Note Title

15.2 Power Screws

Power screws, also referred to as linear actuators, or translation screws, or lead screws, are used to convert rotary motion to relatively slow linear motion along the screw axis. The purpose of many power screws is to obtain a great mechanical advantage in lifting weight, as in screw-type jacks, or to exert large forces, as in presses and tensile test machines, and C-clamps. The purpose of others such as micrometer screws or the lead screw of a lathe, is to obtain precise positioning of the axial movement.

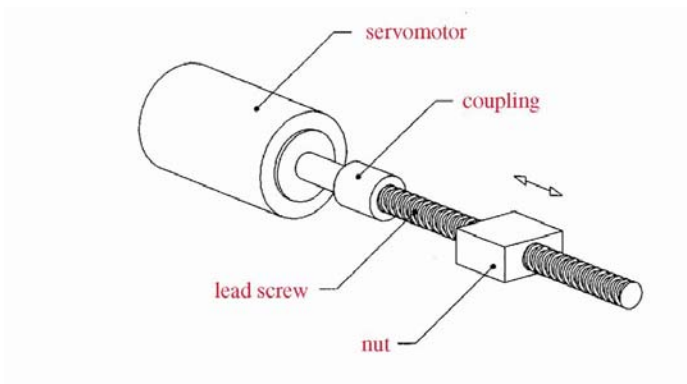


Figure 14-5
Servomotor-Driven Lead Screw for Use as a Positioning Device *Courtesy of J. Karsberg, Gill*

Square, Acme, and Buttress Threads

These are typical thread forms for power screws and are shown in Fig. 15-3a.

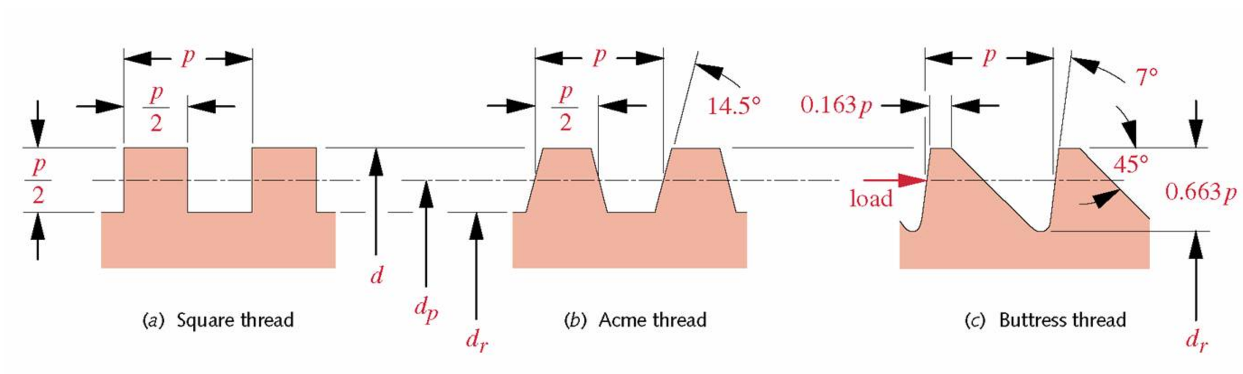


Figure 14-3
Square, Acme, and Buttress Threads.

Square threads provide the greatest strength and efficiency and eliminate any radial component of force between the screw and nut. However, they are more difficult to cut. The Acme thread has a 29° included angle, making it easier to manufacture. The Acme thread is a common choice if power screw must take load in both directions. If the axial load on the screw is unidirectional, the buttress thread is used.

Table 15-3 shows some principal dimensions for standard Acme threads.

