Mechanical Design II

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Lecture # 2

Fasteners and Joints…

- Threaded Fasteners…

The primary function of a fastener is to connect two pieces together and transfer load. There are many types of fasteners. Examples of some requirements for fasteners are listed below:

- Higher strength
- Increased high temperature dependability
- Increased low temperature dependability
- Reduced cost
- Easier maintenance
- Improved corrosion resistance

Types of Joints

Permanent Joints=> welds, rivets, adhesives etc

Non Permanent Joints=>Nut and Bolts, Screws, pins, cotters, flanges etc.

The choice of a fastener is dependent on the design requirements and environment in which the fastener will be used. Attention to various aspects of the fastener must be considered. Some of these are listed below:

- Function of the fastener
- Operating environment of the fastener
- Type of loading on the fastener in service
- Thickness of materials to be joined
- Type of materials to be joined
- Configuration of the joint to be fastened

Mechanical Fasteners

Mechanical fasteners are frequently grouped as listed below:

- Keys and Pins fasteners
- Threaded fasteners
- Rivets
- Blind fasteners
- Adhesives
- Spring retainers
- Locking devices
- Special purpose fasteners

Failure Origins

Frequent locations for fastener failure listed below:

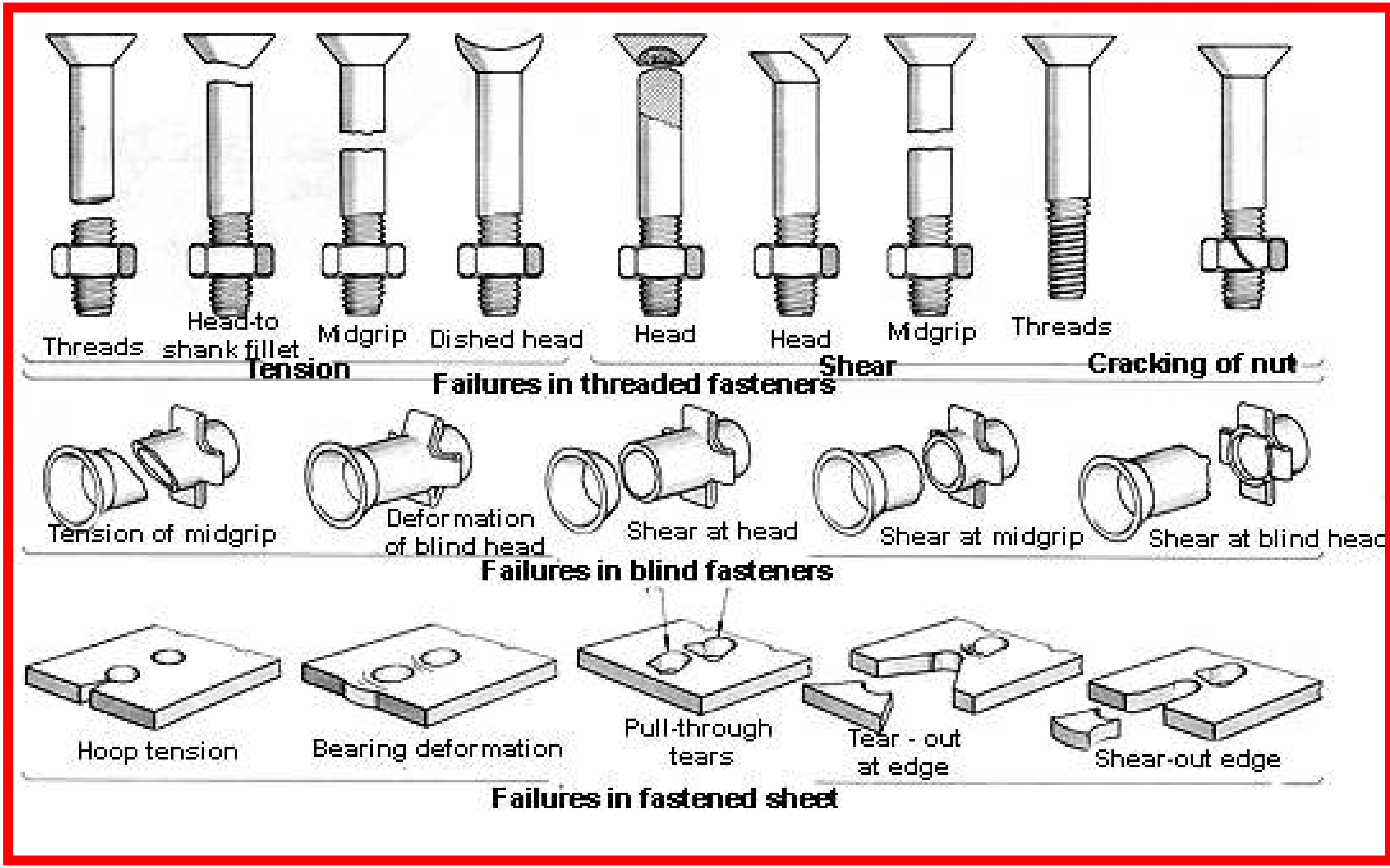
- In the head-to-shank fillet
- Through the first thread inside the nut
- The transition from the thread to the shank

