MECH 344/M
Machine Element Design

Time: M _ _ _ _14:45 - 17:30

Lecture 6

Contents of today's lecture

10

Threaded Fasteners and Power Screws

Introduction

- Multitude of fasteners are available raging from nuts and bots to different varieties. Only a small sample is shown here
- Limit our discussion to design and selection of conventional fasteners (screws, nuts & bolts).
- Primarily used in machine design applications and lot of stresses are encountered.
- Used primarily for holding, or moving (lead screw)
- Loads are tensile, or shear or both
- The economic implications are tremendous.
- the airframe of a large jet aircraft has approximately 2.4 Million fasteners costing about $750,000 in 1978 dollars.

10.2 Thread Forms, Terminology, and Standards

- Figure 10.1 illustrates the basic arrangement of a helical thread wound around a cylinder, as used on screw-type fasteners, power screws, and worms.
- Pitch, lead, lead angle, and hand-of-thread are defined by the illustrations.
- Virtually all bolts and screws have a single thread, but worms and power screws sometimes have double, triple, and even quadruple threads.
- Unless otherwise noted, all threads are assumed to be right-hand.

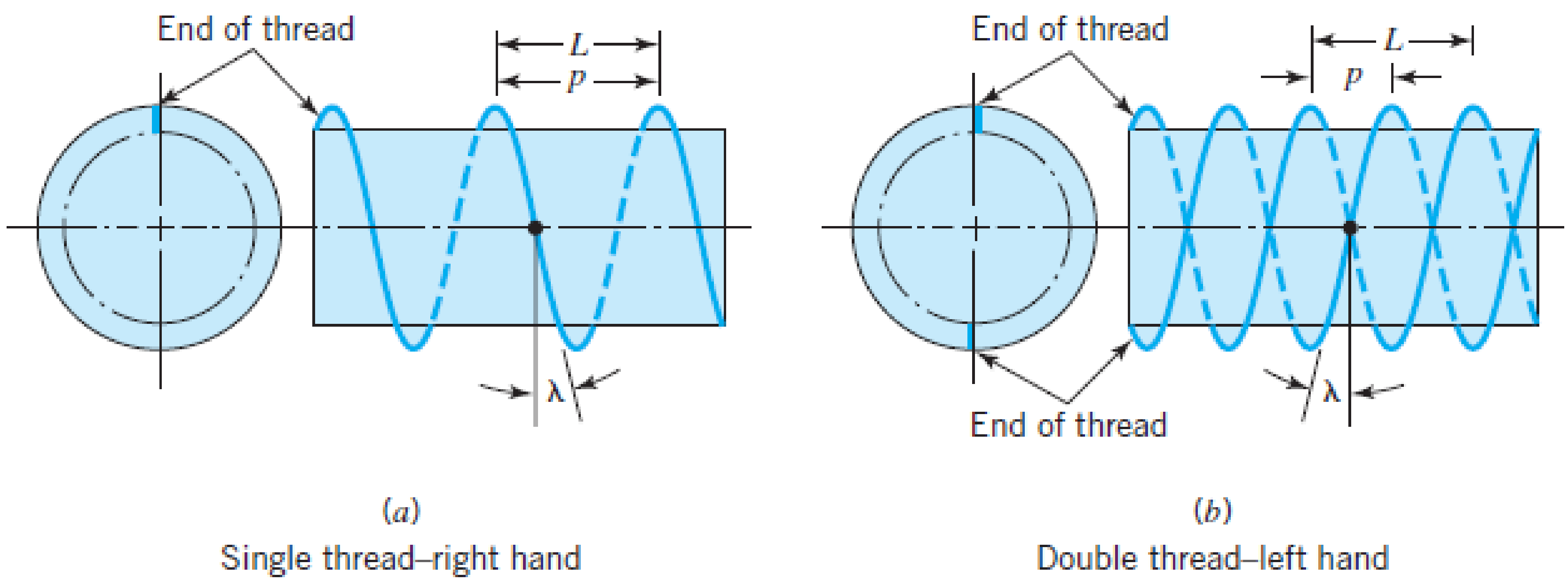


FIGURE 10.1
Helical threads of pitch p, lead L, and lead angle λ.

10.2 Thread Forms, Terminology, and Standards

- Figure 10.2 shows the standard geometry of screw threads used on fasteners.
- This is basically the same for both Unified (inch) and ISO (metric) threads.
- Standard sizes for the two systems are given in Tables 10.1 and 10.2.
- The pitch diameter, d_p, is the diameter of a cylinder on a perfect thread where the width of the thread and groove are equal.
- The stress area tabulated is based on the average of the pitch and root diameters.
- This is the area used for "P/A" stress calculations.
- It approximates the smallest possible fracture area, considering the presence of the helical thread.

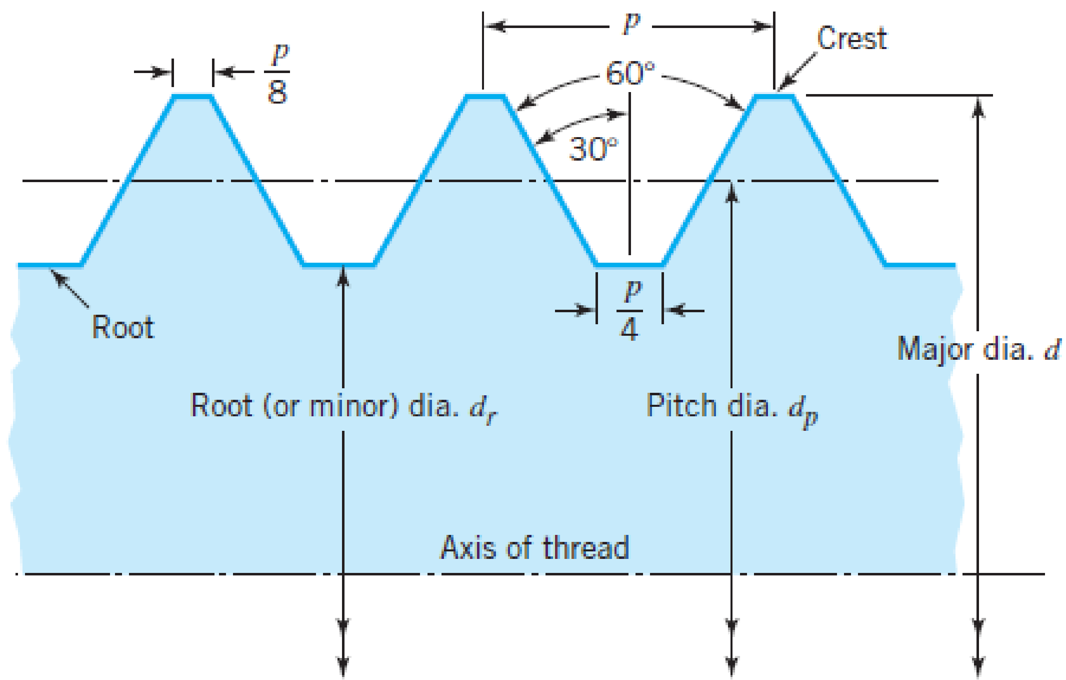


FIGURE 10.2
Unified and ISO thread geometry. The basic profile of the external thread is shown.

TABLE 10.1 Basic Dimensions of Unified Screw Threads

Size	Major Diameter d (in.)	Coarse Threads—UNC Threads per Inch	Coarse Threads—UNC Minor Diameter of External Thread d_r (in.)	Coarse Threads—UNC Tensile Stress Area A_t (in.^{2})	Fine Threads—UNF Threads per Inch	Fine Threads—UNF Minor Diameter of External Thread d_r (in.)	Fine Threads—UNF Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
3(.099)	0.0990	48	0.0734	0.00487	56	0.0771	0.00523
4(.112)	0.1120	40	0.0813	0.00604	48	0.0864	0.00661
5(.125)	0.1250	40	0.0943	0.00796	44	0.0971	0.00830
6(.138)	0.1380	32	0.0997	0.00909	40	0.1073	0.01015
8(.164)	0.1640	32	0.1257	0.0140	36	0.1299	0.01474
10(.190)	0.1900	24	0.1389	0.0175	32	0.1517	0.0200
12(.216)	0.2160	24	0.1649	0.0242	28	0.1722	0.0258
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599
$\frac{9}{16}$	0.5625	12	0.4603	0.182	18	0.4943	0.203
$\frac{5}{8}$	0.6250	11	0.5135	0.226	18	0.5568	0.256
$\frac{3}{4}$	0.7500	10	0.6273	0.334	16	0.6733	0.373
$\frac{7}{8}$	0.8750	9	0.7387	0.462	14	0.7874	0.509
1	1.0000	8	0.8466	0.606	12	0.8978	0.663
$1\frac{1}{8}$	1.1250	7	0.9497	0.763	12	1.0228	0.856
$1\frac{1}{4}$	1.2500	7	1.0747	0.969	12	1.1478	1.073
$1\frac{3}{8}$	1.3750	6	1.1705	1.155	12	1.2728	1.315
$1\frac{1}{2}$	1.5000	6	1.2955	1.405	12	1.3978	1.581
$1\frac{3}{4}$	1.7500	5	1.5046	1.90			
2	2.0000	$4\frac{1}{2}$	1.7274	2.50			
$2\frac{1}{4}$	2.2500	$4\frac{1}{2}$	1.9774	3.25			
$2\frac{1}{2}$	2.5000	4	2.1933	4.00			
$2\frac{3}{4}$	2.7500	4	2.4433	4.93			
3	3.0000	4	2.6933	5.97			
$3\frac{1}{4}$	3.2500	4	2.9433	7.10			
$3\frac{1}{2}$	3.5000	4	3.1933	8.33			
$3\frac{3}{4}$	3.7500	4	3.4433	9.66			
4	4.0000	4	3.6933	11.08			

Note: See ANSI standard B1.1-1974 for full details. Unified threads are specified as "$\frac{1}{2}$ in.–13UNC," "1 in.–12UNF."

Table 10.1 Basic Dimensions of Unified Screw Threads

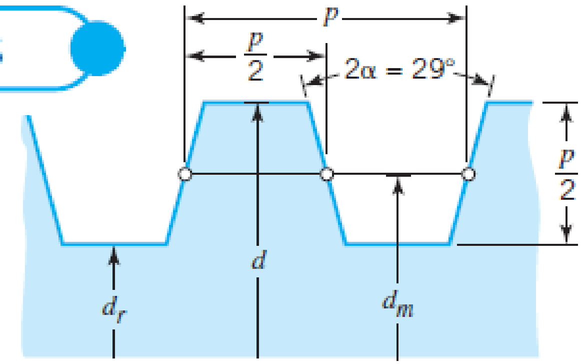
Size	Major Diameter d (in.)	Coarse Threads—UNC Threads per Inch	Coarse Threads—UNC Minor Diameter of External Thread d_r (in.)	Coarse Threads—UNC Tensile Stress Area A_t (in.^{2})	Fine Threads—UNF Threads per Inch	Fine Threads—UNF Minor Diameter of External Thread d_r (in.)	Fine Threads—UNF Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
3(.099)	0.0990	48	0.0734	0.00487	56	0.0771	0.00523
4(.112)	0.1120	40	0.0813	0.00604	48	0.0864	0.00661
5(.125)	0.1250	40	0.0943	0.00796	44	0.0971	0.00830
6(.138)	0.1380	32	0.0997	0.00909	40	0.1073	0.01015
8(.164)	0.1640	32	0.1257	0.0140	36	0.1299	0.01474
10(.190)	0.1900	24	0.1389	0.0175	32	0.1517	0.0200
12(.216)	0.2160	24	0.1649	0.0242	28	0.1722	0.0258
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599

TABLE 10.2 Basic Dimensions of ISO Metric Screw Threads

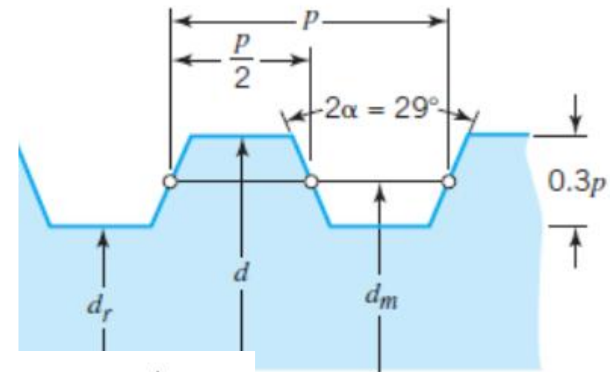
Nominal Diameter d (mm)	Coarse Threads			Fine Threads		
	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

Note: Metric threads are identified by diameter and pitch as "M8 × 1.25."

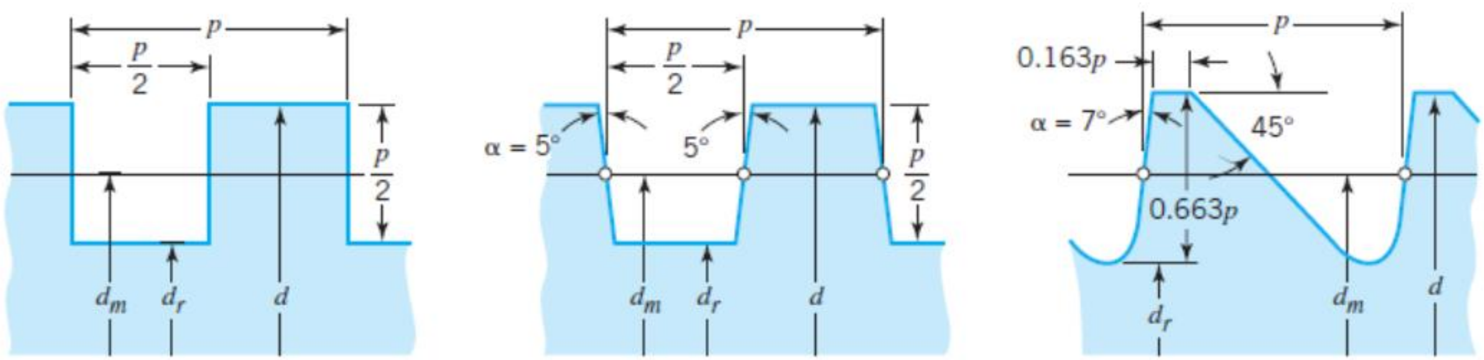
10.2 Thread Forms, Terminology, and Standards



- Standard form of power screws
- Acme is the oldest. Acme stub is easier to heat treat
- Square gives more efficiency but 0° angle difficult
- Modified square with 5° is commonly used
- Buttress is used to resist large axial force in one direction
- For power screws with multiple threads, the number of threads per inch is defined as the reciprocal of the pitch, not the reciprocal of the lead.



Acme stub



(c) Square

(d) Modified square

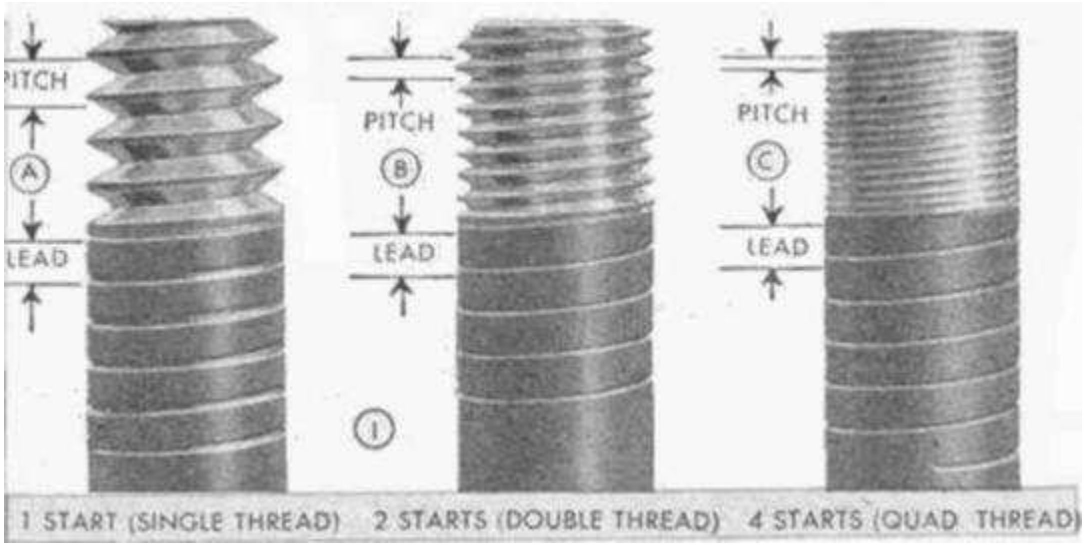
(e) Buttress

FIGURE 10.4

Power screw thread forms. [Note: All threads shown are external (i.e., on the screw, not on the nut); d_m is the mean diameter of the thread contact and is approximately equal to $(d + d_r)/2$.]

10.2 Thread Forms, Terminology, and Standards

An example: let's say you have a 1/2"-8 X 6' with 2 starts. The 1/2" is the diameter and the 8 is the threads per inch, but the difference here is the number of starts. The actual "turns per inch" is actually 4, not 8.



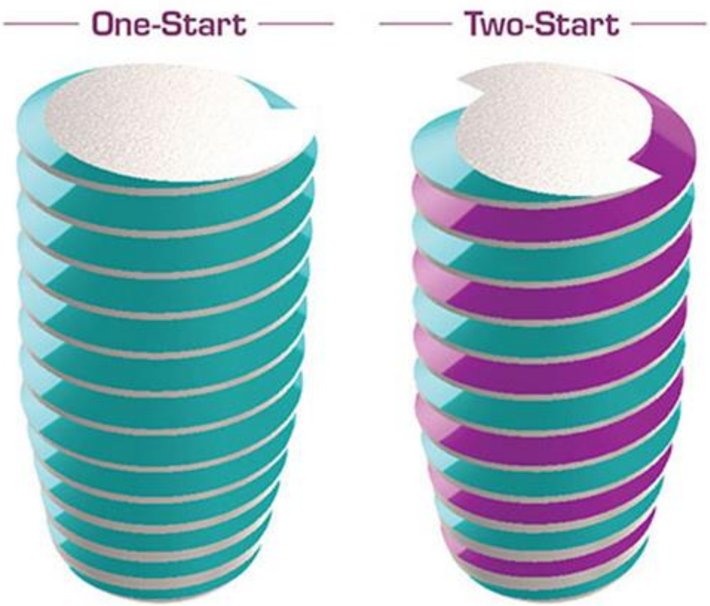


TABLE 10.3 Standard Sizes of Power Screw Threads

Major Diameter d (in.)	Threads per Inch		
	Acme and Acme Stub[a]	Square and Modified Square	Buttress[b]
$\frac{1}{4}$	16	10	
$\frac{5}{16}$	14		
$\frac{3}{8}$	12		
$\frac{3}{8}$	10	8	
$\frac{7}{16}$	12		
$\frac{7}{16}$	10		
$\frac{1}{2}$	10	$6\frac{1}{2}$	16
$\frac{5}{8}$	8	$5\frac{1}{2}$	16
$\frac{3}{4}$	6	5	16
$\frac{7}{8}$	6	$4\frac{1}{2}$	12
1	5	4	12
$1\frac{1}{8}$	5		
$1\frac{1}{4}$	5	$3\frac{1}{2}$	10
$1\frac{3}{8}$	4		10
$1\frac{1}{2}$	4	3	10
$1\frac{3}{4}$	4	$2\frac{1}{2}$	8
2	4	$2\frac{1}{4}$	8
$2\frac{1}{4}$	3	$2\frac{1}{4}$	8
$2\frac{1}{2}$	3	2	8
$2\frac{3}{4}$	3	2	6
3	2	$1\frac{3}{4}$	6
$3\frac{1}{2}$	2	$1\frac{5}{8}$	6
4	2	$1\frac{1}{2}$	6
$4\frac{1}{2}$	2		5
5	2		5

10.3 Power Screws

- Nut turned with applied torque of T lifts load P
- To compensate for the friction between nut and the base, thrust bearings are used
- Another application is shown below
  - For accurate positioning of the nut, based on rotation of the lead screw by servomotor

