Power Screws

Power screws of various descriptions are also commonly encountered machine components. Their engineering and design has much in common with the engineering and design of threaded fasteners.

Thread Forms, Terminology, and Standards

Figure (1) illustrates the basic arrangement of a helical thread wound around a cylinder, as used on screw-type fasteners, power screws, and worms.

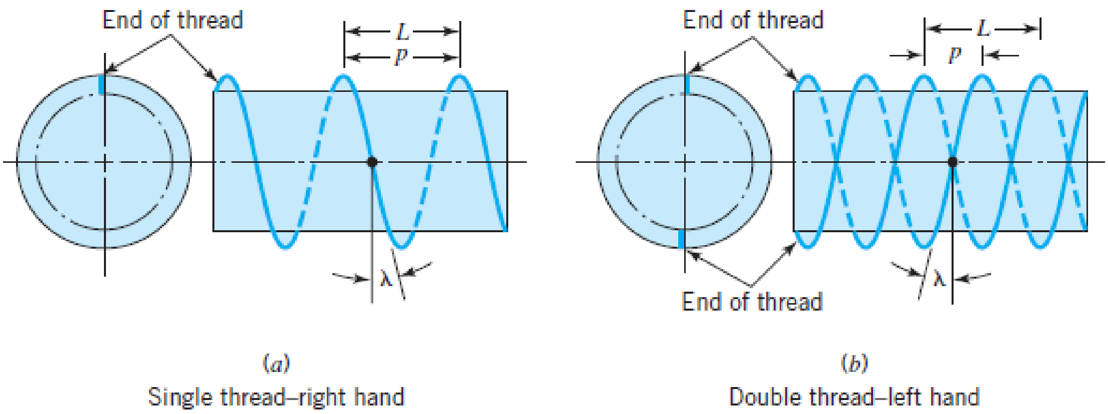


Fig.(1)

The main terminology of the power screw can be defined as follows

Pitch(p): It is the distance measured parallel to the axis of the screw from point on the thread to the corresponding point on the adjacent thread.

Lead(l): It is the distance measured parallel to the axis of the screw

It is the distance measured parallel to the axis of screw which the nut will advance in one revolution of the screw. For the single threaded screw, the lead is same as the pitch while for double threaded screw,the lead is twice of the pitch and so on.

Nominal diameter (d): It is the largest diameter of the screw. It is also called the major diameter.

Core diameter(dr): It is the smallest diameter of the screw thread. It is called the minor diameter.

Helix angle (λ): It is the angle made by the helix of the thread with a plane perpendicular to the axis of the screw. The helix angle is related to the lead and the mean diameter of the screw.

Figure (2) shows the standard geometry of screw threads used on fasteners. This is basically the same for both Unified (inch series) and ISO (International Standards Organization, metric) threads Standard sizes for the two systems are given in Tables (1) and (2). Table (1)shows both the fine thread (UNF, standing for Unified National Fine) and coarse thread (UNC, Unified National Coarse) series. The pitch diameter, dp, 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 .

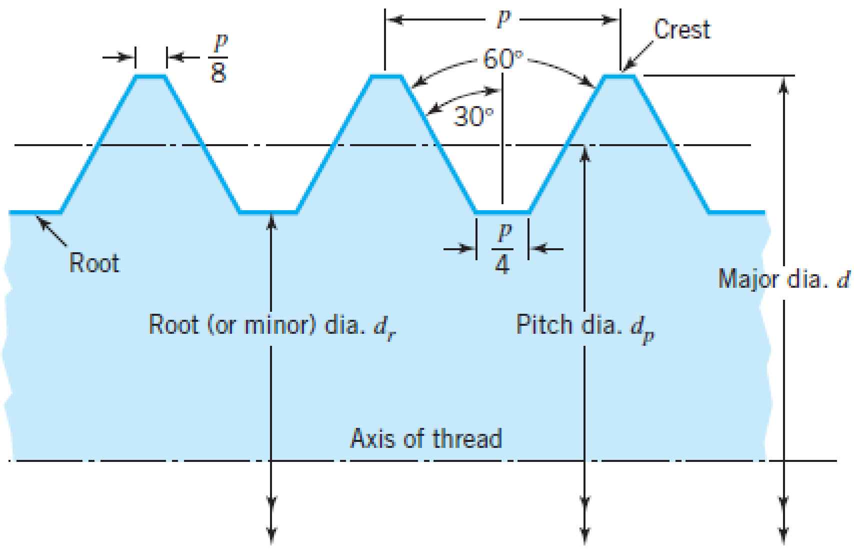


Fig.(2) Unified and ISO thread geometry. The basic profile of the external thread is shown.

Table (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."

Figure (3) illustrates the most of the standard threads used in power screws.