Causes of Fastener Failures

Some causes of fastener failure are listed below:

- Shear
- Overload
- Fatigue
- Corrosion
- Manufacturing discrepancies
- Improper installation

common types of failures in fasteners.



One method of guarding against failure of a new fastener in a critical application is to sufficiently test the fastener system prior to use.

Threaded and Unthreaded Fasteners

Nuts and Bolts,
Screws, studs etc

Keys, cotters, pins,etc

Thread

A thread is a continuous helical ridge formed on the inside (nut) or outside (screw) of a cylinder.

This ridge is called the **crest**.
Between each crest is a space, called the **root**.

Threads are set at an angle to the axis of the bolt or nut.

This slope is called the **helix angle**.

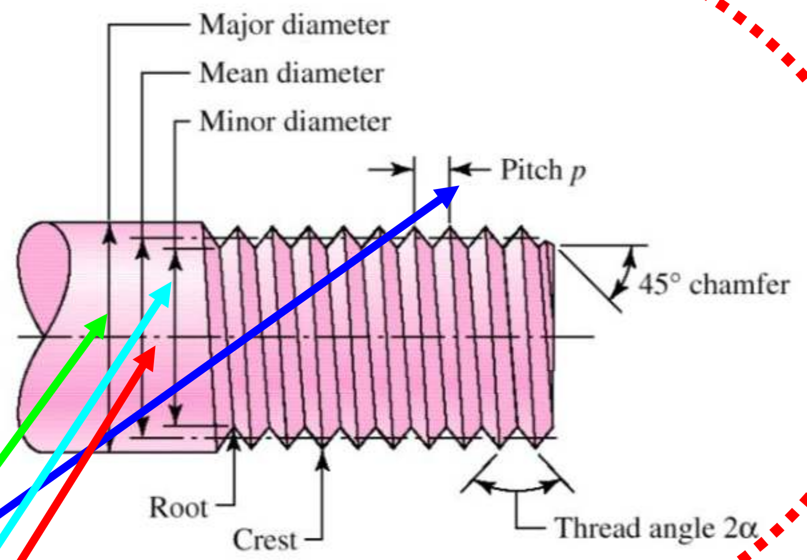
The angle must be sloped, either upward to the right (for right-hand threaded screws) or upward to the left (for left-hand threaded screws).

The thread forms a "V" shape between crests.

The angle of this "V" is called the **thread angle**, and is determined by fastener engineers.

Most screw threads used on a bicycle use a 60-degree thread angle.