Major Diameter (in)	Threads per inch	Thread Pitch (in)	Pitch Diameter (in)	Minor Diameter (in)	Tensile Stress Area (in^2)
0.250	16	0.063	0.219	0.188	0.032
0.313	14	0.071	0.277	0.241	0.053
0.375	12	0.083	0.333	0.292	0.077
0.438	12	0.083	0.396	0.354	0.110
0.500	10	0.100	0.450	0.400	0.142
0.625	8	0.125	0.563	0.500	0.222
0.750	6	0.167	0.667	0.583	0.307
0.875	6	0.167	0.792	0.708	0.442
1.000	5	0.200	0.900	0.800	0.568
1.125	5	0.200	1.025	0.925	0.747
1.250	5	0.200	1.150	1.050	0.950
1.375	4	0.250	1.250	1.125	1.108
1.500	4	0.250	1.375	1.250	1.353
1.750	4	0.250	1.625	1.500	1.918
2.000	4	0.250	1.875	1.750	2.580
2.250	3	0.333	2.083	1.917	3.142
2.500	3	0.333	2.333	2.167	3.976
2.750	3	0.333	2.583	2.417	4.909
3.000	2	0.500	2.750	2.500	5.412
3.500	2	0.500	3.250	3.000	7.670
4.000	2	0.500	3.750	3.500	10.321
4.500	2	0.500	4.250	4.000	13.364
5.000	2	0.500	4.750	4.500	16.800

Table 14-3

Principal Dimensions of American Standard Acme Threads.
See Reference 2 for More Complete Dimensional and Tolerance Information.

One possible application of a power screw as a jack for lifting a load is shown. There will be significant friction between the screw and the nut as well as the nut and base. Ball-thrust bearings are often used to reduce losses due to friction.

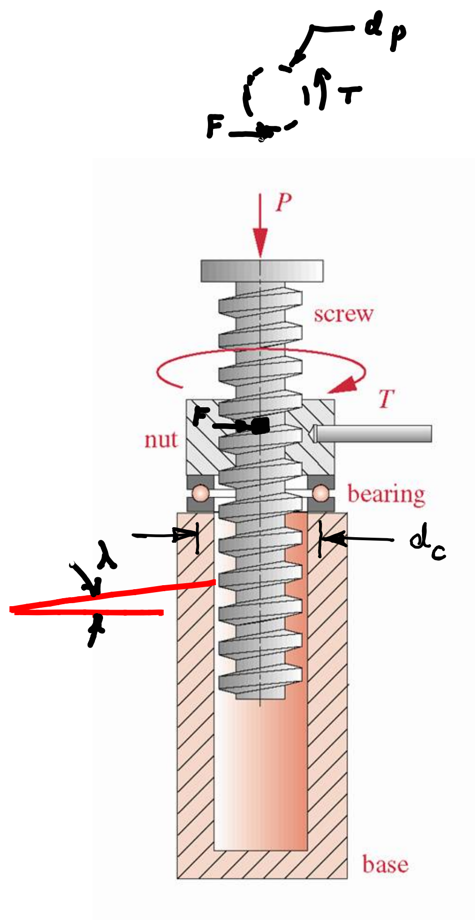
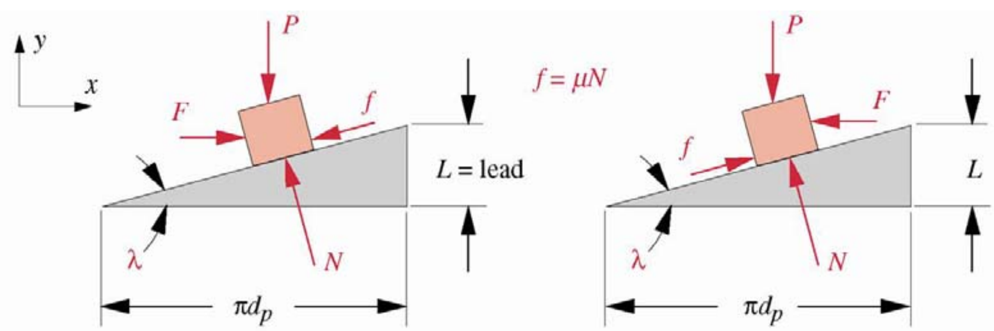


Figure 14-4
An Acme-Thread Power-Screw Jack.