Table (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

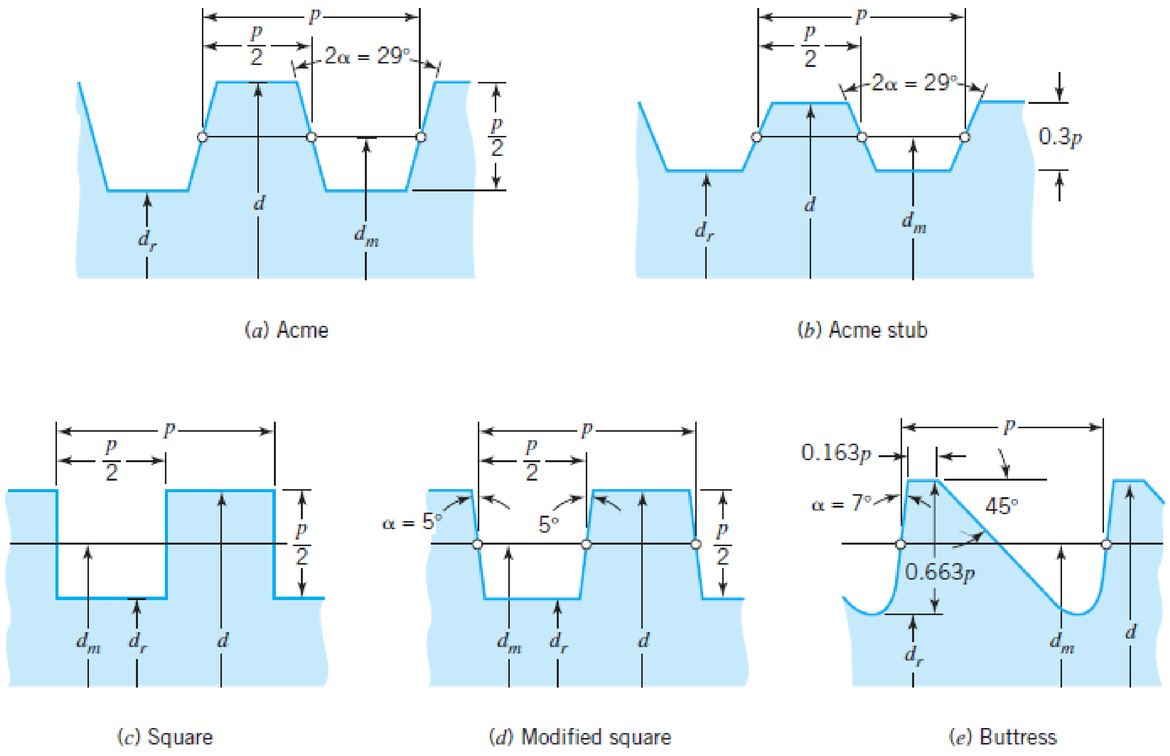


Fig.(3) Power screw thread forms

[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* + *dr*)/2.]

Acme threads are the oldest and are still in common use..

Acme stub is sometimes used because it is easier to heat-treat..

Square thread gives slightly greater efficiency but is seldom used because of difficulties in manufacturing the 0° thread angle **.**

Buttress thread is sometimes used for resisting large axial forces in one direction (the load is carried on the face with the 7° thread angle).

Standard sizes are given in Table (3)

Table (3) Standard Sizes of Power Screw Threads

Major Diameter *d* (in.)	Threads per Inch: Acme and Acme Stub[a]	Threads per Inch: Square and Modified Square	Threads per Inch: 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

Power Screw:

Power screws, sometimes called **linear actuators or translation screws**, are used to convert rotary motion of either the nut or the screw to relatively slow linear motion of the mating member along the screw axis.

The purpose of many power screws is to obtain a great mechanical advantage in lifting weights, as in screw-type jacks, or to exert large forces, as in presses and tensile testing machines, home garbage compactors, 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 (4) shows a simplified drawing of three different screw jacks supporting a weight. Note in each that only the 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 three jacks are basically the same. Figure (4-c) shows that the torque, **F×a**, must be applied to the nut in order to lift a given weight.

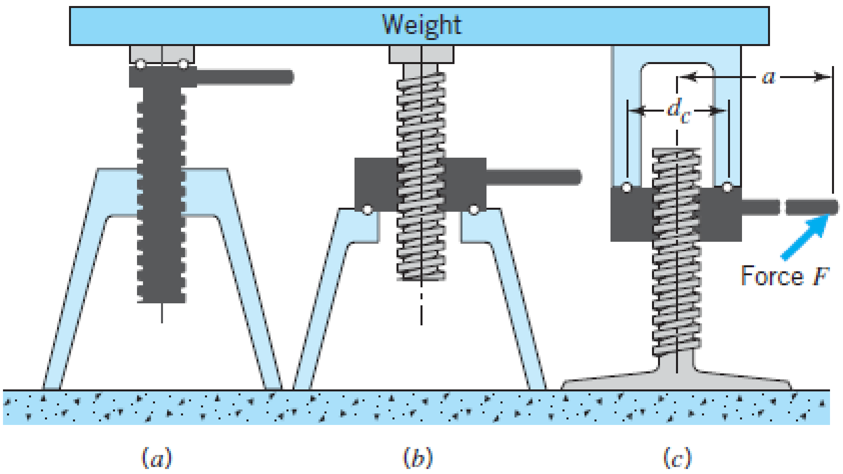


Fig.(4) Weight supported by three screw jacks. In each screw jack, only the shaded member rotates.

