



PDHonline Course S149 (2 PDH)

Analysis of Bolt Torquing

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2020

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Analysis of Bolt Torquing

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Introduction:

During maintenance activities in a power plant environment, the plant engineer is often called upon to make decisions concerning bolts in bolted connections in a variety of situations. Most of these bolts are torqued to achieve a pre-load rather than being tensioned with a tensioner. Examples include: increasing bolt torque beyond vendor manual recommended values to stop leaks, vendor recommendations for torquing are not available, bolt holes need to be enlarged to install replacement components, bolts may require trimming to accommodate replacement component, lack of thread engagement, etc. The ability to perform a component level analysis on a bolt will provide the plant engineer an engineering tool to make the correct technical decision.

- Calculate Tension in bolt when subject to a known torque
- Calculate max stress in bolt when subject to a pre-load and determine its acceptability
- Calculate factor of safety against stripping failure for a known thread engagement
- Determine whether a torque retention device is required
- Calculate nut rotation for a given bolt torque

Definitions:

The *major diameter* is the largest diameter of a screw thread. It is also the nominal size of the bolt.

The *minor diameter* is the smallest diameter of a screw thread. (Also referred to as the root diameter.)

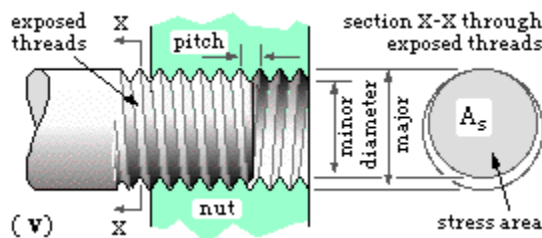
The *lead* is the distance the nut moves parallel to the screw axis when the nut is given one turn. For a single start thread used in bolts, screws, nuts, etc., the lead is the same as the *pitch* and is the inverse of number of threads per inch.

The *pitch diameter* is the diameter of an imaginary coaxial cylinder cutting the threads at a height where the width of the thread and groove are equal. The average of major and minor diameters is approximately equal to the pitch diameter.

The *stress area* is the effective cross-sectional area of the bolt that resists bolt fracture.

The *thread angle* is the included angle between the flanks of a screw thread.

S_m is the design stress intensity for the bolt material per the ASME B & PV Code.



The following table gives bolt geometry and thread engagement data for UNC bolts from 1/4" diameter to 1 1/4" diameter. First row gives the major diameter, second row gives the minor diameter, third row gives the number of threads per inch, and the fourth row gives the distance between nut flats for hex nuts (1/4" to 5/8" sizes), and heavy hex nuts (3/4" to 1 1/4" sizes). The last two rows give the thread stripping areas per inch for external and internal threads respectively. This information is obtained from References 1 & 2.

$$\mathbf{bolt} := \begin{pmatrix} \frac{1}{4} & \frac{3}{8} & \frac{1}{2} & \frac{5}{8} & \frac{3}{4} & \frac{7}{8} & 1 & 1.125 & 1.25 & 1.5 \\ 0.189 & 0.298 & 0.406 & 0.514 & 0.627 & 0.739 & 0.847 & 0.950 & 1.075 & 1.296 \\ 20 & 16 & 13 & 11 & 10 & 9 & 8 & 7 & 7 & 6 \\ \frac{7}{16} & \frac{9}{16} & \frac{3}{4} & \frac{15}{16} & \frac{5}{4} & \frac{23}{16} & \frac{13}{8} & \frac{29}{16} & 2 & \frac{19}{8} \\ 0.368 & 0.576 & 0.779 & 0.998 & 1.21 & 1.43 & 1.66 & 1.88 & 2.11 & 2.58 \\ 0.539 & 0.828 & 1.12 & 1.42 & 1.72 & 2.03 & 2.33 & 2.65 & 2.94 & 3.57 \end{pmatrix}$$

Set the parameter, i , according to the size of the bolt.

$i := 3$ for 1/2" bolt with hex nut as the turning element

Note: The following calculations are based on 1/2" dia. bolt

Determine Torque vs. Tension Relationship

$$\mathbf{major_diameter} := \mathbf{bolt}_{1,i} \cdot \mathbf{in}$$

$$\mathbf{psi} \equiv \frac{\mathbf{lb}}{\mathbf{in} \cdot \mathbf{in}}$$

$$\mathbf{ksi} \equiv \mathbf{psi} \cdot 1000$$

$$\mathbf{minor_diameter} := \mathbf{bolt}_{2,i} \cdot \mathbf{in}$$

d_m is the effective contact diameter for the threads and is the average of major and minor diameters; It is also approximately equal to the pitch diameter.

