Power Screws

- Thread Standards and Definitions
- Stress Areas of Threads
- Mechanics of Power Screws (Square Threads)
- Self Locking Condition
- Power Screw Efficiency
- Power Screws with Acme Threads
- Presence of Collar Friction
- Body Stresses –Determine if it is safe for the Load

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Thread Standards and Definitions

- ***Pitch*** – distance between adjacent threads. Reciprocal of threads per inch
- ***Major diameter*** – largest diameter of thread
- ***Minor diameter*** – smallest diameter of thread
- ***Pitch diameter*** – theoretical diameter between major and minor diameters, where tooth and gap are same width

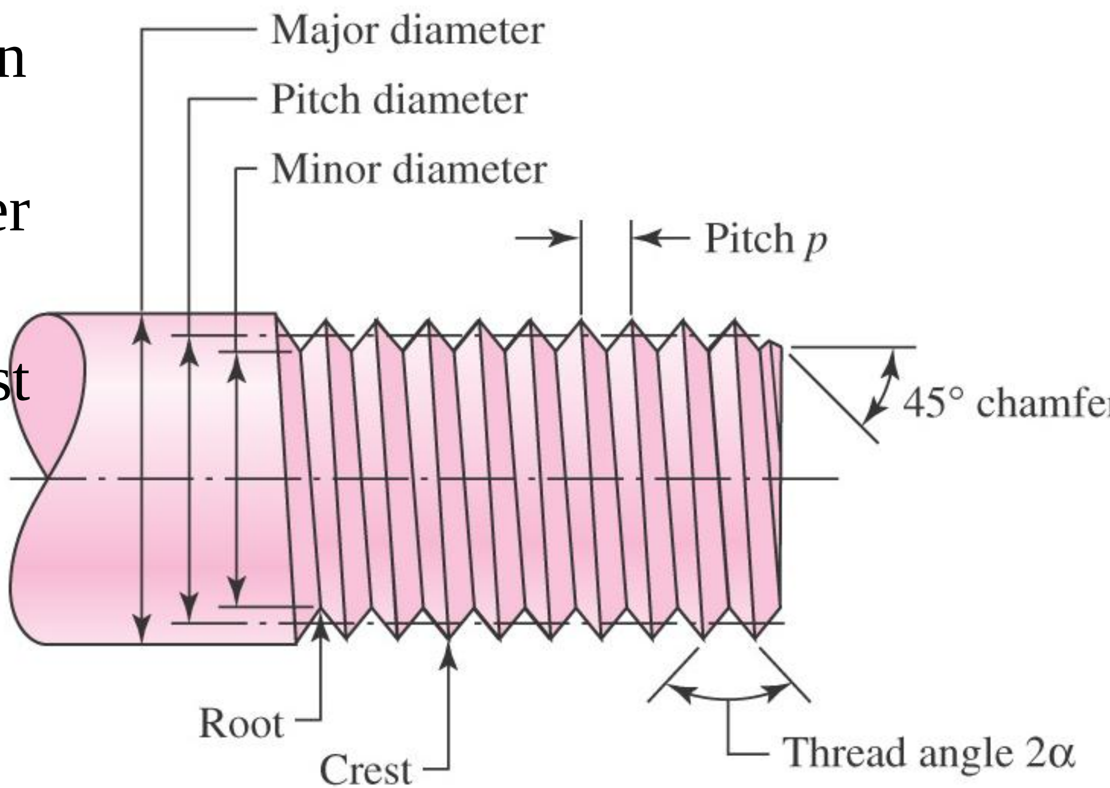

Fig. 8–1

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Standardization

- The *American National* (*Unified*) thread standard defines basic thread geometry for uniformity and interchangeability
- **American National (Unified) thread**
  - UN normal thread
  - UNR greater root radius for fatigue applications
- **Metric thread**
  - M series (normal thread)
  - MJ series (greater root radius)

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Standardization

- **Coarse series UNC**
  - General assembly
  - Frequent disassembly
  - Not good for vibrations
  - The “normal” thread to specify
- **Fine series UNF**
  - Good for vibrations
  - Good for adjustments
  - Automotive and aircraft
- **Extra Fine series UNEF**
  - Good for shock and large vibrations
  - High grade alloy
  - Instrumentation
  - Aircraft

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Standardization

- Basic profile for metric M and MJ threads shown in Fig. 8–2
- Tables 8–1 and 8–2 define basic dimensions for standard threads

