Analysis of Bolting in Flanged Connections

by

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NOMENCLATURE

Symbol		Definition	Unit
A_b	=	cross-sectional area of the bolts using the root diameter of the thread or least diameter to the bottom of the slots	in^2
A_m	=	total required cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2}	in^2
A_{m1}	=	total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions	in^2
	=	W_{m1}/S_b	
A_{m2}	=	total cross-sectional area of bolts at root of thread or section of least diameter under stress, required or gasket sealing	in^2
b	=	Effective gasket or joint-contact-surface seating width	in
b_0	=	Basic gasket seating width	in
D	=	Major diameter of bolt	in
d_r	=	Root diameter of bolt thread	in
F_p	=	Bolt pre-load	lb_f
G	=	Diameter of location of gasket load reaction. Defined as follows: When $b_0 \leq 1/4$ in. (6mm), G = mean diameter of gasket contact face. When $b_0 < 1/4$ in. (6mm), G = outside diameter of gasket contact face less 2*b*.	in
H	=	Total hydrostatic end force	lb_f
	=	$0.785G^2P$	
H_P	=	Total joint-contact surface compression load 2b x 3.14 G*m*P	lb_f
K_b	=	Nominal bending stress concentration factor	-
K_d	=	Direct stress concentration factor	-
K_f	=	Fillet stress concentration factor	-
K_t	=	Transverse shear stress concentration factor	-
m	=	Gasket factor	-
n_e	=	Effective number of load bearing threads	-
p	=	Bolt thread pitch	in
P	=	Internal design pressure	psi
r_o	=	Major radius	in
	=	D/2	
r_p	=	Pitch radius	in
r_r	=	Root radius	in

	=	$d_r/2$	
S	=	Assembly stress	psi
S_a	=	Allowable bolt stress at atmospheric temperature	psi
S_b	=	Allowable bolt stress at design temperature	psi
W	=	Flange design bolt load, for the operating conditions or gasket seating, as may apply.	lb_f
W_{m1}	=	Minimum required bolt load for the operating conditions. For flange pairs used to contain a tubesheet for a floating head or a U-tube type of heat exchangers, or for any other similar design, W_{m1} shall be the larger of the values as individually calculated for each flange, and that value shall be used for both flanges.	lb_f
W_{m2}	=	Minimum required bolt load for gasket seating. For flange pairs used to contain a tubesheet for a floating head or U-tube type of head exchanger, or for any other similar design where the flanges or gaskets are not the same, W_{m2} shall be the larger of the values calculated for each flange and that value shall be used for both flanges.	lb_f
w	=	Width used to determine the basic gasket seating width b_0, based upon the contact width between the flange facing and the gasket	in
y	=	Gasket or joint-contact-surface unit seating load	psi
σ_1	=	Principle stress 1	psi
σ_2	=	Principle stress 2	psi
σ_3	=	Principle stress 3	psi
σ_b	=	Nominal thread bending stress	psi
σ_{dir}	=	Direct stress in root cross-section	psi
σ_{eq}	=	Thread root stress	psi
σ_f	=	Fillet stress	psi
σ_i	=	Initial bolt stress	psi
σ_x	=	Plane stress in x-direction	psi
σ_z	=	Plane stress in z-direction	psi
$\tau_{r-\max}$	=	Maximum transverse shear stress due to thread bending	psi
τ_{zx}	=	Shear stress on plane	psi
FEA	=	Finite Element Analysis	-
COSMOS	=	FEA software	-

ANSI	=	American National Standards Institute
ASME	=	American Society of Mechanical Engineers

ABSTRACT

A model of a bolted flange was created, using the standards that have been produced from the ANSI (American National Standards Institute), which is published by the ASME (American Society of Mechanical Engineers) and sponsored jointly by the ASME and the Society of Automotive Engineers [6].

These standards define nearly every aspect of the bolted flange; from the dimensions, tolerances, materials, and practices used for manufacturing the flanges, as well as the dimensions, tolerances, materials and practices for the bolts to be installed in the flanges.

The ASME code has been developed based on a simple equilibrium calculation of an initial bolt load, a contact pressure of gasket, axial force resulting from internal pressure, as well as best practices learned over time. The objective of this report is to take the recommendations of the ANSI/ASME for pre-loading of the bolted flange, and then analyzing the bolting stresses to validate their practice.

In addition, FEA models were created and examined to compare the classical bolt analysis theory to what was determined using finite element analysis. The areas that had been concentrated on were the fillet connecting the bolt head to the bolt shank, and the first three load bearing threads of the bolt-nut connection. Several iterations were performed, each time increasing the mesh of the finite element model. As the mesh increased, naturally the number of elements would increase, converging the stress given from the FEA model to the exact theoretical solution.

The exact theoretical solution was not precisely matched in the area of the root of the load bearing threads, due to the limitations of the software that was available. However, several supporting references confirm plastic yielding occurring at the root of the first load bearing thread.

1. INTRODUCTION

Due to the recent focus on the production of electrical power without the release of harmful carbon dioxide emissions, emphasis has been placed on the use of higher temperature systems for the production of energy. Bolt-nut connectors (Figure 1) are one of the basic types of fasteners used in machines and structural systems used in high temperature power generation systems. These joints play an important role in the safety and reliability of the systems [1]. It is known that the behavior of real axisymmetric bolted joints in tension is much more complicated than that the conventional theory describes [2]. It is also known that the load distribution among the threads in a threaded connection is not uniform. Most of the applied load is carried by the first three engaged threads [3]. A number of parameters need to be considered when determining the load distribution, which includes the form of the threads, the pitch of the threads, the number of engaged threads, and the boundary conditions.

Additionally, when irregularities such as notches, holes, or grooves exist in the geometry of any structural member, the stress distribution in the neighborhood of the irregularity is altered. Such irregularities are called stress raisers, and the regions in which they occur are called areas of stress concentration [4]. Very few stress intensity factor solutions for threaded connections have been published. Of these, most are difficult to use for comparative purposes because of obscure features of analyses or presentation [5]. In most situations, stress concentrations factors for notches (equivalent to a thread) are obtained experimentally [6].

The objectives of this project is to calculate the bolt loads established by the ASME pressure vessel code, and compute the classical stress calculations in the bolts to shank fillet and the load bearing threads. A comparison of the stresses to an FEA model will also be conducted, and conclusions drawn to support or refute the requirements established by the ASME code.