Turning the nut in Figure (4-c) forces each portion of the nut thread to climb an inclined plane. This plane can be represent by unwinding (or developing) a portion of one turn of the screw thread, as shown in the lower-left portion of Figure (5). If a full turn were developed, a triangle would be formed, illustrating the relationship

$$\tan \lambda = \frac{L}{\pi d_m} \qquad (1)$$

Where

λ= lead angle

L= lead

d_m= mean diameter of thread contact

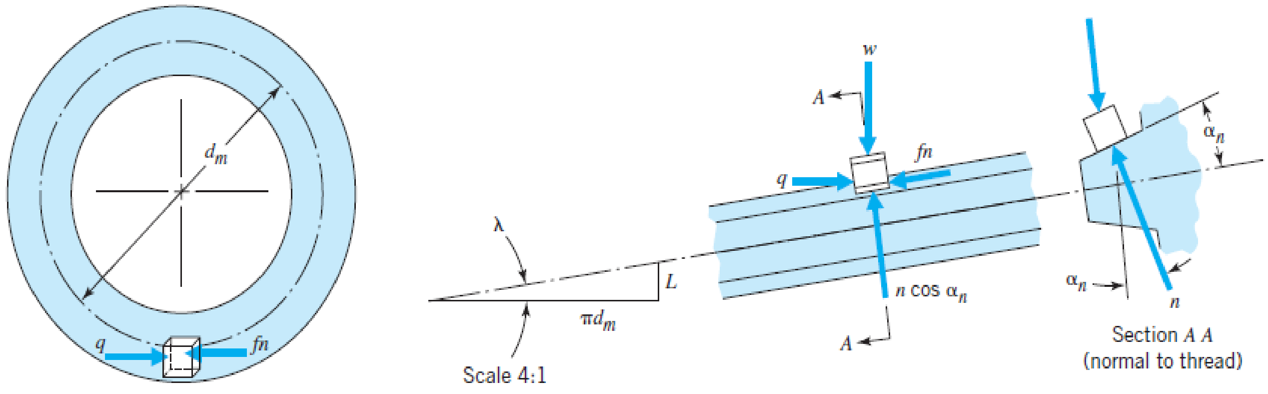


Fig.(5) Screw thread forces.

A segment of the nut is represented in Figure (5) by the small block acted upon by load *w* (a portion of the total axial load W), normal force *n* (shown in true view at the lower right), friction force (f×n), and tangential force q. Note that force q times $d_m/2$ represents the torque applied to the nut segment. Summing tangential forces acting on the block (i.e., horizontal forces in the lower left view) gives

$$\sum F_t = 0: \quad q - n(f \cos \lambda + \cos \alpha_n \sin \lambda) = 0 \qquad (2)$$

Summing axial forces (vertical forces in the lower left view) gives

$$\sum F_a = 0: \quad w + n(f \sin \lambda - \cos \alpha_n \cos \lambda) = 0 \qquad (3)$$

From equation (3) the normal force can be evaluated as:

$$n = \frac{w}{\cos \alpha_n \cos \lambda - f \sin \lambda} \tag{4}$$

Combining Eqs. 2,3 and 4, we have

$$q = w\frac{f \cos \lambda + \cos \alpha_n \sin \lambda}{\cos \alpha_n \cos \lambda - f \sin \lambda} \tag{5}$$

The torque corresponding to force q is q(d_m/2). Since the small block represents a typical segment of nut thread, integration over the entire thread surface in contact results in the same equations except that q, w, and n are replaced by Q, W, and *N*, where the latter represent the *total* tangential, vertical and normal loads, respectively, acting on the thread. **Thus the equation for torque required to lift load W is**

$$T = Q\frac{d_m}{2} = \frac{Wd_m}{2}\frac{f \cos \lambda + \cos \alpha_n \sin \lambda}{\cos \alpha_n \cos \lambda - f \sin \lambda} \tag{6}$$

Note that torque *T* is also equal to (F×a) in Figure (4).

Most applications of power screws require a bearing surface or **thrust collar** between stationary and rotating members. In Figure (4) this function is served by the ball thrust bearing of diameter d_c. In many cases, a simple thrust washer is used.

If the coefficient of friction of the collar washer or bearing is f_c , then the added torque required to overcome collar friction is $Wf_cd_c/2$, and the total **torque required to lift the load W for general case is**

$$T = \frac{Wd_m}{2}\frac{f\pi d_m + L \cos \alpha_n}{\pi d_m \cos \alpha_n - fL} + \frac{Wf_cd_c}{2} \tag{7}$$

and for the special case of the square thread the lifting torque can be calculated as:

$$T = \frac{Wd_m}{2}\frac{f\pi d_m + L}{\pi d_m - fL} + \frac{Wf_c d_c}{2} \tag{8}$$

The preceding analysis pertained to *raising* a load or to turning the rotating member "against the load." The analysis for *lowering* a load (or turning a rotating member "with the load") is exactly the same except that the directions of *q* and (f×n) (Fig.5) are reversed. In general the total **torque required to lower the load *W* is**

$$T = \frac{Wd_m}{2}\frac{f\pi d_m - L\cos\alpha_n}{\pi d_m \cos\alpha_n + fL} + \frac{Wf_c d_c}{2} \tag{9}$$

For power screw with square thread the lowering torque can be evaluated as:

$$T = \frac{Wd_m}{2}\frac{f\pi d_m - L}{\pi d_m + fL} + \frac{Wf_c d_c}{2} \tag{10}$$

Values of Friction Coefficients

When a ball or roller thrust bearing is used, f_c is usually low enough that collar friction can be neglected, thus eliminating the second term from the preceding equations. When a plain thrust collar is used, values of f and f_c vary generally between about 0.08 and 0.20 under conditions of ordinary service and lubrication and for the common materials of steel against cast iron or bronze. This range includes both starting and running friction, with starting friction being about one-third higher than running friction.

