -1.3862D 04   SRH=  3.6057D 03   SRF= -2.6284D 03
ZG= -3.2936D-03  ZC= -6.5445D-03  OFHG=  3.2509D-03  YO=  5.1963D-03  Y1=  3.3302D-03  THETA= -5.5111D-03
   CALCULATIONS FOR PRESSURE LOADING

SLSO=  1.3738D 04  SLSI=  1.5983D 04  SCSO=  1.9611D 04  SCSI=  2.0255D 04
SLLO=  4.1257D 03  SLLI=  1.5875D 04  SCLO=  1.2882D 04  SCLI=  1.6407D 04
 STH=  1.0972D 04   STF=  9.6346D 02   SRH= -2.2395D 03   SRF= -1.9665D 02
ZG= -1.6355D-03  ZC= -3.2175D-03  OFHG=  1.5810D-03  YO=  6.4542D-03  Y1=  4.8517D-03  THETA= -1.7703D-03
   CALCULATIONS FOR TEMPERATURE LOADING

SLSO=  3.2681D-03  SLSI= -3.2681D-03  SCSO= -7.9635D-02  SCSI= -8.1396D-02
SLLO= -4.6835D-02  SLLI=  4.6809D-02  SCLO= -1.3070D-01  SCLI= -1.0262D-01
 STH=  5.9686D-02   STF= -3.7010D-02   SRH= -1.2182D-02   SRF=  7.5540D-03
ZG= -1.5911D-08  ZC= -3.1085D-08  OFHG=  1.5274D-08  YO= -3.3590D-08  Y1= -4.8608D-08  THETA= -1.7103D-08
  CALCULATIONS FOR COMBINED LOADING,M2 OR M2P FOR ITYPE=1 OR 2, W2 FOR ITYPE=3,  =   5.3914D 05

SLSO=  4.9730D 04  SLSI= -2.0109D 04  SCSO=  4.5692D 04  SCSI=  2.4740D 04
SLLO=  3.6499D 04  SLLI= -1.6499D 04  SCLO=  3.2388D 04  SCLI=  1.6489D 04
 STH=  2.2092D 04   STF= -1.6024D 04   SRH=  2.1791D 03   SRF= -3.4176D 03
ZG= -5.6727D-03  ZC= -1.1238D-02  OFHG=  5.5649D-03  YO=  1.2822D-02  Y1=  8.9327D-03  THETA= -6.0730D-03
```

TABLE A1: SHEET 4 OF 4.

```
       TAPERED HUB FLANGE
   CALCULATIONS FOR MOMENT LOADING

SLSO=  1.2404D 04  SLSI= -1.2404D 04  SCSO=  1.4247D 04  SCSI=  6.8043D 03
SLLO=  2.1856D 04  SLLI= -2.1656D 04  SCLO=  7.5797D 03  SCLI= -5.5337D 03
 STH=  5.3361D 03   STF= -1.5339D 04   SRH=  1.4377D 04   SRF= -1.2045D 04
ZG= -4.8318D-03  ZC= -8.5095D-03  QFHG=  3.6777D-03  YO=  4.2540D-03  YI=  4.1345D-04  THETA= -3.4350D-03
   CALCULATIONS FOR PRESSURE LOADING

SLSO=  1.9024D 04  SLSI=  9.7081D 03  SCSO=  2.6149D 04  SCSI=  2.3354D 04
SLLO=  5.2105D 03  SLLI=  6.9141D 03  SCLO=  9.9806D 03  SCLI=  1.0492D 04
 STH=  7.8672D 03   STF=  1.6893D 03   SRH= -1.8339D 03   SRF= -3.9380D 02
ZG= -2.2851D-03  ZC= -3.8523D-03  QFHG=  1.5672D-03  YO=  8.2618D-03  YI=  3.4020D-03  THETA= -1.7810D-03
   CALCULATIONS FOR TEMPERATURE LOADING

SLSO=  8.6395D-02  SLSI= -8.6395D-02  SCSO=  3.5650D-02  SCSI= -1.6187D-02
SLLO=  9.9963D-03  SLLI= -9.9963D-03  SCLO= -1.1271D-01  SCLI= -1.1871D-01
 STH=  6.0091D-02   STF= -2.0516D-02   SRH= -1.4008D-02   SRF=  4.7824D-03
ZG= -2.9815D-08  ZC= -5.0263D-08  QFHG=  2.0448D-08  YO=  3.9330D-09  YI= -4.6765D-08  THETA= -2.3238D-08
  CALCULATIONS FOR COMBINED LOADING,M2 OR M2P FOR ITYPE=1 OR 2,  M2 FOR ITYPE=3,  =   7.2462D 05

