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BACKGROUND OF ANSI B16.5 PRESSURE-TEMPERATURE RATINGS

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ABSTRACT

The history of the development of dimensions and ratings in ANSI B16.5-1968 is outlined. Comparisons are made between the ratings and the design basis for flanges given in the ASME Boiler Code, Section VIII, Division 1. While the ratings do not meet the criteria of the ASME Code, both theory and service experience indicate that the criteria given in that code are neither necessary nor sufficient for flanged joints in piping subjected not only to internal pressure but also to thermal gradients and loadings imposed on the joint by the attached pipe. Several recurring questions with respect to the ratings are discussed.

INTRODUCTION

The scope of ANSI B16.5, "Steel Pipe Flanges and Flanged Fittings"[1] is summarized in Table 1. Pressure-temperature ratings are given for this wide scope of flanges, flanged fittings and valves. The pressure ratings depend only on:

(1) The pressure class

(2) The flange, flanged fitting or valve body material

(3) The service temperature (contained fluid temperature)

The pressure-temperature ratings given in API 600, "Flanged and Butt-Welding-End Steel Gate and Plug Valves for Refinery Use"[2] are identical to those given in B16.5, hence the discussion of the ratings given herein is, to a major extent, also applicable to API 600 ratings.

During the author's some twenty-five years of association with the pressure-temperature ratings of B16.5, several questions concerning the ratings have repeatedly been asked. These are listed below.

(1) Why are B16.5 ratings not proportional to allowable stresses in the ASME Boiler Code?

(2) Why is the 150-lb class rated differently than all of the other pressure classes?

(3) Why do field problems with leakage of 3" and 8" 150-lb class flanges occur?

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(4) Why are B16.5 ratings acceptable under the ASME Boiler Code in those cases where it can be shown that they do not meet the rules given in the ASME Boiler Code, Section VIII, Div. 1, Appendix II, "Rules for Bolted Flange Connections"?

(5) What is an appropriate initial bolt stress for B16.5 flanged joints?

(6) What is the effect of the modulus of elasticity of flanges, bolts, and gaskets on the performance of a flange joint?

This paper will review the background of the ratings in a broad sense, not necessarily in the order of, or specifically directed to, the above questions. However, the summary herein does give answers to the specific questions.

FLANGED FITTING AND VALVE BODY WALL THICKNESS

Table 1 indicates a significant aspect of B16.5 ratings; i.e., the dimensional coverage of flanges, bolting, and gaskets is almost complete whereas the dimensional coverage of fitting bodies and valves is restricted to minimum wall thicknesses and center-to-face dimensions. The minimum wall thicknesses tabulated in B16.5 are from 0.10" to 0.20" heavier than those given by the equation:

$$t = 1.5 \left[\frac{P_p d}{2S - 1.2\, P_p} \right] \qquad (1)$$

where t = calculated thickness, inches

P_p = primary service pressure, pounds per square inch

(P_p = class designation, e.g., for 150-lb class, P_p = 150 psi)

d = inside diameter of fitting or port opening of valve (as taken from B16.5 tables), inches

S = stress of 7000 psi.

Comparisons of t calculated by equation (1) with tabulated maximum wall thicknesses in B16.5 and API 600 are shown in Table 2. The reader will probably recognize that equation (1) is equivalent to equations used in the ASME Boiler Code[3] for calculation of the required minimum wall thickness of cylindrical shells, with an additional factor of 1.5, and zero corrosion allowance. For reasons that will become apparent later, the author prefers to interpret the allowable stress of 7000 psi as an allowable stress of 8750 psi for the material at the primary rating temperature, multiplied by a casting quality factor of 0.8; i.e. 8750 x 0.8 = 7000 psi.

An implied assumption in equation (1) is that the "shape factor" to compensate for the fact that flanged fitting bodies and valve bodies are not cylinders is 1.5. If this is accurate, then the thickness in excess of that required by equation (1) could be considered as corrosion allowance. In the case of API 600, the wall thickness in excess of that required by B16.5 is specifically identified as an additional corrosion allowance. However, it is not necessarily true that the shape factor of 1.5 is adequate for all fitting bodies and valves. This is recognized in B16.5 by the words:

> "Additional metal thickness needed for assembly stresses, valve closing stresses, shapes other than circular, and stress concentrations must be determined by individual manufacturers since these factors vary widely. In particular, 45 degree laterals, tru Ys, crosses, etc., may require additional reinforcement to compensate for inherent weakness in products of this shape."

In the case of valves, considerations of deformation limits may impose additional restrictions on minimum appropriate body wall thickness so that the valve will operate and seat properly with not only pressure loading but also when subjected to loads by the attached pipe.

In general, therefore, there are some significant dimensions of fitting and valve bodies which are not established in B16.5. Further, from the standpoint of establishing the validity of pressure-temperature ratings, even if those dimensions were established, an acceptable theoretical method of assessing the safe internal pressure of such complex shapes is not available. In contrast, the dimensions of flanges, bolting, and gaskets are reasonably well established in B16.5. Further, there are, and have been for many years, acceptable theoretical methods of assessing the safe internal pressure capacity of flanged joints. Accordingly, as discussed in the next section, the ratings in B16.5 have been and continue to be established, extrapolated and/or rationalized on the basis of the strength of the flanged joints. The user obtains assurance of the structural adequacy of flanged fittings and valves for the rated pressures and temperatures by the required hydrostatic of 1.5 times the 100 F rating pressure and, perhaps to a lesser extent, by the specified minimum body wall thicknesses.

HISTORICAL DEVELOPMENT OF DIMENSIONS AND RATINGS*

Prototype Cast Iron Flanges and Flanged Fittings

The dimensional standardization of ASA B16.5 flanges can be traced back to their prototype cast iron flanges and flanged fittings. The first step toward standardization of cast iron flanges was taken by the American Society of Mechanical Engineers in about 1887 when a committee was appointed to obtain the views of manufacturers of pumps, steam engines, valves, and fittings on the possibility of standardizing dimensions of flanges. Agreement was reached on a standard template for flange drilling; this became known as the "ASME Standard" used for 75 psi and 200 psi. In 1901, a Manufacturers Standard for pressures up to 250 psi came into existence. These early standards apparently fixed only the bolt circles and the number and size of bolts. Center-to-end of flanged fittings and their metal thicknesses were dependent upon the individual manufacturers.

From 1900 to 1910, an increasing need was felt for a more complete standardization of flanges and flanged fittings. The ASME, in conjunction with other interested organizations, agreed on dimensions for 125-and 250-lb cast iron flanges and flanged fittings. In 1911, this was approved by the U.S. Department of Commerce and it became known as the U.S. Standard.

The dimensions of the U.S. Standard were not acceptable to several principal manufacturers, resulting in the formation, in 1910, of a group known as the Committee of Manufacturers on Standardization of Fittings and Valves. This Committee later became the present Manufacturers Standardization Society of the Valve and Fittings Industry. The result of this Committee work was the publication in 1912 of a manufacturer's standard differing in many respects from the U.S. Standard published in 1911. During the next two years, compromise standard dimensions of pipe flanges and flanged fittings for 125 and 250 psi steam working pressure were evolved and published as the "American Standard for Pipe Flanges, Fittings and Their Bolting".

Soon after the publication of the American Standard in 1914, it became apparent that higher pressure hydraulic flange and fitting

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* This discussion is abstracted, in part, from Tube Turns Piping Engineering Papers 6.01 (September, 1948) and 6.02 (November, 1953). These papers (now out of print) were prepared by J.D. Mattimore, A.R.C. Markl, and the author of this paper. Background data were supplied by several members of ASA B16 Code Committees.

standards were needed. A subgroup was established by the Committee of Manufacturers on Standardization of Fittings and Valves to develop these new standards; specifically to be equal in strength to the three weights of steel pipe then in use; i.e., standard weight, extra strong, and double extra strong. This resulted in three classes of fittings known as the 800-, 1200- and 3000-lb Hydraulic Standards. The 800- and 1200-lb classes were intended to be made of either semisteel (high-strength cast iron) or cast steel; the 3000-lb class was to be made of cast steel. The three classes were given cold nonshock ratings of 800, 1200 and 3000 psi and pressure ratings under shock conditions of 500, 800 and 2000 psi.

The standards discussed above are significant in that they served as prototypes for the present ASA B16.5 flanges. Whether these early dimensions were related in any way to a "design basis" is not known. By 1900, the theory of plates and shells was well advanced; however, it seems quite unlikely that such theories were used in developing flange standards. One might expect some rough proportioning of the total bolt cross-sectional area to the total pressure load; as discussed later such a rough proportioning did exist. With regard to flange thicknesses, bolt circles, and flange outside diameters, it might be surmised that casting limitations, core-shift allowances, and minimum practical casting thicknesses were significant during the development stage from about 1850 to 1900.

The years from 1914 to 1923 saw the development of high-pressure steam plants. By 1923 several power plants were built to operate at 400 psi with 650 F total temperature and some 600-psi steam plants were under construction. During the same period, higher pressures and temperatures were being used in the rapidly growing oil refining industry. Cast iron was unsuitable for temperatures much above 450 F. These higher temperature requirements, therefore, led to a need for new standards for steel fittings and flanges. In the spring of 1920, the American Engineering Standards Committee (later to become the American Standards Association) organized a Sectional Committee on the Standardization of Pipe Flanges and Fittings; the predecessor of the present ANSI Sectional Committee B16. An organizational meeting of Subcommittee 3 was held on October 26, 1923, at which time a comprehensive program for the standardization of steel flanges and flanged fittings was launched, marking the beginning of the present American National Standard B16.5.