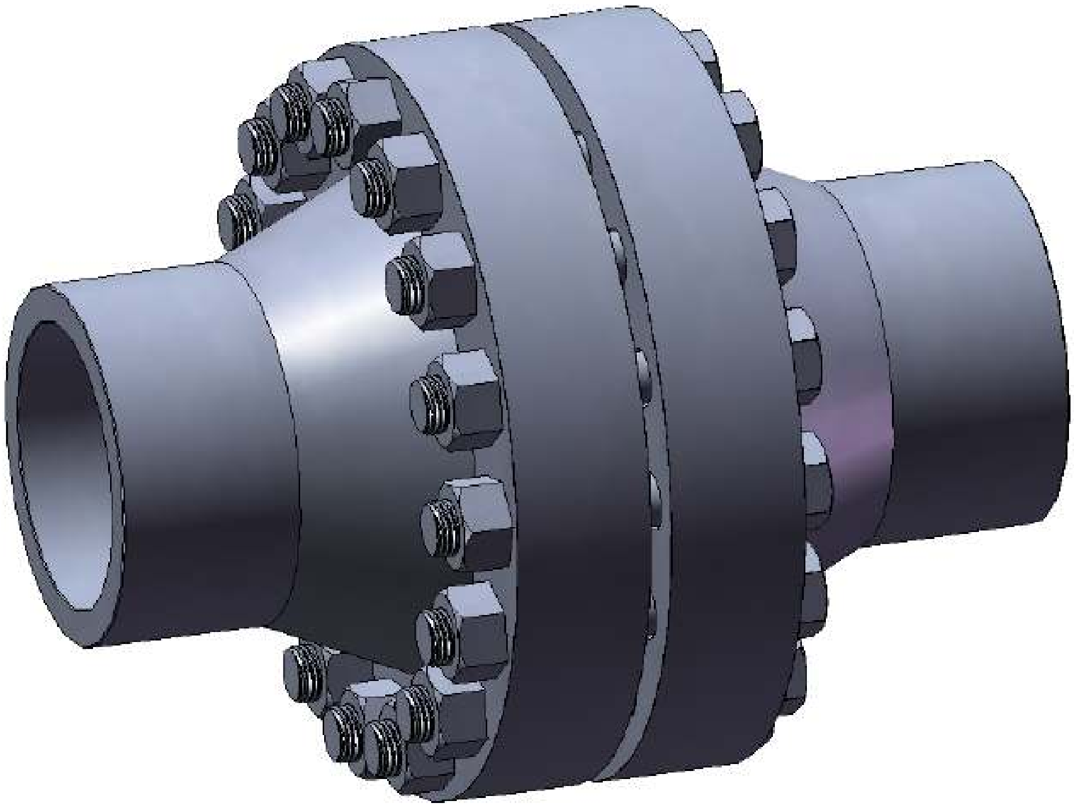


Figure 1: Image of a Bolted Flange Joint

Flange designs (Figure 1) have been the subject of a lot of criticism for the past decade [7]. Of major concern is the lack of the traditional design procedures [8] to quantify the tightness and the effect of temperature. While history shows that most flanges have been in service effectively, problems still arise. Relaxation of the gasket due to thermal transients and creep are the main cause to blame for leakage [9], and a loss of more than half of the initial operating gasket load is commonly encountered in flange joints with certain gaskets operating at relatively high temperatures [7]. A comprehensive survey [9] conducted in 1985 showed that high operational temperatures and thermal transients are one of the major sources of flanged joint failure [10].

Ceramic materials are far more resistant to the failure modes of high temperatures than are metals. It may be more practical from a materials stand point to choose ceramics for the fabrication of a certain component, although it may be more practical to choose metal from a manufacturing standpoint. This decision logic will require the use of bolting to mate the two components. The bolted flange connections are widely used in energy and process plants, and the design procedure used to produce them is specified in the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code Section VIII Division 1 [8]. This design procedure is based on a simple equilibrium calculation of an initial bolt load, a contact pressure of gasket and an axial force resulting from internal pressure [11]. The dimensions for the required bolting are given by the American National Standards Institute (ANSI), while the required material and design stress can be found in Section II Part D of the ASME code.

When analyzing a bolt preload, a vast range of actual preloaded values that can be seen by standard torquing methods. The most common method of loading a bolt is a torque wrench. This method may establish an actual preload of +/- 30% to the value that was set on the torque wrench [6].

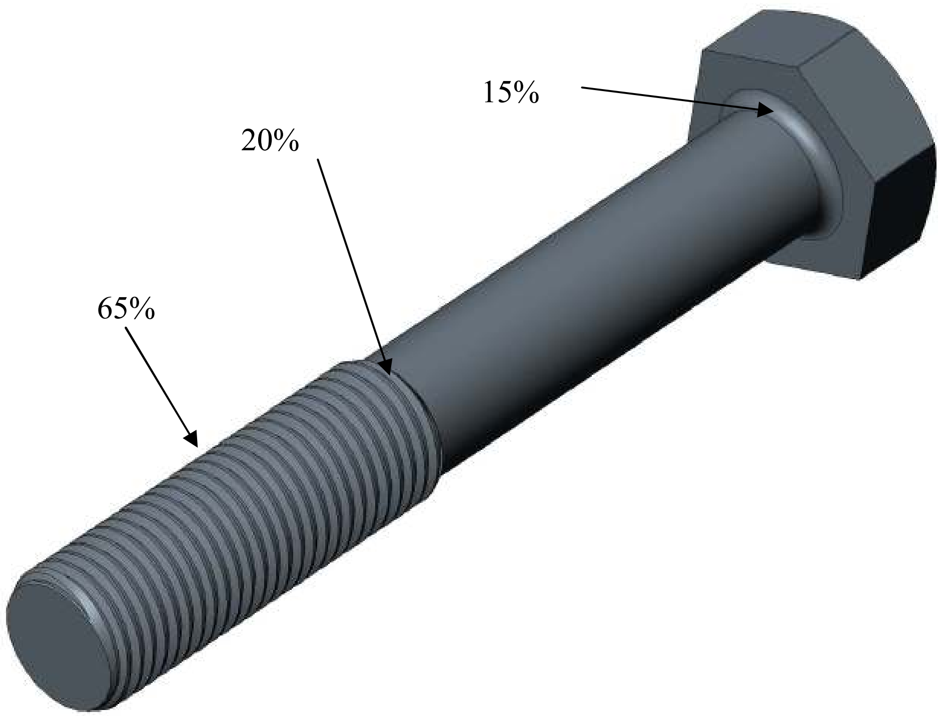


Figure 2: Bolt-Nut Connector Failures

Of all fasteners used in structures, the bolt-nut connectors are one of the most basic types. It has been estimated [12] that bolt nut connector failures are distributed as follow (Figure 2) [1]:

1. 15% under the head
2. 20% at the end of the thread
3. 65% in the thread at the nut face

By using a reduced bolt shank, the situation with regard to fatigue failures of the second type can be improved significantly [13, 14]. Also, with a reduced shank, a larger fillet radius can be provided under the head thereby improving the design with regard to failures of the first type [12]. These recommendations were implemented in the design of the bolt that was analyzed in this study.

This paper utilizes the analytical theory established by the ASME Code, Section VIII, Division 1, and Appendix II. This simpler analysis can be compared to Division 2 of the code, which gives the designer greater freedom of choice, but requires a detailed design analysis to prove the safety of the proposed configuration [15].

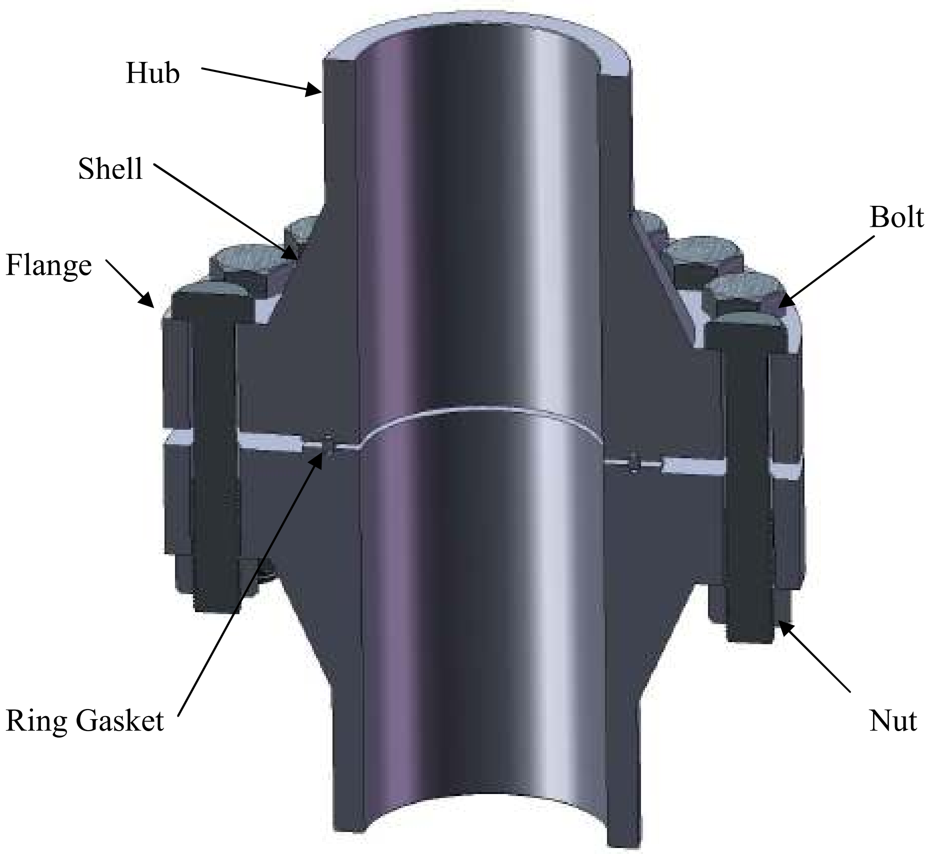


Figure 3: Cross Section of a Bolted Flange Joint

The ASME code has standardized a large variety of flanges, ranging from a ½ inch up to a 24 inch pipe flanges. For the purpose of this report we will be examining the bolting recommended for a 12 inch bolted flange, as seen in Figure 3 above. This figure also highlights the terminology used in describing the flange. A very large array of bolting materials can be selected based on the operating temperature of the flange, but since we are interested in energy production applications, we will use a high strength carbon steel bolting material, SA-540 B24, since higher temperature has little impact on its yield strength.

