Mechanical Design II

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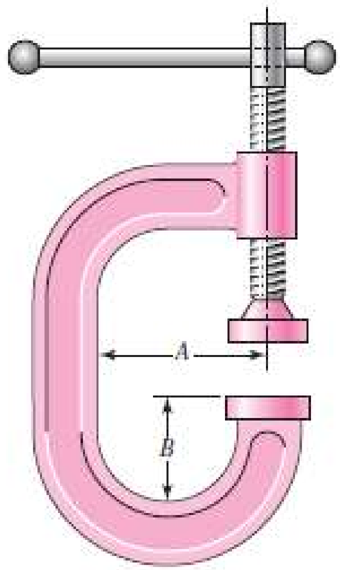
Lecture # 3

Fasteners and Joints…

- Threaded Fasteners…

8-8 The C clamp shown in the figure for Prob. 8–7 uses a $\frac{5}{8}$ in-6 Acme thread. The frictional coefficients are 0.15 for the threads and for the collar. The collar, which in this case is the anvil striker's swivel joint, has a friction diameter of $\frac{7}{16}$ in. Calculations are to be based on a maximum force of 6 lbf applied to the handle at a radius of $2\frac{3}{4}$ in from the screw centerline. Find the clamping force.



Problem 8–7

Home work

8-9 Find the power required to drive a 40-mm power screw having double square threads with a pitch of 6 mm. The nut is to move at a velocity of 48 mm/s and move a load of $F = 10$ kN. The frictional coefficients are 0.10 for the threads and 0.15 for the collar. The frictional diameter of the collar is 60 mm.

8-10 A single square-thread power screw has an input power of 3 kW at a speed of 1 rev/s. The screw has a diameter of 36 mm and a pitch of 6 mm. The frictional coefficients are 0.14 for the threads and 0.09 for the collar, with a collar friction radius of 45 mm. Find the axial resisting load F and the combined efficiency of the screw and collar.

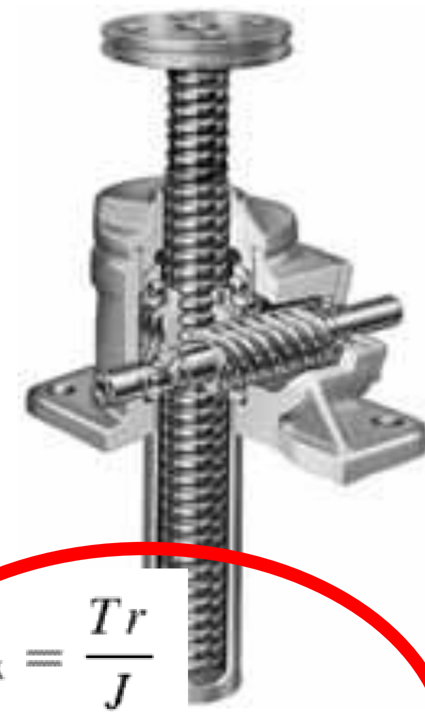
Stresses In power Screws

Nominal Body stresses in power screws can be related to thread parameters as follows

$$\tau = 16T/\pi d_r^3$$

The axial stress σ in the body of the screw due to load F is

$$\sigma = F/A = 4F/\pi d_r^2$$

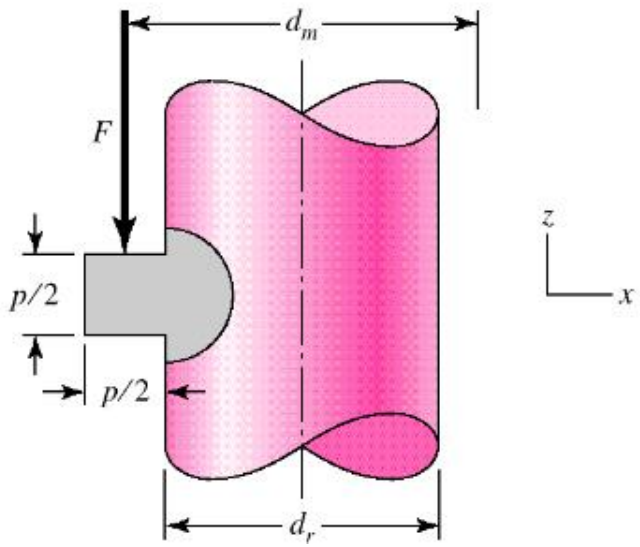


$$\tau_{max} = \frac{Tr}{J}$$

$$J = \pi D^4 / 32$$

Nominal thread Stresses in power Screws can be related to thread parameters as follows