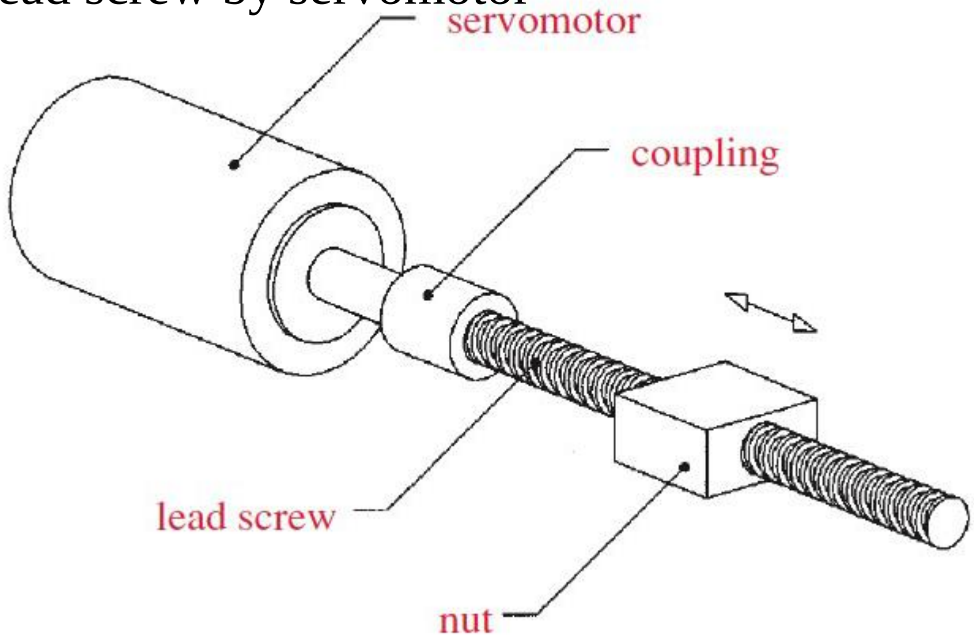


FIGURE 15-5

Servomotor-Driven Lead Screw for Use as a Positioning Device *Courtesy of J. Karsberg,*

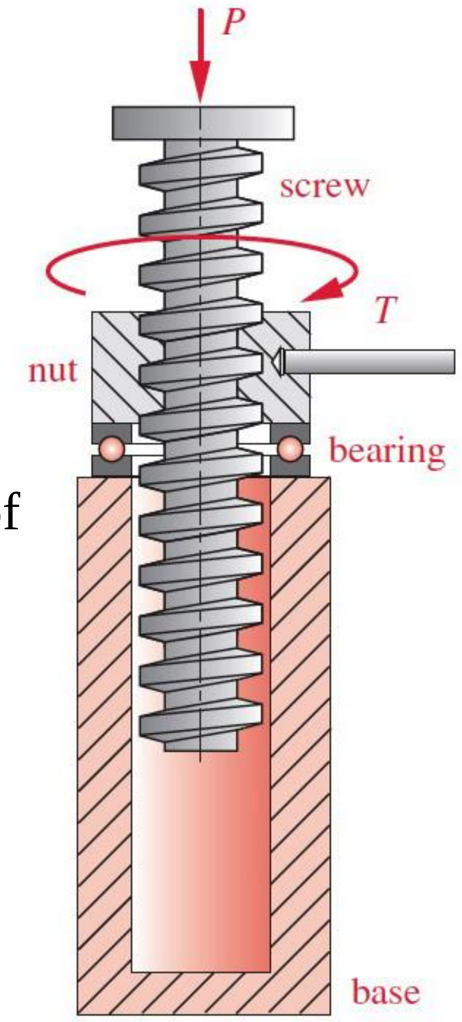


FIGURE 15-4

An Acme-Thread Power-Screw Jack

10.3 Power Screws

- shaded member connected to the handle rotates, and that a ball thrust bearing transfers the axial force from the rotating to a nonrotating member.
- All 3 jacks being same, Figure 10.5c for determining the torque, Fa, that must be applied to the nut in order to lift a given weight.
- Turning the nut in Figure 10.5c forces each portion of the nut thread to climb an inclined plane.

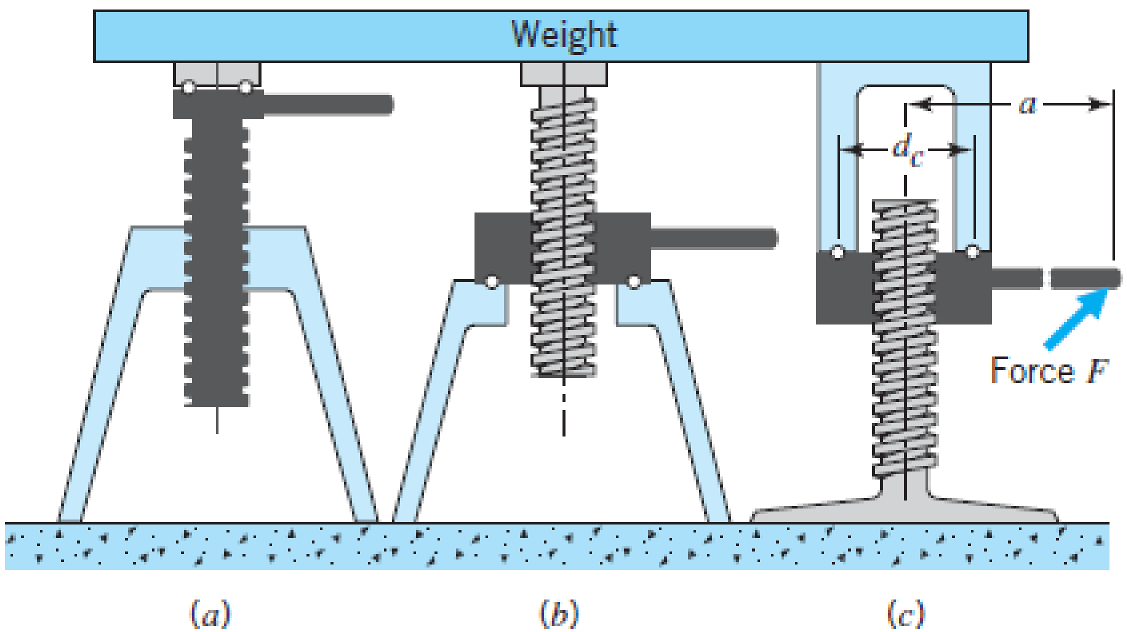


FIGURE 10.5
Weight supported by three screw jacks. In each screw jack, only the shaded member rotates.

10.3 Power Screws

- Turning the nut forces each portion of the nut thread to climb an inclined plane.
- If a full turn were developed, a triangle would be formed, illustrating tan λ

$$\tan \lambda = \frac{L}{\pi d_m} \tag{10.1}$$

- A segment of the nut is represented by the small block acted upon by load w, normal force n, friction force fn, and tangential force q.
- force q times $d_m/2$ represents the torque applied to the nut segment.

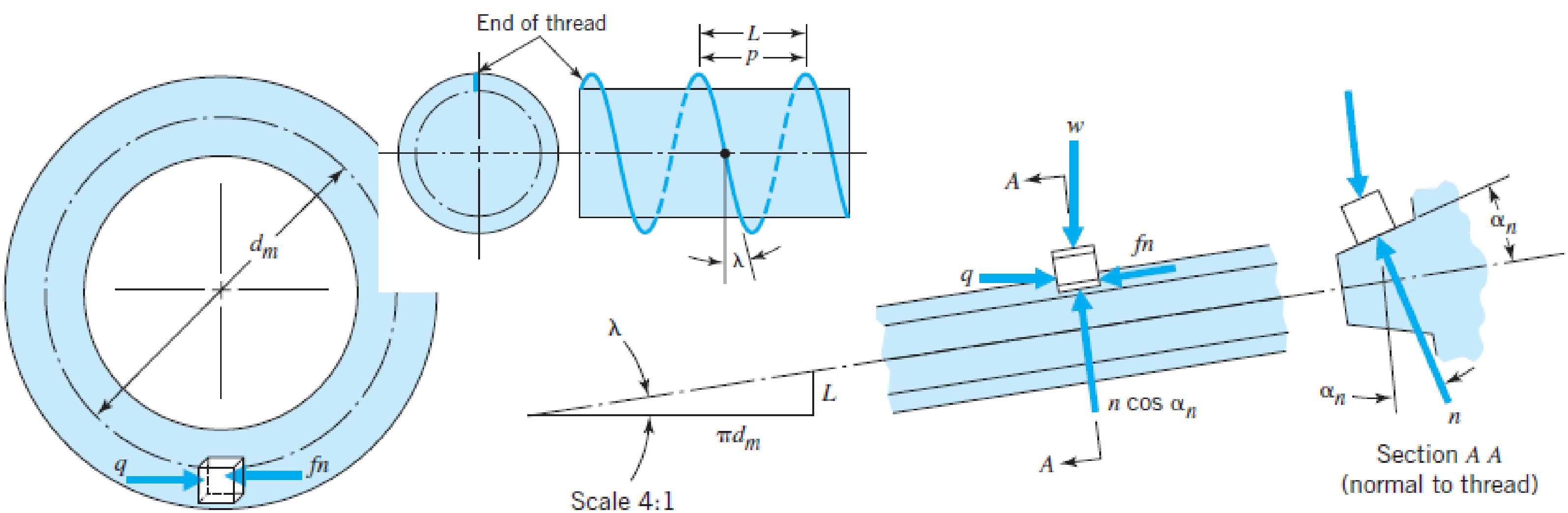


FIGURE 10.6
Screw thread forces.

λ = lead angle

L = lead

d_m = mean diameter of thread contact

10.3 Power Screws

- Summing the tangential forces

$$\sum F_t = 0: \quad q - n(f \cos\lambda + \cos\alpha_n \sin\lambda) = 0 \tag{a}$$

- Summing the axial forces

$$\sum F_a = 0: \quad w + n(f \sin\lambda - \cos\alpha_n \cos\lambda) = 0$$

$$n = \frac{w}{\cos\alpha_n \cos\lambda - f \sin\lambda} \tag{b}$$

Combining Eqs. a and b, we have

$$q = w\frac{f \cos\lambda + \cos\alpha_n \sin\lambda}{\cos\alpha_n \cos\lambda - f \sin\lambda} \tag{c}$$

- With torque for q being $q(d_m/2)$ and q, n, w are acting on a small segment of the nut, integrating this to full nut and changing the notations to Q, N, W, the torque T required to lift a load W is

$$T = Q\frac{d_m}{2} = \frac{Wd_m}{2}\frac{f \cos\lambda + \cos\alpha_n \sin\lambda}{\cos\alpha_n \cos\lambda - f \sin\lambda} \tag{10.2}$$

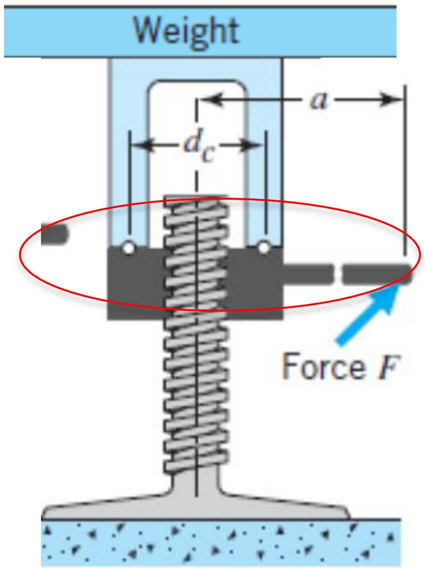
10.3 Power Screws

- Since L is more commonly referred to in threads than λ, dividing the numerator and denominator by cos λ and then substituting L/πdm for tan λ.

$$T = \frac{Wd_m}{2}\frac{f\pi d_m + L\cos\alpha_n}{\pi d_m \cos\alpha_n - fL} \tag{10.3}$$

- Since a bearing or thrust washer (dia d_c) is used friction adds to the torque required
- If the coefficient of friction of the collar washer or bearing is f_c then

$$T = \frac{Wd_m}{2}\frac{f\pi d_m + L\cos\alpha_n}{\pi d_m \cos\alpha_n - fL} + \frac{Wf_c d_c}{2} \tag{10.4}$$



- For a square thread, $\cos\alpha_n = 1$ this simplifies to

$$T = \frac{Wd_m}{2}\frac{f\pi d_m + L}{\pi d_m - fL} + \frac{Wf_c d_c}{2} \tag{10.4a}$$

10.3 Power Screws

- For lowering the load, the directions of q and fn are reversed giving

$$T = \frac{Wd_m}{2} \frac{f\pi d_m - L\cos\alpha_n}{\pi d_m \cos\alpha_n + fL} + \frac{Wf_c d_c}{2} \qquad \textbf{(10.5)}$$

- For a square thread, this simplifies to

$$T = \frac{Wd_m}{2} \frac{f\pi d_m - L}{\pi d_m + fL} + \frac{Wf_c d_c}{2} \qquad \textbf{(10.5a)}$$

- f_c can be (because very low) neglected if ball or roller thrust bearing is used and the second portion of the term does not come into play

- f & f_c can vary between .08 to .2 if plain thrust collar is used (if roller bearing used, f_c can be neglected). This range includes both starting and running friction, with starting friction being 1 and 1/3rd higher than running friction

10.3 Power Screws

- Self locking implies positive torque to lower the load

$$T = \frac{Wd_m}{2}\frac{f\pi d_m - L\cos\alpha_n}{\pi d_m \cos\alpha_n + fL} + \frac{Wf_c d_c}{2} \qquad \textbf{(10.5)}$$

- Neglecting collar friction, screw can be self locking if T>=0

$$f \geqq \frac{L\cos\alpha_n}{\pi d_m} \qquad \textbf{(10.7)}$$

- For square threads

$$T = \frac{Wd_m}{2}\frac{f\pi d_m - L}{\pi d_m + fL} + \frac{Wf_c d_c}{2} \qquad \textbf{(10.5a)}$$

$$f \geqq \frac{L}{\pi d_m}, \qquad \text{or} \quad f \geqq \tan\lambda$$

10.3 Power Screws

- Work output divided by work input is the efficiency
- Work output in 1 revolution is load times distance which is WL
- Work input is the torque in one revolution which is 2πT
- So efficiency e = WL/ 2πT

$$\text{Efficiency, } e = \frac{L}{\pi d_m} \frac{\pi d_m \cos \alpha_n - fL}{\pi f d_m + L \cos \alpha_n} \qquad \textbf{(10.8)}$$

- For a square thread

$$e = \frac{L}{\pi d_m} \frac{\pi d_m - fL}{\pi f d_m + L} \qquad \textbf{(10.8a)}$$

- Simplified to

$$e = \frac{\cos \alpha_n - f \tan \lambda}{\cos \alpha_n + f \cot \lambda} \qquad \textbf{(10.9)}$$

for square threads

$$e = \frac{1 - f \tan \lambda}{1 + f \cot \lambda}$$

10.3 Power Screws

- As f increases; e lowers
- Efficiency tends to 0 as lead angle approaches 0, as load does not move much in the vertical plane
- Efficiency tends to 0 as lead angle approaches 90, as the plane more perpendicular and requires a lot of torque to move the object even slightly
- Ball bearing screws reduce f

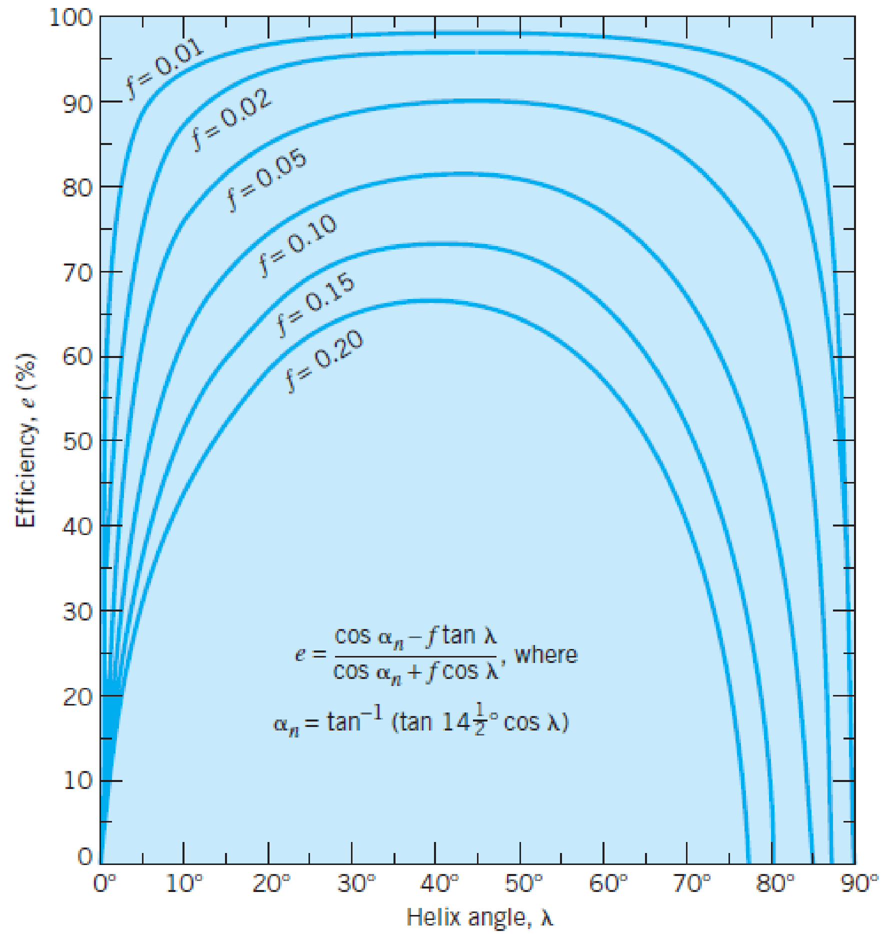


FIGURE 10.8
Efficiency of Acme screw threads when collar friction is negligible. (Note: Values for square threads are higher by less than 1 percent.)

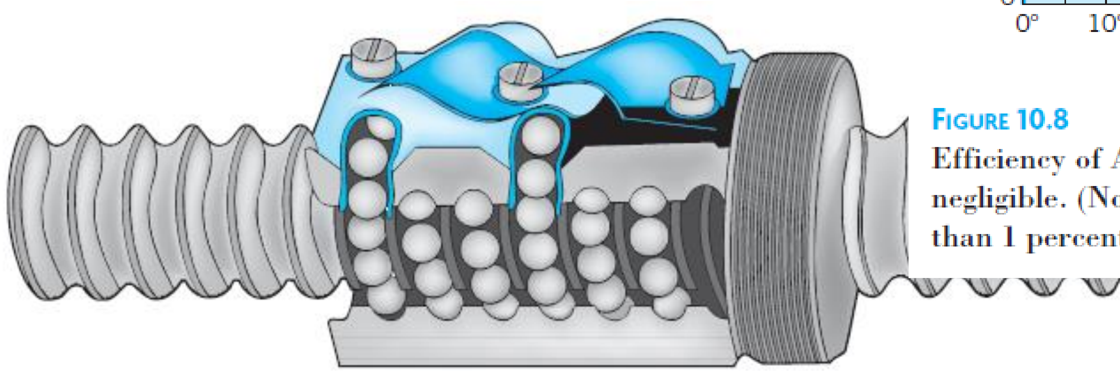


FIGURE 10.9
Ball-bearing screw assembly with a portion of the nut cut away to show construction. (Courtesy Saginaw Steering Gear Division, General Motors Corporation.)

SAMPLE PROBLEM 10.1 Acme Power Screw

A screw jack (Figure 10.10) with a 1-in., double-thread Acme screw is used to raise a load of 1000 lb. A plain thrust collar of $1\frac{1}{2}$-in. mean diameter is used. Coefficients of running friction are estimated as 0.12 and 0.09 for f and f_c, respectively.

a. Determine the screw pitch, lead, thread depth, mean pitch diameter, and helix angle.

b. Estimate the starting torque for raising and for lowering the load.

c. Estimate the efficiency of the jack when raising the load.

SOLUTION

Known: A double-thread Acme screw and a thrust collar, each with known diameter and running friction coefficient, are used to raise a specified load.