Ideally, a bolt will experience only a longitudinal stress from the installation preload. As stated, that is ideal, and it should be recognized that a prying action can occur with an eccentrically loaded bolt, such as one of a bolted flange. If the stiffness of a flange joint is high, then prying action can be regarded as negligible [15]. Because of the size of our bolted flange connection, and that the ASME code compensates for these stresses, in the calculations that follow; the prying action will be neglected.

2. THEORETICAL ANALYSIS

As stated previously, the objectives of this project is to calculate the bolt loads established by the ASME pressure vessel code, and compute the classical stress calculations in the bolts to shank fillet and the load bearing threads. The ASME design procedure is based on a simple equilibrium calculation of an initial bolt load, a contact pressure of gasket and an axial force resulting from internal pressure [11]. A comparison of the stresses to an FEA model will also be conducted, and conclusions drawn to support or refute the requirements established by the ASME code.

When designing a bolted flange, it is necessary to consider the in-service clamping force required for the assembly. The initial clamp has to be high enough to compensate for all of the mechanisms which may reduce the clamping force to the in-service level, including [15]:

1. Embedment relaxation
2. Elastic interactions
3. Creep of metal parts, gaskets, etc.
4. External Tensile Loads
5. Hole interference
6. Resistance of joint members to being pulled together
7. Prevailing torque
8. Differential thermal expansion

The entire surface of the gasket is assumed to not be completely loaded as a result of flange rotation. So an "effective area" is computed based on an "effective width" of the gasket. Naturally, a great number of types of gaskets exist, and the effective gasket width, b_0, can be calculated using Table 2-5.1 and Table 2-5.2 of 2008a Section VIII – Division 1 of the ASME Code. These tables are replicated in Appendix A of this report.

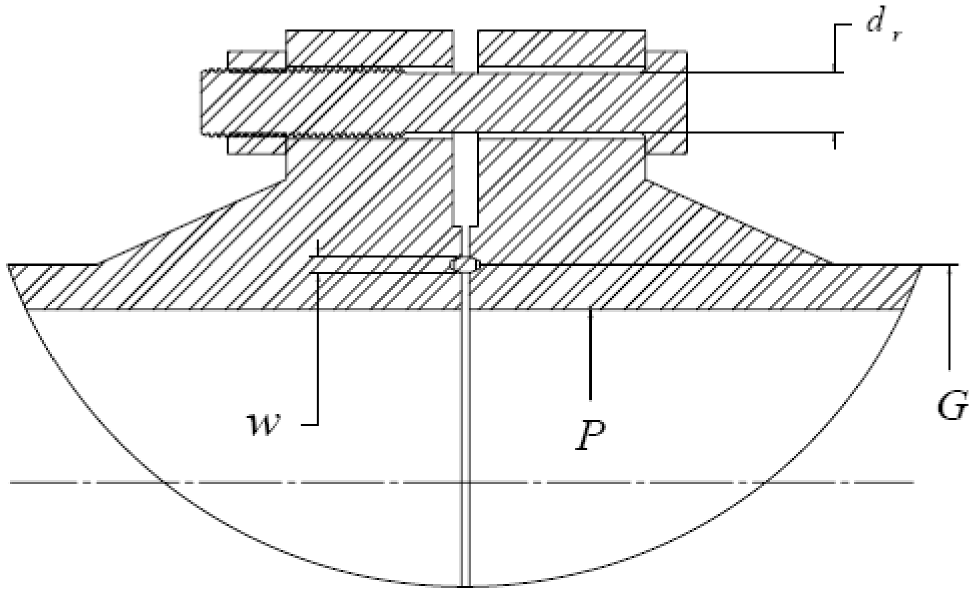


Figure 4 – Bolt Load Analysis

For the purposes of this report, a metal ring gasket will be analyzed, the most common gasket used in high temperature operations [11]. Figure 4 gives a body diagram representing the inputs for the bolt load analysis. The equation below gives b_0, the *basic gasket seating width*, as a function of *w*, the *width of the ring gasket*, and is taken from the ASME code.

$$b_0 = \frac{w}{8} \tag{1}$$

Since the basic gasket seating width of our ring gasket will not exceed ¼ in, the *effective width of the gasket*, *b,* as seen in Table 2 of this report [15], will be equal to the *basic gasket seating width*.

$$b = b_0 \tag{2}$$

When designing a bolted flange connection, it is necessary to calculate the bolt loads based off of a number of requirements for our situation. In general, the calculations should be made for each of the following design conditions. The first condition being the *minimum required bolt load for seating of the gasket* (W_{m2}), and the second being the *minimum required bolt load for the operating conditions* (W_{m1}).

Here, according to [8]

$$W_{m1} = H + H_P \tag{3}$$

where:

$$H = 0.785G^2 P \tag{4}$$

$$H_P = 2b \times 3.14GmP \tag{5}$$

The variable H is the *total hydrostatic end force*, whereas H_P is the *total contact surface compression load*. The *internal design pressure* is P, while G is the *diameter at the location of the gasket load reaction*. The term m is a *gasket factor*, obtained from Table 2-5.1 of 2008a Section VIII – Division 1 of the ASME Code, along with the y term is the *minimum design seating stress*. These constants can take various values, depending on the type and material of the gasket being used.

$$W_{m2} = 3.14bGy \tag{6}$$

The *total required cross sectional area of bolts* (A_m) and the *cross-sectional area of the bolts* (A_b) using the root diameter, are calculated from the required bolt loads from above, and the *allowable bolt stress at atmospheric temperature* (S_a), and the *allowable bolt stress at design temperature* (S_b). The allowable bolt stress values are obtained from Subpart 1 Section II Part D of the ASME code. The total cross sectional area of bolts A_m required for both the operating conditions and gasket seating is the greater of the values for A_{m1} and A_{m2}, where A_{m1} is *the total cross sectional area of bolts at the root of the thread required for the operating conditions*, and A_{m2} is the *total cross-sectional area of the bolts at the root of the thread required for gasket seating*. The bolts to be used shall be made such that the *actual total cross-sectional area of bolts, A_b,* will not be less than A_m.

$$A_{m1} = \frac{W_{m1}}{S_b} \tag{7}$$

$$A_{m2} = \frac{W_{m2}}{S_a} \tag{8}$$

$$A_r = \pi \frac{d_r^2}{4} \tag{9}$$

$$A_b = n_b A_r \tag{10}$$

Where d_r is *the root diameter of the bolts* being used in the bolted flange, A_r is the *cross sectional area using the root diameter,* and n_b is the *total number of bolts* in the flange. As mentioned earlier in this section, the ANSI codes will size the bolts as well as the flanges, which is where we will find our reference for root diameter and total number of bolts.

The *flange design bolt load*, W, is calculated as follows, and is then used to calculate what will be used as the individual *bolt load*, W_b.

$$W = \frac{(A_m + A_b)S_a}{2} \tag{11}$$

$$W_b = \frac{W}{n_b} \tag{12}$$

Section VIII of the ASME code contains a non-mandatory Appendix S that has established design conditions for bolted flange connections. It recommends that the bolts realize a desired stress. Using this recommendation, it is also possible to determine the bolting preload, F_p, using the *nominal diameter of the bolt*, *D*, and the expected *assembly stress, S*.

$$S = \frac{45000}{\sqrt{D}} \tag{13}$$

$$F_p = S \times A_r \tag{14}$$

Once we determine the bolt preload, we will then analyze the two greatest areas of concern, the stress under the head of the bolt, and the stress in the root of the load bearing threads.

The *fillet stress*, σ_f, underneath the head of the bolt (Figure 5) will be calculated using references to determine the *fillet stress concentration factor*, K_f, and multiplying it by the *nominal stress in the bolt*, σ_i. Figure 4 gives the location being referred to when considering the fillet stress of the bolt. These are computed from the from the following equations, fifteen (15) and sixteen (16).