The bearing stress results from the crushing force between the screw surface and the adjacent nut surface developed by lifting and supporting the load F.



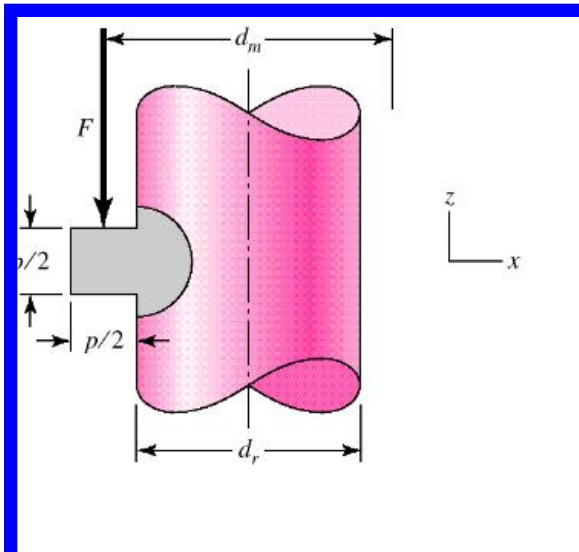
The bearing Stress σ_B is

$$\sigma_B = \frac{F}{\pi\, d_m n_t p/2} = \frac{2F}{\pi\, d_m n_t p}$$

Where n_t is the number of engaged threads

Geometry of a Square Thread useful in finding bending and transverse Shear Stresses at the thread root (Shigley's BOOK)

The maximum bending stress occurs at the root of the thread. It is calculated by assuming the thread is a simple cantilever beam built in at the root. The load is assumed to act at mid point on the thread.



The bending stress σ_b at the root of the thread is found from,

$$\frac{I}{c} = \frac{(\pi\, d_r n_t)(p/2)^2}{6}$$

$$M = Fp/4$$

$$\sigma_b = \frac{M}{I/c} \;\Rightarrow\; \sigma_b = \frac{Fp}{4}\,\frac{24}{\pi\, d_r n_t p^2} = \frac{6F}{\pi\, d_r n_t p}$$

The section under bending has a length = $\pi . d_r . n_t$

The width of the section at the thread root = p/2.

The Moment of Inertia at the root $I = \pi . d_r . n_t . (p/2)^3 / 12$

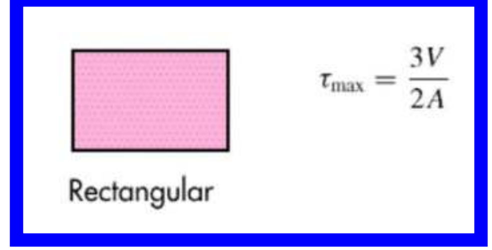
The distance from the centroid the the most remote fibre . = p/4.

