Design of Power Screws

Thread Standards and Definitions

Pitch **(ρ):** distance between adjacent threads measured parallel to thread axis. Reciprocal of threads per inch

Major diameter **(D):** largest diameter of screw thread

Minor diameter **(D_1):** also called "*root diameter*", is the smallest diameter of the screw thread.

Pitch diameter **(D_2):** also called "*mean diameter*", the average diameter of the screw thread, or it is the theoretical diameter between major and minor diameters, where tooth and gap are same width

Lead **(l):** the distance a nut moves parallel to the screw axis when it rotates one full turn.

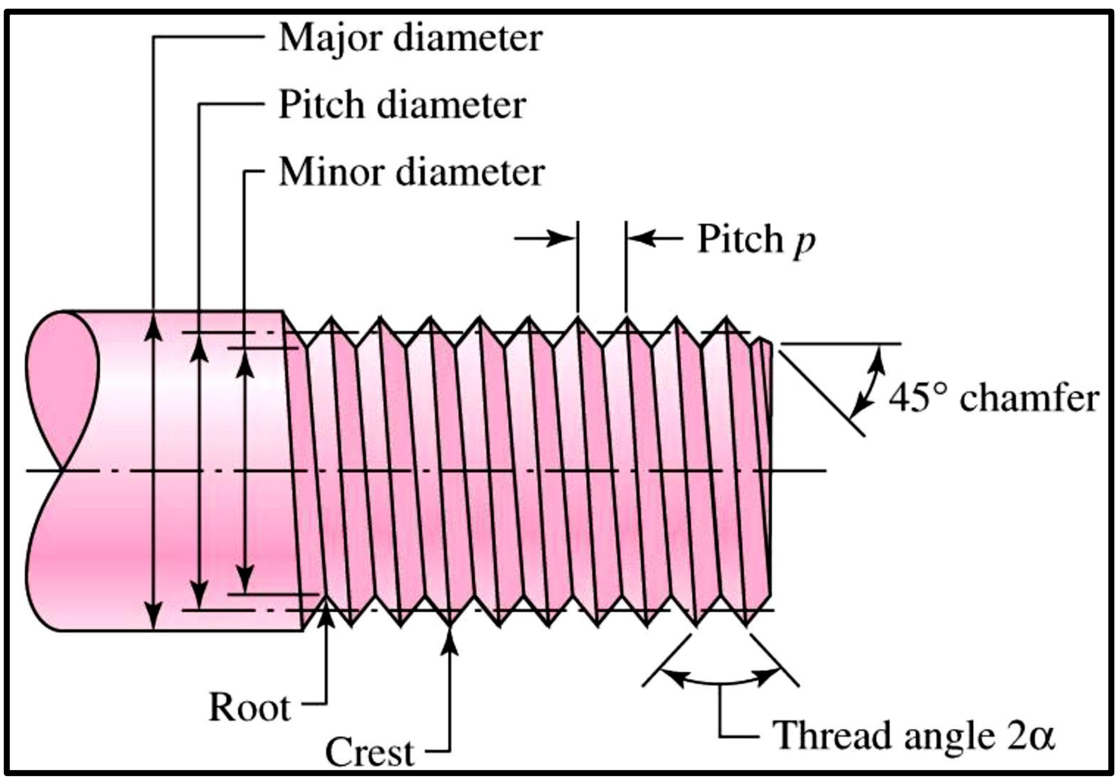


Fig. (8–1)

- For a single thread screw the lead is same as the pitch.
- For multiple thread screws (*two or more threads run beside each other*) the lead equals the pitch multiplied by the number of threads.

- All threads are usually right-handed unless otherwise is indicated.

Tensile Stress Area

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

<u>Standardization</u>

- **Bolts** are standardized and there are two standards: *Metric* (***ISO***) and *American* (***Unified***). In both standards the thread angle is 60°.

***Metric (ISO)*:**

- ❖ There are two standard profiles ***M*** (normal thread) and ***MJ*** (greater root radius)where both have a similar geometry but the ***MJ*** has a rounded fillet at the root and a larger minor diameter and therefore it has a better fatigue strength.
- Metric bolts are specified by the *major diameter* and the *pitch* (*both in mm*).

