In considering thread strength, the concepts of “stress” and “strain” — the basics of metal strength evaluation—are necessary. This article mainly utilizes stress to explain how to evaluate thread strength. In the cases where threads are fastened via Torque Method, the load exerting on the bolt is applied in more severe conditions than using bolt heaters and other methods that apply tension directly. I am going to clarify the reason by utilizing the so-called “Mises stress” that is widely used for strength evaluation. Failures of threaded fasteners mostly occur around thread portions. Accordingly, I will explain the reason, why thread rupture is likely to occur at the thread root near the nut bearing surface, by depicting Mises stress distributions around the engaged threads. Furthermore, I will show “the expansion of plastic zone during bolt fastening” obtained by Finite Element Analysis, which may be the criterion for determining the magnitude of axial bolt force.

Fastener Expert 101 Series:

Static Strength of Threaded Fasteners

The Basics of Thread Strength

by Toshimichi Fukuoka

Metal Strength & Thread Strength

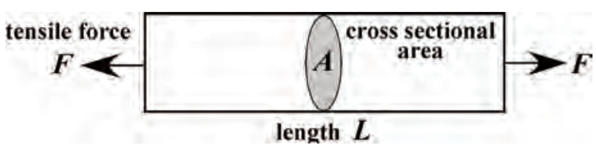


Fig. 1

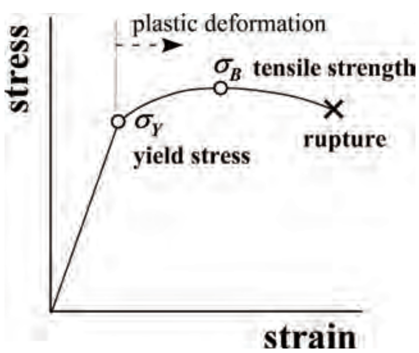


Fig. 2

Figure 1 shows a cylindrical bar subjected to axial tensile load F. Setting the cylindrical bar length as L, the cross sectional area as A, and the elongation due to the load as u, we can express stress σ and strain ε in the following equation.

$$\sigma = F / A, \ \varepsilon = u / L \quad (1)$$

Referring to the above equation, stress can be expressed as “the force exerting on unit area”, and strain as “the ratio of the deformation to the original length”. If we represent load F by N (Newton) and length by mm, the unit of stress becomes N/mm^2. The magnitude of stress in the case of 1N of force exerting on 1mm^2 is 1MPa. **Figure 2** shows a “stress-strain diagram” for carbon steel that is widely used for threaded fasteners. “Stress-strain diagram” can be derived by applying tensile load on a test specimen having a cylindrical bar shape until it ruptures. From this graph we can learn various mechanical behaviors of the materials. The maximum stress that appears in the graph is called “tensle strength σ_B”. The turning point where the straight line in the graph turns into a curved one is called “yield stress σ_Y”. The latter results from the occurrence of the so-called “plastic deformation” phenomenon where the deformations resulting from large loads won’t be recovered even if the load is removed.

The strength of threaded fasteners is commonly expressed by a single number, such as 10.9, 8.8, 4.6, which includes the information on tensile strength σ_B and yield stress s_Y. For example, a bolt labelled 10.9 has 1000MPa of tensile strength and 900MPa of yield stress. Namely, the value before the decimal point is to be multiplied by 100 to obtain tensile strength, and then the tensile strength multiplied by the number 0.9 represents the yield stress. Accordingly, the strength classes of 10.9 and 8.8 correspond to alloy steel and medium carbon steel bolts with high strength, and those of 4.6 and 4.8 correspond to low carbon steel bolts.

When using high strength bolts, we can apply the same level of axial force with smaller bolts. Accordingly, it is preferable to use thread materials having the strength as high as possible. However, “delayed fracture” becomes a problem when using the materials with tensile stress exceeding 1200MPa. Delayed fracture is a phenomenon where the materials turn brittle due to the effect of hydrogen, and after experiencing some service conditions, followed by the completion of fastening operation, the materials break abruptly in the brittle fracture mode. On the other hand, with the development of processing technology in recent years, threaded fasteners that have tensile strength beyond 1200MPa and are free from delayed fracture start to be manufactured lately.