Power Screw Force and Torque Analysis

Square Threads: A screw thread is essentially an inclined plane that has been wrapped around a cylinder to create a helix.

Unwrapping one revolution:



(a) Lifting the load up the plane (b) Lowering the load down the plane

Figure 14-6
Force Analysis at the Screw-Nut Interface.

where

- λ = **Lead angle**
- **F = Tangential force acting on the nut**
- **N = Normal force**
- **f = Friction force opposing the motion**
- μ = **Coefficient of friction between nut and screw**

Note that

$$\tan\lambda = \frac{L}{\pi d_p} \quad \text{and} \quad f = \mu N \qquad (15.3)_M$$

For lifting load:

$$\Sigma F_x = 0: \qquad F = N(\mu \cos\lambda + \sin\lambda) \qquad (15.4a)$$

$$\Sigma F_y = 0: \qquad N = \frac{P}{\cos\lambda - \mu \sin\lambda} \qquad (15.4b)$$

Combine (15.4 A) and (15.4b):

$$F = P \frac{\mu \cos\lambda + \sin\lambda}{\cos\lambda - \mu \sin\lambda} \qquad (15.4c)$$

The screw torque T_{Su} for lifting the load is:

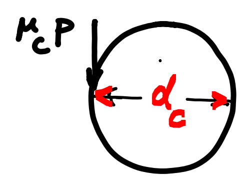
$$T_{s_u} = F\frac{d_p}{2} = \frac{P d_p}{2}\,\frac{\mu \cos\lambda + \sin\lambda}{\cos\lambda - \mu \sin\lambda} \qquad (15.4d)$$

To write this in terms of the lead, L, divide the numerator and the denominator by cos λ and use Eq. 15.3 to find

$$T_{s_u} = \frac{P d_p}{2}\,\frac{\mu \pi d_p + L}{\pi d_p - \mu L} \qquad (15.4e)$$

This torque is only to overcome the friction between the screw and the nut. The torque required to turn the thrust collar is

$$T_c = \mu_c P \frac{d_c}{2}$$



where

d_c = mean diameter of the thrust collar
m_c = coefficient of friction in the thrust bearing

The total torque T_u to lift the load with a square thread is

$$T_u = T_{su} + T_c = \frac{P d_p}{2}\,\frac{\mu \pi d_p + L}{\pi d_p - \mu L} + \mu_c P \frac{d_c}{2} \qquad (15.4g)$$

Similarly, we can repeat the above analysis to show that the total torque T_d to lower the load is

$$T_d = T_{s_d} + T_c = \frac{P d_p}{2}\,\frac{\mu \pi d_p - L}{\pi d_p + L} + \mu_c P \frac{d_c}{2} \qquad (15.4h)$$

Note: The so-called coefficient of "starting" friction is about 1/3 higher than the coefficient of "running" friction, e.g., if $\mu_{running}$ = 0.1, then $\mu_{starting}$ = 0.1+1/3(0.1) = 0.133.

Acme Threads: For this thread form, the normal force between screw and nut is angled in two planes: at the lead angle λ , and also at the α = 14.50° angle of the Acme thread as shown in Fig. 15-7.