***Example*:** M10X1.5(*10 mm* major diameter and *1.5 mm* pitch).

❖ ***Table (8-1)*** gives the standard sizes of *Metric* bolts along with the effective tensile stress area and the root diameter area (*which is used when the bolt is subjected to shear loading*).

The *American National* (*Unified*): thread standard defines basic thread geometry for uniformity and interchangeability

- ❖ There are two standard profiles ***UN*** ***(***normal thread) and ***UNR*** ***(***greater root radius for fatigue applications), where the ***UNR*** has a filleted root and thus better fatigue strength.
- Unified threads are specified by the *major diameter* (*in inch*) and the *number of threads per inch* (N).

➢ ***Example***: $0.25 - 20\ UNC$

Diameter *(N)* *Profile* *Coarse* *or* ***F*** *(Fine)*

Table (8–1): **Diameters and Areas for Metric Threads**

Table 8–1

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

Nominal Major Diameter d mm	Coarse-Pitch Series: Pitch p mm	Coarse-Pitch Series: Tensile-Stress Area A_t mm²	Coarse-Pitch Series: Minor-Diameter Area A_r mm²	Fine-Pitch Series: Pitch p mm	Fine-Pitch Series: Tensile-Stress Area A_t mm²	Fine-Pitch Series: Minor-Diameter Area A_r mm²
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980

❖ ***Table (8 -2)*** gives the standard sizes along with the tensile stress areas and root diameter areas (*used for shear loading*) for *Unified* bolts (*Coarse and Fine series*).

- Note that for diameters smaller than *1/4 inch*, the size is designated by size numbers rather than diameter.

❖ Note that there is Coarse-pitch and Fine-pitch (*more threads*) where the fine-pitch has better tensile strength.

Table (8–2): **Diameters and Areas for Unified Screw Threads**

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

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

- High grade alloy
- Instrumentation
- Aircraft

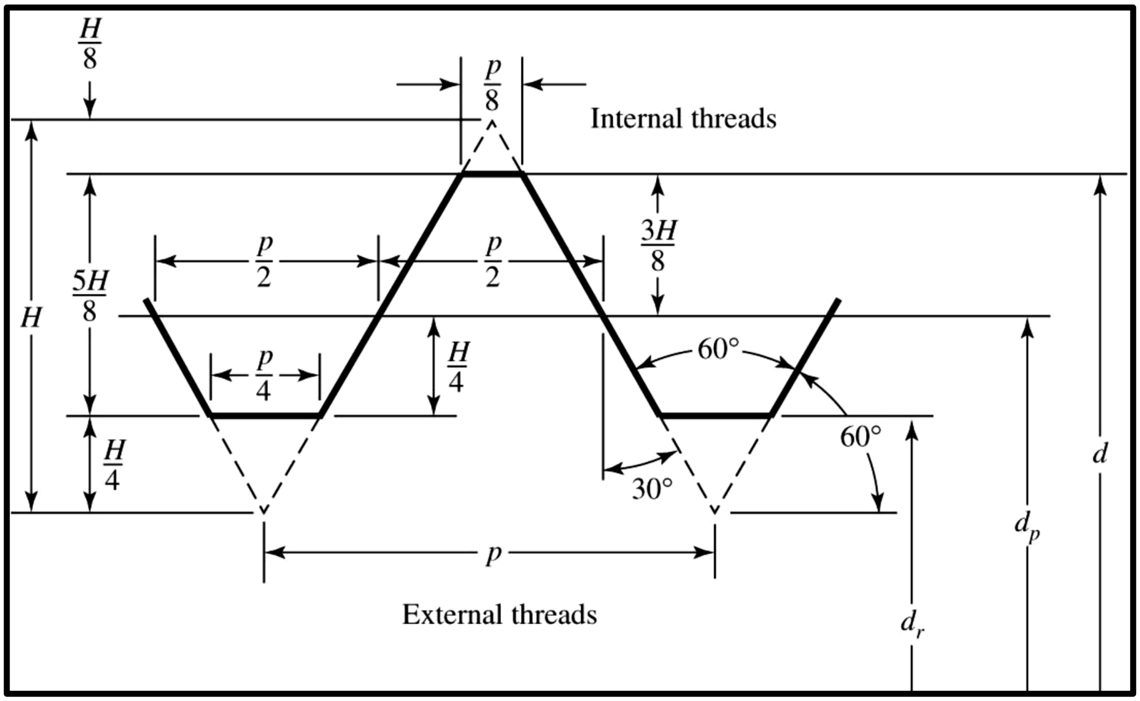


Fig. (8–2)