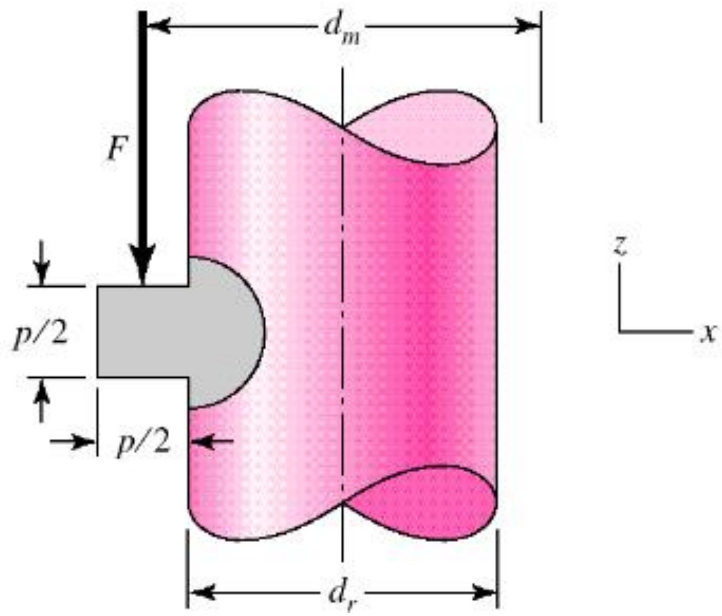
The Bending Moment M = F.p/4

The maximum bending stress is therefore..

$$\sigma_b = \frac{Fp}{4} \frac{24}{\pi d_r n_t p^2} = \frac{6F}{\pi d_r n_t p}$$

Bending Stress





stress τ at the centre of the root of the thread due to load *F* is

$$\tau = \frac{3V}{2A} = \frac{3}{2}\,\frac{F}{\pi d_r n_t p}$$

$$=> \frac{F}{\pi d_r n_t p}$$

And at the top of the root it is zero

Example 8.1

A square- Thread power screw has a major diameter of 32mm and a pitch of 4mm with double threads, and it is to be used in an application similar to that in the figure. The given data include $f=f_c$ =0.08, d_c=40mm, and F=6.4kN per screw..

1. Find the threads depth,thread width,pitch diamter,minor diameter and lead.
2. Find the torque required to raise and lower the load.
3. Find the efficiency during lifting the load.
4. Find the body stresses,torsional and compressive.
5. Find the bearing stress.
6. Find the thread stresses bending at the root ,shear at the root, and von mises stress and maximum shear stress at the same location.

Solution continue

(*d*) The body shear stress τ due to torsional moment T_R at the outside of the screw body is

Answer
$$\tau = \frac{16T_R}{\pi d_r^3} = \frac{16(26.18)(10^3)}{\pi(28^3)} = 6.07 \text{ MPa}$$

The axial nominal normal stress σ is

Answer
$$\sigma = -\frac{4F}{\pi d_r^2} = -\frac{4(6.4)10^3}{\pi(28^2)} = -10.39 \text{ MPa}$$

(*e*) The bearing stress σ_B is, with one thread carrying $0.38F$,

Answer
$$\sigma_B = -\frac{2(0.38F)}{\pi d_m(1)p} = -\frac{2(0.38)(6.4)10^3}{\pi(30)(1)(4)} = -12.9 \text{ MPa}$$

(*f*) The thread-root bending stress σ_b with one thread carrying $0.38F$ is

$$\sigma_b = \frac{6(0.38F)}{\pi d_r (1) p} = \frac{6(0.38)(6.4)10^3}{\pi(28)(1)4} = 41.5 \text{ MPa}$$

The transverse shear at the extreme of the root cross section due to bending is zero. However, there is a circumferential shear stress at the extreme of the root cross section of the thread as shown in part (*d*) of 6.07 MPa. The three-dimensional stresses, after Fig. 8–8, noting the y coordinate is into the page, are

$$\sigma_x = 41.5 \text{ MPa} \qquad \tau_{xy} = 0$$

$$\sigma_y = 0 \qquad \tau_{yz} = 6.07 \text{ MPa}$$

$$\sigma_z = -10.39 \text{ MPa} \qquad \tau_{zx} = 0$$

Equation (5–14) of Sec. 5–5 can be written as

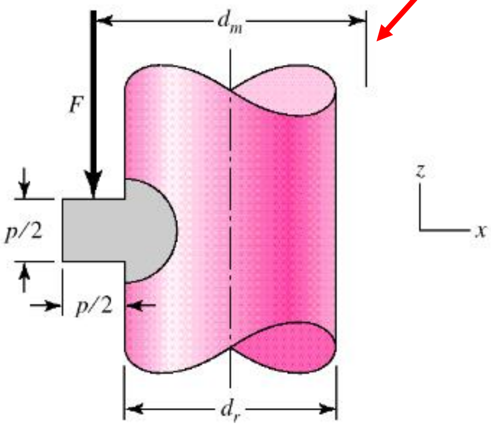
Answer

$$\sigma' = \frac{1}{\sqrt{2}}\{(41.5-0)^2 + [0-(-10.39)]^2 + (-10.39-41.5)^2 + 6(6.07)^2\}^{1/2}$$

