* Power screw

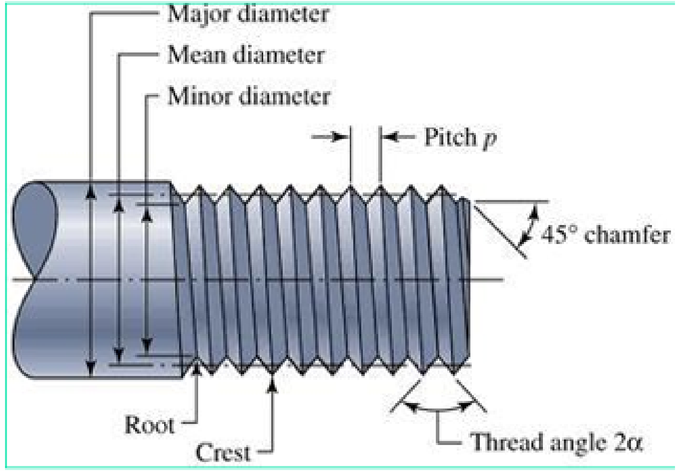
* Fasteners

Screws, Fasteners, and the Design of Non-Permanent Joints

Thread Standards and Definitions

The terminology of screw threads

- ***Pitch (P):*** ***the distance between adjacent threads measured parallel to thread axis.***
- ***Major diameter (D):*** ***the largest diameter of the screw thread.***
- ***Minor diameter (D_1):*** ***also called "root diameter", is the smallest diameter of the screw thread.***
- ***Mean diameter (D_2):*** ***also called "pitch diameter", the average diameter of the screw thread (considering the theoretical full height of the threads).***
- ***Lead (l):*** ***the distance a nut moves parallel to the screw axis when it rotates one full turn.***



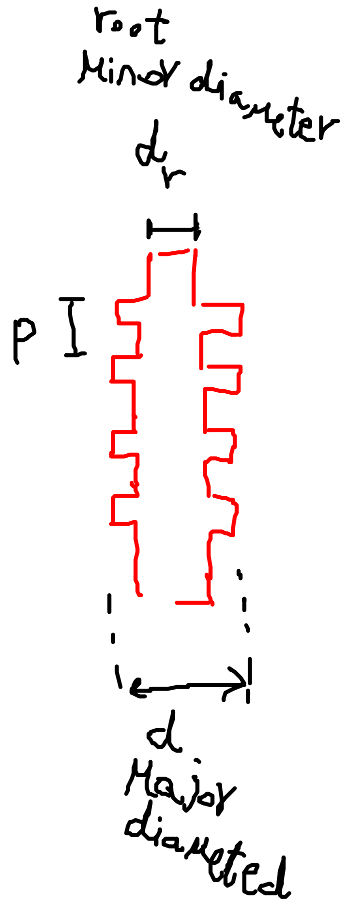
d

d_m — $0.5P$

d_r — $0.5P$

$L = Z * P$

1 single, 2 double, 3 triple



$d_m = \frac{d + d_r}{2}$

Mean diameter

$d - d_r = P$

Thread Standards and Definitions

The terminology of screw threads

✔ A ***multiple-threaded*** ***product is one having*** ***two or more threads*** ***cut beside each other***

✔ In a ***double-threaded***, ***lead = twice pitch***

✔ In a ***triple-threaded screw***, ***lead = 3 times the pitch***

✔ **All threads are usually right-handed unless otherwise is indicated**

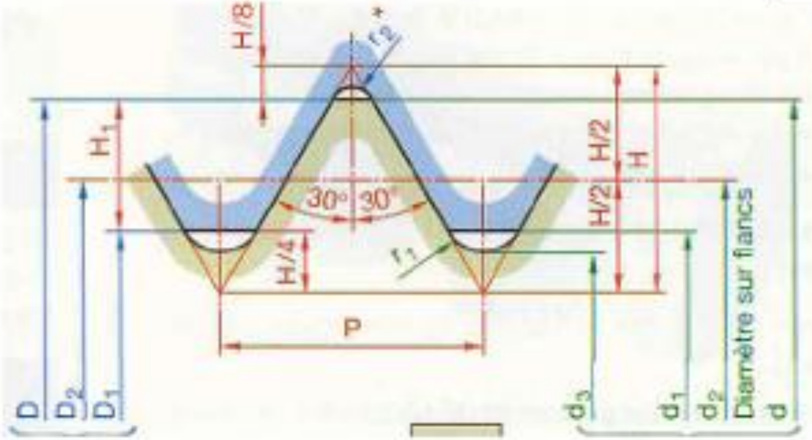
✔ **If the bolt is turned cw, the bolt advances toward the nut.**

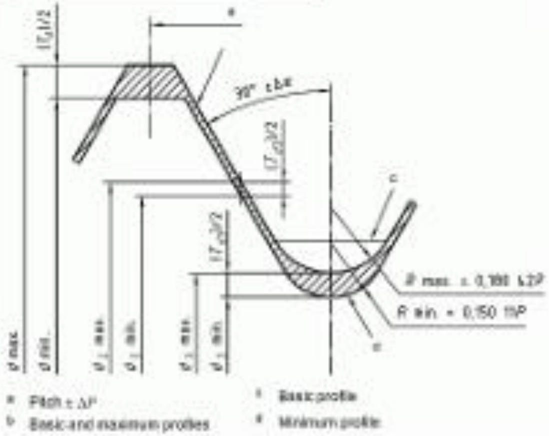
Standardization

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

Metric (ISO): **ISO refers to the International Organization for Standardization, a worldwide federation of national standards bodies**

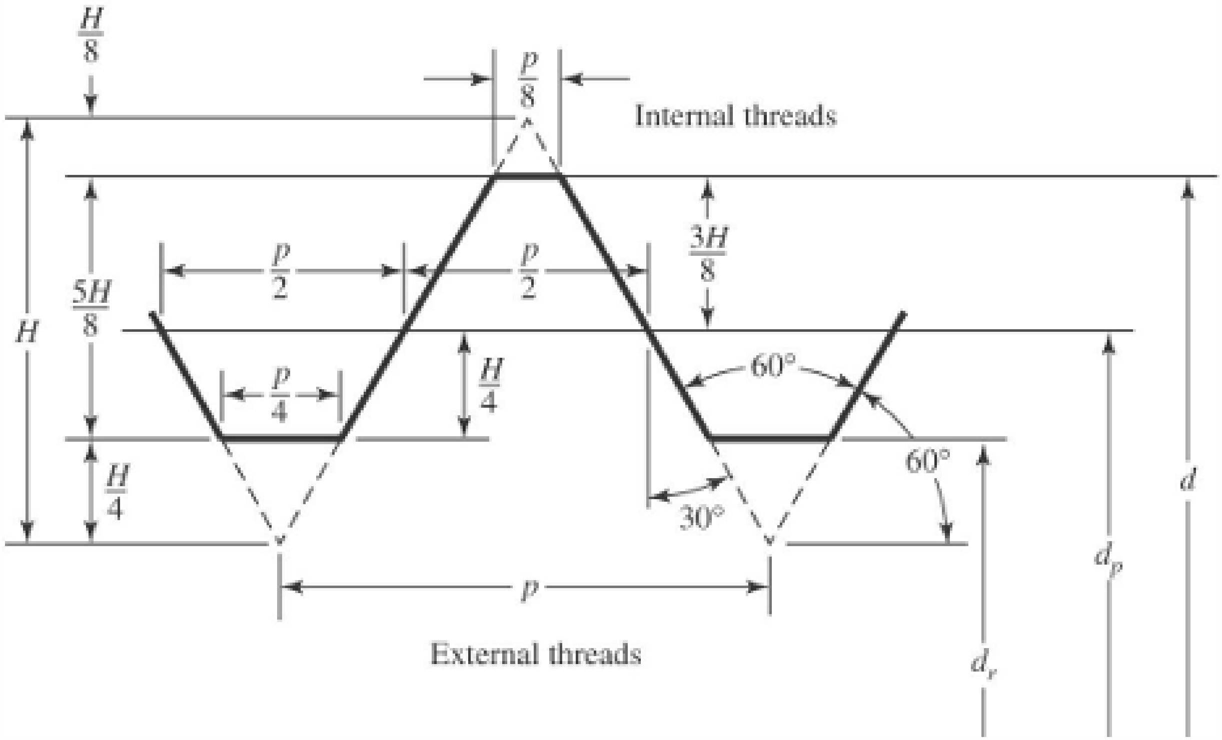
⏩There are two standard profiles <u>M and MJ</u> where both have a similar geometry but the MJ has a rounded fillet *at the root and* a larger minor diameter and therefore it has a better <u>*fatigue strength*</u>.





Standardization

Basic profile for metric M and MJ threads



Metric bolts are specified by the *major diameter* *and the pitch (both in mm).*

- *Example*: $M10 \times 1.5$ (*10 mm* major diameter and *1.5 mm* pitch).

Profile

Standardization

Basic profile for metric M and MJ threads

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

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

↳ **Note that there is <u>Coarse-pitch</u> and <u>Fine-pitch</u> (*more threads) where the fine-pitch has better <u>tensile strength</u>***

Standardization

Coarse-pitch vs Fine-pitch

Fine Thread

- Since they have larger stress areas the bolts are stronger in tension
- Their larger minor diameters develop higher torsional and transverse shear strengths
- They can tap better in thin-walled members
- With their smaller helix angle, they permit closer adjustment accuracy

Coarse Thread

- Stripping strengths are greater for the same length of engagement
- Improved fatigue resistance
- Less likely to cross thread
- Quicker assembly and disassembly
- Tap better in brittle materials
- Larger thread allowances allow for thicker platings and coatings

Standardization

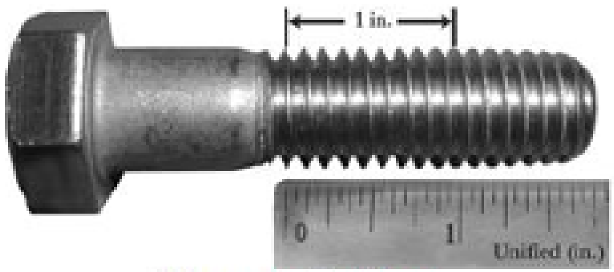
American (Unified): American Standard for Unified Screw Threads

- **There are <u>two standard profiles</u> *UN and UNR where the UNR has a <u>filleted root</u> and thus <u>better fatigue strength</u>*.**

- **Unified threads are specified by the *<u>major diameter</u>* *(in inch) and the number of threads per inch (N).***

  - <u>*Example*</u>: $0.25 - 20\ UNC$

    Diameter → 0.25; (N) → 20; *Profile* → UN; *Coarse <u>or</u> F (Fine)* → C



Threads Per Inch

Standardization

- ❖ *<u>Table 8 -2</u>* gives the <u>standard sizes</u> along with the <u>tensile stress areas</u> and <u>root diameter areas</u> (*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.