A solution using the Finite Element Analysis (FEA) code COSMOS will also be developed for comparison to the calculated solution. The FEA will be performed using the element type and number of elements that gave the most accurate results, compared to the results given from the classical analysis.

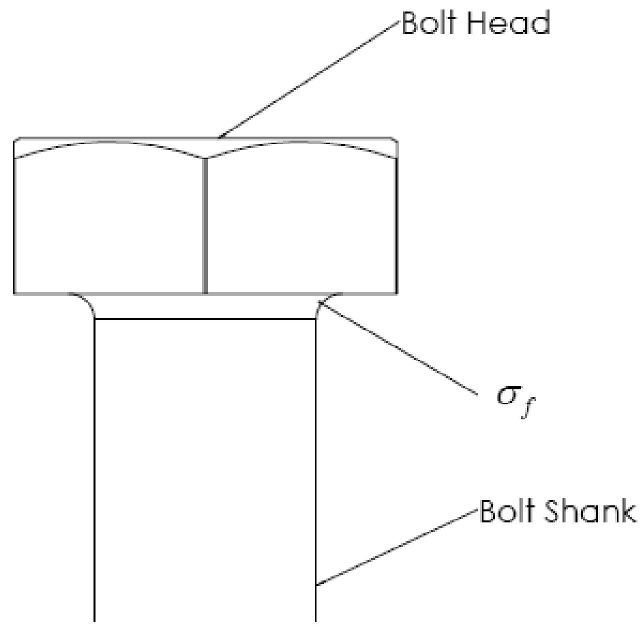


Figure 5 – Bolt Fillet

$$\sigma_i = \frac{F_p}{A_r} \tag{15}$$

$$\sigma_f = K_f \sigma_i \tag{16}$$

The stress in the root of the load bearing threads (Figure 6) can be calculated by using the equations below. These equations assume that the z-axis is in the axial direction of the bolt, the x-axis is in the radial direction of the bolt, and the y-axis is in the tangential direction.

The *root radius* (r_r), *pitch radius* (r_p), and *major radius* (r_o), will all have to be known for determining these stresses, as well as the *effective number of threads that carry the load*, n_e, and the *thread pitch*, p.

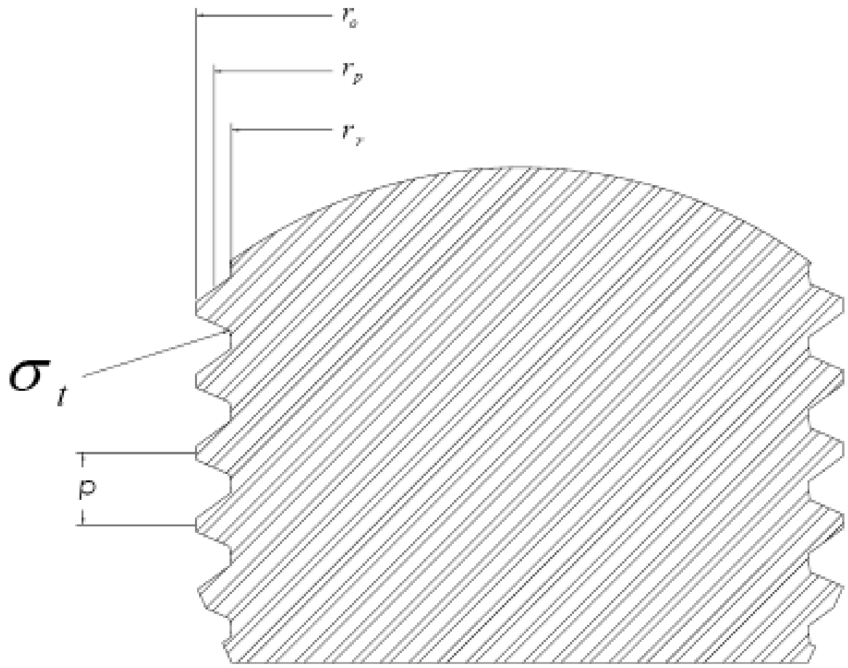


Figure 6 – Bolt Thread

Using the bolt pre-load recommended from the ASME code, and the thread dimensions given established in the ANSI, the stresses that will accumulate in the thread root can be determined. The stresses that will gather are the *direct stress in the root cross-section* (σ_{dir}), the *maximum transverse shear stress due to bending* of the thread ($\tau_{r-\max}$), and the *nominal thread bending stress* (σ_b).

$$\sigma_{dir} = \frac{F_p}{\pi\left(r_0^2 - r_r^2\right)n_e} \tag{17}$$

$$\tau_{r-\max} = \frac{3F_p}{2\pi r_r p n_e} \tag{18}$$

$$\sigma_b = \frac{12F_p\left(r_p - r_r\right)}{\pi r_r n_e p^2} \tag{19}$$

Furthermore, since the root of the threads will be a stress riser, a stress concentration factor will have to be applied. These will be referenced later on in the discussion, and include K_t, K_d, and K_b.

When multiplying the individual stress concentration factors times the corresponding stress, the maximum plane and shear stresses are determined.

$$\sigma_z = K_d \sigma_{dir} \tag{20}$$

$$\sigma_x = K_b \sigma_b \tag{21}$$

$$\tau_{zx} = K_t \tau_{r-\max} \tag{22}$$

The calculated stresses can then be used to produce the principal stresses by solving the cubic equation.

$$\begin{aligned}&\sigma^3 - \sigma^2\left(\sigma_x + \sigma_y + \sigma_z\right) + \sigma\left(\sigma_x\sigma_y + \sigma_y\sigma_z + \sigma_z\sigma_x - \tau_{xy}^2 - \tau_{yz}^2 - \tau_{zx}^2\right)\\ &-\left(\sigma_x\sigma_y\sigma_z + 2\tau_{xy}\tau_{yz}\tau_{zx} - \sigma_x\tau_{yz}^2 - \sigma_y\tau_{zx}^2 - \sigma_z\tau_{xy}^2\right) = 0\end{aligned} \tag{23}$$

For our particular states of stress at our critical point, this equation is reduced to,

$$\sigma^3 - \sigma^2\left(\sigma_x + \sigma_z\right) + \sigma\left(\sigma_z\sigma_x - \tau_{zx}^2\right) = 0 \tag{24}$$

After solving for the three principal stresses, the equivalent Von Mises stress (σ_{eq}) can then be determined. This is the effective stress that exists at the root of the load bearing threads.

$$\sigma_{eq} = \sqrt{\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2} \tag{25}$$

A solution using the finite element analysis (FEA) code COSMOS will be used for comparison to the above calculated solutions. The sensitivity of the results to the number of elements will be investigated.

3. RESULTS AND DISCUSSION

For the purposes of this report we will be examining a 12” pipe flange. The code specifies that a flange of this size will be bolted together using a 2”-4.5UNC hex head bolt. The specifications for these threads can be found in a Machinery’s Handbook [16].

The ASME/ANSI code also governs the specification and design of a bolted flange under ANSI B16.5 and the ring gasket under ASME/ANSI B16.20. The code specifies the dimensions and material of the flange, as well as the necessary bolting and bolt patterns that will be required for the calculations.

A 12” pipe flange as established by the ANSI code, will have a *gasket width* of .4375. The *diameter at the location of the gasket load reaction*, *G*, is determined from both the ANSI code B16.5 that establishes the sizing of the flange, as well as Table 2-5.2 (Table 1 and Table 2 of this report) of Section VIII-Division 1 of the ASME code. The values for *m*, the *gasket factor*, and y, the *gasket seating load*, are determined from Table 2-5.1 (Table 3 of this report) of Section VIII-Division 1 of the ASME code. For the purposes of this report a value of 6.5 has been selected for *m,* and a value of 26,000 psi for y, since we will assume the gasket is made of either a stainless or nickel based alloy.

The maximum allowable tensile stress values permitted for different materials are given in Subpart 1 of Section II, Part D. Table 3 of Part D covers the maximum allowable stress values for high alloy steels used for bolting. This is what is referenced to determine the *allowable bolt stress at atmospheric temperature* (S_a), and the *allowable bolt stress at design temperature* (S_b).