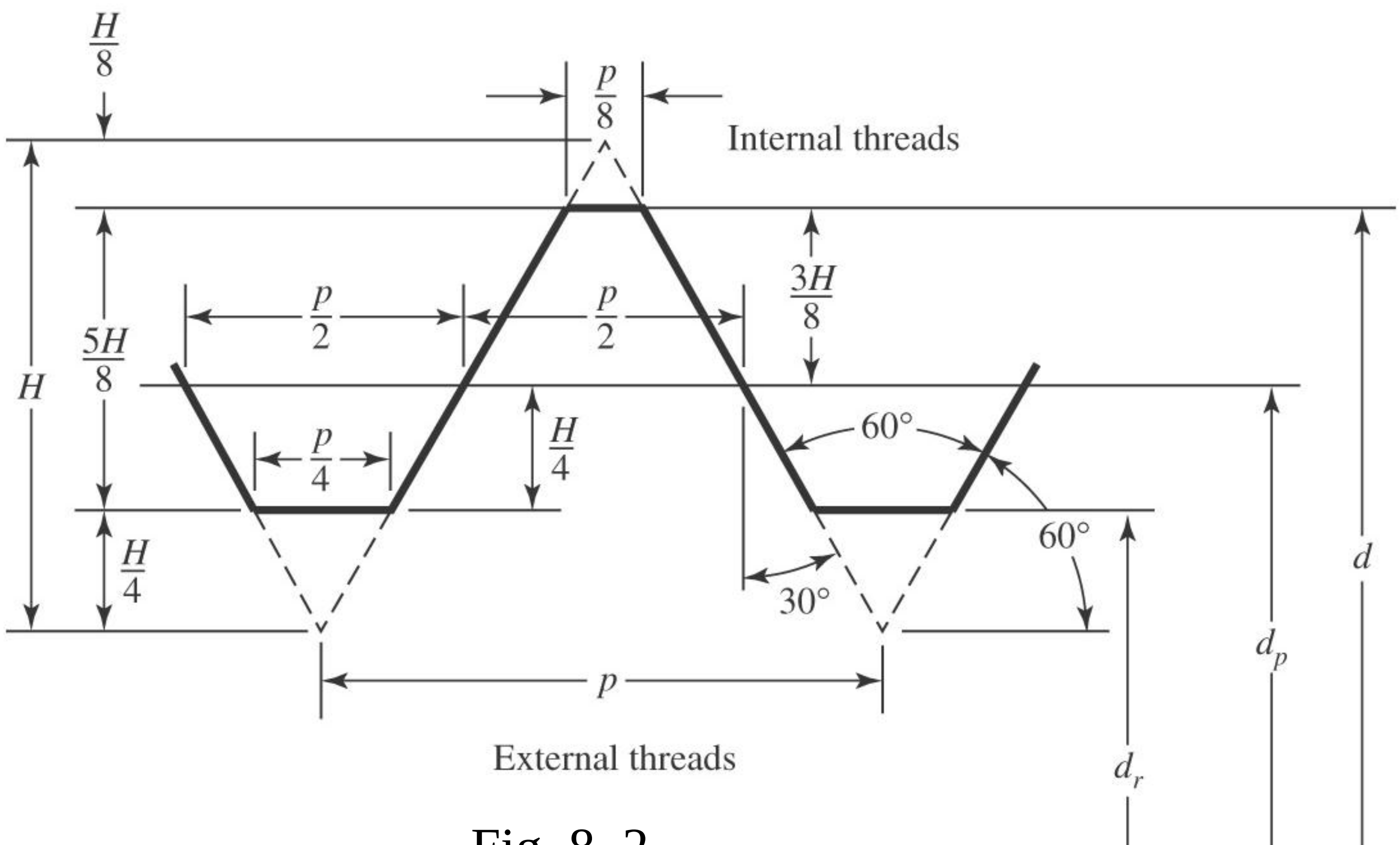

Fig. 8–2

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Diameters and Areas for Metric Threads

Table 8–1

Diameters and Areas of Coarse-Pitch and Fine-Pitch Metric Threads.*

Nominal Major Diameter d mm	Coarse-Pitch Series: Pitch p mm	Coarse-Pitch Series: Tensile-Stress Area A_t mm^2	Coarse-Pitch Series: Minor-Diameter Area A_r mm^2	Fine-Pitch Series: Pitch p mm	Fine-Pitch Series: Tensile-Stress Area A_t mm^2	Fine-Pitch Series: Minor-Diameter Area A_r mm^2
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980

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Diameters and Areas for Unified Screw Threads

Table 8–2

Size Designation	Nominal Major Diameter in	Coarse Series—UNC Threads per Inch N	Coarse Series—UNC Tensile-Stress Area A_t, in^2	Coarse Series—UNC Minor-Diameter Area A_r, in^2	Fine Series—UNF Threads per Inch N	Fine Series—UNF Tensile-Stress Area A_t, in^2	Fine Series—UNF Minor-Diameter Area A_r, in^2
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	0.373	0.351
$\frac{7}{8}$	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
$1\frac{1}{4}$	1.2500	7	0.969	0.890	12	1.073	1.024
$1\frac{1}{2}$	1.5000	6	1.405	1.294	12	1.581	1.521

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Tensile Stress Area

- The tensile stress area, A_t , is the area of an unthreaded rod with the same tensile strength as a threaded rod.
- **It is the effective area of a threaded rod to be used for stress calculations.**
- The diameter of this unthreaded rod is the average of the pitch diameter and the minor diameter of the threaded rod.

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Square and Acme Threads

- **Square and Acme threads are used when the threads are intended to transmit power**