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

Standardization

Square and Acme Threads: used when threads are intended to **transmit power**

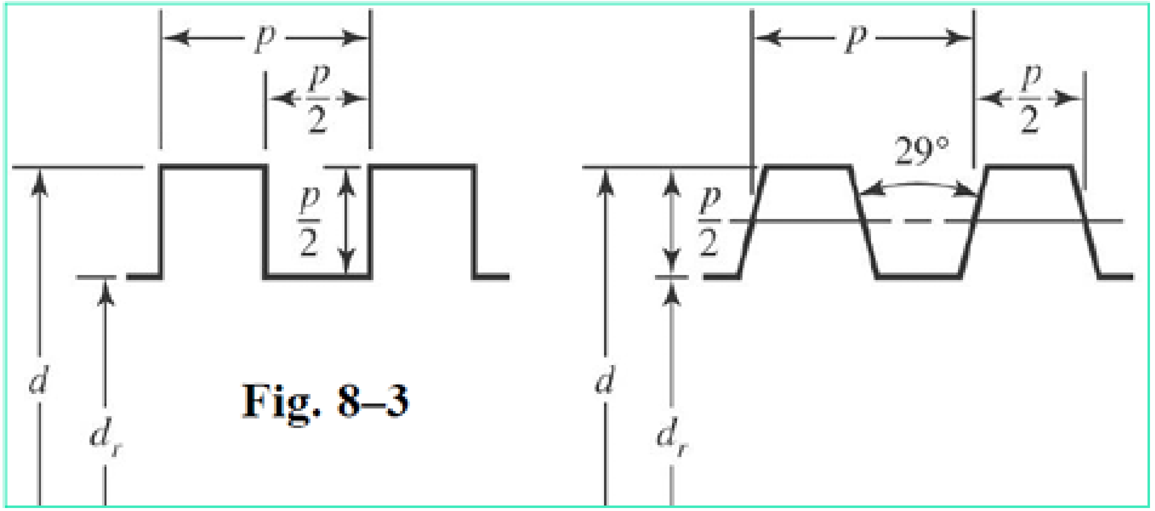


Fig. 8–3

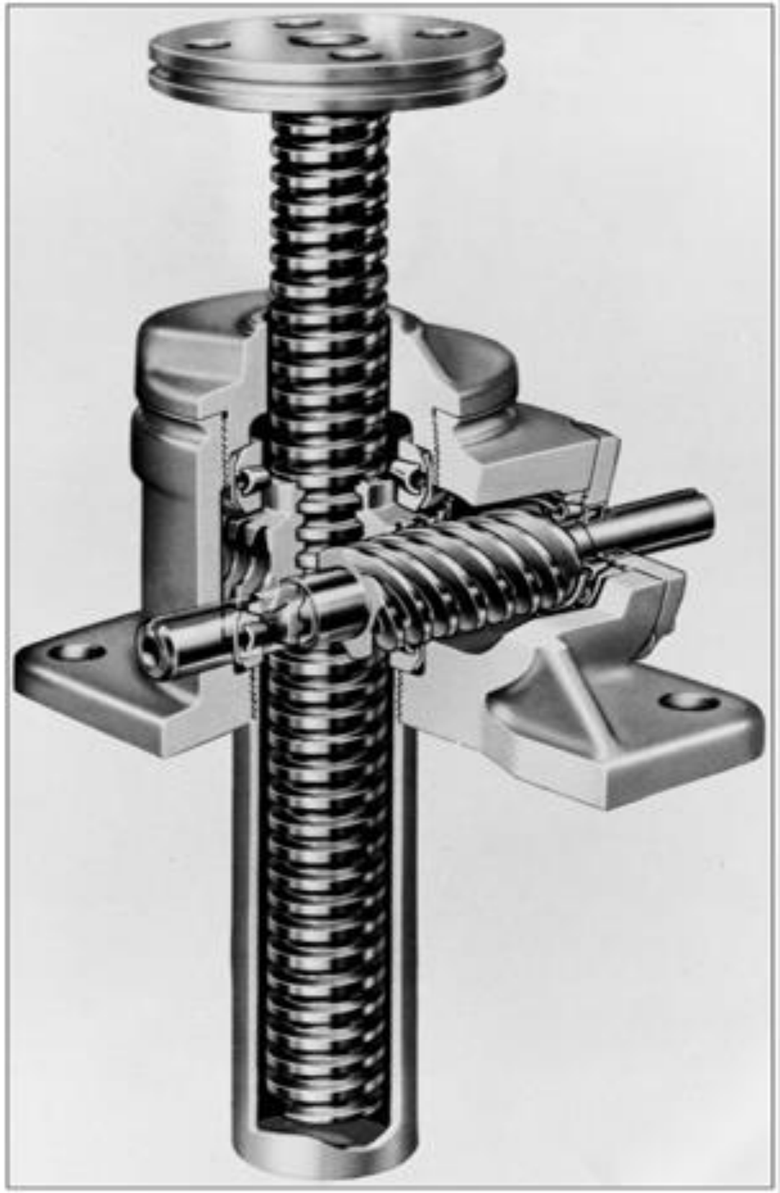
Table 8-3 gives the standard diameters and associated pitch for *Acme thread power screws*

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	$\frac{1}{16}$	$\frac{1}{14}$	$\frac{1}{12}$	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{6}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$

The Mechanics of Power Screws

Power screw

- **Used to change angular motion into linear motion**
- **Usually transmits power**
- **Examples include vises, presses, jacks, lead screw on lathe**



* Torque required to raise the load

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

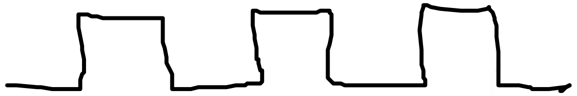
F → load

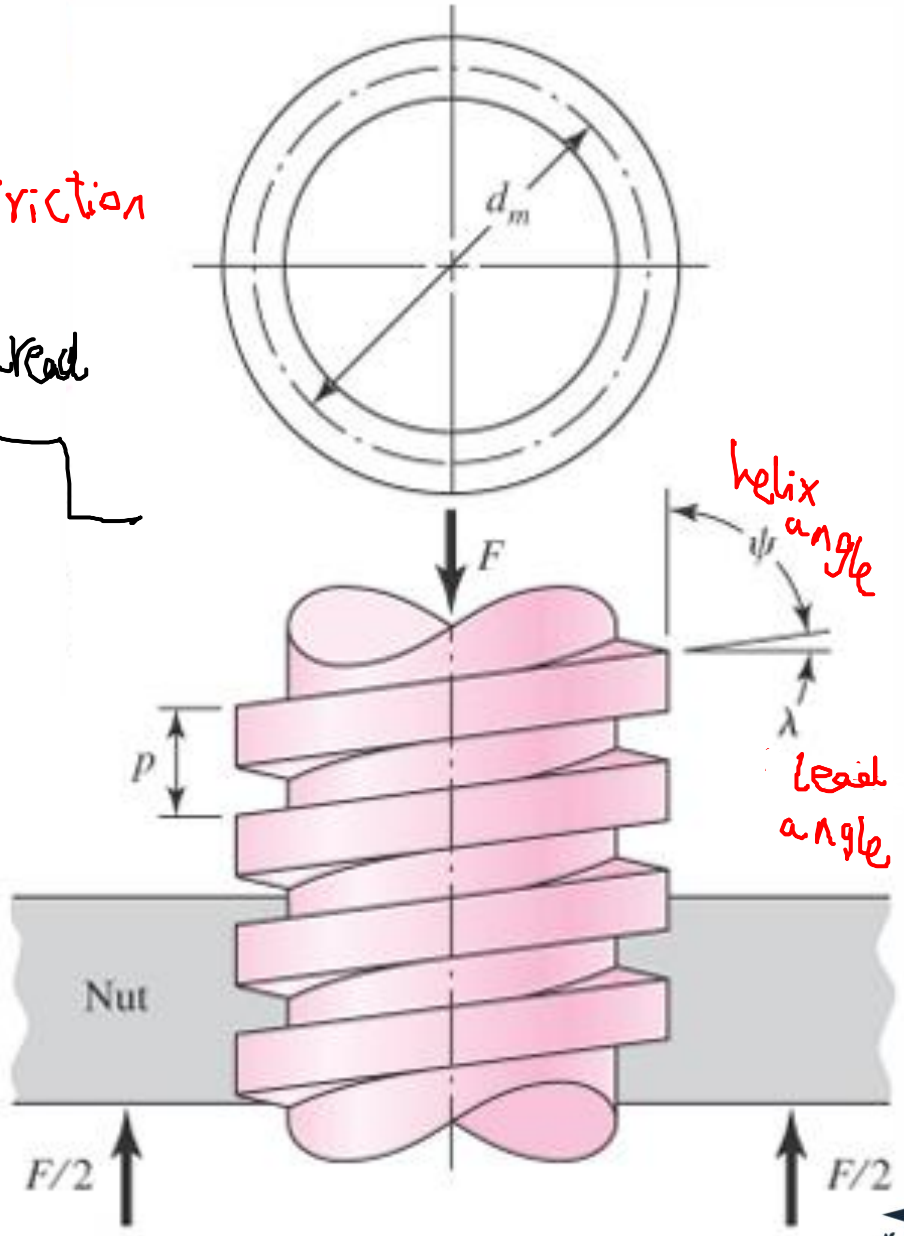
f → Coeficient of friction

* Torque required to lower the load

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

* for square thread





Self-Locking condition

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

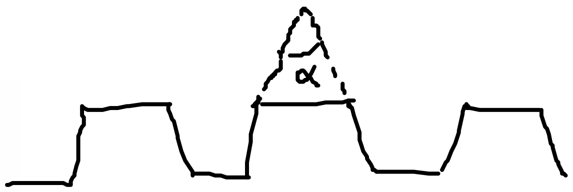
$$\tan\lambda = \frac{L}{\pi d_m}$$

$$e = \frac{T_o}{T_R} = \boxed{\frac{Fl}{2\pi T_R}}$$

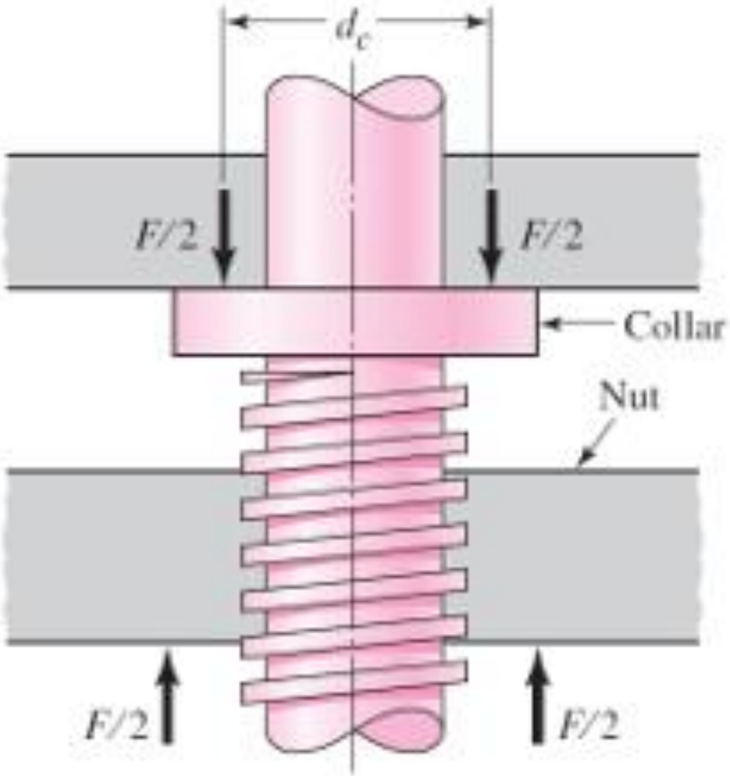
efficiency

* for ACME Thread

$$\boxed{T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m \sec\alpha}{\pi d_m - f l \sec\alpha}\right)}$$



* Collar friction



$$\boxed{T_c = \frac{F f_c d_c}{2}}$$

Torque required to overcome Collar friction

$T_{total} = T_R + T_C \rightarrow$ raise

$T_{total} = T_L + T_C \rightarrow$ lower

Normal stress: $\sigma = \dfrac{F}{A} = \dfrac{4F}{\pi\ d_r^{\ 2}}$

Tension or Compression

Shear due to the torque: $\tau = \dfrac{Tc}{J} = \dfrac{16T}{\pi\ d_r^{\ 3}}$

Maximum at the root

bearing stress

$$\sigma_B = -\frac{F}{\pi d_m n_t p/2} = -\frac{2F}{\pi d_m n_t p}$$

$n_t \rightarrow$ عدد الأسنان اللي شايلة الحمل

$n_t = 1 \rightarrow F = 0{,}38 * \text{load}$

bending stress

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

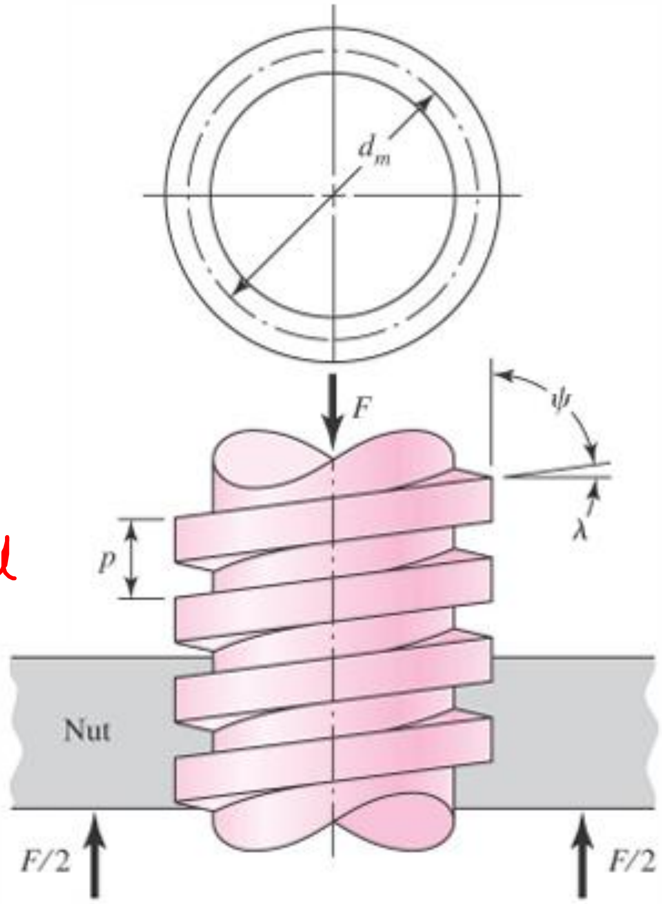
$$\sigma_b = \frac{M}{Z} = \frac{Fp}{4}\frac{24}{\pi d_r n_t p^2} = \frac{6F}{\pi d_r n_t p}$$

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

Max at the top surface of the root

$$\tau = \frac{3V}{2A} = \frac{3}{2}\frac{F}{\pi d_r n_t p/2} = \frac{3F}{\pi d_r n_t p}$$

Max at the center of the root
& zero at the top surface



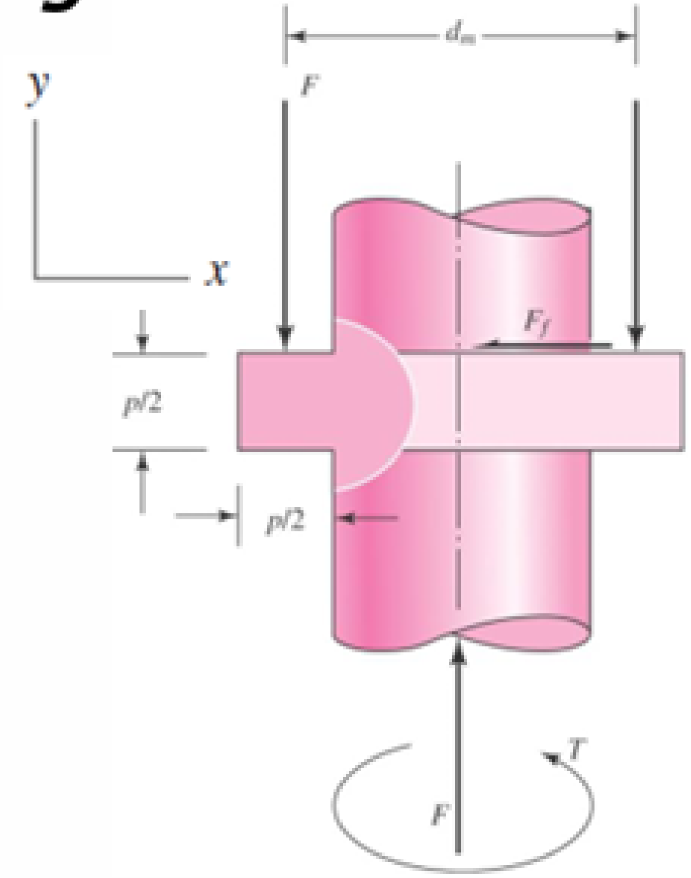
◆Consider stress element at top of the root #plane#

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

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

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

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



Obtain von Mises stress from

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

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.

