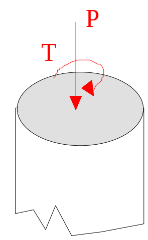
Static Screw Stresses

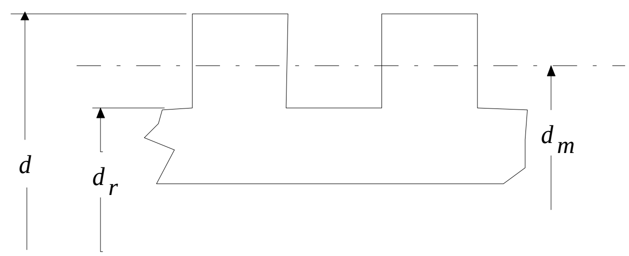


A power screw is generally subjected to a torque and an axial load. Both axial normal and shear stresses result. These stresses are related to the applied loads.

Torsion: $\tau = \dfrac{Tc}{J}$

where $c = d/2$ and $J = \pi d^4/32$

Therefore, $\tau = \dfrac{16T}{\pi d^3}$



For a power screw or threaded fastener, we generally use

$$d = d_r$$

which results in a higher (conservative) shearing stress.

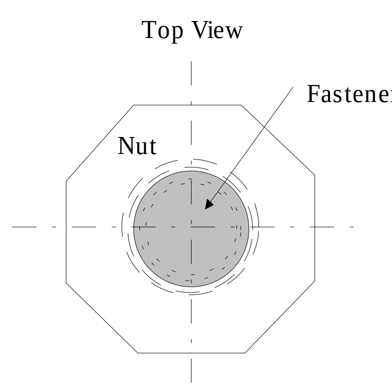
Axial Stress: $\sigma = \dfrac{P}{A}$

Where A is the effective area. For threaded fasteners this is generally the tensile stress area A_t (found in the tables for thread sizes). For a power screw we generally use

$$A = \pi d^2/4$$

where $d = d_r$, resulting in a conservative stress.

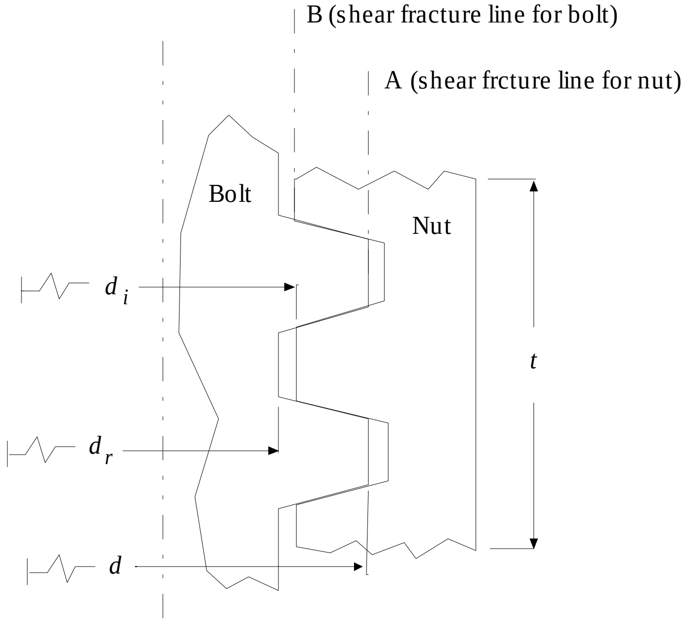
Bolt/Nut Interface- Thread Shear and Bearing Stress



At the interface of the bolt and the nut, several diameters are present. For the bolt, the root diameter and the major diameter are external diameters. For the nut, the threads are cut on the inside and the critical diameter is the inside diameter (d_i), which is approximately the same the as the root diameter (d_r) of the bolt. At the interface, a shear failure can result in either the nut or the bolt.
The location of these possible fracture planes is shown in the figure on the next page.

Bearing Stress:

At the interface of the nut and the bolt (where the threads for each are in contact) a bearing stress is developed. This bearing stress is computed by using the projected area over which the two surfaces are in contact. The projected area is approximated by



$$A_p = \frac{\pi}{4}\left(d^2 - d_i^2\right)$$

where d is the major diameter of the bolt and d_i is the inside diameter (minor diameter) of the nut. For threaded fasteners we can approximate d_i as d_r of the bolt. The number of threads in contact is given by the nut thickness divided by the pitch of the threads (t/p).

The bearing stress is therefore given by

$$\sigma_{br} = \left(\frac{P}{A_p}\right)\left(\frac{p}{t}\right) \text{ or } \sigma_{br} = \frac{4P}{\pi\left(d^2 - d_i^2\right)}\left(\frac{p}{t}\right)$$

Shear Stress (stripping): Two cases are possible

Case 1. The nut is weaker than the bolt material in shear (generally the case). The overload will strip the threads along the cylindrical surface A in the figure above.

Case 2. The bolt material is weaker in shear than the nut material. The failure will occur along surface B in the figure above.

The shear area is generally taken to be $A_s \approx \pi d\,(0.75t)$, where t is the nut thickness and d is the major diameter.

The bolt tensile strength and thread-stripping strength are roughly the same if the nut thickness is $t \approx 0.47d$. The nut is generally softer than the bolt. In order to allow for *slight* yielding of the top threads and thus distribute the load more uniformly among the threads in contact, the standard nut thickness is approximated by

$$t = \frac{7}{8}d$$

Simple Shear: Let j = a fraction of the nut thickness. For simple shear

$$\tau = \frac{P}{A} = \frac{P}{\pi d_r (1.5d)\, j}$$

For Square threads: $j = \dfrac{0.5p}{p} = 0.5$

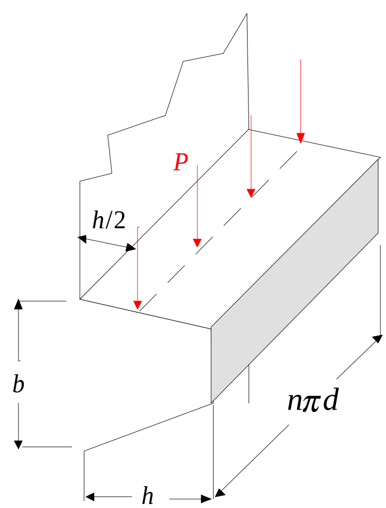
For Acme threads: $j = \dfrac{0.5p + 0.3p\tan 14.5^{o}}{p} = 0.5776$

For Acme Stub threads: $j = \dfrac{0.5p + 0.5p\tan 14.5^{o}}{p} = 0.629$

For Modified Square threads: $j = \dfrac{0.5p + 0.5p\tan 2.5^{o}}{p} = 0.5218$

Thread Stresses

The stresses in the threads are estimated by assuming the thread is a short cantilever beam projecting from the root cylinder of the fastener. The total screw load P is assumed to be concentrated at the mean radius.



The normal bending stress at the root is given by

$$\sigma_b = \frac{Mc}{I}$$

where $c = b/2$, $M = Ph/2$, $I = \pi n \dfrac{db^3}{12}$

n = number of thread turns subjected to load.

Substituting

$$\sigma_b \cong \frac{(Ph/2)(b/2)}{\dfrac{\pi n d b^3}{12}} = \frac{3Ph}{\pi n d b^2}$$

For a square thread, $h = p/2$ and $b = p/2$. Therefore

$$\sigma_b \cong \frac{6P}{\pi n d p}$$