Stresses in Body of Power Screws (Static Screw Stresses)

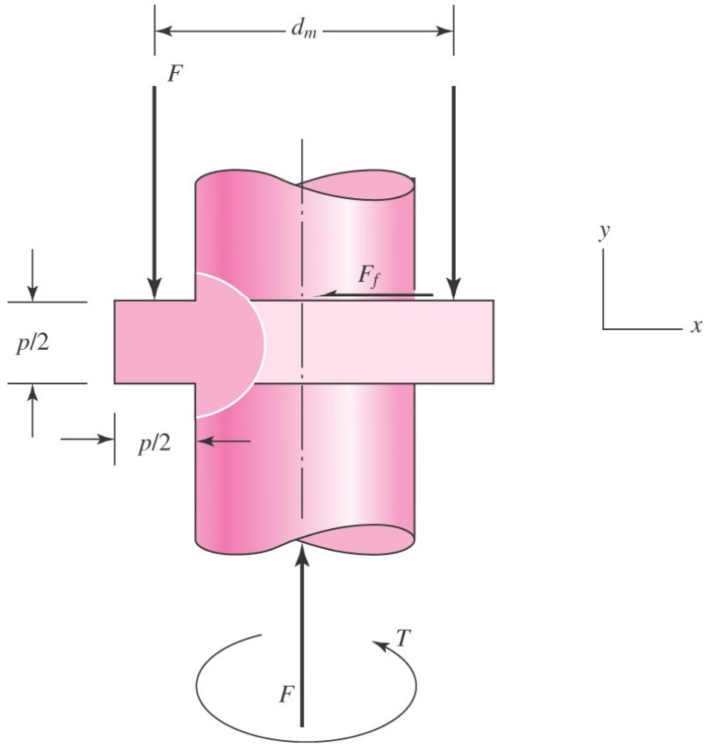
1-Maximum nominal shear stress in torsion of the screw body τ_{yz}

Torsion

$$\tau = \frac{Tc}{J}$$

where $c = d/2$ and $J = \pi d^4/32$

Therefore, $\tau = \dfrac{16T}{\pi d^3}$



For a power screw or threaded fastener, we generally use

$$\tau = \frac{16T}{\pi d_r^3} \qquad (8\text{–}7)$$

2-Axial Stress:

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

Where A is the effective area. For threaded fasteners this is generally the tensile stress area A_t (from tables). For power screws we use d_r.

$$A = \pi d_r^2 / 4$$

Axial stress in screw body σ_y

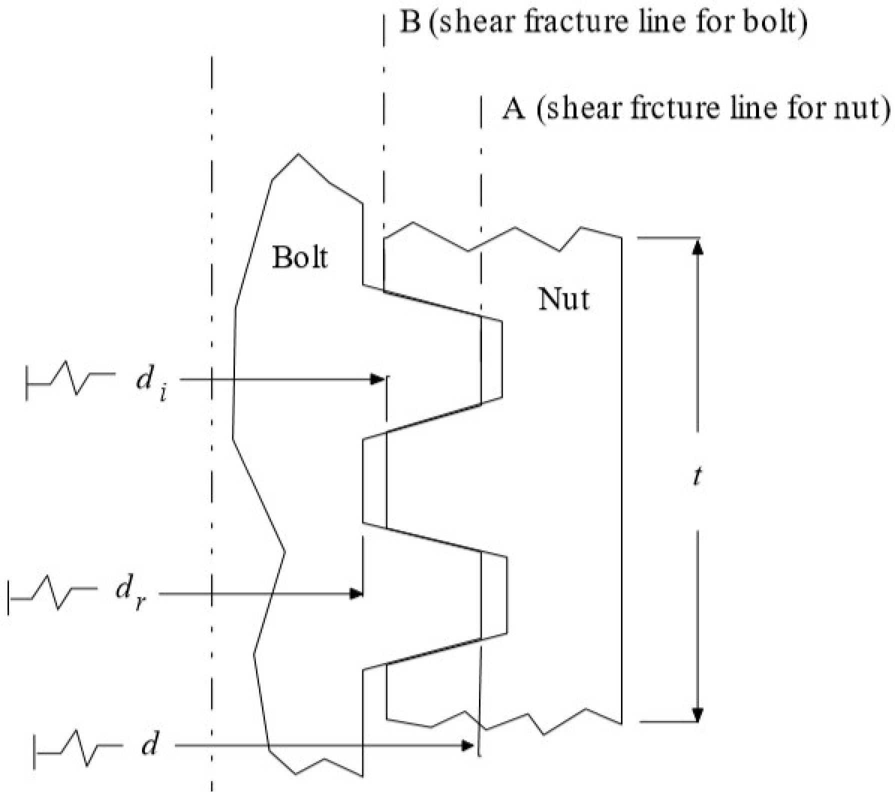
$$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2} \qquad (8\text{–}8)$$