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

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

Answer

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

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

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

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

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

Answer

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

$d = 32$ mm

$d_m = 30$ mm

$d_r = 28$ mm

$P = 4$ mm, $L = 8$ mm

$$T_R = \frac{6{,}4 * 10^3 * 0{,}03}{2}\left(\frac{0{,}008 + \pi * 0{,}08 * 0{,}03}{\pi * 0{,}03 - 0{,}08 * 0{,}008}\right) + \frac{6{,}4 * 10^3 * 0{,}08 * 0{,}04}{2}$$

$$= 15{,}94 + 10{,}24 = \boxed{26{,}18 \text{ N}}$$

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

$$T_L = \frac{Fd_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + fl}\right) + \frac{Ff_c d_c}{2} = \frac{6{,}4 * 10^3 * 0{,}03}{2}\left(\frac{\pi * 0{,}08 * 0{,}03 - 0{,}008}{\pi * 0{,}03 + 0{,}08 * 0{,}008}\right) + \frac{6{,}4 * 10^3 * 0{,}08 * 0{,}04}{2}$$

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

Answer

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

$$= -0{,}466 + 10{,}24 = \boxed{9{,}77\ \mathrm{N.m}}$$

The minus sign in the first term indicates that the screw alone is not self-locking and would rotate under the action of the load except for the fact that the collar friction is present and must be overcome, too. Thus the torque required to rotate the screw "with" the load is less than is necessary to overcome collar friction alone.

(c) The overall efficiency in raising the load is

Answer

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

$$e = \frac{6{,}4 * 10^3 * 0{,}008}{2\pi(26{,}18)} = 0{,}311 \rightarrow \boxed{31{,}1\%}$$

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

Answer

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

The axial nominal normal stress σ is

Answer

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

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

Answer

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

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

Answer

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

(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

$$\sigma_x = 41.5 \text{ MPa} \qquad \tau_{xy} = 0$$

$$\sigma_y = -10.39 \text{ MPa} \qquad \tau_{yz} = 6.07 \text{ MPa}$$

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

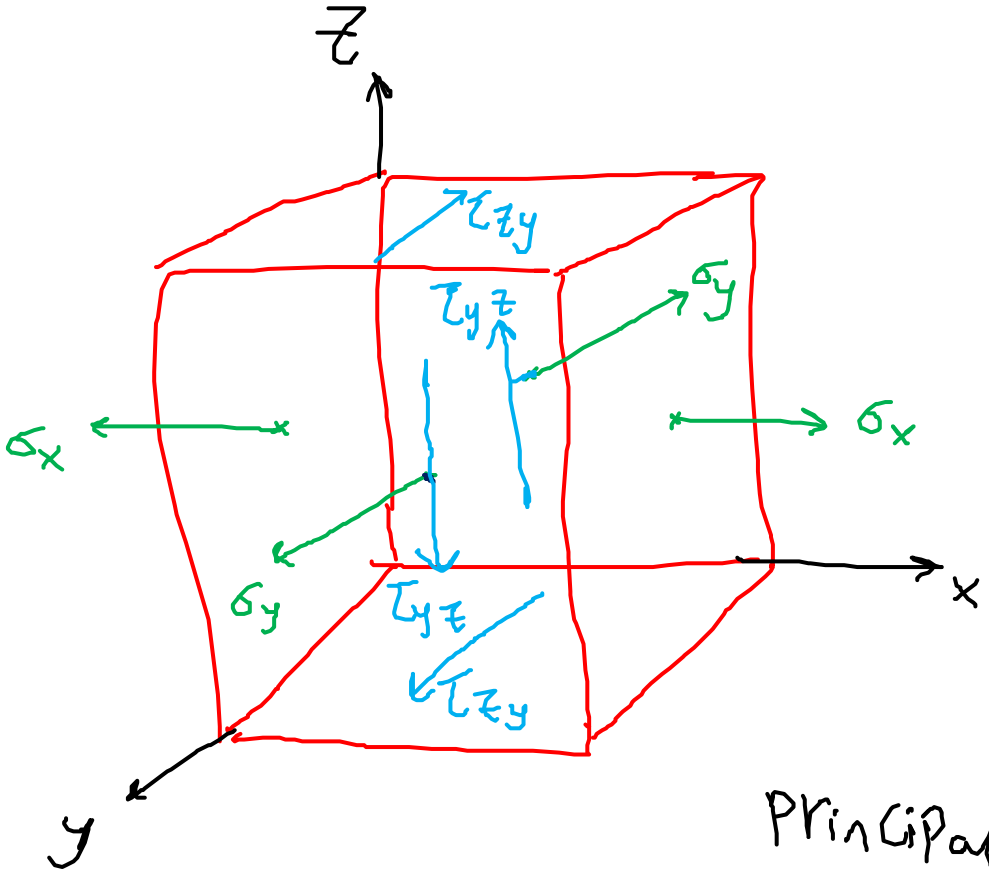
For the von Mises stress, Eq. (5–14) of Sec. 5–5 can be written as

Answer

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

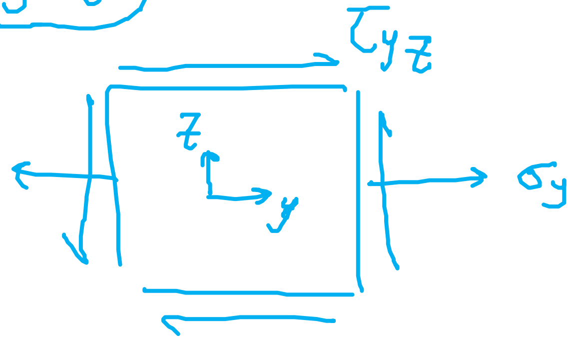
$$= 48.7 \text{ MPa}$$

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



Principal stresses

y - z



$$\sigma_{1,2} = \frac{\sigma_y + \sigma_z}{2} \pm \sqrt{\left(\frac{\sigma_y - \sigma_z}{2}\right)^2 + \tau_{yz}^2}$$

$\sigma_y = -10{,}39\ MPa$

$\sigma_z = \text{Zero}$

$\tau_{yz} = 6{,}07\ MPa$

$$\sigma_{1,2} = \frac{-10{,}39}{2} \pm \sqrt{\left(\frac{-10{,}39}{2}\right)^2 + (6{,}07)^2}$$

$\sigma_1 = 2{,}79\ MPa$, $\sigma_2 = -13{,}18\ MPa$

$\sigma_3 = \sigma_x = 41{,}5\ MPa$

* Von - Mises

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

$$\boxed{\sigma' = \frac{1}{\sqrt{2}}\sqrt{(\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_1-\sigma_3)^2}}$$

$$\sigma' = \frac{1}{\sqrt{2}}\sqrt{(2{,}79+13{,}8)^2 + (2{,}79-41{,}5)^2 + (-13{,}18-41{,}5)^2} = 48{,}7\,MPa$$

* Maximum shear stress

$$\tau_{Max} = \frac{\sigma_{Max} - \sigma_{Min}}{2} = \frac{41{,}5 - (-13{,}18)}{2} = \boxed{27{,}3\,MPa}$$

$\sigma_{Max} \rightarrow$ The largest of $\sigma_1, \sigma_2, \sigma_3$

$\sigma_{Min} \rightarrow$ The smallest of $\sigma_1, \sigma_2, \sigma_3$

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

Answer

$$\sigma' = \left\{\frac{[41.5 - 2.79]^2 + [2.79 - (-13.18)]^2 + [-13.18 - 41.5]^2}{2}\right\}^{1/2}$$
$$= 48.7\ \text{MPa}$$

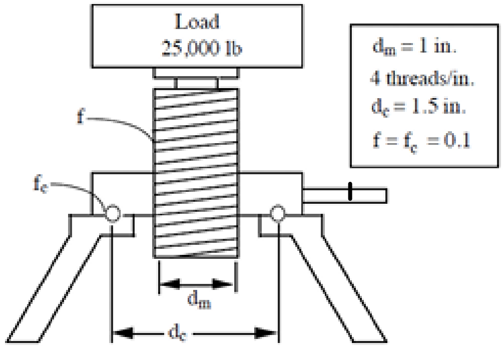
(h) The maximum shear stress is given by Eq. (3–16), where $\tau_{\max} = \tau_{1/3}$, giving

Answer

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

Exercise 1

A square-threaded, single thread power screw is used to raise a known load. The screw has a mean diameter of 1 in. and four threads per inch. The collar mean diameter is 1.5 in. The coefficient of friction is estimated as 0.1 for both the thread and the collar



- Determine the major diameter of the screw
- Estimate the screw torque required to raise and lower (T_R and T_L) the load
- If collar friction is eliminated, determine the minimum value of thread coefficient of friction f needed to prevent the screw from overhauling (self-locking condition)

$d_m = 1$ in, $P = 0{,}25$ in, $d_c = 1{,}5$ in

$f = f_c = 0{,}1$, $L = 0{,}25$ in

* $d = d_m + 0{,}5P = 1 + 0{,}5 * 0{,}25 = 1{,}125$ in

*

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

$$T_R = \frac{25000 * 1}{2}\left(\frac{0{,}25 + \pi * 0{,}1 * 1}{\pi * 1 - 0{,}1 * 0{,}25}\right) + \frac{25000 * 0{,}1 * 1{,}5}{2}$$

$$= 4138{,}24 \text{ lb.in}$$

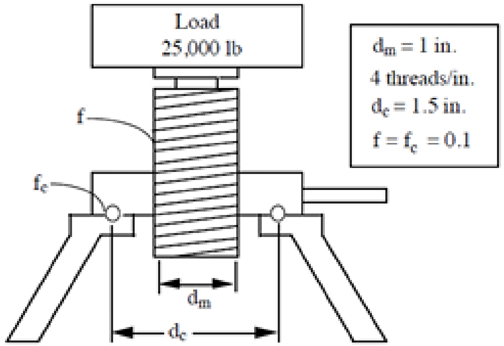
$$T_L = \frac{Fd_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + f l}\right) + \frac{F f_c d_c}{2}$$

$$T_L = \frac{25000 * 1}{2}\left(\frac{\pi * 0{,}1 * 1 - 0{,}25}{\pi * 1 + 0{,}1 * 0{,}25}\right) + \frac{25000 * 0{,}1 * 1{,}5}{2}$$

$$= 253{,}26 + 1875 = 2128{,}26 \text{ lb.in}$$

Exercise 1

A square-threaded, single thread power screw is used to raise a known load. The screw has a mean diameter of 1 in. and four threads per inch. The collar mean diameter is 1.5 in. The coefficient of friction is estimated as 0.1 for both the thread and the collar



- Determine the major diameter of the screw
- Estimate the screw torque required to raise and lower (T_R and T_L) the load
- If collar friction is eliminated, determine the minimum value of thread coefficient of friction *f* needed to prevent the screw from overhauling (self-locking condition)