Find:

a. Determine the screw pitch, lead, thread depth, mean pitch diameter, and helix angle.

b. Estimate the starting torque for raising and lowering the load.

c. Calculate the efficiency of the jack when raising the load.

TABLE 10.3 Standard Sizes of Power Screw Threads

Major Diameter d (in.)	Threads per Inch: Acme and Acme Stub[a]
$\frac{1}{4}$	16
$\frac{5}{16}$	14
$\frac{3}{8}$	12
$\frac{3}{8}$	10
$\frac{7}{16}$	12
$\frac{7}{16}$	10
$\frac{1}{2}$	10
$\frac{5}{8}$	8
$\frac{3}{4}$	6
$\frac{7}{8}$	6
1	5

Schematic and Given Data:

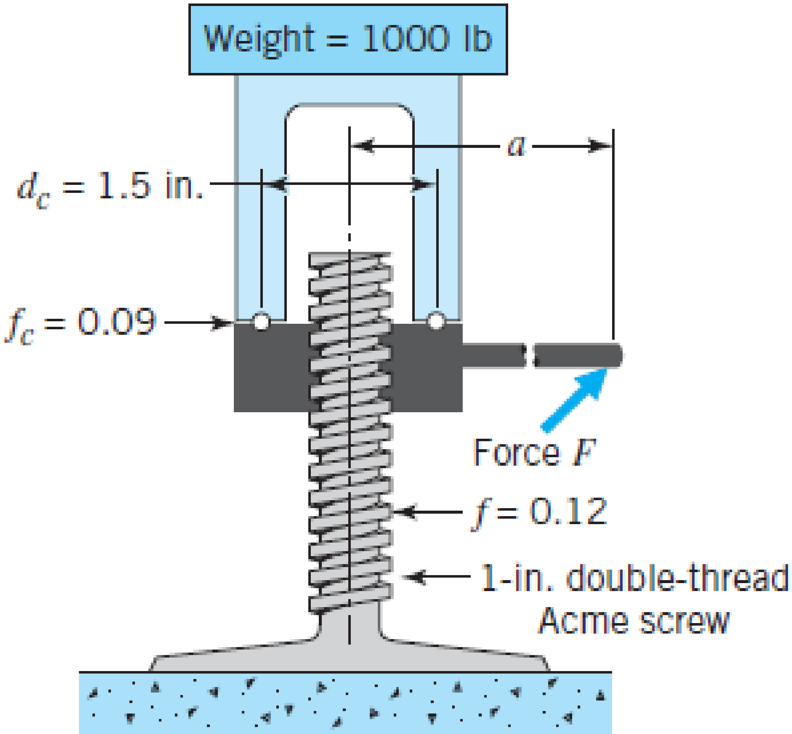


FIGURE 10.10
Screw jack lifting a nonrotating load.

Assumptions:

1. The starting and running friction remain steady.
2. Starting friction is about one-third higher than running friction.

Analysis:

a. From Table 10.3, there are five threads per inch, hence $p = 0.2$ in.
Because of the double thread, $L = 2p$, or $L = 0.4$ in.
From Figure 10.4*a*, thread depth $= p/2 = 0.1$ in.
From Figure 10.4*a*, $d_m = d - p/2 = 1\text{ in.} - 0.1\text{ in.} = 0.9$ in.
From Eq. 10.1, $\lambda = \tan^{-1} L/\pi d_m = \tan^{-1} 0.4/\pi(0.9) = 8.05°$.

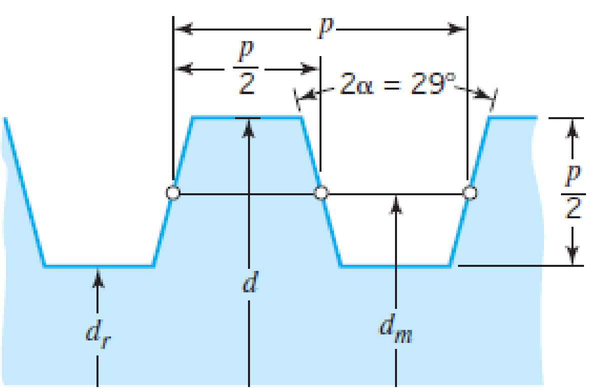
b. For starting, increase the given coefficients of friction by about one-third, giving $f = 0.16$ and $f_c = 0.12$. Equation 10.4a for square threads could be used with sufficient accuracy, but we will illustrate the complete solution, using Eq. 10.4 for the general case.

First, find α_n from Eq. 10.6:

$$\begin{aligned}\alpha_n &= \tan^{-1}(\tan\alpha\cos\lambda)\\ &= \tan^{-1}(\tan 14.5° \cos 8.05°) = 14.36°\end{aligned}$$

Then, substituting in Eq. 10.4 gives

$$\begin{aligned}T &= \frac{Wd_m}{2}\,\frac{f\pi d_m + L\cos\alpha_n}{\pi d_m\cos\alpha_n - fL} + \frac{Wf_c d_c}{2}\\ &= \frac{1000(0.9)}{2}\,\frac{0.16\pi(0.9) + 0.4\cos 14.36°}{\pi(0.9)\cos 14.36° - 0.16(0.4)} + \frac{1000(0.12)(1.5)}{2}\\ &= 141.3 + 90; \qquad T = 231.3\text{ lb}\cdot\text{in.}\end{aligned}$$



(*a*) Acme

(*Comment:* With reference to Figure 10.10, this would correspond to a force of 19.3 lb on the end of a 12-in. handle. If Eq. 10.4a is used, the answer is only slightly less: 228.8 lb · in.). For lowering the load, use Eq. 10.5:

$$T = \frac{Wd_m}{2}\,\frac{f\pi d_m - L\cos\alpha_n}{\pi d_m \cos\alpha_n + fL} + \frac{Wf_c d_c}{2}$$

$$= \frac{1000(0.9)}{2}\,\frac{0.16\pi(0.9) - 0.4\cos 14.36^\circ}{\pi(0.9)\cos 14.36^\circ + 0.16(0.4)} + 90$$

$$= 10.4 + 90; \qquad T = 100.4 \text{ lb} \cdot \text{in.}$$

(*Comment:* Equation 10.5a gives a torque of 98.2 lb · in.)

c. Repeating the substitution in Eq. 10.4, but changing the coefficient of friction to the running values of 0.12 and 0.09, indicates that to raise the load, once motion is started, the torque must be 121.5 + 67.5 = 189 lb · in. Substituting once more in Eq. 10.4, but changing both friction coefficients to zero, indicates that the torque must be 63.7 + 0 = 63.7 lb · in. to raise the load. Efficiency is the ratio of friction-free torque to actual torque, or

$$e = \frac{63.7}{189} = 33.7 \text{ percent}$$

Comment: If a ball thrust bearing were used so that collar friction could be neglected, the efficiency would increase to 63.7/121.5 = 52 percent. This would correspond to the efficiency of the screw itself and agrees with the plotted value in Figure 10.8.

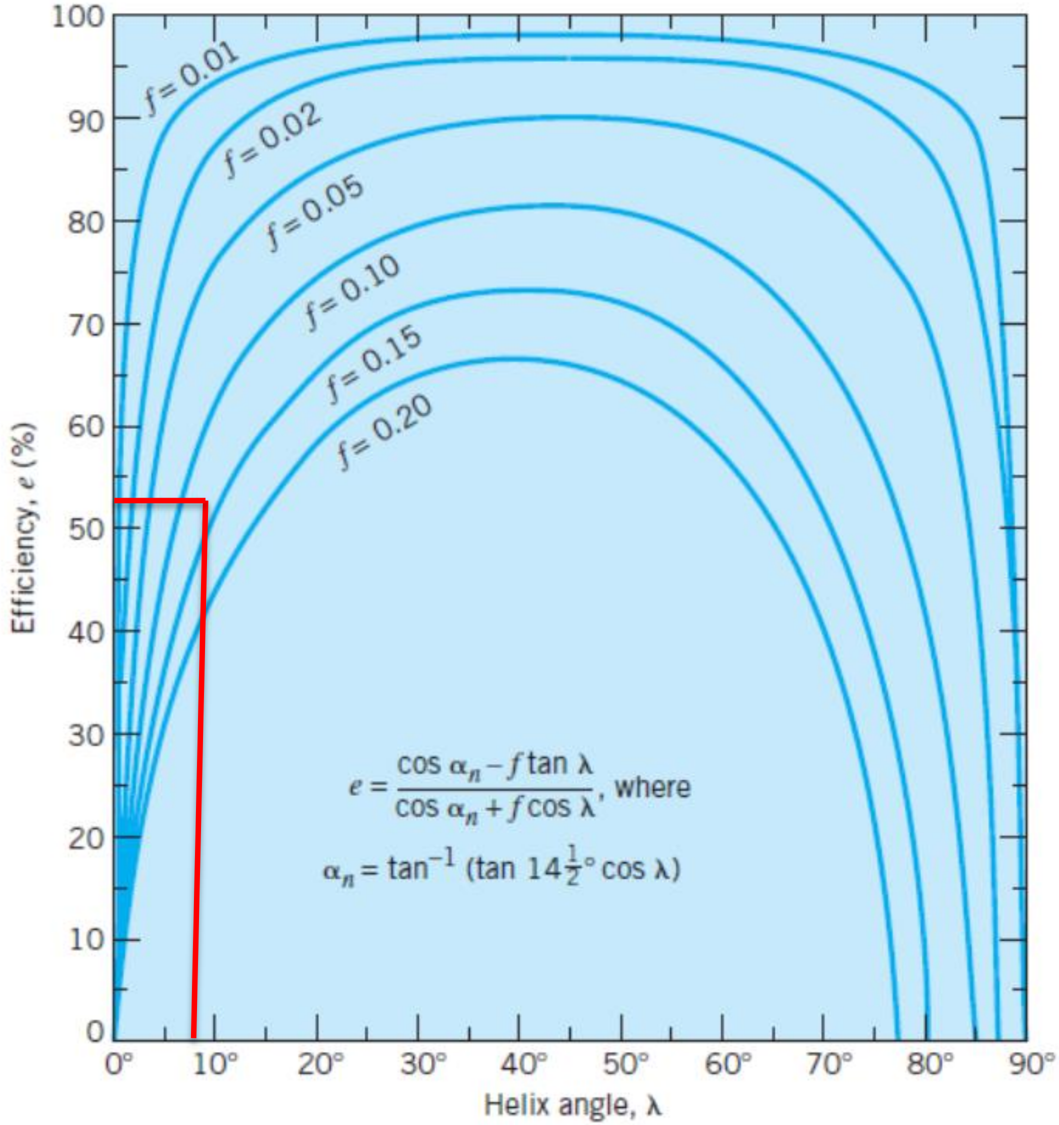
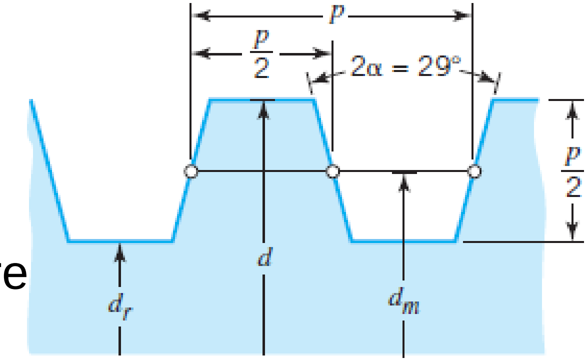


Figure 10.8

Efficiency of Acme screw threads when collar friction is negligible. (Note: Values for square threads are higher by less than 1 percent.)

10.4 Static Screw Stresses

10.4.1 Torsion



(a) Acme

- For power screws and threaded fasteners the stress are

- **Torsion** while tightening

$$\tau = \frac{Tc}{J} = \frac{16T}{\pi d^3} \qquad (4.3, 4.4)$$

- where d is root diameter, d_r , obtained from Figure 10.4 (for power screws) or Tables 10.1 and 10.2 (for threaded fasteners).

- If the screw or bolt is hollow, where d_i represents the inside diameter.

- Where collar friction is negligible, the torque transmitted through a power screw is the full applied torque.
- With threaded fasteners, the equivalent of substantial collar friction is normally present, in which case it is customary to assume that the torque transmitted through the threaded section is approximately half the wrench torque.

$$J = \pi\,(d_r^4 - d_i^4)/32$$

10.4 Static Screw Stresses

10.4.2 Axial Load

- Power screws are subjected to direct P/A tensile and compressive stresses; threaded fasteners are normally subjected only to tension.
- The effective area for fasteners is the tensile stress area A_t (Table 10.1 & 10.2).
- For power screws axial stresses are not critical; so A_t, approximated based on d_r
- Threaded fasteners should always have enough ductility to permit local yielding at thread roots without damage. So non uniform load distribution is ok for static stresses. But not fatigue.

10.4.3 Combined Torsion and Axial Load

- The combination of the stresses can be the distortion energy theory used as a criterion for yielding.
- With threaded fasteners, it is normal for some yielding to occur at the thread roots during initial tightening.

10.4.4 Thread Bearing (Compressive) Stress, and Its Distribution Among the Threads in Contact

- Figure shows "force flow" through bolt & nut

- Compression between the threads exists at threads numbered 1, 2, and 3.

- This type of direct compression is often called bearing, and the area used for P/A stress calculation is the projected area that, for each thread, is $\pi(d^2 - d_i^2)/4$.

- The number of threads in contact is seen from the figure to be t/p.

$$\sigma = \frac{4P}{\pi(d^2 - d_i^2)} \frac{p}{t} \qquad (10.10)$$

- Diameter d_i is the minor diameter of the internal thread. For threaded fasteners this can be approximated by d_r , (Table 10.1)

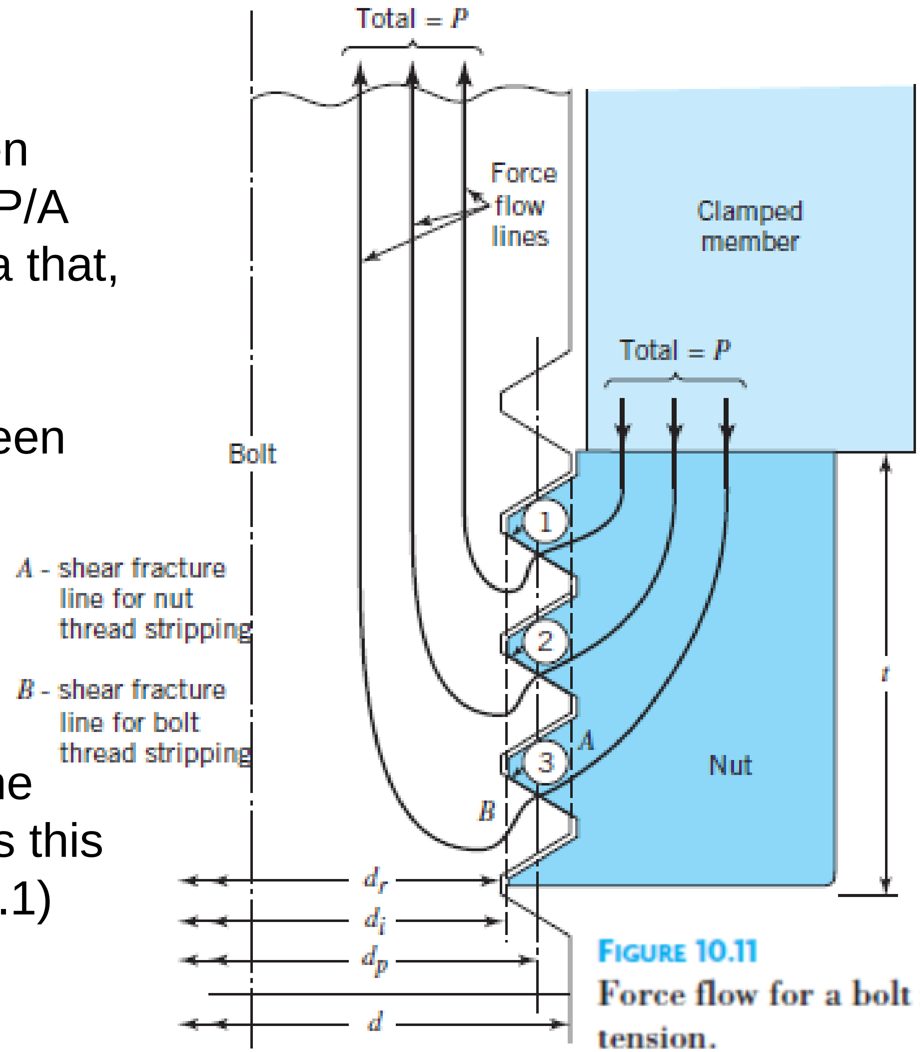


FIGURE 10.11
Force flow for a bolt in tension.

10.4.4 Thread Bearing (Compressive) Stress, and Its Distribution Among the Threads in Contact

$$\sigma = \frac{4P}{\pi(d^2 - d_i^2)} \frac{p}{t} \qquad (10.10)$$

- Equation 10.10 gives an average value of bearing stress. Not uniformly distributed due to threads bending and manufacturing variations from the theoretical geometry.

- Figure 10.11 reveals two important factors causing thread 1 to carry more than its share of the load:

1. The load is shared among the 3 threads as redundant load-carrying members. The shortest (and stiffest) path is through 1.

2. The applied load causes the threaded portion of the bolt to be in tension, whereas the mating portion of the nut is in compression. The resulting deflections slightly increase bolt pitch and decrease nut pitch. This tends to relieve the pressure on threads 2 and 3.

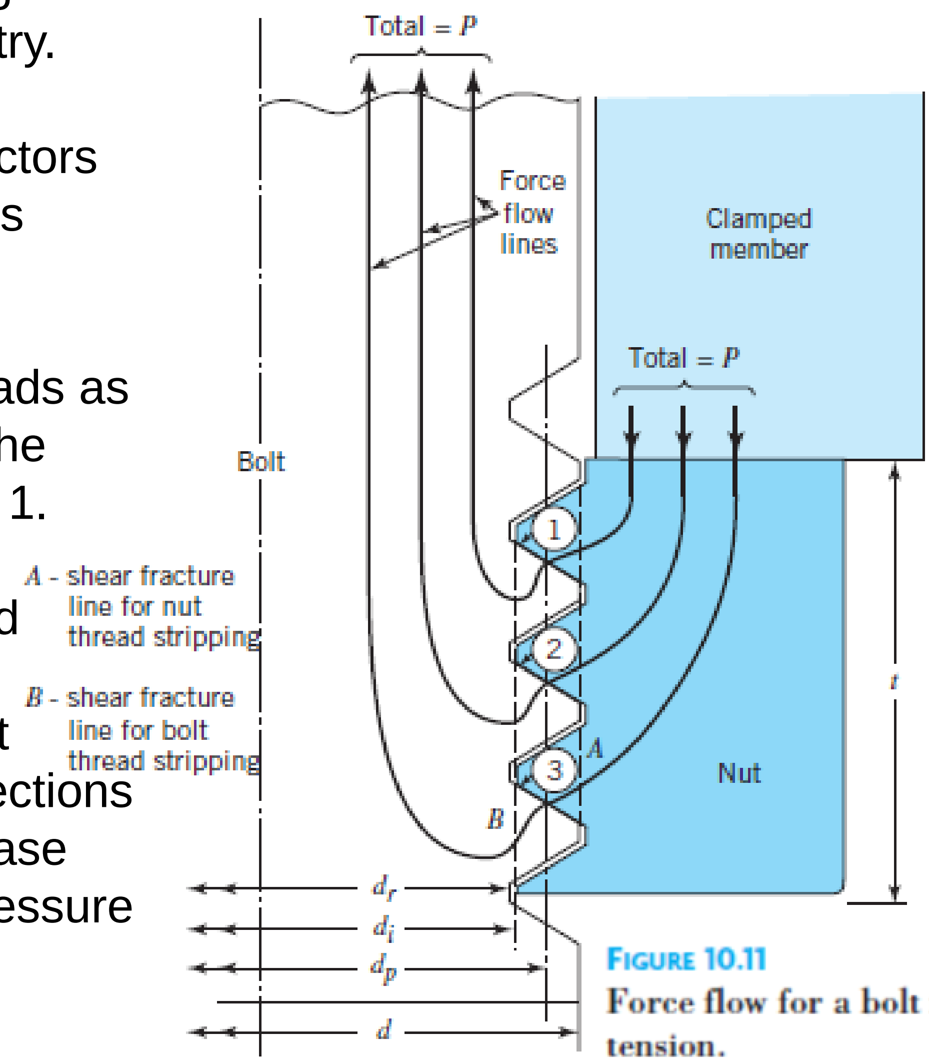


Figure 10.11 Force flow for a bolt in tension.