SLSO=  3.9454D 04  SLSI= -1.0722D 04  SCSO=  4.9614D 04  SCSI=  3.4561D 04
SLLO=  4.1208D 04  SLLI= -2.9083D 04  SCLO=  2.2465D 04  SCLI=  1.3771D 03
 STH=  1.6656D 04   STF= -2.3575D 04   SRH=  2.1846D 04   SRF= -2.0233D 04
ZG= -1.0243D-02  ZC= -1.7868D-02  QFHG=  7.6245D-03  YO=  1.5268D-02  YI=  4.0829D-03  THETA= -7.4387D-03
```

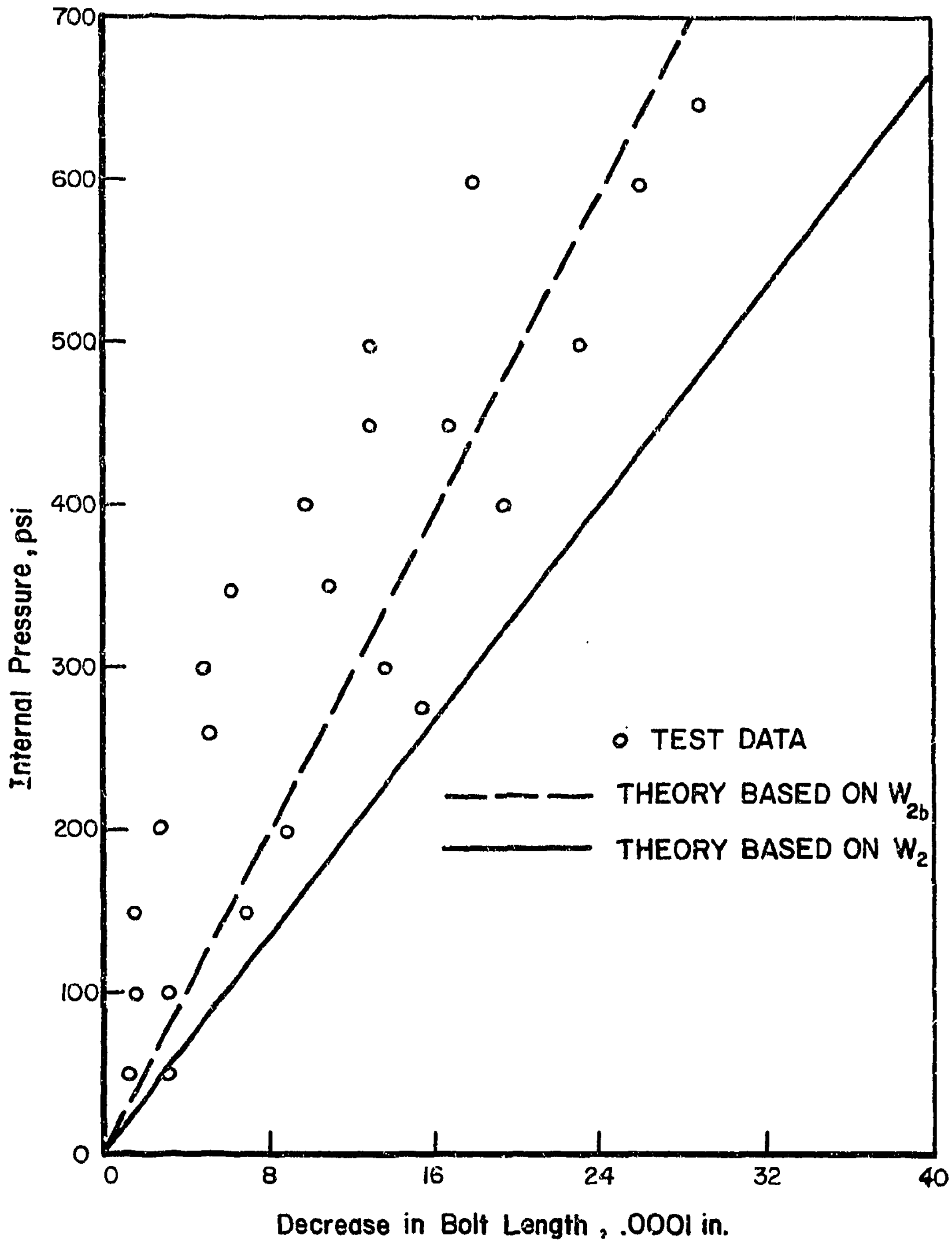

FIGURE A2. COMPARISON OF THEORETICAL AND MEASURED CHANGE IN BOLT LENGTH WITH PRESSURE, REFERENCE (7) TEST DATA.

where δ_b = decrease in bolt length, (W1-W2) = change in bolt load due to 1,000 psi pressure = 314,900 lb as calculated by FLANGE, P = internal pressure, A_b = bolt area = 11.732 sq in. and ℓ = effective bolt length* = 6.125", E = 30 x 10^6 psi.

The theoretical change in bolt length with pressure is shown in Figure A2. It can be seen that the theory overpredicts the rate of decrease of bolt load. However, examination of Figure A1 indicates that the full radial effect of internal pressure would not be expected to fully develop in the test assembly. For this effect to fully develop, the length of cylindrical shell between the flange and knuckle radius on the head should be greater than about $4\sqrt{dt}$ = $4\sqrt{25 \times .422}$ = 13". This length in the test assembly was only 4". Accordingly, the head probably restricted the radial expansion of the cylindrical shell. The theoretical change in bolt load, excluding the radial effect of internal pressure, is shown in Table A1 as (W1-W2B). Using this value gives the dashed line shown in Figure A2; this agrees reasonably well with the test data.

Flange Rotation due to Bolt Load

Test data are shown in Figure A3 and A4 for flanges C-1 and C-2, respectively. The theoretical rotation for the input moment of

* The effective bolt length is taken as the length between nut surfaces plus one bolt diameter.

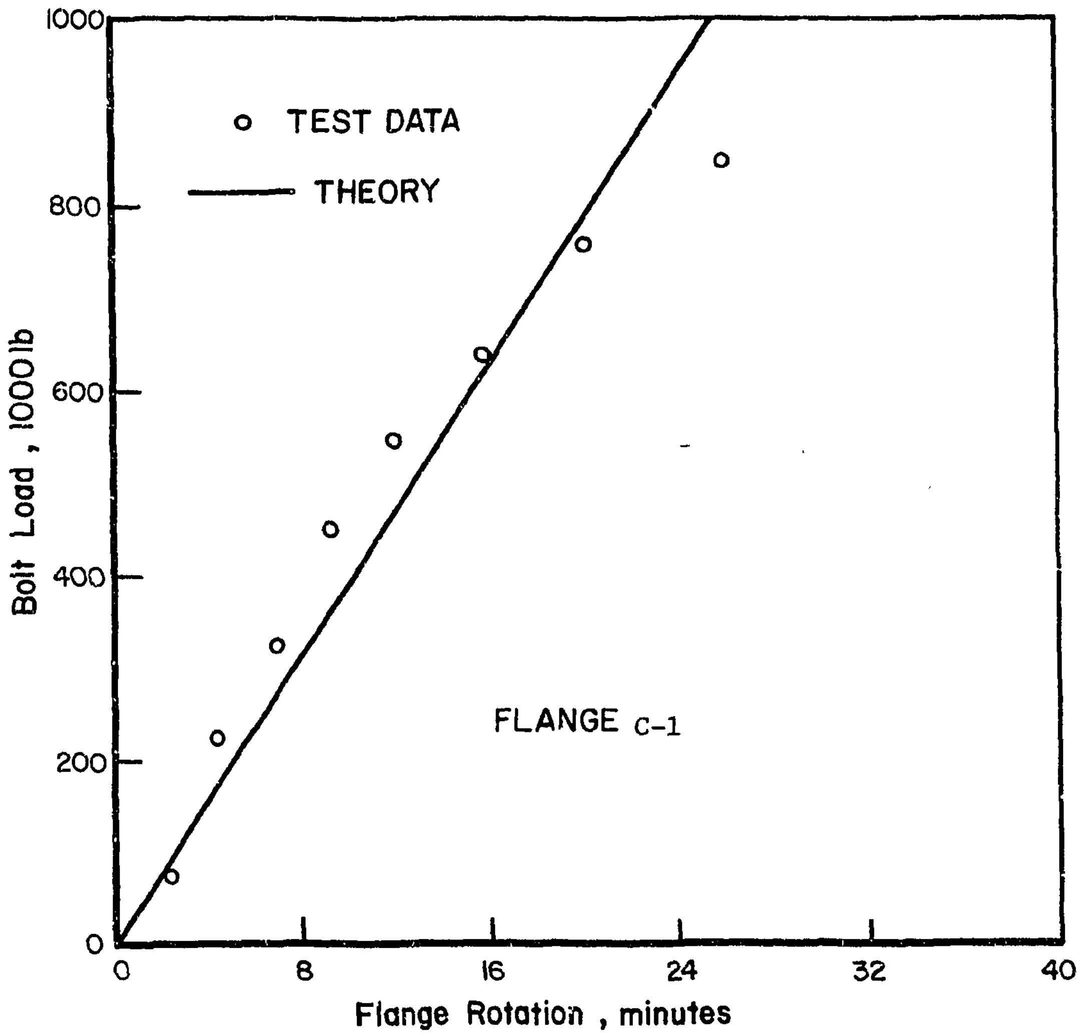

FIGURE A3. COMPARISON OF THEORETICAL AND MEASURED FLANGE ROTATIONS DUE TO BOLT LOAD, FLANGE C-1 REFERENCE (7) TEST DATA

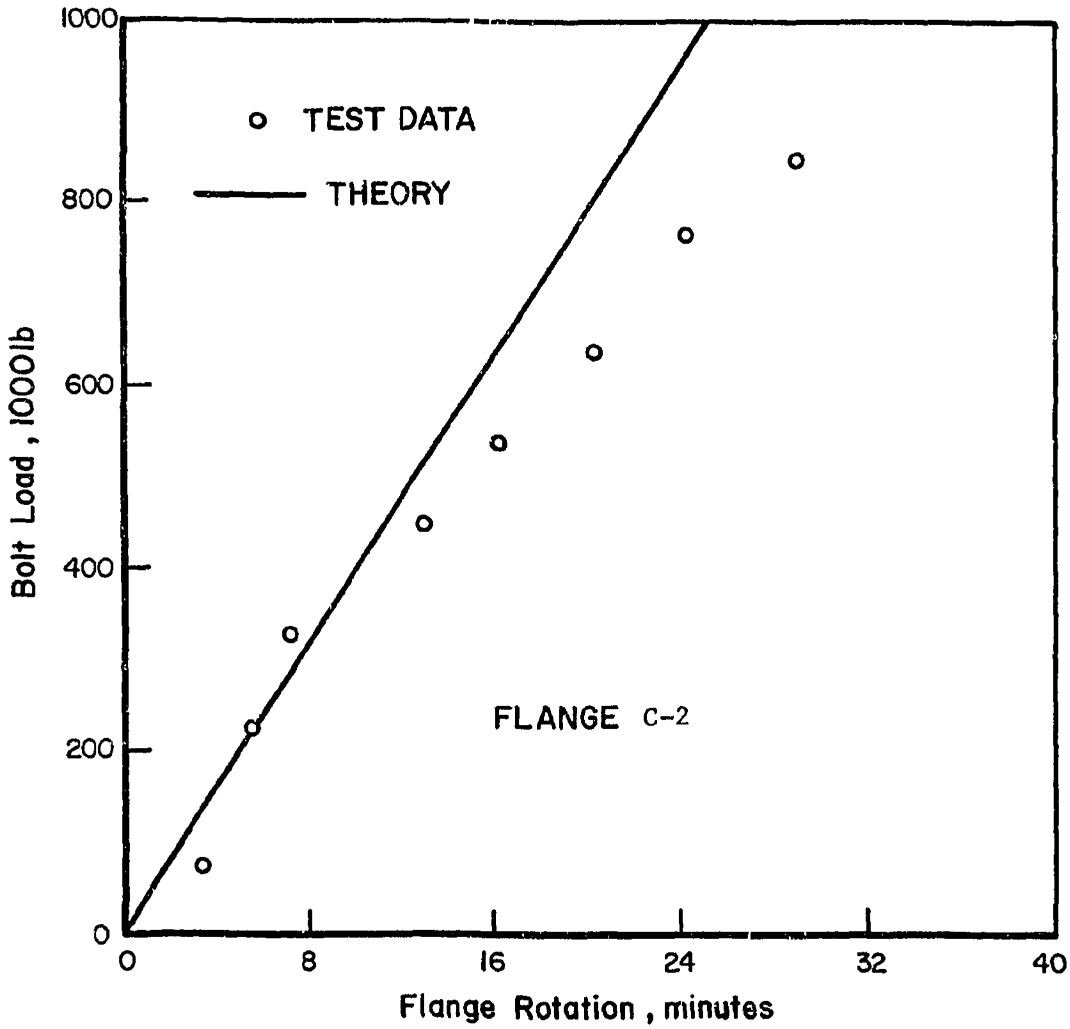

FIGURE A4. COMPARISON OF THEORETICAL AND MEASURED FLANGE ROTATIONS DUE TO BOLT LOAD, FLANGE C-2 REFERENCE (7) TEST DATA

439,900 in.-lb is shown in Table A1 as THETA = 3.5115×10^{-3} and 3.4345×10^{-3} for flanges C1 and C2, respectively.* With $M = Wh_G$, where h_G = 0.9375" for this joint:

$$\theta = \frac{\theta}{M} Wh_G$$

$$\theta = \frac{3.5115 \times 10^{-3}}{439,900} \times 0.9375 \text{ W radians}$$

$$\theta = 7.484 \times 10^{-3} \text{W radians} = 2.573 \times 10^{-5} \text{ W minutes}$$

for flange C-1, and similarly for flange C-2

$$\theta = 2.516 \times 10^{-3} \text{ W minutes.}$$

These theoretical lines are shown in Figures A3 and A4. The test data and theory agree reasonably well at low loads. At higher loads, the test data indicate** that yielding of the flanges might have occurred. The calculated stresses are high enough to indicate the possibility of such yielding.

---

* The negative signs in Table A1 for "THETA" is the sign convention used in the basic theory indicating rotation in the direction of moment due to bolt load.

** Shifting of the bolt load inward or gasket effective diameter outward would also produce a non-linear effect; however, the curvature of the test data is opposite to what would occur from this effect.

The tests were apparently run by increasing the bolt load in steps and, at each step increasing the pressure in steps; perhaps as high as 650 psi with 850,000 lb bolt load. Under these loads, the maximum stresses are:

Flange C-1, SLSO (Longitudinal, small end of hub, outside

$$SLSO = \frac{29,374}{439,900} \times 850,000 \times .9375 + \frac{13,738}{1,000} \times 650 = 62,140 \text{ psi}$$

Flange C-2, SLLO (longitudinal, large end of hub, outside surface)

$$SLLO = \frac{21,858}{439,900} \times 850,000 \times .9375 + \frac{5,211}{1,000} \times 650 = 42,980 \text{ psi.}$$

The materials used for the flanges and attached shell are not described in Reference (7). Presumably they were carbon steel with a yield strength in the range of 30,000 to 50,000 psi.

Flange Rotation Due to Pressure

Test data are shown in Figure A5 and A6 for flanges C-1 and C-2, respectively. The theoretical increase in flange rotation due to internal pressure is given by:

$$\Delta\theta = \frac{\theta p}{P} \Delta P + \frac{\theta m}{M} \frac{\Delta M}{P} \Delta P \qquad \text{(A1)}$$

From Table A1, we obtain:

Flange	θ_p/P	θ_m/M	$\Delta M/P$
C-1	$\frac{1.7703 \times 10^{-3}}{1000}$	$\frac{3.5115 \times 10^{-3}}{439900}$	$\frac{(5.3867-4.3995) \times 10^{5}}{1000}$
C-2	$\frac{1.7810 \times 10^{-3}}{1000}$	$\frac{3.4354 \times 10^{-3}}{439900}$	$\frac{(7.2416-4.3995) \times 10^{5}}{1000}$

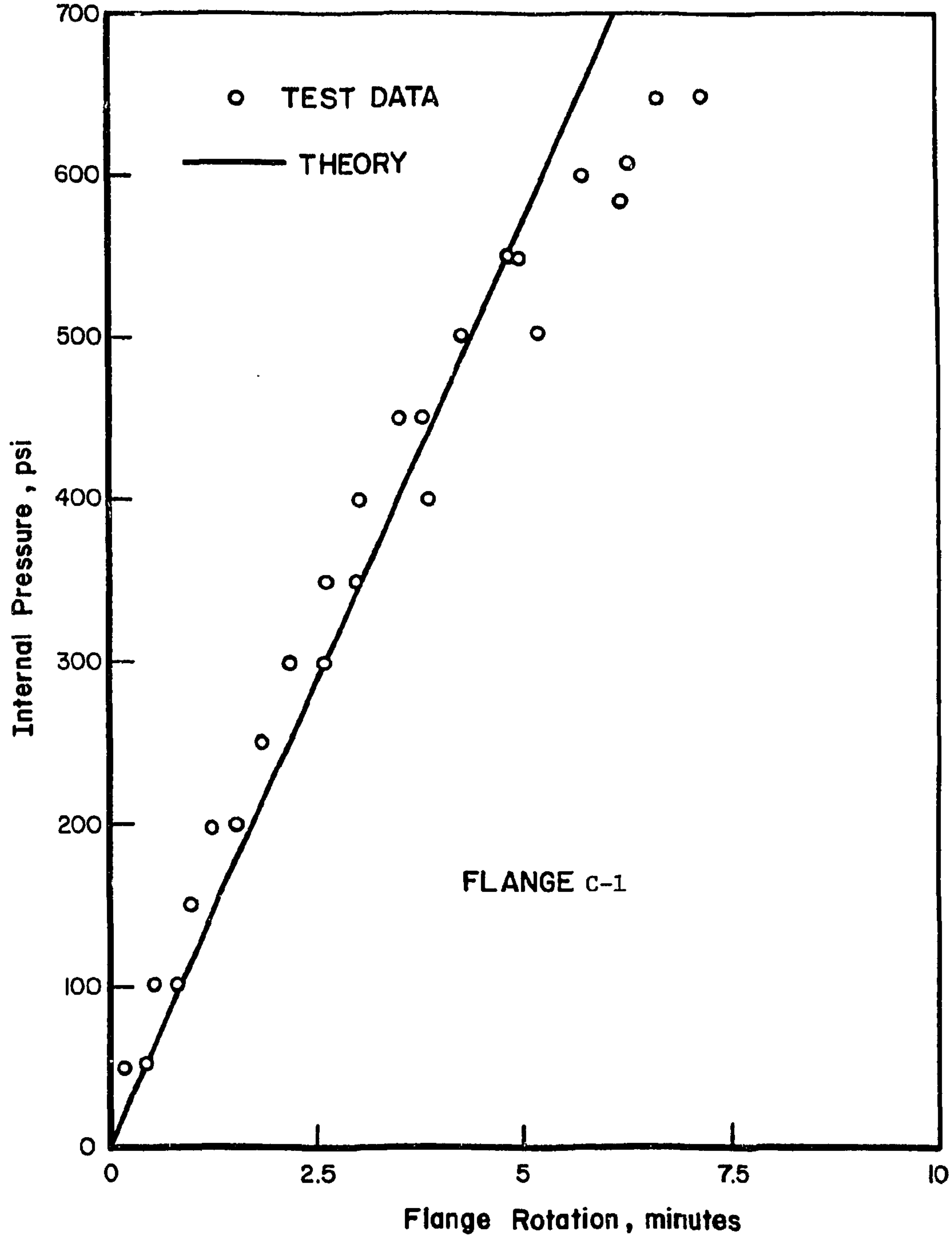

FIGURE A5. COMPARISON OF THEORETICAL AND MEASURED FLANGE ROTATIONS DUE TO INTERNAL PRESSURE, FLANGE C-1, REFERENCE (7) TEST DATA

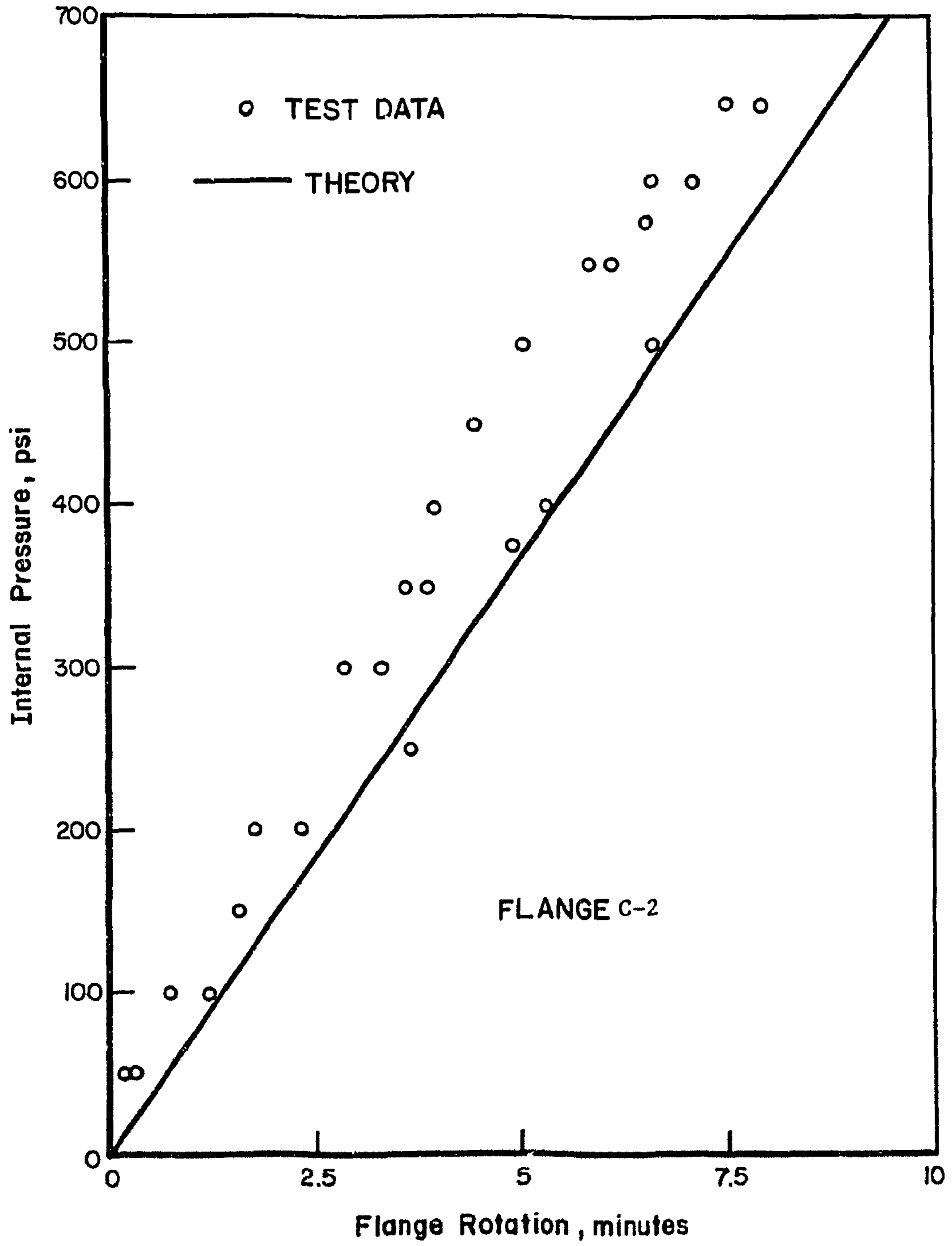

FIGURE A6. COMPARISON OF THEORETICAL AND MEASURED FLANGE ROTATIONS DUE TO INTERNAL PRESSURE, FLANGE C-2, REFERENCE (7) TEST DATA

Using these values in Equation (A1) gives:

For Flange C-1

$$\Delta\theta = (1.770 \times 10^{-6} + 7.982 \times 10^{-9} \times 98.72)\Delta P = 2.558 \times 10^{-6} \ \Delta P \text{ radians}$$

$$= 0.00879 \ \Delta P \text{ minutes}$$

For Flange C-2

$$\Delta\theta = (1.781 \times 10^{-6} + 7.810 \times 10^{-9} \times 284.2) \ \Delta P = 4.006 \times 10^{-6} \ \Delta P \text{ radians}$$

$$= 0.0138 \ \Delta P \text{ minutes}$$

These lines are shown in Figures A5 and A6. The theory is in good agreement with the test data for flange C-1, with again some test data evidence of yielding at high pressure (actually, high pressure combined with high bolt load). The theory over predicts the flange rotation of flange C-2. The test data do not indicate yielding of flange C-2.

Tests on a Large Flanged Joint with O-Ring Gasket

Reference (8) gives test data for the flanged joint shown in Figure A7. Experimental data consist of:

(1) Plot of bolt stresses versus internal pressure. Bolt stresses were determined by strain gages on the bolts.

(2) Flange rotation due to internal pressure.

(3) Longitudinal stress on outer surface of hub due to internal pressure.

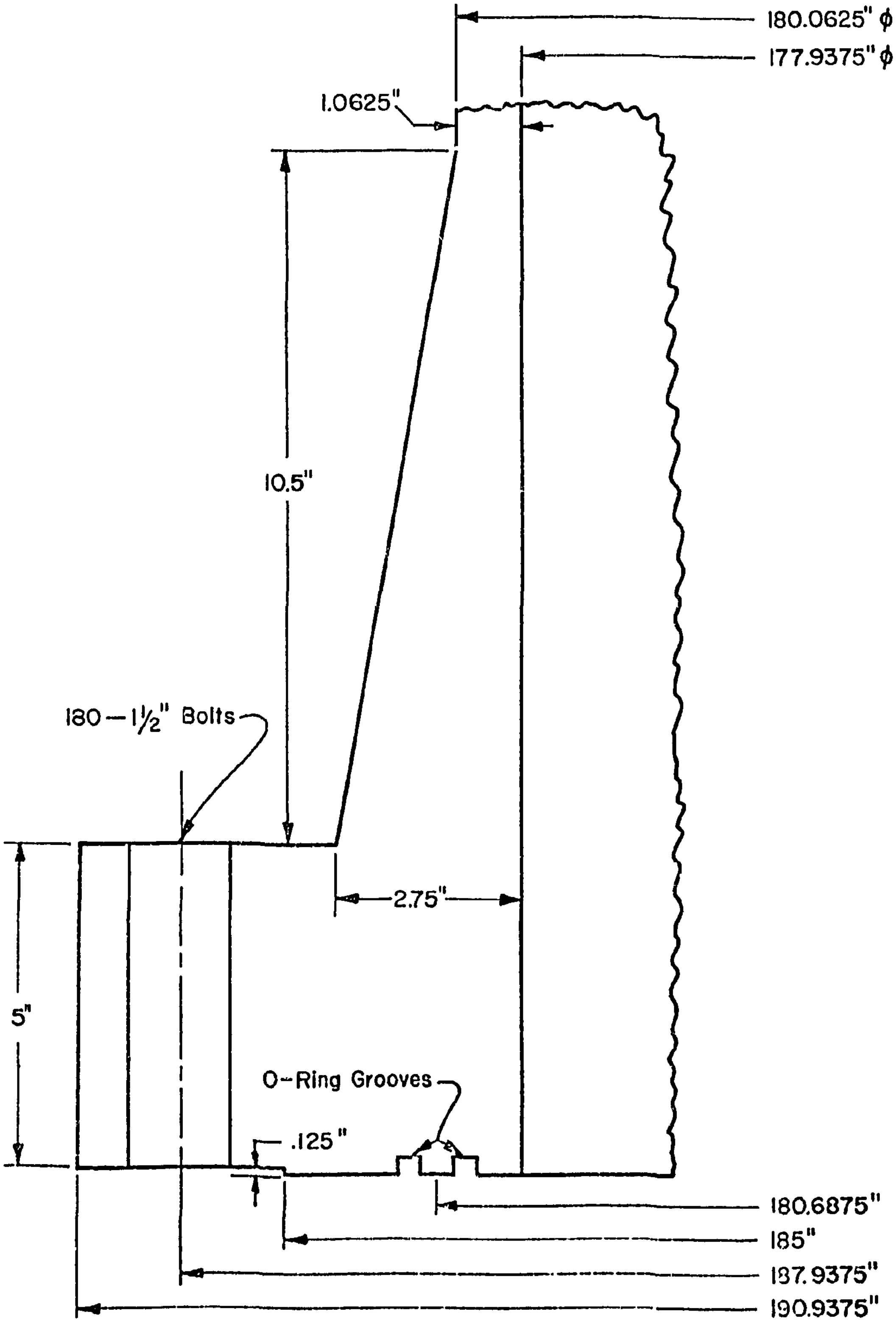

FIGURE A7. DIMENSIONS OF REFERENCE (8) FLANGES, IDENTICAL PAIR USED IN JOINT.

Printout from the computer program FLANGE for the flanged joint is shown in Table A2. The input data to FLANGE (printed out on the first page of Table A2) is based on:

(a) Dimensions as shown in Figure A7.

(b) Modulus of elasticity = 30×10^6 for flanges, bolts, and "gasket".

The elastomeric O-ring gasket is assumed to act only as a seal, hence, the gasket is simply the 1/8" raised face.

(c) Bolt area = 253.4 sq in.

(d) Pressure = 100 psi, Moment = $h_G \times A_b \times S_b$ = 7,920,000 in.-lb

W1 = initial bolt load = $A_b \times S_b$ = 5,392,352 lb.

The somewhat arbitrary aspect of the input consists of appropriate modeling of the "effective gasket diameter". Looking at Figure A7, and considering the rotation of the flanges, one would intuitively expect that most of the load between flanges would be carried by a relatively narrow annular ring of the raised face near the outside of the raised face. To investigate this, the following effective gasket diameters were assumed.

Effective Gasket Diameter, in.	h_G, in.	Gasket O.D.	Gasket I.D.	Initial Moment, in.-lb
185	1.46875	186.4	183.6	7,920,000
184	1.96875	185.4	182.6	10,616,000
183	2.46875	184.4	181.6	13,312,000
182	2.96875	183.4	180.6	16,009,000
180.7	3.61875	182.1	179.3	19,514,000

TABLE A2. OUTPUT FROM COMPUTER PROGRAM FOR LARGE FLANGED JOINT, FIGURE A7.

```
 FLANGE     FLANGE     FLANGE      PIPE      HUB AT      HUB        BOLT     PRESSURE,
 O.D.,A     I.D.,B    THICK.,T  WALL,GO    BASE,G1    LENGTH,H  CIRCLE,C     P
193.93750 177.93750   5.00000   1.06250    2.75000   10.50000 187.93750    100.000

 MOMENT   COEFF. OF    DELTA     MOD. OF MEAN GASKET  ITYPE      IBOND     ICODE      MATE
         THERMAL EXP.          ELASTICITY DIAMETER
7.9200 06 6.0000-06 1.0000-03 3.0000 07 1.8500 02      1          2          0          2

     BSIZE          YB            EB            TB            XGO           XGI           AB
   1.50000 00    3.00000 07    6.00000-06    0.0           1.86400 02    1.83600 02    2.53400 02
     VO            YG            EG            TG            FACE          PBE
   1.25000-01    3.00000 07    6.00000-06    0.0           0.0           0.0
     WI            TF            TFP           YF2           YFP2          YB2           YG2
   5.39240 06    0.0           0.0           3.00000 07    3.00000 07    3.00000 07    3.00000 07

    FLANGE JOINT BOLT LOAD CHANGE DUE TO APPLIED LOADS, IDENTICAL PAIR

      FLANGE JOINT SIDE ONE (PRIMED QUANTITIES)

QFHG=  5.64260-10  QPHG=  1.41000-05  QTHG=  5.99240-05   XB =  1.77940 02       GO=  1.06250 00       TH  =  5.00000 00
      YM =  3.00000 07          YF2 =  3.00000 07    EF =  6.00000-06

      FLANGE JOINT SIDE TWO (UNPRIMED QUANTITIES)

QFHG=  5.64260-10  QPHG=  1.41000-05  QTHG=  5.99240-05   XB =  1.77940 02       GO=  1.06250 00       TH  =  5.00000 00
      YM =  3.00000 07          YF2 =  3.00000 07    EF =  6.00000-06

        BOLTING

BOLT LENGTH=  1.16250 01  BOLT AREA=  2.53400 02  BOLT CIRCLE=  1.87940 02
      YB =  3.00000 07         YB2 =  3.00000 07    EB =  6.00000-06

        GASKET

  VO  =  1.25000-01  XGO =  1.86400 02  XGI =  1.83600 02
     YG  =  3.00000 07         YG2 =  3.00000 07    EG =  6.00000-06
```

TABLE A2: SHEET 2 OF 3.