Thread Fastening Strength

Since a thread is processed along a spiral, its strength is relatively lower than a cylindrical bar with the diameter equal to the nominal diameter. When evaluating the thread strength, therefore, we adopt a value called stress area, and substitute threads with complex geometry for cylindrical bars. By using thread nominal diameter d and thread pitch P, the diameter d_s of stress area A_s can be expressed as follows.

$$d_s = d - 0.9382P \quad (2)$$

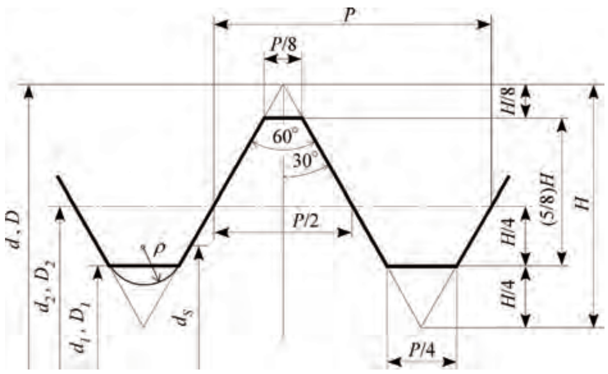


Fig. 3

Figure 3 shows the basic profile of triangular threads. The magnitude of d_s is between the minor diameter d_1 and pitch diameter d_2 of the threads.

The strength of the engaged threads is evaluated by the stress σ_{th} obtained by dividing axial bolt force by stress area A_s.

$$\sigma_{th} = F_b / A_s \quad (3)$$

The above equation is applicable to the cases of the tightening operation being conducted using bolt heaters or hydraulic tensioners, in which the axial bolt force is applied as a direct tension. On the other hand, if we use Torque Method for fastening, in addition to the tensile stress due to axial bolt force, shear stress τ_{th} generated by the torque T_1 exerted on the threaded portion will occur as well. I have explained T_1 in my third article on Fastener World Magazine.

$$\tau_{th} = 16T_1 / (\pi d_s^{\ 3}) \quad (4)$$

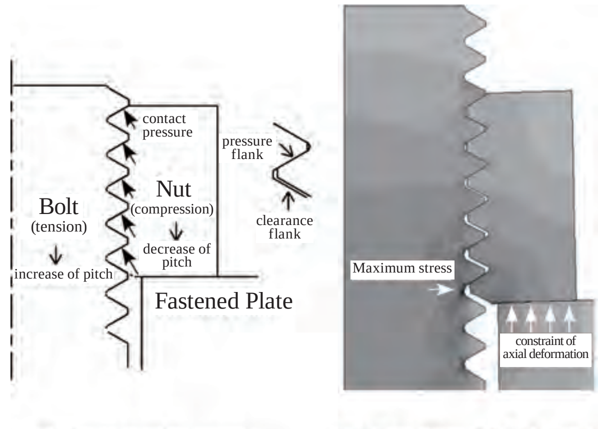
In the case where fastening is performed with Torque Method, the bolt strength is evaluated by an equivalent stress called Mises stress.

$$\bar{\sigma} = \sqrt{\sigma_{th}^{\ 2} + 3\tau_{th}^{\ 2}} \quad (5)$$

Here, plastic deformation is supposed to begin once Mises stress reaches the yield stress σ_Y of the materials. As a calculation example, in the case of a M16 bolt with target axial stress of 100MPa and COF at thread surface being 0.12, we find that Mises stress on the engaged threads is 154MPa using the above equations.

Stress & Load Distributions on Engaged Threads

The rupture of threaded fasteners mostly initiates from the engaged threads. The reason can be explained by the characteristic stress & load distributions on engaged threads. As shown in **Figure 4(a)**, in fastening a bolt,



(a) forces exerting around engaged threads (b) Mises stress distribution and deformation pattern

Fig. 4

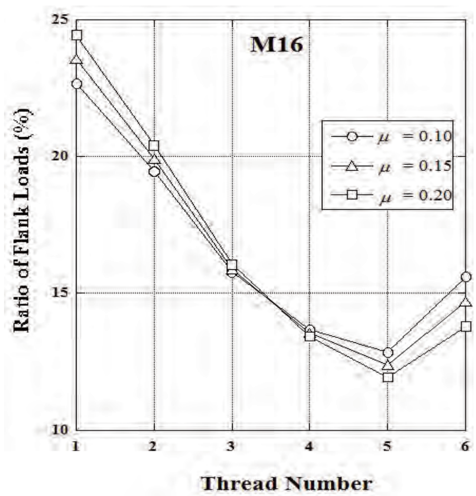


Fig. 5

tension force exerts on the bolt and compression one on the nut. Then, thread pitch of bolt increases and that of nut decreases, and only one side of the thread surfaces is in contact. Thread surfaces in contact and out of contact each other are designated as pressure flank and clearance flank, respectively. Accordingly, Mises stress on the engaged threads present a characteristic distribution pattern as in **Figure 4(b)**. The maximum stress occurs around the first bolt thread root that is nearest to the nut bearing surface, and the location is consistent with the place where fatigue fracture is likely to occur.