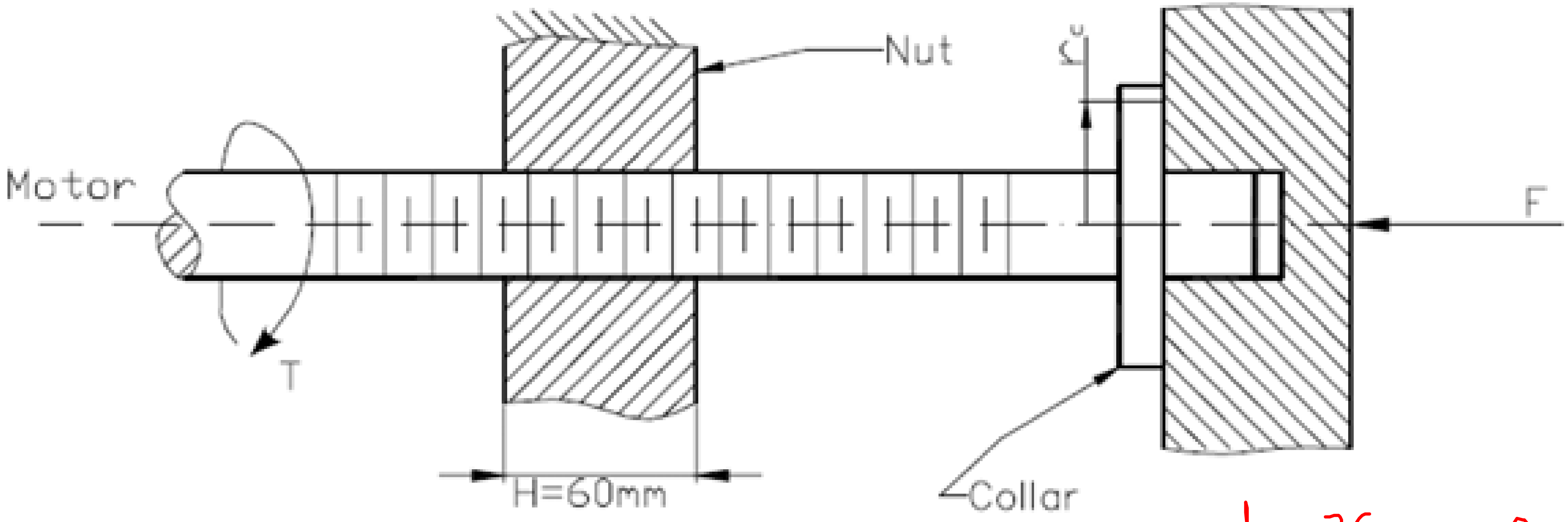
$$* \tan\lambda = \frac{L}{\pi d_m} = \frac{0{,}25}{\pi * 1} = 0{,}07957$$

$$f > 0{,}07957$$

↳ for self locking screw

Exercise 2

A single square-thread power screw has an input power of 3 kW at a speed of 1 rev/s. The screw has a major diameter of 36 mm and a pitch of 6 mm. The frictional coefficients are 0.14 for the threads and 0.09 for the collar, with a collar friction radius of 45 mm.



$d = 36\,mm$, $P = 6\,mm$

$d_m = 33\,mm$, $d_r = 30\,mm$

$L = 6\,mm$, $d_c = 90\,mm$

$f = 0{,}14$, $f_c = 0{,}09$

.1. Find the axial resisting load, **F**,

.2. Find the overall efficiency, **e**.

2.2 Indicate whether the screw is self-locking or not.

$*$ Power = $T * \omega$ $\quad 3*10^3 = T * 2\pi(1)$ $\quad T_{total} = 477{,}7\ N\cdot m$

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

$$477{,}7 = F\left[\frac{0{,}033}{2}\left(\frac{0{,}006 + \pi * 0{,}14 * 0{,}033}{\pi * 0{,}033 - 0{,}14 * 0{,}006}\right) + \frac{0{,}09 * 0{,}09}{2}\right]$$

$$F = 65067{,}56\ N$$

$*$

$$e = \frac{T_o}{T_R} = \boxed{\frac{Fl}{2\pi T_R}}$$

$$e = \frac{65067{,}56 * 0{,}006}{2\pi * 477{,}7} = 0{,}13 \rightarrow \simeq \boxed{13\%}$$

$$\tan\lambda = \frac{L}{\pi d_m} = \frac{6}{\pi * 33} = 0{,}057 < 0{,}14 \xleftarrow{f}$$

$\therefore$ Self locking screw

Threaded Fasteners

The purpose of a bolt or screw is to clamp "*fasten*" *<u>two or more parts</u> together.*

1. The dimensions <u>of bolts and screws</u> are <u>standardized</u> and there are several head styles that are being used

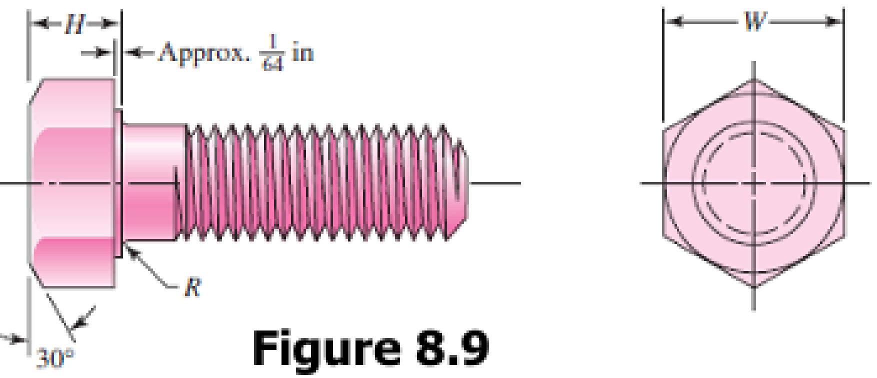


Figure 8.9

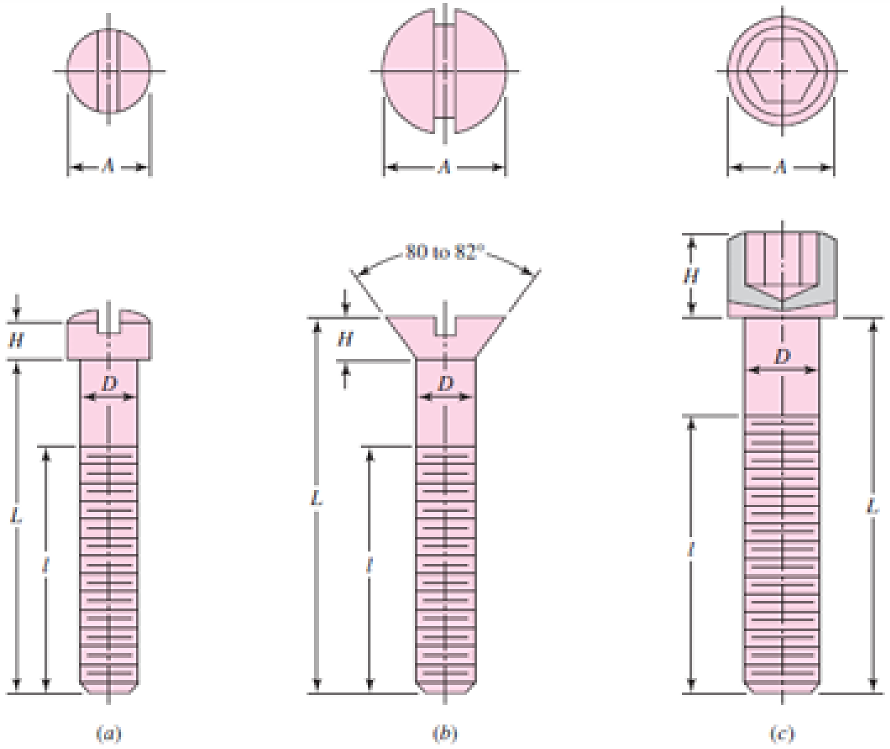


Figure 8.10

Figures 8-9 and 8-10 show some of the common head styles for bolts and cap screws

Threaded Fasteners

- The terms bolt and screw are sometimes used interchangeably and they can refer to the same element. In general, a bolt is used with a nut while a screw is used with a threaded hole (*Not a standard definition).*

bolt with a nut

screw with a threaded hole

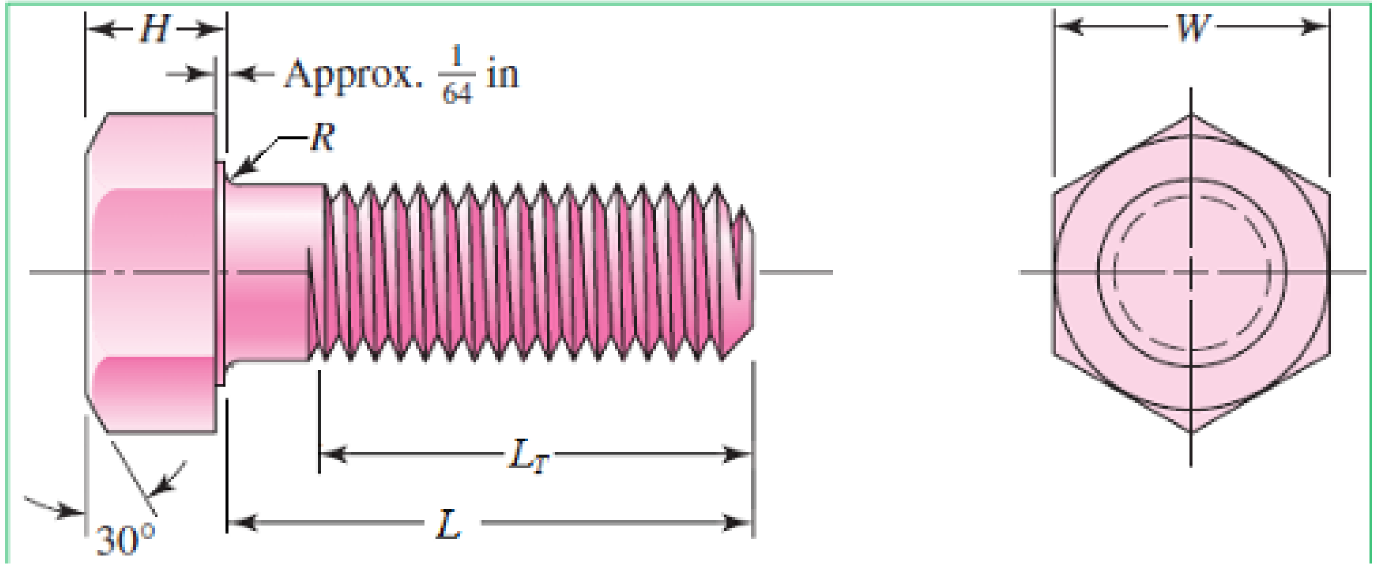
Threaded Fasteners

2. Washers must be used under bolts heads in order to prevent the sharp corner of the hole from biting into bolt head fillet where that increases stress concentration.

Hexagon-Head Bolt

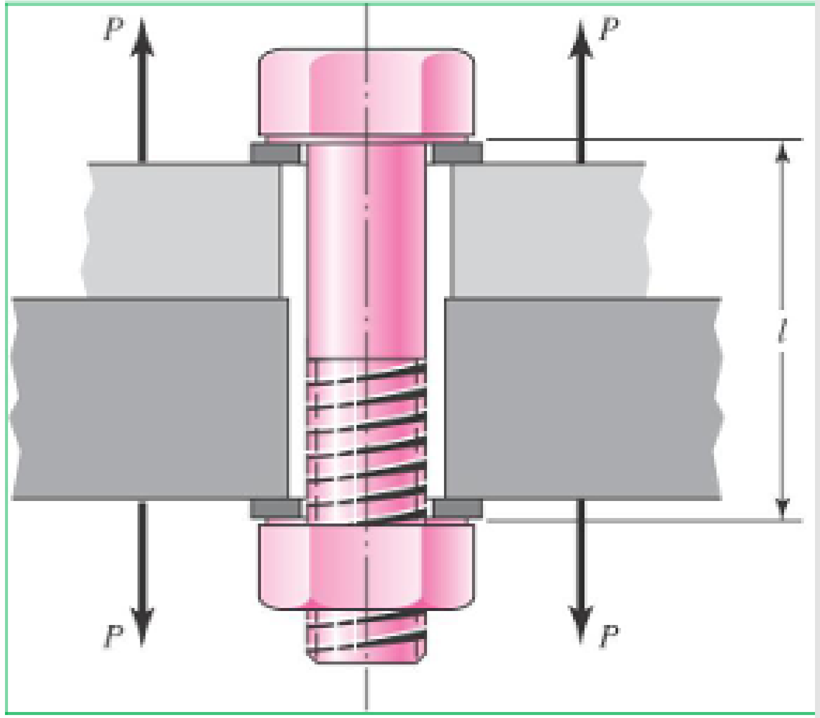
- **Hexagon-head bolts are one <u>of the most common</u> for engineering applications**

- **Standard dimensions are included in <u>Table A–29</u>**
- ***W is usually about <u>1.5</u> times nominal diameter***
- **Bolt length *L is measured from below the head***



Joints with External Load (*Tension Joints)*

◆ are used to clamp two, or more, parts together where these parts are subjected to an external force trying to separate them.



➢ When a bolt and a nut are used to make a joint, the nut is usually tightened to grip the joint firmly.