10.4.4 Thread Bearing (Compressive) Stress, and Its Distribution Among the Threads in Contact

- To obtain nearly equal distribution of loads among the threads in contact, especially when considering fatigue loading is done by:

1. Make nut softer than bolt so that the highly loaded first thread will deflect, transferring the load to the other threads. Maybe increase the number of threads in contact in order to maintain strength.

2. Make Nut Pitch > bolt pitch so that the two pitches are equal after the load is applied. Mfg precision important to make sure that the nut and bolt can be readily assembled

3. Modifying the nut design as shown in Figure 10.12. Here, the nut loading puts the region of the top threads in tension, thus causing elastic changes in pitch that approximately match the changes in bolt pitch. Such special nuts are expensive and have been used only in critical applications involving fatigue loading.

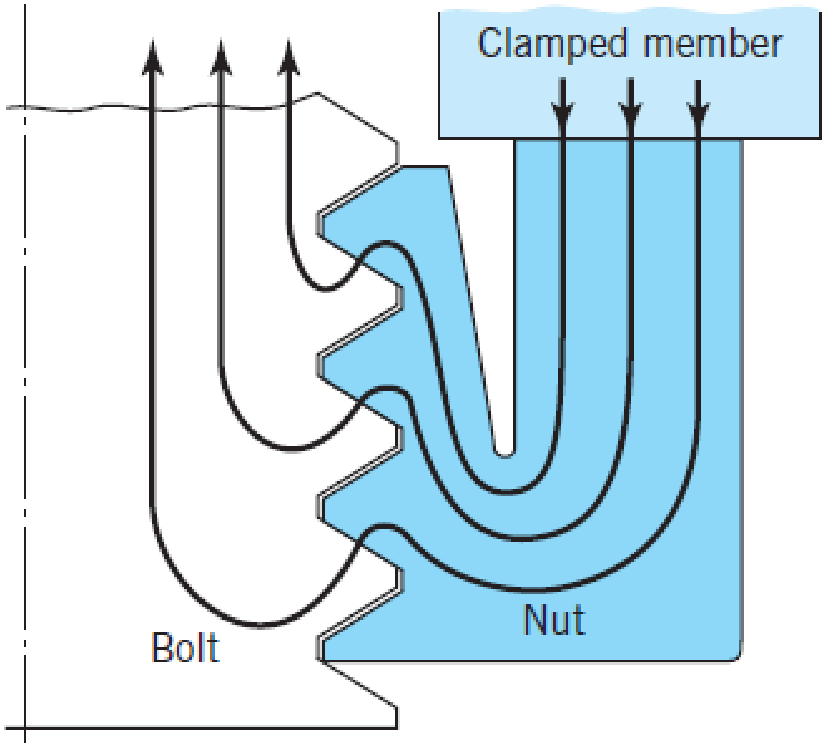


Figure 10.12
A special nut provides more nearly equal distribution of load among threads in contact.

10.4.5 Thread Shear ("Stripping") Stress and Nut Thickness Requirement

- With reference to Figure 10.11, if the nut is weaker than bolt in shear (common), a sufficient overload would "strip" the nut threads along cylindrical surface A.
- If the bolt is weaker in shear, the failure surface would be B.
- From the thread geometry shown in Figure 10.2, the shear area is = $\pi d\,(0.75t)$, where d is the diameter of the shear fracture surface.

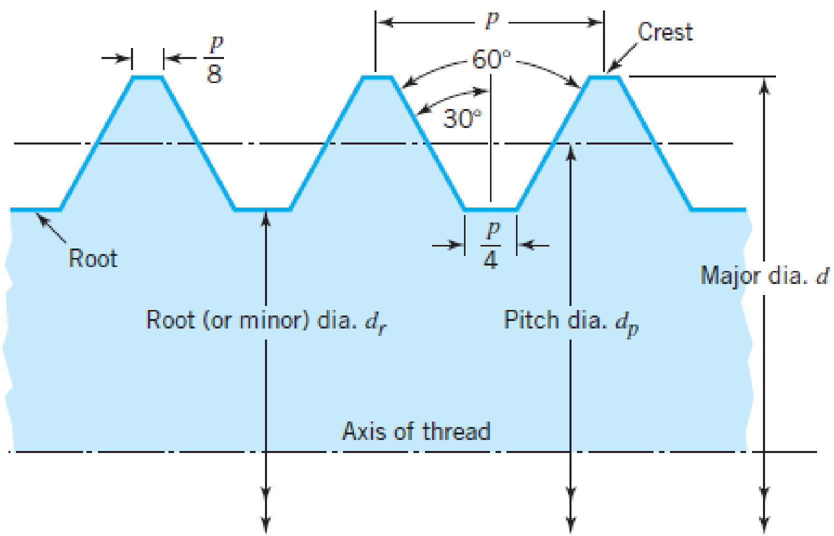


FIGURE 10.2
Unified and ISO thread geometry. The basic profile of the external thread is shown.

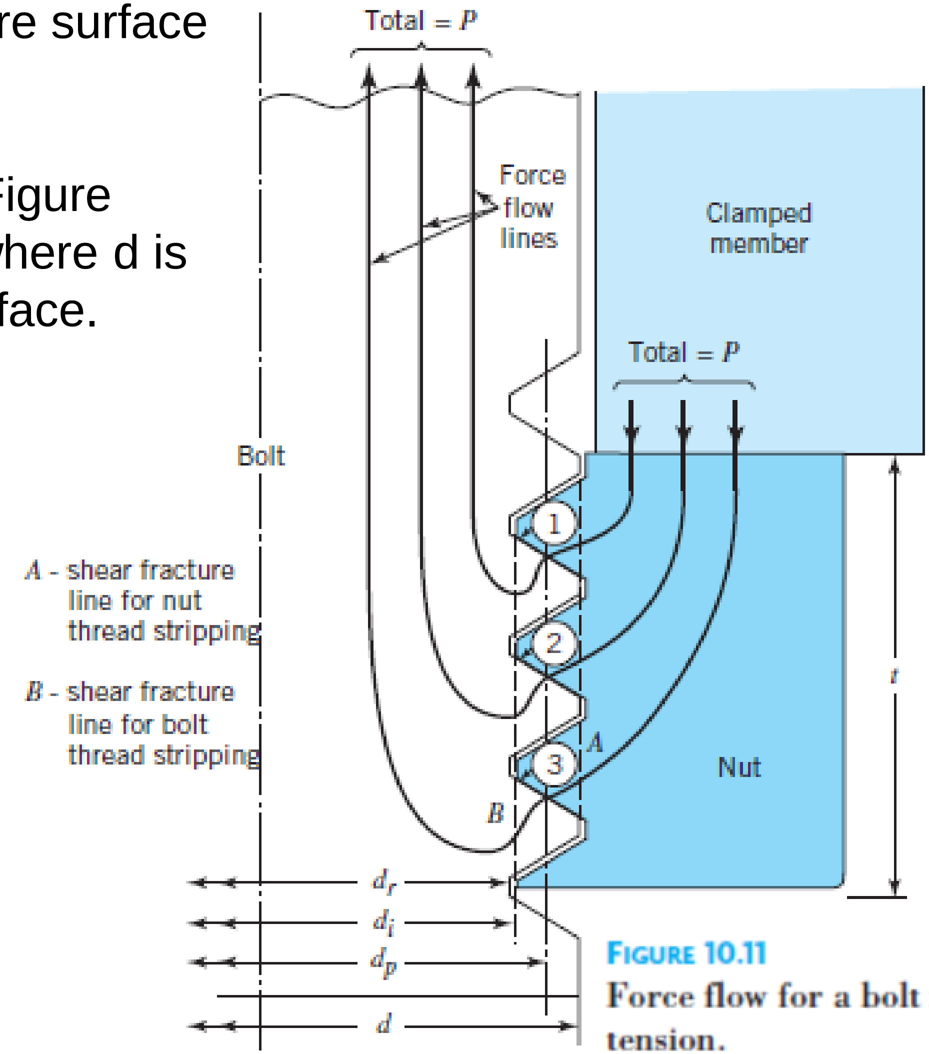


FIGURE 10.11
Force flow for a bolt in tension.

10.4.5 Thread Shear ("Stripping") Stress and Nut Thickness Requirement

- With reference to Figure 10.11, if the nut is weaker than bolt in shear (common), a sufficient overload would "strip" the nut threads along cylindrical surface A.
- If the bolt is weaker in shear, the failure surface would be B.
- From the thread geometry shown in Figure 10.2, the shear area is = πd (0.75t), where d is the diameter of the shear fracture surface.
- The nut thickness (or depth of engagement in a tapped hole) needed to provide a balance between bolt tensile strength and thread stripping strength if bolt and nut strength are same.
- The bolt tensile force required to yield the entire threaded cross section is

$$F_{\text{bolt}} = A_t S_y \approx \frac{\pi}{4}(0.9\,d)^2 S_y$$

- d is the major dia of the thread

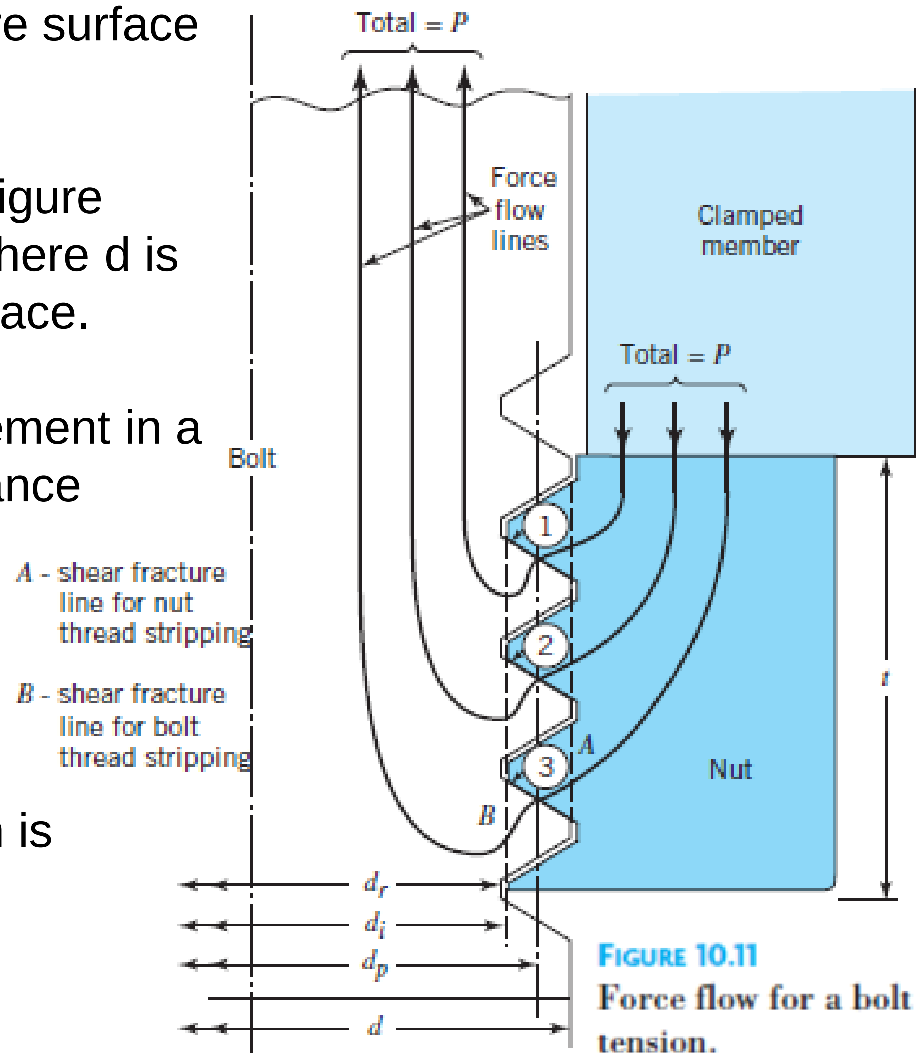


FIGURE 10.11 Force flow for a bolt in tension.

10.4.5 Thread Shear ("Stripping") Stress and Nut Thickness Requirement

- With reference to Figure 10.11, the bolt tensile load required to yield the entire thread-stripping failure surface of the nut based on parabolic stress distribution is

$$F_{\text{nut}} = \pi d(0.75t)S_{sy} \approx \pi d(0.75t)(0.58S_y)$$

- where t is the nut thickness.

$$F_{\text{bolt}} = A_t S_y \approx \frac{\pi}{4}(0.9\, d)^2 S_y$$

- $F_{bolt} = F_{nut}$ indicates bolt tensile and thread-stripping strengths are balanced when the nut thickness is approximately

$$t = 0.47d$$

- Nuts are usually softer than bolts to allow slight yielding of top thread(s) and thus distribute the load more uniformly, the standard nut thickness is approximately

- t = 7/8 d or .875d

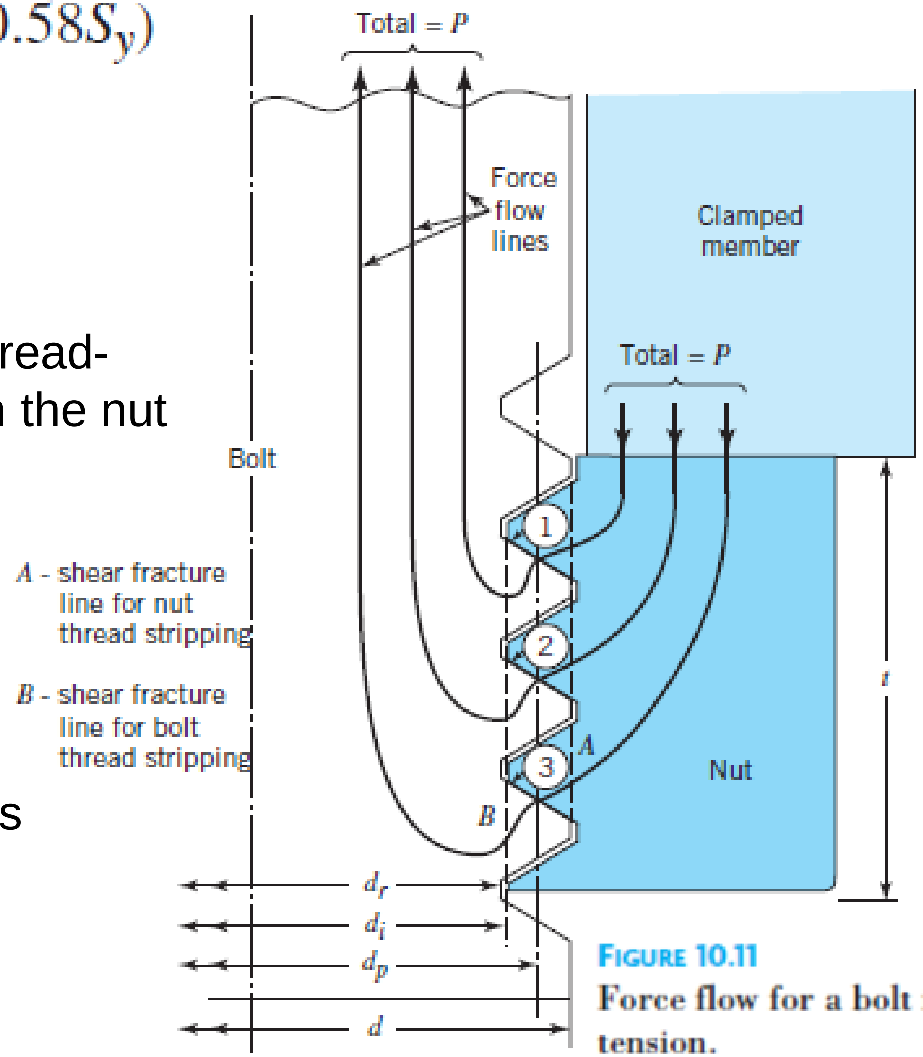


FIGURE 10.11
Force flow for a bolt in tension.

10.4.6 Transverse Shear Loading and Providing Transverse Alignment

- Bolts are sometimes subjected to transverse shear loading fig (4.3, 4.4)
- Shear loads are transmitted by friction, where friction load-carrying capacity is = bolt tension X clamped interface coeff of friction
- For the double shear, the friction load capacity would be twice this amount.
- Sometimes bolts are required to provide precise alignment of mating members and are made with a pilot surface as shown in Figure 10.13.

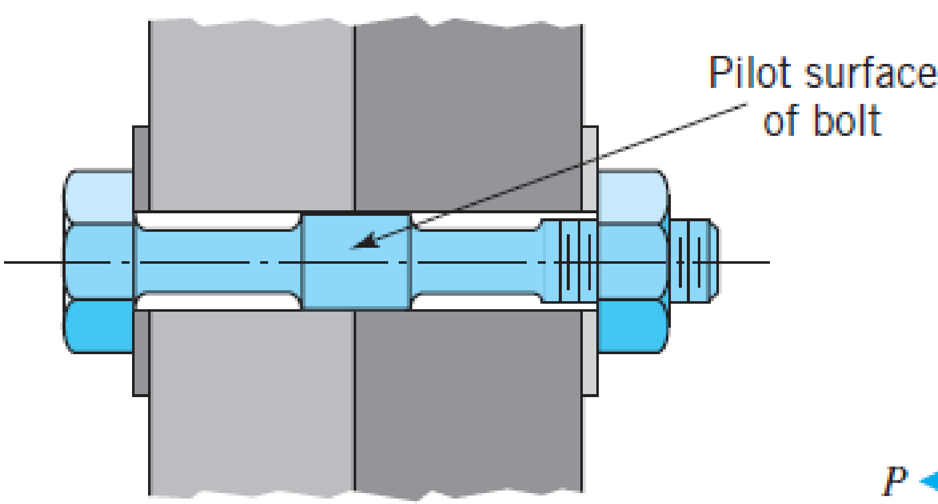
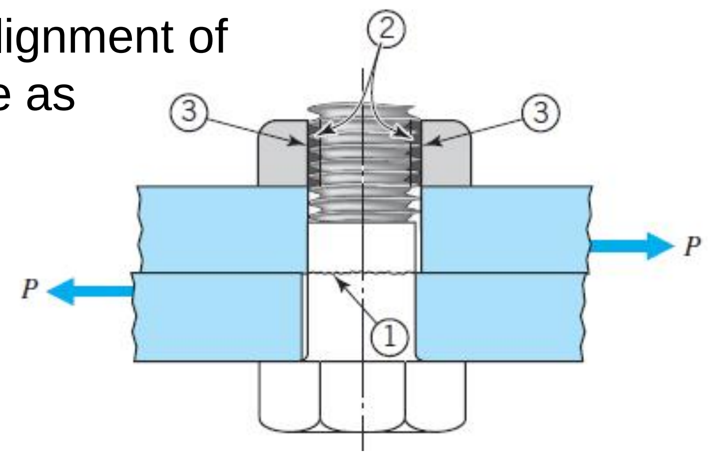
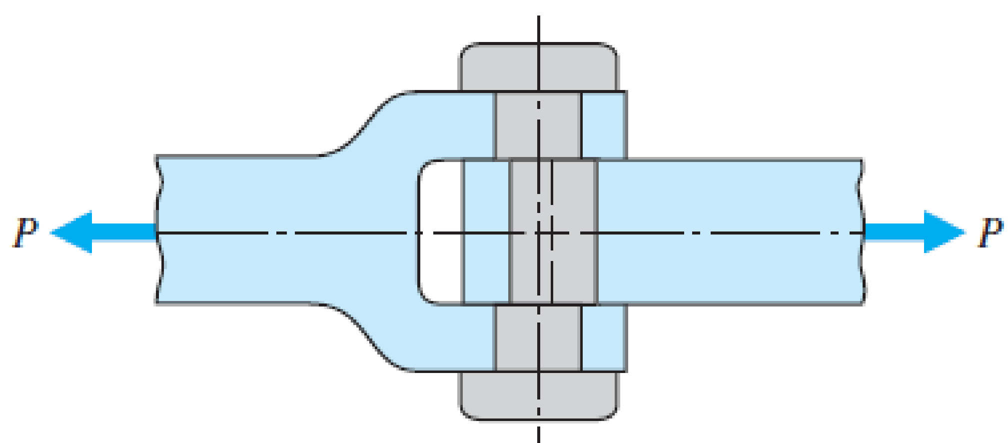


FIGURE 10.13
Bolt with pilot surface.