1927 Standard B16.e

In order to provide a uniform basis of design, semiempirical rules were established for the required bolt area and flange thickness. Prototype flanges were modified (to some extent) to comply with these rules. The pressure classes eventually included, along with their prototypes, were:

Primary Rating Pressure, 750 F	Prototype
250 psi	250-lb Cast Iron Standard
400 psi	250-lb Cast Iron Standard, with larger bolts and thicker flanges
600 psi	800-lb Hydraulic Standard
900 psi	1200-lb Hydraulic Standard
1350 psi	3000-lb Hydraulic Standard

According to the design rules, the bolt area was to be

$$A_b = \text{larger of: } \frac{A_{rf}}{7000} P_p \qquad (2)$$

$$\text{or } \frac{A_p + 12A_{st}}{14{,}000} P_p \qquad (3)^*$$

where A_b = total bolt area, sq. in.

A_{rf} = area to outside diameter of the raised face, sq. in.

A_p = area to inside diameter of small tongue facing, sq. in.

P_p = primary rating pressure, psi

A_{st} = area of small tongue facing, sq. in.

The alternate equations for bolt area seem contradictory in that equation (2) appears to contain an allowable bolt stress of 7000 psi; equation (3) an allowable bolt stress of 14,000 psi. At that time, however, a widely used rule-of-thumb for designing flange bolting provided for doubling the pressure; by this rule-of-thumb, equation (2) could be written:

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* The factor of 12 in Equation (3) corresponds to a gasket "m-factor" such as those given in the ASME Boiler Code, Section VIII, Div. 1. The value of 12 is considerably higher than any m-factor given at present in the ASME Code.

$$A_b = \frac{A_{rf} \times 2P_p}{14,000} \qquad (2a)$$

This would indicate that the allowable bolt stress intended in both equations (2) and (3) was 14,000 psi.

Actually, however, the bolting of the 250-lb class (later to become the 300-lb class) was taken directly from the existing 250-lb Cast Iron Standard. Equations (2) and (3) were used only as a rough lower limit for bolt areas for the 250-lb and 400-lb classes. The 600-, 900-, and 1350-lb classes follow equations (2) and (3) quite closely.

The flange thickness was to be obtained by the equation:

$$t = \sqrt{\frac{15(C-G)\ A_b}{7(\pi C-Nd)}} \qquad (4)$$

where t = flange thickness

C = bolt circle

G = outside diameter of small tongue facing

N = number of bolts

d = diameter of bolt holes

A_b = total bolt area.

Equation (4) was derived by considering the cross-section of the flange ring (hub ignored) as a cantilever beam, fixed at the bolt circle and loaded along the outside diameter of the small tongue facing. The "cantilever" design equation (4) contains an empirical factor based on tests of 8 and 16 inch sizes of 400-, 600-, and 900-lb tentative designs of cast steel rings (no hubs). The yield strength of the flange material was assumed to be 36,000 psi and the bolts were assumed to be tightened to 20,000 psi stress.

The actual flange thicknesses of larger sizes and higher pressure series are about the same as given by equation (4). The thicknesses of the 250-lb class were taken directly from the 250-lb Cast Iron Standard; these and the actual thicknesses of smaller sizes, in general, are significantly greater than required by equation (4). Possibly the thicknesses of smaller size flanges were well established by their prototypes and only the thicknesses of larger sizes were actually established by equation (4).

In addition to the primary pressure ratings at 750 F, ratings were given at 450 F (1.2 to 1.3 times primary ratings) and at air temperature (1.67 to 2.0 times primary ratings). The higher ratings at lower temperatures were presumably based in part on prior experience. The yield strength of carbon steel increases with decreasing temperature. In addition, and perhaps more important, thermal gradient stresses and bending loads imposed by the pipe on the flanged joint would, in general, decrease with decreasing (to ambient) temperature. Test pressures were set at 1.5 times the air temperature rating for the 250- and 600-lb class; 1.33 for the 400-, 900-, and 1350-lb class.

1932 Standard B16.e

The 1932 Standard added the 150-lb class, based on the 125-lb Cast Iron Standard. The 250-lb class was raised to the 300-lb class. The 1350-lb class was raised to the 1500-lb class and extended to include sizes 14 through 24 inches.

The bolting for the 150- and 1500-lb classes was checked against equations (2) and (3); an increase in bolt size was deemed necessary only for the 12 inch size in the 1500-lb class although the bolt stresses in the several sizes are slightly above the presumed limit of 14,000 psi. The 150-lb class bolting is more than ample by equations (2) and (3); stresses range from 400 to 11,400 psi.

The 1932 edition added dimensions for screwed and lapped flanges in all series. An analytical method given by Waters and Taylor[4] provided an improved means of designing flanges. This analysis considers the flange ring as an annular plate, the hub as a uniform wall cylinder, with shear loads and moments at the juncture of the cylinder to the ring such that continuity at the juncture is obtained.

Screwed flanges in the 150-lb class were given hub lengths the same as the 125-lb cast iron flanges (B16a-1928) and hub outside diameters, in sizes 1 through 8 inch, also the same as 125-lb cast iron. In the larger sizes, the 150-lb class had slightly larger hub diameters than the corresponding cast iron flanges. Screwed flanges in the 300-lb class were given slightly (1/16 to 1/4 inch) longer hubs than their 250-lb cast iron prototypes, with the same hub diameters as the 250-lb cast iron in sizes through 12 inches; slightly larger diameters in the larger sizes. The hub dimensions of the smaller size screwed flanges (through 12 inch in the 150 lb, through 8 inch in the 300-, 400-, 600-, and 900-lb class, and through 3 inch in the 1500-lb class) were the same as for lapped flanges. In the larger sizes, the hubs of screwed flanges were made significantly shorter than the corresponding lapped flanges. These large sizes of screwed flanges are, therefore, relatively weak.

The primary ratings remained at 750 F, except for the 150-lb class which was rated at 150 psi at 500 F and 100 psi at 750 F. The comparatively low ratings of the 150-lb class continues to this day.

1935 Standard B16.e

The 1935 Addendum dealt entirely with pressure-temperature ratings. Ratings below 750 F were given in 50 F increments, starting with a secondary rating point of 100 F instead of the prior "at or near the ordinary range of air-temperatures". The ratio of secondary to primary rating pressure was set at 1.67 for all classes except the 150 lb, which was decreased from 1.67 to 1.53; i.e., from 250 psi to 230 psi.

New ratings were added for temperatures above 750 F; up to 850 F for steam service, to 1000 F for oil service. The ratings above 750 F were based on consideration of creep strength.

Test pressures were set at 1.5 times the air temperature rating for the 150-, 300-, 400-, and 600-lb classes; 1.4 for the 1500-lb class; 1.33 for the 900-lb class.

1939 Standard B16.e

The pressure-temperature ratings were expanded in two ways:

(1) For the first time, ratings were established for materials other than carbon steel, specifically for carbon-molybdenum and "equivalent steels". Standard faced carbon-moly steel flanges were given the ratings established for five percent chrome-moly steel in Section 3 of the Code for Pressure Piping, ASA B31.1-1935. Their primary rating temperature was 900 F; above this temperature the ratings paralleled the decrease in creep strength of the material. The secondary rating pressure was twenty percent higher than for carbon steel, presumably on the basis of the higher yield strength of the five percent chromium material. Ratings between 100 F and 900 F were linearly interpolated.

(2) Separate ratings were established for flanges with ring-joint gaskets. These ratings were taken from the 1937 issue of API Standard 5-G-3. Experience and tests[5] indicated that flanged joints with ring-joint gaskets withstood higher internal pressures without leakage than the same flanged joints with other types of gaskets and facings in use at the time. The ring-joint ratings

were set at 6/5 of ratings established for standard facings for carbon-moly and equivalent materials and for carbon steel at 100 F. Two anomalies which appeared in these ratings were:

(a) For carbon steel flanges above the primary rating temperature, no distinction was made between standard facings and ring-joint facings.

(b) The hydrostatic test pressures for carbon steel flanges were unchanged from the 1935 standard. For ring-joint flanges, this greatly reduced the margin available between maximum working pressure and test pressure; for example, the 900-lb class had a test pressure only 11 percent higher than the secondary (100 F) rating pressure. (This anomaly remained until the 1949 edition.)

The size range and types of flanges covered in ASA B16.e was increased in the 1939 edition as listed below:

(1) The 1500-lb class was extended to include companion flanges in sizes 14 through 24 inches. In setting new dimensions, specifically the hubs of lapped flanges, methods developed for the 1932 issue were used.

(2) The size range in all classes was extended downward to 1/2 inch. The dimensions were obtained by extrapolation; the number of bolts was set at a minimum of four and the minimum size at 1/2 inch. The 400- and 900-lb class constituted exceptions: 400 lb, 3 1/2 inch and smaller, were made the same as the 600 lb; 900 lb, 2 1/2 inch and smaller, were made the same as the 1500-lb class.

(3) Welding neck flanges were added in all classes. They were given the same dimensions as other types except for the hub length and the diameter at the welding end, which was made the same as the outside diameter of matching pipe. For 150- and 300-lb class welding neck flanges, the hub lengths were made practically the same as had been established by the Heating and Piping National Contractors in 1930; the dominant consideration was to provide sufficient length to prevent warping the flange face during welding (Oxy-acetylene welding, in use at that time, tends to produce more warpage than present-day electric arc welding). Hub lengths of higher pressure classes were made roughly parallel to those established for the 150- and 300-lb class. The hubs of the welding neck flanges were considered to provide about the same strength as lap-joint flanges; hence, no check calculations were made at that time.

(4) Slip-on flanges were added in the 150- and 300-lb class. These were made to be the same overall dimensions as screwed flanges and, hence, could be considered at least equal in strength.

(5) Blind flanges were added in all pressure classes. Their thickness was made the same as that of the flange ring of companion flanges. It is not known whether they were checked at the time by the formulas provided for flat heads in the ASME Boiler Construction Code or the API-ASME Unfired Pressure Vessel Code, but checks made later show them to be roughly equivalent in strength to the other types.