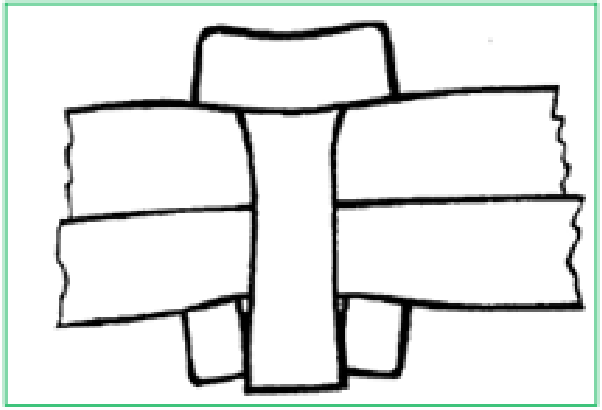
➢ This tightening of the nut introduces a tensile force in the bolt (*called the pre-load) and a compressive force (having the same value) in the clamped material.*

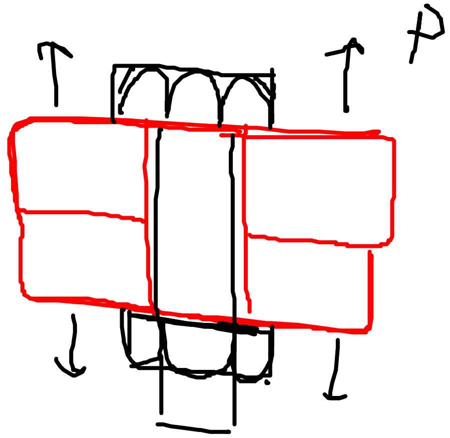
Joints with External Load (*Tension Joints)*

P

bolt P_b Member P_M

➢ When the <u>external "*separating" force* </u>is *applied to the joint it will be divided between the <u>bolt</u> (where it increases the tension in the bolt) and the <u>clamped material</u> (where it reduces the compression in the material).*





- In order to find the <u>portion</u> of the external load carried by the bolt and the portion carried by the material, <u>spring methodology</u> is used where <u>the bolt and the clamped material</u> are represented <u>as two springs</u> in parallel.

(شد) bolt

(ضغط) Member

F_i → pre load

- Therefore, the portion of the external load carried by each of the two springs (*representing the bolt and the clamped material) depends on the stiffness (spring rate) of each of the two springs.*

Joints with External Load (*Tension Joints)*

Procedure to Find Bolt Stiffness

Given fastener diameter d and pitch p in mm or number of threads per inch

Washer thickness: t from Table A–32 or A–33

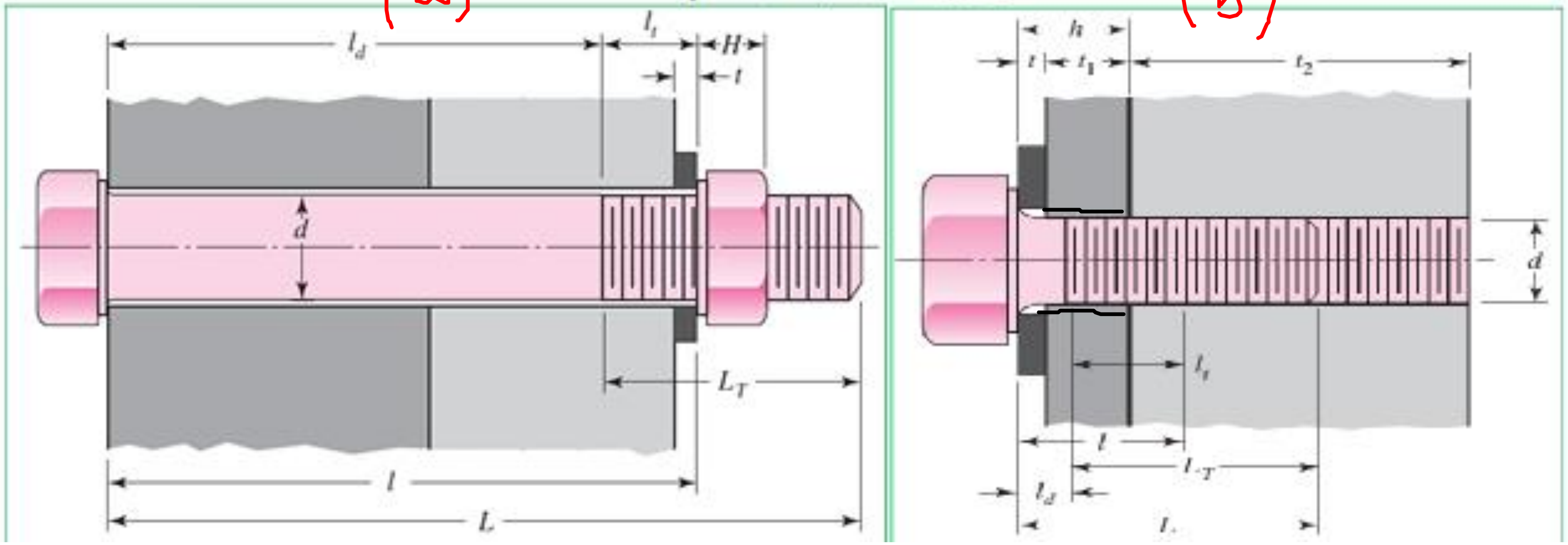
Nut thickness [Fig. (a) only]: H from Table A–31

Grip length:

For Fig. (a): l = thickness of all material squeezed between face of bolt and face of nut

For Fig. (b): $l = \begin{cases} h + t_2/2, & t_2 < d \\ h + d/2, & t_2 \geq d \end{cases}$

(a) (b)



Joints with External Load (*Tension Joints*)

Procedure to Find Bolt Stiffness

Fastener length (round up using Table A–17*):

For Fig. (*a*): $L > l + H$

For Fig. (*b*): $L > h + 1.5d$

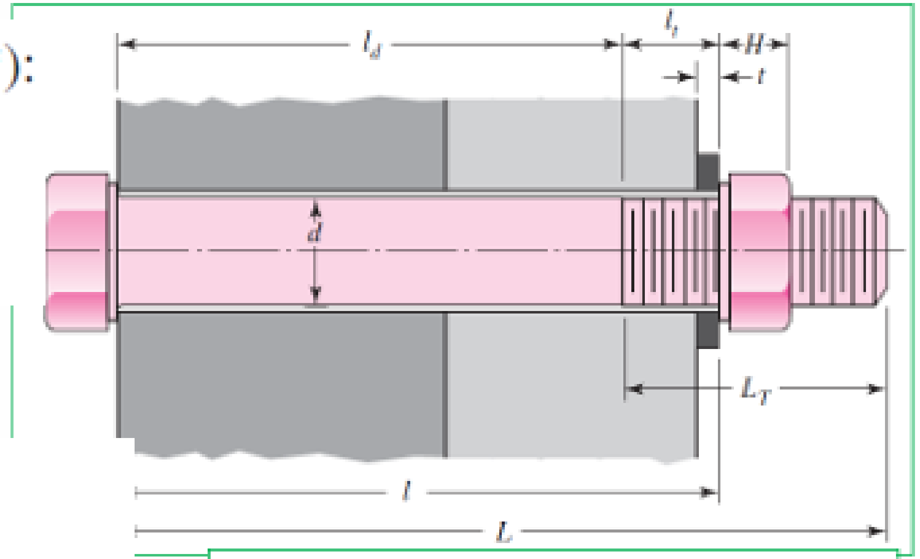
Threaded length L_T:

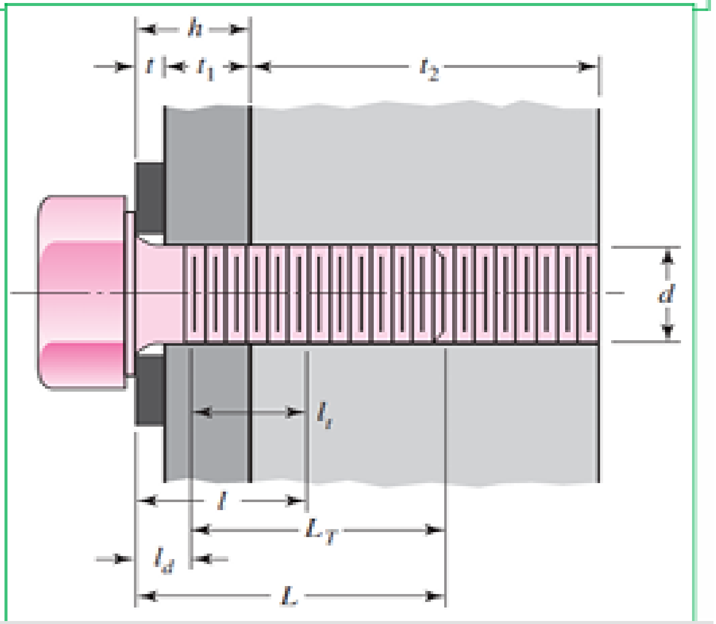
Inch series:

$$L_T = \begin{cases} 2d + \frac{1}{4}\text{ in}, & L \le 6 \text{ in} \\ 2d + \frac{1}{2}\text{ in}, & L > 6 \text{ in} \end{cases}$$

Metric series:

$$L_T = \begin{cases} 2d + 6\text{ mm}, & L \le 125\text{ mm}, d \le 48\text{ mm} \\ 2d + 12\text{ mm}, & 125 < L \le 200\text{ mm} \\ 2d + 25\text{ mm}, & L > 200\text{ mm} \end{cases}$$





Joints with External Load (*Tension Joints*)

Procedure to Find Bolt Stiffness

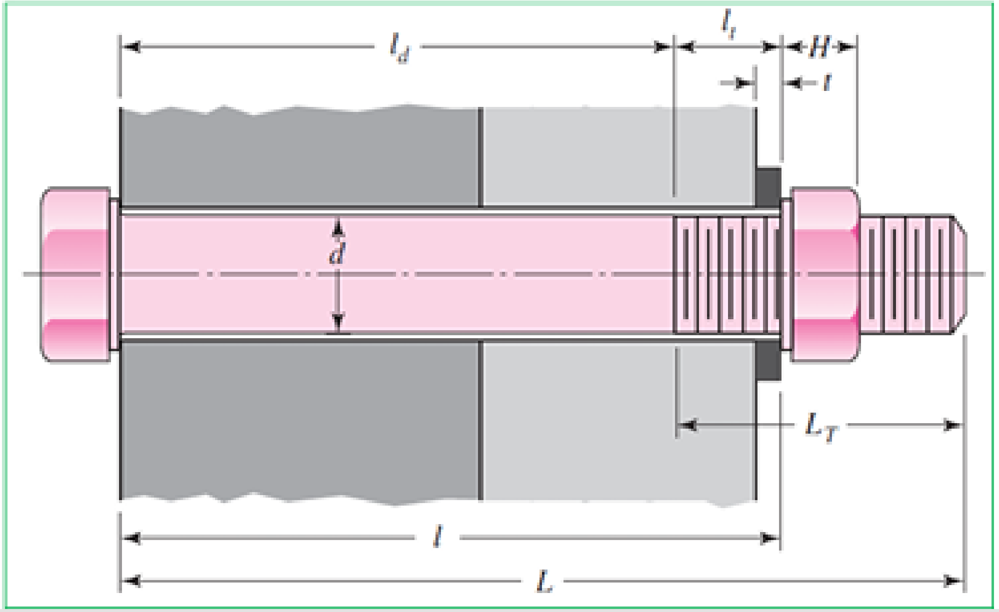
Length of unthreaded portion in grip: $l_d = L - L_T$

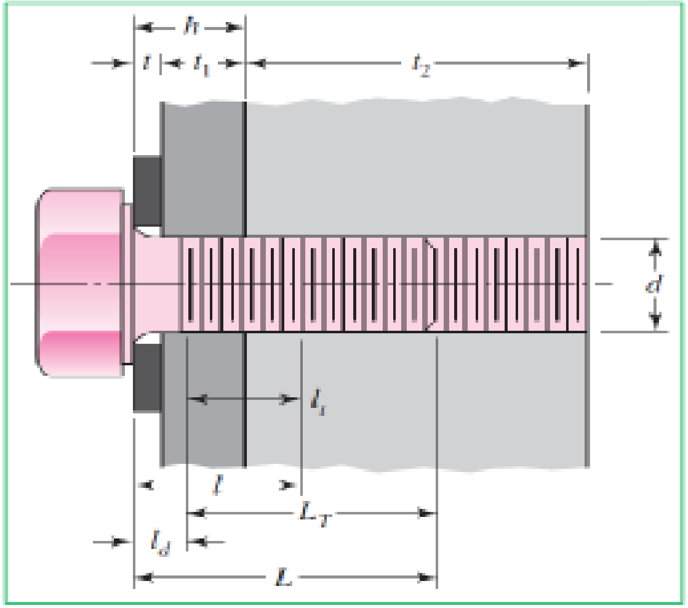
Length of threaded portion in grip: $l_t = l - l_d$

Area of unthreaded portion: $A_d = \pi d^2/4$

Area of threaded portion: A_t from Table 8–1 or 8–2

Fastener stiffness: $k_b = \dfrac{A_d A_t E}{A_d l_t + A_t l_d}$





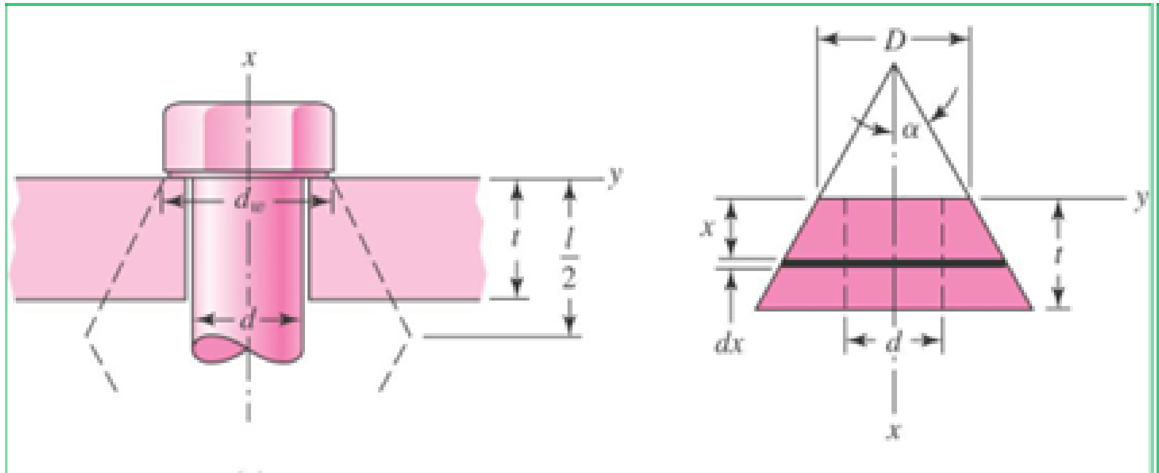
Joints – Member Stiffness

The clamped members will be treated as springs in series in order to find the total stiffness.