Thread nomenclature



- *Pitch*, denoted by p, is the distance, parallel to the screw axis, between corresponding points on adjacent thread forms having uniform spacing.
- *Major diameter*, denoted by d, is the largest (outside) diameter of a screw thread.
- *Minor diameter*, denoted by d_r or d_1, is the smallest diameter of a screw thread.
- *Pitch diameter*, denoted by d_m or d_2, is the imaginary diameter for which the widths of the threads and the grooves are equal.

The lead *l* is the distance the nut moves parallel to the screw axis when the nut is given one turn. For a single thread, as in Fig. the lead is the same as the pitch.

A *multiple-threaded* product is one having two or more threads cut beside each other (imagine two or more strings wound side by side around a pencil).

Standardized products such as screws, bolts, and nuts all have single threads; a *double-threaded* screw has a lead equal to twice the pitch, a *triple-threaded* screw has a lead equal to 3 times the pitch, and so on.

(Shigly et al,2004)

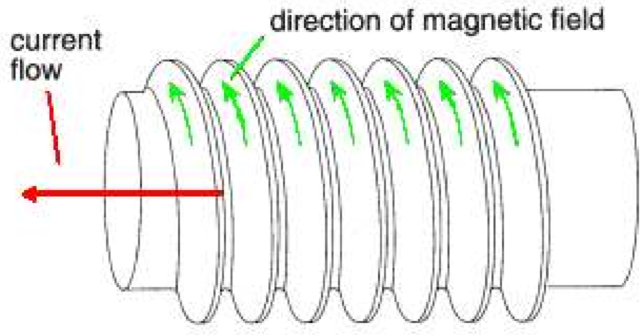
Left Hand / Right Hand Thread

Right-hand thread:
The common direction clockwise) which is used to secure a nut to a bolt

Left Handed /Right Handed Threads

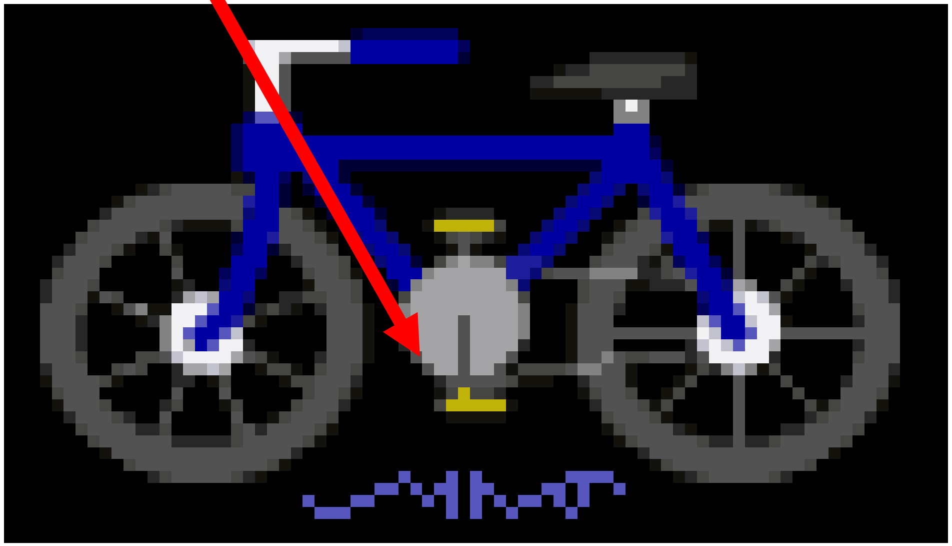
Most threads on bolts have right-hand thread meaning that turning the bolt clockwise inserts or tightens the bolt; counterclockwise loosens or removes the bolt.

But in left-hand thread the reverse is true. Turning the bolt clockwise loosens or removes the bolt.