10.4.7 Column Loading of Power Screws and Associated Design Details

- Long power screws loaded in compression must be designed for buckling. It is important first to make sure that it is necessary to subject the screws to compression or a simple redesign allows it to be in tension
- Often, a simple redesign permits the screws to be in tension.
- For example, Figure 10.14a shows a press with the screws in compression.
- Figure 10.14b shows an alternative design with the screws in tension.
- The second is obviously to be preferred.

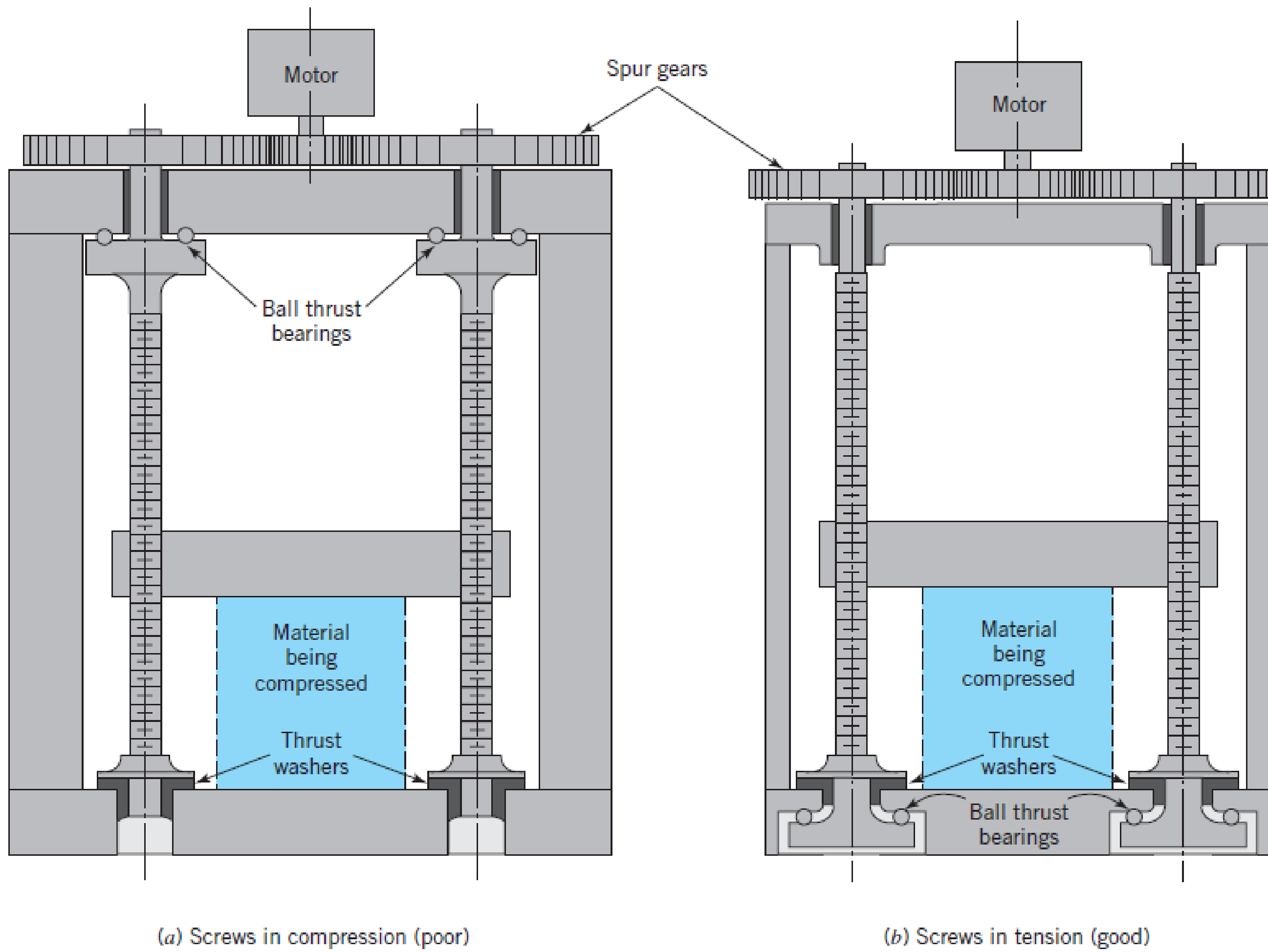


FIGURE 10.14
Alternative screw press arrangements.

10.5 Threaded Fastener Types

- Classified based on intended use, thread type, head style, strength
- Based on intended use
  - Blots - Used with a nut for assembly
  - Machine screws - Or cap screw, threads into a tapped hole
  - ANSI definition - bolt is stationary while nut engages. But screw engages in a tapped hole
  - Studs - Headless fastener threaded on both ends

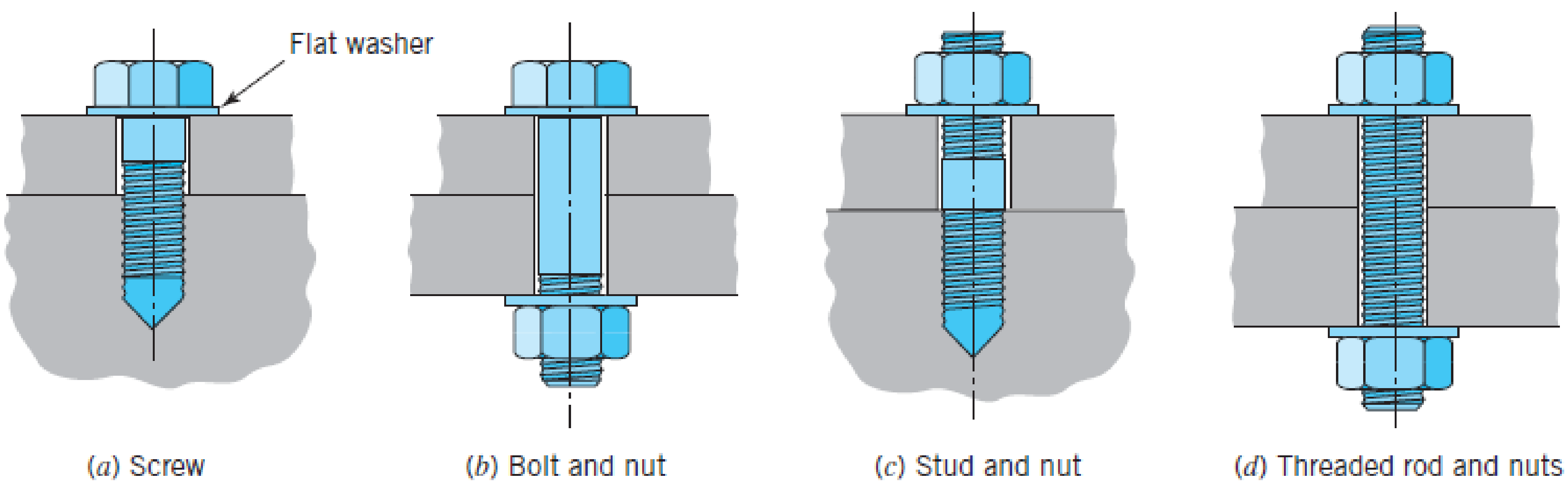


FIGURE 10.15
Basic threaded fastener types.

10.5 Threaded Fastener Types

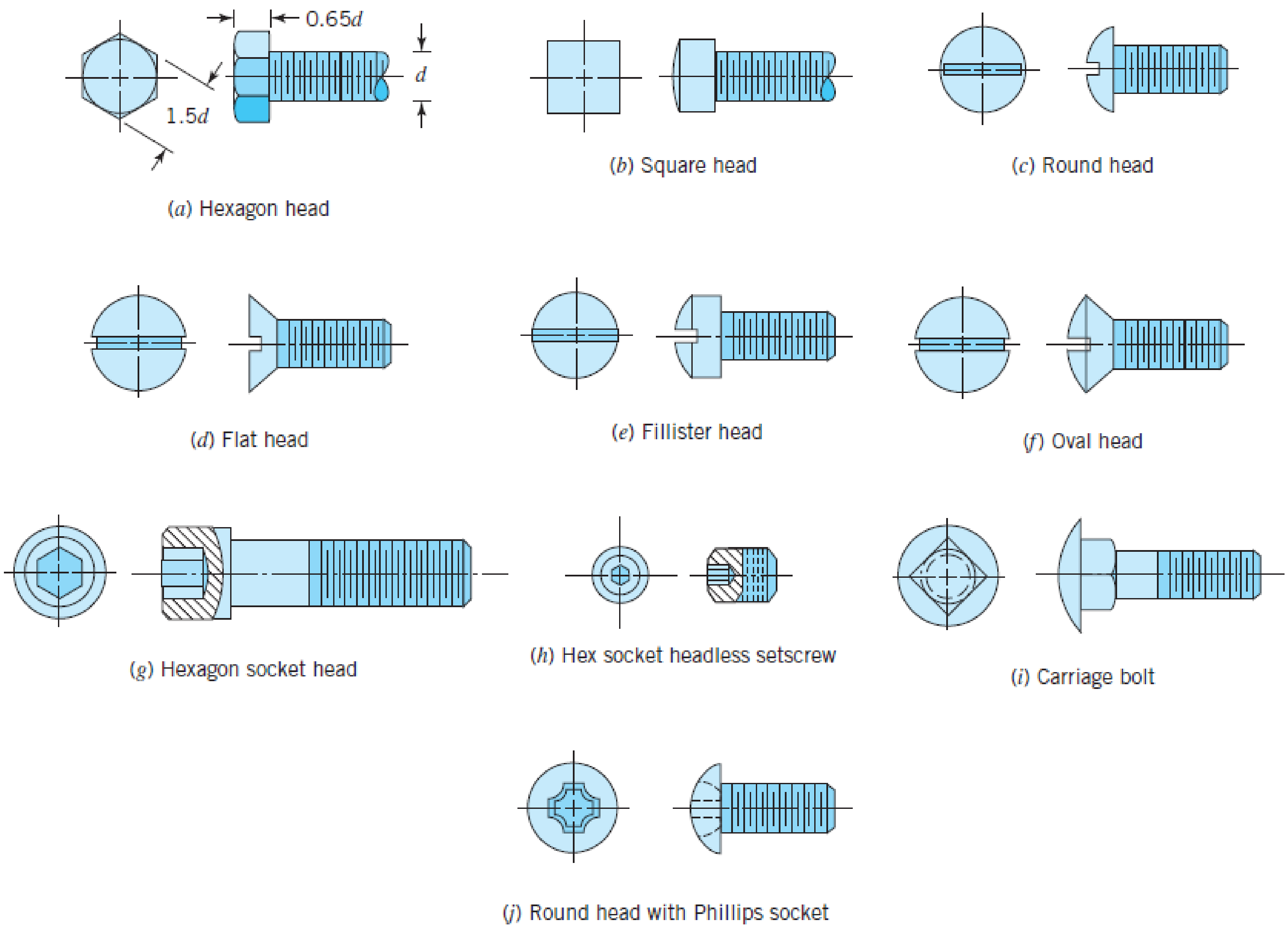


FIGURE 10.16
Some common screw (and bolt) head types.

10.5 Threaded Fastener Types

- Need for screws that are resistant to tampering by unauthorized personnel
- An almost endless number of special threaded fastener designs continue to appear. Some are specially designed for a specific application.
- Others embody proprietary features that appeal to a segment of the fastener market.
- Not only is ingenuity required to devise better threaded fasteners, but also to use them to best advantage in the design of a product.

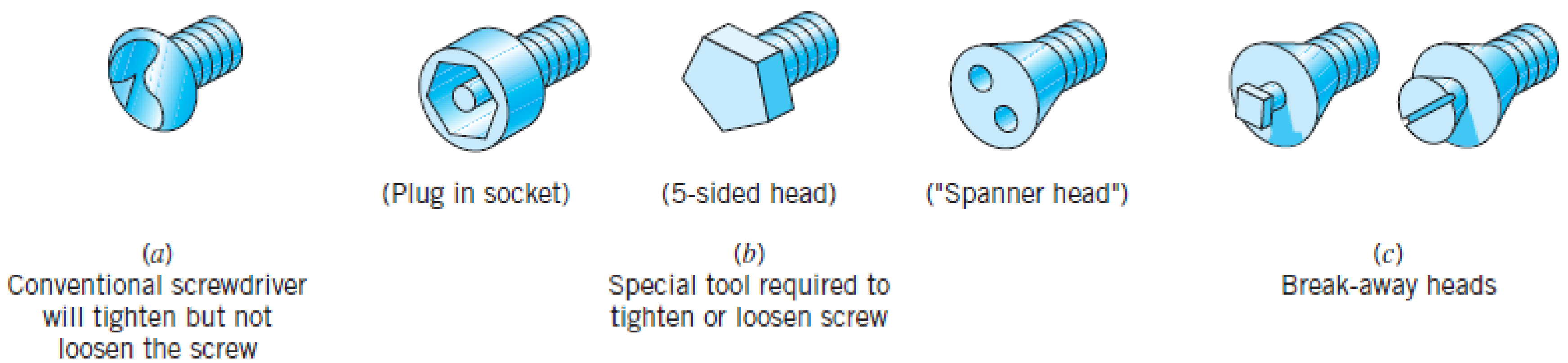
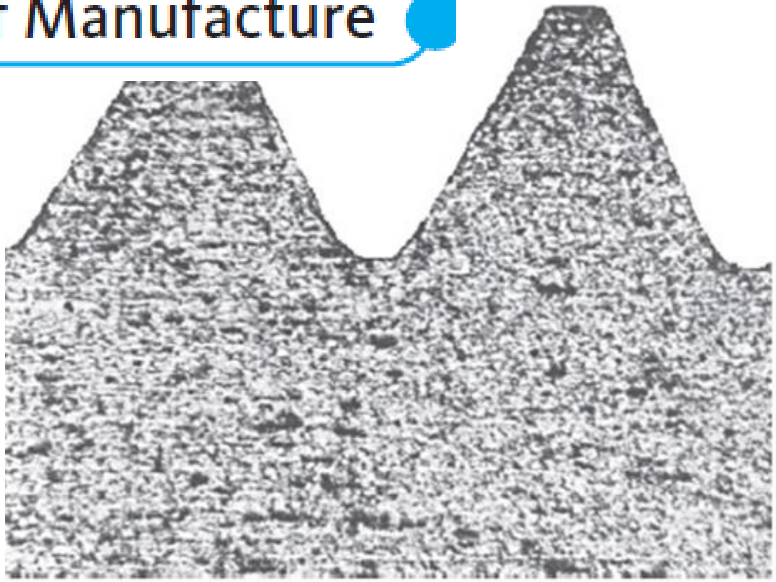


FIGURE 10.17
"Tamper-resistant" screw heads.

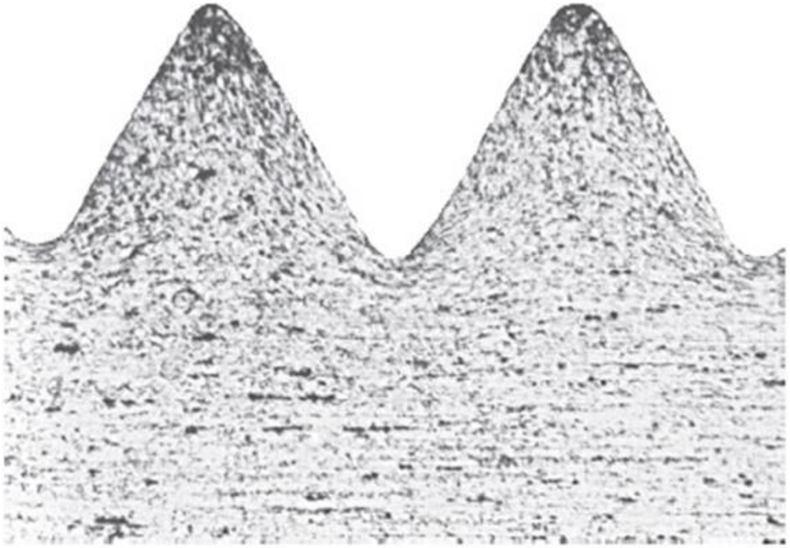
10.6 Fastener Materials and Methods of Manufacture

- Thread Cutting
- Thread rolling/forming
- Head forming in upsetting process
- Nuts: chipless tapping

- Mostly made of steel
- Specifications standardized as in tables 10.4 and 10.5
- Aluminum is also common
- Rolled threads are stronger than cut threads and in case of higher loads, rolled threads should be used



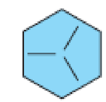
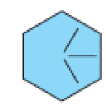
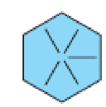
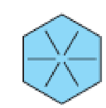
(*a*) Cut threads



(*b*) Rolled threads

10.6 Fastener Materials and Methods of Manufacture

Table 10.4 Specifications for Steel Used in Inch Series Screws and Bolts

SAE Grade	Diameter d (in.)	Proof Load (Strength)[a] S_p (ksi)	Yield Strength[b] S_y (ksi)	Tensile Strength S_u (ksi)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max	Grade Identification Marking on Bolt Head
1	$\frac{1}{4}$ thru $1\frac{1}{2}$	33	36	60	18	35	B70	B100	None
2	$\frac{1}{4}$ thru $\frac{3}{4}$	55	57	74	18	35	B80	B100	None
2	Over $\frac{3}{4}$ to $1\frac{1}{2}$	33	36	60	18	35	B70	B100	None
5	$\frac{1}{4}$ thru 1	85	92	120	14	35	C25	C34	
5	Over 1 to $1\frac{1}{2}$	74	81	105	14	35	C19	C30	
5.2	$\frac{1}{4}$ thru 1	85	92	120	14	35	C26	C36	
7	$\frac{1}{4}$ thru $1\frac{1}{2}$	105	115	133	12	35	C28	C34	
8	$\frac{1}{4}$ thru $1\frac{1}{2}$	120	130	150	12	35	C33	C39	

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.

[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.

Source: Society of Automotive Engineers standard J429k (1979).

10.6 Fastener Materials and Methods of Manufacture

TABLE 10.5 Specifications for Steel Used in Millimeter Series Screws and Bolts

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34
9.8	1.6 thru 16	650	—	900	—	—	C27	C36
10.9	6 thru 36	830	940	1040	9	35	C33	C39
12.9	1.6 thru 36	970	1100	1220	8	35	C38	C44

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J1199 (1979).

10.7 Bolt Tightening and Initial Tension

- Screws and nut-bolt assemblies should ideally be tightened with an initial tensile force F_i nearly = full "proof load," which is the maximum tensile force that does not produce a normally measurable permanent set. (This is < the tensile force producing a 0.2 percent offset elongation associated with S_y)

- On this basis initial tensions are specified in accordance with the equation

$$F_i = K_i A_t S_p \tag{10.11}$$

- where A_t is the tensile stress area of the thread, S_p is the "proof strength" of the material (Tables 10.4 and 10.5), and K_i is a constant (0.75 to 1.0).

- For ordinary applications involving static loading, let $K_i \cong 0.9$, or

$$F_i = 0.9 A_t S_p \tag{10.11a}$$

1. For loads tending to separate rigid members, the bolt load cannot be increased very much unless the members do actually separate, and the higher the initial bolt tension, the less likely the members are to separate.

2. For loads tending to shear the bolt, the higher the initial tension the greater the friction forces resisting the relative motion in shear.