3-Bearing stress in threads,

At the interface of the nut and the bolt (where the threads for each are in contact) a bearing stress is developed. This bearing stress is computed by using the projected area over which the two surfaces are in contact.

$$\sigma_B = -\frac{F}{\pi d_m n_t p/2} = -\frac{2F}{\pi d_m n_t p} \qquad (8\text{–}10)$$

A screw thread is subjected to the localized compressive stress at profile of threads of nut and screw.



d as r d of the bolt.

n :The number of threads in contact is given by the nut thickness divided by the pitch of the threads

n= t/p

4-Bending stress at root of thread,σ_x

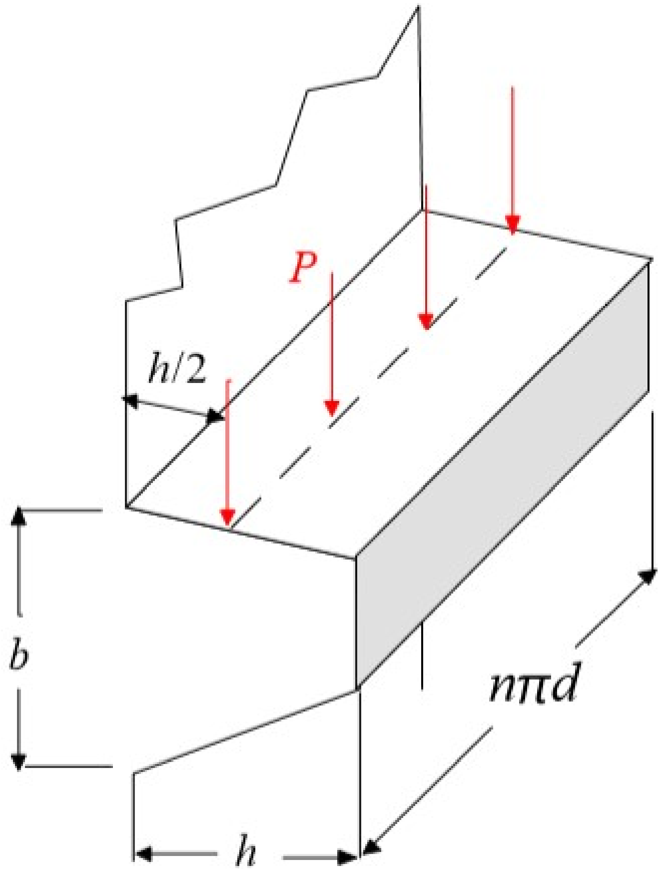
$$\sigma_b = \frac{Mc}{I}$$

$$Z = \frac{I}{c} = \frac{(\pi d_r n_t)\,(p/2)^2}{6} = \frac{\pi}{24} d_r n_t p^2$$