```
LOADINGS

INITIAL BOLT LOAD=  5.3924D 06  BOLT TEMP.=  0.0          FLANGE ONE TEMP.=  0.0          FLANGE TWO TEMP.=  0.0
GASKET TEMP.=  0.0          DELTA=  1.0000D-03   DELTAP=  1.0000D-03  PRESSURE=  1.0000D 02

   RESIDUAL BOLT LOADS AFTER THERMAL-PRESSURE LOADS

AXIAL THERMAL,W2A=  5.3924D 06  MOMENT SHIFT,W2B=  2.6334D 06

TOTAL PRESSURE,W2C=  1.7499D 06   DELTA THERMAL,W2D=  5.3924D 06

COMBINED,W2=  1.7499D 06

W1-W2A=  0.0          W1-W2B=  2.7590D 06  W1-W2C=  3.6425D 06  W1-W2D=  3.7548D 01  W1-W2=   3.6425D 06

W2A/W1=  1.0000D 00  W2B/W1=  4.8835D-01  W2C/W1=  3.2451D-01  W2D/W1=  9.9999D-01  W2/W1=  3.2451D-01

    INITIAL AND RESIDUAL MOMENTS AFTER THERMAL PRESSURE LOADS.

M1=  7.9201D 06  M2A=  7.9201D 06  M2B= 1.1683D 07  M2C=  1.0386D 07  M2D=  7.9200D 06  M2=  1.0386D 07

M2BP=  1.1683D 07  M2CP=  1.0386D 07  M2P=  1.0386D 07
```

TABLE A2: SHEET 3 OF 3.

```
      TAPERED HUB FLANGE
   CALCULATIONS FOR MOMENT LOADING

SLSO=  9.0589D 03  SLSI= -9.0589D 03  SCSO=  8.0258D 03  SCSI=  2.5905D 03
SLLO=  8.3767D 03  SLLI= -8.3767D 03  SCLO=  1.8688D 03  SCLI= -3.1572D 03
 STH=  3.7688D 02   STF= -6.5534D 03   SRH=  3.4036D 03   SRF= -2.9688D 03
ZG= -1.0644D-02  ZC= -1.5113D-02  QFHG=  4.4690D-03  YO=  1.5742D-02  Y1= -1.9104D-03  THETA= -2.9766D-03
   CALCULATIONS FOR PRESSURE LOADING

SLSO=  5.6500D 03  SLSI=  2.7235D 03  SCSO=  7.4892D 03  SCSI=  6.6113D 03
SLLO=  1.5196D 03  SLLI=  1.7166D 03  SCLO=  2.8577D 03  SCLI=  2.9171D 03
 STH=  2.3524D 03   STF=  7.4179D 02   SRH= -1.6560D 02   SRF= -5.2220D 01
ZG= -3.4211D-03  ZC= -4.8310D-03  QFHG=  1.4100D-03  YO=  1.7184D-02  Y1=  7.1237D-03  THETA= -9.7548D-04
   CALCULATIONS FOR TEMPERATURE LOADING

SLSO=  1.0322D-01  SLSI= -1.0322D-01  SCSO=  3.8043D-02  SCSI= -2.3890D-02
SLLO=  1.2045D-02  SLLI= -1.2045D-02  SCLO= -1.1217D-01  SCLI= -1.1940D-01
 STH=  6.2897D-02   STF= -5.5648D-03   SRH= -4.4270D-03   SRF=  3.9174D-04
ZG= -1.4539D-07  ZC= -2.0532D-07  QFHG=  5.9924D-08  YO=  2.0987D-08  Y1= -3.4336D-07  THETA= -4.1458D-09
  CALCULATIONS FOR COMBINED LOADING,M2 OR M2P FOR ITYPE=1 OR 2, W2 FOR ITYPE=3,  =   1.0386D 07