(6) The 2500-lb class was added in sizes 1/2 through 12 inches. The following design basis* was used for the flanges and fittings of this class:

(a) The inside diameter of the fitting and flanges was made slightly larger than the inside diameter of the pipe calculated by the Barlow formula, using an allowable stress of 7000 psi and a corrosion allowance.

(b) The outside diameter of the flange hub was made equal to the outside diameter established for the fittings on the basis of a fitting wall 1.5 times as thick as the calculated pipe wall, plus an allowance for core shift.

(c) The size and number of bolts was determined so that the bolt stress required to contain the hydrostatic end load did not exceed 7000 psi (equation 1).

(d) The bolt circle was selected so as to provide the necessary wrench clearances.

(e) The outside diameter of the flange was made just large enough to provide nut bearing surface.

(f) The hub length, measured from the center of the flange thickness to the end of the hub was made equal to $\sqrt{Dt}$, where D = flange ID, t = hub thickness; but not less than the length of the 1500-lb flange of the same size.

The test pressure of the 2500 lb class was set at 1.4 times the air temperature rating for raised face flanges. The same test pressure was assigned for ring-type joint facing, resulting in a ratio of test pressure to air temperature rating pressure of 1.17.

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* As given in the report, "Development of 2500-pound Fittings and Flanges", by E.C. Petrie of Crane Co. (April 4, 1936).

1943 Standard B16.e

In 1941, Subcommittees 3 and 4 of ASA B16 appointed a special subgroup to undertake an analytical review of the ratings established in ASA B16.e-1939. In 1937, an improved method* by Waters, et al.[6] of calculating stresses became available. The subgroup, using the improved analysis, set out to analyze most, if not all, of the types, sizes, and facings of flanges covered by ASA B16.e. At that time, this was an ambitious undertaking although today, thanks to high-speed digital computers, the stresses in the several thousands of possible combinations of types, sizes, and facing-gaskets could be calculated in a few minutes of computer time.

Unfortunately, before the work planned was well under way the United States became involved in World War II. Utmost conservation of materials, particularly the scarce alloys, became a necessity. Time being of the essence, the original extensive program of analysis had to be abandoned in favor of limiting probing calculations. From these calculations, a "representative" flange stress at the primary pressure was obtained. This representative stress* was then used to establish primary rating temperatures from allowable stress-temperature curves based on a safety factor of 4 and a 90 percent quality factor applying to both forgings and castings.

As a result of these studies, the following changes in ratings were made in ASA B16.e-1943.

(1) The primary rating temperatures of carbon steel flanges were raised from 750 to 850 F for ring-joint facings. Primary rating temperatures for carbon-moly flanges remained at 900 F and 950 F for standard and ring-joint facings, respectively.

(2) The secondary ratings (100 F) of carbon steel flanges were increased to equal those for carbon-moly flanges, with a corresponding increase in test pressures.

(3) Ratings between 100 F and the primary rating temperature (in previous editions, obtained by linear interpolation) were made roughly proportional to the yield strength of the flange material as a function of temperature. This resulted in a large increase in allowable working pressure at and near 450 F.

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* This method is now incorporated in the ASME Boiler Code[3]. Discussion of stresses in B16.5 flanged joints calculated by this method and the significance of the representative flange stress is covered later herein in the section on "Calculation of Stresses by the ASME Method".

The ratings of the 150-lb class remained unchanged, except that steam ratings were brought up to the 1939 oil ratings; the differences between oil and steam ratings were thereby abolished. It was not deemed necessary to introduce ratings for carbon-moly steel flanges in the 150-lb class.

No new flange types were introduced in the 1943 issues, but slip-on flanges, made to threaded flange overall dimensions, were permitted in all pressure classes at the same ratings as other types.

1949 Standard B16.e

Following the end of World War II, the special subgroup of Subcommittees 3 and 4, appointed in 1941, was asked to review the ratings on both the 1939 edition and 1943 edition and decide which should be used pending a proposed complete revision of the standard. This review resulted in Supplement No. 1 to ASA B16.e-1939, issued under the designation ASA B16.e6-1949.

The ratings issued in 1943 were, for the most part, reaffirmed. Service temperature limitations, however, were modified to reflect differences in steam power plant and oil refinery practice. The oil industry continued use of the 1943 standard upper temperature limits, the power industry reverted to the former limit of 850 F for carbon steel flanges and set limits of 900 and 950 F for carbon-moly with standard and ring-joint facings, respectively. This decision was influenced, in part, by adverse experiences with graphitization at elevated temperatures.

A major change in ratings consisted of establishing test pressures for all classes and facings at 1.5 times the secondary rating, paralleling pressure vessel code practice where the test pressure is normally 1.5 times the cold allowable working pressure.

The 1949 issue made no changes in flange types except that slip-on flanges in classes above 300 lb were eliminated.

1953 Standard B16.5

An extensive analytical study was made of all types, sizes, and facings included in ASA B16.e, completing the study started in 1941. Computations were made following the ASME Boiler Code[3] rules, with a number of proven types of gaskets and facings.

Based on these calculations, the committee deemed that the stress in ASA B16.5 flanged joints at the primary rating pressure could be represented by a stress of 8750 psi. The selection of a representative stress of 8750 psi was motivated by the need for extrapolation of existing carbon steel and carbon-moly flange ratings to a variety of other alloy steels; such ratings to be consistent with the allowable stresses for the various alloys established in the ASME Boiler Construction Code, and with the basic physical properties of the materials. First, therefore, the selected representative stress had to be consistent with established ratings for carbon and carbon-moly steels.

That the representative flange stress of 8750 psi was consistent with the ASME Code allowable stresses for carbon steel (SA105 Gr II) and carbon-moly steel (SA182 Gr F1) was established by noting the Code allowable stresses for carbon steel at 850 F and for carbon-moly steel at 950 F were essentially equal to 8750 psi. These temperatures were the primary rating temperatures established in ASA B16.e for ring-type joints. With respect to the secondary (100 F) rating of 2.4 times the primary rating pressure established in ASA B16.e for ring-joint facings, the representative stress at this pressure is 2.4 x 8750 = 21,000 psi. This stress is approximately 60 percent of the minimum yield strength found in tests of typical carbon and carbon-moly steels meeting flange material specifications.

These two correlations formed a basis for using the representative stress of 8750 psi as a means of extrapolating carbon and carbon-moly ratings to other alloy steels. In addition, it was noted that ASA B16.e ratings between 100 F and 650 F were roughly parallel to the yield strength of carbon steel at temperature. These correlations led to a rating procedure given in Appendix D of the 1953 issue of ASA B16.5. Briefly, the procedure consists of:

Up to 650 F:	Pressure ratings same as for carbon steel
650 F to primary rating temperature:	Pressure ratings obtained by linear interpolation
Primary rating temperature:	Temperature at which ASME Code allowable stress equals 8750 psi
Above primary rating temperature:	Rating pressure = $(S_c/8750)P_p$ S_c = ASME code allowable stress P_p = primary rating pressure
All temperatures:	Ratings shall in no case exceed $(0.6\ S_y/8750)\ P_p$, where S_y is the yield strength at temperature.

The rating procedure given above was used to establish "Class A" ratings for carbon steel and 15 alloy steels. From 100 to 650 F, ratings for all materials* were made the same as carbon steel, except for Type 304 material which, because of its relatively low yield strength at moderate temperatures, was assigned substantially lower ratings in this temperature range. At temperatures above the primary rating temperature, rating pressures are proportional to ASME Code allowable stresses for the particular material.

It is notable that the rating procedure uses a design criteria of excessive plastic deformation at all temperatures; i.e., yield strength at low temperatures, creep strength at high temperatures, since Code stresses at high temperatures are based on creep resistance.

In prior issues of ASA B16.e, two sets of ratings were given: one for ring-type joints, the other for "standard facings". The ratings for ring-type joints were generally about 6/5 of those for standard facings. In the 1953 issue, these two sets of ratings became Class A and Class B. Class A included: (a) ring-joint facing, (b) small tongue and groove facing used with any type gasket, (c) large tongue and groove facing used with any type gasket except flat solid metal, and (3) other facings and gaskets which result in no increase in bolt load or flange moment over these resulting from the facing-gasket combinations listed in (a) through (c). Class B ratings were applied to all facings and gaskets other than those listed under Class A rating.

Ratings of the 150-lb class constituted an exception to the rating procedure. This pressure class was retained at its long-established rating level.

With regard to flange types, coverage in ASA B16.5-1953 was enlarged by the addition of socket welding flanges in small sizes (1/2 to 2-1/2 or 3 inches) and the reinstatement of slip-on flanges in 400; 600; and 900-lb class and small sizes (1/2 to 2-1/2 inches) of the 1500-lb class. Socket welding flanges were made dimensionally the same as screwed flanges except for the bore and socket details.

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* The logic of this procedure might be questioned as applied to flange materials of high yield strength; e.g., ASTM A182 Grade F9, with a minimum specified yield strength of 70,000 psi. However, there are other considerations in flanged joint design as discussed later herein.

1957 Standard B16.5

The 1953 issue included under Class A ratings: "Other facings and gaskets which result in no increase in bolt load or flange moment over those resulting from the facing-gasket combinations listed in (a) through (c) above". The facing-gasket combinations listed in (a) through (c) were ring-type joints; small tongue and groove with any type gasket and large tongue and groove with any type gasket, except flat solid metal. The most common type of facing, i.e., raised face, could be given Class A ratings by restricting gasket dimensions to those of a small tongue and groove, for flat solid metal gaskets, or to those of a large tongue and groove for all other gaskets. This follows from ASME Boiler Code flange bolt load and moment calculations, which are dependent on the gasket dimensions, not the facing dimensions.

In order to more explicitly define the dimensions of gaskets usable with raised faces and acceptable for Class A ratings, extensive calculations of bolt loads and moments were made of various gaskets and gasket dimensions. These calculations followed the rules given in the ASME Code, including the gasket m-factors given therein. This work resulted in MSS SP-47, "Limiting Dimensions of Raised Face Flange Gaskets Which Meet the Requirements of ASA B16.5 for Class A Ratings". This MSS standard was incorporated almost entirely in the 1957 issue of ASA B16.5 as Appendix E. Because Appendix E includes practically all commonly used gaskets, Class B ratings became obsolete and were dropped.

