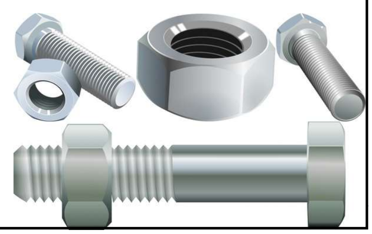
IPE 303
Product Design I

Chapter 8: Screws, Fasteners, and the Design of Nonpermanent Joints

R. M. SHAHBAB
Lecturer, Department of IPE, BUET
Email: sshahbab@ipe.buet.ac.bd



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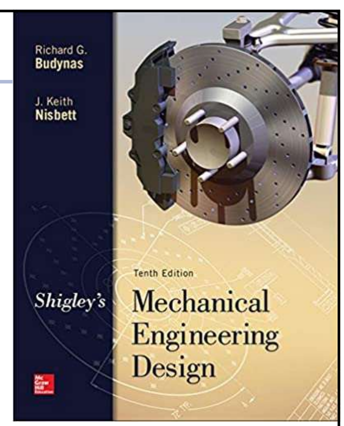

Reference

Shigley's Mechanical Engineering Design, 10th Edition

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Mechanism of Nut-Bolt and power screw

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Mechanism of Nut-Bolt and power screw

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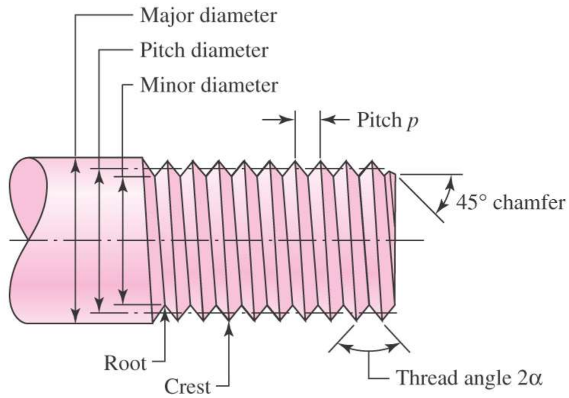
8-1 Thread Standards and Definitions

- ***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
- ***Lead, l*** – the distance the nut moves parallel to the screw axis when the nut is given one turn.
- **Lead = number of threads × pitch**



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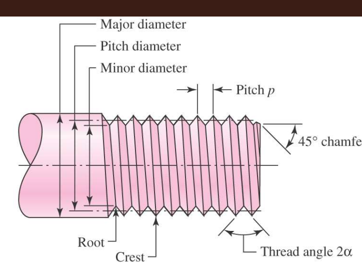
8-1 Thread Standards and Definitions



- Bolts are standardized and there are two standards –
- American (Unified) thread
  - **UN** - normal thread
  - **UNR**- greater filleted root radius for better fatigue strength
- Metric (ISO) thread
  - **M series** - normal thread
  - **MJ series** - rounded fillet at the root and a large minor diameter for better fatigue strength

In both standards the thread angle is 60°

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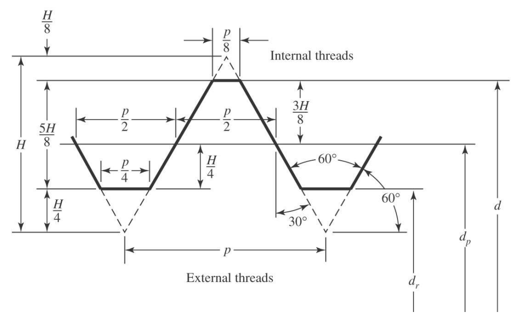
8-1 Thread Standards and Definitions

Metric (ISO):

- Two basic profiles for metric (ISO) **- M and MJ.**
- Specified by the **major diameter** and the **pitch** (both in mm).

  Example: $M10 \times 1.5$

  M profile — Major diameter — Pitch



- Table 8-1 gives the standard sizes along with **Tensile stress areas (A_t)** and **Root diameter areas (A_r)** (for shear loading) for Metric bolts.
- Note that, there is Coarse pitch and Fine pitch series.

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8-1 Thread Standards and Definitions

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 mm²	Coarse-Pitch Series: Minor-Diameter Area A_r mm²	Fine-Pitch Series: Pitch p mm	Fine-Pitch Series: Tensile-Stress Area A_t mm²	Fine-Pitch Series: Minor-Diameter Area A_r mm²
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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8-1 Thread Standards and Definitions

American (Unified)

- Specified by **major diameter** (in inch) and the **number of thread per inch** (N).

  Example: ${}^{1}/_{4} - 20\ UNC$

  major diameter

  number of thread per inch

  Coarse series UN Profile

Note: UNF = Fine series UN Profile.

- Table 8-2 gives the standard sizes along with **Tensile stress areas (A_t)** and **Root diameter areas (A_r)** (for shear loading) for UN bolts.
- Note that, for diameter ≤ ${}^{1}/_{4}$ inch, the size is designated by size numbers rather than diameter.