Left-hand thread is used on applications where the normal turning motion of the object is such that there is a possibility that the object could back out and be removed

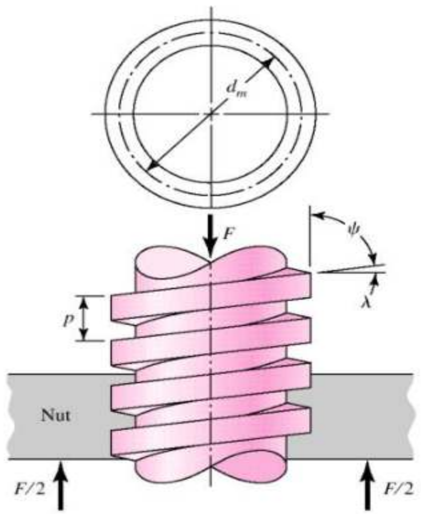
For instance, the left pedal of a bicycle turns around the crank arm in a counterclockwise direction. If it had right-hand thread, the pedal could fall off. Thus all left pedals have left hand thread so that the motion of the pedal helps to keep the pedal tight on the crank arm.

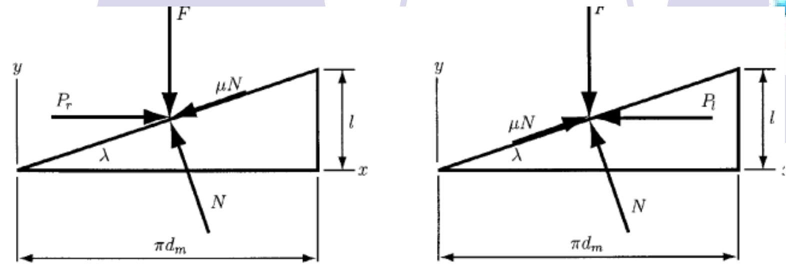


Last Lecture Recap

Lathes, Screw Jack etc

Power Screws





(a) (b)

$$T_r = P_r \frac{d_m}{2} = \frac{Fd_m}{2}\left(\frac{l + \pi\mu d_m}{\pi d_m - \mu l}\right).$$

$$T_l = \frac{Fd_m}{2}\left(\frac{\pi\mu d_m - l}{\pi d_m + \mu l}\right).$$

Self Locking Condition

$$\pi\mu d_m > l.$$

$$\mu > \tan\lambda.$$

Power Screws….

An expression for efficiency is also useful in the evaluation of power screws. If we let $f = 0$ we obtain

$$T_0 = \frac{Fl}{2\pi}$$

which, since thread friction has been eliminated, is the torque required only to raise the load. The efficiency is therefore

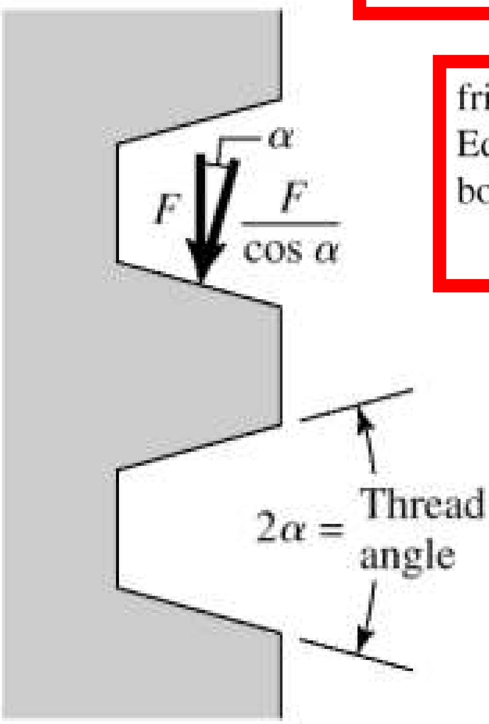
$$e = \frac{T_0}{T} = \frac{Fl}{2\pi T}$$

Note: f is the coefficient of friction same a μ

Text book is using f so that's why I am switching to this

For ACME Threads

In the case of Acme or other threads,
the normal thread load is inclined to the axis because of the thread angle 2α and the lead angle λ. Since lead angles are small, this inclination can be neglected and only the effect of the thread angle (Fig. 8–7*a*) considered. The effect of the angle α is to increase the