SLSO=  1.7529D 04  SLSI= -9.1556D 03  SCSO=  1.8014D 04  SCSI=  1.0008D 04
SLLO=  1.2503D 04  SLLI= -9.2679D 03  SCLO=  5.3082D 03  SCLI= -1.2231D 03
 STH=  2.8467D 03   STF= -7.8517D 03   SRH=  4.2975D 03   SRF= -3.9452D 03
ZG= -1.7379D-02  ZC= -2.4650D-02  QFHG=  7.2702D-03  YO=  3.7826D-02  Y1=  4.6182D-03  THETA= -4.8768D-03
```

The "gasket" width is kept constant at 1.4". This is not significant as the elastic response of the strip is negligible compared to that of the bolts for any reasonably large gasket width. The calculations for an effective diameter of 185" is shown in Table A2; the pertinent results of the other calculation are included in the following discussion.

Change in Bolt Stress with Pressure

There are two regimes involved in these tests; i.e., below and above the critical pressure. The critical pressure can be calculated by the equation:

$$P_c = \frac{S_b A_b}{A_p} \qquad \text{(A2)}$$

where S_b = bolt stress, A_b = bolt area and A_p = pressure area = $(\pi/4)(180)^2$ = 25,447 sq in. in these tests. For the elastomeric O-ring,

P_c is that pressure at which the metal-to-metal interface load is zero; at pressures above P_c a separation of the metal faces occurs. The initial value of S_b in these tests was 9.5 tons/in.^{2}. The results from FLANGE show that S_b decreases as the pressure increases. For G = 185", the results shown in Table A2 give W2 = 1,749,800 lb for P = 100 psi; giving S_{b2} = 1,749,800/(253.4 x 2240) = 3.083 ton/in.^{2} at 100 psi or, in terms of P,

$$S_b = 9.5 - 6.417 \frac{P}{100} \qquad \text{(A3)}$$

Substituting S_b from Equation (A3) into Equation (A2) and solving for P gives the critical pressure, P_c.

$$P_c = (9.5 - 6.417 \frac{P_c}{100}) \frac{253.4 \times 2240}{25,447}$$

$$= 87.15 \text{ psi}$$

These relationships are shown graphically in Figure A8 for the various assumed values of G_e. The values of G_e, P_c, and S_b at P_c are:

G_e, in.	P_c, psi	S_b at P_c, ton/in.^{2}
185	87.15	3.907
184	97.13	4.354
183	110.51	4.954
182	125.75	5.638
180.7	147.07	6.593

It can be seen in Figure A8 that, as might intuitively be expected, the effective gasket diameter is somewhat smaller than the raised face outside diameter; an effective gasket diameter of about 183" agrees best with the test data.

Above P_c, the increase in bolt stress is given by Equation (A2).

Flange Rotation Due to Pressure

Below P_c, the theoretical increase in flange rotation due to pressure is given by Equation (A1). For G = 185", Table A2 gives $\theta_p/P = 9.7548 \times 10^{-4}/100$ and $\theta_m/M = 2.9766 \times 10^{-3}/7.92 \times 10^6$. Table A2 also shows $M_2 = 1.0386 \times 10^7$ for $\Delta P = 100$ psi, hence, $\Delta M = M_2 - M_1 = 1.0386 \times 10^7 - 0.792 \times 10^7$ for P = 100 psi. Accordingly, Equation (A1) gives:

$$\Delta\theta = 9.7548 \times 10^{-6}\, \Delta P + 3.758 \times 10^{-10}(1.0386 - 0.7920) \times 10^7 \frac{\Delta P}{100}$$

$$= 1.9022 \times 10^{-5}\, \Delta P \text{ radians}$$

$$= 0.0654\, \Delta P \text{ minutes} \qquad \text{(A4)}$$

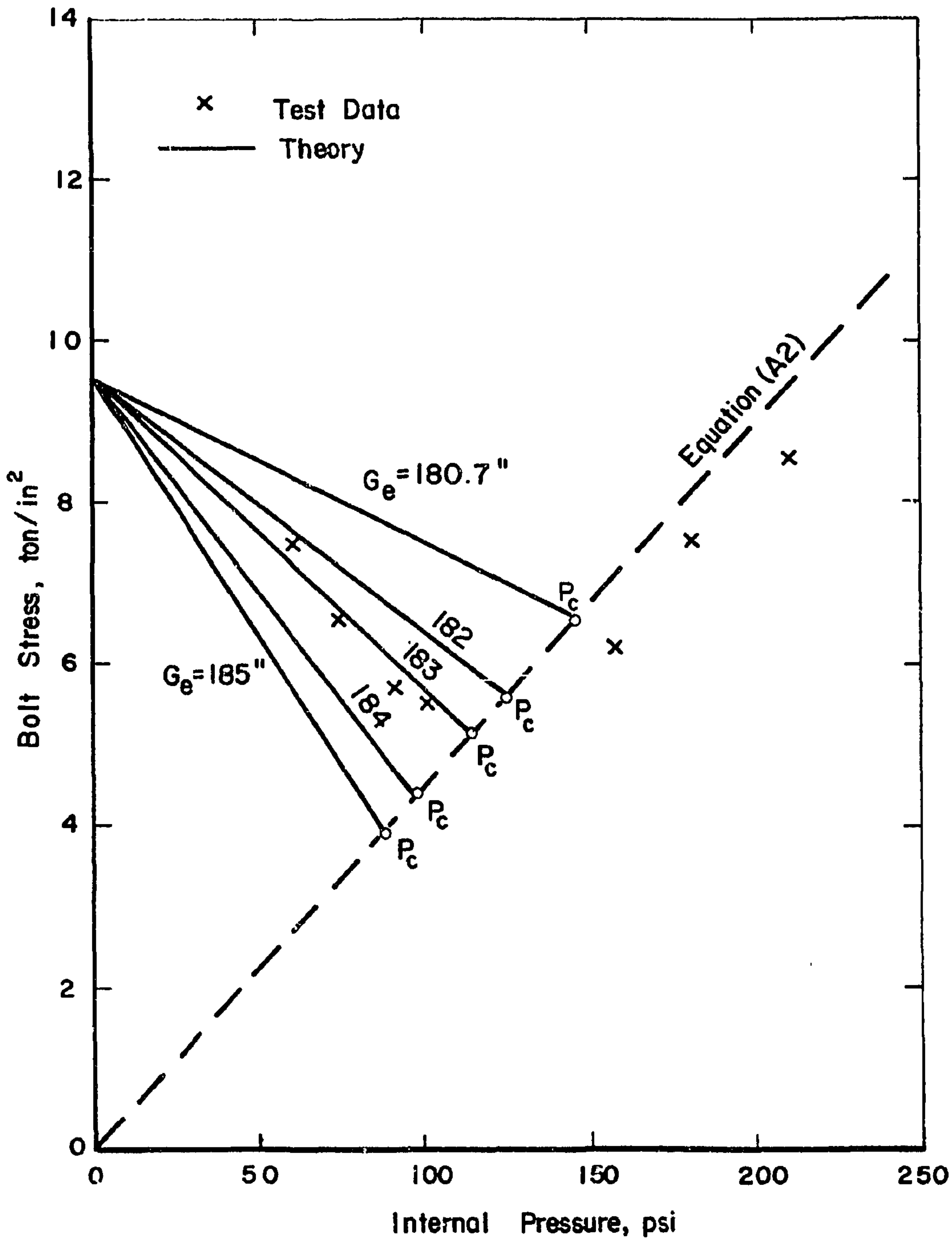

FIGURE A8. COMPARISON OF THEORETICAL AND MEASURED CHANGE IN BOLT STRESS WITH PRESSURE, REFERENCE (8) TEST DATA.

Equation (A4) is shown in Figure A9, along with theoretical lines for G = 185, 184, 183, 182 and 180.7 inches. These lines terminate at P_c. Above P_c, the rotation is given by Equation (A1) but ΔM is obtained by the equation:

$$\Delta M = A_p h_D \Delta P \qquad \text{(A5)}$$

where

$$h_D = \text{moment arm to midsurface of attached shell}$$
$$= \left[C - (B + g_o)\right] /2$$
$$= \left[187.9375 - (177.9375 + 1.0625)\right] /2 = 4.469''$$

With A_p = 25,447 in.^{2}, Equation (A5) gives

$$\Delta M = 113{,}722 \Delta P \text{ in.-lb}$$

Equation (A1) then gives

$$\Delta\theta = (9.75 \times 10^{-6} + 3.758 \times 10^{-10} \times 113{,}722)\Delta P, \text{ radians}$$

$$= 5.2492 \times 10^{-5} \Delta P \text{ radians}$$

$$= 0.1805 \ \Delta P \text{ minutes.}$$

The value of $\Delta\theta / \Delta P$ does not depend upon the choice of G but the value of P_c does. The complete curves are shown in Figure A9. As in Figure A8 for bolt stress change, Figure A9 indicates that an effective gasket diameter of about 183" agrees best with the test data.

Flange Stresses Due to Pressure

Test data are given (only) for longitudinal stresses on the outside surface of the hub. These are stated to be stresses obtained above P_c and are for a pressure change of 100 psi. The theoretical value of stress is obtained by:

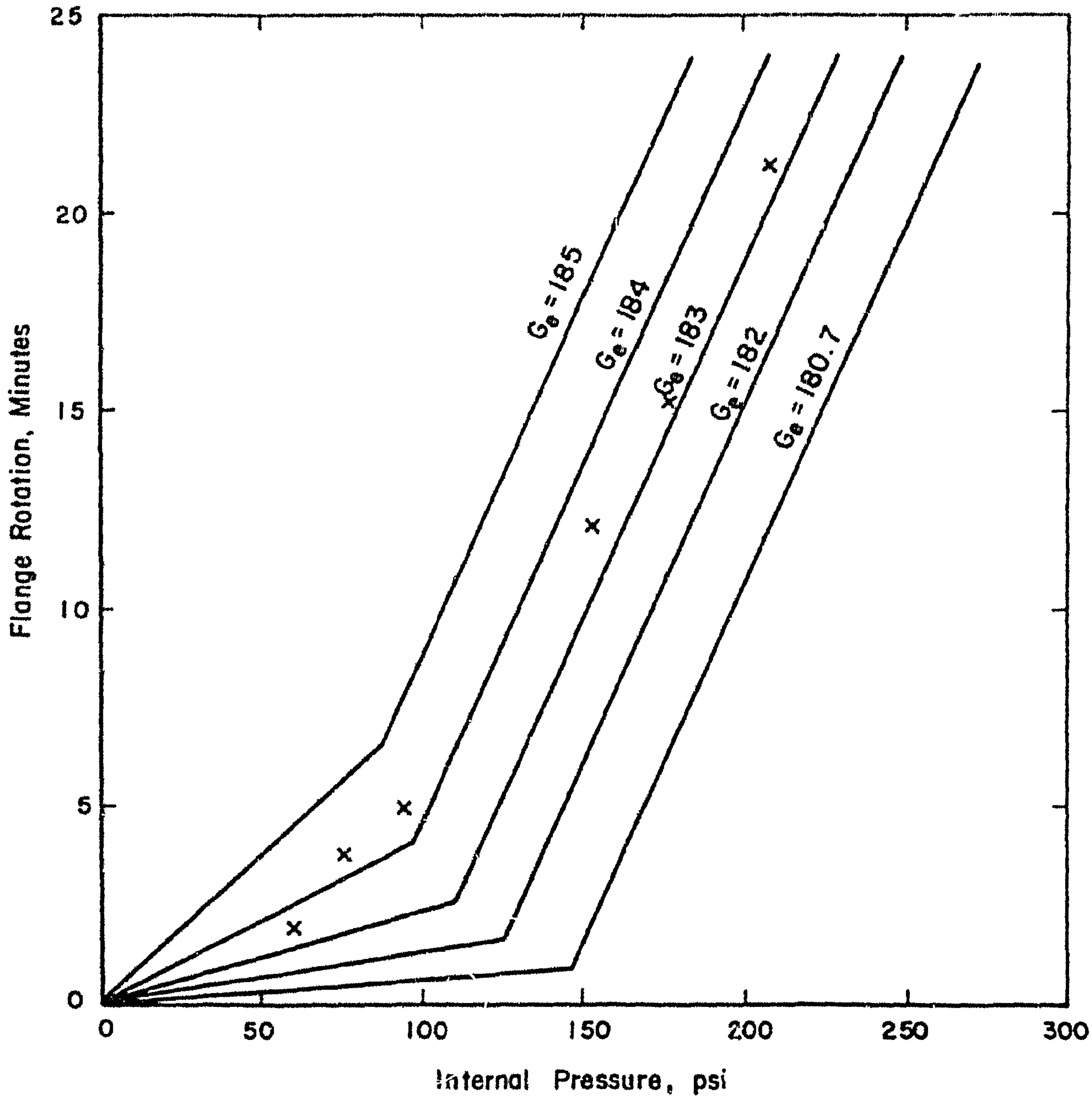

FIGURE A9. COMPARISON OF THEORETICAL AND MEASURED FLANGE ROTATION DUE TO INTERNAL PRESSURE, REFERENCE (8) TEST DATA.

$$\sigma = \frac{\sigma_p}{P} \times 100 + \frac{\sigma_m}{M} \frac{\Delta M}{\Delta P} \times 100 \qquad (A6)$$

The value of ΔM (above P_c) is $A_p h_D$ P = 113,722 P in.-lb. The value of the longitudinal stress at the juncture of the hub with the shell is given in Table A2 as SLSO = 5,650 psi for a pressure of 100 psi; and SLSO = 9,059 for a moment of 7.92×10^6 in.-lb. Accordingly, for this location and surface, Equation (A6) gives:

$$\sigma = 5{,}650 + \frac{9059}{7.92 \times 10^6} \times 113{,}722 \times 100 = 18{,}660 \text{ psi}$$

Similarly, from Table A2 and Equation (A6) the longitudinal stress on the outside of the hub at the juncture with the flange ring (identified as SLLO in Table A2) is

$$\sigma = 1519 + \frac{8377}{7.92 \times 10^6} \times 113{,}722 \times 100 = 13{,}550 \text{ psi}$$

The calculated stresses, along with the test data, are shown in Figure A10. Additional data along the hub and attached shell were calculated using the general axisymmetric shell program, MOLSA [17]. It is apparent in Figure A10 that the calculated stresses are substantially higher than the measured stresses. One of several possible reasons for this lies in the assumption as to the location of the bolt load application line. The theory assumes that this is along the bolt circle, C. However, as the flange ring rotates, the nuts on the bolts shift the load inward. Calculated stresses assuming that the bolt load application line lies along the inside of the bolts are also shown in

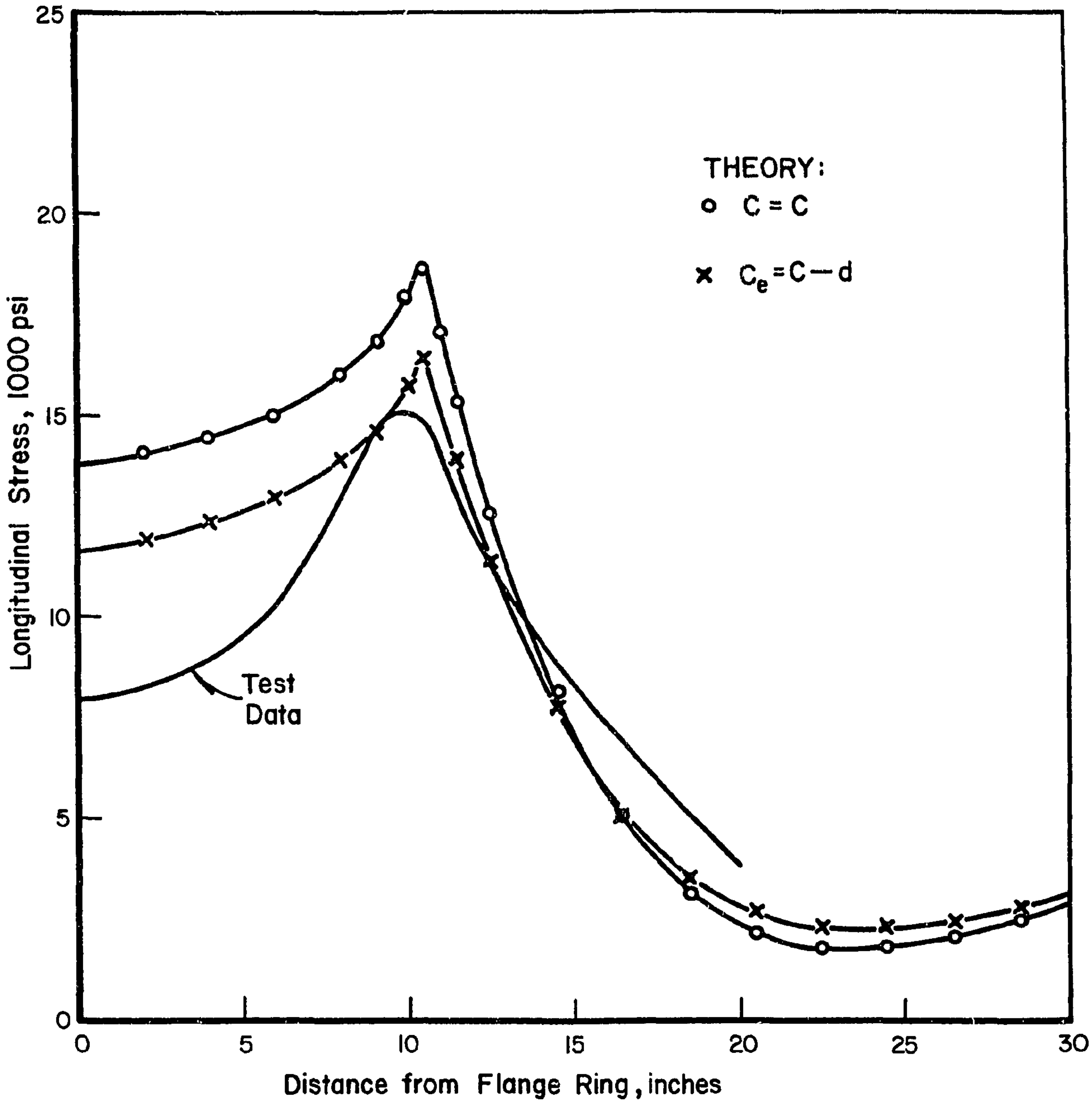

FIGURE A10. COMPARISON OF THEORETICAL AND MEASURED STRESSES DUE TO 100 PSI INTERNAL PRESSURE, REFERENCE (8) TEST DATA, STRESSES ARE ALONG OUTSIDE SURFACE OF HUB

Figure A10 as $C_e = C-d$. These are calculated as previously discussed except using $h_D = 4.469 - \frac{d}{2} = 4.469 - 0.75 = 3.719$ in. Considering the basis of the theory, the agreement between measured and calculated stresses is as good as should be anticipated.

Tests on ANSI B16.5 Flanged Joints with Asbestos Gaskets, Bolt Stress Change with Pressure

Reference (9), Figure 10, gives test data on the variation of bolt stress (measured with strain gages) with internal pressure. The data for welding neck flanges are shown in Figure A11, along with theoretical results obtained using the computer program FLANGE.

In this set of tests, there are two major uncertainties in application of the theory:

(1) What is the effective gasket diameter, G_e?

(2) What is the effective modulus of elasticity of the asbestos gasket material?

The gasket widths in these tests are such that the value of G_e is a lesser problem than in the previously discussed tests of the large flanges with O-ring gaskets. Accordingly, in this set of calculations we have used the gasket mean diameter as G_e.

The modulus of elasticity of an asbestos gasket varies with bolt stress, time under load, and temperature. For new gaskets, tested within a few hours after installation, Reference (10) gives an equation for approximating the gasket modulus of such "newly installed" gaskets which is about 50,000 to 100,000 psi for the tests under consideration. Reference (3) suggests that the effective modulus of elasticity of asbestos gaskets may be around 1,000,000 psi. In some flanged joints, the bolt stress variation with pressure would not be much different for

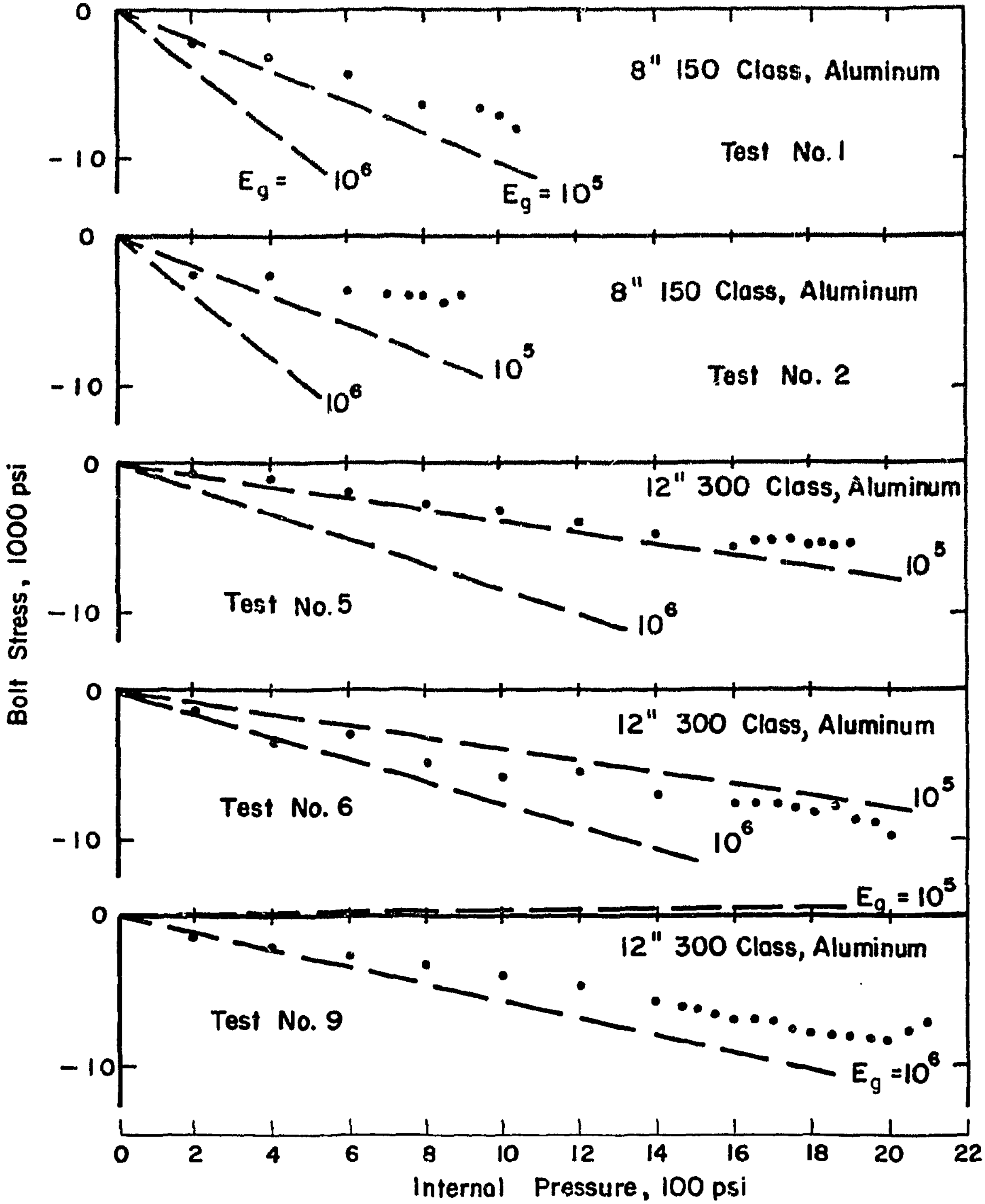

FIGURE A11. COMPARISONS OF THEORETICAL AND MEASURED CHANGE IN BOLT STRESS WITH PRESSURE, REFERENCE (9) TEST DATA.

$E_g = 10^5$ psi than for $E_g = 10^6$ psi. However, for the particular flanged joints under consideration, there is a significant difference. Figure A11 shows the calculated bolt stress variations assuming $E_g = 10^5$ or $E_g = 10^6$ psi.

If we assume that the effective modulus of elasticity of the asbestos gaskets is proportional to the initial gasket stress, and the initial bolt stress is the same for all tests, then the 8" flanges should have a smaller modulus than the 12" flanges. This is because the ratio of bolt area to gasket area is 0.123 for the 8" flange; 0.346 for the 12" flanges. A qualitative trend in this direction can be seen in Figure A11.

It is pertinent to note that further increase in the assumed value of E_g above 10^6 psi does not lead to a greatly increased prediction of the rate of bolt stress change with pressure. This point can be illustrated by considering the values of $\Delta S_b/$ P for $E_g = 10^5$, 10^6 and 30×10^6 as tabulated below.

Flanged Joint	$\Delta S_b/\Delta P$ for E_g of:		
	10^5 psi	10^6 psi	30×10^6 psi
8" Aluminum	10.2	19.9	21.3
12" Aluminum	4.0	7.5	8.0
12" Steel	-0.29	5.2	6.1

LEAKAGE PRESSURE TESTS

Because leakage is usually the "failure criterion" for flanged joints, one might expect to find numerous test data in which the leakage pressure of a flanged joint was determined. Such is not the case. Back in 1936 the British "Pipe Flanges Research Committee" published a few results of this type; but, the tests were not really on flanged joints because the load was externally applied and controlled; not applied by tightening the bolts. However, they did find [11] that the following relationship held for a variety of metal gaskets

$$(\pi/4)\ G_o^2\ P_L = W \qquad \text{(A7)}$$

where

G_o = gasket outside diameter

P_L = leakage pressure

W = total load

Petrie[5] published results of leakage pressure tests where the load was applied by tightening the bolts in a flanged joint. Details of the joints that he tested are not given nor is the method of measuring the bolt stress described. His results, for flat gaskets, are shown in Table A3. Petrie introduced the idea of a "performance factor", shown in the last column of Table A3 and defined as:

$$\frac{W_p}{W_b} = \frac{(\pi G_o^2/4)P_L}{S_{b1}A_b} \qquad \text{(A8)}$$

TABLE A3. LEAKAGE PRESSURE TESTS OF FLAT GASKETS IN FLANGED JOINTS, DATA FROM REFERENCE (5).

Flange O.D., in.	Bolts, No. and Size	Gasket Material	O.D., in.	I.D., in.	S_{b1} psi	P_L, psi	W_p/W_b
14.625	12-1 1/4	Akro Metal	9.0625	7.375	40,500	5,000	0.712
"	"	"	"	"	48,000	6,500	0.780
"	"	"	"	"	66,000	10,000	0.874
12.5	16-5/8	Akro Metal	8.875	7.75	35,000	1,600	0.877
"	"	"	"	"	70,000	2,200	0.602
"	"	"	"	"	105,000	3,300	0.604
"	"	"	"	"	120,000	4,000	0.643
25.5	16-1 7/8	Asbestos	14.25	13.50	21,400	3,900	0.908
"	"	"	"	"	28,500	5,000	0.872
"	"	"	15.00	13.50	35,700	6,000	0.927
"	"	Akro Metal	14.25	13.50	28,500	5,600	0.976
"	"	"	"	"	46,800	6,200	0.660
"	"	Copper	14.25	13.50	46,800	6,200	0.660
"	"	Copper	15.00	13.50	46,800	5,800	0.617

where

G_o = gasket outside diameter

P_L = leakage pressure

S_{b1} = initial bolt stress

A_b = total bolt area

Equation (A8) is, of course, an axial force equilibrium equation; valid if S_{b2} at P_L is equal to S_{b1} at $P = 0$. If S_{b2} decreases as P increases, then W_p/A_b will be less than unity. Equation (A8) also assumes that a gasket cannot carry a tensile load; an assumption of doubtful general validity for asbestos gaskets.

Table A3 shows W_p/W_b consistently less than unity; in contrast to Reference (11) tests where W_p/W_b (with load controlled externally) was essentially equal to unity. This suggests that in Petrie's tests the bolt load did decrease with pressure. However, details of the flanges are not known, hence no theoretical comparisons can be made.

Reference (11) states that leakage pressure tests were made using water as the pressurizing fluid and that "the first signs of hydraulic leakage were easily and accurately determined by visual observation". Reference (5) does not state what fluid was used nor how leakage was determined. The writer would guess that the fluid was water; with leakage detected by visual observation.

During the years between 1937 and to-date, only a few papers have been published giving data on leakage pressures of flange joints. One of these[9], along with data from unpublished reports of Tube Turns (Division of Chemetron), are discussed in the following. These are all tests on ANSI B16.5 flanged joints and thus are particularly relevant to this report. The test procedure was generally as follows.

(1) A flanged joint was made up with a new asbestos gasket. The bolts were tightened with a torque wrench to a desired stress level. The bolt stress was determined either directly with strain gages in the bolts or indirectly by calculations of torque versus bolt load.

(2) Using water as the pressurizing fluid, the pressure was raised in steps until leakage occurred as evidenced by drops of water emerging from the flanged joint within a short time after applying the pressure step.