8-1 Thread Standards and Definitions

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

8-1 Thread Standards and Definitions

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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8-1 Thread Standards and Definitions

Square and Acme Threads

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

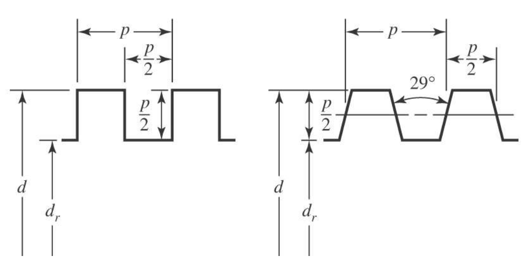

Table 8-3 Preferred Pitches for Acme Threads

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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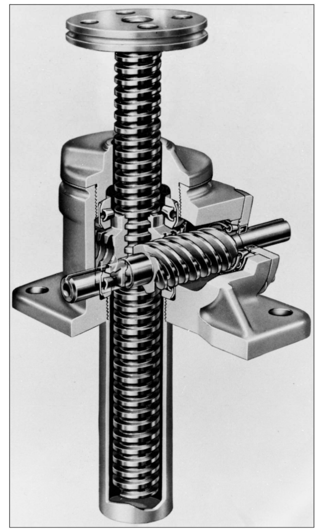
8-2 The Mechanics of Power Screws

Power screw

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



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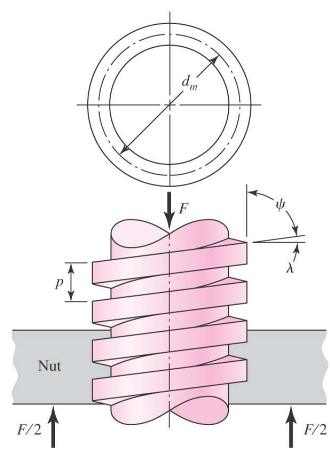
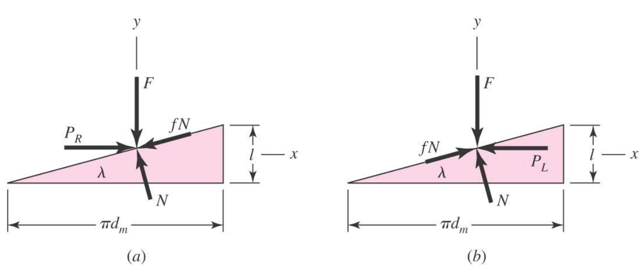
8-2 The Mechanics of Power Screws

- Find expression for **torque required to raise or lower** a load with **Square Thread**
- Find expression for **torque required to raise or lower** a load with **Acme Thread**
- Find expression for **Self-locking Condition**
- Find expression for **Power Screw Efficiency**
- Find expression for **Collar Friction**
- Find the **Stresses in Threads of Power Screws**





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8-2 The Mechanics of Power Screws

Raising and Lowering Torque

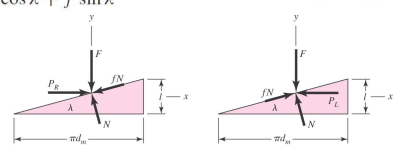
- Noting that the **torque is the product of the force and the mean radius**,

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

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

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

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



(a) (b)

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

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

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8-2 The Mechanics of Power Screws

Power Screws with Acme Threads

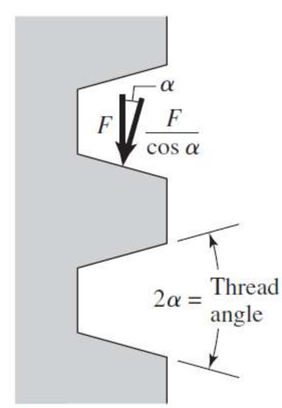
- If Acme threads are used instead of square threads, the **thread angle** creates a **wedging action**.
- The friction components are increased.
- The torque necessary to raise a load (or tighten a screw) is found by dividing the friction terms in Eq. (8–1) by cosα.

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

- For power screws, the Acme thread is not as efficient as the square thread, because of the additional friction
- However, it is often preferred because it is easier to machine and can be adjusted against wear.