Square and Acme Threads

Square and Acme threads (Trapezoidal thread forms) are used when the threads are intended to transmit power

Table (8-3): the standard diameters and associated pitch for *Acme* thread power screws

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

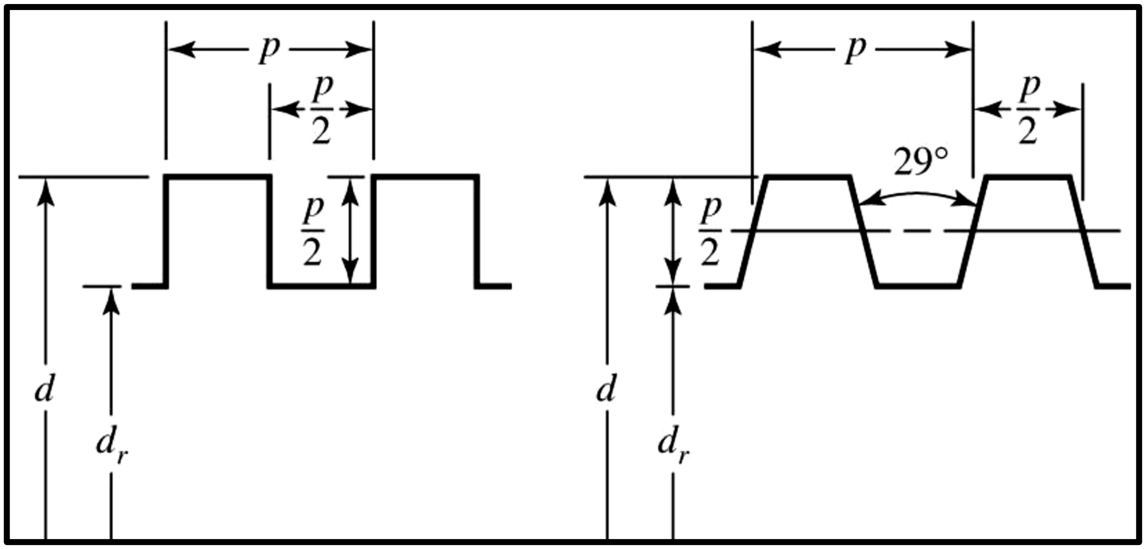


Fig. (8–3)

Mechanics of Power Screws

- ***Power screw***
  - ◦Used to change angular motion into linear motion, where it is similar in principle to screws and bolts.
  - ◦Usually transmits power
  - ◦Examples include vises, presses, jacks, lead screw on lathe

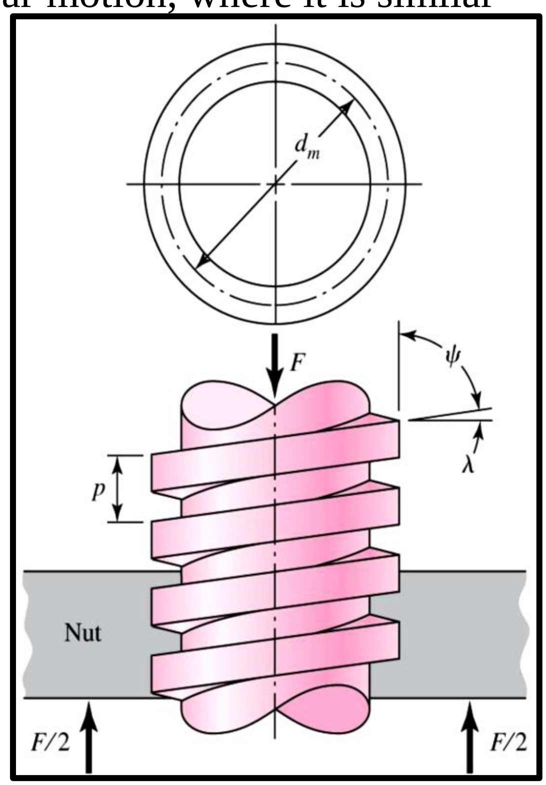


Fig. (8–5)