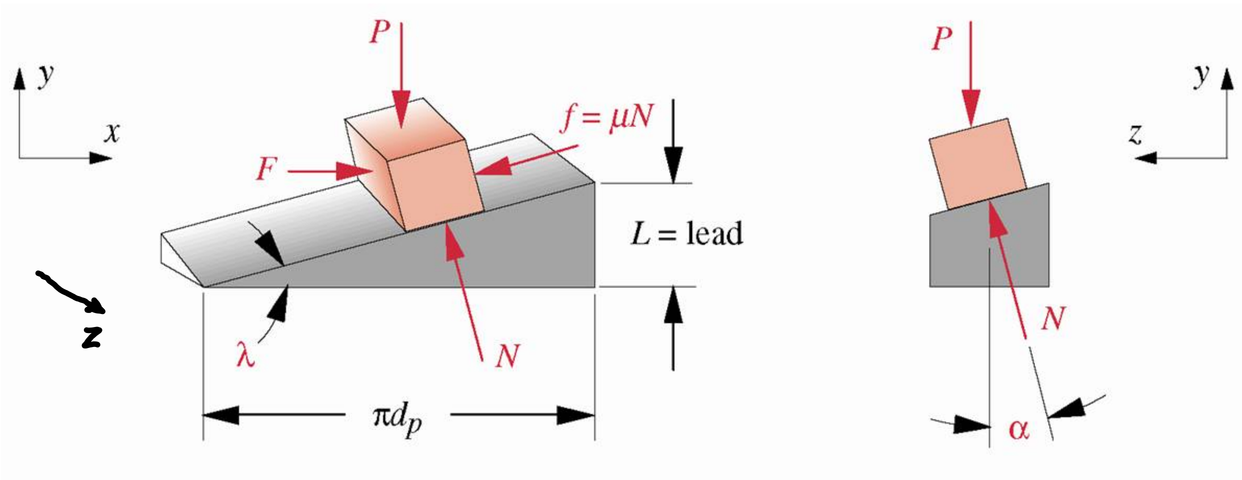


Figure 14-7

Force Analysis for an Acme-Thread Screw-Nut Interface.

If we require equilibrium of the block as we did for the square thread, we find

$$T_u = T_{s_u} + T_c = \frac{P d_p}{2} \frac{\mu \pi d_p + L \cos\alpha}{\pi d_p \cos\alpha - \mu L} + \mu_c P \frac{d_c}{2} \qquad (15.5a)$$

$$T_d = T_{s_d} + T_c = \frac{P d_p}{2} \frac{\mu \pi d_p - L \cos\alpha}{\pi d_p \cos\alpha + \mu L} + \mu_c P \frac{d_c}{2} \qquad (15.5b)$$

when α = 0 in these equations, they reduce to those for square thread.

Self-Locking and Back-Driving of Power Screws

Consider the torque required to lower a load, T_{Sd}:

$$T_{s_d} = \frac{P d_p}{2} \frac{\mu \pi d_p - L \cos\alpha}{\pi d_p \cos\alpha + \mu L} \qquad (15.6)$$

If $T_{Sd} < 0$, i.e.,

$$\mu \pi d_p < L \cos\alpha \qquad \text{or} \qquad \mu < \tan\lambda \cos\alpha$$

then the load P is lowered on its own. This is called back-driving or overhauling. If $T_{Sd} >= 0$, i.e.,

$$\mu \pi d_p \geq L \cos\alpha \qquad \text{or} \qquad \mu \geq \tan\lambda \cos\alpha \qquad (15.6a)$$

then the screw is called self-locking. This is useful in applications like jacks.

Caution: These are for **static** loading only. Vibration due to dynamic loading can cause a self-locking screw to back drive (overhaul).

Efficiency of Power Screws

In the **ideal** case that $\mu = 0$, we have (neglecting the collar friction term) from Eq. 15-5a:

$$T_o = \frac{PL}{2\pi}$$

The efficiency is defined as

$$e = \frac{T_o}{T} = \frac{PL}{2\pi T}$$

Another way to interprete this relation is:

> **PL** = Work output of a power screw for one revolution of the rotating member
> **$2\pi T$** = Work input

$$\therefore \quad \text{Efficiency}, \; e = \frac{W_{out}}{W_{in}} = \frac{PL}{2\pi T}$$

Substituting for T from 15.5a:

$$e = \frac{L}{\pi d_p} \frac{\pi d_p \cos\alpha - \mu L}{\pi \mu d_p + L \cos\alpha} \qquad (15.7d)$$

In terms of the lead angle, λ, from Eq. 15.3:

$$e = \frac{\cos\alpha - \mu \tan\lambda}{\cos\alpha + \mu \cot\lambda} \qquad (15.7e)$$