Raising Torque for square threads

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



(a)

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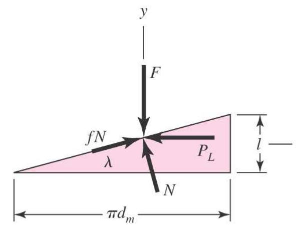
8-2 The Mechanics of Power Screws

Self-locking Condition

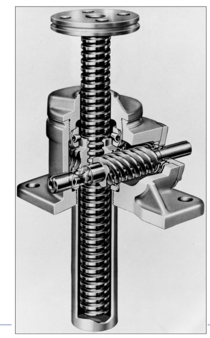
$$T_L = \frac{F\,d_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + f l}\right) \qquad (8\text{–}2)$$



(b)

- 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.

$$f > \tan\lambda \qquad (8\text{–}3)$$



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8-2 The Mechanics of Power Screws

Raising Torque for square threads

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

Power Screw Efficiency

- 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 therefore

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

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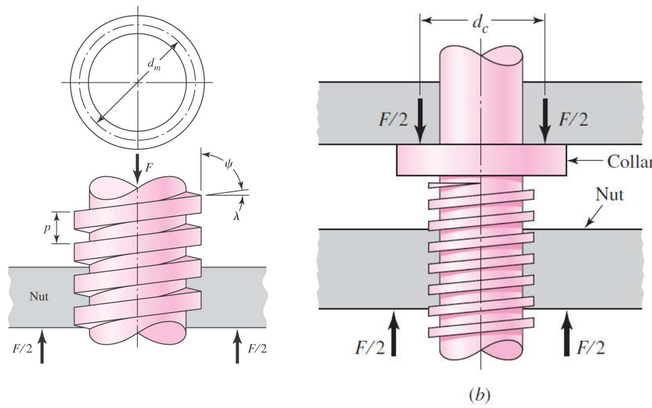
8-2 The Mechanics of Power Screws

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

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



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8-2 The Mechanics of Power Screws

Stresses in Body of Power Screws

- Maximum nominal shear stress in torsion of the screw body

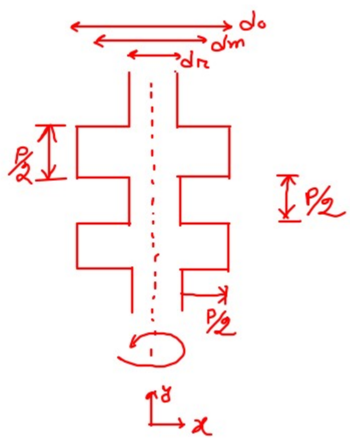
$$\tau = \frac{16T}{\pi d_r^3}$$

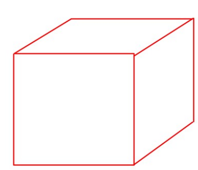
- Axial stress in screw body

$$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2}$$

- Bending stress at root of thread,

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





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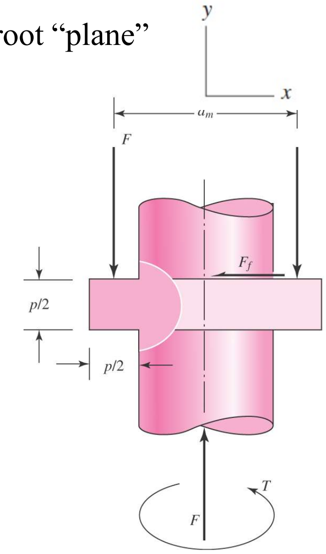
8-2 The Mechanics of Power Screws › Stresses in Body of Power Screws

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

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

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

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



- Obtain von Mises stress from Eq. (5–14),

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

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8-2 The Mechanics of Power Screws › Stresses in Body of Power Screws

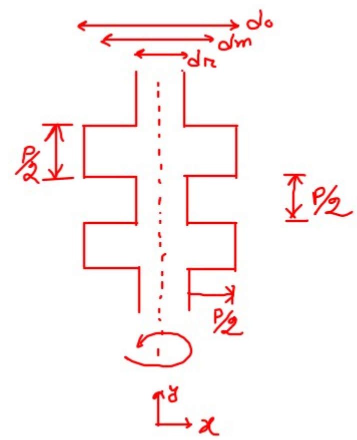
- Bearing stress in threads,

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

where n_t is number of engaged threads

- Transverse shear stress at center of root of thread,

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



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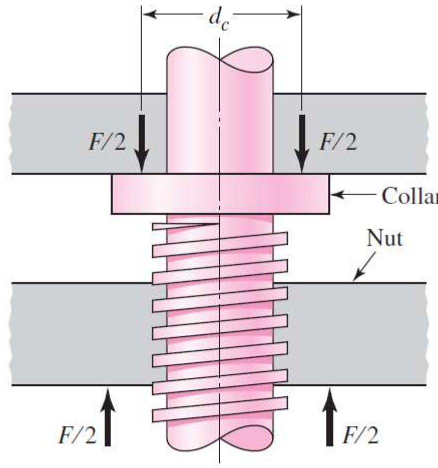
8-2 The Mechanics of Power Screws

Thread Load distribution

- 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 $n_t = 1$.



(b)

Example 8-1

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.

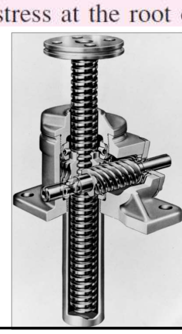


Fig. 8–4