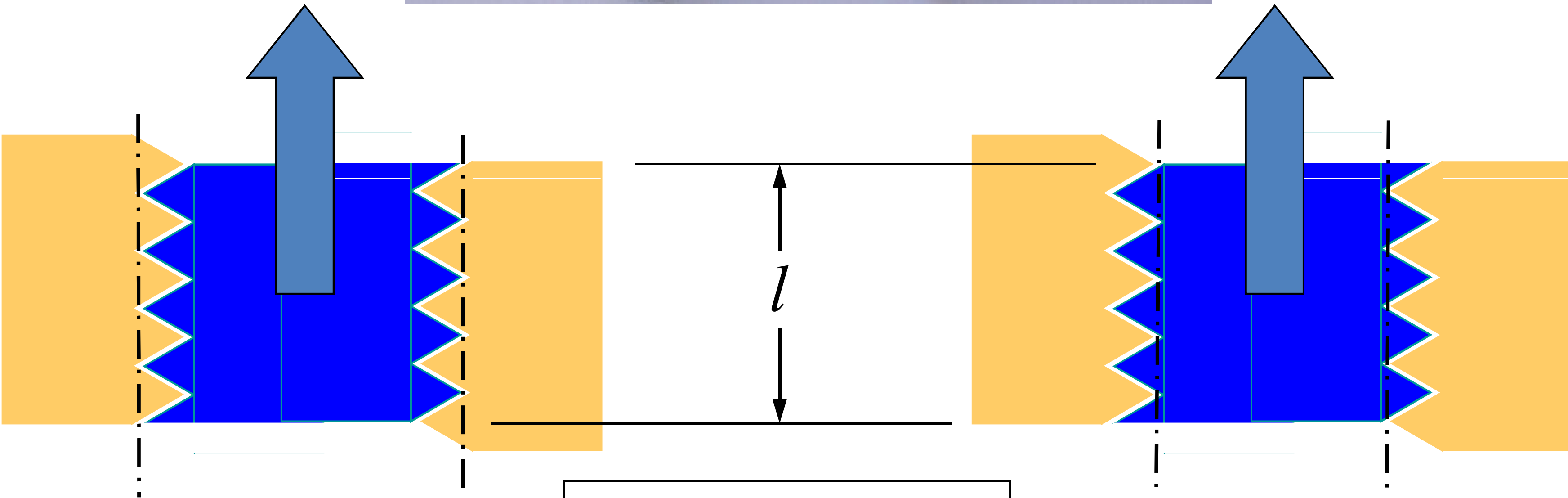
$$M = \frac{Fp}{4}$$

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

(8–11)



Experiments indicate the first thread carries 38% of the load, the second thread 25%, and the third thread 18%. The seventh thread is free of load. To find the largest stress in the first thread of a screw-nut combination, use 0.38*F* in place of *F*, and set $n_t = 1$.



Shear of Nut Threads

$$A_{shear} = \pi d_{crest} l$$

The shear strength of the bolt and nut material may not be the same.