Several new materials were included in the 1957 issue; however, since these were for subzero service (rated at the secondary rating pressure), no additions to the 1953 Class A rating tables were necessary.

In 1957, a Task Group was appointed by the chairman of Subcommittee 4 of ASA B16 to develop ratings for nonferrous flanges and flanged fittings dimensionally made to ASA B16.5. The rating basis used by this Task Group was analogous to the rating procedure given in Appendix D of ASA B16.5-1953, i.e.,

$$P = \frac{S_f}{21,000} P_s \tag{5}$$

where P = pressure rating for nonferrous flange at temperature, T

P_s = 100 F rating of carbon steel flange of corresponding pressure class as given in ASA B16.5-1957

S_f = allowable flange stress for the flange material at temperature, T

S_f = lowest of:

(a) 60 percent of the yield strength at temperature, T

(b) 100 percent of the stress to produce a secondary creep rate of one percent in 100,000 hours at temperature, T

(c) 100 percent of the stress to produce rupture in 100,000 hours at temperature, T.

The stress value of 21,000 psi comes from 2.4 x 8750 = 21,000 psi corresponding to the representative stress of carbon steel flanges at their secondary (100 F) rating pressure. Again, it is significant to note that the representative stress of 21,000 psi is used as a means of extrapolating ratings of carbon steel flanges to other materials and does not infer that ASA B16.5 flanges are stressed to this value under typical bolting-up and loading conditions. Application of this set of rating rules to carbon steel flanges would, of course, produce practically the same pressure-temperature ratings as given in ASA B16.5-1957 for carbon steel flanges.

The ratings obtained from the rating procedure were published as "1960 Addendum to B16.5-1957". These ratings covered nine nonferrous alloys. It should be noted that these ratings apply to flanged joints, not flanged fittings or valves. Also, the tabulated ratings for 6061-T6 aluminum alloy apply to flanges which either are not welded in installation (lapped joint, blind, threaded) or are not significantly reduced in strength by the welding process (welding neck). Slip-on and socket-welding types were rated at two-thirds of the tabulated values.

The ratings given in the 1960 Addendum (B16.5a-1960) are for wrought materials, except for the aluminum bronze alloy, ASTM B148, Alloy 9A. Flanged fittings and valves would usually be made of cast materials. Consideration was given to ratings of cast, nonferrous materials; however, at that time, suitable cast, nonferrous material specifications, and allowable stresses were not established (except for aluminum-bronze).

The 1960 Addendum also included ratings for two ferrous alloys: 304L and 316L. These ratings were obtained in accordance with the procedure given in Appendix D of ASA B16.5-1953 and -1957.

1961 Standard B16.5

The 1961 edition of ASA B16.5 made only a few editorial changes from the 1957 edition. The rating tables were expanded to include austenitic alloys 304L and 316L, formerly given as part of the 1960 Addendum to the 1957 issue. The nonferrous flange ratings continue to be shown as a 1960 Addendum (B16.5a) to ASA B16.5.

1968 Standard B16.5

The 1968 edition of B16.5 dropped the nonferrous flanged joint ratings inasmuch as they were inappropriate to a standard entitled "Steel Pipe Flanges . . .". The nonferrous ratings formerly in B16.5a-1960 are to be a part of a new standard, B16.31 entitled "Nonferrous Flanges". The 1968 edition permits minimum wall thicknesses less than 1/4 inch; in prior editions 1/4 inch was deemed to be the least feasible minimum wall thickness.

Summary of Historical Development

Flange dimensions and, in particular, their bolting dimensions established in some cases in the 1880's, have remained static for many years. This, of course, is the purpose of a standard; producing major economic advantages to both manufacturer and user of flanges, flanged fittings, and valves. For the manufacturer, standardization of dimensions permits full recovery of investment in patterns, dies, jigs, etc.; part of the savings obtained thereby are passed on to the consumer. For both manufacturer and user, standardization leads to large savings in engineering and design costs. Further advantages accrue to the user in the assurance of procuring matching parts for replacement or additions to existing plants.

Within this framework of established dimensions, the B16.5 Standard has shown remarkable progress in, on the one hand, greater diversity in coverage of types of flanges, sizes, and materials; and, on the other hand, in self-consistency in ratings and consistency with the ASME Boiler Code design concepts. The pressure-temperature ratings have been extended over a wider temperature range (atmospheric temperature to 750 F in 1927, subzero to 1500 F in 1968). Also, rating pressures have shown a general increase over the years, with attendant savings to users in that lower (and less expensive) classes can be used for some service conditions. For example, in 1927 the 300-lb class (raised face, carbon steel) was rated at 325 psi at 450 F; in 1961 this same class is rated at 650 psi at 450 F.

Several factors have perhaps contributed to increased rating pressures. First, years of experience with ASA flanges have increased confidence in their pressure capacity. Secondly, over the past 30 years the quality and reliability of flange, bolting, and gasket materials have improved. Finally, in critical piping systems, the piping system as a whole is designed with more care so that excessive forces imposed on the flanged joints by the piping are avoided.

CALCULATION OF STRESSES BY THE ASME METHOD

The ASME Boiler Code method of designing/analyzing a flanged joint may be considered as consisting of two steps.

(1) Determination of the required bolt area, A_m, by use of the equations:

$$A_m = \text{larger at } W_{m1}/S_b \text{ or } W_{m2}/S_a \quad (6)$$

$$W_{m1} = (\pi/4)\, G^2 P + 2\pi bGmP \quad (7)$$

$$W_{m2} = \pi bGy \quad (8)$$

where S_b = allowable bolt stress at design temperature

S_a = allowable bolt stress at atmospheric temperature

G = gasket diameter

b = gasket seating width

y = gasket seating load

m = gasket factor

P = design pressure.

(2) Analysis of the stresses in a flange with a given set of dimensions. The flange and its bolting are acceptable provided that all of the following limitations are met.

(a) $S_h \leq 1.5\, S_f$, (b) $S_t \leq S_f$, (c) $S_r \leq S_f$,

(d) $(S_h + S_t)/2 \leq S_f$, (e) $(S_h + S_r)/2 \leq S_f$.

These criteria must be met for:

(a) Moment M_{op} applied to flange, for which S_f = allowable stress for the flange material at design temperature

(b) Moment M_{gs} applied to flange, for which S_f = allowable stress for the flange material at atmospheric temperature.

The stresses S_h, S_r, and S_t are calculated in accordance with rules given in the Code. The method is that developed by Waters, et al[6]. The location of these stresses is shown in Figure 1. It is significant to note that the stresses so calculated are <u>not</u> due to internal pressure loading; they are due to tightening the bolts to the extent that the moment thereby imposed is equal to M_{op} or M_{gs}; these moments are defined in the Code.

Results of application of this analysis are shown in Table 3 for B16.5 welding neck flanges under the following conditions:

(1) The thickness, g_o, of the pipe welded to the flange is given by the equation

$$g_o = (P_p D/2S) + 0.05 \qquad (9)$$

where P_p = primary rating pressure

D = pipe outside diameter

S = 8570 psi.

(2) The tapered length of the hub was taken as equal to Y-C-1.5 g_o, where Y-C is the total hub length; Y and C are dimensions tabulated in B16.5.

(3) The gasket factors are m = 2.75, y = 3700. Gasket ~~outside diameter is equal to the pipe outside diameter.~~
(the raised face diameter; gasket inside diameter is equal to)

(4) Bolt material allowable stress is 25,000 psi at 100 F, 17,000 psi at the operating pressure.

(5) Design pressure is equal to the primary rating pressure, P_p.

Examination of Table 3 shows that B16.5 flanged joints have ample bolting, as judged by the ASME Code criteria.

For carbon steel flange materials, the primary rating temperature* (i.e., that temperature at which the rated pressure is P_p) is 850 F. Carbon steel flanges made to SA105 Gr I are so rated. The allowable stress given in the ASME Code for this material is 15,000 psi at 100 F (atmospheric), 8600 psi at 850 F. Those flanged joints made of SA105 Gr I material with S_{op} greater than 8600 psi or S_{gs} greater than 15,000 psi do not meet the ASME Code criteria; these are underlined in Table 3.

B16.5 flange pressure ratings at 100 F are 2.4 times the primary pressure*. As S_{op} is proportional to the pressure, it follows that if 2.4 S_{op} is greater than 15,000 psi, the B16.5 flanged joint does not meet the ASME Code criteria. This is equivalent to a stress S_{op} in Table 3 being greater than 15,000/2.4 = 6250 psi. It is apparent that there are additional B16.5 flanged joints which are rated higher at 100 F than would be permitted by the ASME Code design method.

Table 3, of course, covers only a sample of one particular type of flange with one particular gasket. B16.5 covers many types of flanges and gaskets. The following tabulation gives the ranges of calculated stresses at primary rating pressures for all sizes and classes of a few combinations of flange types and gaskets.

Flange Type	Facing	Gasket m-Factor	S_{op}, psi Min.	S_{op}, psi Max.
Welding neck	Small tongue and groove	5.5	2075	14,950
Welding neck	Ring type joint	5.5	2025	12,900
Slip-on	Small tongue and groove	5.5	3400	16,950
Slip-on	Ring type joint	5.5	1900	14,800
Blind	Small tongue and groove	5.5	2600	16,190

Obviously, there is a large variation of stresses in B16.5 flanged joints as calculated by the ASME Code method. However, a rough average of the controlling stresses in all of the combinations of flange types is 8750 psi. As was discussed in the preceding section, 8750 psi is the "representative" flange stress used in extrapolation of established ratings for carbon and carbon-moly flange materials to obtain ratings for

* Except for the 150 lb class.

other flange materials. Its primary justification, of course, lies in its relationship to ratings backed up by significant prior field experience; nevertheless, the "representative" stress is approximately the average S_{op} calculated by the ASME Code method.