10.7 Bolt Tightening and Initial Tension

- Tightening of a imparts torsional stress to bolt, along with the initial tensile stress.
- During initial use, the bolt usually “unwinds” very slightly, relieving most of torsion.

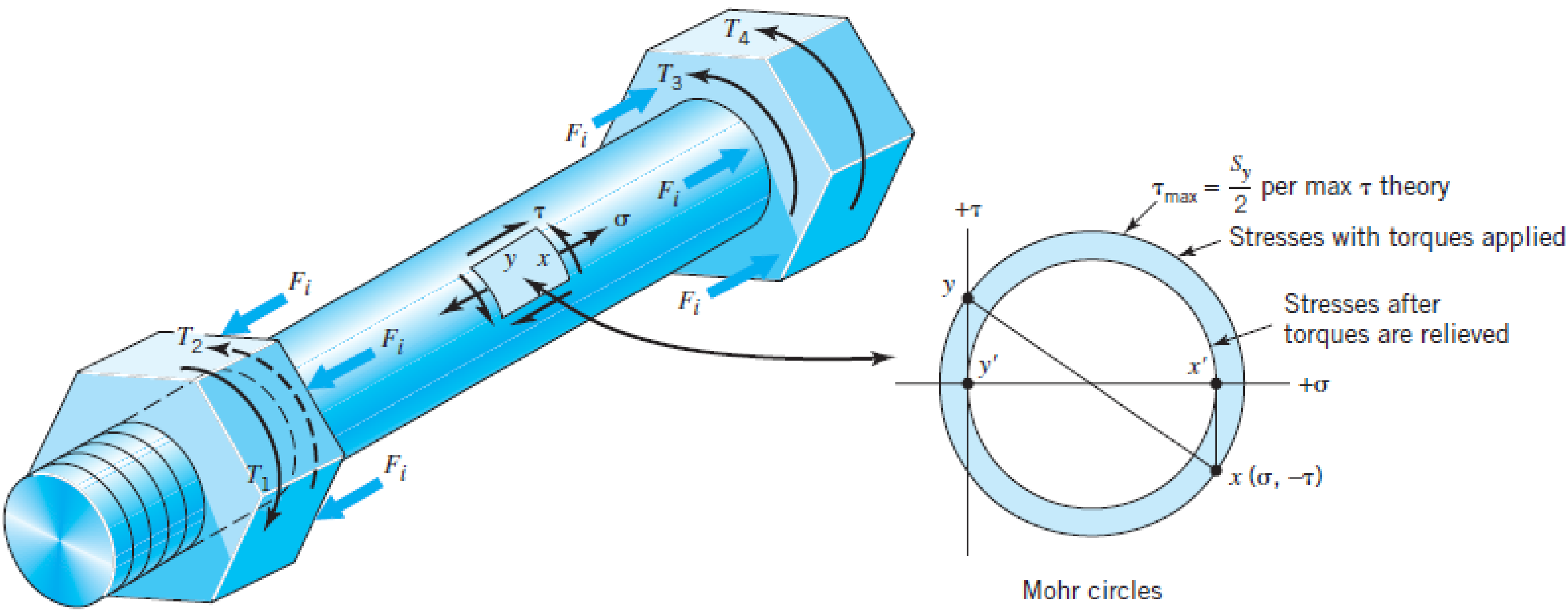


FIGURE 10.18
Bolt loads and stresses that are due to initial tightening of a nut. $\sum M = 0$ for the bolt and nut assembly shown; that is, $T_1 = T_2 + T_3 + T_4$ (where T_1 = nut wrench torque), T_2 = nut face friction torque $= fF_ir_n$ (where r_n is the effective radius of nut face friction forces), T_3 = bolt head friction torque $\leqq fF_ir_h$ (where r_h is the effective radius of bolt head friction forces), T_4 = wrench torque required to keep bolt head from turning. Note that $T_4 = 0$ if $fF_ir_h > T_1 - T_2$.

10.7 Bolt Tightening and Initial Tension

- the initial tension that can be achieved with a given bolt
- the amount of elongation that can be achieved before over tightening fractures the bolt.
- Accurate determination of bolt tensile load during tightening is difficult (micrometer or drilling and strain guage)
- The most common method of tightening a bolt a measured amount is probably to use a torque wrench.
- Accuracy limited. Normal torque wrench controls initial tension within ±30%; with special care, ± 15% is reasonable.

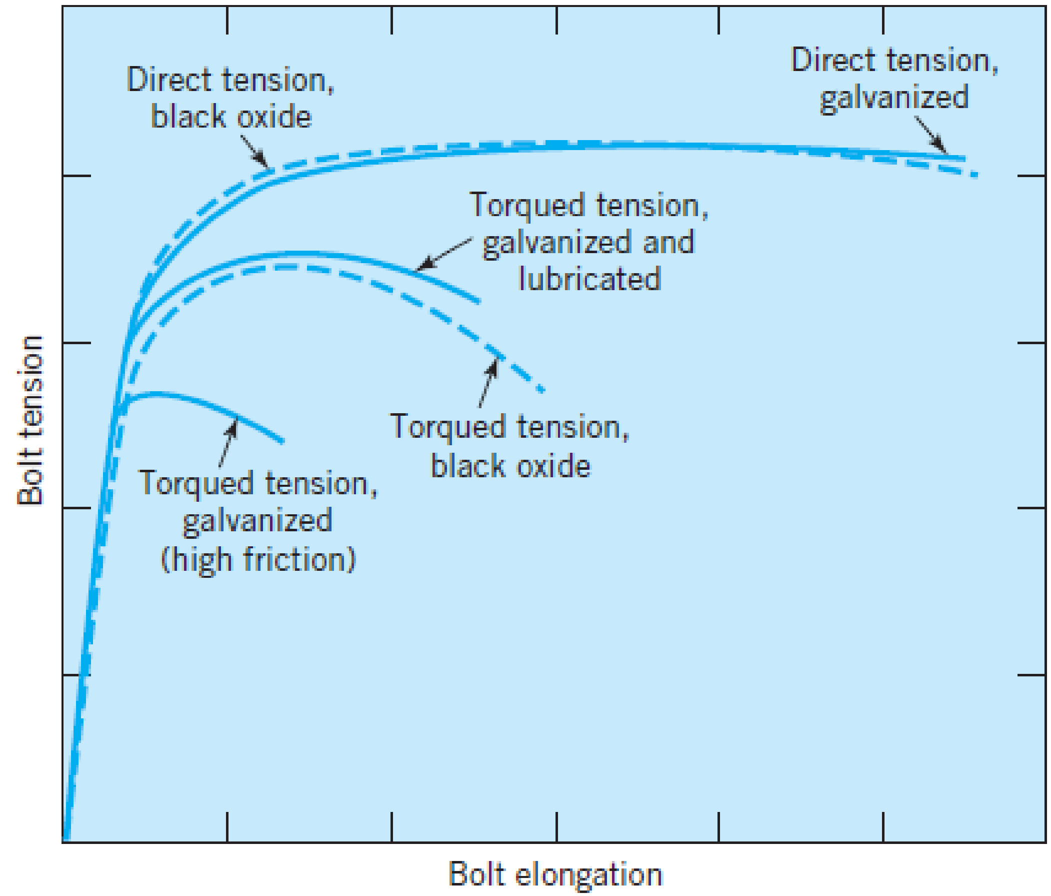


Figure 10.19
Bolt tension versus elongation, resulting from tightening by torquing versus direct tensioning, and for black oxide versus galvanized surfaces [5]. (Note: Direct tension is produced by hydraulic loading; hence, no torsional stresses are produced.)

10.7 Bolt Tightening and Initial Tension

- An equation relating torque to initial tension can be from Eq. 10.4 by recognizing that load W of a screw jack as to F_i for a bolt, and that collar friction in the jack as friction on the flat surface of the nut.

- When we use 0.15 for both f and f_c , in Eq. 10.4, for standard screw threads,

$$T = 0.2F_i d \qquad (10.12)$$

- where d is the nominal major diameter of the thread.

- This is approximate relationship, on “average” conditions of thread friction.

- A common way to tighten a screw or nut is

$$F_i\,(\text{lb}) \approx 16{,}000d\,(\text{in.}) \qquad (e)$$

- While the tension increases with d^2 and torsion with d^3 the F_i is dependent on d; So small bolts twist and large bolts remain undertightened

- When “rigid” parts are bolted, the elastic deflection of the parts <.01mm. Should the loading cause any creep, much of the bolt initial tension will be lost.

- 5% lost in first few minutes and another 5% lost in next few weeks

10.8 Thread Loosening and Thread Locking

- The following are among the factors influencing whether or not threads loosen.

1. The greater the helix angle (i.e., the greater the slope of the inclined plane), the greater the loosening tendency. Thus, coarse threads tend to loosen more easily than fine threads.

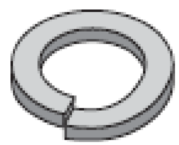
2. The greater the initial tightening, the greater the frictional force that must be overcome to initiate loosening.

3. Soft or rough clamping surfaces tend to promote slight plastic flow which decreases the initial tightening tension and thus promotes loosening.

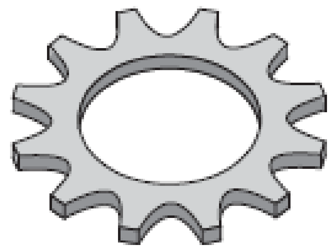
4. Surface treatments and conditions that tend to increase the friction coefficient provide increased resistance to loosening.

- The problem of thread loosening has resulted in numerous and ingenious special designs and design modifications, and it continues to challenge the engineer to find effective and inexpensive solutions.

10.8 Thread Loosening and Thread Locking

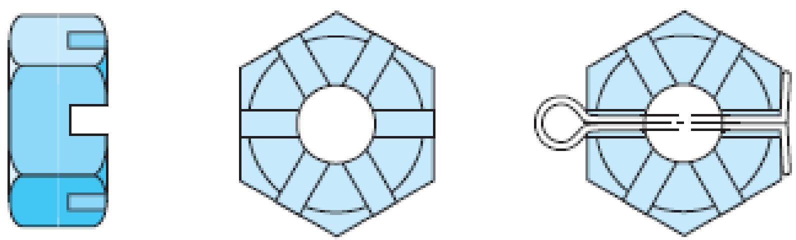


(*a*)
Helical (split) type

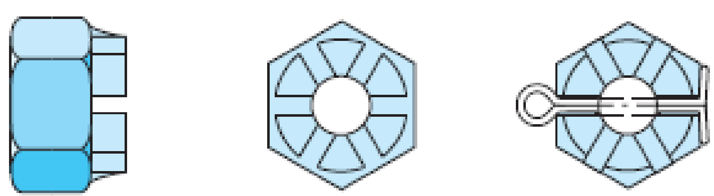


(*b*)
Twisted-tooth type
(Teeth may be external,
as in this illustration,
or internal.)

FIGURE 10.20
Common types of lock washers.



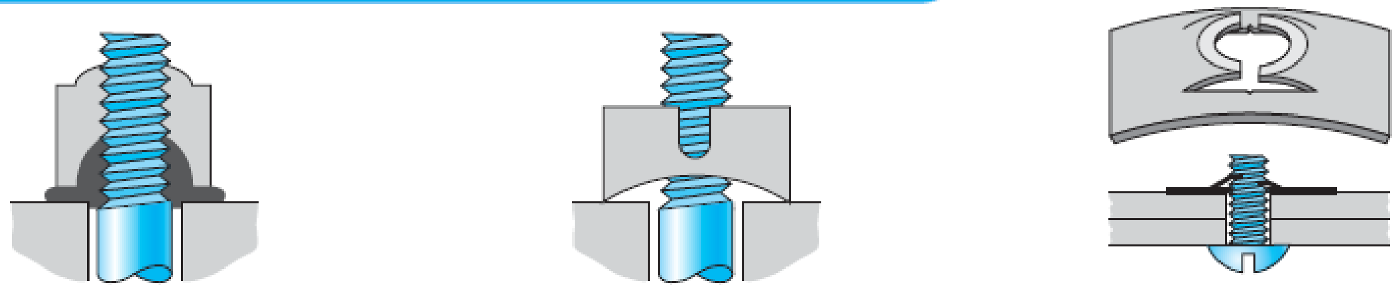
(*a*)



(*b*)

FIGURE 10.21
(*a*) Slotted and (*b*) castle nuts. Each is also shown with a drilled bolt and a cotter pin.

10.8 Thread Loosening and Thread Locking

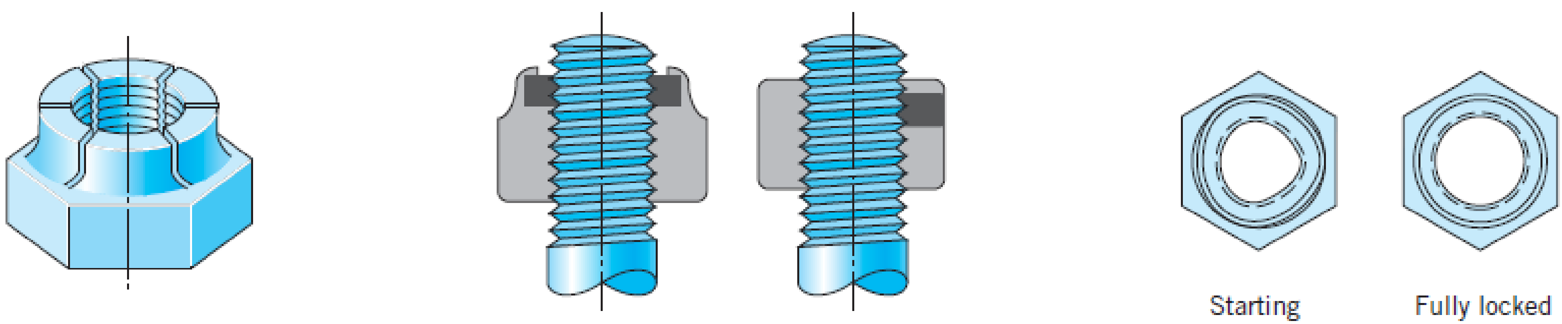


(*a*)
Insert nut (Nylon insert is compressed when nut seats to provide both locking and sealing.)

(*b*)
Spring nut (Top of nut pinches bolt thread when nut is tightened.)

(*c*)
Single thread nut (Prongs pinch bolt thread when nut is tightened. This type of nut is quickly applied and used for light loads.)

FIGURE 10.22
Examples of free-spinning locknuts.



Spring- top nut
(Upper part of nut is tapered. Segments press against bolt threads.)

(*a*)

Nylon-insert nuts
(Collar or plug of nylon exerts friction grip on bolt threads.)

(*b*)

Distorted nut (Portion of nut is distorted to provide friction grip on bolt threads.)

(*c*)

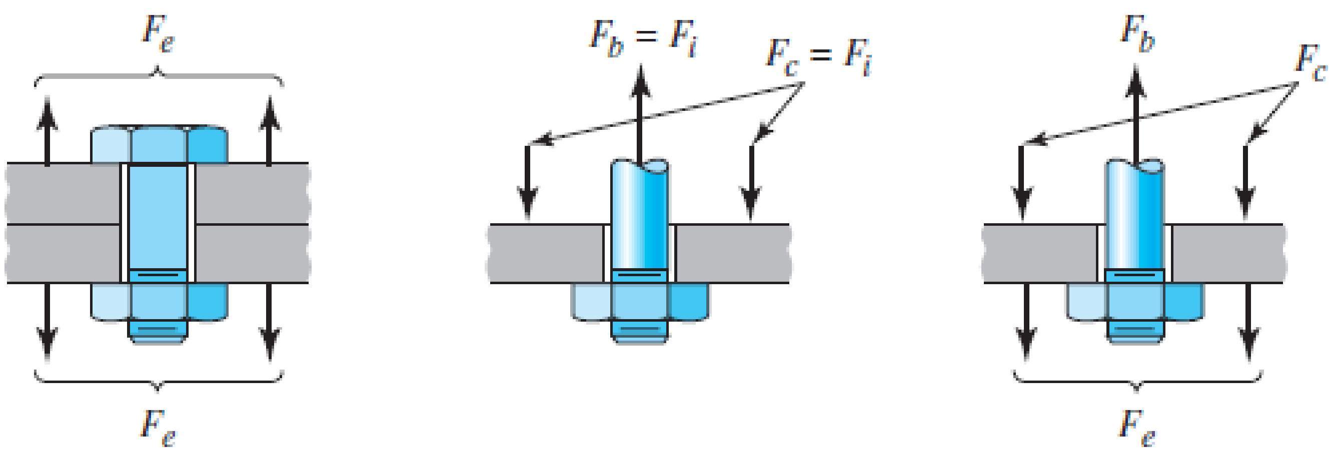
FIGURE 10.23
Examples of prevailing-torque locknuts. (Courtesy SPS Technologies, Inc.)

10.9 Bolt Tension with External Joint-Separating Force

- Bolts are typically used to hold parts together against to forces that pull, or slide
- Figure 10.24a shows the general case with external force F_e tending to separate
- Figure 10.24b shows a portion of this assembly as a free body. In this figure the nut has been tightened, but the external force has not yet been applied.
- The bolt axial load F_b = clamping force F_c = initial tightening force F_i .
- Figure 10.24c shows after F_e has been applied.
- Equilibrium considerations require one or both of the following:

1. an increase in F_b
2. a decrease in F_c .

- The relative magnitudes of the changes in F_b and F_c depend on the relative elasticities involved.



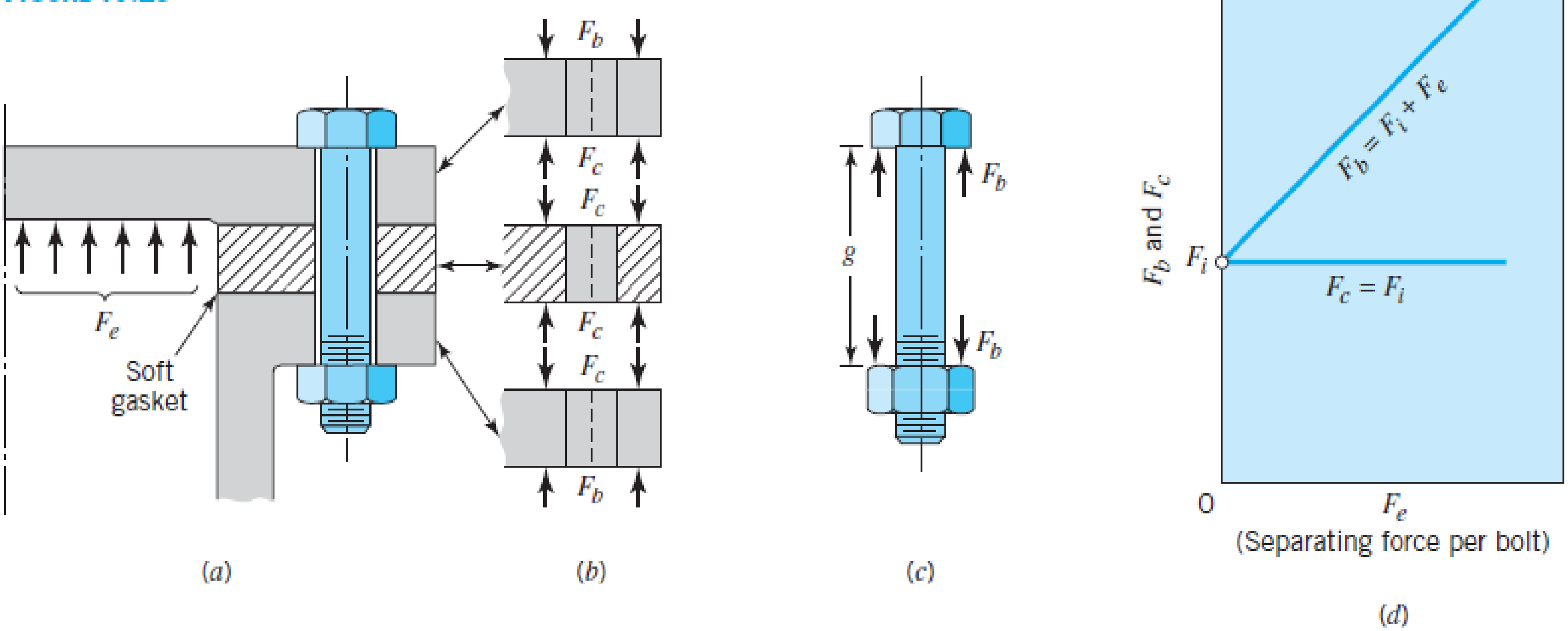
(a) Complete joint (b) Free body without external load (c) Free body with external load

FIGURE 10.24
Free-body study of bolt tensile loading.