(3) Provided the gasket remained intact, the pressure was then reduced to zero, the bolts were tightened to a higher level and the leakage pressure was determined again.

Tables A4 thru A9 give results from these tests. Each table shows the flange material, bolt material, pipe material, and gasket material and dimensions. The flange dimensions are specified in ANSI B16.5.

Each table gives:

(a) The initial bolt stress, S_{b1}. It should be emphasized that this is not necessarily and generally is not the bolt stress at pressure.

(b) The leakage pressure, P_L.

(c) The performance factor, W_p/W_b, as defined by Equation (A8).

(d) Calculated (Computer program FLANGE, IBOND = 2) maximum flange stresses. Stresses are shown for the initial bolt load, for pressure and for the sum of the two loads. The latter (combined) stress is valid only if the initial bolt load does not change with pressure.

Table A4: LEAKAGE PRESSURE TESTS, 12" 300 class, WN to WN,

Reference (9), Test No. 9

Flange Material A181 Grade 1; U.T.S = 73,400 psi, Y.S. = 42,000 psi

Bolt Material A193 Grade B7

Pipe Material (.375" wall) A106 Grade B; U.T.S. = 65,000 psi Y.S. = 42,600 psi

Gasket: 1/16" thick asbestos, G_o = 15", G_i = 13.5"

Total Bolt Area = 16 x .728 = 11.648 in.^{2}, $\pi G_o^2/4$ = 176.7 in.^{2}

S_{b1} psi (1)	P_L psi (2)	W_p/W_b (3)	Calculated Max. Flange Stress, psi Bolt Load	Pressure	Combined
10,000	700	1.06	6,920	630	7,550
24,000	1,500	0.95	16,600	1,360	17,960
36,500	2,100	0.87	25,300	1,900	27,200
47,500	2,500	0.80	32,900	2,260	35,160
57,000	2,700	0.72	39,400	2,440	41,840
70,000	2,900	0.63	48,400	2,620	51,020

(1) S_{b1} = initial bolt stress
(2) P_L = leakage pressure
(3) W_p/W_b = performance factor, see Equation (A8).

Table A5: LEAKAGE PRESSURE TESTS, 12" - 300 class, WN to WN,

Reference (9), Test No. 5

Flange Material: 6061-T6; U.T.S. = 45,400 psi, Y.S. = 38,800 psi

Bolt Material: A193 Grade B7 0.500" wall.

Pipe Material: (0.500" wall) 6061-T6; U.T.S. = 46,600 psi, Y.S. = 40,100 psi

Gasket: 1/16" thick asbestos, G_o = 15", G_i = 13.5"

Total Bolt Area = 16 x.728 = 11.648 in.^{2}, $\pi G_o^2/4$ = 176.7 in.^{2}.

S_{b1} psi	P_L psi	W_p/W_b	Calculated Max. Flange Stress, psi Bolt Load	Pressure	Combined
6,000	300	0.76	4,010	320	4,330
10,000	550	0.83	6,680	590	7,270
13,000	800	0.93	8,700	860	9,560
15,000	950	0.96	10,020	1,030	11,050
17,500	1,100	0.95	11,690	1,190	12,880
22,500	1,200	0.81	15,030	1,300	16,330
27,500	1,350	0.74	18,380	1,460	19,840
36,000	1,600	0.67	24,050	1,730	25,780
40,000	1,800	0.68	26,730	1,940	28,700
48,000	1,850	0.58	32,070	2,000	34,070
93,000	2,850	0.46	62,140	3,080	65,220

Table A6: LEAKAGE PRESSURE TESTS, 8" - 150 class, WN to WN, 0.322 THICK ATTACHED PIPE (ALUMINUM)

Reference (9), Test No. 1

Flange Material: 6061-T6; U.T.S. = 47,800 psi, Y.S. = 43,100 psi

Bolt Material: A193 Grade B7

Pipe Material: (0.322" wall) 3003-F; U.T.S. = 14,000 psi Y.S. = 5,000 psi

Gasket: 1/16" thick asbestos, G_o = 10.625", G_i = 9.375"

Total Bolt Area: = 8 x 0.302 = 2.416 in.^{2}, $\pi G_o^2/4$ = 88.66 in.^{2}

S_{b1} psi	P_L psi	W_p/W_b	Calculated Max. Flange Stress		
			Bolt Load	Pressure	Combined
9,500	300	1.16	3,290	500	3,790
17,000	600	1.30	5,890	1,060	6,950
31,000	920	1.09	10,750	1,630	12,380
59,000	1,070 *	--	20,460	1,890	22,350

* Pipe ruptured at 1,070 psi.

Table A7: LEAKAGE PRESSURE TESTS, 3" WELDING NECK TO SLIP-ON, 150 CLASS, 0.216" THICK ATTACHED PIPE.

Reference (12)

Flange Material: A181 Grade 1; U.T.S. = 77,000 psi, Y.S. = 35,900 psi

Bolt Material: A193 Grade B7

Pipe Material: A106 Grade B

Gasket: 1/16" thick asbestos, G_o = 5", G_i = 3.56"

Total bolt area = 4 x .202 = 0.808 in.^{2}, $\pi G_o^2/4$ = 19.63 in.^{2}

S_{b1} psi	P_L, psi	W_p/W_b	Calculated Max. Flange Stress, psi Bolt Load	Pressure	Combined
10,000	675	1.64	3,640	50	3,690
22,500	1,160	1.25	8,200	90	8,290
34,170	2,410	1.71	12,450	180	12,630
42,500	2,990	1.71	15,480	220	15,700
54,000	4,000	1.80	19,670	300	19,970
66,000	4,200	1.55	24,040	320	24,360
79,303	4,700	1.44	28,790	350	29,140
87,500	5,200	1.44	31,880	390	32,270
80,700*	4,500	1.35	29,400	340	29,740
89,700	5,150	1.39	32,680	390	33,070
98,700	5,400	1.33	35,960	410	36,370
107,600	5,350	1.21	38,110	400	38,510
116,600	5,550	1.16	42,480	420	42,900
125,600	5,950	1.15	45,760	450	46,210
134,600	6,175	1.11	49,030	460	49,490
143,500	6,400	1.08	52,280	480	52,760
152,500	6,700	1.07	55,560	500	56,060
161,500	6,950	1.05	58,830	520	59,350

* These and following tests were made using 3/4" rather than 5/8" bolts. Bolt stresses have been converted to equivalent on 5/8" bolts.

Table A8: LEAKAGE PRESSURE TESTS, 4" 900 CLASS WELDING NECK TO WELDING NECK, CAPS WELDED TO FLANGES

Reference (13)

Flange Material: A105 Grade 2; U.T.S. = 63,000 psi, Y.S. = 30,600 psi

Bolt Material: A193 Grade B7

Pipe Material: A106 Grade B (caps)

Gasket: 1/16" thick asbestos, G_o = 6.1875", G_i = 4.5"

Total Bolt Area = 8 x .728 = 5.824 in.^{2}. $\pi G_o^2/4$ = 30.07 in.^{2}.

S_{b1} psi	P_L, psi	W_p/W_b	Calculated Max. Flange Stress, psi: Bolt Load	Pressure	Combined
8,400	350	0.22	9,400	-30	9,370
16,900	4,500	1.37	18,920	-440	18,480
25,400	6,500	1.32	28,440	-640	27,800
33,800	8,300	1.27	37,840	-810	37,030
42,300	9,500	1.16	47,350	-930	46,420
50,700	10,500	1.07	56,760	-1030	55,730
55,800	11,000	1.02	62,460	-1080	61,380

Table A9: LEAKAGE PRESSURE TESTS, 8" 150 CLASS, WELDING NECK TO WELDING NECK AND 8" FLANGES WITH REDUCED FLANGE THICKNESS

Reference (14).

Bolt Material: A193 Grade B7

Pipe Material: A106 Grade B, 0.322" thick

Gasket: 1/16" thick asbestos, G_o = 10.625", G_i = 9.375"

Total Bolt Area ≐ 8 x .302 = 2.416 in.^{2} $\pi G_o^2/4$ = 88.66 in.^{2}

Test No.	Yield Strength psi	Flange Thickness, in.	S_{b1} psi (1)	P_L, psi	W_p/W_b	Calculated Max. Flange Stress, psi Bolt Load	Pressure	Combined
1	55,400	1.17	96,000	3,250	1.24	33,270	5580	38,850
2	88,800	1.17	96,000	2,980	1.14	33,270	5110	38,380
3	35,500	1.17	78,500*	2,460	1.25	26,400	4220	30,620
4	35,500	1.17	83,000*	2,930	1.30	27,900	5030	32,930
5	88,800	0.75	130,400	3,220	0.91	101,230	-4510	96,720
6	88,800	0.50	103,500*	3,000	1.06	182,600	-3990	178,610
7	35,500	0.50	41,400*	1,080	0.96	73,040	-1440	71,600

(1) * Indicates that load/pressure was sufficient to produce measurable yielding of the flanges.

(2) Maximum stress is LLO for tests 1, 2, 3, and 4; RH for tests 5, 6, and 7; see Figure 3 of text.

Some significant aspects of the tests are

(1) The flanges have a pressure capacity much greater than the standard hydrostatic test pressure of 1.5 times the 100 F rated pressure, P_h: as shown by the following tabulation.

Flanges	Max. P_L, psi	P_h, psi	$\frac{P_L}{P_h}$,
12" 300 WN, A181 Grade 1 Steel	2900	1080	2.7
12" 300 WN, 6061-T6 Aluminum	2850	1080	2.6
8" 150 WN, 6061-T6 Aluminum	1070 +	413	2.6 +
3" 150 WN Slip-on A181 Gr. 1	6950	413	16.9
4" 900 WN, 35,500 Y.S. Material	11000	3240	3.4
8" 150 WN, 35,500 Y.S. Material	2460	413	6.0
" " 55,400 Y.S. Material	3250 +	413	7.9 +
" " 88,000 Y.S. Material	2980 +	413	7.2 +

(2) The "performance factor", W_p/W_b, generally decreases with increasing bolt load. This factor is theoretically equal to unity if:

(a) The bolt load at pressure is equal to the initial bolt load.

(b) The gasket cannot transfer tensile forces.

The performance factors less than unity may be due to decrease in bolt load which, in turn, may be due to either elastic effects or yielding of some part of the flanged joint; i.e., flanges, bolts, or attached pipe. Factors higher than unity can be due to increase in bolt stress with pressure, which is theoretically possible; particularly if the effective modulus of elasticity of the gasket material is small. Qualitatively, one would expect the effective modulus of the gasket to be small for low loads and large for high loads; leading to the trend of decreasing values of W_p/W_b with increasing initial bolt stress. However, we speculate that some of the results were influenced by "sticking of the gasket" so that the gasket can transfer tensile forces. Anyone who has taken apart a flanged joint using an asbestos gasket will recognize that, after removing the bolts, it is often necessary to pry the joint apart.

(3) The values of maximum calculated flange stresses indicate that the flanges are capable of carrying loads substantially higher than the load to produce local yielding. One exception to this is shown in Table A9, Test No. 3. The calculated maximum stress is 26,400 psi; flange material yield strength is 35,500 psi. Test No. 4 was a retest of the flanged joint used in Test No. 3. It also indicated yielding at unexpectedly low loads. In several of the tests described in Tables A4 through A9, yielding of the attached pipe may have occurred and thereby influenced the test results. However, in Test No. 3 of Table A9, the yield pressure of the pipe would be expected to be about 2 x 40,000 x 0.322/8.303 = 3,100 psi and hence, should not have yielded at the flanged-joint test pressure of 2,460 psi.

TEST DATA WITH PIPE BENDING MOMENT

We have not found any published data on the effect of pipe bending moments on flanged joints. References (18), (19), and (20) give theoretical approaches for estimating the effect of pipe bending moments, but do not include test data.

Data from unpublished reports of Tube Turns Division of Chemetron are presented in the following and compared with text Equations (7) and (17). These are "leakage pressure tests", but with the addition of a pipe bending moment. The data are for tests on 4" 300 class, 3" 150 class, and 12" 150 class B16.5 welding neck flanged joints. The flanges were made of SA-181 Grade 1 material; bolts were SA-193 Grade B7 material; gaskets were 1/16" thick asbestos with diameters as shown below.

Flange	Gasket O.D., in.	Gasket I.D., in.
4" 300	6-3/16"	4"
3" 150	5"	3-9/16"
12" 150	15"	12.25"

The attached pipe was SA-106 Grade B material with wall thicknesses of 0.237", 0.216", and 0.250" for the 4-300, 3-150, and 12-150 flanged joints, respectively.

The test procedure was generally as follows.

(1) A flanged joint was made up with a new asbestos gasket. The bolts were tightened with a torque wrench to a desired stress level. The bolt stress was determined either directly with strain gages on the bolts or indirectly by calibration of torque vs bolt load.

(2) Using water as a pressurizing fluid, the pressure was raised in steps until leakage occurred as evidenced by drops of water emerging from the flanged joint within a short time after applying the pressure step. The leakage pressure so obtained is the zero pipe bending moment leakage pressure plotted in Figure A12 through A17.

(3) The pressure was then reduced to zero and a bending force was applied to one of the attached pipes; the other attached pipe was anchored. The axial distance from the load application to the gasket of the flanged joint was 35", 20", and 60" for the 4-300, 3-150, and 12-150 flanged joints, respectively. The applied moment is given in the References and is the force times the distance from the point of force application to the gasket. The leakage pressure was then determined as in Step (2).

(4) Step (3) was repeated with higher bending forces.

(5) The bolts were then further tightened and steps (2) thru (4) were repeated.

(6) An upper limit on applied pressures was imposed of 1,650 psi for the 4" - 300 class and 625 psi for the 12" - 150 class.

The results are shown in Figures A12 thru A17 as plots of pipe bending moment versus leakage pressure.

Figures A12, A13, and A14 also show the pipe bending moment versus leakage pressure relationships from text Equation (7), using $S_m = S_{b1}$, and S_{b1} = initial bolt stress. It is apparent that Equation (7), which is the Code NB-3647.1(b) limitation of pipe bending moments as controlled by bolt stress, is quite conservative with respect to the test data. For example, in Figure A12 Equation (7) implies that the joint is not capable of carrying any pressure with a pipe bending moment of 82,000 in.-lb with an initial bolt stress of 25,100 psi whereas the test data indicate that the flanged joint can hold more than 1,650 psi pressure at these conditions.

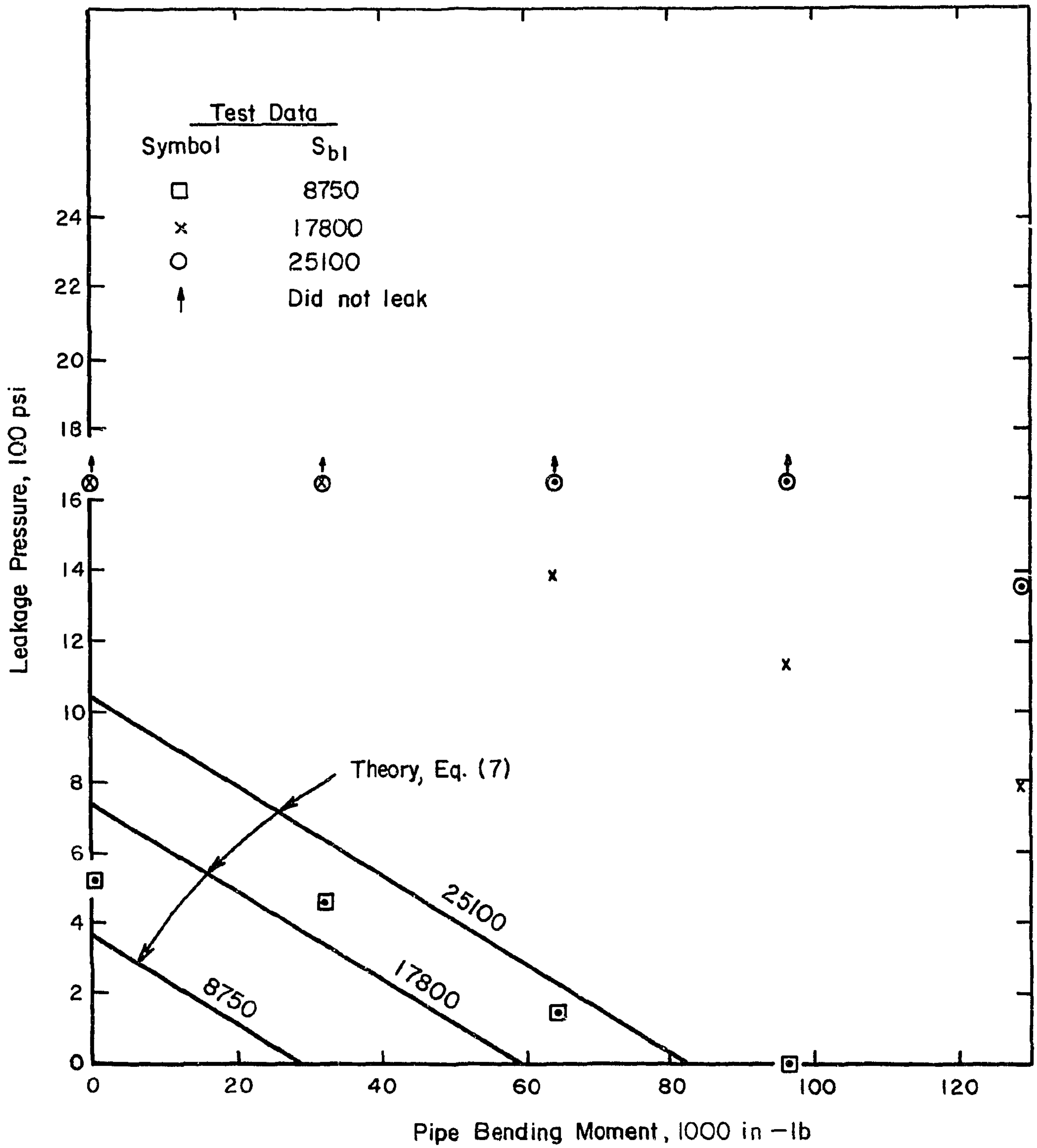

FIGURE A12: LEAKAGE PRESSURES, 4"-300 FLANGED JOINT, TEST DATA FROM REFERENCE (10), THEORY FROM TEXT EQ. (7).

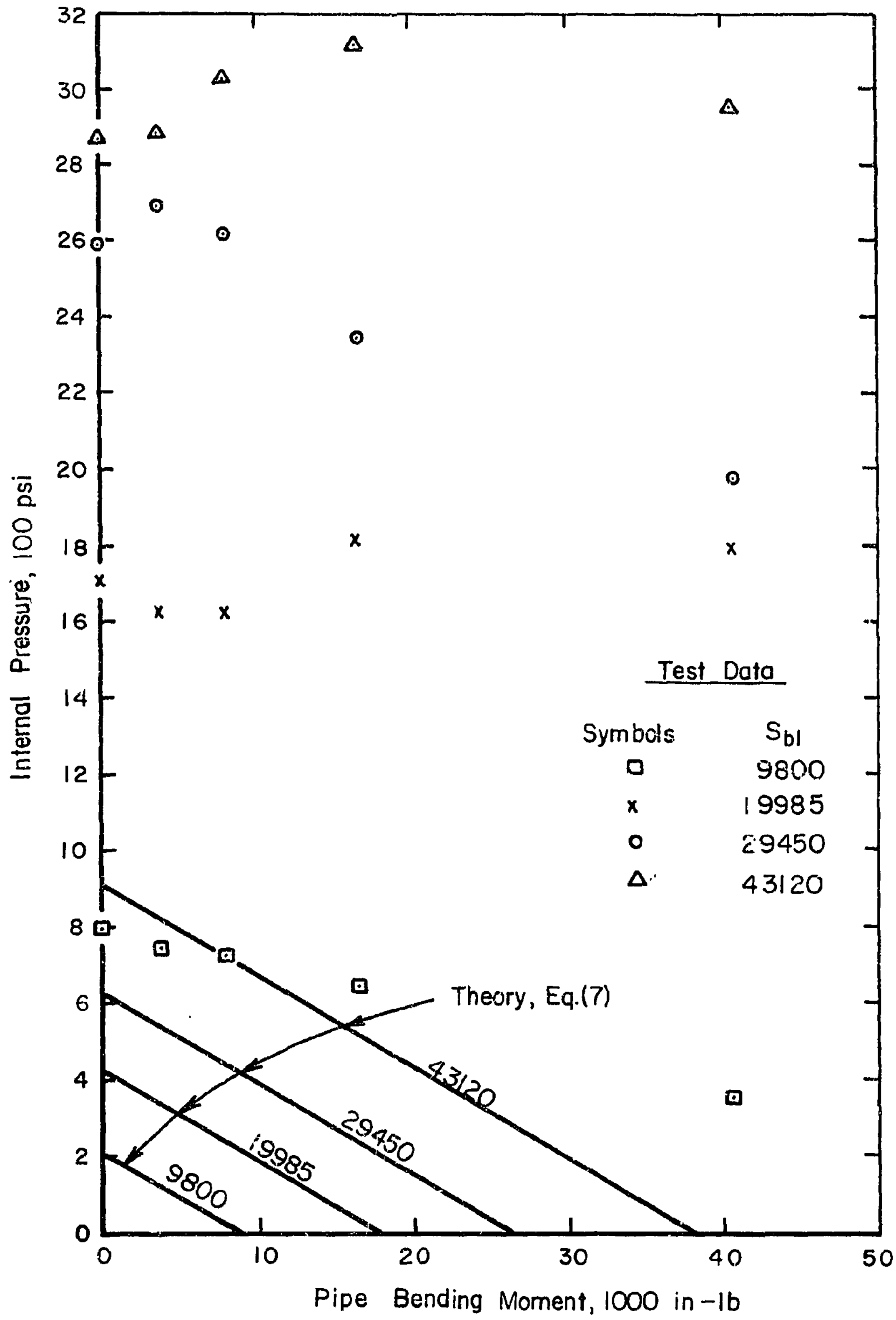

FIGURE A13: LEAKAGE PRESSURES, 3"-150 FLANGED JOINT, TEST DATA FROM REFERENCE (12), THEORY FROM TEXT EQ. (7).

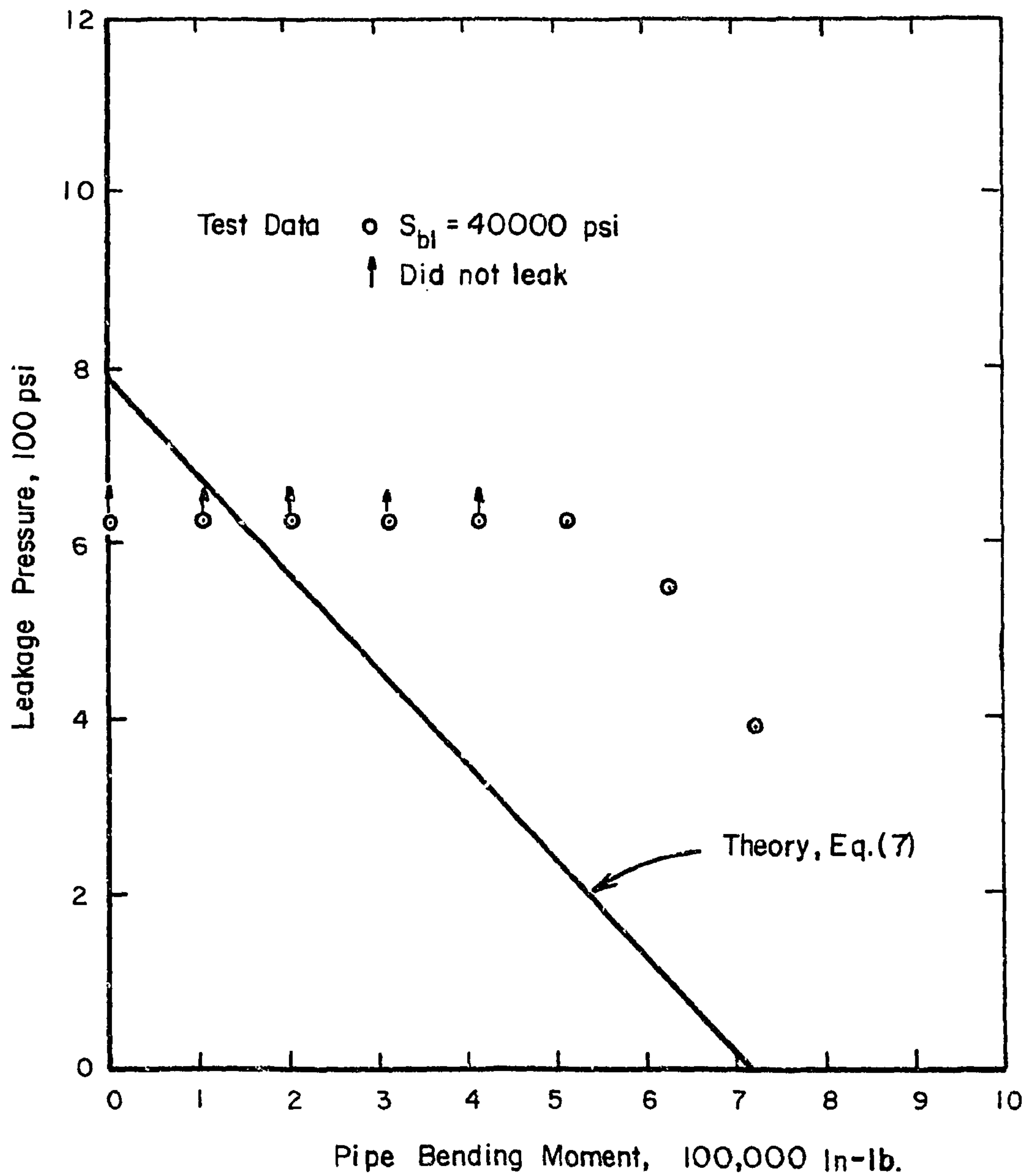

FIGURE A14: LEAKAGE PRESSURES, 12"-150 FLANGED JOINT, TEST DATA FROM REFERENCE (15), THEORY FROM TEXT EQ. (7).

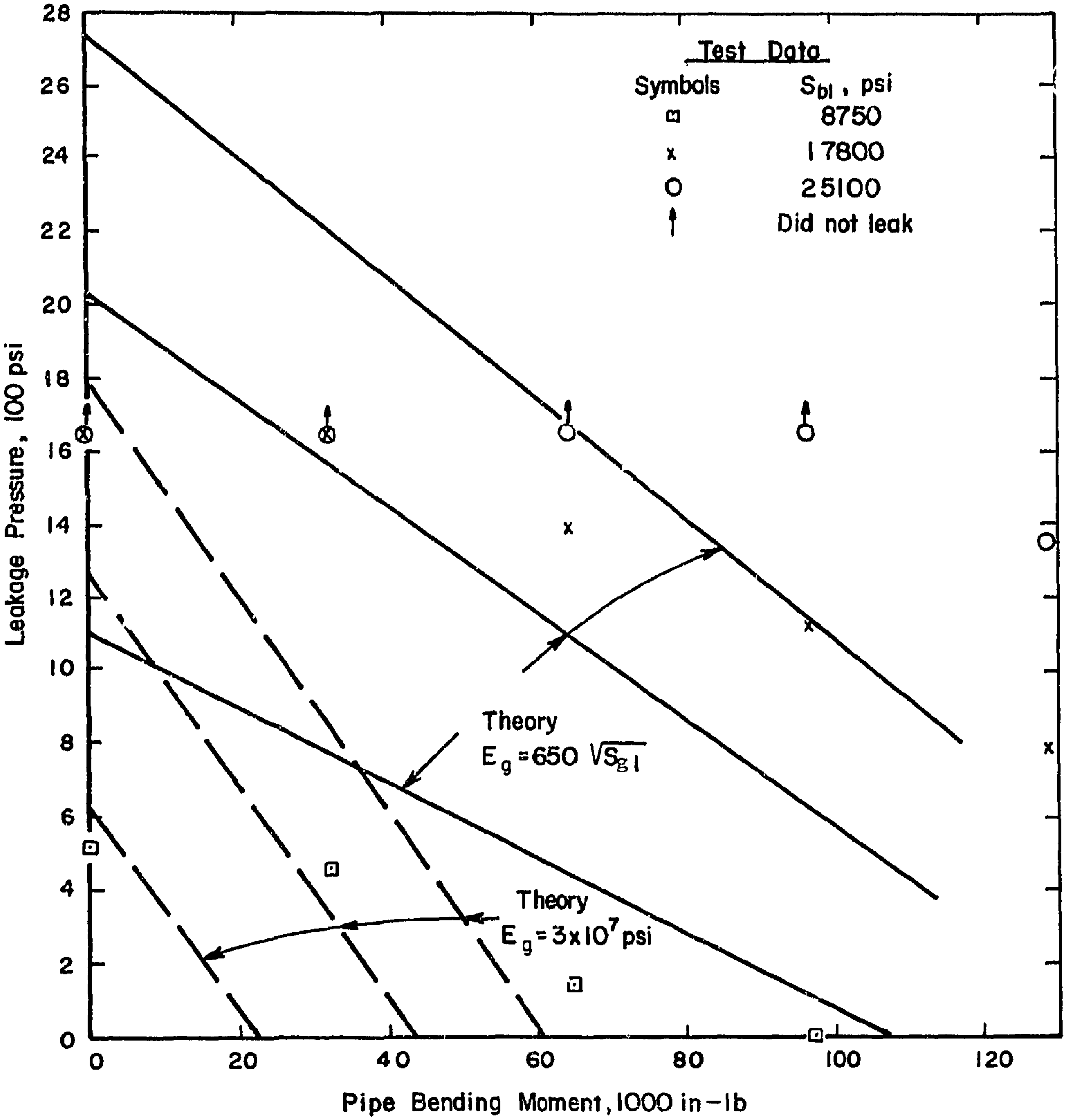

FIGURE A15: LEAKAGE PRESSURES, 4"-300 FLANGED JOINT, TEST DATA FROM REFERENCE (10), THEORY FROM TEXT EQ. (17).

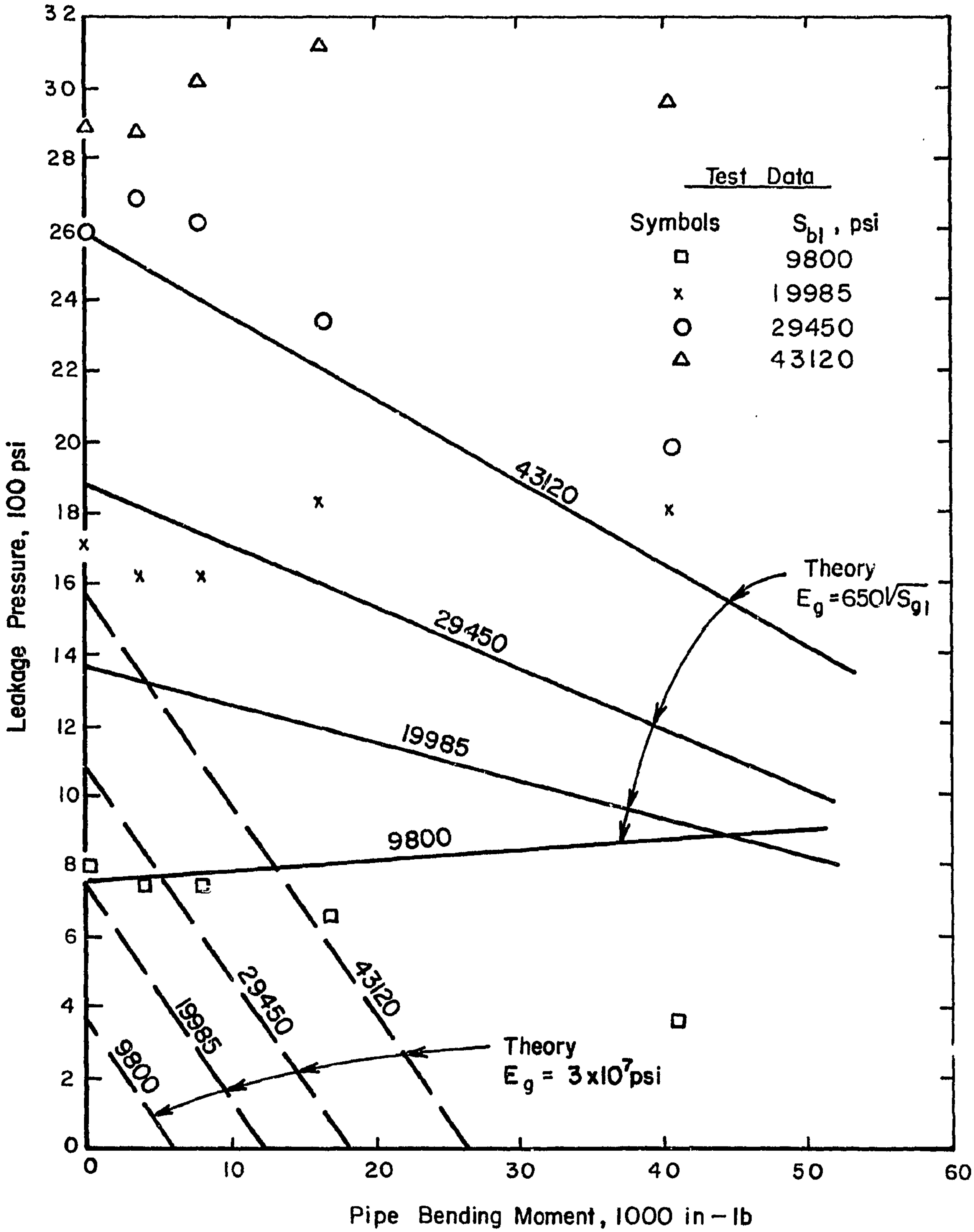

FIGURE A16: LEAKAGE PRESSURES, 3"-150 FLANGED JOINT, TEST DATA FROM REFERENCE (12), THEORY FROM TEXT EQ. (17).

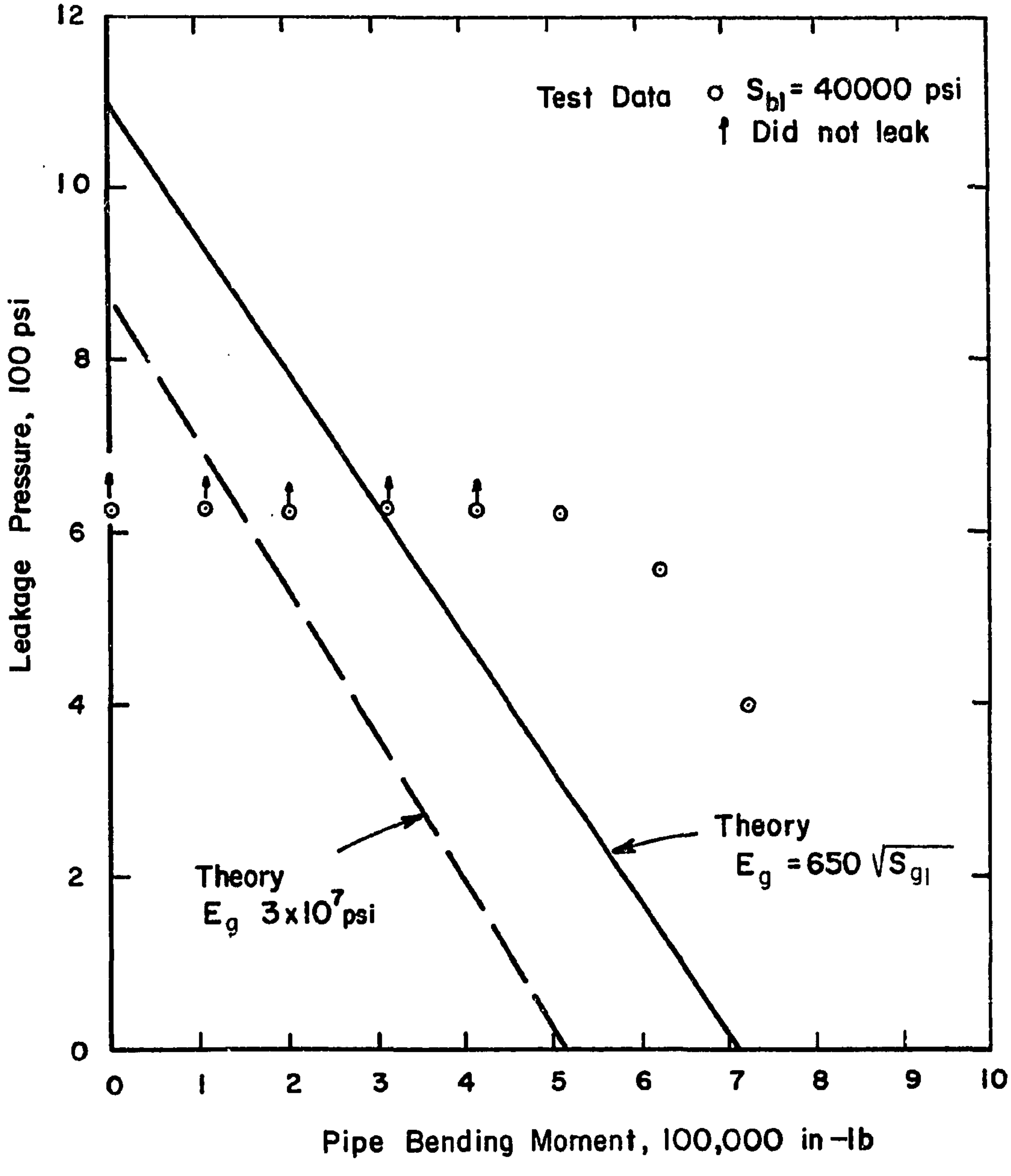

FIGURE A17: LEAKAGE PRESSURES, 12"-150 FLANGED JOINT, TEST DATA FROM REFERENCE (15), THEORY FROM TEXT EQ. (17).

Figures A15, A16, and A17 show the test data along with two sets of theoretical pipe bending moment vs pressure relationships. Both are based on Equation (17) of the text; with the "40,000" replaced by the initial bolt stress, S_{b1}, used in the tests. The first set, shown as dashed lines in Figure A15, A16, and A17, are based on a modulus of elasticity of the gasket of $E_g = 3 \times 10^7$ psi. These lines are also quite conservative with respect to the test data.