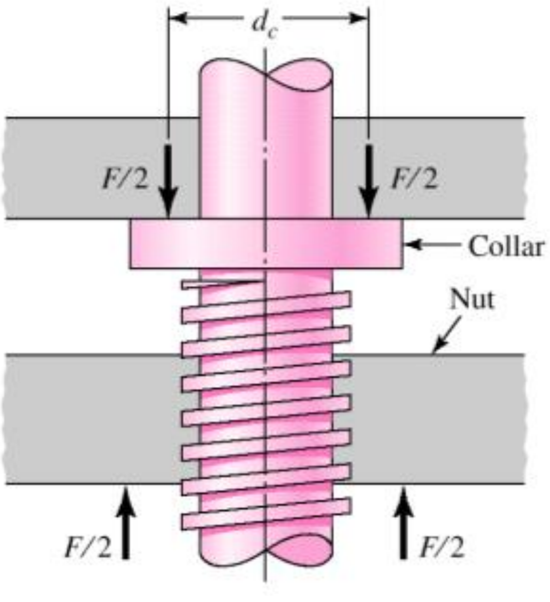


frictional force by the wedging action of the threads. Therefore the frictional terms in Eq. (8–1) must be divided by $\cos\alpha$. For raising the load, or for tightening a screw or bolt, this yields

$$T = \frac{F d_m}{2}\left(\frac{l + \pi f d_m \sec\alpha}{\pi d_m - f l \sec\alpha}\right)$$

remember that it is an approximation because the effect of the lead angle has been neglected.

Usually a third component of torque must be applied in power-screw applications. When the screw is loaded axially, a thrust or collar bearing must be employed between the rotating and stationary members in order to carry the axial component.



$$T_c = \frac{F f_c d_c}{2}$$

Chapter 8, Problem 4.

A single-threaded 25-mm power screw is 25 mm in diameter with a pitch of 5 mm. A vertical load on the screw reaches a maximum of 6 kN. The coefficients of friction are 0.05 for the collar and 0.08 for the threads. The frictional diameter of the collar is 40 mm. find the overall efficiency and the torque to “raise” and “lower” the load.

Solution:

Given $F = 6$ kN, $l = 5$ mm, and $d_m = 22.5$ mm, the torque required to raise the load is found using Eqs. (8-1) and (8-6)

$$T_R = \frac{6(22.5)}{2}\left[\frac{5 + \pi(0.08)(22.5)}{\pi(22.5) - 0.08(5)}\right] + \frac{6(0.05)(40)}{2}$$
$$= 10.23 + 6 = 16.23 \text{ N} \cdot \text{m} \quad \textit{Ans.}$$

The torque required to lower the load, from Eqs. (8-2) and (8-6) is

$$T_L = \frac{6(22.5)}{2}\left[\frac{\pi(0.08)22.5 - 5}{\pi(22.5) + 0.08(5)}\right] + \frac{6(0.05)(40)}{2}$$
$$= 0.622 + 6 = 6.622 \text{ N} \cdot \text{m} \quad \textit{Ans.}$$

Since T_L is positive, the thread is self-locking. The efficiency is

Eq. (8-4): $$e = \frac{6(5)}{2\pi(16.23)} = 0.294 \quad \textit{Ans.}$$

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 $f = f_c = 0.08$, $d_c = 40$mm, 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.

$$T_r = P_r \frac{d_m}{2} = \frac{Fd_m}{2}\left(\frac{l + \pi\mu d_m}{\pi d_m - \mu l}\right).$$