It is not to be inferred that the stresses cited in the preceding are necessarily the highest stresses that may be calculated for B16.5 flanged joints. As will be discussed in the next section, the initial bolt stress applied to B16.5 flanged joints is typically about 40,000 psi. The values of S_{40} shown in Table 3 are the calculated controlling stresses due to an initial bolt stress of 40,000 psi. The calculated stresses shown in Table 3 are not the highest that can be calculated as existing in B16.5 flanged joints. The following example illustrates this point and, more important, serves to illustrate the significance of the calculated stresses.

The highest value of S_{40} shown in Table 3 is for the 24 inch 600-lb class. The wall thickness of the attached pipe (g_o), was based on the equation: $g_o = (PD_o/2S) + 0.05$, where S = 8750 psi; for this size and class the value of g_o is 0.873 inch. This particular size and class is frequently used in gas transmission pipelines; however, in such applications, the pipe would normally be thinner, e.g., g_o = 0.438 inch. The calculated stresses due to an initial bolt stress of 40,000 psi in the 24 inch 600-lb class with g_o = 0.873 inch or g_o = 0.438 inch are tabulated below.

g_o, in.	S_{ho}, psi	S_{hl}, psi	S_r, psi	S_t, psi	Controlling Stress
0.873	75,500	26,200	28,600	30,500	53,000
0.438	115,000	23,500	21,400	46,200	80,600

It is apparent that a significant increase in maximum stress, as well as in the ASME controlling stress, occurs when the attached pipe wall thickness is decreased from 0.873 inch to 0.438 inch.

The question arises: Can the bolts in a 24 inch 600-lb welding neck flange, with attached pipe wall of 0.438 inch, be tightened to 40,000 psi stress. Despite the calculated stress of 115,000 psi, the answer is yes. The author has supervised the installatation of such flanged joints; no difficulty was encountered in applying the bolt stress and, insofar as the author is aware, these flanged joints are performing satisfactorily some twelve years after installation. The reason for the satisfactory performance of such flanges lies in the distribution of the stresses. Firstly, S_{ho}, S_{hl}, S_r, and S_t are bending stresses, the average stress through the hub wall or ring thickness is essentially zero. Secondly, S_{ho} is also a local stress in the axial direction. Accordingly, while local zones of yielding may occur when such joints are tightened to 40,000 psi bolt stress, there is no gross deformation of the flanges. The next section of this paper will further discuss the significance of local yielding and gross plastic deformation on the pressure capacity of a flanged joint.

In summary of the calculation of stresses by the ASME Code method, it is apparent that many B16.5 flanges do not meet the Code stress criteria at their rated pressures. However, many of those flanges have been used in flanged joints at or near their rated pressures for a number of years. Accordingly, it does not appear that meeting the ASME Code criteria is necessary to obtain a serviceable flanged joint.

Attention is called to the results shown in Table 3 for the 3 inch and 8 inch 150-lb class flanged joints. It may be noted that both of these meet the ASME Code criteria. As many readers are probably aware, these two sizes of the 150-lb class have a rather long history of being difficult to keep tight in the field. On this basis, it may be said that the ASME Code criteria are neither necessary nor, in the case of pipeline-flanged joints, sufficient. In the next section, in which a more complete analysis of flanged joints and their loadings is given, some reasons for the relative weakness of 3 inch and 8 inch 150-lb class joints will be discussed.

LOAD CAPACITIES OF B16.5 FLANGED JOINTS

The ASME Code method provides generally adequate, albeit somewhat arbitrary, guidance for checking the adequacy of a flanged joint for pressure loading. However, flanged joints in pipelines may also be subjected to significant loadings imposed on the joint by the attached pipe (called external moments herein). In addition, flanges in pipelines are more likely to be subjected to severe thermal gradients than flanged joints in pressure vessels. In the following, we will describe a more fundamental approach to analyzing the behavior of flanged joints than is used by the ASME Code and give results of the application of this method to a sampling of B16.5 welding neck flanges.

Let us consider the relatively simple case of initial bolt loading followed by internal pressure loading. We would like to establish the relationship that exists between the initial bolt load and the leakage pressure of a flanged joint.

Figure 2 represents a typical, although somewhat idealized, set of test results on a flanged joint with a flat asbestos gasket*. In this test, the bolts are tightened to some low stress level and the internal pressure is increased until leakage is observed, this pressure being the leakage pressure, P_L. The internal pressure is then dropped to zero, the bolts are further tightened to a higher stress level, the leakage pressure again determined, and the process is repeated until a curve, as shown in Figure 2, is obtained.

In the initial stages of the test, Figure 2 indicates that the leakage pressure is essentially zero. The bolt load is not sufficient to "seat" the gasket. This part of the flanged joint performance is at least approximately represented by the ASME Code term W_{m2} (equation (8) herein), although the actual value of the seating load depends at least as much on the planeness of the flange faces and the amount of nicks or scratches on the faces as it does on the gasket characteristics.

As the test proceeds, as indicated in Figure 2, we eventually reach a sufficient bolt load to "seat" the gasket. We now find that the leakage pressure increases essentially in proportion to the initial bolt load. It is necessary to define what is meant by leakage pressure. In tests such as described above, using water as a pressurizing fluid,

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* Test data on the type discussed herein and shown in Figure 2 is given by George, Rodabaugh and Holt[7]. This paper gives the results of tests on 8" 150-lb and 12" 300-lb classes, with flange materials of A181 Gr I and 6061-T6 aluminum alloy.

after the gasket is seated it is observed that below a certain pressure, no water emerges from the flange joint, although, if one held the test conditions for an hour or so, one or two drops of water might emerge. However, as the pressure is increased further, one reaches a pressure at which the leakage, rather than being a slow diffusion, becomes profuse. The action of the flanged joint in this respect is analogous to that of a spring-loaded relief valve. It is not to be inferred that no leakage occurs below the leakage pressure as this obviously depends on the contained fluid and the means of leak detection; e.g., helium, with leakage detected by a mass spectrometer.

It will be noted that on Figure 2 there is a straight line labeled $P_L = W_1/A_p$, where P_L = leakage pressure, W_1 = initial bolt load, and A_p = pressure to the outside of the gasket. It has been shown by Roberts[8] that the theoretical leakage pressure for a flanged joint with a flat gasket is the pressure at which the pressure times the area to the outside of the gasket is equal to the bolt load. Figure 2 shows that the straight line portion of the test results lie along a line that is lower than the line $P_L = W_1/A_p$ which indicates that the bolt load at pressure, W_2, is less than the initial bolt load, W_1. This is a typical result for B16.5 flanged joints. In terms of the ASME Code method, the spread between these two straight lines is at least crudely represented by the term $2\pi bGmP$ in the calculation of W_{m1}. Actually, however, the value of W_2 can be calculated with reasonable engineering accuracy and, for typical B16.5 flanges with typical flat gaskets, it is more dependent upon the elastic characteristics of the flanges and bolts than it is upon the gasket.

The third portion of the P_L versus W_1 curve in Figure 2 shows the relationship becoming nonlinear and dropping below the straight line labeled $P_L = W_2/A_p$. We now have reached the stage where the initial bolt load is sufficient to cause local yielding at some locations in the flanges. Plastic stress redistribution and strain hardening are such that the bolts can be tightened further and a stable condition exists. However, the addition of the pressure load causes the stress to increase slightly but since the stress pattern is already at yield conditions, further yielding takes place and the consequent reduction in bolt load is greater than calcualted on a purely elastic basis. It should be noted that the amount of yielding of the flange need only be very small in order to produce a significant reduction in bolt load. It is also significant to note that, for a given pressure level, this is a self-limiting process, i.e., the bolt load will reduce to that level which the flanges are capable of carrying and no further.

Figure 2 also shows a set of data obtained by reducing the pressure and bolt load to zero and then repeating the test. Typically, the initial stage of gasket seating will not be observed and, in the final stage, because of plastic stress redistribution and strain hardening in the flanges in the first test, the straight line portion of the curve will extend higher.

The limit to a test as described above arises either from the "limit load" of the flange or yielding of the bolts. In the first case, the person tightening the bolts will notice that he must rotate the nuts more to obtain a desired increment in bolt load and he will have trouble getting all the bolts up to the same load level. If he keeps going, he will eventually bend the flanges until the outer edges touch. In the second case, if strain gages are used on the bolts as is almost essential in these tests, he will note a very large increment of strain as he tightens the bolts. The limit of yielding of the bolting is easy to calculate.* The limit load or "plastic collapse load" of flanges, however, is a much more difficult problem and, to the author's knowledge, no attempt has been made to establish such limits for B16.5 flanges. It should be noted that the bolt load that produces a maximum stress in the flange equal to the flange material yield strength is not the plastic collapse bolt load; that load may be three or four times as high, as indicated by the previous discussion of the 24 inch 600-lb welding neck flanged joint with 0.438 inch wall pipe.

From the preceding discussion, it is apparent that the bolt stress required to balance the pressure load is a significant aspect of B16.5 flanged joints. Table 4 shows those bolt stresses for all B16.5 flanged joints when the pressure is equal to the primary rating pressure. The bolt stresses at the 100 F rating pressure are 2.4 times those shown and at a test pressure of 1.5 times the 100 F rating pressure are 3.6 times those shown. This indicates that initial bolt stresses of 3.6 x 7200 = 25,920 psi would be sufficient for most sizes and classes and sufficient for the 16 inch 900 lb class provided that W_2 were equal to W_1.