TPI is threads per inch.

l is lead.

d_c is the effective contact diameter between the nut and joint surface and is the average of major diameter and the distance between parallel nut flats for a hex nut.

$$\mathbf{major_diameter} = 0.5 \mathbf{in}$$

$$\mathbf{d}_m := \frac{\mathbf{major_diameter} + \mathbf{minor_diameter}}{2}$$

$$\mathbf{TPI} := \frac{\mathbf{bolt}_{3,i}}{\mathbf{in}}$$

$$\mathbf{Nut_OD} := \mathbf{bolt}_{4,i} \cdot \mathbf{in} \quad \text{this is the distance between nut flats for hex nuts}$$

$$\mathbf{l} := \frac{1}{\mathbf{TPI}}$$

$$\mathbf{d}_c := \frac{\mathbf{major_diameter} + \mathbf{Nut_OD}}{2}$$

$$\alpha := 30 \cdot \mathbf{deg}$$

2α is the angle of the thread and is 60 degrees for UNC thread

$$\mu := 0.15$$

μ is the coefficient of friction assuming non-lubricated steel bolt and is assumed to be the same for friction between threads and between nut and joint surface. A more realistic coefficient of friction for fasteners that has been in service may be 0.35-0.40 rather than 0.15.

Given an applied torque, T, the preload force F in the bolt is given by

$T = F \times \text{Factor}$ where:

$$\text{Factor} := \frac{d_m}{2} \cdot \left(\frac{l + \pi \cdot \mu \cdot d_m \cdot \sec(\alpha)}{\pi \cdot d_m - \mu \cdot l \cdot \sec(\alpha)} \right) + \frac{\mu \cdot d_c}{2} \quad \text{See Reference 3; "long formula"}$$

This expression can be mathematically simplified as $\frac{l}{2 \cdot \pi} + \frac{\mu \cdot d_m}{2 \cdot \cos(\alpha)} + \frac{\mu \cdot d_c}{2}$ such that

- the first term represents the amount of torque to stretch the bolt and compress the joint
- the second term represents the amount of torque required to overcome the friction between the nut and bolt threads
- the third term represents the amount of torque required to overcome the friction on the face of nut when multiplied by the preload F.

From this expression, it can be seen that bolt and thread geometry and the coefficient of friction significantly influence the relationship between torque and preload. If the above expressions are evaluated it will be seen that only about 10% of the applied torque works to achieve the preload and the remainder works to overcome friction.

Further, it should be noted that only the torques represented by the first two terms will create a twist in the body of the bolt. Evaluating the above expression,

$$\text{Factor} = 0.099in$$

Suppose we apply a torque of 20 ft.lb.(240 in.lbs)

$$T := 240 \cdot \text{in} \cdot \text{lb}$$

$$F := \frac{T}{\text{Factor}}$$

$$F = 2428.287b$$

There is a simple empirical formula ("short formula") that is widely used in the industry to calculate the relationship between Torque and Tension. It is given by:

$$\text{Factor} := \text{Nut_factor} \cdot \text{major_diameter} \quad \text{where } \text{Nut_factor} := 0.20 \text{ for unlubricated NEW steel fasteners}$$

$$\text{Factor} = 0.1in$$

Nut factors are determined empirically. For values of Nut factor for various lubricated fasteners. See Reference 4.

$$F := \frac{T}{\text{Factor}}$$

$$F = 2400lb$$

This short formula gives results which compares well with the results of the long formula. However, it is important for the Nut factor to be selected carefully, since it significantly affects the results.

Calculate maximum stress on the bolt during the torquing process:

We can either use the results of the long or short formula for the value of F for this analysis.

$$F := 2400 \text{ lb}$$

Tensile stress will be calculated using the stress area (d_s is the diameter based on stress area):

$$A_s := 0.7854 \left(\text{major_diameter} - \frac{0.9743}{TPI} \right)^2$$

$$\text{Tensile_stress} := \frac{F}{A_s} \quad A_s = 0.142 \text{ in}^2$$

$$d_s := \sqrt{\left(\frac{4 \cdot A_s}{\pi} \right)}$$

$$\text{Tensile_stress} = 16.913 \text{ ksi}$$

Allowable Tensile Stress:

There is no clear guidance for the maximum allowable average tensile stress for preload. For ASME components, ASME III NB-3230 could be used. Provided differential thermal expansion does not create additional stresses, the maximum allowable average tensile stress could be considered as $2 \times S_m$ since internal pressure would not increase the bolt loading. For ASTM A193 B7 at a service temperature of 200 deg. it would be 2×32.6 ksi and about 60% of yield.

Structural bolts such as ASTM A325 bolts are pre-loaded to 70% of ultimate tensile strength or about 90% of yield strength for slip critical connections and would yield during the torquing process. Hence, they should not be re-used since cumulative deformations would lead to bolt rupture.

$$\text{Thread_torque} := T - \frac{\mu \cdot d_c \cdot F}{2}$$

Note: The full applied torque is not experienced by the cross-section of the bolt since some of this torque goes to overcome friction under the nut head.

$$\text{Shear_stress} := \frac{16 \cdot \text{Thread_torque}}{\pi \cdot d_s^3}$$

The shear stress is the thread torque divided by the polar section modulus. Thus,

$$\text{Shear_stress} = 8.456 \text{ ksi}$$

The shear stress is the highest at the periphery of the bolt cross-section which is also subject to an axial tensile stress. For ASME components, the rules of NB-3232.2 could be used to determine the maximum allowable stress. This requires calculating the stress intensity which is defined as twice the maximum shear stress or the difference between the principal stresses.