Values of Thread Angle in the Normal Plane

Figure (6) shows the thread angle measured in the normal plane (α_n, as used in the preceding equations) and in the axial plane (α, as usually specified, and as shown in Figure 3). It follows from Figure (6) that

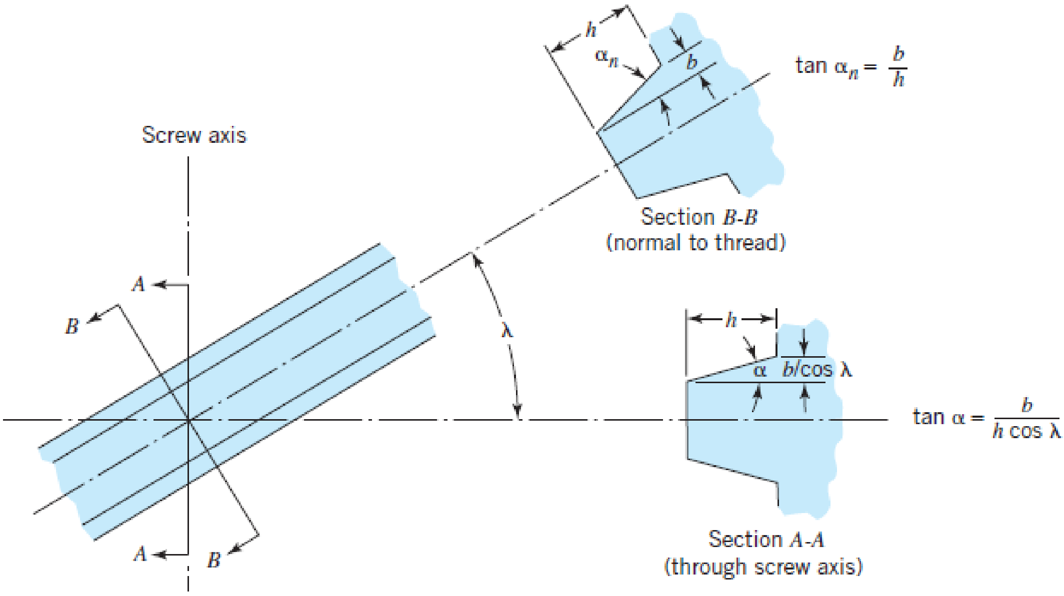


Fig.(6) Comparison of thread angles measured in axial and normal planes (α and α_n).

$$\tan \alpha_n = \tan \alpha \cos \lambda \qquad \textbf{(11)}$$

For small helix angles, cos λ is often taken as unity.

Overhauling and Self-Locking

A **self-locking** screw is one that requires a positive torque to lower the load, and an **overhauling** screw is one that has low enough friction to enable the load to lower itself; that is, a negative external lowering torque must be maintained to keep the load from lowering**.** If collar friction can be neglected, Eq. (9) for T≥0 shows that a screw is self-locking if

$$f \geqq \frac{L \cos \alpha_n}{\pi d_m} \qquad \textbf{(12)}$$

For a square thread, this simplifies to

$$f \geqq \frac{L}{\pi d_m}, \qquad \text{or} \qquad f \geqq \tan \lambda \qquad \textbf{(13)}$$

Efficiency

The work output of a power screw for one revolution of the rotating member is the product of force times distance, or (W×L). Corresponding work input is (2πT). The ratio (WL/2πT) is equal to efficiency. Substituting the expression for *T* in Eq. (7), with collar friction neglected, gives

$$\text{Efficiency, } e = \frac{L}{\pi d_m} \frac{\pi d_m \cos \alpha_n - fL}{\pi f d_m + L \cos \alpha_n} \quad \textbf{(14)}$$

or, for the case of the square thread,

$$e = \frac{L}{\pi d_m} \frac{\pi d_m - fL}{\pi f d_m + L} \quad \textbf{(15)}$$

1. The higher the coefficient of friction, the lower the efficiency.

2. Efficiency approaches zero as lead angle approaches zero because this represents a condition corresponding to the case where a large amount of friction work is required when moving the “block”

3. Efficiency again approaches zero as lead angle λ approaches 90°, and efficiency also decreases slightly as the thread angle α_n is increased from zero (square thread) to (Acme thread). Furthermore, efficiency would also approach zero if the thread angle were to approach 90°.