The *total number of bolts* (n_b) used on the flange, as well as the thru hole in the flange, is determined by the sizing, given by the ANSI code. For our 12” flange, there are sixteen (16) bolts. The root diameter of these bolts is also established by the ANSI code, and is given as 1.735 inches for a 2”-4.5UNC thread.

Using these values obtained from the ASME and the ANSI code, and equations one (1) through twelve (12), the *bolt load* (*W*) is determined to be 41,613 lbs. However, Non-mandatory Appendix S of Section VIII – Division 1 of the ASME code recommends an *assembly stress*, *S*, of the flange, that is seen in equation 13. Using equation 14, the *bolt pre-load (F_p)*, is now determined to be 75,229 lbs.

Although this might seem high, it is important to remember that this is a two inch major diameter bolt, which is much larger than bolts used in many major commercial applications. Bolts are recommended to be tightened to approximately 80 to 90 percent of their yield strength [6]. For comparison, a bolt this size would result in an approximate preload of 225,000 lbs for customary commercial applications.

The required stress concentration factors are calculated from Figures 11, 12, and 13 [12], and are located in Appendix B of this report. These referenced sources have been determined experimentally, by such means as photo elasticity or precision strain gage, and documented for design use [4].

After determining the stress concentration factor, we can conclude that the fillet stress is 58,886 psi. A finite element model was created to estimate the stresses in the bolt fillet and to compare them against the results of the calculations above.

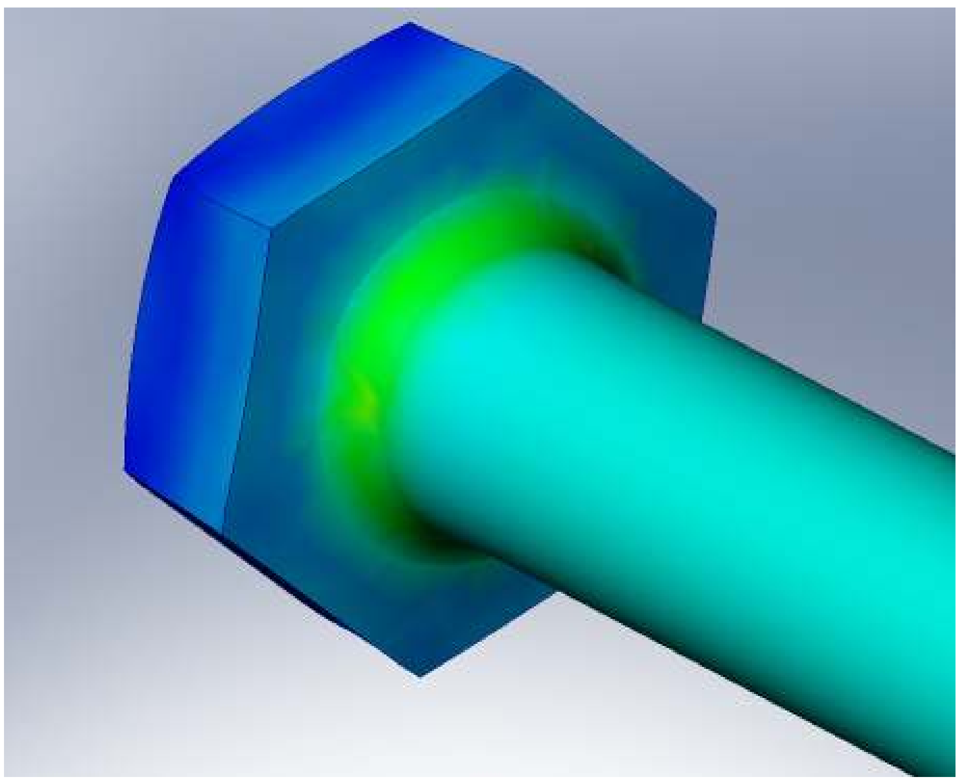


Figure 7 – FEA Model of Bolt Fillet [COSMOS]

Using 75,000 nodes, the finite model (Figure 7) gave a fillet stress of 58,415 psi, a difference of less than 1 %, with respect to the classical stress calculations. It is worth noting however that the number of elements had to be increased significantly to get these results. Figure 8 below shows the results of FEM computations using different meshes.

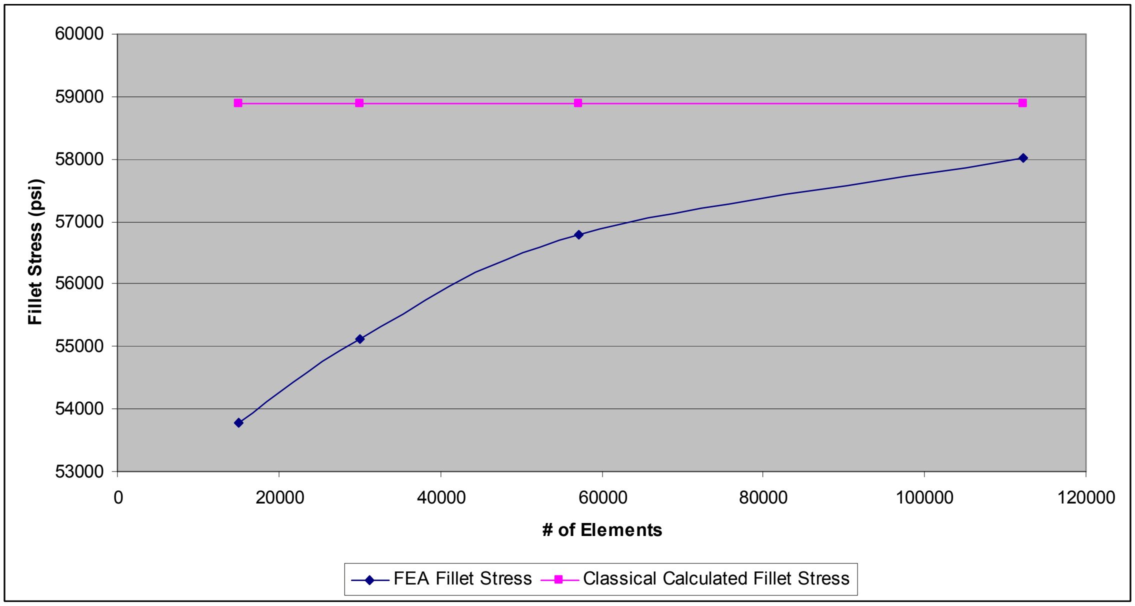


Figure 8 – Number of Elements v. Fillet Stress

We will use the same stress concentration reference for determining the stress in the root of the thread. Another noteworthy consideration is that not all threads of the bolt will carry the load applied to the bolt. This stress is not uniformly distributed because of factors such as bending of the threads as cantilever beams and manufacturing variations from the theoretical geometry [6].

Several factors affect the actual stress concentration factor for the root of a threaded section. Threads that are formed by rolling between dies that force the material to cold-form into the threaded contour of the die grooves are stronger than cut or ground threads in fatigue and impact because of cold working, favorable (compressive) residual stresses at the thread roots, and a more favorable grain structure [6]. Additionally, the maximum fatigue strength is increased if the threads are rolled subsequent to heat treatment, so that the ensuing work hardening and compressive residual stresses are not lost.

The stress concentration factor K_b, for the bending of the thread, and as a function of notch angle is obtained from Figure 12. Stress concentration factors vary depending on whether the element is in bending or axial loading. The chart in Figure 13 will be used to determine the stress concentration factors for K_d and K_t, since these stress are generated as a result of the axial load.

After determining the stress concentration factors for the root of the thread, we can then utilize equations 20 through 25 to determine the Von Mises stress in the root of the first load carrying thread. This calculated stress was 368,480 psi. This seems remarkably high, and would suggest that yielding has occurred in this location. A comparison was then run using finite element analysis to confirm this expectation. Figure 9 shows computed Von Mises stress of the thread root using the FEM model.