Fig. 8–3

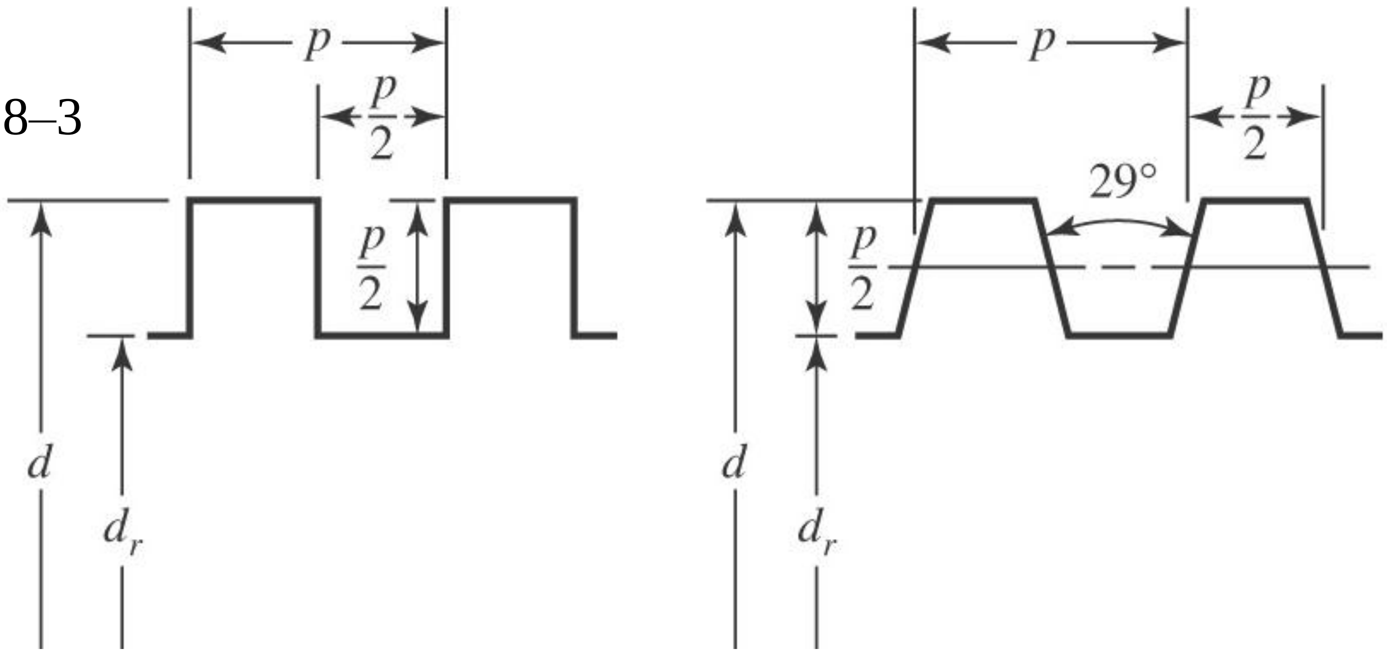

Table 8–3 Preferred Pitches for Acme Threads

d, in	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3
p, in	$\frac{1}{16}$	$\frac{1}{14}$	$\frac{1}{12}$	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{6}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$

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Mechanics of Power Screws

- ***Power screw***
  - Used to change angular motion into linear motion
  - Usually transmits power
  - Examples include vises, presses, jacks, lead screw on lathe

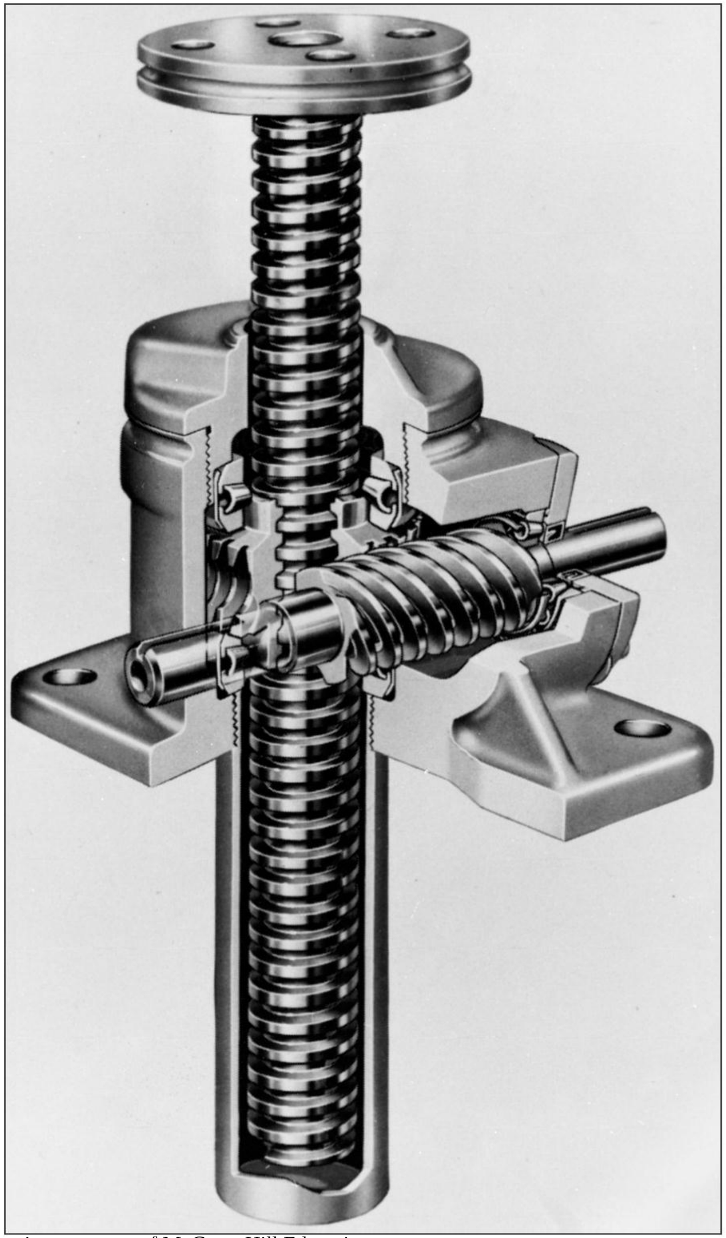

Fig. 8–4

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Mechanics of Power Screws

- Find expression for torque required to raise or lower a load
- Unroll one turn of a thread
- Treat thread as inclined plane
- Do force analysis