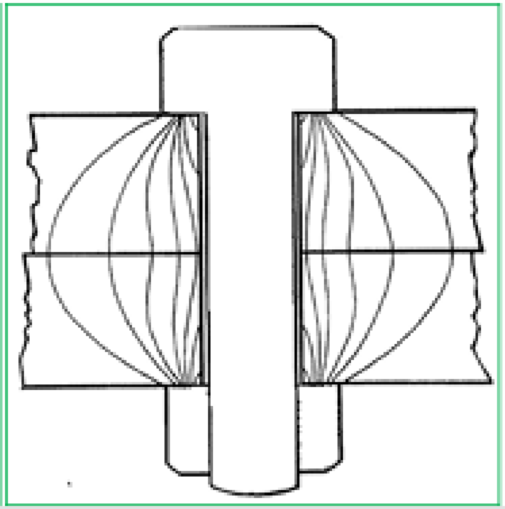
$$1/k = 1/k_1 + 1/k_2 + 1/k_3$$

- **If one of the clamped members is a <u>soft gasket</u> (*which has a very small stiffness compared to other members), the stiffness of other members can be neglected and <u>only the gasket stiffness</u> is used.*)**

- **Experimental investigation showed that the <u>area subjected to compressive stress</u> in the clamped zone has a <u>conical shape</u> with <u>half apex angle</u> of about *30°*.**



(a) (b)



Joints with External Load (*Tension Joints)*

Joints – Member Stiffness

For members made of the <u>same material</u> (*same*), *the* effective stiffness is found to be:

$$\boxed{k_m = \frac{\pi E d \tan\gamma}{2 \ln \dfrac{(l \tan\gamma + d_w - d)(d_w + d)}{(l \tan\gamma + d_w + d)(d_w - d)}}}$$

d_w is the washer face diameter

- Knowing that the standard washer diameter is *50%* greater than the bolt diameter ($d_w = 1.5d$) and $\gamma = 30°$, the effective member stiffness can be <u>simplified</u> as:

$$\boxed{k_m = \frac{0.5774\, \pi\, E\, d}{2\, ln\left(5\, \dfrac{0.5774\, l + 0.5\, d}{0.5774\, l + 2.5\, d}\right)}}$$

Where (l) is the total thickness of the clamped members.

Joints with External Load (*Tension Joints*)

Joints – Member Stiffness

◆ <u>**Alternatively**</u>**, k_m, can be found using a** <u>**curve fit equation**</u> **(*obtained from FEA) which gives a very close value:***

$$\boxed{k_m = Ed\left[A\, e^{Bd/l}\right]}$$

➢ **The *constants A and B* depend on the material being used and their values are given in <u>*Table 8-8.*</u>**

Table 8–8

Stiffness Parameters of Various Member Materials†

Material Used	Poisson Ratio	Elastic GPa	Modulus Mpsi	A	B
Steel	0.291	207	30.0	0.787 15	0.628 73
Aluminum	0.334	71	10.3	0.796 70	0.638 16
Copper	0.326	119	17.3	0.795 68	0.635 53
Gray cast iron	0.211	100	14.5	0.778 71	0.616 16
General expression				0.789 52	0.629 14

†*Source:* J. Wileman, M. Choudury, and I. Green, "Computation of Member Stiffness in Bolted Connections," *Trans. ASME, J. Mech. Design*, vol. 113, December 1991, pp. 432–437.

Tension Joints—The External Load

- During bolt preload
  - ✓ bolt is stretched
  - ✓ members in grip are compressed
- When external load P is applied
  - ✓ Bolt stretches an additional amount δ
  - ✓ Members in grip uncompress same amount δ

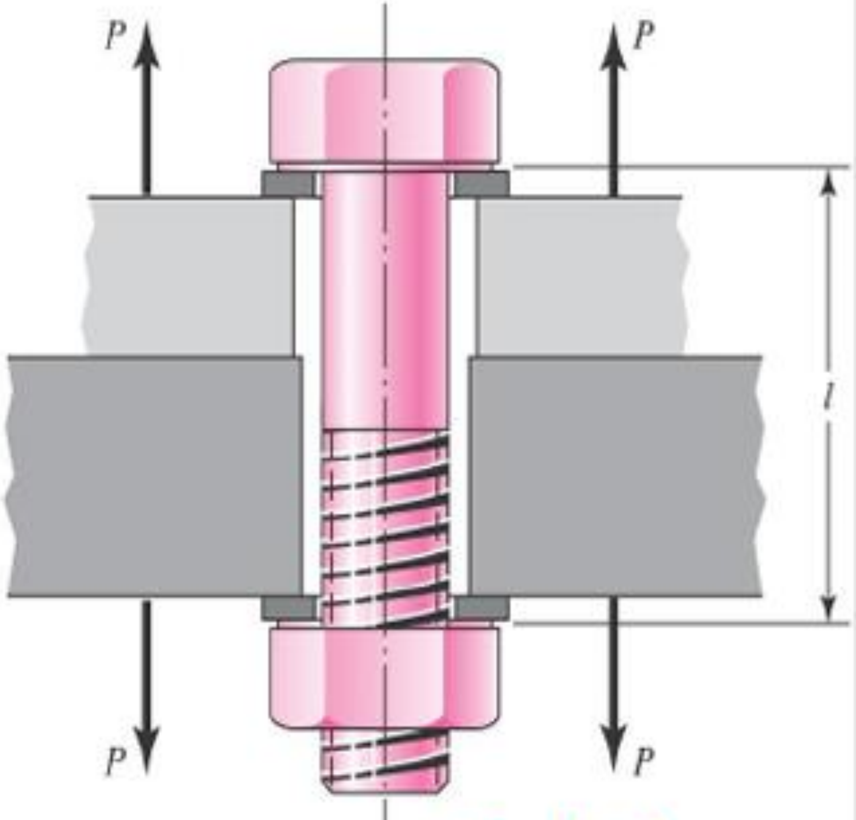


Fig. 8–13

$$\delta = \frac{P_b}{k_b} \quad \text{and} \quad \delta = \frac{P_m}{k_m}$$

$$P_m = P_b \frac{k_m}{k_b}$$

$F_i \rightarrow$ Preload

Member

bolt

$P \rightarrow$ external load

P_m

P_b

Tension Joints—The External Load

- **Since $P = P_b + P_m$**

$$P_b = \frac{k_b P}{k_b + k_m} = C\,P \qquad \text{(d)}$$

$$P_m = P - P_b = (1 - C)\,P \qquad \text{(e)}$$

- ***C*** **is defined as the *stiffness constant* of the joint**

$$C = \frac{k_b}{k_b + k_m} \qquad \text{(f)}$$

- ***C*** **indicates the proportion of external load *P* that the bolt will carry. A good design target is around 0.2.**

Table 8–12

Computation of Bolt and Member Stiffnesses. Steel members clamped using a $\frac{1}{2}$ in-13 NC steel bolt. $C = \dfrac{k_b}{k_b + k_m}$

	Stiffnesses, M lbf/in			
Bolt Grip, in	k_b	k_m	C	$1 - C$
2	2.57	12.69	0.168	0.832
3	1.79	11.33	0.136	0.864
4	1.37	10.63	0.114	0.886

$$\underbrace{P}_{P_b \; + \; P_m}$$

$$P_b = C\,P$$

$$P_m = (1 - C)\,P$$

$$C = \frac{k_b}{k_b + k_m}$$

Relating Bolt Torque to Bolt Tension (*Preload*)

- **Table 8–15:** recommended Torque factors K for use with Eq. (8-27)
- ***K* = 0.2** when bolt condition is not stated

$\alpha = 30°$

Bolt Condition	*K*
Nonplated, black finish	0.30
Zinc-plated	0.20
Lubricated	0.18
Cadmium-plated	0.16
With Bowman Anti-Seize	0.12
With Bowman-Grip nuts	0.09

Mean diameter

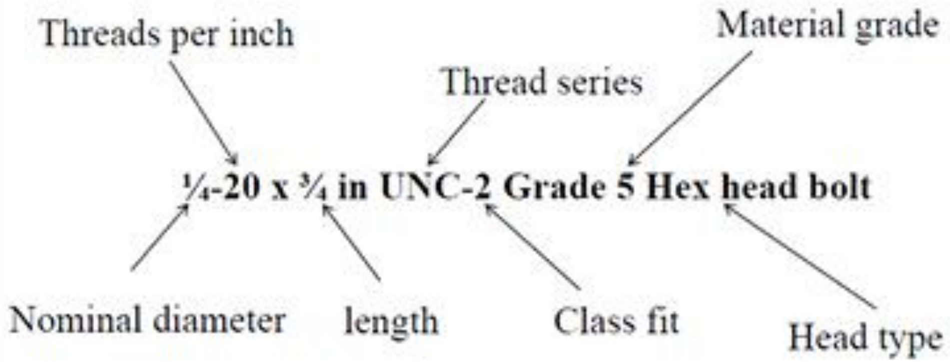
$$K = \left(\frac{d_m}{2d}\right)\left(\frac{\tan\lambda + f\sec\alpha}{1 - f\tan\lambda\sec\alpha}\right) + 0.625 f_c \qquad (8\text{–}26)$$

$$T = K F_i d \qquad (8\text{–}27)$$

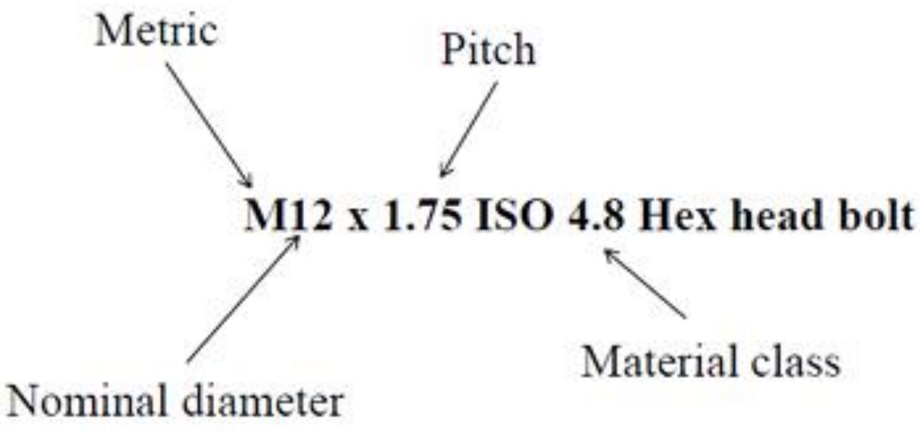
$$\tan\lambda = \frac{l}{\pi d_m}$$

Bolt Materials

Unified Bolt Specification



Metric Bolt Specification



Tension Loaded Bolted Joints: Static Factors of Safety

For a bolt subjected to an external load " P" and having a preload "F_i ", the tensile stress is found as:

P_b

$$\textbf{Axial Stress:} \quad \sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t}$$

The limiting value of σ is the *Proof Strength "S_p", therefore the static factor of safety "n_p" can be calculated as:*

Proof strength (Table 8-9, 8-10, 8-11)

$$\textbf{Yielding Factor of Safety:} \quad n_p = \frac{S_p}{\sigma_b} = \frac{S_p}{(CP + F_i)/A_t} = \frac{S_p A_t}{CP + F_i} \qquad (8\text{–}28)$$

$\frac{P_{total}}{N}$ = الـ load لكل مسمار

N ← عدد المسامير

Tension Loaded Bolted Joints: Static Factors of Safety

However, such factor of safety is not very useful since it is expressed based on the stress on the bolt which is due to both the pre-load and external load. A more useful definition will be a "*load factor*" *of safety that indicates how many times the external load can be increased (since the pre-load value will remain constant) and that can be found as:*

Load Factor:

$$\frac{C n_L P + F_i}{A_t} = S_p \qquad n_L = \frac{S_p A_t - F_i}{CP} \quad (8\text{–}29)$$

Where n_L is the *Load Factor (i.e. the load " P" can be increased "n_L" times for the stress to reach S_p).*

Tension Loaded Bolted Joints: Static Factors of Safety

◆ **It's also necessary to ensure that <u>separation will not occur</u> (*if separation occurs, the bolt will carry the entire load).***

- Separation occurs when:

$$F_m = 0 = (1 - C)n_o P - F_i$$

Thus,

$$\boxed{n_o = \frac{F_i}{P(1 - C)}}$$

Where n_o is the *<u>Load Factor</u> guarding against joint <u>separation</u>*.