Shear of Bolt Threads

$$A_{shear} = \pi d_{root} l$$

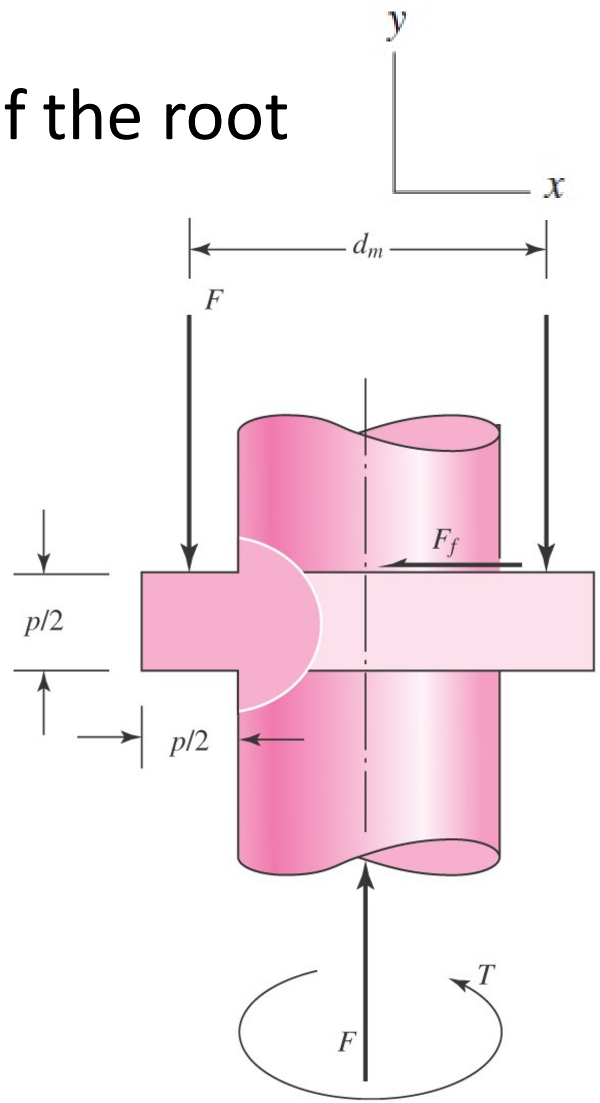
Stresses in Threads of Power Screws

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

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

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

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



Obtain von Mises stress from

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

A screw clamp similar to the one shown in the figure has a handle with diameter $\frac{3}{8}$ in made of cold-drawn AISI 1006 steel. The screw is $\frac{3}{4}$ in-10 UNC made of steel (dry) and the nut material is steel

A force will be applied to the handle at a radius of 3.5 in from the screw centerline."

The clamp will accommodate parts up to 6 in high.

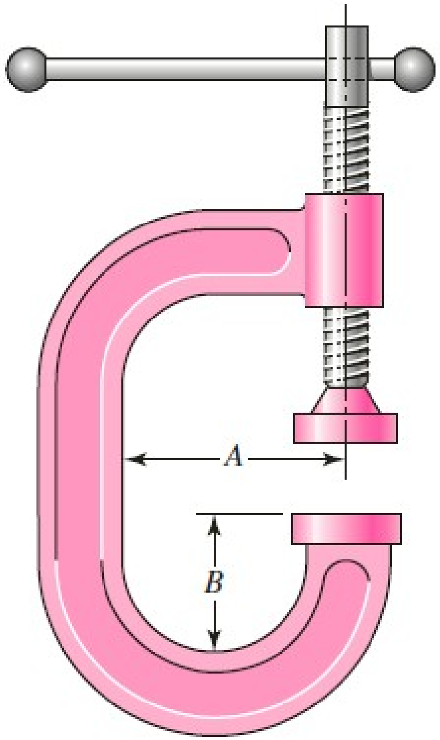
(*a*) What screw torque will cause the handle to bend permanently?

(*b*) What clamping force will the answer to part (*a*) cause if the collar friction is neglected

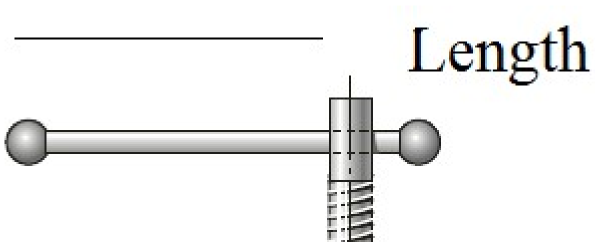
c) calcuate the stresses in the screw threads

Given that nt=4

ACME thread



Problem 8–7



Torque=Length×Force

Length=3.5 inch

T=3.5Force

Bending Moment=distance to the centre of the screw ×Force = (3.5-(3/4)/2)Force=3.125Force

$$Yield-Strength=41000=\frac{32M}{\pi d^3}=\frac{32\times 3.125F}{\pi(0.1875)^3}$$

Force=8.5 lb.f

d/2 wronge

T=3.5×8.5=30 lbf.in

Table 8–2

Diameters and Area of Unified Screw Threads UNC and UNF*

Size Designation	Nominal Major Diameter in	Coarse Series—UNC: Threads per Inch N	Coarse Series—UNC: Tensile-Stress Area A_t in²	Coarse Series—UNC: Minor-Diameter Area A_r in²	Fine Series—UNF: Threads per Inch N	Fine Series—UNF: Tensile-Stress Area A_t in²	Fine Series—UNF: Minor-Diameter Area A_r in²
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