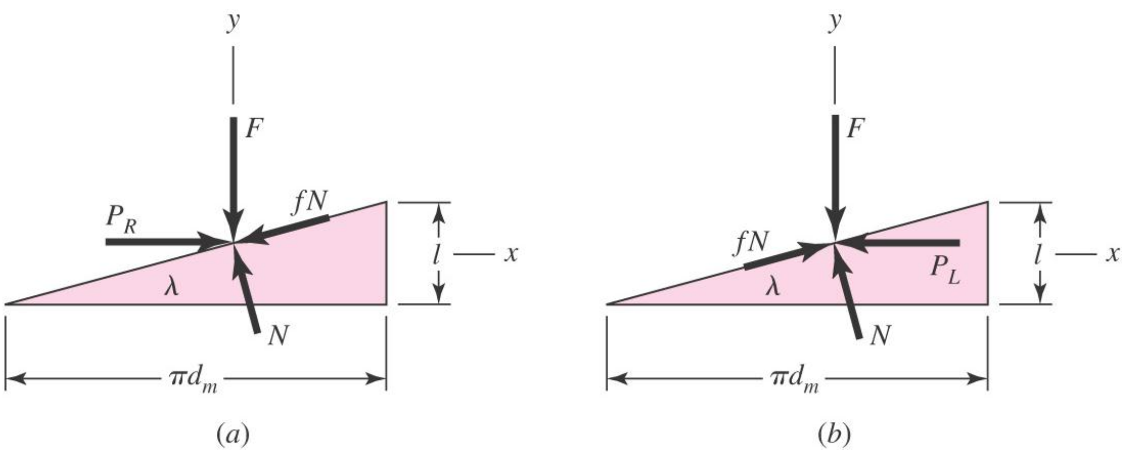

Fig. 8–6

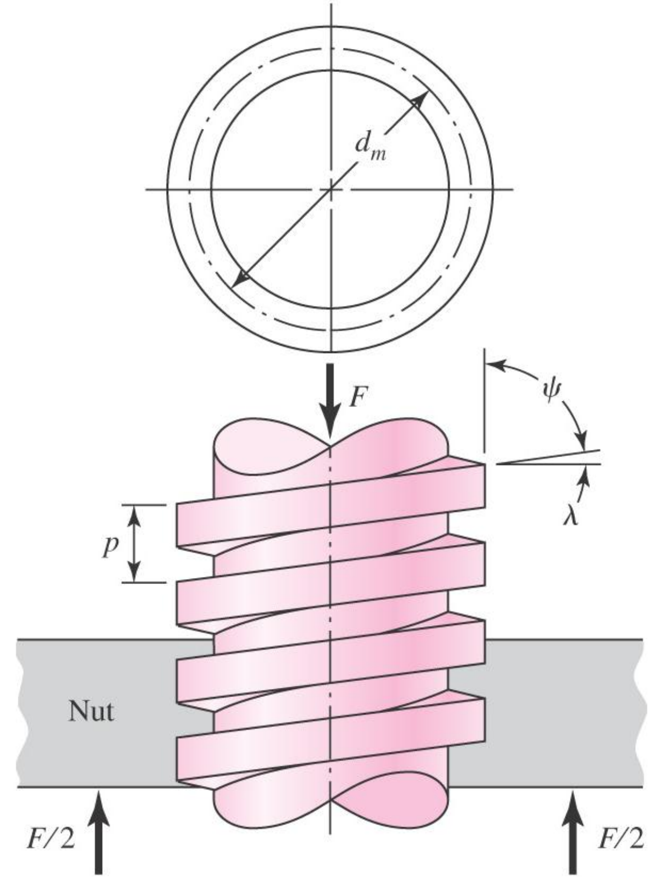

Fig. 8–5

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Mechanics of Power Screws

- **For raising the load**

$$\sum F_x = P_R - N\sin\lambda - fN\cos\lambda = 0$$

$$\sum F_y = -F - fN\sin\lambda + N\cos\lambda = 0$$

(a)

- **For lowering the load**

$$\sum F_x = -P_L - N\sin\lambda + fN\cos\lambda = 0$$

$$\sum F_y = -F + fN\sin\lambda + N\cos\lambda = 0$$

(b)

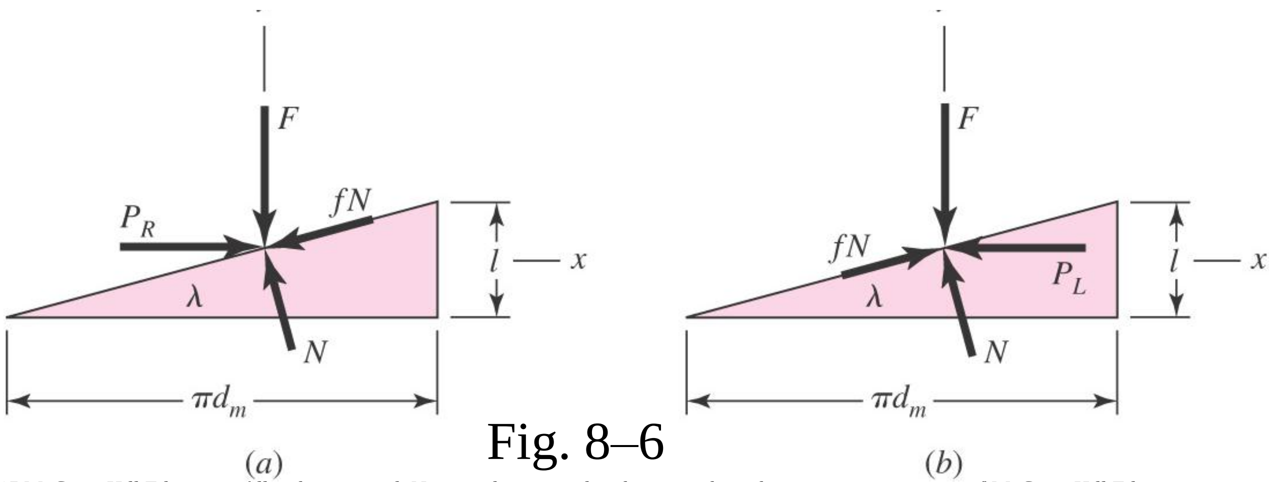

Fig. 8–6

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Mechanics of Power Screws

- Eliminate N and solve for P to raise and lower the load

$$P_R = \frac{F(\sin\lambda + f\cos\lambda)}{\cos\lambda - f\sin\lambda} \qquad (c)$$

$$P_L = \frac{F(f\cos\lambda - \sin\lambda)}{\cos\lambda + f\sin\lambda} \qquad (d)$$