10.9 Bolt Tension with External Joint-Separating Force

- Figure 10.25a shows a plate bolted on a pressure vessel with soft gasket so soft that the other parts can be considered infinitely rigid in comparison.
- When the nut is tightened to produce initial force F_i, the rubber gasket compresses; the bolt elongates negligibly.
- Figures 10.25b and 10.25c show details of the bolt and the clamped surfaces. Note the distance defined as the grip g. On initial tightening, $F_b = F_c = F_i$.
- Figure 10.25d shows the change in F_b and F_c as separating load F_e is applied.
- The elastic stretch of the bolt caused by F_e is so small. The clamping force F_c does not diminish and the entire load F_e goes to increasing bolt tension

FIGURE 10.25 F_b and F_c versus F_e per bolt for soft clamped members—rigid bolt.



10.9 Bolt Tension with External Joint-Separating Force

- Figure 10.26 illustrates the clamped members are “rigid” with precision-ground mating surfaces and no gasket, The bolt has a center portion made of rubber.
- Here the initial tightening stretches the bolt; it does not significantly compress the clamped members. (Sealing accomplished by a rubber O-ring).
- Figure 10.26d shows F_e is balanced by reduced F_c without increase in F_b.
- The only way the tension in the rubber bolt can be increased is to increase its length, and this cannot happen without an external force great enough to separate physically the mating clamped surfaces. (Note also that as long as the mating surfaces remain in contact, the sealing of the O-ring is undiminished.)

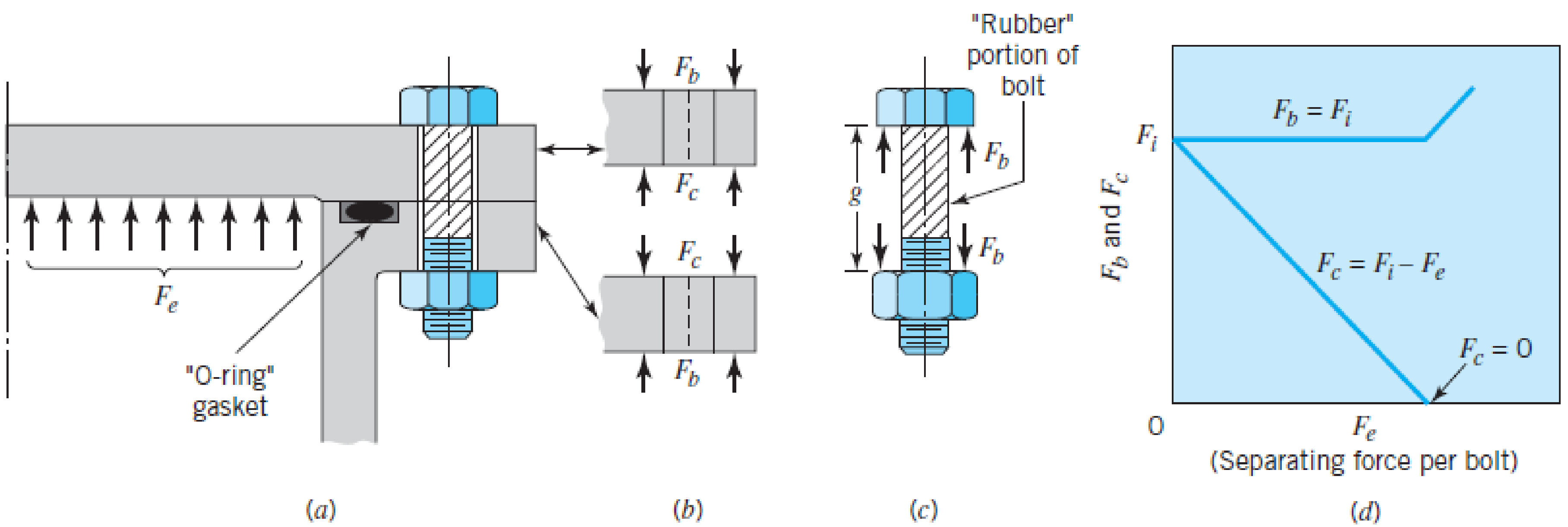


FIGURE 10.26

F_b and F_c versus F_e per bolt for rigid clamped members—soft bolt.

10.9 Bolt Tension with External Joint-Separating Force

- The extreme cases can be only approximated.
- In the realistic case in which both the bolt and the clamped members have applicable stiffness. Joint tightening both elongates the bolt and compresses the clamped members.
- When F_e is applied, the bolt and clamped members elongate by δ (g + δ for both)
- From Figure 10.24 the F_e = increased F_b + the decreased F_c, or

$$F_e = \Delta F_b + \Delta F_c \tag{f}$$

$$\Delta F_b = k_b \delta \quad \text{and} \quad \Delta F_c = k_c \delta \tag{g}$$

- Where k_b and k_c are spring constants of bolt and clamped material. So substituting

$$F_e = (k_b + k_c)\delta \quad \text{or} \quad \delta = \frac{F_e}{k_b + k_c} \tag{h}$$

$$\Delta F_b = \frac{k_b}{k_b + k_c} F_e \quad \text{and} \quad \Delta F_c = \frac{k_c}{k_b + k_c} F_e \tag{i}$$

- From figures 10.25 and 10.26

$$F_b = F_i + \frac{k_b}{k_b + k_c} F_e \quad \text{and} \quad F_c = F_i - \frac{k_c}{k_b + k_c} F_e \tag{10.13}$$

10.9 Bolt Tension with External Joint-Separating Force

1. When the external load is sufficient to bring the F_c to zero (A), $F_b = F_e$. So figure shows $F_c = 0$ and $F_b = F_e$ for F_e in excess of A.

2. When F_e is alternately dynamic, fluctuations of F_b and F_c can be found from figure

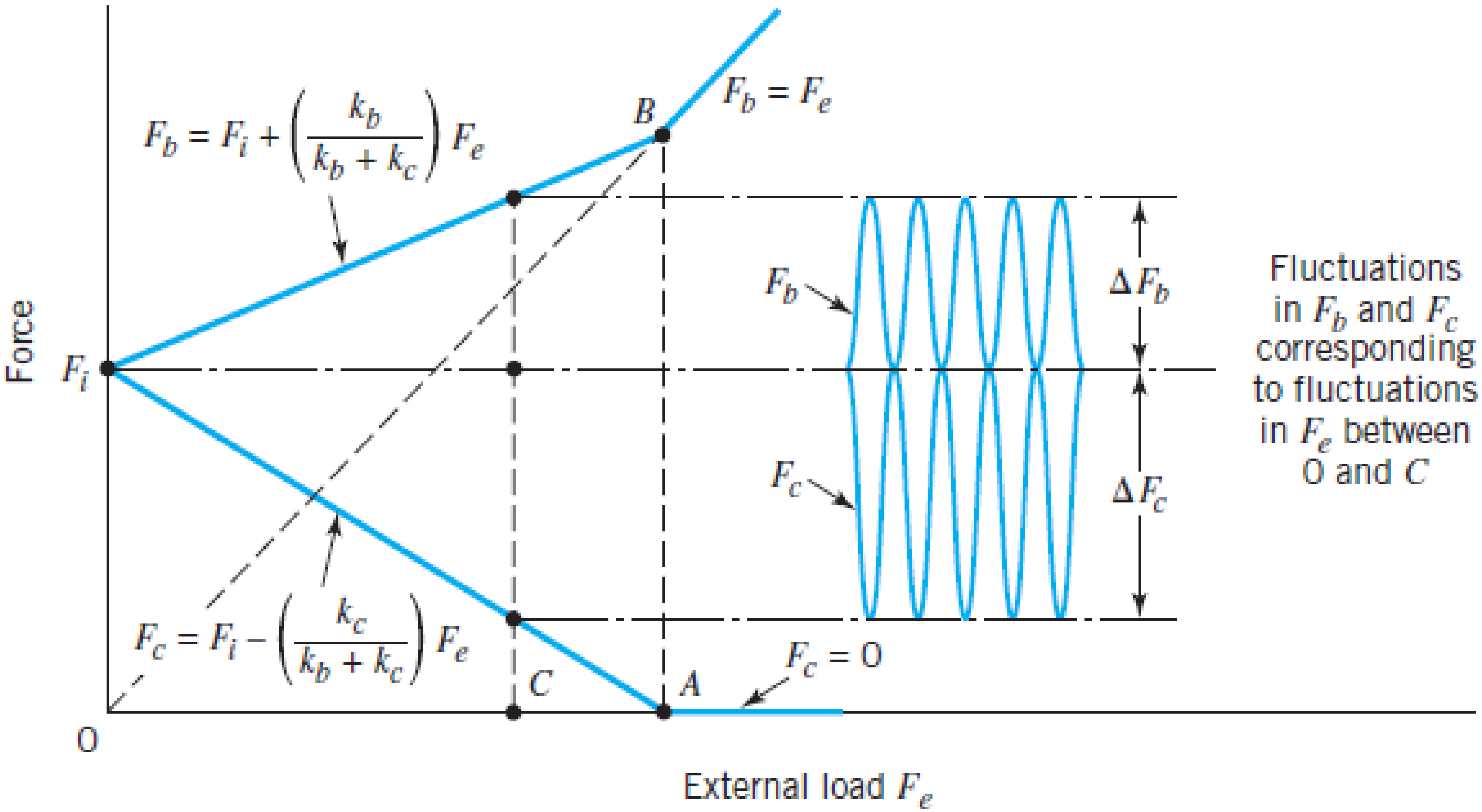


FIGURE 10.27
Force relationships for bolted connections.

$$F_b = F_i + \frac{k_b}{k_b + k_c} F_e \quad \text{and} \quad F_c = F_i - \frac{k_c}{k_b + k_c} F_e \qquad (10.13)$$

10.9 Bolt Tension with External Joint-Separating Force

- We need k_b and k_c. From the basic axial deflection ($\delta = PL/AE$) and for spring rate ($k = P/\delta$)

$$k_b = \frac{A_b E_b}{g} \quad \text{and} \quad k_c = \frac{A_c E_c}{g} \tag{10.14}$$

- where the grip g represents the effective length for both. Two difficulties that commonly arise in estimating k_c are

1. The clamped members may consist of a stack of different materials, representing "springs" in series. For this case,

$$1/k = 1/k_1 + 1/k_2 + 1/k_3 + \cdots \tag{10.15}$$

2. The effective CSA of the clamped members is not easy to determine. (irregular shapes, or if they extend a substantial distance from the bolt axis) An empirical procedure sometimes used to estimate A_c is illustrated in Figure.

- One method for estimating the effective area of clamped members (for calculating k_c). Effective area A_c is approximately equal to the average area of the dark grey section.

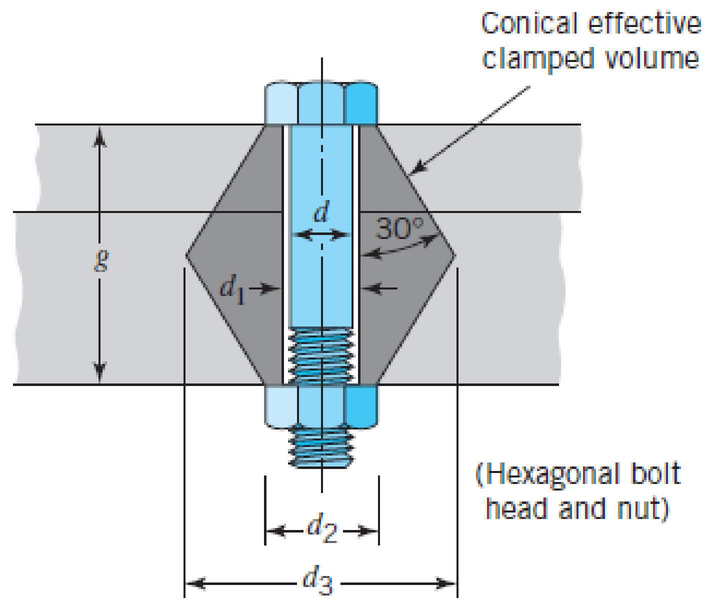


FIGURE 10.28

10.9 Bolt Tension with External Joint-Separating Force

$$A_c = \frac{\pi}{4}\left[\left(\frac{d_3 + d_2}{2}\right)^2 - d_1^2\right]$$

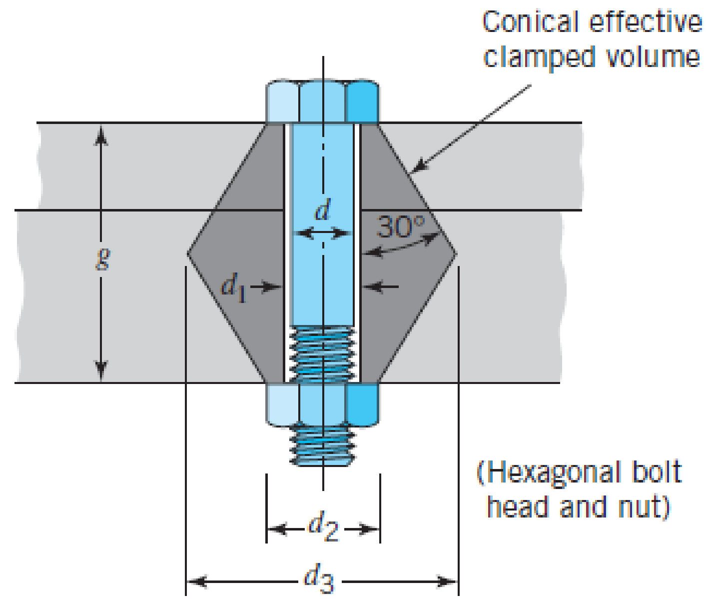


Figure 10.28

$d_1 \approx d$ (for small clearances)

$d_2 = 1.5d$ (for standard hexagonal-head bolts—see Figure 10.16)

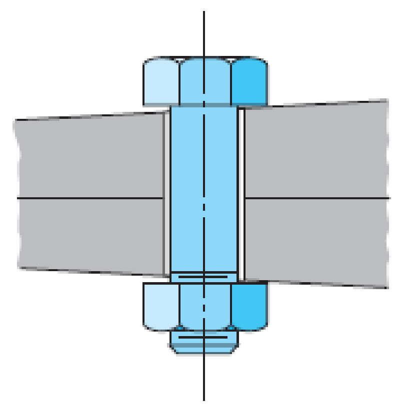
$d_3 = d_2 + g \tan 30° = 1.5d + g \tan 30°$

$$A_c = \frac{\pi}{16}(5d^2 + 6dg \tan 30° + g^2 \tan^2 30°) \approx d^2 + 0.68dg + 0.065g^2$$

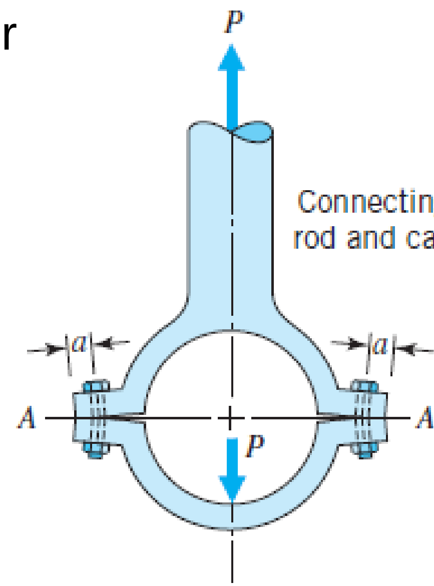
- An effective experimental procedure for determining the ratio of k_b and k_c for a given joint is to use a bolt equipped with an electric-resistance strain gage or to monitor bolt length ultrasonically.
- This permits a direct measurement of F_b both before and after F_e is applied.
- Some handbooks contain rough estimates of the ratio k_c/k_b for various general types of gasketed and ungasketed joints.
- For a “typical” ungasketed joint, k_c is sometimes taken as 3 k_b, but with careful joint design $k_c = 6k_b$.

10.10 Bolt (or Screw) Selection for Static Loading

- The primary loading applied to bolts is tensile, shear, or a combination of the two.
- Some bending is usually present because the clamped surfaces are not exactly parallel to each other and perpendicular to the bolt axis (Figure 10.29a) and because the loaded members are somewhat deflected (Figure 10.29b).
- Most times screws and bolts are selected rather arbitrarily. Such is the case with noncritical applications with small loads
- Almost any size would do, including sizes considerably smaller than the ones used.
- Selection is a matter of judgment, based on factors such as appearance, ease of handling and assembly, and cost.
- Even in bolt applications with known significant loads, larger bolts than necessary are used because a smaller size “doesn’t look right,” and the cost penalty of using the larger bolts is minimal.



(*a*)
Bolt bending caused by nonparallelism of mating surfaces. (Bolt will bend when nut is tightened.)



(*b*)
Bolt bending caused by deflection of loaded members. (Note tendency to pivot about *A*; hence, bending is reduced if dimension *a* is increased.)

FIGURE 10.29
Examples of nonintended bolt bending.

SAMPLE PROBLEM 10.2D Select Screws for Pillow Block Attachment—Tensile Loading

Figure 10.30 shows a ball bearing encased in a "pillow block" and supporting one end of a rotating shaft. The shaft applies a static load of 9 kN to the pillow block, as shown. Select appropriate metric (ISO) screws for the pillow block attachment and specify an appropriate tightening torque.

SOLUTION

Known: A known static tensile load is applied to two metric (ISO) screws.

Find: Select appropriate screws and specify a tightening torque.

Schematic and Given Data:

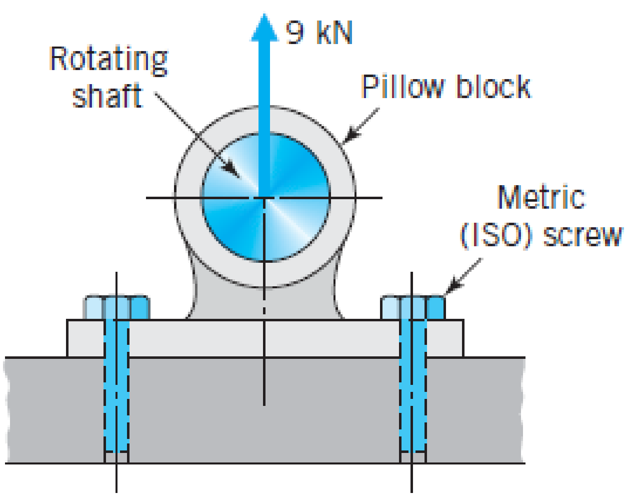


FIGURE 10.30
Pillow block attached by two machine screws.

Decisions/Assumptions:

1. A relatively inexpensive class 5.8 steel is chosen for the screw material.
2. The load of 9 kN is shared equally by each screw.
3. No bending of the machine screws (bolts) takes place; that is, the bolt load is axial tension.

Design Analysis:

1. Any class of steel could have been used, but there appears no reason to specify a costly high-strength steel. Class 5.8, with a proof strength of 380 MPa (Table 10.5), was chosen.
2. The nominal load for each of the two bolts is 4.5 kN. Reference to Section 6.12 indicates that if screw failure would not endanger human life, cause other damage, or entail costly shutdown, a safety factor of 2.5 would be reasonable. Since in this case the cost of using a larger safety factor is trivial, and since failure might prove rather costly, let us use "engineering judgment" and increase the safety factor to 4. Then, the "design overload" for each bolt is 4.5 kN $\times$ 4, or 18 kN.