Rolling Contact

Figure(7) shows **a ball-bearing screw**; the sliding friction between screw and nut threads has been replaced with approximate rolling contact between the balls and the grooves in the screw and nut. This decreases friction drastically, **with efficiencies commonly 90 percent or higher**. Because of the low friction, ball-bearing screws are usually overhauling. This means that **a brake must be used** to hold a load in place. On the other hand, it also means that the screw is reversible in that linear motion can be converted to relatively rapid rotary motion in applications for which this is desirable. Operation is smooth, without the “slip-stick” action commonly observed in regular power screws (because of differences between static and sliding friction).

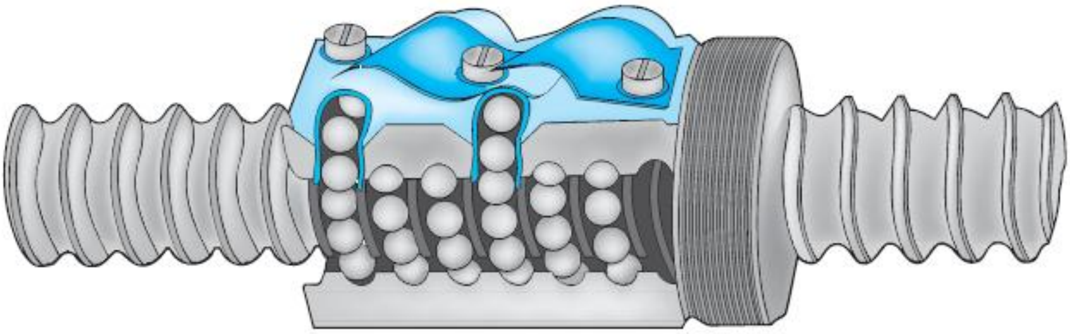


Fig.(7) Ball-bearing screw assembly

Ball-bearing screws are **commonly used** in aircraft landing gear retractors, large jet aircraft engine thrust reverser actuators, automatic door closers, antenna drives, hospital bed adjustors, machine tool controls, and numerous other applications

<u>Ex.1:</u>

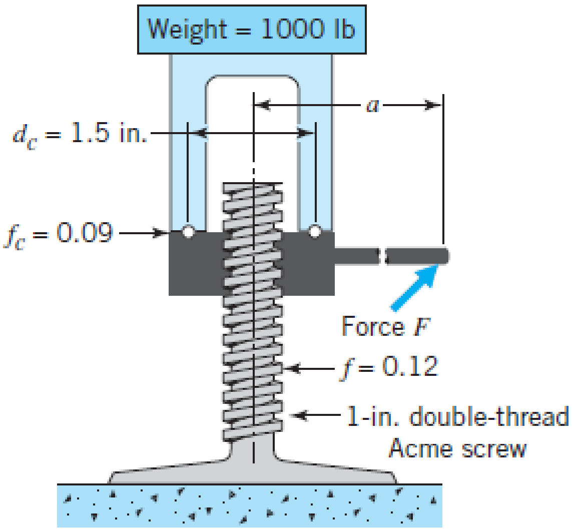
A screw jack shown in figure (8) with a 1-in., double-thread Acme screw is used to raise a load of 1000 lb. A plain thrust collar of mean 1.5 in diameter is used. Coefficients of running friction are estimated as 0.12 and 0.09 for f and f_c respectively.

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.

Assume that the starting friction is about one-third higher than running friction.



Fig,(8) Screw jack lifting a nonrotating load.

Solution:

a. From table (3) there are five threads per inch , hence

$$p = \frac{1}{5} = 0.2 \text{ in}$$

Because of the double thread ,

$$L = 2 \times p = 2 \times 0.2 = 0.4\ in$$

From figure (3-a) thread depth = $\frac{p}{2} = \frac{0.2}{2} = 0.1 in$

From figure (3-a) , $d_m = d - \frac{p}{2} = 1 - \frac{0.2}{2} = 0.9\ in$

$$\lambda = tan^{-1} = \frac{L}{\pi \times d_m} = tan^{-1} \frac{0.4}{\pi \times 0.9} = 8.05^o$$

b. For starting, increase the given coefficients of friction by about one-third, giving $f = 0.16$ and $f_c = 0.12$.

Use equation (7) to evaluate the required torque

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

α_n can be calculated from equation (11)

$$\alpha_n = tan^{-1}(tan\alpha\ cos\lambda)$$

$$\alpha_n = tan^{-1}(tan14.5\ cos8.05) = 14.36$$

$$T = \frac{1000(0.9)}{2}\frac{0.16\pi(0.9) + 0.4cos14.36}{\pi(0.9)cos14,36 - 0.16(0.4)} + \frac{1000(0.12)(1.5)}{2}$$

=141.3+90=231.3 lb.in

If a handle of 12in was used the required force is

$F = \frac{T}{12} = \frac{231.3}{12} = 19.3lb$

The torque required to lowering the load can be calculated from equation (9) as:

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

$$T = \frac{1000(0.9)}{2}\frac{0.16\pi(0.9) - 0.4cos14.36}{\pi(0.9)cos14,36 - 0.16(0.4)} + \frac{1000(0.12)(1.5)}{2}$$

=10.4+90;   T=100.4 lb.in

Static Screw Stresses:

1- Torsion

Power screws in operation are subjected to torsional stresses

$$\boldsymbol{\tau = \frac{Tc}{J} = \frac{16T}{\pi {d_r}^3}} \qquad \textbf{(16)}$$

where *d* is the thread root diameter, d_r , obtained from Figure (3).