The calculations using $E_g = 3 \times 10^7$ psi were made because this is the assumed modulus used in calculations for the tables shown in the text. In our opinion, $E_g = 3 \times 10^7$ psi is an appropriate representation of the characteristics of an asbestos gasket in a B16.5 flanged joint after the bolts have been tightened to 40,000 psi and the joint has been in service for several years. However, at this point, we are attempting to see how theory correlates with test data, when the tests were partly run with low bolt stresses and the tests were run immediately (within a few hours) after making up the joint.

Reference (10) summarizes tests on various 1/16" and 1/8" thick asbestos gasket materials compressed between platens to determine the effective modulus of elasticity. Not all details of these tests are known (e.g., size of test piece; time-effects, if any) and they are non-linear; E_g increases with gasket stress, S_g. However, over the range of S_g from 200 to 6,000 psi, E_g is at least approximately represented by the equation:

$$E_g = 650 \sqrt{S_{g1}} \tag{A9}$$

where $S_{g1} = S_{b1}A_b/A_g$.

The solid lines shown in Figures A15, A16, and A17 are based on a_1 and a_2 values calculated using E_g from Equation (A9). It can be seen in Figures A15, A16, and A17 that use of these more realistic (but still very approximate) values for E_g gives significantly improved correlation with the test data.

One notes in Figure A16 that the solid line for S_{b1} = 9,800 slopes upward. The theory indicates that with S_{b1} = 9,800 and the corresponding S_g = 818 psi, E_g = 18,590 psi, the joint is "self-sealing" for pipe bending moments. We do not see such a trend in the test data for S_{b1} = 9,800 psi but such a trend is apparent for S_{b1} = 19,985 psi and S_{b1} = 43,120 psi.

Another bit of pertinent data is given in the paper by Markl and George[(16)]. In running cyclic pipe bending loading fatigue tests on 4" - 300 class flanged joints with 0.237" wall attached pipe, they note that an initial bolt stress of 40,000 psi was sufficient to prevent leakage, but that at an initial bolt stress of 24,000 psi was not sufficient for "specimens subjected to high deflections". Their results indicate that they imposed bending stress of S_{pb} up to 60,000 psi in the attached pipe. In most tests, the internal pressure was essentially zero (25" head of water to detect leakage). In some tests, an internal pressure of 600 psi was applied.

Values of S_{pb} calculated by text Equations (7) and (17) are as follows

S_{b1}, psi	Pressure, psi	Calculated S_{pb} (psi) Eq. (7)	Eq. (17)
24,000	0	20,950	15,430
	600	8,350	10,010
40,000	0	34,920	25,720
	600	22,320	20,300

Both Equations (7) and (17) agree with the test data in that they indicate that a bolt stress of 24,000 psi is <u>not</u> sufficient to withstand high (S_{pb} = 40,000 to 60,000 psi) deflections. However, both equations are conservative with respect to the test data, which indicated that, with a bolt stress of 40,000 psi, the joints did not leak at values of S_{pb} up to 60,000 psi. The equations indicate leakage when S_{pb} is around 30,000 psi(P = 0) or around 20,000 psi (P = 600 psi).

One additional aspect from the Markl and George[(16)] paper that is worth comment is their fatigue tests results using welding neck flanges. In the text*, the conclusion is reached that insofar as fatigue evaluation of the flange subjected to cyclic pipe bending moments is concerned, the stresses in the flange due to tightening the bolts can be ignored. An example is cited where S_{pb} (range) was assumed to be 24,200 psi and, using the Code evaluation method, the usage fraction per cycle was 1/60,000 or a design life of 60,000 cycles. From Figure 9 of Markl and George's paper, the cycles to failure (stress amplitude of 12,100 psi) is 3,400,000 cycles. Recognizing that the Code method incorporates a factor of safety of 20 on cycles, the value of 3,400,000/20 = 170,000 compares adequately with the Code value of 60,000 cycles.

In summary of the test data versus theory comparisons:

(1) The Code theory, Equation (7), is conservative with respect to the test data.

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* Chapter on FLANGE STRESSES, Basic Theory Flange Stress, Effect of Pipe Bending Moment on Flange Stresses.

(2) The basic theory, Equation (17) using $E_g = 3 \times 10^7$ psi, is conservative with respect to the test data.

(3) The basic theory, Equation (17) using $E_g = 650\sqrt{S_{g1}}$, instead of $E_g = 3 \times 10^7$ psi, gives better agreement with the test data and, considering the many uncertainties involved in the tests and approximations involved in the theory, the agreement is about as good as should be expected.

APPENDIX B

DEVELOPMENT OF SIMPLE EQUATIONS FOR LIMITATION OF PIPE BENDING AND TORSIONAL MOMENTS

TABLE OF CONTENTS

INTRODUCTION

The text of this report presents two methods of evaluating the pressure and pipe bending moment capacity of flanged joints; i.e., the method given in NB-3647.1 and the method incorporated in the computer program FLANGE. The NB-3647.1 method for limiting pipe bending moments, despite its complexity, can only be described as a very crude approximation. The computer program FLANGE, while theoretically sounder, never-the less uses rough approximations in the analysis for pipe bending moments. As discussed in Appendix A, both methods appear to be highly conservative with respect to available test data.

In this appendix, we will develop simple equations for limitation of pipe bending moments starting with consideration of bolting strength requirements for normal and upset conditions. We then consider the flange strength requirements and then examine limits for emergency and faulted conditions. The equations developed herein are included as recommendations for Code revisions in Appendix C.

BOLTING STRENGTH REQUIREMENTS

In the text, we have discussed criteria for minimum bolt area which involve the concept of a "leakage pressure"; i.e., Equations (8) and (10). Hopefully, the bolts will be prestressed to $S_{bl} > S_{bc}$ so that gross leakage will not occur. However, the Code does not directly impose any minimum on the bolt prestress. Practically, a minimum bolt prestress becomes a necessity in order to pass the hydrostatic test and/or not leak too much in service. The aim of the Code rules is to provide a flanged joint which has sufficient strength so that the bolts can be prestressed sufficiently to prevent gross leakage under sustained loadings encountered in normal or upset conditions. In addition, the Code rules should provide a flanged joint which will withstand dynamic loads encountered in normal or upset conditions and, after the dynamic loads stop, the joint will return to a condition where gross leakage will cease.

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Finally, the Code rules should provide a flanged joint of sufficient strength so that the bolts do not break as a result of emergency or faulted conditions.

The stress in the bolts due to internal pressure is conservatively given by the equation:

$$\sigma_{bp} = (\pi/4)\, D_f^2\, P/A_b \tag{B1}$$

where

σ_{bp} = bolt stress due to pressure
D_f = outside diameter of raised face
P = internal pressure
A_b = total bolt area

The stress in the bolts due to a pipe bending moment can be developed by assuming that the bolt area is uniformly distributed along the bolt circle. The imaginary cylindrical shell so formed has a thickness of $A_b/\pi C$, where A_b = total bolt area, C = bolt circle diameter. The maximum stress in this imaginary cylindrical shell due to pipe bending moment is then:

$$\sigma_{bm} = \frac{M}{Z} = \frac{M_{pb}}{(\pi/4)\, C^2\, A_b/(\pi C)} = \frac{M_{pb}}{C\, A_b/4} \tag{B2}$$

Equation (B2) is based on the assumption that the neutral axis coincides with a centerline of the imaginary pipe. This is not necessarily the case for a flanged joint, but Equation (B2) will not be unconservative by more than 11 percent, regardless of the neutral axis location. An equation analogous to (B2) could also be developed using the discrete bolt areas; the value of σ_{bm} would then depend on the orientation of the neutral axis. If we assume the bolt diameter is small compared to the bolt circle diameter, then Equation (B2) gives the same answer as the discrete bolt area approach when a pair of bolts are on the neutral axis. If the bolts straddle the neutral axis, Equation (B2) gives higher stresses than the discrete bolt area approach; by factors of 1.41, 1.08, and 1.04 for 4, 8, and 12 bolts, respectively.

Having Equations (B1) and (B2), we now seek a simple equation to establish the maximum bending moment that can be imposed on the flanged joint as limited by the strength of the bolting.

According to ANSI B16.5 (Par. 6.9.6) the stress calculated by Equation (B1) does not exceed 7,000 psi when the pressure is equal to the primary rating pressure. Table B1 shows the calculated values of σ_{bp} using Equation (B1). It can be seen from Table B1 that the statement in B16.5 is essentially correct; only the 16" and 20" Class 900 slightly exceed 7,000 psi. It can also be seen from Table B1 that, for most sizes and classes, the value of σ_{bp} is well below 7,000 psi. However, the highest bolt stress due to pressure, in relationship to Code allowable bolt stresses, occurs at the 100 F rating pressure rather than at the primary rating pressure. The 100 F rating pressure does not exceed 2.4 times the primary rating pressure, hence, the upper bound to σ_{bp} at the 100 F rating pressure is 2.4 x 7,000 = 16,800 psi. If the Code allowable bolt stress at 100 F is 20,000 psi or higher, we can conservatively assume that the bolt strength needed to hold the design pressure will not exceed the Code allowable bolt stress, S_b.

The lower limit on S_b of 20,000 psi at 100 F is deemed desirable so that the bolts can be prestressed to 40,000 psi without violation of NB-3232. In the remainder of this portion on bolting strength requirements, we will use S_b to indicate the allowable bolt strength as given in Code Table I-1.3 However, from flange strength considerations, we cannot use a value of S_b greater than 25,000 psi. Accordingly, in examples and Tables, we will be using S_b = 25,000 psi and, in our final recommendations, S_b will be replaced by 25,000 psi.

Having assigned 1.0 S_b of the total bolt strength to pressure loading, what remains for moment loading? In NB-3232.1, we note that the bolt stress is permitted to be 2.0 S_b. The extra 1.0 S_b, for our purpose, can be allotted to bolt strength needed to hold the pipe bending moment. We then simply use Equation (B2), with S_b used in place of σ_{bm}, and solve

TABLE B1: BOLT STRESSES BY THE EQUATION $\sigma_{bp} = (\pi/4)\ D_f^2\ \overline{P}/A_b$

Nom. Size	σ_{bp} = bolt stress at primary rating pressure, psi						
	150	300	400	600	900	1,500	2,500
1/2	440	880	--	1,770	--	1,840	3,070
3/4	670	830	--	1,660	--	2,780	4,630
1	930	1,170	--	2,330	--	2,810	4,690
1-1/4	1,460	1,820	--	3,640	--	4,390	5,570
1-1/2	1,930	1,610	--	3,220	--	4,420	5,570
2	1,920	1,920	--	3,830	--	4,620	5,850
2-1/2	2,480	1,660	--	3,320	--	4,550	5,740
3	3,640	2,440	--	4,880	5,270	5,060	6,600
3-1/2	2,200	2,950	--	4,250	--	--	--
4	2,790	3,730	3,590	5,380	4,650	6,070	6,690
5	2,610	5,210	5,010	5,720	5,090	5,600	6,630
6	3,520	4,700	4,510	5,150	5,850	6,140	6,690
8	5,500	5,290	5,360	6,090	5,760	6,600	6,960
10	3,810	4,340	4,380	5,150	6,220	6,930	6,200
12	5,270	4,550	4,760	5,710	6,890	6,250	7,000
14	4,710	4,270	4,460	5,390	6,640	5,680	--
16	4,570	4,340	4,650	5,740	7,200	5,870	--
18	4,460	4,660	5,000	6,180	6,765	6,170	--
20	4,280	5,590	4,930	6,180	7,050	6,160	--
24	4,710	5,190	4,910	6,330	6,110	6,250	--

$\overline{P}$ = primary rating pressure, psi

A_b = total bolt area, sq-in.

D_f = outside diameter of raised face, inches.

for M_{pb} to obtain:

$$M_{pb} \leq C\ A_b S_b/4 \tag{B3}$$

Equation (B3) provides a limit on M_{pb} based on strength of the bolts; it does not necessarily limit M_{pb} so that gross leakage will not occur. There are no test data available in which flanged joints were loaded with pressure and/or pipe bending moment sufficient to break (or even yield) the bolts. However, the data given in Appendix A show that moments well in excess of these permitted by Equation (B3) have been applied to flanged joints.

Equation (B3) provides a very simple method for limiting M_{pb} with respect to bolt strength. It would be very useful to obtain a similar simple equation for limiting M_{pb} with respect to joint leakage. NB-3647.1 provides a method for limiting M_{pb} but its complexity is not compatible with its accuracy. The computer program FLANGE is deemed to be a more accurate method, but it can not be made a part of Code rules. Let us see what happens if we consider Equation (B3) as representing the essential variables (C, A_b, S_b) needed to represent joint leakage, but multiply the right hand side by one-half; i.e.,

$$M^*_{pb} = C\ A_b\ S_b/8 \tag{B4}$$