- TR=Tr+Tc

$$T_c = \frac{F f_c d_c}{2}$$

- TL=

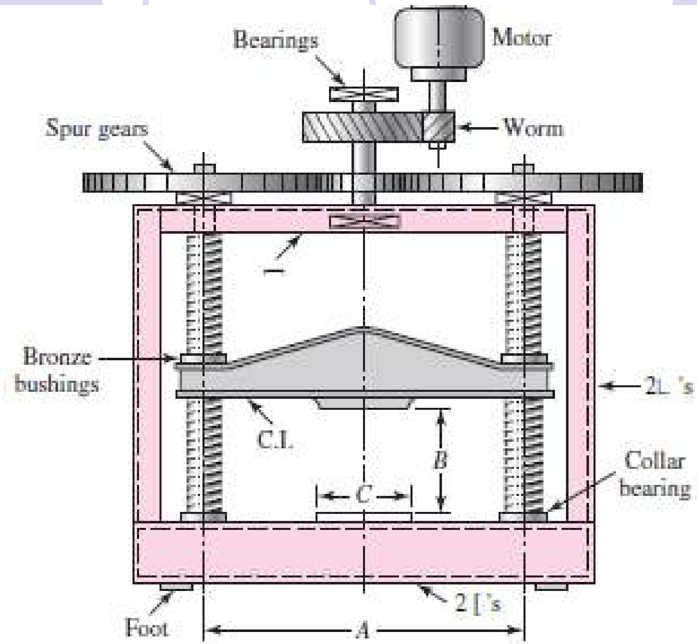
- e=Fl/(2*pi*TR)

$$T_l = \frac{Fd_m}{2}\left(\frac{\pi\mu d_m - l}{\pi d_m + \mu l}\right).$$

8–1 A power screw is 25 mm in diameter and has a thread pitch of 5 mm.

(*a*) Find the thread depth, the thread width, the mean and root diameters, and the lead, provided square threads are used.

(*b*) Repeat part (*a*) for Acme threads.



Problem 8–5

8–6 The press shown for Prob. 8–5 has a rated load of 5000 lbf. The twin screws have Acme threads, a diameter of 3 in, and a pitch of $\frac{1}{2}$ in. Coefficients of friction are 0.05 for the threads and 0.06 for the collar bearings. Collar diameters are 5 in. The gears have an efficiency of 95 percent and a speed ratio of 75:1. A slip clutch, on the motor shaft, prevents overloading. The full-load motor speed is 1720 rev/min.

(*a*) When the motor is turned on, how fast will the press head move?

(*b*) What should be the horsepower rating of the motor?

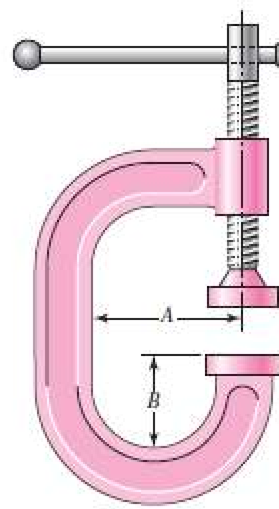
8–7 A screw clamp similar to the one shown in the figure has a handle with diameter $\frac{3}{16}$ in made of cold-drawn AISI 1006 steel. The overall length is 3 in. The screw is $\frac{7}{16}$ in-14 UNC and is $5\frac{3}{4}$ in long, overall. Distance A is 2 in. The clamp will accommodate parts up to $4\frac{3}{16}$ in high.

(*a*) What screw torque will cause the handle to bend permanently?

(*b*) What clamping force will the answer to part (*a*) cause if the collar friction is neglected and if the thread friction is 0.075?

(*c*) What clamping force will cause the screw to buckle?

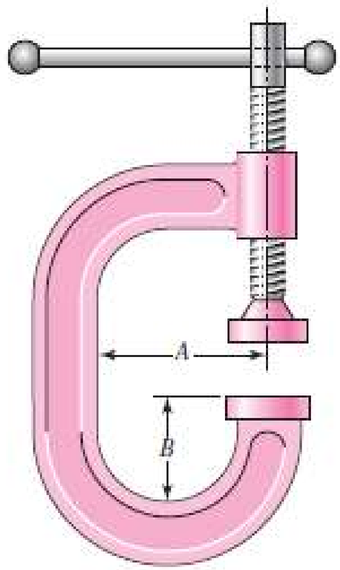
(*d*) Are there any other stresses or possible failures to be checked?