To calculate the value of W_2, it is necessary to consider the joint as a whole, including the elastic characteristics of the flanges, bolts, and gaskets. The joint is a statically indeterminate structure and it is necessary to match displacements. The details of the analysis are too lengthy to include in this paper; the interested reader should refer to the paper by Wesstrom and Bergh[9] and the discussion by Rodabaugh thereof. In the following, we will give some examples of the results of the application of the theory (with some additions to include thermal gradients) to typical B16.5 flanged joints. It is perhaps pertinent to note that up to a few years ago, the detailed analysis of B16.5 flanged joints would be

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* In calculating yielding of bolts, the effect of the torsional stress must be included.

prohibitively expensive. However, with the use of a digital computer and an appropriate computer program, it takes roughly one-half second of computer time* to carry out the analysis of a flanged joint.

Table 5 shows values of $W_1 - W_2$ for a sample of B16.5 welding neck flange joints. W_1 is the initial bolt load, W_2 is the subsequent bolt load at the following conditions.

(1) Internal pressure equal to the primary rating pressure, P_p, psi

(2) Initial condition modulus of elasticity of all parts = 30,000,000 psi, pressurized condition modulus of elasticity of all parts = 23,000,000 psi (corresponding to assumed temperature increase from atmosphere to 850 F)

(3) Bolts 50 F hotter than flanges and gasket

(4) Pipe and flange hub average temperature 100 F hotter than the average temperature of the flange ring

(5) Coefficient of thermal expansion of all parts = 0.000006/F

(6) External moment that produces a nominal bending stress, S_{em}, in the attached pipe of 8750 psi

The magnitude of the loads listed above are more-or-less representative of loadings that are applied to B16.5 flanged joints in steam piping, although (3) and (4) would seldom, if ever, occur at the same time.

The change in bolt load, $W_1 - W_2$, does not depend upon W_1 as long as W_2 is greater that the critical bolt, W_c. The purpose of Table 5 is to illustrate what initial bolt loads must be applied so the subsequently

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* These computations were run on a CDC 6400 computer using a program developed by the author entitled FLANGE. The input consists of flange, bolt and gasket dimensions and material properties (modulus of elasticity, coefficient of thermal expansion, uses Poisson's ratio of 0.3). The program computes (in one of several options) F, V and f factors (ASME Code factors) and stresses corresponding to ASME Code stresses plus others and the change in bolt loads and moments as a function of the input loads (pressure, thermal gradients, external moments).

applied loads, thermal gradients, and change in modulus of elasticity do not reduce the bolt load W_2 below W_c. If W_2 becomes less than W_c, leakage will occur with a flat gasket or the faces will separate with an elastomeric O-ring gasket.

The column headed W_c is the critical bolt load calculated by the equation:

$$W_c = (\pi/4)\, G_o^{\,2}\, P_p + \pi D^2 g_o\, S_{em}/C \qquad (10)$$

where G_o = gasket outside diameter (raised-face diameter in these calculations)

P_p = primary rating pressure

D = pipe diameter (outside diameter used in calculations)

g_o = pipe wall thickness

S_{em} = nominal bending stress produced in the attached pipe by the external moment

C = bolt circle diameter

The minimum value of W_1, to prevent W_2 from becoming less than W_c, is equal to $W_1 - W_2 + W_c$; this is shown in Table 5 in the column headed $(W_1)_{min}$. The corresponding minimum initial bolt stresses are shown in the column headed $(S_{b1})_{min}$. The next column, S_{ASME}, shows bolt stresses at the bolt loads calculated by the ASME Code rules; it can be seen that S_{ASME} ranges from about one-half to one-fifth of the load indicated by the analysis.*

In field installations of B16.5 flanged joints the initial bolt stress is seldom controlled; the pipe fitter simply tightens the bolts to what he considers to be an appropriate amount. Petrie[5] indicates that this initial bolt stress (psi) is approximately given by the equation:

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* However the ASME Code, Appendix S, does recognize that initial bolt stresses may be and perhaps should be higher than the allowable bolt stresses.

$$(S_{bl})_p = \frac{45,000}{\sqrt{d}} \qquad (11)$$

where d = bolt diameter, inches

The bolt stress $(S_{bl})_p$ is shown in the last column of Table 5. For most joints, $(S_{bl})_p$ is sufficient as judged by comparison with $(W_1)_{min}$.

The flange stresses are also quite different under operating conditions than under initial bolt-up conditions. The column headed S_{load} in Table 3 gives the calculated flange stresses corresponding to that combination of conditions used for calculating W_2. These stresses are the sum of the individual stresses due to (a) moment loads, (b) pressure and (c) thermal gradients. The stresses are shown for the stress combination (last column of Table 3) which controlled, by ASME rules, for initial bolt loading although the inclusion of stresses due to pressure and thermal gradients in some cases changes the controlling stress combination. It will be noted that S_{load} is significantly lower than S_{40}. This is because the reduction in bolt load from W_1 to W_2 reduces stresses more than the additional stresses due to pressure and thermal gradients.

It should be noted that Table 5 and S_{load} of Table 3 are examples for the specific set of conditions previously listed. The actual conditions that will exist depend not only upon the magnitudes of loads but also upon their time sequence. For example, in a pipeline during heat-up the bolts might be 50 F cooler than the flange rings; which would produce an increase in bolt load ($W_2 > W_1$) and an increase in flange stress. The analysis method can give an engineering evaluation of the effects of these loads and conditions in varying time sequences, as well as effects of using flanged joints made up of different materials (e.g., aluminum flange to stainless steel flange); obviously the various possible combinations are too numerous to be covered herein. Table 5 serves its primary purpose in showing that, by-and-large, an initial bolt stress of about 40,000 psi in B16.5 flanges is often necessary and generally sufficient for B16.5 flanged joints with the pressure ratings given therefor.

Figure 3 shows the external moment capacity of B16.5 flanged joints in the form of S_{em} (stress in attached pipe due to external moment) plotted against nominal size. This relationship was calculated using equation (10) with g_o taken as the larger of standard weight pipe wall thickness or g_o as calculated by equation (9), and with W_c set equal to $\sigma_b A_b$. The value of σ_b now represents the bolt stress at operating conditions (loads of pressure and moment) and S_{em} is the corresponding calculated maximum bending stress which can be imposed on the joint as limited by leakage at the joint. Figure 3 is based on P equal to the 100 F rating pressure and σ_b = 40,000 psi.

The assumption that g_o is not less that standard weight is particularly significant in that 150-lb flanged joints are seldom used with pipe thinner than standard weight. Such pipe is much thicker than required for the pressure. In general, the moments in a piping system are proportional to the pipe moment of inertia; accordingly, the lower pressure classes are likely to be subjected to higher moments in proportion to their pressure rating.

The two horizontal dashed lines in Figure 3, labeled S_A, represent magnitudes of pipe bending stress permitted by USAS B31.1.0-1967 (Par. 102.3.2) for ASTM A106 Grade B pipe at temperatures up to 650 F. The upper line is based on the assumption that the longitudinal stresses due to pressure, weight, and other sustained loads are negligible and the number of cycles is less than 7000, in which case S_A = 1.0 (1.25 x 15,000 + 1.25 x 15,000) = 37,500 psi. The lower line is based on the assumption that the sum of the longitudinal stresses due to pressure, weight and other sustained loads is equal to S_{ha}, in which case S_A = 1.0 (1.25 x 15,000 + 0.25 x 15,000) = 22,500 psi. Where the calculated moment capacity of the joint is below these lines, the implication is that such joints cannot withstand bending monents otherwise permissible in straight pipe portions of the piping system.

In the preceding discussion some engineering evaluations have been presented, which were based on linear elastic theory plus some elementary equilibrium assumptions. However, B16.5 primary rating temperatures are sufficiently high so that the loaded flanges and bolting surely undergo significant creep (or, more precisely, relaxation). While a reasonably valid creep/relaxation analysis of a flanged joint is within the state-of-the-art, the analysis is quite expensive and, insofar as the author is aware, no such analyses have been made on anything as prosaic as a B16.5 flanged joint. However, it is informative to consider conditions in which it is assumed that relaxation has proceeded to the extent that the bolt stress is 15,000 psi. This stress is roughly the ASME Code allowable stress for SA193 Grade B7 bolts at temperatures corresponding to the primary rating temperatures of carbon and low alloy steel flange materials. Figure 4 shows moment capacity for the residual bolt stress of 15,000 psi and with the pressure equal to the primary rating pressure.

As in Figure 3, ANSI B31.1 piping code allowable bending stresses are shown; in Figure 4 for a temperature of 850 F. It is apparent in Figure 4, even more so than in Figure 3, that some B16.5 flanged joints are unable to withstand bending moments otherwise permissible in the straight portions of the piping system.

Both Figures 3 and 4 show that the 150-lb class of flanged joints is relatively weak with respect to moments imposable by standard weight pipe. The 8 inch size is represented by the lowest point in Figures 3 and 4. The 3 inch size is the weakest of sizes 5 inch and smaller and probably little attention is given to providing adequate flexibility in most 3 inch piping systems using 150-lb flanged joints. Another, and possibly equally important aspect of 3 and 8 inch 150-lb flanges is that the bolt spacing is relatively large. This aspect is not covered by the ASME Code method or the more complete analysis used herein; in both approaches it is assumed that the desired loading at the bolts can be approximated by a line load along the bolt circle and that the effect of the bolt holes has negligible effect on the flange strength.

It is appropriate now to take a broad look at the problem of pressure-temperature ratings of B16.5 flanged joints. B16.5 covers a large range of sizes, classes, types of flanges, flange materials, bolting materials, and gaskets. Any one of these flanged joints poses a complex analytical problem because of the basically statically redundant nature of a flanged joint and the several types of significant loadings which may be imposed on the joint in service. The rating temperatures extend into the creep range of the materials.

The analysis methods discussed herein, based entirely on elastic theory, give at least a rough engineering evaluation of B16.5 flanged joints. This evaluation indicates that the pressure capacity in a given pressure class decreases with increasing size. The external moment capacity, in relationship to the external moments imposable by the pipe with which the flanges are normally used, generally decreases with increasing size (with notable exceptions of the 3" and 8" - 150 lb) and generally increases with increasing pressure class.