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

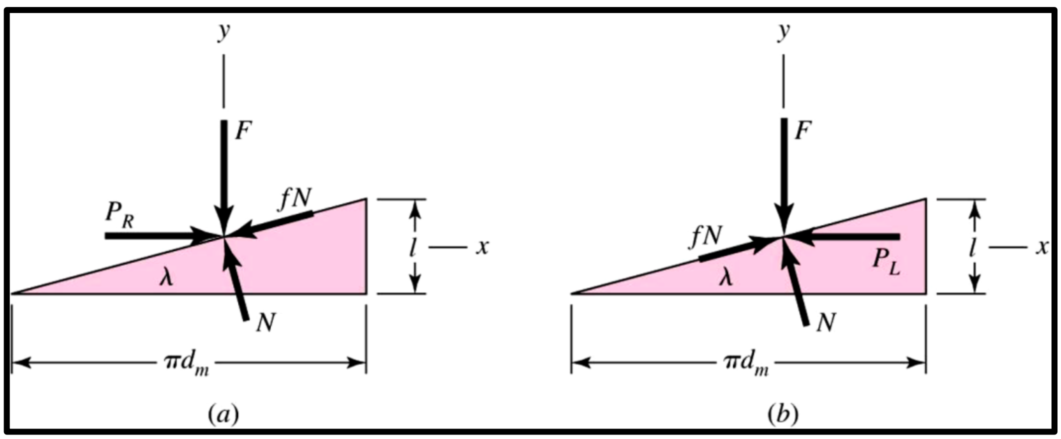
Raising and Lowering Load

p **(Pitch) =** l **(Lead: for single thread screws)**

λ **: Lead angle,** ψ **: Helix angle**

d_m **: Mean diameter**

- To find the torque needed to raise the load (P_R) or needed to lower the load (P_L), let one thread of the screw to be unrolled (*assuming square thread*).



- For raising the load and using static equilibrium equations

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

$$\sum F_y = -F - fN\sin\lambda + N\cos\lambda = 0 \qquad (a)$$

- For lowering the load

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

$$\sum F_y = -F + fN\sin\lambda + N\cos\lambda = 0 \qquad (b)$$

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

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

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

- Divide numerator and denominator by cosλ and use relation

  $\tan\lambda = l/\pi\, d_m$

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

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

Raising and Lowering Torque

- Noting that the torque is the product of the force and the mean radius. To find the torque (torsional moment) needed to raise the load (T_R) or needed to lower the load (T_L), let one thread of the screw to be unrolled (*assuming square thread*). Knowing that $T = P(d_m/2)$:

The torque needed to raise the load : *(The torque is used to raise the load and to overcome thread friction)*

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

The torque needed to lower the load: *(The torque is used to overcome a part of the friction)*

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

Self-locking Condition

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

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

Power Screw Efficiency

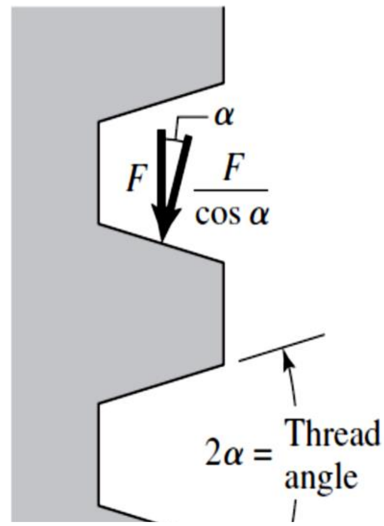
- The efficiency is important in evaluating power screws.
- If $\boldsymbol{f = 0}$ (*no friction*) then all the applied torque is transferred into force (*100% efficiency*) and the torque needed to rise the load (T_R) with no friction losses becomes from Eq. (8–1) with $f = 0$:

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

- The efficiency of the power screw is therefore

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

Power Screws with Acme Threads



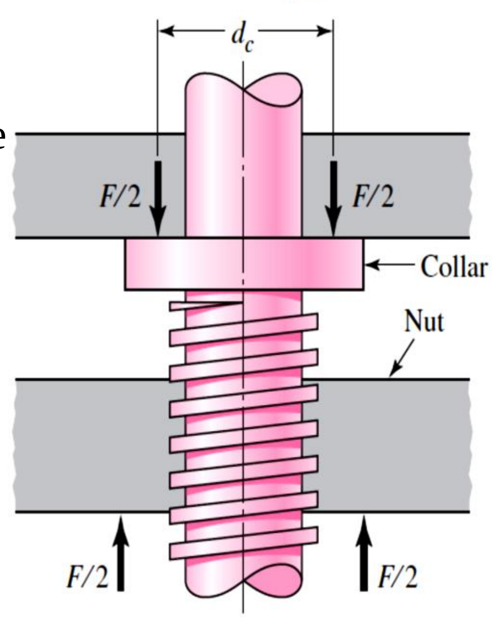
- If Acme threads are used instead of square threads, the thread angle (α) creates additional wedging force which increases the frictional forces (F becomes $F/\cos\alpha$)

- Thus all frictional terms in Eq. (8–1) are divided by ($\cos\alpha$) therefore the torque necessary to raise a load (or tighten a screw) (T_R) becomes:

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

- Due to the increased friction the efficiency of *Acme* thread is less than that of *Square* threads.
- However *Acme* threads are commonly used because they are easier to machine and split-nuts (*to compensate for wear*) can be used.

(a)



(b)

Collar Friction

In many cases a Collar (*sliding friction bearing*) is used to support the load (as seen in the figure), and thus additional component of torque (T_C) is needed to overcome the friction between the collar

and load plate.

The collar torque is found as:(Assuming the load is concentrated at the mean collar diameter *dc)*

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

Where, f_C : coefficient of friction for the collar
dc: collar mean diameter

- ❖ ***Table (8-5)*** gives the coefficients of sliding (*dynamic*) and starting (*static*) friction for some common metal pairs (*The best is for bronze on bronze, but since bronze have relatively low strength it is not commonly used for the screw*).
- ❖ ***Table (8-6)*** gives the coefficients of friction (*sliding and starting*) for thrust collars.

Table 8–5

Coefficients of Friction *f* for Threaded Pairs

Source: H. A. Rothbart, *Mechanical Design and Systems Handbook*, 2nd ed., McGraw-Hill, New York, 1985.

Screw Material	Nut Material: Steel	Bronze	Brass	Cast Iron
Steel, dry	0.15–0.25	0.15–0.23	0.15–0.19	0.15–0.25
Steel, machine oil	0.11–0.17	0.10–0.16	0.10–0.15	0.11–0.17
Bronze	0.08–0.12	0.04–0.06	—	0.06–0.09

Table 8–6

Thrust-Collar Friction Coefficients

Source: H. A. Rothbart, *Mechanical Design and Systems Handbook*, 2nd ed., McGraw-Hill, New York, 1985.

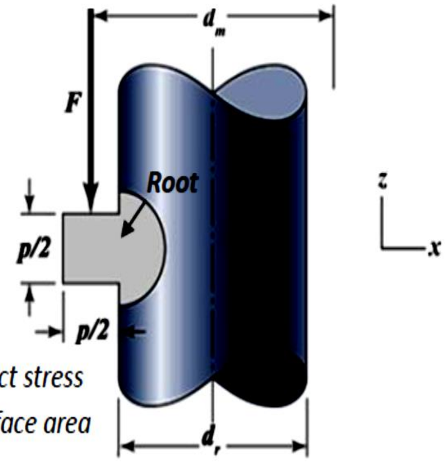
Combination	Running	Starting
Soft steel on cast iron	0.12	0.17
Hard steel on cast iron	0.09	0.15
Soft steel on bronze	0.08	0.10
Hard steel on bronze	0.06	0.08