- Divide numerator and denominator by $\cos\lambda$ and use relation **$\tan\lambda = l/\pi d_m$**

$$P_R = \frac{F[(l/\pi d_m) + f]}{1 - (fl/\pi d_m)} \qquad (e)$$

$$P_L = \frac{F[f - (l/\pi d_m)]}{1 + (fl/\pi d_m)} \qquad (f)$$

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Raising and Lowering Torque

- Noting that the torque is the product of the force and the mean radius,

$$T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) \tag{8–1}$$

$$T_L = \frac{F d_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + f l}\right) \tag{8–2}$$

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Self-locking Condition

$$T_L = \frac{F\,d_m}{2}\left(\frac{\pi f\,d_m - l}{\pi\,d_m + f\,l}\right) \tag{8–2}$$

- If the lowering torque is negative, the load will lower itself by causing the screw to spin without any external effort.
- If the lowering torque is positive, the screw is *self-locking*.
- **Self-locking condition is $\pi f\, d_m > l$**
- Noting that $l / \pi\, d_m = \tan \lambda$, the self-locking condition can be seen to only involve the coefficient of friction and the lead angle.

$$f > \tan\lambda \tag{8–3}$$

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Collar Friction

- An additional component of torque is often needed to account for the friction between a collar and the load.
- Assuming the load is concentrated at the mean collar diameter d_c

$$T_c = \frac{F f_c d_c}{2} \qquad (8\text{–}6)$$

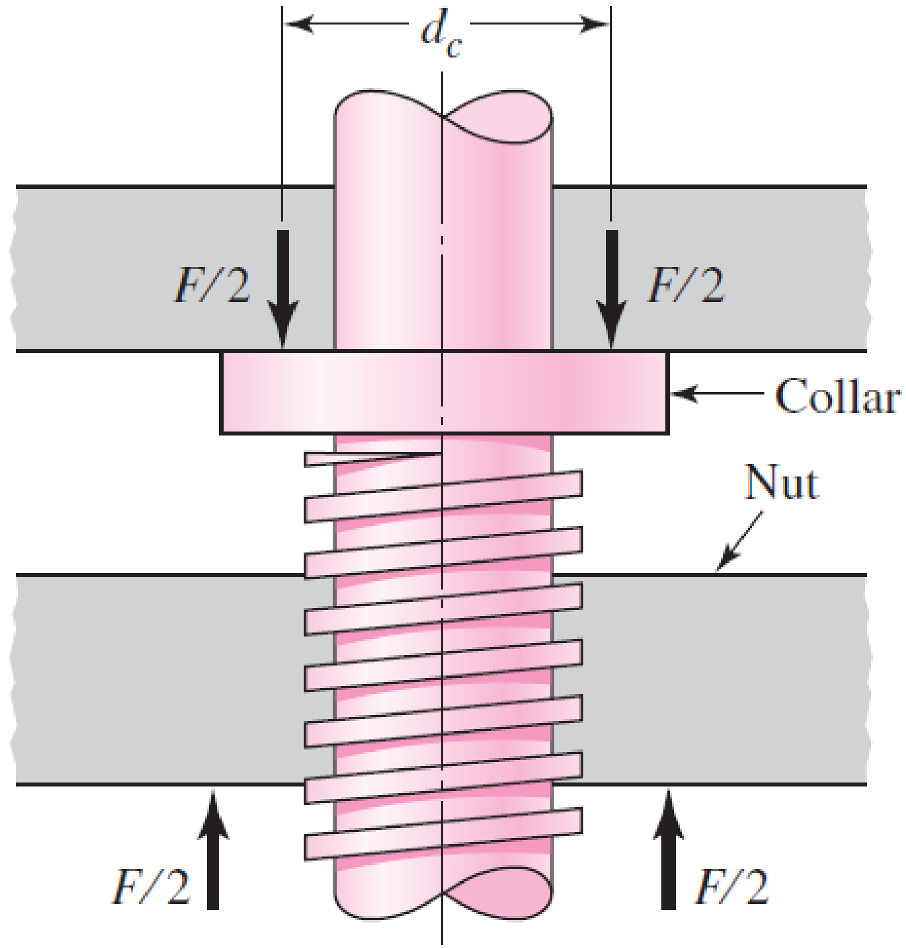

Fig. 8–7(b)

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Power Screw Efficiency

$$T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) \qquad (8\text{–}1)$$

- The torque needed to raise the load with no friction losses can be found from Eq. (8–1) with $f = 0$.

$$T_0 = \frac{F l}{2\pi} \qquad (g)$$

- The efficiency of the power screw is:

$$e = \frac{T_0}{T_R} = \frac{F l}{2\pi T_R} \qquad (8\text{–}4)$$

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Power Screws with Acme Threads

- If Acme threads are used instead of square threads, the thread angle creates a wedging action.
- The friction components are increased.
- The torque necessary to raise a load (or tighten a screw) is found by dividing the **friction terms** in Eq. (8–1) by cosα.

$$T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) \quad (8\text{–}1)$$

$$T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m \sec\alpha}{\pi d_m - f l \sec\alpha}\right) \quad (8\text{–}5)$$