*This table was compiled from ANSI B1.1-1974. The minor diameter was found from the equation $d_r = d - 1.299\,038p$, and the pitch diameter from $d_p = d - 0.649\,519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

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

dm=d-Pitch/2= 0.7 in

dr=d- P/4= 0.65 in

Power Screw Friction Coefficients

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

$$T_R = \frac{F_{\text{clamp}}(0.6850)}{2}\left(\frac{0.1 + \pi(0.2)(0.6850)(1.155)}{\pi(0.6850) - 0.2(0.1)(1.155)}\right)$$

$$T_R = 0.096\, F_{\text{clamp}}$$

$$F_{\text{clamp}} = \frac{T_R}{0.096} = \frac{29.7}{0.096} = 309.38 \text{ lbf}$$

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

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

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

Power Screw Safe Bearing Pressure

-In order to reduce wear of the screw and nut, the bearing pressure on the thread surfaces must be within limits.
- In the design of power screws, the bearing pressure depends upon the materials of the screw and nut, relative velocity between the nut and screw and the nature of lubrication

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

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. (5–4). 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.

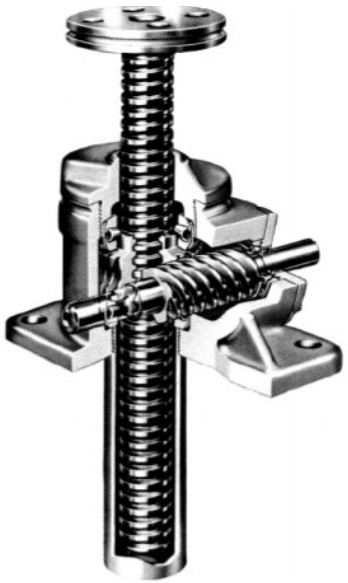
(*b*) Find the torque required to raise the load.

c Find the body stresses, torsional and compressive.

(*d*) Find the thread stresses bending at the root, shear at the root, and von Mises stress and maximum shear stress at the same location.

(e) Find the bearing stress.

(*f*) factor of safety



$$\sigma_x = \frac{6F}{\pi d_r n_t p}$$ bending stress

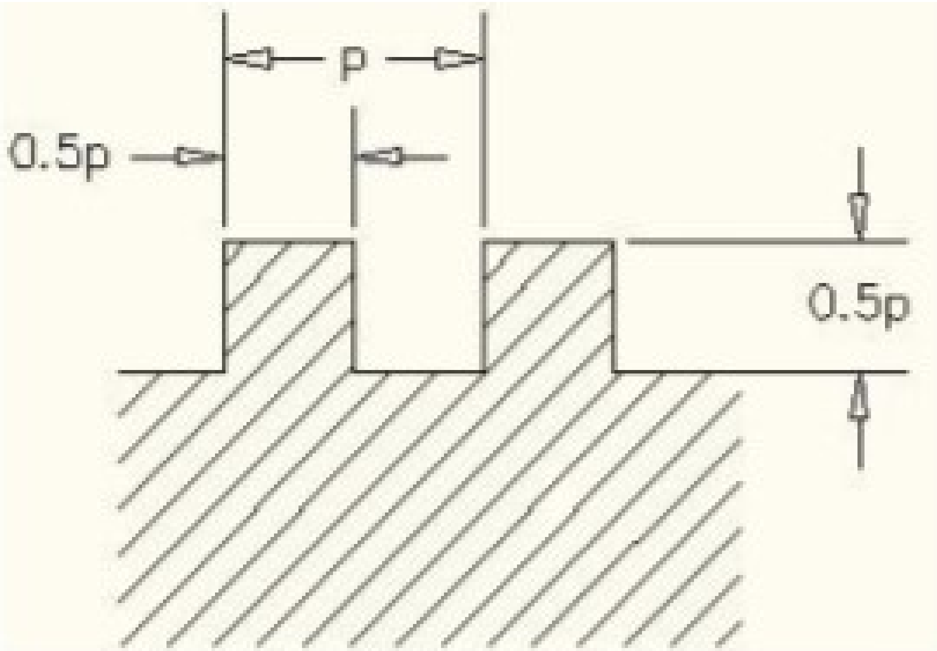
$$\sigma_y = -\frac{4F}{\pi d_r^2}$$ axial normal stress