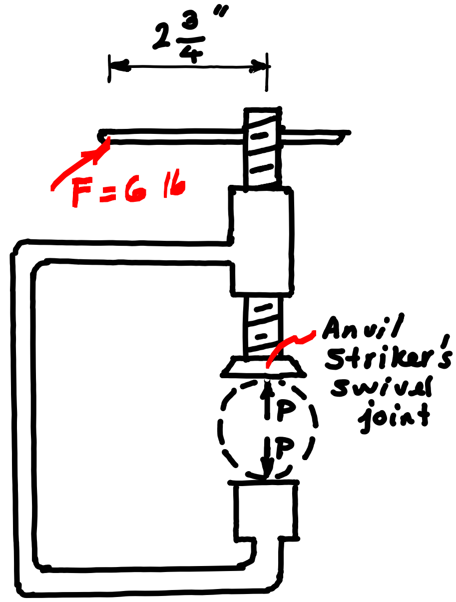
The efficiencies of standard Acme screws for an assumed friction coefficient of $\mu = 0.15$ are between 18% and 36% for a range of lead angles between 2° and 5° (see Table 15-4 in text).

Example: A 5/6"-6 Acme thread is used in a C-clamp. Given that

$$\mu = \mu_c = 0.15, \text{ and } d_c = 7/16",$$

find the clamping force, P.

(Note: The collar in this case is the anvil striker's swivel joint)



15.3 Stresses in Threads

Various stresses that act on power screws and threaded fasteners are considered here. Theoretically, when a nut engages a thread, all the threads should share the load. In reality,
however, inaccuracies in thread spacing cause practically all the load to be taken by the first threads. Thus, to be conservative, the worst case of one thread-pair taking the entire load is usually assumed.

Axial Stress

Power screws are subjected to both tension and compression. Fasteners are normally subjected only to tension:

$$\sigma = \frac{P}{A_t}$$

where A_t is the "tensile stress area" given in Tables 15-1, 15-2, and 15-3 of text.

For power screws subjected to compressive loads, if possible, design should be modified so that they are always in tension. If not, buckling analysis should be performed by, e.g., using J. B. Johnson's column formula.

Thread Shear ("Stripping") Stress

Here, a possible shear failure mode is stripping of the threads either out of the nut or off the screw (see text, pp.875).

Torsional Stress

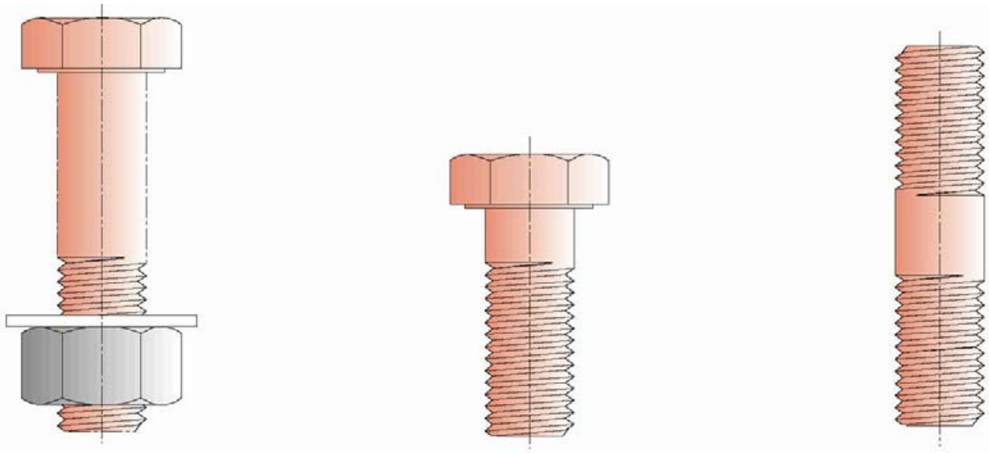
When a torque is transmitted through a nut, a torsional stress can be developed in the screw. we have

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

This accomodate for the worst-case of high thread friction at the screw-nut interface.