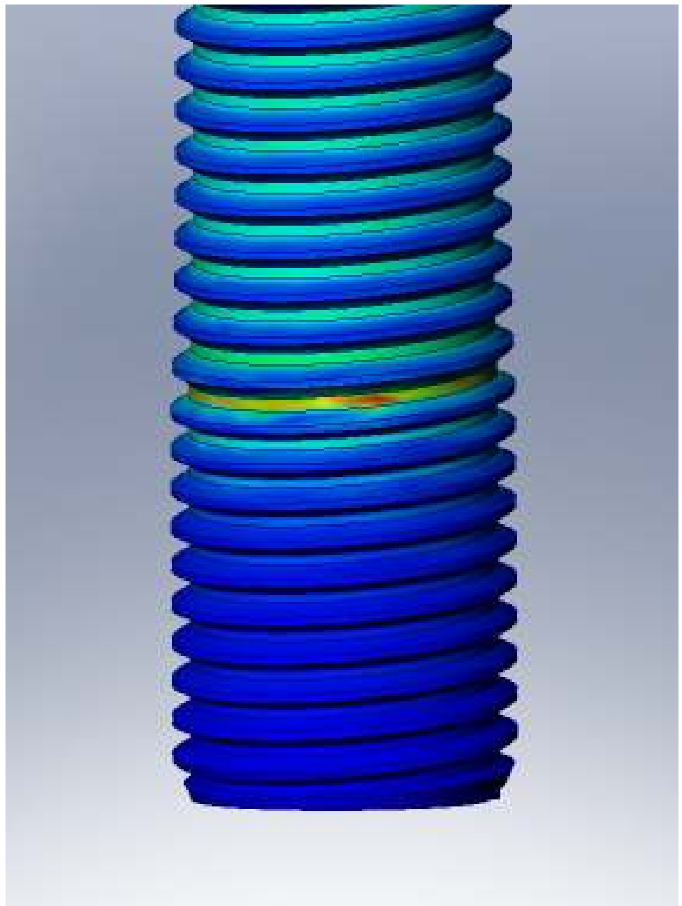


Figure 9 – FEA of Thread Root [COSMOS]

Performing the finite element analysis on the thread root produced a stress of 135,189.68 psi, which although indicated yielding, was not in the vicinity of the calculated value. Upon increasing the number of elements, the value had appeared to approach the calculated value, but due to the limitations of the software being used, could not realize enough elements to obtain agreement with the value obtained, using the referenced equations. Figure 10 shows a graph indicating the convergence of the calculated stress to the FEA determined stress.

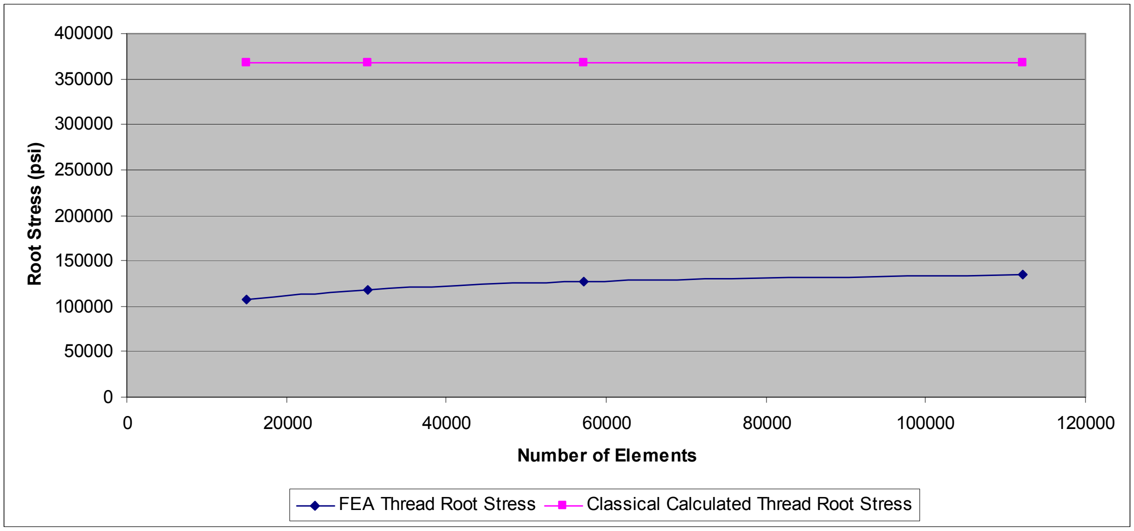


Figure 10 – Number of Elements v. Root Stress

The initial speculation could be that the stresses calculated from the ASME code equations are incorrect. But upon looking into the subject, it is possible and acceptable for yielding to occur at the thread root. The distribution of the axial stress near the ends of the loaded portion of a screw is far from uniform. This is not a concern now because axial stresses are quite low, and because threaded fasteners should always have enough ductility to permit local yielding at thread roots without damage [6].

It is generally considered that the first three threads will carry nearly the entire bolt load, and that the first thread will bear the most load of the three. Two important factors causing this are:

- The load is communal among the three threads as redundant load-carrying members. The shortest path is through the first thread [6].
- The applied load causes the threaded segment of the bolt to be in tension, whereas the mating portion of the nut is in compression. The resulting deflections slightly increase bolt pitch and decreases nut pitch. This tends to relieve the pressure on the second and third threads [6].

The recommendation using *assembly stress*, D, is from a non-mandatory section of the ASME code, would allow us to use the calculated *bolt load*, W_b. This would allow us to reduce our bolt pre-load to 40,124 lbs, nearly cutting our stresses in half. Although this might seem logical, many other problems arise by not having a high initial tension.

With threaded fasteners, it is normal for yielding to crop up at the root of the bolt threads. The rationale behind this is that the higher the initial tension, the less likely the joint members will separate. Also, the higher the initial tension, the greater the friction forces to resist the relative motion in shear [6].

Tests have shown that a typical joint loses about 5 percent of its initial tension with in a few minutes, and that various relaxation effects result in the loss of about another 5 percent within a few weeks [6]. In some cases, it is necessary to retighten bolts periodically during operation.

It is important to recognize that tightening of bolts to full yield strength should not be specified, because of the inaccuracies of standard bolt pre-loading methods, although it is advantageous to obtain a high initial bolt tension. It is significant to be aware that the diminutive amount of yielding that occurs when bolts are tightened to the full proof load is not detrimental to any bolt material of adequate ductility. The advantages to tightening this forcefully are:

- The dynamic load on the bolt is reduced because the effective area of the clamped members is larger [6].
- There is a maximum protection against overloads which cause the joint to separate [6].
- There is a maximum protection against thread loosening [6].

4. CONCLUSION

Multiple variables are involved in calculating the stresses in a bolted flange, and rely on a wide variety of factors that have an effect on these variables. The most important variables being the thread forming method affecting the stress concentration, and the friction factor and tool accuracy affecting the actual preload established in the bolt. This explains why the ANSI/ASME code is based on a conservative and simpler analysis.

Even though the code will specify a pre-load, the variation in actually establishing that pre-load can range from +/- 30%, and a wide range of factors affect the stress concentration on the thread root. Many factors affect the accuracy of torquing, but as an example, the friction coefficient is a main driver for the large variation. But in a testament to the variable interactions, when the coefficient at the thread interface is increased, the stress concentration factor decreases [1]. The most accurate way to determine the stress concentration factor is through physical testing, but even with this approach, the variances in manufacturing can still produce different stress concentrations.

The finite element analysis was not effective in replicating the ASME code computed stress at the thread root. However, this may be due to limitations of the particular software version that was available. The stress risers in the root of the thread were so minute that the program could not generate enough elements to properly resolve the particular area of stress concentration. Increasing the number of elements did help, but the percent difference, in computed stress was still high in this particular case.

The finite element analysis was effective at replicating the theoretical stress at the fillet radius from bolt head to shank transition. However, due to the size of the bolt, this fillet radius was relatively large, and did not create a highly concentrated stress riser.

Although the stresses calculated from the classical equations were accurate, the area of most concern, the first load bearing thread, could not be validated with the use of the finite element model used in the study. Even if this scenario changes, a large safety margin needs to be considered due to the large array of variables that effect establishing the pre-load as well as the stress concentration on the thread root.