$$\tau_{yz} = \frac{16T}{\pi d_r^3}$$ shear stress

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

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

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



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

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

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

The body shear stress τ due to torsional moment T_R

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

The axial nominal normal stress σ_y is

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

bending stress σ_b

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

Experiments indicate the first thread carries 38% of the load, the second thread 25%, and the third thread 18%. The seventh thread is free of load. To find the largest stress in the first thread of a screw-nut combination, use 0.38*F* in place of *F*, and set $n_t = 1$.

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

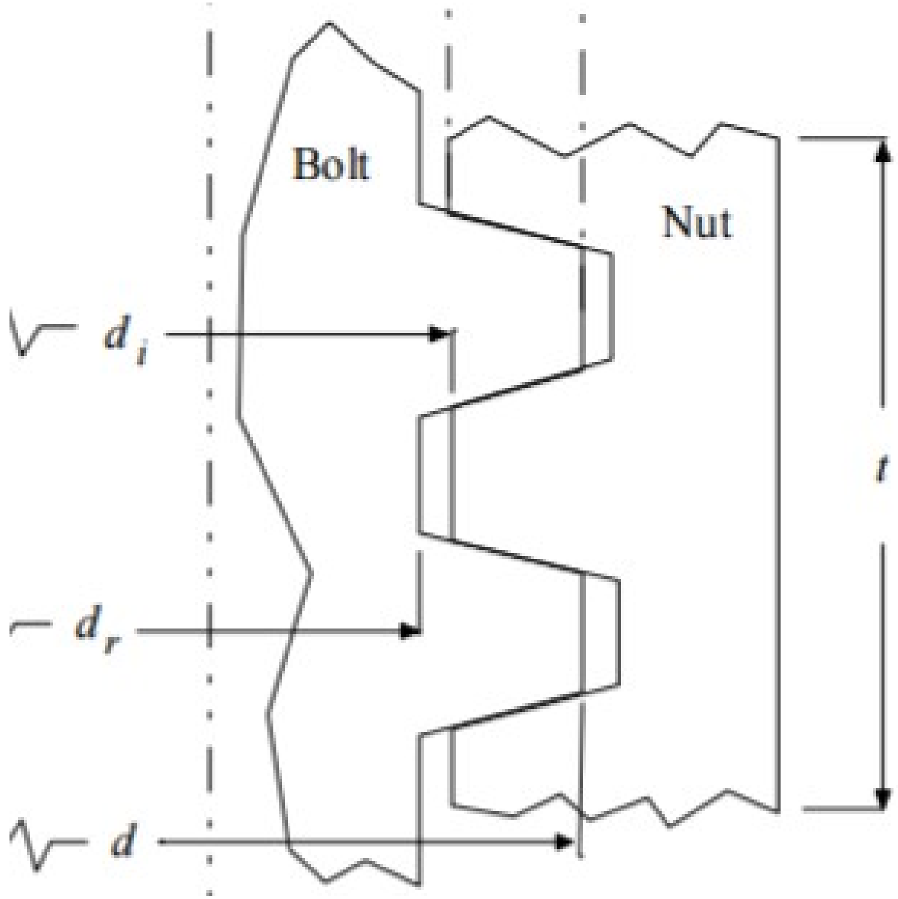
$$\sigma' = (1/\sqrt{2})\{(41.5-0)^2+[0-(-10.39)]^2+(-10.39-41.5)^2+6(6.07)^2\}^{1/2}$$