Thread Deformation in Screw-Nut Combination

- It is necessary to find the stresses developed in the power screw while performing its function to ensure its safety.
- The stresses in the body of the power screw are found as:

Normal stress: $\sigma = \frac{F}{A} = \frac{4F}{\pi\ {d_r}^2}$ *Tension or Compression*

Shear due to the torque: $\tau = \frac{Tc}{J} = \frac{16T}{\pi\ {d_r}^3}$ *Maximum at the root*



- If the screw is loaded in compression, then buckling should be considered also.
  - ✓ *Johnson* or *Euler* formula can be used according to the slenderness ratio (*use the root diameter*).

- The threads are also subjected to stresses which are:

Bearing stress: $\sigma_B = \frac{-F}{A} = -\frac{2F}{\pi d_m n_t p}$ *Compressive contact stress over the entire surface area*

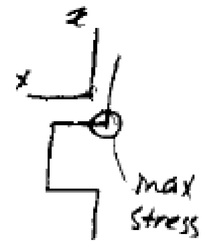
where n_t is the *number of engaged threads*

Bending stress: $\sigma_b = \frac{Mc}{I} = \frac{6F}{\pi d_r n_t p}$ *Max at the top surface of the root*

Transverse shear: $\tau = \frac{3V}{2A} = \frac{3F}{\pi d_r n_t p}$ *Max at the center of the root & zero at the top surface*

- Power screw thread is in compression, causing elastic shortening of screw thread pitch.
- Engaging nut is in tension, causing elastic lengthening of the nut thread pitch.
- Consequently, the engaged threads cannot share the load equally between the engaged threads.
- Experiments indicate the first thread carries 38% of the load, the second thread 25%, and the third thread 18%. The seventh thread is free of load (*assuming the number of engaged threads is six or more*).
- To find the largest stress at the root of the first thread of a screw-nut combination, and in the analysis we do not divide the load by the number of engaged threads (*nt*) but rather we do the analysis based on the use of $0.38F$ in place of F, and set $nt = 1$.

- The critical stress occurs at the top of the root and it's found according to *Von Misses* knowing that:



Bending $\longrightarrow$ $\sigma_x = \dfrac{6F}{\pi d_r n_t p}$, $\tau_{xy} = 0$

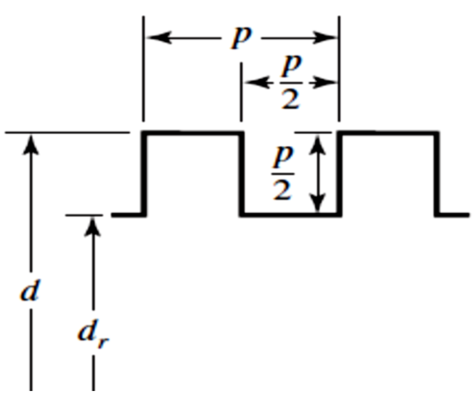
$\sigma_y = 0$, $\tau_{yz} = \dfrac{16T}{\pi\, d_r^{\,3}}$ $\longleftarrow$ *Torsion*

Axial $\longrightarrow$ $\sigma_z = -\dfrac{4F}{\pi\, d_r^{\,2}}$, $\tau_{zx} = 0$

Critical when this is compression and bending is (+)

Example 8-1

A square-thread power screw has a major diameter of 32 mm and a pitch of 4 mm with double threads, and it is to be used in an application similar to that in Fig. 8–4. The given data include $f = f_c = 0.08$, $d_c = 40$ mm, and $F = 6.4$ kN per screw.
(*a*) Find the thread depth, thread width, pitch diameter, minor diameter, and lead.
(*b*) Find the torque required to raise and lower the load.
(*c*) Find the efficiency during lifting the load.
(*d*) Find the body stresses, torsional and compressive.
(*e*) Find the bearing stress.
(*f*) Find the thread bending stress at the root of the thread.
(*g*) Determine the von Mises stress at the root of the thread.
(*h*) Determine the maximum shear stress at the root of the thread.