➢ **<u>Both</u> load factors n_L & n_0 should be calculated, and the <u>smaller</u> of the two will be the load factor of safety for the joint.**

Tension Loaded Bolted Joints: Static Factors of Safety

- According to the standard, the <u>recommended</u> value of preload is given as:

$$\boxed{F_i} = \begin{cases} 0.75\,F_p & for\ nonpermanent\ connections\ (reused\ fasteners) \\ 0.9\,F_p & for\ permanent\ connetions \end{cases}$$

Where F_p is the <u>*Proof Load*</u>: $\boxed{F_p = S_p A_t}$

➢ If the preload "F_i" is set <u>according to the recommended value</u>, then it is <u>not likely</u> that separation will happen before the stress reaches the proof strength.

For other materials, an approximate value is $S_p = 0.85 S_y$

Example 8-3

A $\frac{3}{4}$ in-16 UNF $\times 2\frac{1}{2}$ in SAE grade 5 bolt is subjected to a load P of 6 kip in a tension joint. The initial bolt tension is $F_i = 25$ kip. The bolt and joint stiffnesses are $k_b = 6.50$ and $k_m = 13.8$ Mlbf/in, respectively.

(*a*) Determine the preload and service load stresses in the bolt. Compare these to the SAE minimum proof strength of the bolt.

(*b*) Specify the torque necessary to develop the preload, using Eq. (8–27).

(*c*) Specify the torque necessary to develop the preload, using Eq. (8–26) with $f = f_c = 0.15$.

Solution From Table 8–2, $A_t = 0.373\ \text{in}^2$.

(*a*) The preload stress is

Answer

$$\sigma_i = \frac{F_i}{A_t} = \frac{25}{0.373} = 67.02\ \text{kpsi}$$

The stiffness constant is

$$C = \frac{k_b}{k_b + k_m} = \frac{6.5}{6.5 + 13.8} = 0.320$$

From Eq. (8–24), the stress under the service load is

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t} = C\frac{P}{A_t} + \sigma_i$$

Answer

$$= 0.320\frac{6}{0.373} + 67.02 = 72.17\ \text{kpsi}$$

From Table 8–9, the SAE minimum proof strength of the bolt is $S_p = 85$ kpsi. The preload and service load stresses are respectively 21 and 15 percent less than the proof strength.

(*b*) From Eq. (8–27), the torque necessary to achieve the preload is

Answer

$$T = KF_i d = 0.2(25)(10^3)(0.75) = 3750 \text{ lbf} \cdot \text{in}$$

(*c*) The minor diameter can be determined from the minor area in Table 8–2. Thus $d_r = \sqrt{4A_r/\pi} = \sqrt{4(0.351)/\pi} = 0.6685$ in. Thus, the mean diameter is $d_m = (0.75 + 0.6685)/2 = 0.7093$ in. The lead angle is

$$\lambda = \tan^{-1}\frac{l}{\pi d_m} = \tan^{-1}\frac{1}{\pi d_m N} = \tan^{-1}\frac{1}{\pi(0.7093)(16)} = 1.6066°$$

For $\alpha = 30°$, Eq. (8–26) gives

$$T = \left\{\left[\frac{0.7093}{2(0.75)}\right]\left[\frac{\tan 1.6066° + 0.15(\sec 30°)}{1 - 0.15(\tan 1.6066°)(\sec 30°)}\right] + 0.625(0.15)\right\}25(10^3)(0.75)$$

$$= 3551 \text{ lbf} \cdot \text{in}$$

which is 5.3 percent less than the value found in part (*b*).

Tension Loaded Bolted Joints: Static Factors of Safety

Example 8-4

Figure 8–19 is a cross section of a grade 25 cast-iron pressure vessel. A total of N bolts are to be used to resist a separating force of 36 kip.

(a) Determine k_b, k_m, and C.

(b) Find the number of bolts required for a load factor of 2 where the bolts may be reused when the joint is taken apart.

(c) With the number of bolts obtained in part (b), determine the realized load factor for overload, the yielding factor of safety, and the load factor for joint separation.

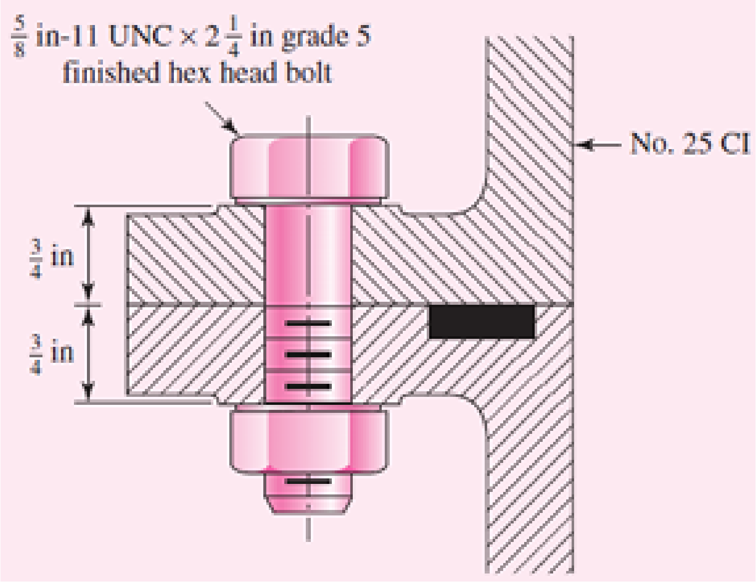


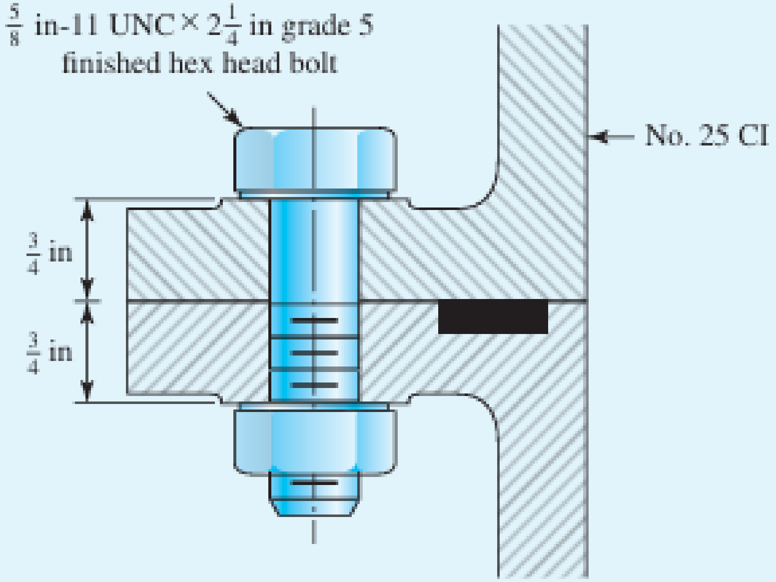
Fig. 8–19

Solution (*a*) The grip is $l = 1.50$ in. From Table A–31, the nut thickness is $\frac{35}{64}$ in. Adding two threads beyond the nut of $\frac{2}{11}$ in gives a bolt length of

$$L = \frac{35}{64} + 1.50 + \frac{2}{11} = 2.229 \text{ in}$$

From Table A–17 the next fraction size bolt is $L = 2\frac{1}{4}$ in. From Eq. (8–13), the thread length is $L_T = 2(0.625) + 0.25 = 1.50$ in. Thus, the length of the unthreaded portion

Figure 8–19



in the grip is $l_d = 2.25 - 1.50 = 0.75$ in. The threaded length in the grip is $l_t = l - l_d = 0.75$ in. From Table 8–2, $A_t = 0.226\ \text{in}^2$. The major-diameter area is $A_d = \pi(0.625)^2/4 = 0.3068\ \text{in}^2$. The bolt stiffness is then

Answer

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.3068(0.226)(30)}{0.3068(0.75) + 0.226(0.75)}$$
$$= 5.21\ \text{Mlbf/in}$$

From Table A–24, for no. 25 cast iron we will use $E = 14$ Mpsi. The stiffness of the members, from Eq. (8–22), is

Answer

$$k_m = \frac{0.5774\pi E d}{2\ln\left(5\dfrac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)} = \frac{0.5774\pi(14)(0.625)}{2\ln\left[5\dfrac{0.5774(1.5) + 0.5(0.625)}{0.5774(1.5) + 2.5(0.625)}\right]}$$
$$= 8.95\ \text{Mlbf/in}$$

If you are using Eq. (8–23), from Table 8–8, $A = 0.778\,71$ and $B = 0.616\,16$, and

$$k_m = EdA\exp(Bd/l)$$
$$= 14(0.625)(0.778\,71)\exp[0.616\,16(0.625)/1.5]$$
$$= 8.81\ \text{Mlbf/in}$$

which is only 1.6 percent lower than the previous result.

From the first calculation for k_m, the stiffness constant C is

Answer

$$C = \frac{k_b}{k_b + k_m} = \frac{5.21}{5.21 + 8.95} = 0.368$$

(b) From Table 8–9, $S_p = 85$ kpsi. Then, using Eqs. (8–31) and (8–32), we find the recommended preload to be

$$F_i = 0.75A_tS_p = 0.75(0.226)(85) = 14.4 \text{ kip}$$

For N bolts, Eq. (8–29) can be written

$$n_L = \frac{S_pA_t - F_i}{C(P_{\text{total}}/N)} \tag{1}$$

or

$$N = \frac{Cn_LP_{\text{total}}}{S_pA_t - F_i} = \frac{0.368(2)(36)}{85(0.226) - 14.4} = 5.52$$

Answer Six bolts should be used to provide the specified load factor.

(c) With six bolts, the load factor actually realized is

Answer
$$n_L = \frac{85(0.226) - 14.4}{0.368(36/6)} = 2.18$$

From Eq. (8–28), the yielding factor of safety is

Answer
$$n_p = \frac{S_pA_t}{C(P_{\text{total}}/N) + F_i} = \frac{85(0.226)}{0.368(36/6) + 14.4} = 1.16$$

From Eq. (8–30), the load factor guarding against joint separation is

Answer
$$n_0 = \frac{F_i}{(P_{\text{total}}/N)(1 - C)} = \frac{14.4}{(36/6)(1 - 0.368)} = 3.80$$

Problem 3

$N=6$

For a bolted assembly with six bolts, the stiffness of each bolt is k_b = 3 Mlbf/in and the stiffness of the members is k_m =12 Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. Assume the bolts are preloaded to 75 percent of the proof load.

(*a*) Determine the yielding factor of safety.

(*b*) Determine the overload factor of safety.

(*c*) Determine the factor of safety based on joint separation.

$$C = \frac{k_b}{k_b + k_m} = \frac{3}{3+12} = 0,2$$

$$P_{total} = 80 \text{ kips}$$

$$F_i = 0,75 * S_p * A_t$$ $F_p \rightarrow$ Proof load

table 8-9 $\downarrow$ $S_p = 120$ kpsi ; table 8-2 $A_t = 0,1419 \text{ in}^2$

$$F_i = 0,75 * 120 * 0,1419 = 12,77 \text{ kip}$$

$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{120 * 0,1419}{0,2 * \left(\frac{80}{6}\right) + 12,77} = 1,1$$

$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{120 * 0,1419 - 12,77}{0,2 * \left(\frac{80}{6}\right)} = 1,6$$

$$\boxed{n_o = \frac{F_i}{P(1-C)}} = \frac{12,77}{\left(\frac{80}{6}\right)(1-0,2)} = 1,197$$

Problem 4

For the bolted assembly of Prob. 3, it is desired to find the range of torque that a mechanic could apply to initially preload the bolts without expecting failure once the joint is loaded. Assume a torque coefficient of $K = 0.2$.

(a) Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts.

(b) Determine the minimum bolt preload that can be applied while avoiding joint separation.

(c) Determine the value of torque in units of lbf · ft that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b).

a) $F_{i_{Max}} = F_p = S_p * A_t = 120 * 0,1419 = 17 \text{ Kip}$

b) $F_{i_{Min}} = P(1-C) = \left(\frac{80}{6}\right)(1-0,2) = 10,66 \text{ Kip}$

c) $F_{Mean} = \frac{17+10,66}{2} = 13,8 \text{ Kip}$

$T = K F_i d = 0,2 * 13,8 * 0,5 = 1,38 \text{ Kip.in}$

Problem 5

For a bolted assembly with eight bolts, the stiffness of each bolt is k_b =1.0 MN/mm and the stiffness of the members is k_m = 2.6 MN/mm per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. Assume the external load is equally distributed to all the bolts. It has been determined to use M6 ×1 class 5.8 bolts with rolled threads.