Table B2 compares allowable pipe bending stresses as obtained from Equation (B4) with allowable pipe bending stresses obtained by the NB-3647.1 procedure and from the computer program FLANGE. It can be seen in Table B2 that Equation (B4) is a reasonably good approximation of the moment limits calculated using the computer program FLANGE.

To check Equation (B4) experimentally, we would like to have test data on flanged joints in which the bolt prestress is 40,000 psi,

TABLE B2: COMPARISON OF BENDING STRESS IN TYPICAL [1] ATTACHED PIPE

		S_{pb} (1,000 psi)		
Class	Size	Eq. (B4)	FLANGE[2]	NB-3647.1[2]
150	4	11.8	13.7	13.6
	8	5.3	7.5	6.5
	16	8.3	12.6	11.5
	24	10.6	16.3	16.1
300	4	18.5	20.4	14.2
	8	12.2	14.1	9.1
	16	18.6	24.8	19.0
	24	17.1	22.1	17.3
400	4	25.6	20.1	20.0
	8	16.0	19.1	11.7
	16	21.0	27.0	20.3
	24	18.9	24.2	19.3
600	4	27.7	27.0	13.8
	8	21.0	23.0	12.7
	16	19.4	22.6	14.9
	24	17.0	18.1	12.9
900	4	45.4	40.8	24.1
	8	27.7	27.2	15.1
	16	17.1	17.4	9.6
	24	19.3	21.8	13.9
1,500	4	42.7	33.7	13.8
	8	27.9	25.7	11.5
	16	27.3	27.1	15.0
	24	23.4	23.8	13.6
2,500	4	54.8	34.7	10.6
	8	36.0	27.9	10.3
	12	31.5	25.8	10.4

(1) Typical pipe is defined as pipe with wall thickness equal to the larger of std. wt. or $P D_o/17{,}500 + 0.05$.

(2) Values from text Table 9.

internal pressure is equal to the rated pressure and the pipe bending moment is increased until leakage occurs. Data given in Appendix A is not ideal in this sense, but it can be used. Figure A14 represents perhaps the best test for the purpose because the initial bolt stress was 40,000 psi. Equation (B4), with S_b = 25,000 psi, gives M_{pb} = 267,000 in.-lb. It can be seen in Figure A14 that there was no leakage at this moment, even though the pressure was 625 psi rather than the rated pressure of 275 psi at 100 F. Indeed, the data suggest that the moment could be doubled before leakage at 625 psi occurs. Figures A12 and A13 also indicate that Equation (B4) is highly conservative.

It is pertinent at this point to discuss the magnitudes of moments that can be applied to flanged joints in comparison to the moments that can be applied to straight pipe. Table B3 shows the stresses in "typical" attached pipe where the maximum moment is calculated by Equation (B3) with S_b = 25,000 psi. Typical pipe is defined as pipe with wall thickness equal to the larger of std. wt. wall or $\overline{P}\, D_o/17{,}500 + 0.05$.

As a yardstick for comparison, the stress in straight pipe due to thermal expansion and anchor movements is permitted to be 3 S_m. The value of S_m for A-106 Grade B pipe at 100 F is 23.3 ksi; 3 S_m = 69.9 ksi. It is apparent from Table B3 that many small size flanges have an "allowable pipe bending stress" by Equation (B3) that is greater than 69.9 ksi. On the other hand, for many flanged joints, the permissible moment will be severely restricted; e.g., the 8" - 150 class where the allowable moment is 10.6/69.9 times that permitted on 8" Std. wt. A-106 Grade B pipe at 100 F.

Table B4 shows the stresses in "typical" attached pipe where the maximum moment is calculated by Equation (B4) with S_b = 25,000 psi. These values, of course, are simply one-half of those shown in Table B3. It is apparent in Table B4 that only a few small size flanged joints will not be restricted in permissible moment as compared to "typical" straight pipe. The larger sizes of the 150 class are severely restricted; e.g., the 8" - 150 class is limited to a moment of 0.076 times that for 8" std. wt. A-106 Grade B pipe at 100 F. In the smaller sizes of the 150 class, the 3" size

TABLE B3: BENDING STRESSES IN TYPICAL [1] ATTACHED PIPE WITH MAXIMUM MOMENTS PERMITTED BY EQUATION (B3) USING S_b = 25,000 PSI.

Nom Size	S_{pb} (ksi) for Class Indicated						
	150	300	400	600	900	1,500	2,500
1/2	184	203	--	203	--	564	519
3/4	123	232	--	232	--	327	292
1	74.1	133	--	133	--	276	239
1-1/4	47.0	83.4	--	83.4	--	157	192
1-1/2	37.4	104.2	--	104.2	--	156	192
2	42.8	90.0	--	90.0	--	168	178
2-1/2	26.1	83.4	--	83.4	--	147	154
3	17.6	58.0	--	58.0	86.6	117	127
3-1/2	29.5	45.7	--	63.4	--	--	--
4	23.5	36.9	51.2	55.3	90.9	85.4	110
5	23.6	25.6	35.6	53.1	75.1	83.5	97.6
6	16.9	28.3	39.3	55.9	60.6	67.5	88.5
8	10.6	24.3	32.0	41.9	55.3	55.9	72.0
10	15.0	28.1	37.1	46.7	46.3	49.0	[illegible]4.5
12	12.2	29.5	37.9	40.3	39.8	53.7	62.9
14	14.5	34.6	44.1	41.2	38.5	58.3	--
16	16.7	37.2	42.0	38.8	34.2	54.6	--
18	18.5	38.5	39.4	35.6	36.9	51.8	--
20	20.4	32.3	38.6	34.7	34.0	48.8	--
24	21.2	34.2	37.8	34.0	38.5	46.7	--

(1) Typical pipe is defined as pipe with wall thickness equal to the larger of std. wt. wall or $F D_o/17{,}500 + 0.05$.

TABLE B4: BENDING STRESS IN TYPICAL (1) ATTACHED PIPE WITH MAXIMUM MOMENTS PERMITTED BY EQUATION (B4) USING S_b = 25,000 PSI.

Nom. Size	S_{pb} (ksi) for Class Indicated						
	150	300	400	600	900	1,500	2,500
1/2	91.9	101	--	101	--	282	260
3/4	61.3	116	--	116	--	164	146
1	37.0	66.5	--	66.5	--	138	119
1-1/4	23.5	41.7	--	41.7	--	78.3	96.2
1-1/2	18.7	52.1	--	52.1	--	78.2	96.1
2	21.4	45.0	--	45.0	--	83.8	88.8
2-1/2	13.1	41.7	--	41.7	--	73.4	76.8
3	8.8	29.0	--	29.0	43.3	58.6	63.7
3-1/2	14.7	22.9	--	31.7	--	--	--
4	11.8	18.5	25.6	27.7	45.4	42.7	54.8
5	11.8	12.8	17.8	26.5	37.5	41.8	48.8
6	8.4	14.2	19.6	28.0	30.3	33.7	44.2
8	5.3	12.2	16.0	21.0	27.7	27.9	36.0
10	7.5	14.0	18.6	23.4	23.1	25.0	37.2
12	6.1	14.8	19.0	20.2	19.9	26.8	31.4
14	7.3	17.3	22.1	20.6	19.2	29.2	--
16	8.3	18.6	21.0	19.4	17.1	27.3	--
18	9.2	19.2	19.7	17.8	18.4	25.9	--
20	10.2	16.2	19.3	17.4	17.0	24.4	--
24	10.6	17.1	18.9	17.0	19.3	23.4	--

(1) Typical pipe is defined as pipe with wall thickness equal to the larger of std. wt. wall or $P\,D_o/17{,}500 + 0.05$.

is severely limited. These results correlate with field experience in that leakage problems occur relatively frequently with the 3" and 8" 150 class.

Equation (B4) is deemed to provide a suitable limit for sustained moment loads. If the bolts are prestressed to 40,000 psi, the flanged joint would not be expected to exhibit gross leakage with pressure up to the rated pressure in combination with moments up to that permitted by Equation (B4). For moments which act over a short period of time, the higher moment limit of Equation (B3) is deemed appropriate. Leakage may occur, but the bolts will remain elastic because the total strength requirement is, at most, 2 S_b. The value of S_b is not greater than one-third of the bolt material yield strength.

FLANGE STRENGTH REQUIREMENTS

Table 18 of the text shows calculated flange stresses for a bolt preload stress of 40,000 psi. As long as the pressure and pipe bending loads do not reduce the gasket stress to zero, the flange stresses (except in the hub) will not change significantly. By limiting the pressure to the rated pressure and the sustained moment to $S_b/2$, we are essentially limiting the loads so that a prestress of 40,000 psi will not be exceeded. The worst case is the 16" 900 class at 100 F for which $\sigma_{bp} + \sigma_{bm} =$ 7,200 x 2.4 + 12,500 = 29,780 psi. For short-time moments, the worst case limit is 7,200 x 2.4 + 25,000 = 42,280 psi. B16.5 flanges are deemed to have ample strength to withstand these loads.

In the above, we have taken S_b = 25,000 psi, the allowable value for SA-193 Grade B7 at temperatures up to 700 F in Code Table I-7.3. If we take S_b from Code Table I-1.3 (Class 1), we note that the allowable stress for SA-193 Grade B7 is 35,000 psi at 100 F and that other bolt materials have allowable stresses as high as 50,000 psi at 100 F. Now, if the flanges were capable of supporting a bolt prestress of 70,000 psi or 100,000 psi, the values of S_b from Table I-1.3 could be used in Equations (B3) and (B4) with correspondingly larger permissible values of M_{pb}. The data in Table 18, along with experience with B16.5 flanged joints, is sufficient to convince the authors that all B16.5 flanges are capable of supporting a bolt prestress of 40,000 psi. We can speculate that most B16.5 flanges are capable of carrying bolt prestresses considerably higher than 40,000 psi. Indeed, Tables A4 through A9 show test data where bolt prestresses were much higher than 40,000 psi; in Table A7, the bolt prestress was as high as 161,000 psi. However, in Table A9, it is noted that flange yielding occurred at a bolt stress of 78,500 psi. Elastic theory cannot be used to justify bolt prestresses in excess of 40,000 psi; note that in Table 18 we have stresses far above the yield strength of many B16.5 flange materials even for a bolt stress of 40,000 psi. Accordingly, the higher allowable bolt stresses given in Table I-1.3 cannot be used because we are not sure of the strength of the flanges. The preceding consideration also leads to the rejection of higher allowable bolt stresses given in Table I-7.3; e.g., for SA-540, Grade B21, Class 1 bolt material.

The flange strength considerations lead to the recommendation that S_b in Equations (B3) and (B4) be replaced with the specific value of 25,000 psi. This does not prohibit the use of bolt material with higher allowable stresses. Further, in order to assure that 40,000 psi prestress can be applied without exceeding the requirement of NB-3232.1, Equations (B3) and (B4) should be restricted to bolt materials which have an allowable bolt stress in Table I-1.3 of 20,000 psi. This will rule out, in particular, SA-193 Grade B8, B8C, B8M and B8T bolt materials.

The change in flange strength as temperature increases must also be considered. The pertinent flange material property is its yield strength. In addition, we recognize, at this point, that available test data and most field experience are with a material like SA-105, with a specified minimum yield strength of 36,000 psi and that there are other flange materials with lower yield strength; e.g., SA-182, Type 304 with a specified minimum yield strength of 30,000 psi. Accordingly, it is recommended that the limit for sustained moment loading during normal and upset conditions be given by the equation:

$$M_{fs} \leq 3,125\ (S_y/36)CA_b \tag{B5}$$

Where S_y is the yield strength (ksi) of the flange material at the design temperature; to be obtained from Code Table I-2.2. For short-time moments:

$$M_{fd} \leq 6,250\ (S_y/36)CA_b \tag{B6}$$

The value of $(S_y/36)$ should not be taken as greater than unity; otherwise, we might violate bolt prestress limitations and/or bolt strength requirements.

It is significant to note that the B16.5 rated pressures decrease with increasing temperature approximately in proportion to the decrease in flange material yield strength with increasing temperature. Accordingly, both allowable pressures and moments decrease with decreasing S_y.

EMERGENCY CONDITIONS

NB-3655.1 states* that the pressure under emergency conditions shall not exceed the design pressure by more than 50 percent. Because the design pressure cannot be higher than the B16.5 rated pressure, the emergency condition pressure cannot exceed the B16.5 test pressure of 1.5 times the 100 F rating pressure. Accordingly, the pressure limit of NB-3655.1 is quite acceptable for B16.5 flanged joints. However, the question arises as to what co-incident pipe bending moment can be permitted. We note that for normal and upset conditions we have used up about all of the assured margin of flange strength.

* NB-3655.1 states that "permissible pressure shall not exceed the design pressure (P) calculated in accordance with Equation (2) of NB-3641.1 by more than 50 percent." The words "calculated, in accordance with Equation (2) of NB-3641.1"should be deleted as they are at best, useless. If the user does calculate P by Equation (2), he will simply end up with the answer that P = P.

Recognizing that emergency conditions require shutdown for correction of the conditions or repair of damage in the system (NB-3113.3), it could be contended that flange yielding would be acceptable and would be part of the "repair of damage". However, there is no basis available for estimating the magnitude of the leak if, for example, we allowed the pressure <u>and</u> the moment to be 50 percent higher than permitted for normal and upset conditions. The bolt stress would then be (for the worst case, 16" - 900 class) 7,200 x 3.6 + 25,000 x 1.5 = 63,420 psi. The authors can speculate that most of, if not all, B16.5 flanges can withstand this loading but there is neither test data nor theoretical calculations that can be used to defend such speculation for <u>all</u> B16.5 flanges. However, we can take advantage of the fact that for most B16.5 flanges, σ_{bp} by Equation (B1) is less than 7,200 psi (see Table B1). We can, therefore, require for emergency conditions that:

(a) The pressure shall not exceed 1.5 times the design pressure, and

(b) $M_{fd} \leq [11,250\ A_b - (\pi/16)\ D_f^2\ P_{fd}]C\ (S_y/36)$ (B7)

where

P_{fd} = pressure under emergency conditions concurrent with M_{fd}
M_{fd} = bending or torsional moment (considered separately) as defined for M_{fs} but including dynamic loadings, in.-lb.
D_f = outside diameter of raised face
S_y = yield strength of flange material at design temperature, ksi. $(S_y/36) \leq 1.00$.

Bending stresses in typical attached pipe calculated by Equation (B7) are shown in Table B5, along with stresses calculated by Equation (B6) and results from the text Table 9. In most flanges, the permissible moment is higher for emergency conditions than for normal/upset conditions, even though the emergency condition pressure is 1.5 times the rated pressure.