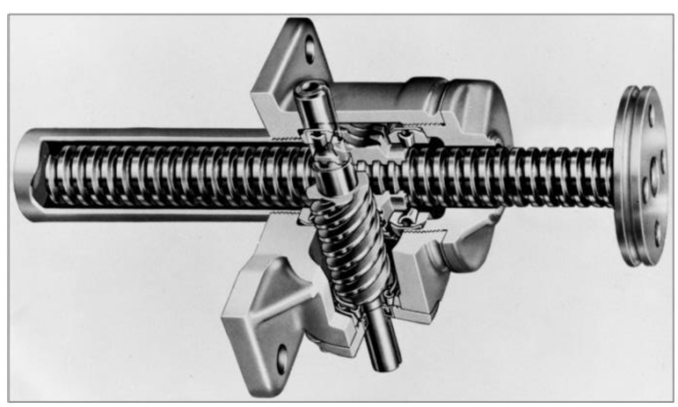


Fig. (8–4)

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

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

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

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

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

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

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

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

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

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

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

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

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

$$e = \frac{F l}{2\pi T_R} = \frac{6.4(8)}{2\pi(26.18)} = 0.311$$

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

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

The axial nominal normal stress σ is

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

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

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

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

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

(g) The transverse shear at the extreme of the root cross section due to bending is zero. However, there is a circumferential shear stress at the extreme of the root cross section of the thread as shown in part (d) of 6.07 MPa. The three-dimensional stresses, after Fig. 8–8, noting the y coordinate is into the page, are

$$\begin{aligned} \sigma_x &= 41.5 \text{ MPa} & \tau_{xy} &= 0 \\ \sigma_y &= -10.39 \text{ MPa} & \tau_{yz} &= 6.07 \text{ MPa} \\ \sigma_z &= 0 & \tau_{zx} &= 0 \end{aligned}$$

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

$$\begin{aligned} \sigma' &= \frac{1}{\sqrt{2}}\{(41.5-0)^2 + [0-(-10.39)]^2 + (-10.39-41.5)^2 + 6(6.07)^2\}^{1/2} \\ &= 48.7 \text{ MPa} \end{aligned}$$

Alternatively, you can determine the principal stresses and then use Eq. (5–12) to find the von Mises stress. This would prove helpful in evaluating $\tau_{\max}$ as well. The principal stresses can be found from Eq. (3–15); however, sketch the stress element and note that there are no shear stresses on the x face. This means that σ_x is a principal stress. The remaining stresses can be transformed by using the plane stress equation, Eq. (3–13). Thus, the remaining principal stresses are

$$\frac{-10.39}{2} \pm \sqrt{\left(\frac{-10.39}{2}\right)^2 + 6.07^2} = 2.79, -13.18 \text{ MPa}$$

Ordering the principal stresses gives $\sigma_1, \sigma_2, \sigma_3 = 41.5, 2.79, -13.18$ MPa. Substituting these into Eq. (5–12) yields

$$\sigma' = \left\{ \frac{[41.5 - 2.79]^2 + [2.79 - (-13.18)]^2 + [-13.18 - 41.5]^2}{2} \right\}^{1/2}$$

$$= 48.7 \text{ MPa}$$

(*h*) The maximum shear stress is given by Eq. (3–16), where $\tau_{max} = \tau_{1/3}$, giving

$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} = \frac{41.5 - (-13.18)}{2} = 27.3 \text{ MPa}$$

<u>Power Screw Safe Bearing Pressure</u>

Table 8–4

Screw Bearing Pressure p_b

Source: H. A. Rothbart and T. H. Brown, Jr., *Mechanical Design Handbook,* 2nd ed., McGraw-Hill, New York, 2006.

Screw Material	Nut Material	Safe p_b, psi	Notes
Steel	Bronze	2500–3500	Low speed
Steel	Bronze	1600–2500	≤10 fpm
	Cast iron	1800–2500	≤8 fpm
Steel	Bronze	800–1400	20–40 fpm
	Cast iron	600–1000	20–40 fpm
Steel	Bronze	150–240	≥50 fpm

Table 8–5

Coefficients of Friction f for Threaded Pairs

Source: H. A. Rothbart and T. H. Brown, Jr., *Mechanical Design Handbook,* 2nd ed., McGraw-Hill, New York, 2006.