15-4 Types of Screw Fasteners

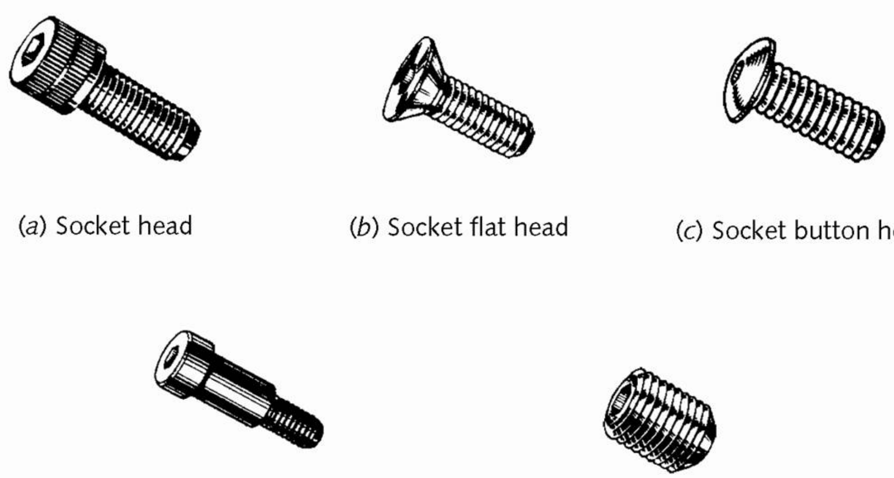
There is a wide variety of screws. They can be classified based on intended use:



(*a*) Bolt, washer, and nut (*b*) Machine (cap) screw (*c*) Stud

Figure 14-10
Bolt and Nut, Machine Screw and Stud.

Classification by head style:



(*a*) Socket head (*b*) Socket flat head (*c*) Socket button head

(*d*) Shoulder screw (*e*) Socket set screw

Figure 14-13
Various Styles of Socket-Head Cap Screws *Courtesy of Cordova Bolt Inc., Buena Park, Calif.*

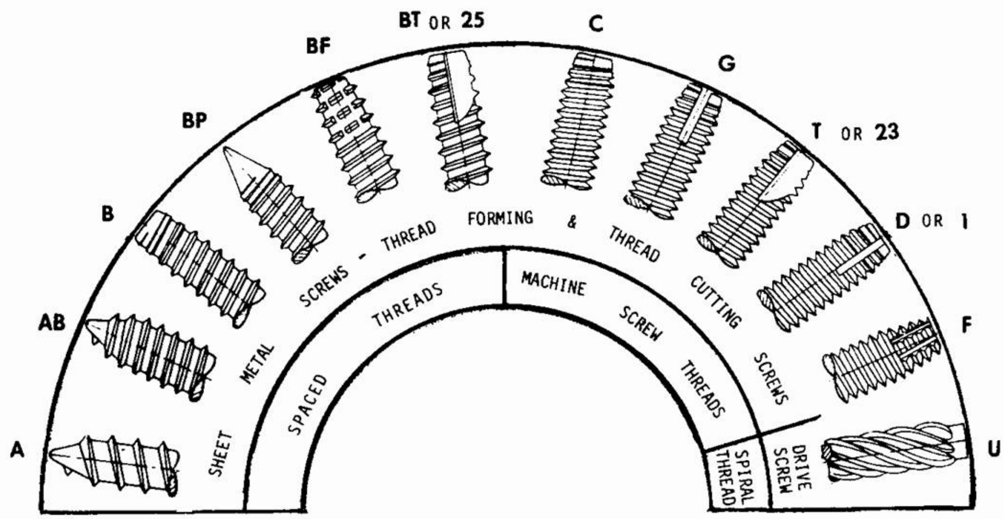


Figure 14-11
Various Styles of Threads Used on Tapping Screws *Courtesy of Cordova Bolt Inc., Buena Park, Calif. 90621.*