2- Normal compressive stress

The body of screw is subjected to an axial force W and torsional moment (T). The direct compressive stress σ_c is given by:

$$\boldsymbol{\sigma_c = \frac{W}{\frac{\pi}{4}{d_r}^2}} \qquad (17)$$

The maximum shear stress due to combined loading can be evaluated as:

$$\boldsymbol{\tau_{max} = \sqrt{(\frac{\sigma_c}{2})^2 + (\tau)^2}} \qquad \textbf{(18)}$$

The threads of the screw, which are engaged with the nut, are subjected to transverse shear stress. The screw will tend to shear off the threads at the root diameter under the action of the load W. the shear area of one thread is $(\pi d_r t)$. The transverse shear stress in the screw is given by**:**

$$\boldsymbol{\tau_s = \frac{W}{\pi d_r t z}} \qquad \textbf{(19)}$$

t= thread thickness at the root diameter.

z= number of threads engaged with the nut.

3- Bearing stress

The bearing stress between the contacting surfaces of the screw and the nut is an important consideration in design. The bearing area between the screw and the nut for one thread is

$[\frac{\pi}{4}(d^2 - {d_r}^2]$. Therefore

$$\boldsymbol{S_b = \frac{W}{[\frac{\pi}{4}(d^2 - {d_r}^2)z]}} \qquad \textbf{(20)}$$

Ex2:

The nominal diameter of a triple threaded square screw is 50mm, while the pitch is 8mm. It is used with a collar having an outer diameter of 100mm and inner diameter as 65mm. The coefficient of friction at the thread surface as well as at the collar surface can be taken as 0.15. The screw is used to raise a load of 15kN, calculate

1- Torque required raising the load.
2- Torque required lowering the load.
3- The force required to raise the load, if applied at a radius of 500mm.

Solution:

1- The raising torque f or the power screw with square thread can be calculated by using equation(8) which can be re written as

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

d_m for square thread can be evaluated as:

d_m=d-0.5p= 50-0.5×8=46mm

L=3×p=3×8=24mm

$$d_c = \frac{D_o + D_i}{2} = \frac{100+65}{2} = 82.5mm$$

$$T = \frac{15000 \times 0.046}{2}\frac{0.15 \times \pi \times 0.046 + 0.024}{\pi \times 0.046 - 0.15 \times 0.024} + \frac{15000 \times 0.15 \times 0.082}{2} = 204.6425N.m$$

2- The torque required to lowering the load for a power screw with square threads can be used from equation (10) which can be rewritten as

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

$$T = \frac{15000 \times 0.046}{2} \frac{0.15 \times \pi \times 0.046 - 0.024}{\pi \times 0.046 + 0.15 \times 0.024} + \frac{15000 \times 0.15 \times 0.082}{2} = 86.58 N.m$$

3- The force required to raise the load can be calculated as

$T = F \times a$

204.64=F×0.5

F=409.285N

Ex 3:

A double threaded power screw with ACME threads is used to raise a load of 300kN. The nominal diameter is 100mm and the pitch is 12mm. The coefficient of friction at the screw threads is 0.15. Neglecting collar friction, calculate

1- Torque required to raise the load.
2- Torque required to lower the load.
3- Efficiency of the screw.

Ex4: A machine shown in figure(9) has a single start, square threads with 22mm nominal diameter and 5mm pitch. The outer and the inner diameters of the collar are 55mm and 45mm respectively. The coefficient of friction of the thread and the collar are 0.15 and 0.17 respectively. The machinist can comfortably exert a load of 125N on the handle at a mean radius of 150mm. Calculate

1- The clamping force developed between the jaws .
2- The overall efficiency of the clamp.

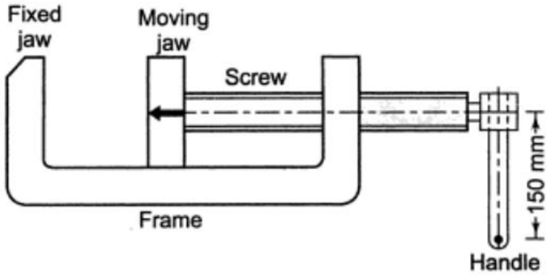


Fig.(9)

Solution:

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

d_m= d-0.5p= 22-0.5×5=19.5mm

L= p=5mm

$d_c = \frac{D_o + D_i}{2} = \frac{55+45}{2} = 50mm$

$$T = F \times a = 125 \times 0.15 = 18.75N.m$$

$$18.75 = \frac{W \times 0.019}{2}\,\frac{0.15 \times \pi \times 0.019 + .005}{\pi \times 0.019 - 0.15 \times 0.005} + \frac{W \times 0.17 \times 0.05}{2}$$

$$18.75 = 2.24 \times 10^{-3}W + 0.00425 = 6.49 \times 10^{-3}W$$

$$W = \frac{18.75}{6.49 \times 10^{-3}} = 2885N$$

The overall efficiency of the power screw can be evaluated as:

$$\eta = \frac{output\ work}{input\ work} = \frac{W \times L}{2\pi \times T} = \frac{2885 \times 0.005}{2\pi \times 18.75} = 0.12\ or\ 12\%$$