APPENDIX A

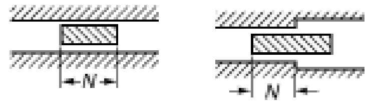
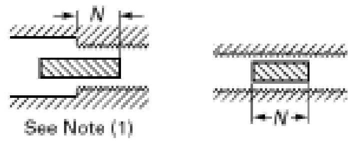
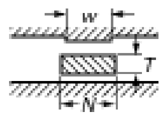
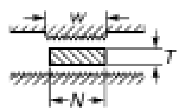
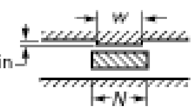
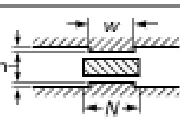
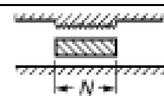
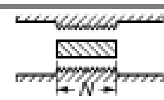
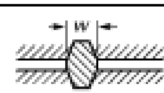
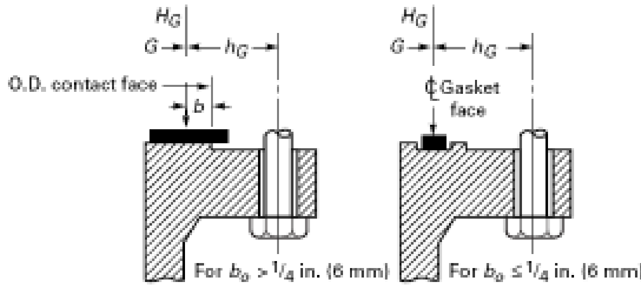
Facing Sketch (Exaggerated)	Basic Gasket Seating Width b_o Column I	Basic Gasket Seating Width b_o Column II
(1a) 	$\frac{N}{2}$	$\frac{N}{2}$
(1b)  See Note (1)		
(1c)  $w \le N$	$\frac{w+T}{2}$; $\left(\frac{w+N}{4}\text{ max}\right)$	$\frac{w+T}{2}$; $\left(\frac{w+N}{4}\text{ max}\right)$
(1d)  See Note (1) $w \le N$		
(2) $^1/_{64}$ in. (0.4 mm) nubbin  $w \le N/2$	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
(3) $^1/_{64}$ in. (0.4 mm) nubbin  $w \le N/2$	$\frac{N}{4}$	$\frac{3N}{8}$
(4)  See Note (1)	$\frac{3N}{8}$	$\frac{7N}{16}$
(5)  See Note (1)	$\frac{N}{4}$	$\frac{3N}{8}$
(6) 	$\frac{w}{8}$	. . .

Table 1 – Effective Gasket Width [Reference 8]

Effective Gasket Seating Width, b

$b = b_o$ when $b_o \leq \frac{1}{4}$ in. (6 mm); $b = C_b\sqrt{b_o}$ when $b_o > \frac{1}{4}$ in. (6 mm)

Location of Gasket Load Reaction



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

Table 2 – Gasket Seating [Reference 8]

Gasket Material	Gasket Factor m	Min. Design Seating Stress y, psi (MPa)	Sketches	Facing Sketch and Column in Table 2-5.2
Corrugated metal:				
Soft aluminum	2.75	3,700 (26)		
Soft copper or brass	3.00	4,500 (31)		(1a),(1b),(1c),(1d); Column II
Iron or soft steel	3.25	5,500 (38)		
Monel or 4%–6% chrome	3.50	6,500 (45)		
Stainless steels and nickel-base alloys	3.75	7,600 (52)		
Flat metal, jacketed mineral fiber filled:				
Soft aluminum	3.25	5,500 (38)		
Soft copper or brass	3.50	6,500 (45)		
Iron or soft steel	3.75	7,600 (52)		(1a),(1b),(1c),[2] (1d)[2];(2)[2]; Column II
Monel	3.50	8,000 (55)		
4%–6% chrome	3.75	9,000 (62)		
Stainless steels and nickel-base alloys	3.75	9,000 (62)		
Grooved metal:				
Soft aluminum	3.25	5,500 (38)		
Soft copper or brass	3.50	6,500 (45)		
Iron or soft metal	3.75	7,600 (52)		(1a),(1b),(1c),(1d), (2),(3); Column II
Monel or 4%–6% chrome	3.75	9,000 (62)		
Stainless steels and nickel-base alloys	4.25	10,100 (70)		
Solid flat metal:				
Soft aluminum	4.00	8,800 (61)		
Soft copper or brass	4.75	13,000 (90)		(1a),(1b),(1c),(1d), (2),(3),(4),(5); Column I
Iron or soft steel	5.50	18,000 (124)		
Monel or 4%–6% chrome	6.00	21,800 (150)		
Stainless steels and nickel-base alloys	6.50	26,000 (180)		
Ring joint:				
Iron or soft steel	5.50	18,000 (124)		
Monel or 4%–6% chrome	6.00	21,800 (150)		(6); Column I
Stainless steels and nickel-base alloys	6.50	26,000 (180)		

NOTES:

(1) This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b given in Table 2-5.2. The design values and other details given in this Table are suggested only and are not mandatory.

(2) The surface of a gasket having a lap should not be against the nubbin.

Table 3 – Gasket Materials and Contact Facings [Reference 8]

APPENDIX B

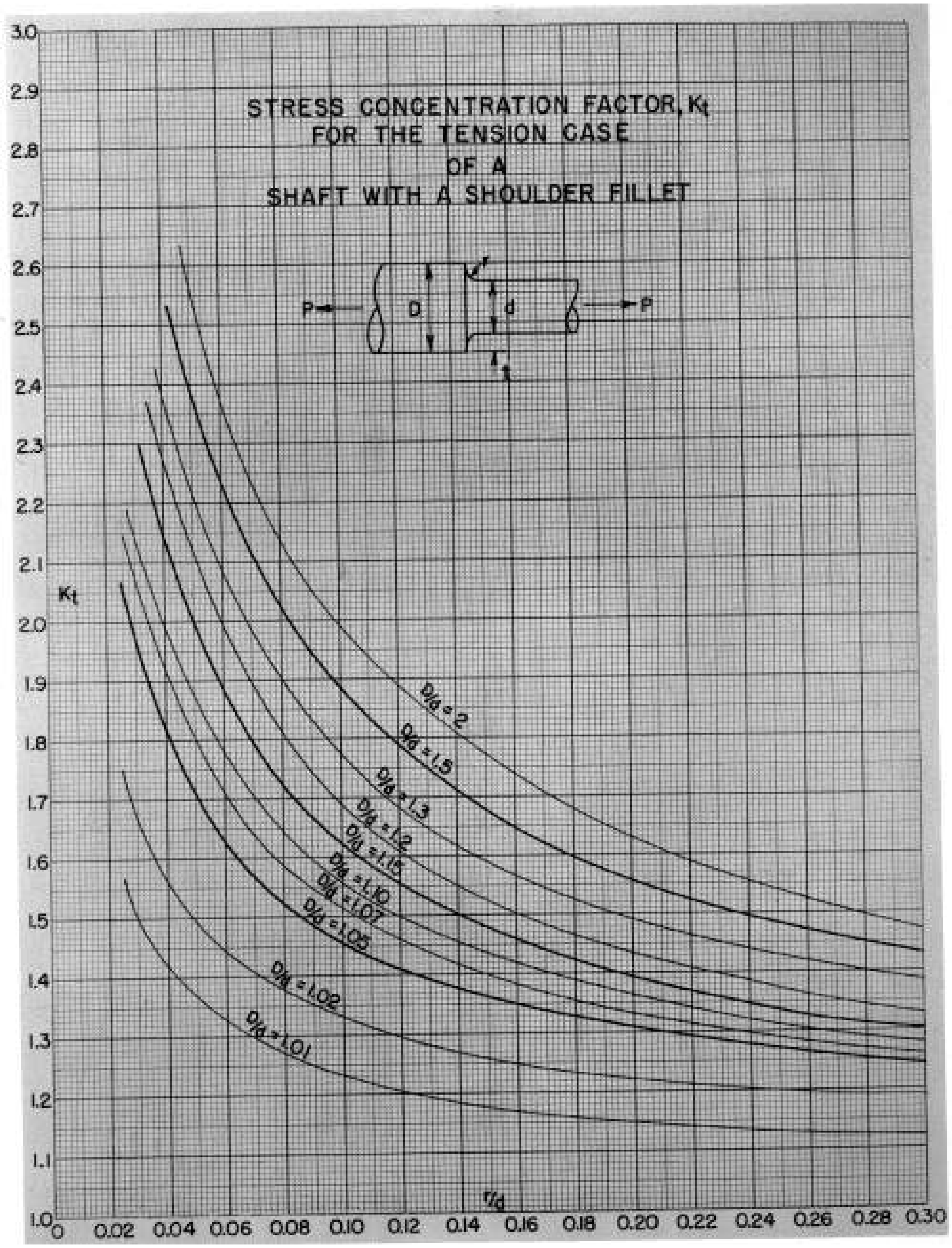


Figure 11 – Fillet Stress Concentration Factor [Reference 12]