Problem 8–7

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



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.

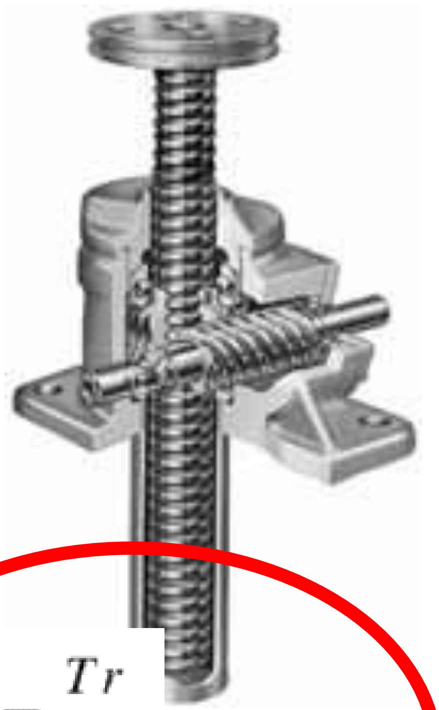
Stresses In power Screws

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

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

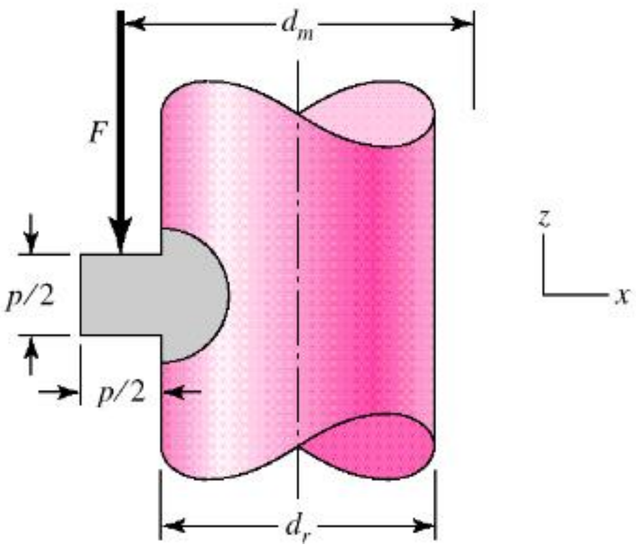


$$\tau_{max} = \frac{Tr}{J}$$

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

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.