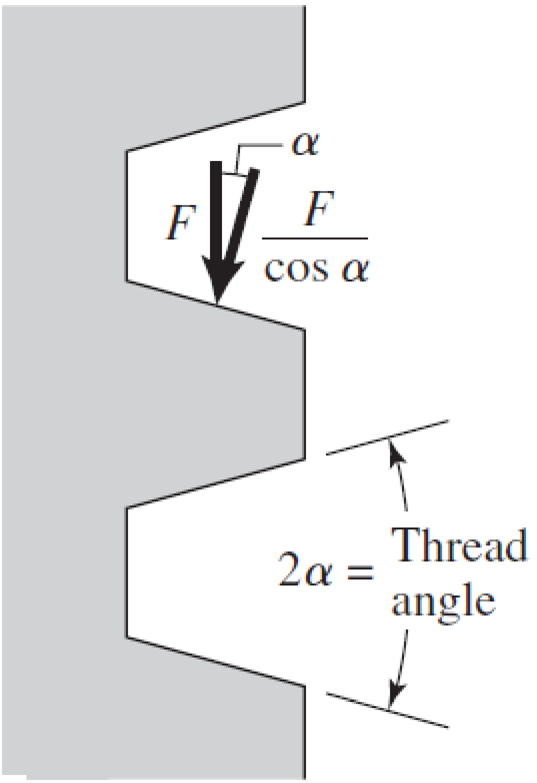

Fig. 8–7 (a)

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Stresses in Body of Power Screws

- Maximum nominal shear stress in torsion of the screw body

$$\tau = \frac{16T}{\pi d_r^3} \qquad (8\text{–}7)$$

- Axial stress in screw body

$$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2} \qquad (8\text{–}8)$$

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Stresses in Threads of Power Screws

- Bearing stress in threads,

$$\sigma_B = -\frac{F}{\pi d_m n_t p/2} = -\frac{2F}{\pi d_m n_t p} \qquad (8\text{–}10)$$

where n_t is number of engaged threads

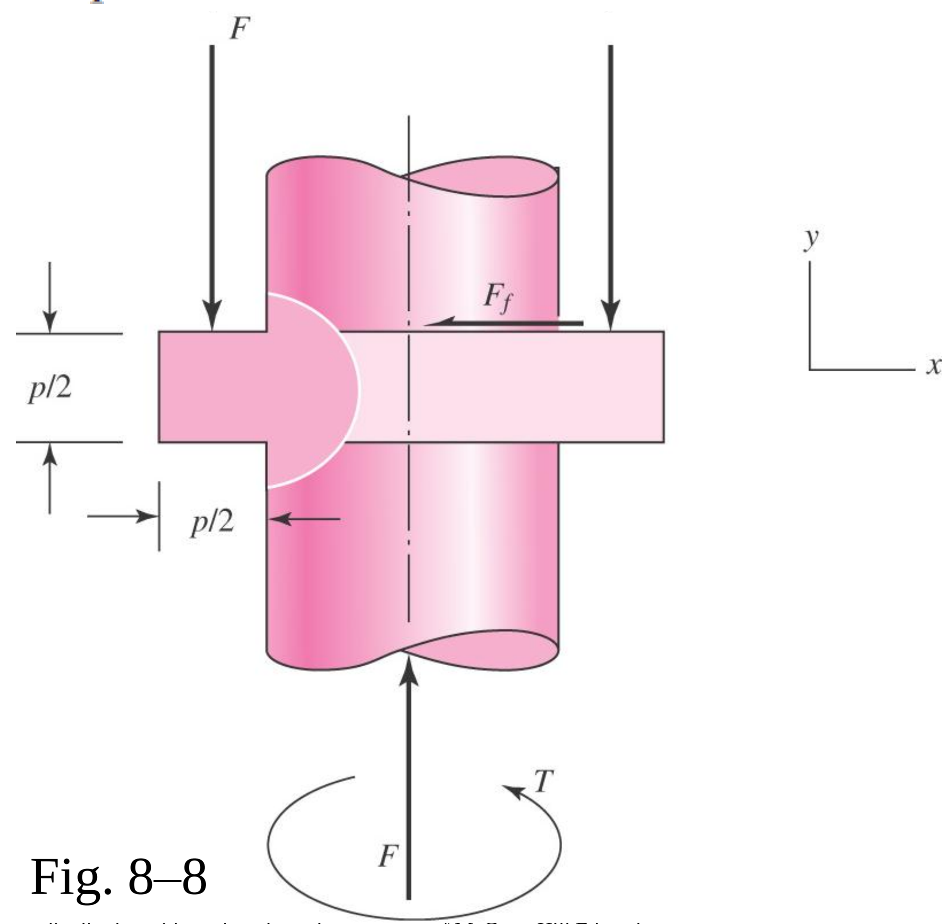

Fig. 8–8

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Stresses in Threads of Power Screws

- Bending stress at root of thread,

$$Z = \frac{I}{c} = \frac{(\pi d_r n_t)(p/2)^2}{6} = \frac{\pi}{24} d_r n_t p^2$$

$$M = \frac{Fp}{4}$$

$$\sigma_b = \frac{M}{Z} = \frac{Fp}{4} \frac{24}{\pi d_r n_t p^2} = \frac{6F}{\pi d_r n_t p} \qquad (8\text{–}11)$$