Screw Material	Nut Material			
	Steel	Bronze	Brass	Cast Iron
Steel, dry	0.15–0.25	0.15–0.23	0.15–0.19	0.15–0.25
Steel, machine oil	0.11–0.17	0.10–0.16	0.10–0.15	0.11–0.17
Bronze	0.08–0.12	0.04–0.06	—	0.06–0.09

Table 8–6

Thrust-Collar Friction Coefficients

Source: H. A. Rothbart and T. H. Brown, Jr., *Mechanical Design Handbook,* 2nd ed., McGraw-Hill, New York, 2006.

Combination	Running	Starting
Soft steel on cast iron	0.12	0.17
Hard steel on cast iron	0.09	0.15
Soft steel on bronze	0.08	0.10
Hard steel on bronze	0.06	0.08

Table A-2

Conversion Factors A to Convert Input X to Output Y Using the Formula $Y = AX$*

Multiply Input X	By Factor A	To Get Output Y	Multiply Input X	By Factor A	To Get Output Y
British thermal unit, Btu	1055	joule, J	mile/hour, mi/h	1.61	kilometer/hour, km/h
Btu/second, Btu/s	1.05	kilowatt, kW	mile/hour, mi/h	0.447	meter/second, m/s
calorie	4.19	joule, J	moment of inertia, lbm $\cdot$ ft^2	0.0421	kilogram-meter^2, kg $\cdot$ m^2
centimeter of mercury (0°C)	1.333	kilopascal, kPa	moment of inertia, lbm $\cdot$ in^2	293	kilogram-millimeter^2, kg $\cdot$ mm^2
centipoise, cP	0.001	pascal-second, Pa $\cdot$ s	moment of section (second moment of area), in^4	41.6	centimeter^4, cm^4
degree (angle)	0.0174	radian, rad	ounce-force, oz	0.278	newton, N
foot, ft	0.305	meter, m	ounce-mass	0.0311	kilogram, kg
foot^2, ft^2	0.0929	meter^2, m^2	pound, lbf†	4.45	newton, N
foot/minute, ft/min	0.0051	meter/second, m/s	pound-foot, lbf $\cdot$ ft	1.36	newton-meter, N $\cdot$ m
foot-pound, ft $\cdot$ lbf	1.35	joule, J	pound/foot^2, lbf/ft^2	47.9	pascal, Pa
foot-pound/second, ft $\cdot$ lbf/s	1.35	watt, W	pound-inch, lbf $\cdot$ in	0.113	joule, J
foot/second, ft/s	0.305	meter/second, m/s	pound-inch, lbf $\cdot$ in	0.113	newton-meter, N $\cdot$ m
gallon (U.S.), gal	3.785	liter, L	pound/inch, lbf/in	175	newton/meter, N/m
horsepower, hp	0.746	kilowatt, kW	pound/inch^2, psi (lbf/in^2)	6.89	kilopascal, kPa
inch, in	0.0254	meter, m	pound-mass, lbm	0.454	kilogram, kg
inch, in	25.4	millimeter, mm	pound-mass/second, lbm/s	0.454	kilogram/second, kg/s
inch^2, in^2	645	millimeter^2, mm^2	quart (U.S. liquid), qt	946	milliliter, mL
inch of mercury (32°F)	3.386	kilopascal, kPa	section modulus, in^3	16.4	centimeter^3, cm^3
kilopound, kip	4.45	kilonewton, kN	slug	14.6	kilogram, kg
kilopound/inch^2, kpsi (ksi)	6.89	megapascal, MPa (N/mm^2)	ton (short 2000 lbm)	907	kilogram, kg
mass, lbf $\cdot$ s^2/in	175	kilogram, kg	yard, yd	0.914	meter, m
mile, mi	1.610	kilometer, km			

*Approximate.

†The U.S. Customary system unit of the pound-force is often abbreviated as lbf to distinguish it from the pound-mass, which is abbreviated as lbm.