Nuts and washers

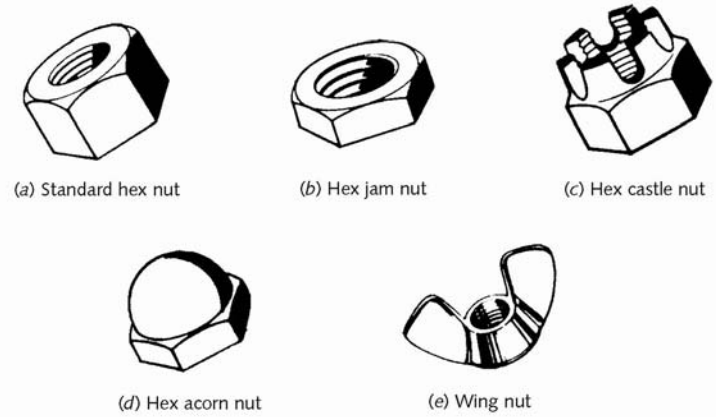


Figure 14-14
Some Styles of Standard Nuts *Courtesy of Cordova Bolt Inc., Buena Park, Calif. 90621.*

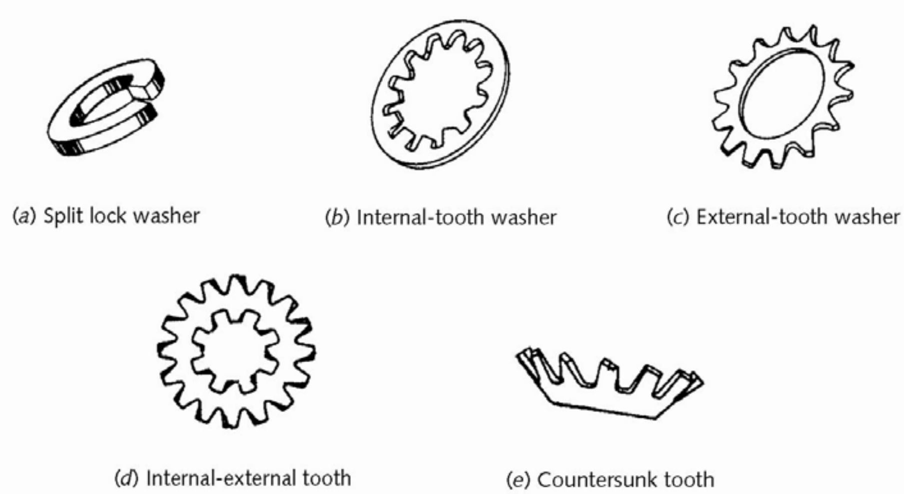


Figure 14-16
Types of Lock Washers *Courtesy of Cordova Bolt Inc., Buena Park, Calif.*

15.5 Manufacturing Fasteners (reading assignment)

15.6 Strengths of Bolts & Screws

Bolts and screws that support load are selected based on their "proof strength" S_p as defined in SAE, ASTM, or ISO specifications. The (minimum) proof strength, S_p, is the stress at which the bolt begins to take a permanent set, and is close to, but lower than, the yield strength of material (see, Table 15-6). The "proof load," $F_p = A_t S_p$.

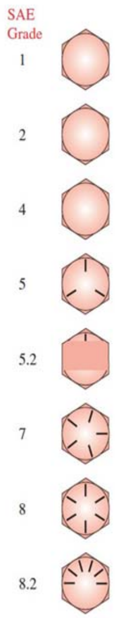


FIGURE 15-19
Head Marks for SAE Bolts

Table 15-6 SAE Specifications and Strengths for Steel Bolts

SAE Grade Number	Size Range Outside Diameter (in)	Minimum Proof Strength (kpsi)	Minimum Yield Strength (kpsi)	Minimum Tensile Strength (kpsi)	Material
1	0.25–1.5	33	36	60	low or medium carbon
2	0.25–0.75	55	57	74	low or medium carbon
2	0.875–1.5	33	36	60	low or medium carbon
4	0.25–1.5	65	100	115	medium carbon, cold drawn
5	0.25–1.0	85	92	120	medium carbon, Q&T*
5	1.125–1.5	74	81	105	medium carbon, Q&T
5.2	0.25–1.0	85	92	120	low-carbon martensite, Q&T
7	0.25–1.5	105	115	133	medium-carbon alloy, Q&T
8	0.25–1.5	120	130	150	medium-carbon alloy, Q&T
8.2	0.25–1.0	120	130	150	low-carbon martensite, Q&T

* Quenched and Tempered.