Ex5:

The construction of a gate valve used in high pressure pipeline is shown in figure(10). The screw is rotated in its place by means of the handle. The nut is fixed to the gate. When the screw rotates, the nut along with the gate moves downward or upward depending upon the direction of rotation of the screw. The screw has single start square thread of 40mm outer diameter and 7mm pitch. The weight of the gate is 5kN. The water pressure in the pipeline induces frictional resistance between the gate and its seat. The resultant frictional resistance in the axial direction is 2kN. The inner and outer diameters of thrust washer are 40 and 80mm respectively. The values of coefficient of friction of the screw and the washer are 0.15 and 0.12 respectively. The handle is rotated by two arms each exerting equal force at a radius of 500mm from the axis of screw. Calculate

1- The maximum force exerted by each arm when the gate is being raised
2- The maximum force exerted by each arm when the gate is being lowered.
3- The efficiency of the gate mechanism.
4- The length of the nut if the permissible bearing pressure is $5 N/mm^2$

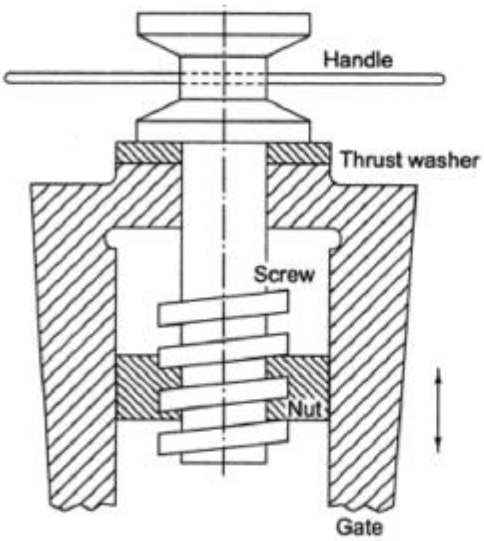


Fig.(10) Gate valve

Solution:

1- The torque required to raise the gate can be evaluated as:

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

$$\boldsymbol{d_m = d - 0.5p = 40 - 0.5 \times 7 = 36.5mm}$$

L=p (since the screw is single start square thread

L=7mm

$$d_c = \frac{D_o + D_i}{2} = \frac{80 + 40}{2} = 60mm$$

Frictional resistance acts opposite to the motion. When the gate is being raised the frictional force acts in downward direction.

Therefore the axial force on the screw consists of addition of the weight of the gate plus the frictional resistance

$$W = 5000 + 2000 = 7000N$$

$$T = \frac{7000 \times 0.036}{2} \frac{0.15 \times \pi \times 0.036 + .007}{\pi \times 0.036 - 0.15 \times 0.007} + \frac{7000 \times 0.12 \times 0.06}{2} = 52.1N.m$$

$$P = \frac{T}{2a} = \frac{52.1}{2 \times 0.5} = 52.1N$$

2- When the gate being lowered the frictional resistance acts in upward direction. Therefore

$$W = 5000 - 2000 = 3000\ N$$

The torque required to lower the gate can be calculated as:

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

$$T = \frac{3000 \times 0.036}{2} \frac{0.15 \times \pi \times 0.036 - .007}{\pi \times 0.036 + 0.15 \times 0.007} + \frac{3000 \times 0.12 \times 0.06}{2} = 15.513N.m$$

The force required can be calculated as

$$P = \frac{T}{2a} = \frac{15.513}{2 \times .5} = 15.531N$$

3- The efficiency of the power screw can be evaluated as

$$\eta = \frac{WL}{2\pi T} = \frac{7000 \times 0.007}{2\pi \times 52.1} = 0.14\ or\ 14\%$$

4- To calculate the length of the nut

$$d_r = d - p = 40 - 7 = 33mm$$

The bearing stress can be expressed as

$$S_b = \frac{W}{\pi \frac{(d^2 - {d_r}^2)}{4} z}$$

z= number of threads

$$\therefore z = \frac{W}{\pi \frac{(d^2 - {d_r}^2)}{4} S_b} = \frac{7000}{\pi \frac{(0.04^2 - 0.033^2)}{4} \times 5 \times 10^6} = 3.4$$

So that four teeth must be used or z=4

The length of the nut=z×L= 4×7=28mm

<u>Ex6:</u> A screw clamp used on the shop floor is shown in in figure (11). The screw has single start square threads of 22mm nominal diameter and 5mm pitch. The coefficient of friction and the collar is 0.15.The mean radius of the friction collar is 15mm. The capacity of the clamp is 750N. The handle is made of steel with S_{yt}=400N/mm^2. It can be assumed that the operator exerts a force of 20N on the handle.