TABLE B5: COMPARISONS OF BENDING STRESSES IN TYPICAL ATTACHED PIPE AT 100 F AND 700 F, SA-105 FLANGE MATERIAL.

		S_{pb} (1,000 psi)					
				Temp = 100 F		Temp = 700 F	
Class	Size	FLANGE (1)	NB-3647.1 (1)	Eq. (B6) (2)	Eq. (B7) (3)	Eq. (B6) (2)	Eq. (B7) (3)
150	4	13.7	13.6	23.5	35.1	16.9	27.6
	8	7.5	6.5	10.6	12.6	7.6	11.1
	16	12.6	11.5	16.7	21.6	12.0	18.2
	24	16.3	16.1	21.2	27.1	15.3	23.0
300	4	20.4	14.2	36.9	46.6	26.5	34.9
	8	14.1	9.1	24.3	25.2	17.5	19.4
	16	24.8	19.0	37.2	43.7	26.8	33.0
	24	22.1	17.3	34.2	36.0	24.6	27.6
400	4	20.1	20.0	51.2	65.8	36.8	48.8
	8	19.1	11.7	32.0	32.8	23.0	25.1
	16	27.0	20.3	42.0	47.5	30.2	35.8
	24	24.2	19.3	37.8	41.3	27.2	31.3
600	4	27.0	13.8	55.3	56.7	39.8	43.6
	8	23.0	12.7	41.9	38.7	30.1	30.3
	16	22.6	14.9	38.8	36.2	27.9	28.1
	24	18.1	12.9	34.0	30.2	24.5	23.8
900	4	40.8	24.1	90.9	102.8	65.4	78.0
	8	27.2	15.1	55.3	53.7	39.8	41.7
	16	17.4	9.6	34.2	26.1	24.6	21.2
	24	21.8	13.9	38.5	35.4	27.7	27.7
1,500	4	33.7	13.8	85.4	79.1	61.4	61.9
	8	25.7	11.5	55.9	47.5	40.2	37.7
	16	27.1	15.0	54.6	52.1	39.3	40.5
	24	23.8	13.6	46.7	42.0	33.6	33.1
2,500	4	34.7	10.6	110.0	91.7	79.1	72.9
	8	27.9	10.3	72.0	57.4	51.8	46.1
	12	25.8	10.4	62.9	49.8	45.3	40.0

(1) Leakage pressure limits, from Table 9 of text.
(2) Strength limits, normal and upset conditions. S_y = 36 ksi at 100 F at 100 F, S_y = 25.9 ksi at 700 F.
(3) Strength limits, emergency condition with co-incident pressure of 1.5 times the B16.5 rated pressure for carbon steel flange material.

Usually, the design pressure is somewhat less than the B16.5 rated pressure; for such cases, Equation (B7) will tend to permit higher moments than Equation (B6). However, the permissible moment for emergency conditions will not exceed 1.8 times that for normal/upset conditions; this occurs for $P_{fd} = 0$.

FAULTED CONDITIONS

NB-3656, by reference in Code Appendix F, states that the pressure under faulted conditions shall not exceed 2.0 times the design pressure. While this may be above the B16.5 test pressure, it is deemed acceptable for B16.5 flanged joints both from a bolt strength and flange strength standpoint. However, for both pressure and moment loadings, it is not apparent that anything more liberal than Equation (B7) can be recommended. While under faulted conditions, all that is required is that the flanged joint restrict leakage enough to permit a safe shutdown, it is not apparent to the authors what that permissible leakage rate might be and, if it were known, how the permissible leakage rate could be correlated with the leakage characteristics of flanged joints where yielding of the flanges might occur.

TORSIONAL MOMENTS

The discussion so far has concerned bending moments. Flanged joints may also be subjected to torsional moments applied by the attached pipe. These torsional moments are resisted by frictional forces between the flange faces and the gasket. Large rotations would be prevented by contact between the bolts and bolt holes but, because bolt diameters are 1/8" less than hole diameters, we should not depend upon the bolt shear resistance when concerned about leakage.

With a coefficient of friction μ between flange faces and gasket, the torque capacity M_{pt} of a flanged joint is

$$M_{pt} = \int_{o}^{2\pi} \frac{G}{2} \cdot dF \qquad \text{(B8)}$$

where dF is the torsional resistance per inch of gasket circumference, and G is the gasket diameter. The unit axial force (lb/inch) on the gasket is $S_{b2}A_b/\pi G$, and $dF = \mu(S_{b2}A_b/\pi G)\ (G/2)\ d\phi$, where ϕ is the co-ordinate angle. Accordingly:

$$M_{pt} = \int_{o}^{2\pi} \left(\frac{G}{2}\right)^2 \frac{\mu\ S_{b2}A_b}{\pi G}\ d\phi = \frac{\mu\ G\ S_{b2}A_b}{2} \qquad \text{(B9)}$$

We want to compare equation (B5) with equation (B9). We note that $G < C$, and S_{b2} (at operating conditions) may be less than the preload bolt stress of (assumed) 40,000 psi. For high loads at the gasket interface, a coefficient of friction of $\mu = 1/2$ appears reasonable. If we assume that $G/C = 1/2$ and use $S_{b2} = 25,000$ psi(the same S_{b2} assumed for bending moment evaluation) then equation (B9) gives

$$M_{pt} = \frac{1}{2} \cdot \frac{1}{2} \cdot \frac{25000}{2}\ C\ A_b = 3125\ C\ A_b \qquad \text{(B10)}$$

Equation (B5) gives $M_{fs} = 3125\ C\ A_b$ (for $S_y = 36$ ksi) hence, insofar as leakage is concerned, we can apply the same limit to torsional moments as for bending moments. The $(S_y/36)$ factor is not needed for torsional moments but, for simplicity, equation (B5) will be recommended for both bending moments and torsional moments--considered separately. These can be considered separately because the bending moment does not reduce the total load on the gasket; it decreases the load on the tension side but increases the load on the compression side.

For short time loadings, where we do not necessarily design to prevent temporary gross leakage, we first observe that the assumptions that $G/C = 0.5$ and $S_{b1} = 25,000$ psi are quite conservative for most B16.5 flanged joints and most operating conditions. G/C is more typically equal to about 0.7. For pressure loading only, S_{b2}/S_{b1} is more typically about 0.75, hence, if $S_{b1} = 40,000$ psi, $S_{b2} = 30,000$ psi. With these more typical ratios, equation (B9) gives $M_{pt} = 5250\ C\ A_b$ in.-lb. This is almost as high as the limit given by the analogous equation (B6) of $M_{fd} = 6250\ C\ A_b$ in.-lb. In addition, large movements are prevented by the bolts. If we assign an allowable shear stress for the bolts of 13,000 psi, the torsional moment capacity of the bolts alone is $13000\ C\ A_b/2 = 6250\ C\ A_b$ in.-lb. Accordingly, equation (B9) is deemed as a suitable limit for short-time torsional moments as well as short-time bending moments.

Insofar as the authors are aware, no test data exists on the torsional moment capacity of flanged joints. While the limits developed in the preceding are deemed to be conservative, it would be highly desirable to obtain some test data.

APPENDIX C

RECOMMENDED CODE REVISIONS

NB-3600

It is recommended that the NB-3600 portion be revised as follows:

(1) Change NB-3651.2 to read:

NB-3651.2 Piping Products for Which Stress Indices Are Not Available. For analysis of flanged joints, see NB-3658. For other piping products for which stress indices are not available, see NB-3680.

(2) Change NB-3647.1 to read:

NB-3647.1 Flanged Joints.

(a) Flanged joints manufactured in accordance with the standards listed in Table NB-3132-1, as limited by NB-3612.1, shall be considered as meeting the requirements of NB-3640.
(b) Flanged joints not included in Table NB-3132-1 shall be designed in accordance with Article XI - 3000, including the use of the appropriate allowable stress given in Table I-7.

(3) Add a New Paragraph NB-3658 to Read as Follows:

NB-3658 Analysis of Flanged Joints

Flanged joints using flanges, bolting, and gaskets as specified in ANSI B16.5 (1968) and using a bolt material having an S_m - value at 100 F not less than 20,000 psi may be analyzed in accordance with the following rules or in accordance with NB-3200. Other flanged joints shall be analyzed in accordance with NB-3200.

NB-3658.1 Design, Normal and Upset Conditions

(a) Bolting. The bolting shall meet the requirements of NB-3232. In addition, the limitations given by Equations (15) and (16) shall be met.

$$M_{fs} \leq 3125 \ (S_y/36) C \ A_b \quad , \qquad (15)$$

where

M_{fs} = bending or torsional moment (considered separately) applied to the joint due to weight, thermal expansion of the piping, sustained anchor movements, relief valve steady-state thrust, and other sustained mechanical loads, in-lb. If cold springing is used, the moment may be reduced to the extent permitted by NB-3672.8.

S_y = yield strength (ksi) of flange material at design temperature (Table I-2.2). The value of $(S_y/36)$ shall not be taken as greater than unity.

C = diameter of bolt circle, in.

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

$$M_{fd} \leq 6250\ (S_y/36)\ C\ A_b \quad , \tag{16}$$

where

M_{fd} = bending or torsional moment (considered separately) as defined for M_{fs} but including dynamic loadings, in-lb.

(b) Flanges. Flanges of ANSI B16.5 flanged joints meeting the requirements of NB-3612.1 are not required to be analyzed under NB-3650. However, the pipe-to-flange welds shall meet the requirements of NB-3652, NB-3653, and NB-3654, using appropriate stress indices from Table NB-3683.2-1.

NB-3658.2 Emergency Conditions

(a) The pressure shall not exceed 1.5 times the design pressure.

(b) The limitation given by Equation (17) shall be met.

$$M_{fd} \leq [11250\ A_b - (\pi/16)\ D_f^2\ P_{fd}]\ C\ (S_y/36) \quad , \tag{17}$$

where

D_f = outside diameter of raised face, in.

P_{fd} = pressure (psi) during the emergency condition concurrent with M_{fd}.

M_{fd}, C, S_y, the limitation on (S_y/36) and A_b are defined in NB-3658.1(a).

(c) Pipe-to-flange welds shall be evaluated by Equation (9) of NB-3652, using a stress limit of 2.25 S_m.

NB-3658.3 Faulted Conditions

(a) The pressure shall not exceed 2.0 times the design pressure.

(b) The limitation given by Equation (17) of NB-3658.2(b) shall be met, where P_{fd} and M_{fd} are pressures (psi) and moments (in-lb) occurring concurrently during the faulted condition.

(c) Pipe-to-flange welds shall be evaluated by Equation (9) of NB-3652, using a stress limit of 3.0 S_m.

NB-3658.4 Testing Conditions. Analysis for Testing Conditions is not required.

NC-3600

It is recommended that the NC-3600 portion be revised as follows.

(1) NC-3651, Add:

(c) For analysis of flanged joints, see NC-3658.

(2) NC-3671.1, Delete Period After "NC-3647" and Add:

and NC-3658.

(3) NC-3673.5(a)

Change equation numbers: (12) to (15) and (13) to (16).

(4) NC-3673.5(b)

Change "Formulas (12) and (13)" to "Equations (15) and (16)".

(5) Change NC-3647.1 to Read:

NC-3647.1 Flanged Joints.

(a) Flanged joints manufactured in accordance with the standards listed in Table NC-3132-1, as limited by NC-3612.1, shall be considered as meeting the requirements of NC-3640.
(b) Flanged joints not included in Table NC-3132-1 shall be designed in accordance with Article XI - 3000.

(6) Add a New Paragraph NC-3658 to Read as Follows:

NC-3658 Analysis of Flanged Joints

Flanged joints using flanges, bolting, and gaskets as specified in ANSI B16.5 (1968) and using a bolt material having an S - value at 100 F not less than 20,000 psi may be analyzed in accordance with the following rules or in accordance with NB-3200. Other flanged joints shall be analyzed in accordance with NB-3200. If the NB-3200 analysis is used, allowable stresses from Table I-7 and I-8 shall be used in place of S_m.

NC-3658.1 Design, Normal and Upset Conditions

(a) The limitations given by Equations (12) and (13) shall be met.

$$M_{fs} \leq 3125\ (S_y/36)\ C\ A_b \quad , \qquad (12)$$

where

M_{fs} = bending or torsional moment (considered separately) applied to the joint due to weight, thermal expansion of the piping, sustained anchor movements, relief valve steady-state thrust and other sustained mechanical loads, in-lb. If cold springing is used, the moment may be reduced to the extent permitted by NC-3673.5.

S_y = yield strength (ksi) of flange material at design temperature (Table I-2.2). The value of $(S_y/36)$ shall not be taken as greater than unity.

C = diameter of bolt circle, in.

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

$$M_{fd} \leq 6250\ (S_y/36)\ C\ A_b \quad , \qquad (13)$$

where

M_{fd} = bending or torsional moment (considered separately) as defined for M_{fs} but including dynamic loadings, in-lb.

(b) Flanges. Flanges of ANSI B16.5 flanged joints meeting the requirements of NC-3612.1 are not required to be analyzed under NC-3650. However, the pipe-to-flange welds shall meet the requirements of NC-3652, using appropriate stress intensification factors from Figure NC-3673.2(b)-1.

NC-3658.2 Emergency Conditions

(a) The pressure shall not exceed 1.5 times the design pressure.

(b) The limitation given by Equation (14) shall be met.

$$M_{fd} \leq [11250\ A_b - (\pi/16)\ D_f^2\ P_{fd}]\ C\ (S_y/36) \quad , \qquad (14)$$

where

D_f = outside diameter of raised face, in.

P_{fd} = pressure (psi) during the emergency condition concurrent with M_{fd}.

M_{fd}, C, S_y, the limitation on $(S_y/36)$ and A_b are defined in NC-3658.1(a).

(c) Pipe-to-flange welds shall be evaluated by Equation (9) of NC-3652, using a stress limit of 1.8 S_h.

NC-3658.3 Faulted Conditions

(a) The pressure shall not exceed 2.0 times the design pressure.

(b) The limitation given by Equation (14) shall be met, where P_{fd} and M_{fd} are pressures (psi) and moments (in-lb) occurring concurrently during the faulted condition.

(c) Pipe-to-flange welds shall be evaluated by Equation (9) of NC-3652, using a stress limit of 2.4 S_h.

NC-3658.4 Testing Conditions. Analysis for Testing Conditions is not required.

ND-3600

It is recommended that ND-3600 be changed as shown for NC-3600; changing NC to ND in all places where NC occurs. In ND-3673.5, the present equation numbers (9) and (10) are to be changed to (15) and (16) in two places. Do not change the phrase: "using Equations (9) and (10) (ND-3650).".