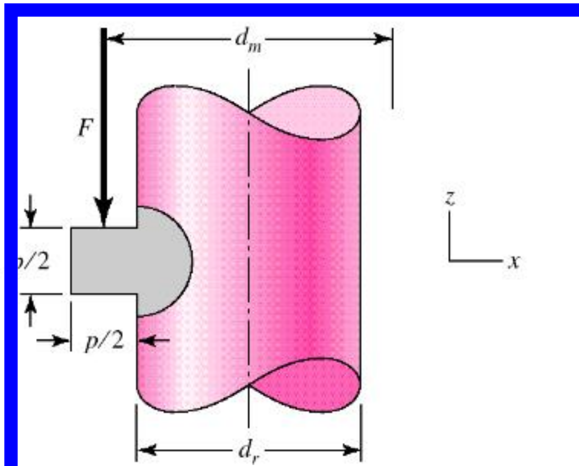
The bearing Stress σ_B is

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

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,

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

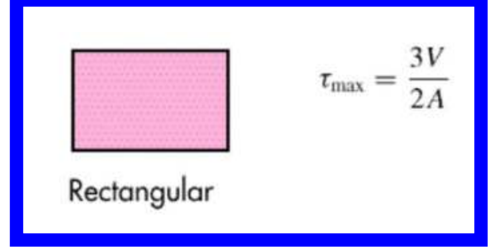
$$M = Fp/4$$

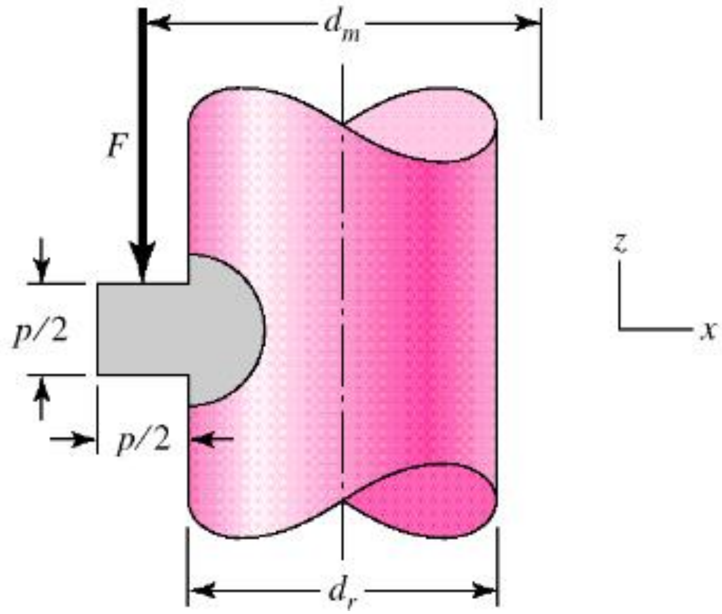
$$\sigma_b = \frac{M}{I/c} \Rightarrow \sigma_b = \frac{Fp}{4}\,\frac{24}{\pi\, d_r n_t p^2} = \frac{6F}{\pi\, d_r n_t p}$$

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

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

$$\Rightarrow \frac{F}{\pi d_r n_t p}$$

And at the top of the root it is zero

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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

$$\begin{aligned} \sigma_x &= 41.5 \text{ MPa} & \tau_{xy} &= 0 \\ \sigma_y &= 0 & \tau_{yz} &= 6.07 \text{ MPa} \\ \sigma_z &= -10.39 \text{ MPa} & \tau_{zx} &= 0 \end{aligned}$$

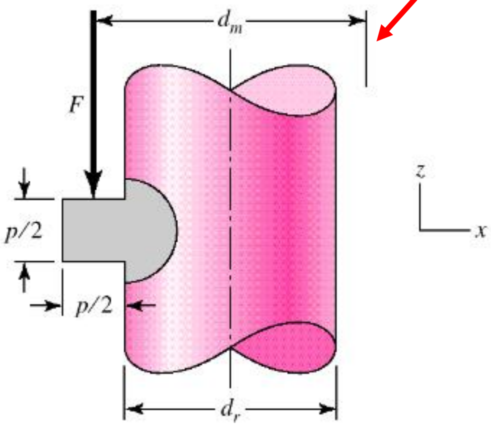
Equation (5–14) of Sec. 5–5 can be written as

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.



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

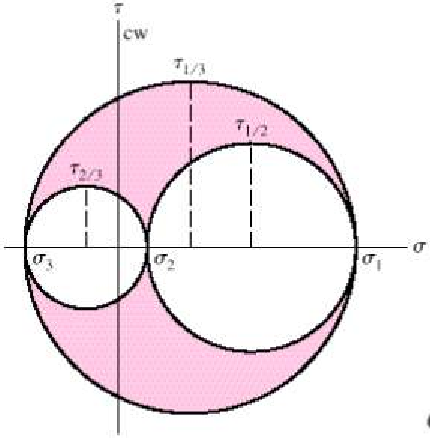
$$\sigma_y = 0 \qquad \tau_{yz} = 0$$

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

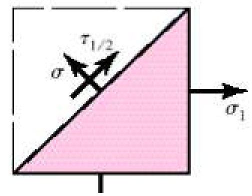
$$\sigma' = \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}\right]^{1/2}$$

Therefore yielding is predicted to occur when

$$\boxed{\sigma' \geq S_y}$$



(*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.