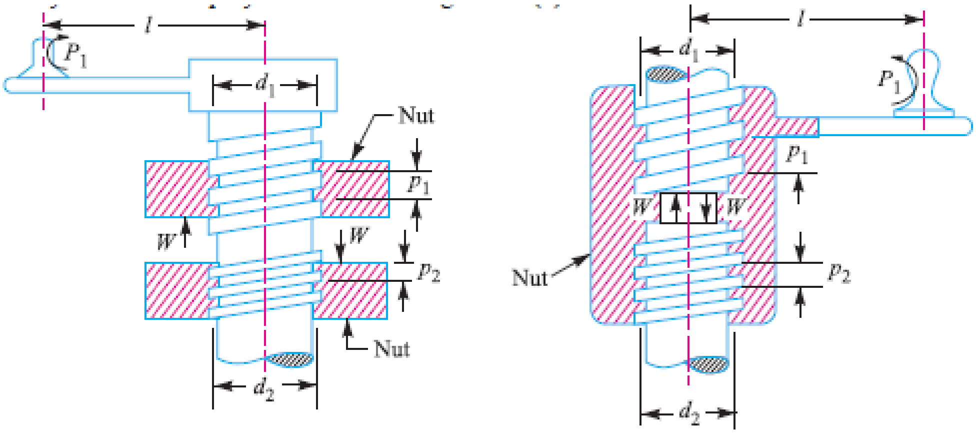
(i) What torque is required to tighten the clamp to full capacity?

(ii) Determine the length and the diameter of the handle such that it will bend with permanent set when the rated capacity of the clamp is exceeded.

Differential and Compound Screws

There are certain cases in which a very slow movement of the screw is required whereas in other cases, a very fast movement of the screw is needed. The slow movement of the screw may be obtained by using a small pitch of the threads, but it results in weak threads. The fast movement of the screw may be obtained by using multiple-start threads, but this method requires expensive machining and the loss of self-locking property. In order to overcome these difficulties, differential or compound screws, as discussed below, are used.

1. Differential screw. When a slow movement or fine adjustment is desired in precision equipment, then a differential screw is used. It consists of two threads of the same hand (*i.e.* right handed or left handed) but of different pitches, wound on the same cylinder or different cylinders as shown in Fig. (11). It may be noted that when the threads are wound on the same cylinder, then two nuts are employed as shown in Fig. (11-*a*) and when the threads are wound on different cylinders, then only one nut is employed as shown in Fig. (11-b).



a: Thread wound on the same cylinder b: Thread wound on the different cylinder

Fig.(11)

Total torque required when there is no friction can be calculated as:

$$T_o = \frac{W}{2\pi}(P_1 - P_2)$$

Where

T_o= output torque(N.m)

W= Applied load(N)

P_1= first pitch(m)

P_2= second pitch(m)

2. Compound screw. When a fast movement is desired, then a compound screw is employed. It consists of two threads of opposite hands (*i.e.* one right handed and the other left handed) wound on the same cylinder or different cylinders, as shown in Fig. (11 -*a*) and (*b*) respectively. In this case, each revolution of the screw causes the nuts to move towards one another equal to the sum of the pitches of the threads. Usually the pitch of both the threads are made equal.

$$T_o = \frac{W}{2\pi}(P_1 + P_2)$$

Design Project

A toggle jack as shown in Fig.(1) is to be designed for lifting a load of 4 kN. When the jack is in the top position, the distance between the center lines of nuts is 50 mm and in the bottom position this distance is 210 mm. The eight links of the jack are symmetrical and 110 mm long. The link pins in the base are set 30 mm apart. The links, screw and pins are made from mild steel for which the permissible stresses are 100 MPa in tension and 50 MPa in shear. The bearing pressure on the pins is limited to 20 N/mm2.Assume the pitch of the square threads as 6 mm and the coefficient of friction between threads as 0.20.

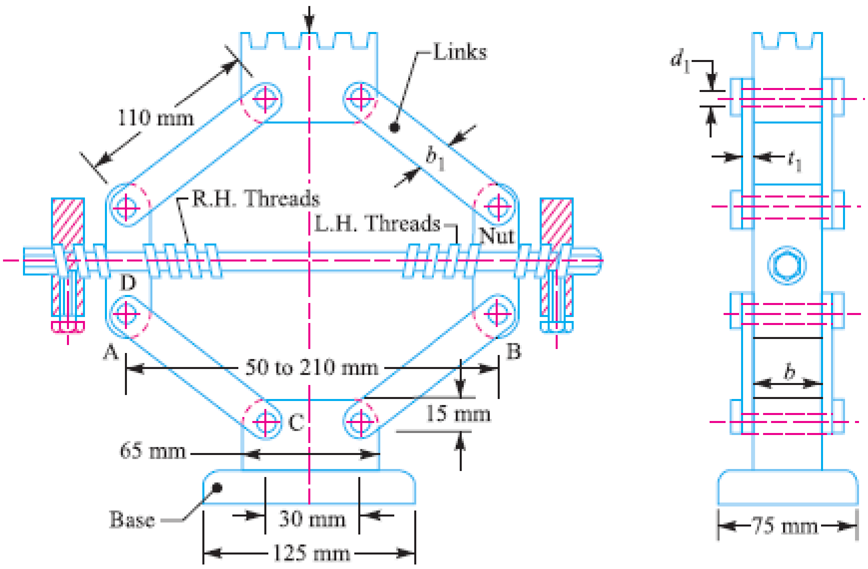


Fig.(1) Toggle Jack