While the above variations indicate that B16.5 flanges are not consistently rated, there appear to be good economic reasons why the smaller sizes should be made relatively stronger than the larger sizes. First, small (e.g., below six-inch sizes) pipelines are seldom "engineered". On the other hand, the large sizes are relatively more likely to be checked, at least to the extent of determining external moments as required by ANSI B31.1.0. Perhaps more important, the moment that may be imposed on large sizes is likely to be limited by the strength and/or flexibility of anchors and/or connected equipment. Accordingly, it seems desirable that the smaller sizes should have relatively higher capacities; thereby, achieving a tradeoff between the cost of the flanged joints and engineering costs.

SUMMARY

(1) Pressure-temperature ratings given in B16.5 have been, and continue to be established, extrapolated and/or rationalized on the basis of the strength of flanged joints.

(2) The historical development shows that flange and bolting dimensions have remained static for many years. This is the primary purpose of a standard; producing major economic advantages to both manufacturers and users. Within this framework of established dimensions, the B16.5 Standard has diversified in coverage of types of flanges, sizes and materials. The pressure-temperature ratings have been extended over a wide range of materials, and rating pressures have been increased in the years from 1927 to 1968.

(3) B16.5 flanged joints do not necessarily meet the criteria given in the ASME Boiler Code. Experience and a more detailed analysis indicate that it is not necessary to meet the ASME Code rules in order to have a satisfactory flanged joint and, on the other hand, meeting the ASME Code rules does not necessarily assure a good flanged joint for use in a pipeline.

(4) Engineering evaluations, based on elastic analysis, indicate that the capacity of B16.5 flanged joints decreases as the size increases. However, from the standpoint of "tradeoff" between flanged joint costs and engineering costs, this aspect appears to be desirable.

Several questions which come up rather often in connection with B16.5 ratings were listed in the Introduction to this paper. Answers to the questions are given in the following. Some of the answers are incomplete and controversial, and should be understood as those of the author, with no official status insofar as the B16.5 Standard is concerned.

(1) Why are B16.5 ratings not proportional to allowable stresses in the ASME Boiler Code?

The question is based on a partially incorrect premise in that ratings at and above the primary rating temperature are proportional to the Code allowable stresses, see the discussion of the 1953 edition of B16.5. However, the ratings from room temperature up to the rating temperature are not proportional to Code allowable stresses. If they were, then, for example, the pressure rating of carbon steel flanges would be constant up to 650 F because the allowable stress for carbon steel is constant up to 650 F. The Code allowable stress is based on one quarter of the ultimate strength at temperature.

The discussion in this paper was intended to bring out the significance of elastic material properties, the material yield strength, the change in modulus with temperature (and creep/relaxation properties at high temperatures). Further, external moment loads are likely to increase as temperature increases. These factors indicate that ratings of flanged joints should decrease in the range of temperature from 100 F to 650 F; B16.5 ratings have done so ever since their inception in 1927. At present, the ratings between 100 F and 650 F decrease roughly in proportion to the decrease in yield strength of carbon steel.

The B16.5 rating method has been criticized as ignoring the tensile strength of the flange material in establishing ratings. That it does so is based on the premise that flanged joints (B16.5 ratings are flanged joint ratings) made of reasonably ductile materials fail by leakage, not rupture. Accordingly, the significant material property is yield strength rather than ultimate strength, along with the analogous creep strength at high temperatures. In the author's opinion, the Boiler Code procedure (potentially giving the same flanged joint rating from 100 F to 650 F) is logically incorrect, while the B16.5 ratings are logically defensible.

(2) Why is the 150-lb class rated differently than all of the other pressure classes?

From an analytical standpoint, Figures 3 and 4 herein give an indication of the relatively low external moment capacities of the 150-lb class. An increase in rated pressures would tend to aggravate this situation. Also, the bolt loading available to seat some types of gaskets is marginal. In addition, the relatively short face-to-face dimensions of 150-lb valves has led to the use of obround bonnets; these also pose problems in up-rating the 150-lb class.

(3) Why do field problems with leakage of 3" and 8" 150-lb class flanged joints occur?

It should be remarked first that most 3" and 8" 150-lb flanges are satisfactory in service. Figures 3 and 4 herein give an indication of the low external moment capacity of such joints. Also, the bolt load available for gasket seating is particularly low in these two sizes. Finally, the relatively large bolt spacing probably contributes to the problem.

(4) Why are B1.6.5 ratings acceptable under the ASME Boiler Code in those cases where it can be shown that they do not meet the rules given in the "Rules for Bolted Flanged Connections"?

The primary answer is based on the generally favorable service experience with B16.5 flanged joints and their ratings. The discussion in this paper was intended to illustrate that flanged joints that fail to meet the ASME Code criteria by a wide margin nevertheless give adequate service and, on the other hand, flanged joints that met the ASME Code criteria were not necessarily good pipeline flanged joints.

(5) What is an appropriate initial bolt stress for B16.5 flanged joints?

Experience and theory indicate that initial bolt stresses of about 40,000 psi is usually necessary and adequate.

(6) What is the effect of modulus of elasticity of flanges, bolts and gaskets on the performance of a flanged joint?

The change in modulus of elasticity with temperature is significant in that the bolt load changes in proportion to the modulus; a decrease in modulus of 20 percent would mean that the initial applied bolt stress would decrease by 20 percent. However, if two dimensionally identical flanged joints were compared, one made of aluminum (E = 10,000,000) and the other of steel (E = 30,000,000), the elastic theory indicates that the performance of the joints would be the same. No generalization can be made about flanged joints made of combinations of materials (e.g., aluminum flange mated to steel flange); however, the analysis method discussed herein would give an engineering evaluation of the characteristics of such flanged joints.

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TABLE 1. SCOPE OF ANSI B16.5

<u>Product Scope</u>

(a) Companion flanges: welding neck, lapped, slip-on-welding, socket-welding, screwed and blind

(b) Flanged fittings: elbows, tee, cross, lateral, reducers, true-Y's

(c) Valves

<u>Pressure Classes</u>

150, 300, 400, 600, 900, 1500, and 2500 lb

<u>Sizes</u>

1/2 through 24" nominal sizes

<u>Dimensional Coverage</u>

(a) Flanges: relatively complete dimensions specified

(b) Bolting: complete dimensional coverage

(c) Flange fitting bodies and valve bodies: center-to-face, face-to-face, and minimum body wall thickness only

<u>Materials Coverage</u>

(a) Flanges, flanged fittings and valves: carbon steel, 10 ferritic alloys, and 6 austenitic alloy steels

(b) Bolting: 4 ASTM specifications for bolts, 2 ASTM specifications for nuts

(c) Gaskets: variety of materials and types, see Appendix E of B16.5

TABLE 2. COMPARISONS OF MINIMUM WALL THICKNESS REQUIREMENTS FOR B16.5 FLANGED FITTINGS AND VALVE BODIES AND API 600 VALVE BODIES

Class	Size	d (1)	t (2)	Tabulated Min. Wall B16.5 t_1	Tabulated Min. Wall API 600 t_2	$\frac{t_1}{t}$	$\frac{t_2}{t}$
150	4	4.000	0.065	0.250	0.438	3.85	6.74
	8	8.000	0.130	0.312	0.500	2.40	3.85
	16	15.250	0.248	0.438	0.688	1.77	2.77
	24	23.250	0.379	0.562	0.812	1.48	2.14
300	4	4.000	0.132	0.312	0.500	2.36	3.79
	8	8.000	0.264	0.438	0.688	1.66	2.61
	16	15.250	0.503	0.688	0.937	1.37	1.86
	24	23.000	0.759	0.938	1.187	1.24	1.56
600	4	4.000	0.271	0.375	0.625	1.38	2.31
	8	7.875	0.534	0.625	1.000	1.17	1.87
	16	14.750	1.000	1.094	1.500	1.09	1.50
	24	22.000	1.491	1.594	2.000	1.07	1.34
1500	4	3.625	0.668	0.750	1.125	1.12	1.68
	8	7.000	1.291	1.406	1.875	1.09	1.45
	16	13.000	2.397	2.500	3.125	1.04	1.30
	24	19.625	3.619	3.719	4.500	1.03	1.24

(1) d = inside diameter of fitting or port opening of valve as given in B16.5 Tables

(2) t = wall thickness calculated by the equation:

$$t = 1.5\left[\frac{P_p d}{14{,}000 - 1.2\,P_p}\right]$$

P_p = primary rating pressure

TABLE 3. BOLT AREAS AND CALCULATED STRESSES, B16.5 WELDING NECK FLANGES

Class	Size	A_m, sq.in. (1)	A_b, sq.in. (2)	S_{op}, psi (3)(8)	S_{gs}, psi (4)(8)	S_{40}, psi (5)	S_{load}, psi (6)	C.S.C. (7)
150	3	0.625*	0.808	2,780	7,600	14,900	8,300	HI, T
	4	0.836*	1.616	3,860	12,420	28,790	13,850	HI, R
	8	1.620*	2.416	4,430	8,390	19,100	6,750	HI, R
	16	3.253	8.816	5,820	13,580	35,800	9,740	R
	24	6.618*	18.58	5,520	11,500	33,200	16,030	HI, R
300	4	0.973	2.416	3,800	8,770	21,600	10,010	HI, R
	8	2.431	5.028	5,010	9,970	23,550	9,990	HI, R
	16	6.476	18.58	5,900	<u>15,300</u>	39,800	24,420	HO, R
	24	13.235	33.72	<u>9,420</u>	<u>21,400</u>	55,500	35,540	HO, R
600	4	1.946	3.352	6,280	11,550	25,000	14,720	HI, R
	8	4.862	8.736	5,760	10,700	23,700	17,240	HI, R
	16	12.959	28.10	7,530	<u>16,110</u>	37,800	23,480	HO, R
	24	26.47	55.30	<u>11,160</u>	<u>22,000</u>	53,000	34,480	HO, T
1500	4	4.865	7.432	<u>8,570</u>	<u>15,200</u>	30,800	17,140	HI, R
	8	12.159	20.16	6,860	12,150	25,600	13,330	HI, R
	16	32.39	68.67	7,190	<u>15,300</u>	34,700	16,800	R
	24	66.18	140.0	6,630	13,450	30,800	36,400	HI, R

(1) ASME Code minimum required bolt area, A_m (see equation (6)). S_b = 17,000 psi, S_a = 25,000 psi. W_{m1}/S_b is the larger except for those values with an asterisk.