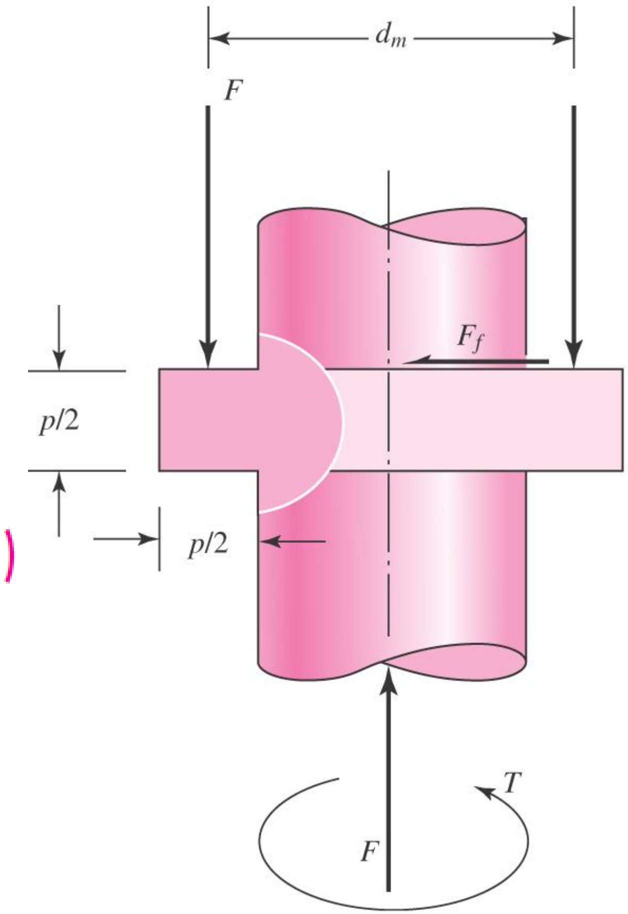

Fig. 8–8

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Stresses in Threads of Power Screws

- Transverse shear stress at center of root of thread,

$$\tau = \frac{3V}{2A} = \frac{3}{2}\frac{F}{\pi d_r n_t p/2} = \frac{3F}{\pi d_r n_t p} \qquad (8\text{–}12)$$

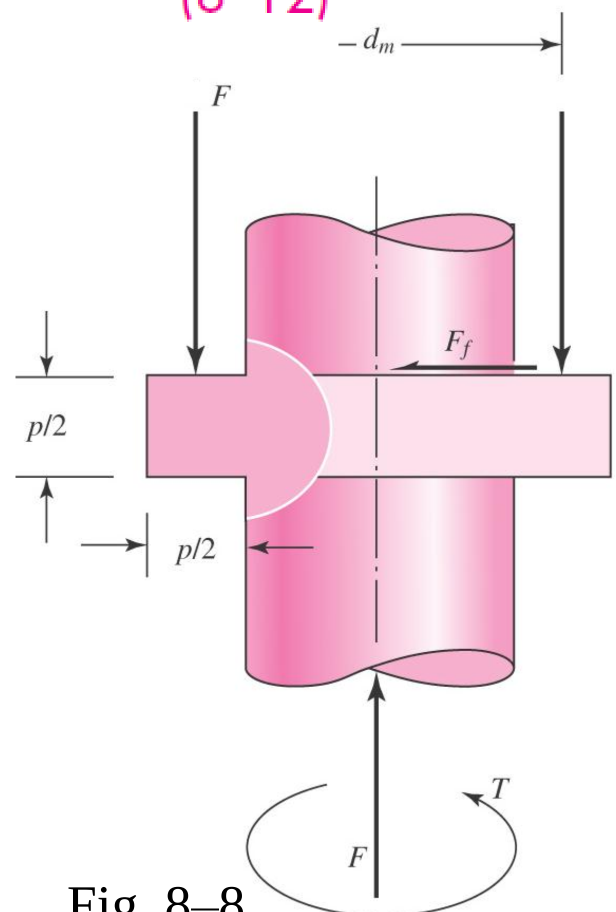

Fig. 8–8

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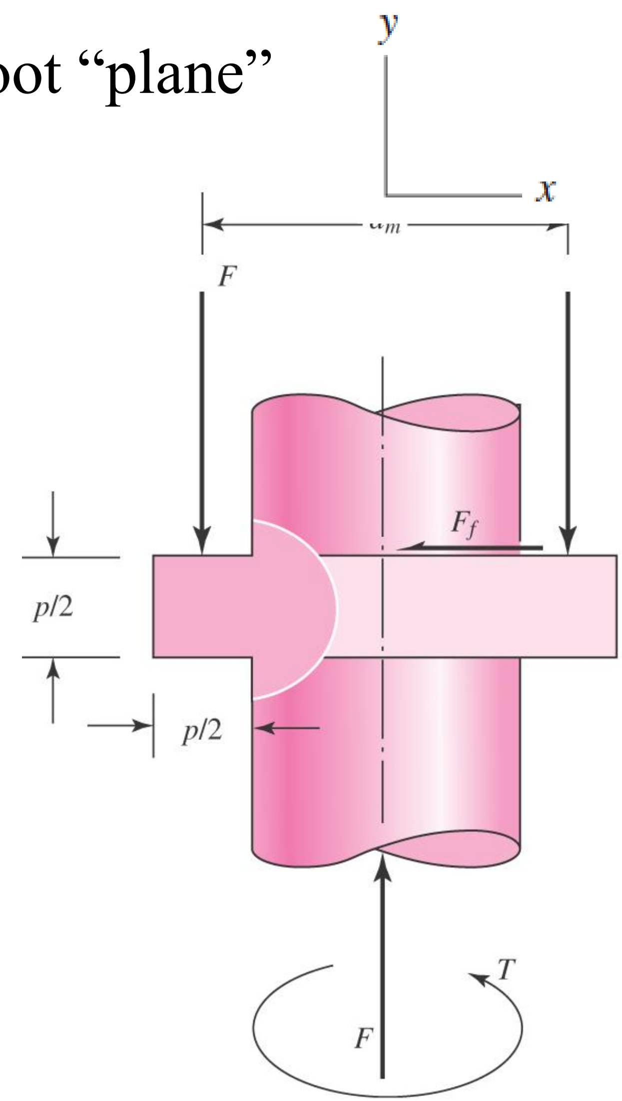
Stresses in Threads of Power Screws

- Consider stress element at the top of the root "plane"

$$\sigma_x = \frac{6F}{\pi d_r n_t p} \qquad \tau_{xy} = 0$$

$$\sigma_y = -\frac{4F}{\pi d_r^2} \qquad \tau_{yz} = \frac{16T}{\pi d_r^3}$$

$$\sigma_z = 0 \qquad \tau_{zx} = 0$$



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Stresses in Threads of Power Screws