$$= 48.7 \text{ MPa}$$

AISI 1010 Sy=305 MPa

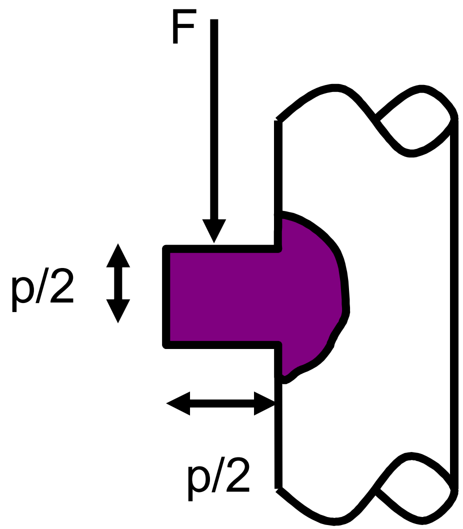
$$n = \frac{Sy}{\sigma'} = \frac{305}{48.7} = 6.4$$

The screws are carburized and heat treated and common materials are AISI 1010, 3310, 4620, and 8620. Nuts are generally made of soft ductile materials such as bronzes and brasses and cast iron due to its rather low friction coefficient.


Bolt
Nut
d_i
d_r
d
t

Thread Stresses – Bearing

$$\sigma_B = \frac{F}{A_{bearing}} = \frac{2F}{\pi d_p n_t p}$$



$A_{bearing} = (p/2)(\pi d_p n_t)$

The bearing stress σ_B is,

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

Example: A screw jack is to lift a load of 80 kN through a height of 400 mm. The elastic strength of screw material in tension and compression is 200 MPa and in shear 10 MPa. The material for nut is phosphor-bronze for which the elastic limit may be taken as 100 MPa in tension, 90 MPa in compression and 80 MPa in shear .The bearing pressure between the nut and the screw is not to exceed 18 N/mm^2. Using a F.S= 2:

1-Design and draw the screw jack. The design should include the design of 1. screw, 2. nut, 3. handle and cup,and 4. body.

Design of screw

$$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2} \qquad (8\text{–}8)$$

$$\frac{\sigma}{F.S} = \frac{F}{A} = \frac{4F}{\pi d_r^2} = \frac{4*\ 8*10^3}{\pi \quad d_r^2} = \frac{200}{2}$$

dr = 32 mm

for safety add 6 mm to the dimensions ,

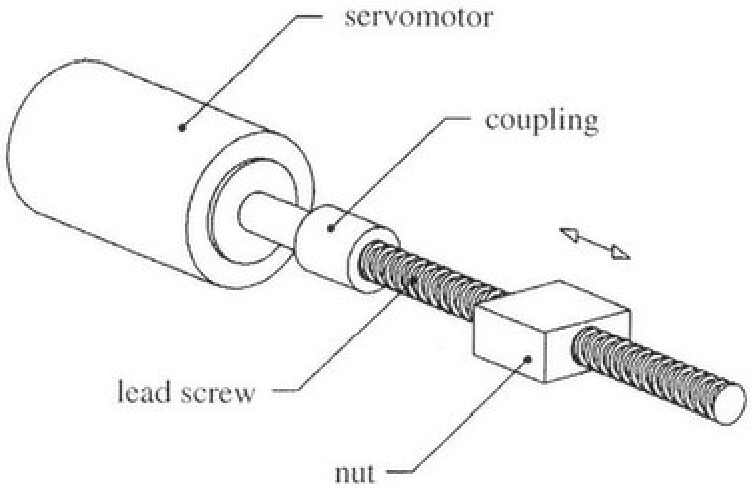
dr = 38 mm d = dr+p= 46 mm dm= dr+p/2= 42 mm

for a pitch = 8 mm

$$\tan \lambda = \frac{Lead}{\pi d_m} = \frac{8}{\pi * 42} = 0.606$$

Assuming coefficient of friction between screw and nut,

$$f = 0.14$$


servomotor
coupling
lead screw
nut