(2) Total bolt root area of B16.5 flanged joint.

(3) Stress at operating condition moment, M_{op}.

(4) Stress at gasket seating condition moment, M_{gs}.

(5) Stress due to an initial bolt stress of 40,000 psi.

(6) Stress at loaded conditions described in text under "Load Capacities of B16.5 Flanged Joints".

(7) C.S.C. = controlling stress combination: HI, T = $(\sigma_{hi} + \sigma_t)/2$; HO, R = $(\sigma_{ho} + \sigma_r)/2$; R = σ_r; HO, T = $(\sigma_{ho} + \sigma_t)/2$; σ_{hi}, etc., are stresses calculated by the ASME Code method; their locations are shown in Figure 1.

(8) Underlined stresses are higher than allowed by the ASME Boiler Code for SA193 Grade B material.

TABLE 4. B16.5 FLANGE BOLT STRESSES BY THE EQUATION $\sigma_b = (\pi/4)G_o^2P_p/A_b$

Nom. Size	σ_b = 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	930	--	1,660	--	2,780	4,830
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	4,350	5,150	6,220	6,930	6,200
12	5,270	4,550	4,760	5,710	6,590	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	5,390	4,930	6,180	7,050	6,160	--
24	4,710	5,190	4,910	6,330	6,110	6,250	--

P_p = primary rating pressure, psi.

A_b = total bolt area, sq-in.

G_o = outside diameter of raised face, inches.

TABLE 5. CALCULATED BOLT STRESSES IN B16.5 WELDING NECK FLANGED JOINTS

Class	Size	$W_1 - W_2$,[1] lb	W_c,[2] lb	$(W_1)_{min}$,[3] lb	$(S_{b1})_{min}$,[4] psi	S_{ASME},[5] psi	$(S_{b1})_p$,[6] psi
150	3	17,100	7,400	24,500	30,300	19,300	56,900
	4	34,200	11,100	45,300	28,000	12,900	56,900
	8	69,900	34,900	104,800	43,400	16,900	52,000
	16	211,100	102,300	313,400	35,500	9,200	45,000
	24	475,800	224,700	700,500	37,700	7,400	40,300
300	4	50,100	18,000	68,100	28,200	8,600	52,000
	8	118,200	57,700	175,900	35,000	8,200	48,100
	16	394,100	182,000	576,100	31,000	5,900	40,300
	24	740,600	403,300	1,143,900	33,900	6,700	36,700
600	4	66,300	31,400	97,700	29,100	9,900	48,100
	8	189,700	104,600	294,300	33,700	9,500	42,400
	16	586,700	338,600	925,300	32,900	7,800	36,700
	24	1,219,000	768,700	1,987,700	35,900	8,100	32,900
1,500	4	129,200	70,600	199,800	26,900	11,100	40,300
	8	378,700	237,100	615,800	30,500	10,200	35,300
	16	1,264,900	763,700	2,028,600	29,500	8,000	28,500
	24	2,660,800	1,730,300	4,391,100	31,400	8,000	24,100

(1) Change in bolt load from initial conditions (W_1) to conditions described in text (W_2).

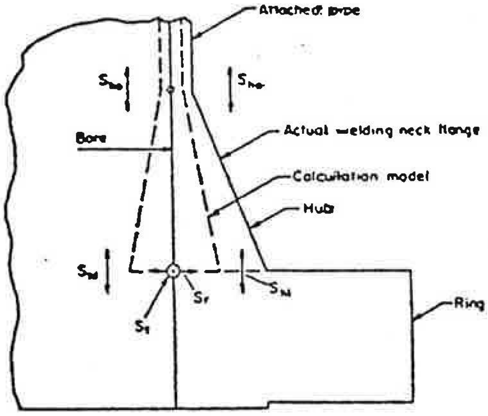
(2) W_c = critical bolt load, equation (10), S_{em} = 8,750 psi.

(3) $(W_1)_{min}$ = initial bolt load required to maintain $W_2 > W_c$.

(4) $(S_{b1})_{min} = (W_1)_{min}/A_b$.

(5) S_{ASME} = larger of W_{m1}/A_b or W_{m2}/A_b, see equations (7) and (8).

(6) $(S_{b1})p$ = typical initial bolt stresses applied by pipe filler in making up B16.5 flanged joints, see equation (11).



S_{ho} and S_{hi} are bending stresses in the hub, the membrane stress is zero
S_t and S_r are almost entirely bending stresses in the flange ring

FIGURE 1. LOCATION AND DIRECTION OF STRESSES CALCULATED BY THE ASME CODE METHOD

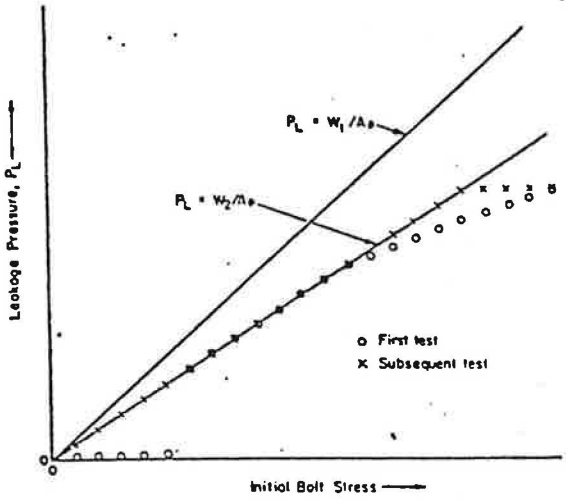


FIGURE 2. IDEALIZED TEST DATA FROM AN INITIAL BOLT LOAD - PRESSURE TEST OF A B16.5 FLANGED JOINT WITH A FLAT GASKET

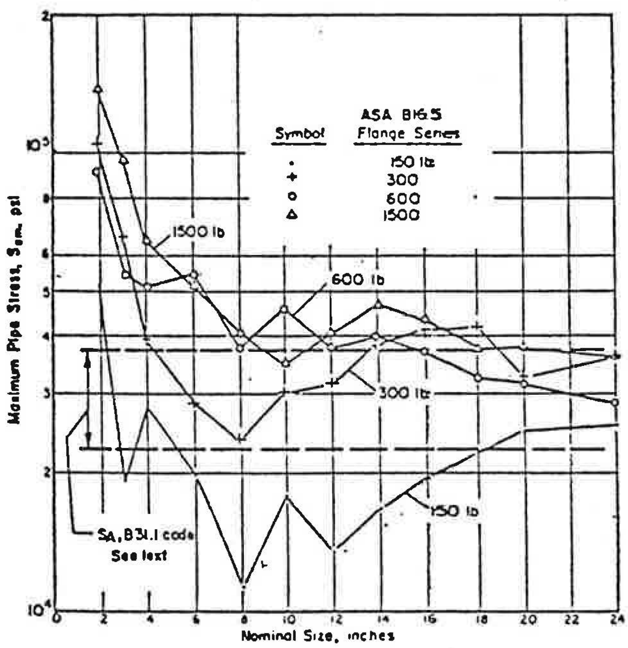


FIGURE 3. MAXIMUM PIPE BENDING STRESSES AS LIMITED BY LEAKAGE OF ANSI B16.5 FLANGED JOINTS, $P = 2.4P_p$, $\sigma_b = 40{,}000$ psi

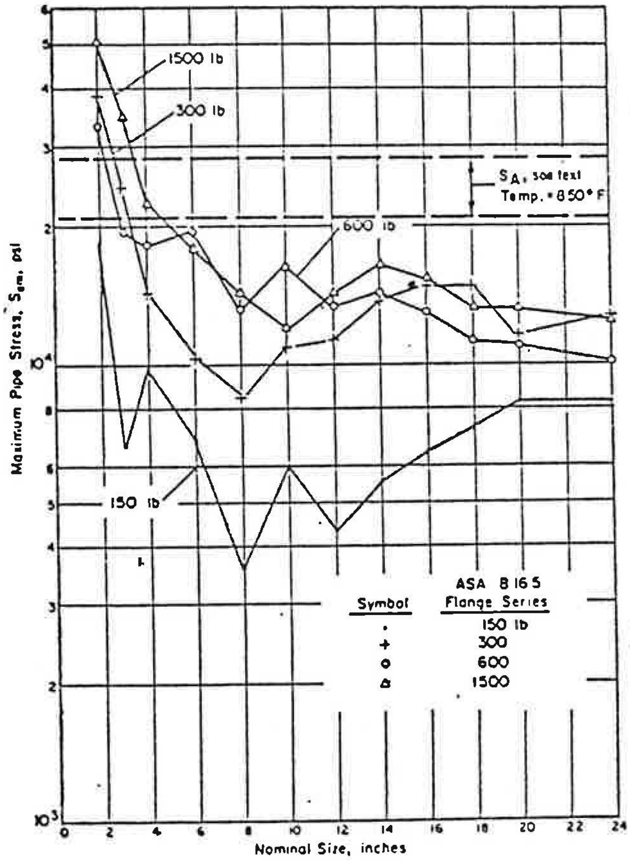


FIGURE 4. MAXIMUM PIPE BENDING STRESSES AS LIMITED BY LEAKAGE OF ANSI B16.5 FLANGED JOINTS, $P = P_p$, $\sigma_b = 15{,}000$ psi