10.6 Fastener Materials and Methods of Manufacture

Table 10.5 Specifications for Steel Used in Millimeter Series Screws and Bolts

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34
9.8	1.6 thru 16	650	—	900	—	—	C27	C36
10.9	6 thru 36	830	940	1040	9	35	C33	C39
12.9	1.6 thru 36	970	1100	1220	8	35	C38	C44

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J1199 (1979).

3. For static loading of a ductile material, stress concentration can be neglected and the simple "$\sigma = P/A$" equation used, with σ being equal to the proof strength when P is equal to the design overload:

$$380 \text{ MPa} = \frac{18{,}000\ N}{A_t} \quad \text{or} \quad A_t = 47.4 \text{ mm}^2$$

4. Reference to Table 10.2 indicates an appropriate standard size of class 5.8 screw to be M10 × 1.5 (for which $A_t = 58.0 \text{ mm}^2$).
5. Initial tightening tension might reasonably be specified (Eq. 10.11a) as

$$F_i = 0.9A_tS_p = 0.9(58.0 \text{ mm}^2)(380 \text{ MPa}) = 19{,}836 \text{ N}$$

6. This corresponds to an estimated tightening torque (Eq. 10.12) of

$$T = 0.2F_id = 0.2(19.8 \text{ kN})(10 \text{ mm}) = 39.6 \text{ N} \cdot \text{m}$$

TABLE 10.2 Basic Dimensions of ISO Metric Screw Threads

	Coarse Threads			Fine Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

Note: Metric threads are identified by diameter and pitch as "M8 × 1.25."

The *shear strengths* of steel bolts of various grades was studied by Fisher and Struik [5], who concluded that a reasonable approximation is

$$S_{us} \approx 0.62 S_u \text{ For direct (not torsional) shear loading.} \tag{10.16}$$

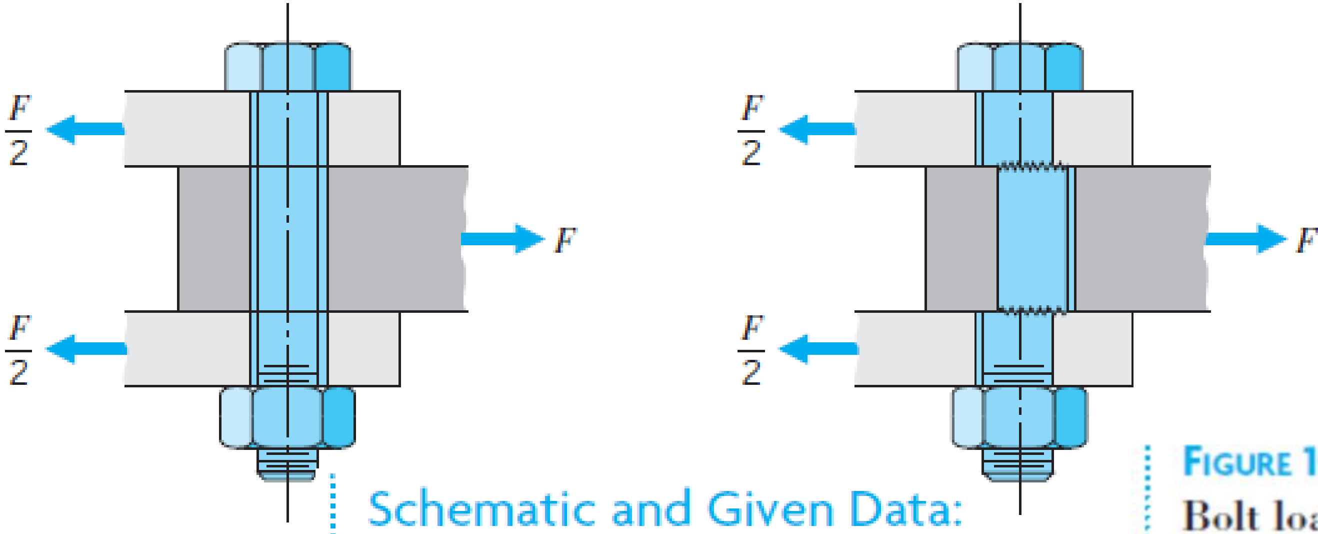
SAMPLE PROBLEM 10.3 Determine Shear Load Capacity of a Bolted Joint

Figure 10.31 shows a $\frac{1}{2}$ in.–13UNC grade 5 steel bolt loaded in double shear (i.e., the bolt has two shear planes, as shown). The clamped plates are made of steel and have clean and dry surfaces. The bolt is to be tightened with a torque wrench to its full proof load; that is, $F_i = S_p A_t$. What force F is the joint capable of withstanding? (Note: This double shear bolt loading is the same as that on the pin in Figure 2.14. It is assumed that the bolt and plates have adequate strength to prevent the other failure modes discussed in connection with Figures 2.14 and 2.15.)

SOLUTION

Known: A specified steel bolt clamps three steel plates and is loaded in double shear.

Find: Determine the force capacity of the joint.



Schematic and Given Data:

(a)

Normal load, carried by friction forces

(b)

Overload, causing shear failure

Figure 10.31
Bolt loaded in double shear.

Assumptions:

1. The bolt is tightened to its full proof load; that is, $F_i = S_p A_t$.
2. The bolt fails in double shear.
3. The bolt and plates have adequate strength to prevent other failure modes.
4. The wrench-torque variation is roughly ± 30 percent.
5. There is a 10 percent initial loss in tension during the first few weeks of service (see Section 10.7).

Analysis:

1. For the $\frac{1}{2}$ in.–13UNC grade 5 steel bolt, Table 10.1 gives $A_t = 0.1419 \text{ in.}^2$ and Table 10.4 shows that $S_p = 85$ ksi. Specified initial tension is $F_i = S_p A_t = 85{,}000 \text{ psi} \times 0.1419 \text{ in.}^2 = 12{,}060$ lb. But with a roughly estimated ± 30 percent torque-wrench variation and 10 percent initial-tension loss during the first few weeks of service (see Section 10.7), a conservative assumption of working value of F_i is about 7600 lb.

2. Reference 5 gives a summary (p. 78) of friction coefficients obtained with bolted plates. The coefficient for semipolished steel is approximately 0.3, and for sand or grit-blasted steel approximately 0.5. Various paints, platings, and other surface treatments can alter the coefficient markedly, usually downward. Here a friction coefficient of 0.4 is assumed. This gives a force required to slip each of the two interfaces of $7600 \text{ lb} \times 0.4 = 3040$ lb. Thus, the value of F required to overcome friction is estimated to be in the region of 6000 lb.

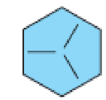
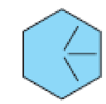
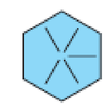
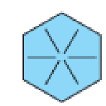
3. Although it is often desirable to limit applied load F to the value that can be transmitted by friction, we should know the larger value of force that can be transmitted through the bolt itself. For the two shear planes involved, this force is equal to $2S_{sy}A$, where A is the area of the bolt *at the shear planes*—in this case, $\pi(0.5)^2/4 = 0.196 \text{ in.}^2$. Taking advantage of the fact that the distortion energy theory gives a good estimate of shear yield strength for ductile metals, we have $S_{sy} = 0.58S_y = 0.58(92 \text{ ksi}) = 53$ ksi. Thus, for yielding of the two shear planes, $F = 2(0.196 \text{ in.}^2)(53{,}000 \text{ psi}) = 21{,}000$ lb.

TABLE 10.1 Basic Dimensions of Unified Screw Threads

		Coarse Threads—UNC			Fine Threads—UNF		
Size	Major Diameter d (in.)	Threads per Inch	Minor Diameter of External Thread d_r (in.)	Tensile Stress Area A_t (in.^{2})	Threads per Inch	Minor Diameter of External Thread d_r (in.)	Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
3(.099)	0.0990	48	0.0734	0.00487	56	0.0771	0.00523
4(.112)	0.1120	40	0.0813	0.00604	48	0.0864	0.00661
5(.125)	0.1250	40	0.0943	0.00796	44	0.0971	0.00830
6(.138)	0.1380	32	0.0997	0.00909	40	0.1073	0.01015
8(.164)	0.1640	32	0.1257	0.0140	36	0.1299	0.01474
10(.190)	0.1900	24	0.1389	0.0175	32	0.1517	0.0200
12(.216)	0.2160	24	0.1649	0.0242	28	0.1722	0.0258
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599

10.6 Fastener Materials and Methods of Manufacture

Table 10.4 Specifications for Steel Used in Inch Series Screws and Bolts

SAE Grade	Diameter d (in.)	Proof Load (Strength)[a] S_p (ksi)	Yield Strength[b] S_y (ksi)	Tensile Strength S_u (ksi)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max	Grade Identification Marking on Bolt Head
1	$\frac{1}{4}$ thru $1\frac{1}{2}$	33	36	60	18	35	B70	B100	None
2	$\frac{1}{4}$ thru $\frac{3}{4}$	55	57	74	18	35	B80	B100	None
2	Over $\frac{3}{4}$ to $1\frac{1}{2}$	33	36	60	18	35	B70	B100	None
5	$\frac{1}{4}$ thru 1	85	92	120	14	35	C25	C34	
5	Over 1 to $1\frac{1}{2}$	74	81	105	14	35	C19	C30	
5.2	$\frac{1}{4}$ thru 1	85	92	120	14	35	C26	C36	
7	$\frac{1}{4}$ thru $1\frac{1}{2}$	105	115	133	12	35	C28	C34	
8	$\frac{1}{4}$ thru $1\frac{1}{2}$	120	130	150	12	35	C33	C39	

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J429k (1979).

$$S_{us} \approx 0.62 S_u$$

4. The estimated 21,000-lb load would bring the shear stress to the yield strength over the entire cross section of the shear planes, and the very small amount of yielding would probably result in losing most or all of the clamping and friction forces. A further increase in load would cause total shear failure, as indicated in Figure 10.31*b*. This total failure load is calculated as in step 3, except for replacing S_{sy} with S_{us}. From Eq. 10.16, $S_{us} \approx 74$ ksi; the corresponding estimated load is $F = 29{,}000$ lb.

Comment: Note that in Figure 10.31 the threaded portion of the bolt does *not* extend to the shear plane. This is important for a bolt loaded in shear. Extending the thread to the shear plane is conservatively considered to reduce the shear area to a circle equal to the thread root diameter; in this case, $A = \pi(0.4056)^2/4 = 0.129$ in.^{2}, which is a reduction of 34 percent.

Sample Problem 10.4 Select Bolts for Bracket Attachment, Assuming Shear Carried by Friction

Figure 10.32 shows a vertically loaded bracket attached to a fixed member by three identical bolts. Although the 24-kN load is normally applied in the center, the bolts are to be selected on the basis that the load eccentricity shown could occur. Because of safety considerations, SAE class 9.8 steel bolts and a minimum safety factor of 6 (based on proof strength) are to be used. Determine an appropriate bolt size.

Solution

Known: Three SAE class 9.8 steel bolts with a specified safety factor are used to attach a bracket of known geometry that supports a known vertical load.

Find: Determine an appropriate bolt size.

Schematic and Given Data:

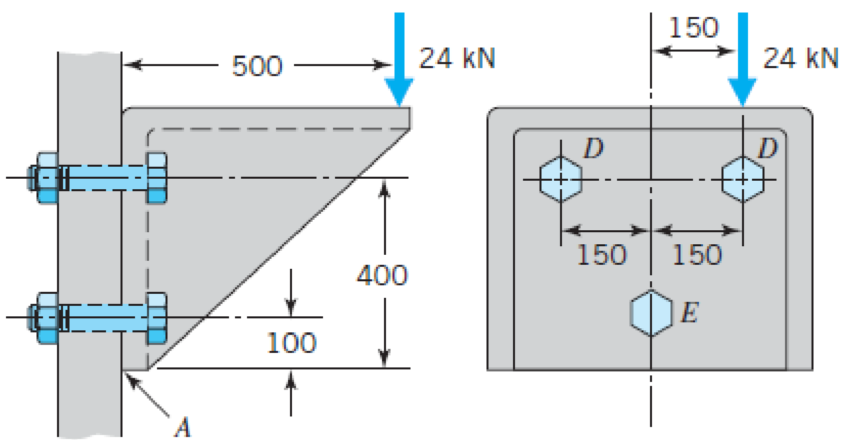


Figure 10.32
Vertically loaded bracket supported by three bolts.

Assumptions:

1. The clamped members are rigid and do not deflect with load.
2. The load tends to rotate the bracket about an axis through point A.
3. The shear loads are carried by friction.

Analysis:

1. With the assumptions of rigid clamped members and shear loads carried by friction, the eccentricity of the applied load has no effect on bolt loading. With the bracket tending to rotate about an axis through point A, the strain (and hence the load) imposed upon the two bolts D is four times that imposed upon bolt E. Let F_D and F_E denote the tensile loads carried by bolts D and E. Summation of moments about point A for the *design overload* of 24 kN(6) = 144 kN gives

$$\begin{aligned} 500(144) &= 100F_E + 400F_D + 400F_D \\ &= 25F_D + 400F_D + 400F_D = 825F_D \end{aligned}$$

   or

$$F_D = 87.27 \text{ kN}$$

2. Class 9.8 steel has a proof strength of 650 MPa. Hence the required tensile stress area is

$$A_t = \frac{87{,}270 \text{ N}}{650 \text{ MPa}} = 134 \text{ mm}^2$$

   Reference to Table 10.2 indicates the required thread size to be M16 $\times$ 2.

Comments:

1. Because of appearance, and to provide additional safety, a larger bolt size might be selected.
2. As in Sample Problem 10.2, the bolt size required is independent of k_b, k_c, and F_i, *except* for the fact that F_i must be large enough to justify the assumption that shear forces are transmitted by friction. With an assumed coefficient of friction of 0.4 and an initial tension (after considering tightening variations and initial relaxation) of at least $0.55S_pA_t$, compare the available shear friction force (using 16-mm bolts) with the applied shear overload:

$$\begin{aligned}\text{Available friction force} &= (3\text{ bolts})(0.55\,S_pA_t)F\\ &= 3(0.55)(650\text{ MPa})(157.27\text{ mm}^2)(0.4)\\ &= 67{,}500\text{ N}\end{aligned}$$

which represents a margin of safety with respect to the 24-kN applied overload, plus the rotational tendency caused by the overload eccentricity. The second effect is dealt with in Sample Problem 10.5.

Sample Problem 10.5 Select Bolts for Bracket Attachment, Neglecting Friction and Assuming Shear Forces Are Carried by the Bolts

Repeat Sample Problem 10.4, except neglect the frictional forces.

Solution

Known: Three SAE class 9.8 steel bolts having a specified safety factor are used to attach a bracket of known geometry that supports a known vertical load.

Find: Select an appropriate bolt size.

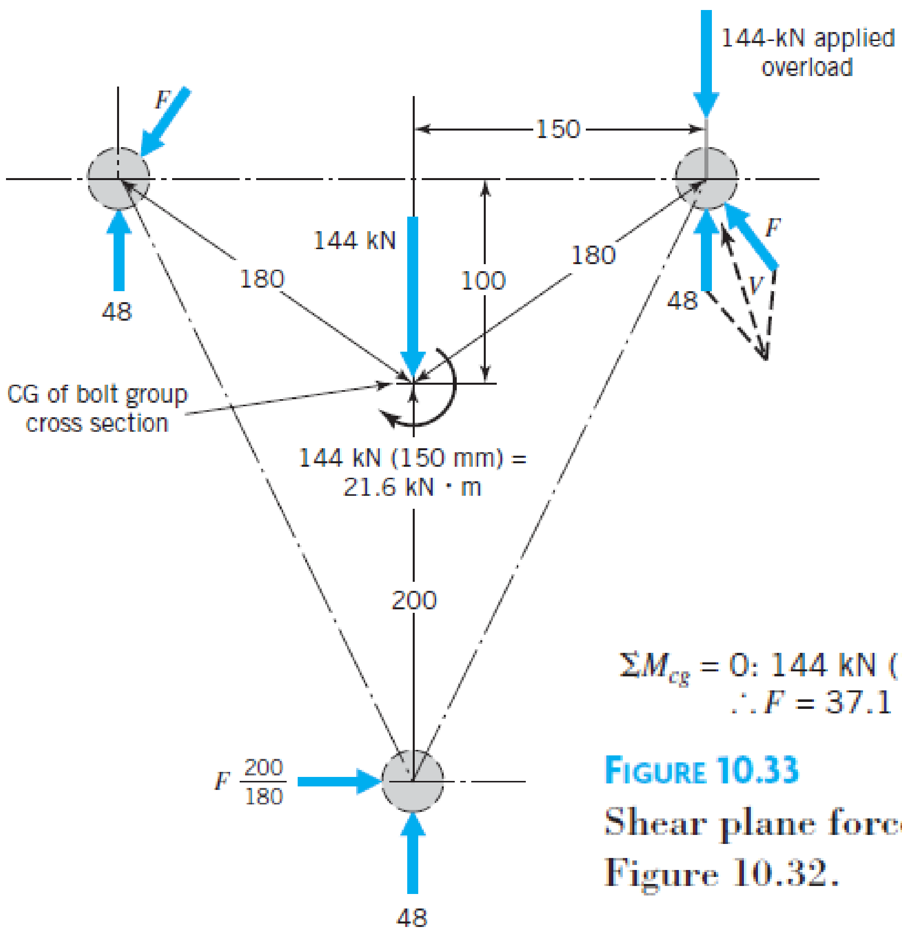
Schematic and Given Data: See Sample Problem 10.4 and Figure 10.32.

Assumptions:

1. The shear forces caused by the eccentric vertical load are carried completely by the bolts.
2. The vertical shear load is distributed equally among the three bolts.
3. The tangential shear force carried by each bolt is proportional to its distance from the center of gravity of the group of bolts.

Analysis:

1. Neglecting friction has no effect on bolt stresses in the *threaded region*, where attention was focused in Sample Problem 10.4. For this problem attention is shifted to the *bolt shear plane* (at the interface between bracket and fixed plate). This plane experiences the tensile force of 87.27 kN calculated in Sample Problem 10.4 in addition to the shear force calculated in the following step 2.
2. The applied eccentric shear force of 24 kN(6) = 144 kN tends to displace the bracket downward and also rotate it clockwise about the center of gravity of the bolt group cross section. For three bolts of equal size, the center of gravity corresponds to the centroid of the triangular pattern, as shown in Figure 10.33.



$$\Sigma M_{cg} = 0\text{: } 144\text{ kN }(150\text{ mm}) = F(180\text{ mm}) + F(180\text{ mm}) + F\left(\frac{200}{180}\text{ mm}\right)(200\text{ mm})$$
$$\therefore F = 37.1\text{ kN}$$

Figure 10.33

Shear plane force and moment equilibrium for the bracket in Figure 10.32.

This figure shows the original applied load (dotted vector) replaced by an equal load applied at the centroid (solid vector) plus a torque that is equal to the product of the force and the distance it was moved. As assumed, each bolt carries one-third of the vertical shear load, plus a tangential force (with respect to rotation about the center of gravity) that is proportional to its distance from the center of gravity. Calculations on the figure show this tangential force to be 37.1 kN for each of the top bolts. The vector sum of the two shear forces is obviously greatest for the upper right bolt. Routine calculation shows $V = 81.5$ kN.

3. The critical upper right bolt is thus subjected to a tensile stress, $\sigma = 87{,}270/A$, and a shear stress, $\tau = 81{,}500/A$. Substitution in the distortion energy equation gives an equivalent tensile stress of

$$\sigma_e = \sqrt{\sigma^2 + 3\tau^2} = \frac{1}{A}\sqrt{(87{,}270)^2 + 3(81{,}500)^2} = \frac{166{,}000}{A}$$

4. Equating this to the proof stress gives

$$\frac{166{,}000}{A} = S_p = 650 \text{ MPa}$$

Therefore,

$$A = 255 \text{ mm}^2$$

5. Finally,

$$A = \frac{\pi d^2}{4}, \qquad \text{or} \quad d = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(255)}{\pi}} = 18.03 \text{ mm}$$

Thus, a *shank* diameter of 18 mm is required.

Comment: In comparing this solution with that of Sample Problem 10.4, note that *for this particular case*, shear plus tension in the bolt shear plane proved to be more critical than tension alone in the threads.