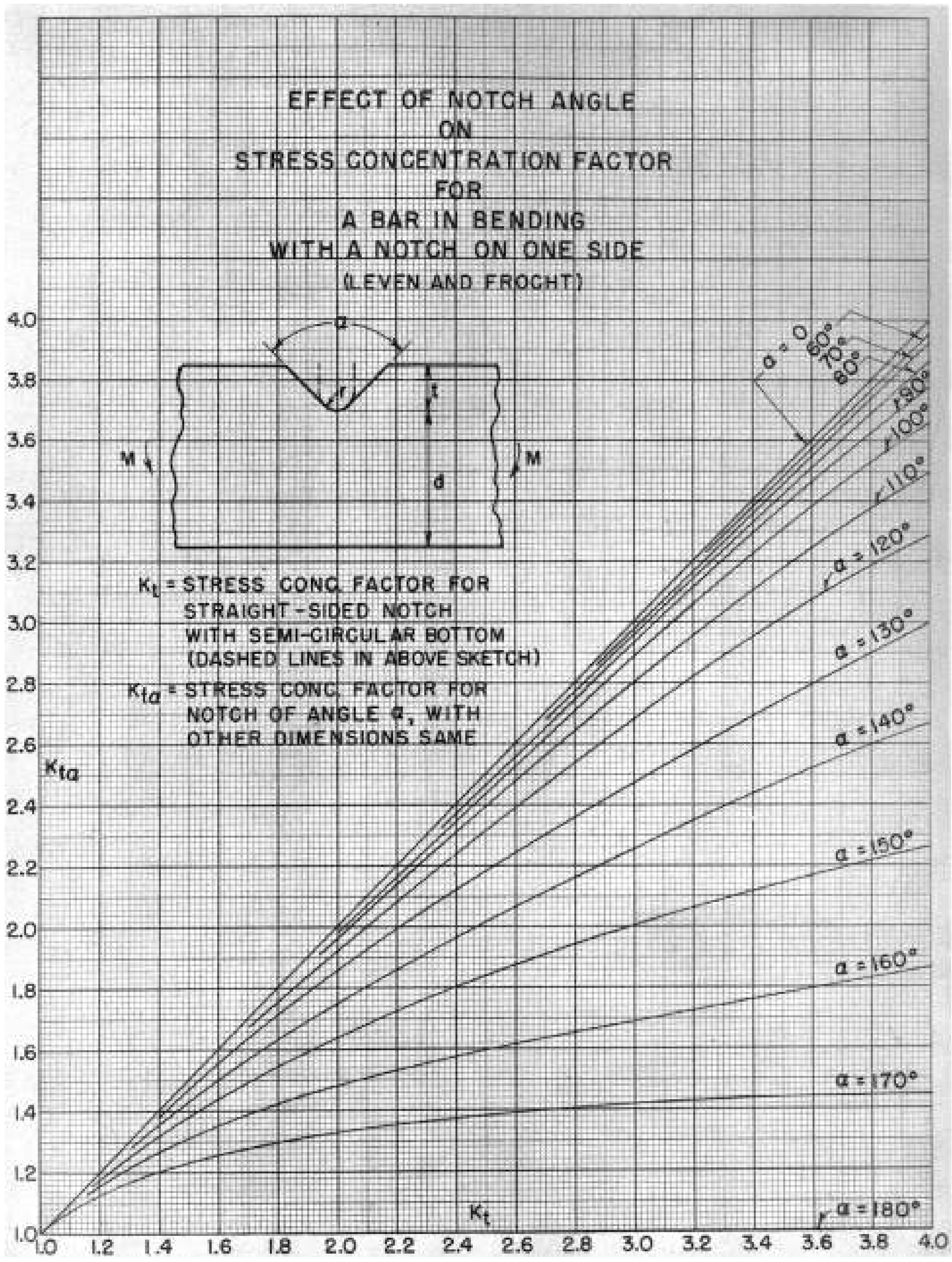


Figure 12 – Root Stress Concentration Factor for Bending [Reference 12]

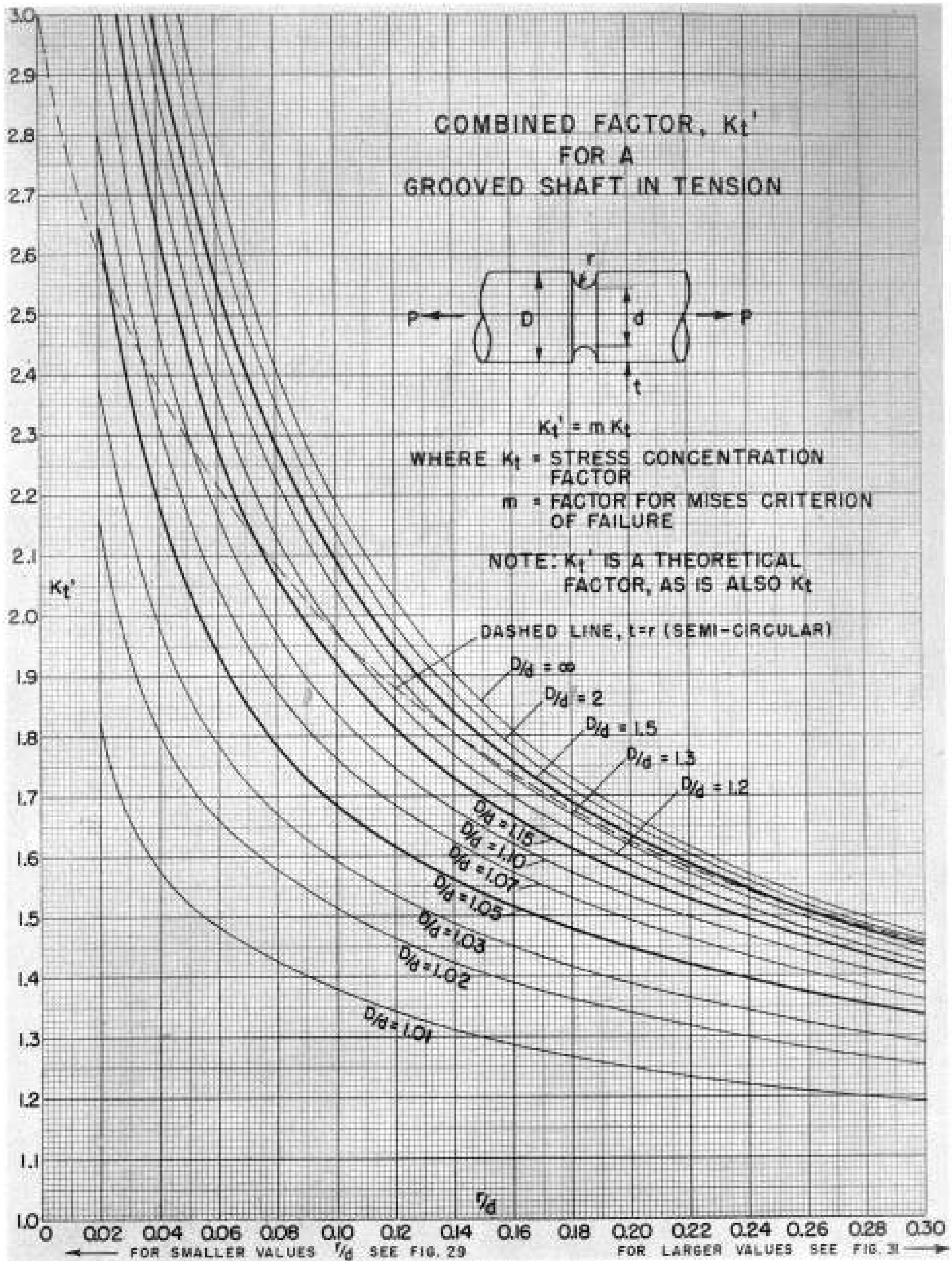


Figure 13 – Root Stress Concentration Factor for Axial Loading [Reference 12]

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