Thus,

$$\mathbf{Stress_intensity} := \sqrt{(\mathbf{Tensile_stress})^2 + 4 \cdot (\mathbf{Shear_stress})^2}$$

$$\mathbf{Stress_intensity} = 23.918\mathbf{ksi}$$

For an ASTM A93, Gr. B7 bolt, the allowable stress intensity would be $3 S_m$, i.e. $3 \times 35 \text{ ksi} = 105 \text{ ksi}$ and has not been exceeded.

However, Code Interpretation III-1-91-15 states that Section NB-3232 does not apply to this situation. Thus, we could instead use Von Mises stress for combining tensile stress with shear and check against yield for ASME or non-ASME components

Von Mises stress is given by combining tensile stress with shear stress as follows:

$$\mathbf{Von_stress} := \sqrt{(\mathbf{Tensile_stress})^2 + 3 \cdot (\mathbf{Shear_stress})^2} \quad \text{See Reference 6.}$$

$$\mathbf{Von_stress} = 22.373\mathbf{ksi}$$

If this is an ASTM A193, Gr. B7 bolt, it would not have reached the plastic range since the von Mises stress is much less than the yield strength of the bolt (105 ksi).

The determination of the required pre-load is outside the scope of this paper since it is dependent on the function of the joint, for eg. gasket seating, joint sealing, etc. A minimum residual pre-load stress of 20% of yield strength of the bolt is recommended by Ref. 8 for torque retention. Otherwise, torque retention devices such as lock nuts, jam nuts, lock washers, staking of threads, locking adhesives, etc would be required.

Thread Engagement Issue

When a bolt is fully engaged into a nut or when a screw has a depth of engagement at least equal to the diameter of the screw, the thread engagement is adequate. Special situations arise when the bolt is not fully engaged into the nut or when the screw is engaged into a much weaker material. Sufficient design margin against failure by thread stripping should be provided.

Calculate Factor of Safety Against Thread Stripping with A36 body

Method I:

$$\mathbf{Thread_engagement} := 0.40 \mathbf{in}$$

$$\mathbf{Ultimate_tensile_strength} := 58 \cdot \mathbf{ksi} \quad (\text{Use the material property for the weaker material})$$

$$\mathbf{Ultimate_shear_strength} := 0.6 \cdot \mathbf{Ultimate_tensile_strength} \quad \begin{array}{l} \text{for Carbon Steel;} \\ \text{for Stainless Steel use 0.55} \end{array}$$

$$\mathbf{Thread_stripping_area} := \frac{\pi \cdot d_m \cdot \mathbf{Thread_engagement}}{3} \quad \text{Using methodology in Reference 7.}$$

$$\mathbf{Shear_Capacity} := \mathbf{Ultimate_shear_strength} \cdot \mathbf{Thread_stripping_area}$$

$$\mathbf{Factor_of_safety} := \frac{\mathbf{Shear_Capacity}}{F}$$

$$\mathbf{Factor_of_safety} = 2.751$$

Method II

$$\text{Thread_stripping_area} := \text{bolt}_{5,i} \cdot \text{Thread_engagement} \cdot \text{in}$$

Note: External threads with methodology in Reference 1 used.

$$\text{Shear_Capacity} := \text{Ultimate_shear_strength} \cdot \text{Thread_stripping_area}$$

$$\text{Factor_of_safety} := \frac{\text{Shear_Capacity}}{F}$$

$$\text{Factor_of_safety} = 4.518$$

Calculation of Nut Rotation

Although it is theoretically possible to calculate the nut rotation as a function of the preload, it does not correlate well with observations for preloads less than yield. If tightening to preloads beyond yield strength is permissible, this is a cost effective and reliable method and is widely used for structural connections when installing A325 or A490 bolts.

However, an approximate amount of nut rotation can be calculated based on Tables 3 and 4 of Reference 9.

The bolt in this example has a preload of 2400 lbs. According to Table 4, the nut rotation for 1/2" A325 bolt would be 1/3 turn or 120 degrees for a bolt less than 4 diameters in length. The required preload per Table 3 is 12000 lbs. Therefore, the rotation for 2400 lbs would be:

$$\text{Nut_rotation} := \frac{F}{12000 \text{ lb}} \cdot 120 \text{ deg}$$

$$\text{Nut_rotation} = 24 \text{ deg} \quad \text{i.e. Approx. } 1/12 \text{ th turn --- } 0.5 \text{ nut flat}$$

Acknowledgements:

The author wishes to thank Dr. Yinyao Jiang, Dr. Peter Turula, and Bill Eccles for helping the author understand some technical points presented in this paper.

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