$$= 48.7 \text{ MPa}$$

The von Mises stress σ' at the top of the root "plane"

is found by first identifying the orthogonal normal stresses and the shear stresses. From the coordinate system of Fig.



$$\sigma_x \quad \frac{6F}{\pi d_r n_t p} \qquad \tau_{xy} = \frac{16T}{\pi d_r^3}$$

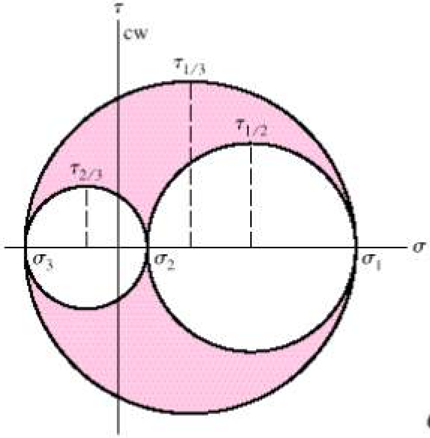
$$\sigma_y = 0 \qquad \tau_{yz} = 0$$

$$\sigma_z = -\frac{4F}{\pi d_r^2} \qquad \tau_{zx} = 0$$

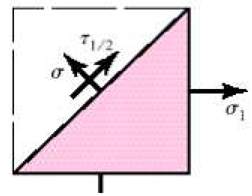
$$\sigma' = \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}\right]^{1/2}$$

Therefore yielding is predicted to occur when

$$\sigma' \geq S_y$$



(a)



$$\sigma^3 - (\sigma_x + \sigma_y + \sigma_z)\sigma^2 + (\sigma_x\sigma_y + \sigma_x\sigma_z + \sigma_y\sigma_z - \tau_{xy}^2 - \tau_{yz}^2 - \tau_{zx}^2)\sigma - (\sigma_x\sigma_y\sigma_z + 2\tau_{xy}\tau_{yz}\tau_{zx} - \sigma_x\tau_{yz}^2 - \sigma_y\tau_{zx}^2 - \sigma_z\tau_{xy}^2) = 0$$

$$\sigma' = \frac{1}{\sqrt{2}}\left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6\left(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2\right)\right]^{1/2}$$

and for plane stress

$$\sigma' = \left(\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2\right)^{1/2}$$

For the biaxial stress state, let σ_A and σ_B be the two nonzero principal stresses. Then, from Eq. (6–7), we get

$$\sigma' = \left(\sigma_A^2 - \sigma_A\sigma_B + \sigma_B^2\right)^{1/2} \tag{6–9}$$

Alternatively, one can recognize that this is a plane-stress situation, find the nonzero principal stresses, and find the von Mises stress using Eq.

For biaxial stress states in which σ_1 and σ_3 have like signs, the internal-friction hypothesis is the same as the maximum-normal-stress hypothesis and failure is predicted by

$$\sigma_1 = S_t \qquad \sigma_1 > 0$$
$$\sigma_3 = -S_c \qquad \sigma_3 < 0$$

The screw-thread form is complicated from an analysis viewpoint. Remember the origin of the tensile-stress area A_t which comes from experiment. A power screw lifting a load is in compression and its threat pitch is *shortened* by elastic deformation. Its engaging nut is in tension and its thread pitch is *lengthened*. The engaged threads cannot share the load equally. Some experiments show that the first engaged thread carries 0.38 of the load, the second 0.25, the third 0.18, and the seventh is free of load. In estimating thread stresses using the equations above, substituting $0.38F$ for F and setting n_t to 1 will give the largest level of stresses in the thread-nut combination.