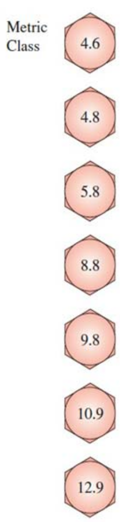


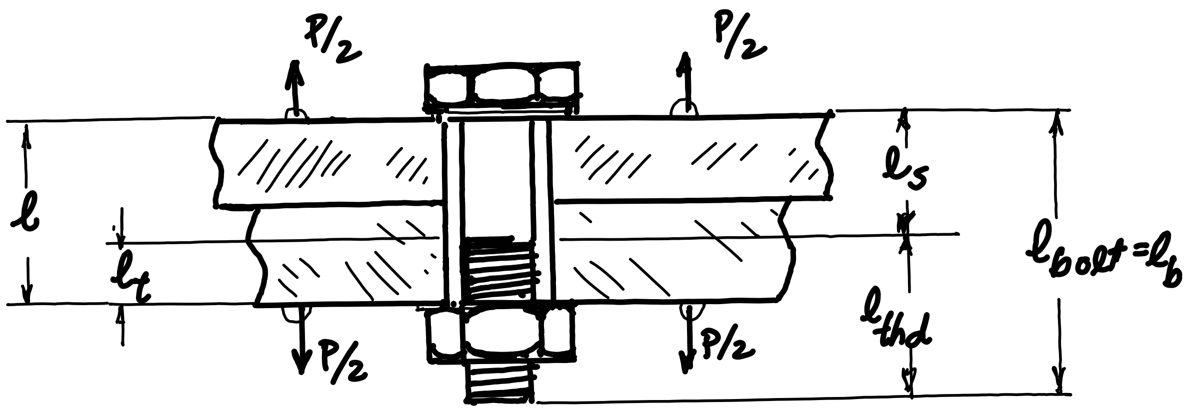
FIGURE 15-20
Head Marks—Metric Bolts

Table 15-7 Metric Specifications and Strengths for Steel Bolts

Class Number	Size Range Outside Diameter (mm)	Minimum Proof Strength (MPa)	Minimum Yield Strength (MPa)	Minimum Tensile Strength (MPa)	Material
4.6	M5–M36	225	240	400	low or medium carbon
4.8	M1.6–M16	310	340	420	low or medium carbon
5.8	M5–M24	380	420	520	low or medium carbon
8.8	M3–M36	600	660	830	medium carbon, Q&T
9.8	M1.6–M16	650	720	900	medium carbon, Q&T
10.9	M5–M36	830	940	1 040	low-carbon martensite, Q&T
12.9	M1.6–M36	970	1 100	1 220	alloy, quenched & tempered

15.7 Preloaded Fasteners in Tension

Bolts & nuts are often used for clamping parts that are being pulled apart by applied loads.



Note that for standard hexagon-head bolts we have the following relations:

For inch series:

$$l_{thd} = \begin{cases} 2d + \frac{1}{4} \text{ in} & \text{for } l \leq 6 \text{ in} \\ 2d + \frac{1}{2} \text{ in} & \text{for } l > 6 \text{ in} \end{cases} \quad (1)$$

For metric series:

$$l_{thd} = \begin{cases} 2d + 6 & \text{for } l \leq 125 \text{ mm}, d \leq 48 \text{ mm} \\ 2d + 12 & \text{for } 125 \text{ mm} < l \leq 200 \text{ mm} \\ 2d + 25 & \text{for } l > 200 \text{ mm} \end{cases} \quad (2)$$