The reason that the large stress occurs at the bolt thread root near the nut bearing surface can be explained with a value called "ratio of flank loads". Ratio of flank loads is the "proportion of the axial load sustained by each thread with respect to the total axial bolt force", assuming that the threads geometry is axi-symmetric. Figure 5 shows an example of the results of analysis. The target is an M16 thread, and the number of engaged threads is 6. Although the distribution pattern is affected by coefficient of friction μ on the thread surface, the first thread is found to sustain nearly 25% of the axial bolt force. Consequently, large stresses occur at the bolt thread root near the nut bearing surface.

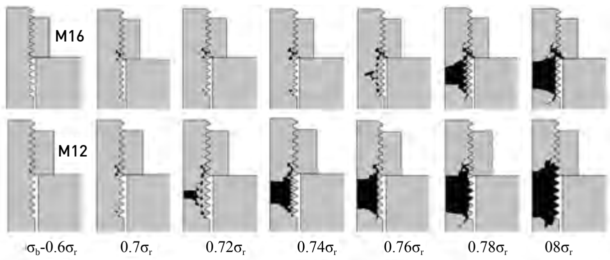


Fig. 6

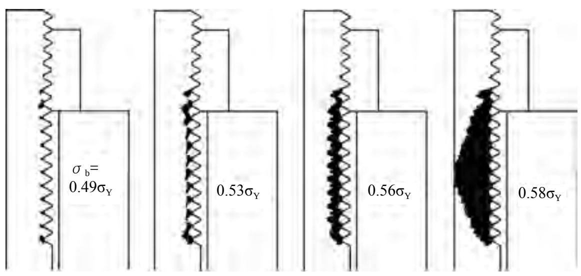


Fig. 7

Plastic Deformation in Threaded Fasteners

When determining the magnitude of axial bolt force, it is common to employ the yield stress of thread materials. As shown in the previous section, fairly larger stresses exert on the engaged threads, comparing to those on the bolt shank. Here, as a criterion for determining axial bolt stress σ_b, a traditional standard proposes that σ_b is set to be 0.6 or 0.7 times yield stress σ_Y. In this section, I will offer a hint for determining axial bolt force by showing the expansion pattern of plastic deformations during bolt fastening. Targeting M16 and M12 bolts, **Figure 6** shows the progression of plastic zone as axial bolt stress is increased. The bolt is subjected to only pure axial force, and the coefficient of friction on the contact surface is 0.15. The magnitude of axial bolt stress σ_b is represented as the ratio to the yield stress σ_Y of the materials. It is found from the Figure that plastic zone starts to expand drastically when axial bolt stress exceeds approximately $0.7\sigma_Y$. Furthermore, this tendency appears more apparently in M12 bolts with small nominal diameter. The reason can be explained with the fact that threaded fasteners are not geometrically similar as explained in my first article in Fastener World Magazine, and that the ratio of the cross sectional area of thread root to that of shank is relatively small on in a M12 bolts with small nominal diameter.

Figure 7 shows the progression of plastic zone of M12 bolts fastened with Torque Method. Although coefficient of friction is a rather large value of 0.2, the extent of plastic zone expansion is larger, comparing to the case where only the axial force is applied. As another significant characteristic, plastic deformation initiates around the first bolt thread root, and then it expands around the thread roots of unengaged threads, and eventually the central portion of the unengaged threads is entirely in plastic zone. This is consistent with the actual on-site phenomena reported rather frequently that an excessively fastened bolt breaks around the central portion of unengaged threads.

Conclusion

This article explains the basic approach to evaluate the strength of threaded fasteners. Particularly, I explained the mechanism of bolt rupture occurring at the thread root near the nut bearing surface by means of the analytical results of stress and load distributions around engaged threads. In addition, by showing the progression of plastic zone in the fastening process, I offered a guideline for determining axial bolt force based on the ratio of yield stress to axial bolt stress.

Reference

Toshimichi Fukuoka, “Threaded Fasteners for Engineers and Design – Solid Mechanics and Numerical Analysis –“, pp.137-157, Corona Publishing Co., Ltd. (2015)