Obtain von Mises stress from:

$$\sigma' = \frac{1}{\sqrt{2}}\left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6\left(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2\right)\right]^{1/2} \qquad (5\text{–}14)$$

Design Equation using Maximum Distortion Energy:

$$\mathbf{n = S_Y/\sigma^1}$$

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Thread Deformation in Screw-Nut Combination

- The engaged threads cannot share the load equally.
- **Experiments indicate the first thread carries 38% of the load, the second thread 25%, and the third thread 18%. The seventh thread is free of load.**
- **To find the largest stress in the first thread of a screw-nut combination, use 0.38F in place of F, and set n_t = 1.**

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Example-Power Screw

A square-thread power screw has a major diameter of 32 mm and a pitch of 4 mm with double threads, and it is to be used in an application similar to that in Fig. 8–4. The given data include $f = f_c = 0.08$, $d_c = 40$ mm, and $F = 6.4$ kN per screw.

(*a*) Find the thread depth, thread width, pitch diameter, minor diameter, and lead.

(*b*) Find the torque required to raise and lower the load.

(*c*) Find the efficiency during lifting the load.

(*d*) Find the body stresses, torsional and compressive.

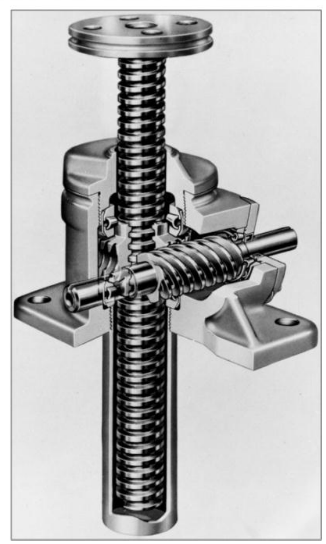
(*e*) Find the bearing stress.

(*f*) Find the thread bending stress at the root of the thread.

(*g*) Determine the von Mises stress at the root of the thread.

(*h*) Determine the maximum shear stress at the root of the thread.

Fig. 8–4



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Example -(continued)

(*a*) From Fig. 8–3*a* the thread depth and width are the same and equal to half the pitch, or 2 mm. Also

$$d_m = d - p/2 = 32 - 4/2 = 30 \text{ mm}$$ Answer

$$d_r = d - p = 32 - 4 = 28 \text{ mm}$$ Answer

$$l = np = 2(4) = 8 \text{ mm}$$ Answer

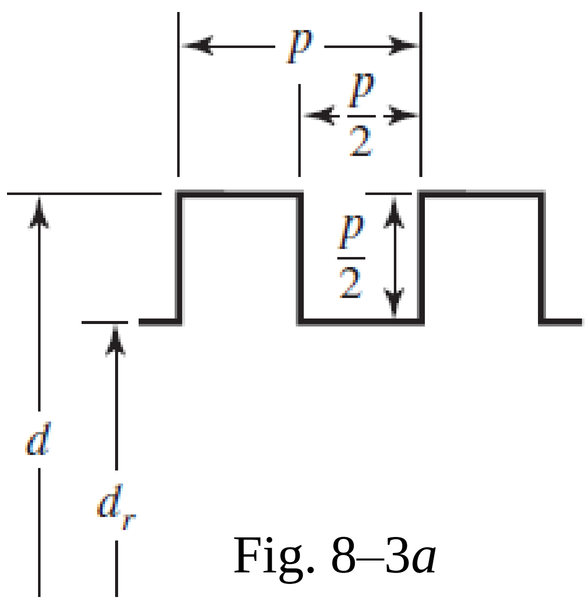

Fig. 8–3*a*

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Example - (continued)

(b) Using Eqs. (8–1) and (8–6), the torque required to turn the screw against the load is

$$T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) + \frac{F f_c d_c}{2}$$

$$= \frac{6.4(30)}{2}\left[\frac{8 + \pi(0.08)(30)}{\pi(30) - 0.08(8)}\right] + \frac{6.4(0.08)40}{2}$$

$$= 15.94 + 10.24 = 26.18\ \text{N} \cdot \text{m}$$ Answer

Using Eqs. (8–2) and (8–6), we find the load-lowering torque is

$$T_L = \frac{F d_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + f l}\right) + \frac{F f_c d_c}{2}$$

$$= \frac{6.4(30)}{2}\left[\frac{\pi(0.08)30 - 8}{\pi(30) + 0.08(8)}\right] + \frac{6.4(0.08)(40)}{2}$$

$$= -0.466 + 10.24 = 9.77\ \text{N} \cdot \text{m}$$ Answer

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Example - (continued)

(*c*) The overall efficiency in raising the load is

$$e = \frac{Fl}{2\pi T_R} = \frac{6.4(8)}{2\pi(26.18)} = 0.311 \qquad \text{Answer}$$

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Example - (continued)

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

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

The axial nominal normal stress σ is

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

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

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

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

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

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Example - (continued)

(*g*) 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

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

$$\sigma_y = -10.39 \text{ MPa} \qquad \tau_{yz} = 6.07 \text{ MPa}$$

$$\sigma_z = 0 \qquad \tau_{zx} = 0$$

For the von Mises stress, Eq. (5–14) of Sec. 5–5 can be written as

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

$$= 48.7 \text{ MPa}$$ Answer

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Power Screw – Assignment

A single-thread power screw is 25-mm in diameter and has thread pitch of 5-mm. A vertical load on the screw reached 5 kN.
The coefficients of friction are 0.06 for the collar and 0.09 for the threads. The frictional diameter of the collar is 45 mm. For a square thread, find:

a. Mean and root diameters
b. Determine if the screw will self-lock.
c. The torques require to raise the load and to lower the load
d. The lifting efficiency of the screw
e. The minimum yield strength of the screw material based on maximum distortion energy and a factor of safety of 5.0

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