(a) Determine the maximum external load P_{max} that can be applied to the entire joint without exceeding the proof strength of the bolts.

(b) Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

$K_b = 1 \frac{MN}{mm}$

$K_m = 2{,}6 \frac{MN}{mm}$

$N = 8$

table 8.1 $\longrightarrow$ $A_t = 20{,}1\,mm^2$

$\sim$ 8.11 $\longrightarrow$ $S_p = 380\,MPa$

$C = \frac{k_b}{k_b + k_m} = \frac{1}{1+2{,}6} = 0{,}278$

$F_i = 0{,}75\, S_p A_t$

a) $n_p = \frac{S_p A_t}{C p + F_i} = 1$

$P = \frac{S_p A_t - F_i}{C} = \frac{380 * 20{,}1 - 0{,}75 * 380 * 20{,}1}{0{,}278}$ $6868{,}7\,N$

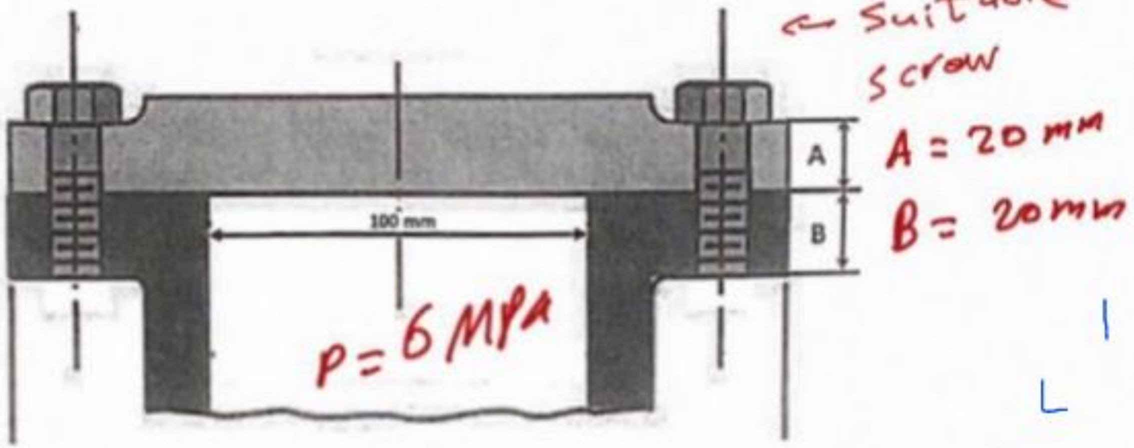
$P_{total} = 8 * 6868{,}7 = \boxed{54949{,}6\,N}$

b) $n_o = \frac{F_i}{P(1-C)} = 1$

$P = \frac{F_i}{1-C} = \frac{0{,}75 * 380 * 20{,}1}{1 - 0{,}278} = 7934{,}2\,N$

$P_{total} = 8 * 7934{,}2 = \boxed{63473{,}6\,N}$

The figure illustrates the connection of a cylinder head to a pressure vessel using 10 screws that are threaded into tapped holes in the grade 30 cast-iron cylinder. The effective sealing diameter is 100 mm. Other dimensions are: A = 20mm, B = 20mm. The cylinder is used to store gas at a static pressure of 6 MPa. ISO class 9.8 bolts with a diameter of 12 mm have been selected.



Q1.1 Calculate and select a suitable screw length from the preferred sizes in Table A–17 (neglect the washer's thickness).

Q1.2 Calculate the stiffness constants K_b and K_m (use E = 207 GPa for screw and E=100 GPa for cast-iron)

Q1.3. Determine the yielding factor, the load factor, and the joint separation factor for this selection (assume non-permanent connection), conclude.

$* \ L > h + 1.5d$

$L > 20 + 1.5 * 12 \qquad L > 38$

table A-17 → $L = 40mm$

$* \ l = h + \frac{d}{2} = 20 + \frac{12}{2} = 26mm$

$L_T = 2d + 6 = 2*12 + 6 = 30mm$

$l_d = L - L_T = 40 - 30 = 10mm$

$l_t = l - l_d = 26 - 10 = 16 \ mm$

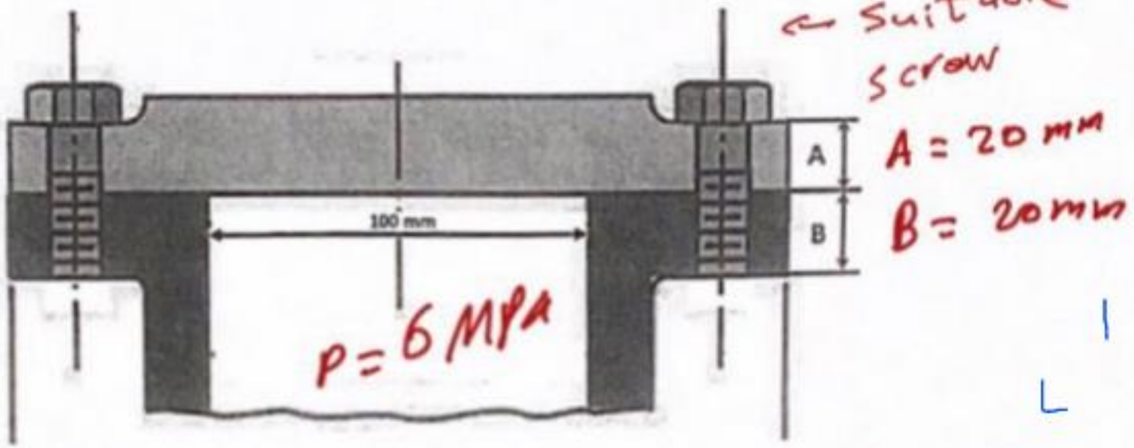
$A_d = \frac{\pi}{4}(12)^2 = 113.1 \ mm^2$

table 8-1 → $A_t = 84.3 \ mm^2$

$$K_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{113.1 * 84.3 * 207*10^3}{113.1 * 16 + 84.3 * 10} = 744027.1 \frac{N}{mm}$$

$$= 0.744 \frac{MN}{mm}$$

The figure illustrates the connection of a cylinder head to a pressure vessel using 10 screws that are threaded into tapped holes in the grade 30 cast-iron cylinder. The effective sealing diameter is 100 mm. Other dimensions are: A = 20mm, B = 20mm. The cylinder is used to store gas at a static pressure of 6 MPa. ISO class 9.8 bolts with a diameter of 12 mm have been selected.



Q1.1 Calculate and select a suitable screw length from the preferred sizes in Table A–17 (neglect the washer's thickness).

Q1.2 Calculate the stiffness constants K_b and K_m (use E = 207 GPa for screw and E=100 GPa for cast-iron)

Q1.3. Determine the yielding factor, the load factor, and the joint separation factor for this selection (assume non-permanent connection), conclude.

$$\boxed{k_m = \frac{0.5774\,\pi\,E\,d}{2\,ln\left(5\,\frac{0.5774\,l + 0.5\,d}{0.5774\,l + 2.5\,d}\right)}}$$

$$K_m = \frac{0{,}5774\,\pi * 100*10^3 * 12}{2\,ln\left(5\,\frac{0{,}5774*26+0{,}5*12}{0{,}5774*26+2{,}5*12}\right)}$$

$$K_m = 1283394{,}72\ \frac{N}{mm}$$

$$= 1{,}28\ \frac{MN}{mm}$$

$$C = \frac{K_b}{K_b+K_m} = \frac{0{,}744}{0{,}744+1{,}28} = 0{,}3675$$

table 8-11 $\rightarrow$ $S_p = 650\ MPa$, $F_i = 0{,}75 * 650 * 84{,}3 = 41096{,}25\ N$

$P_{total} = \frac{\pi}{4}(100)^2 * 6 = 47123{,}88\ N$, $P_{bolt} = \frac{47123{,}88}{10} = 4712{,}3\ N$

$$n_p = \frac{S_p A_t}{CP + F_i}$$

$$n_p = \frac{650 * 84{,}3}{0{,}36 * 4712{,}3 + 41096{,}25} = \boxed{1{,}28}$$

$$n_L = \frac{S_p A_t - F_i}{CP}$$

$$n_L = \frac{650 * 84{,}3 - 41096{,}25}{0{,}36 * 4712{,}3} = \boxed{8{,}07}$$

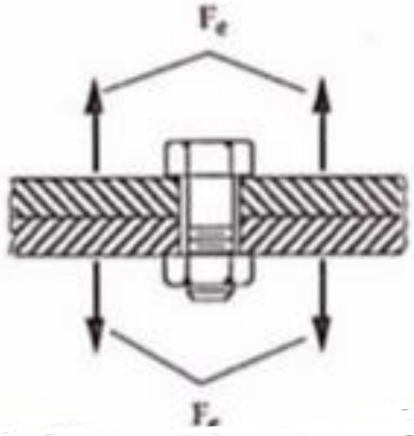
$$\boxed{n_o = \frac{F_i}{P(1 - C)}}$$

$$n_o = \frac{41096{,}25}{4712{,}3(1 - 0{,}36)} = \boxed{13{,}62}$$

Question 4 (2 pts)

The bolt is used to joint two parts of a machine as shown below. It has initially tightened to provide an initial clamping force of 2000 lb. Determine the external separating force that would cause the clamping force to be reduced to 500 lb. Suppose that $k_m = 6k_b$

$$C = \frac{k_b}{k_m + k_b} = \frac{k_b}{7k_b} = 0{,}1428$$



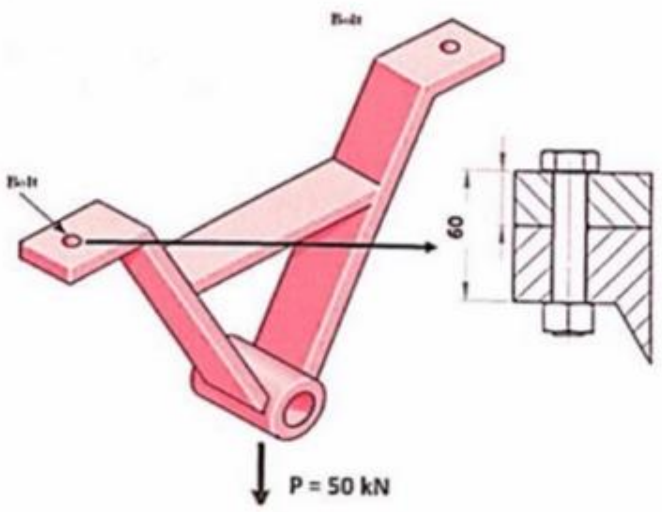
$$F_{total\ member} = -F_i + (1-C)P = -500$$

$$-2000 + (1-0{,}1428)P = -500$$

$$\boxed{P = 1750\ \text{lb}}$$

Question 3 (17 PTS)

The upside-down steel frame shown in the figure below is to be bolted to steel beams on the ceiling of a machine room using ISO grade 5.8 bolts. This frame is to support 50 kN radial load. The grip length is 60 mm. The bolts are size M10 × 1.5 coarse pitch. (use E = 207 Gpa for steel)



Q3.1. What tightening torque should be used if the connection is permanent? (3 pts)

table 8.11 → $S_p = 380\,MPa$

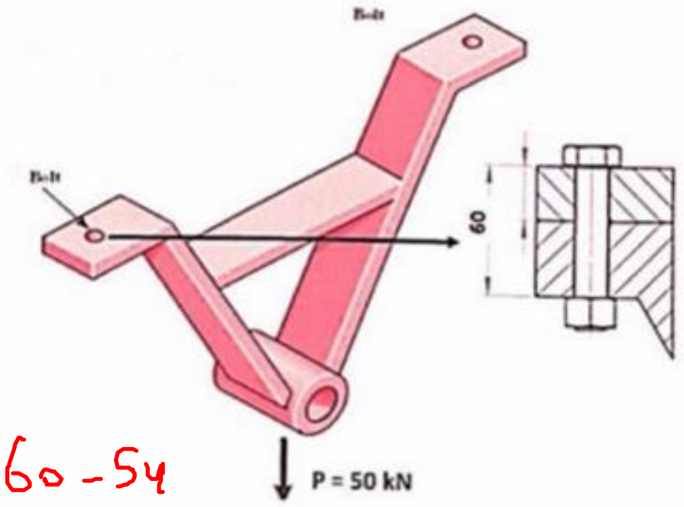
table 8.1 → $A_t = 58\,mm^2$

$F_i = 0,75\, S_p A_t = 16530\,N$

$T = K F_i d = 0,2 * 16530 * 10 = \boxed{33060\,N}$

Question 3 (17 PTS)

The upside-down steel frame shown in the figure below is to be bolted to steel beams on the ceiling of a machine room using ISO grade 5.8 bolts. This frame is to support 50 kN radial load. The grip length is 60 mm. The bolts are size M10 × 1.5 coarse pitch. (use E = 207 Gpa for steel)



Q3.2. Calculate the bolt stiffness (round the total length to next standard) (4 pts)

Table A-31 → Nut thickness $(H) = 8.4\,mm$

$l_p = 60\,mm$ (grib length)

$L > l_p + H \qquad L > 68.4\,mm$

Table A-17 → $L = 80\,mm$ Fastener length

$L_T = 2d + 6 = 2 \times 10 + 6 = 26\,mm$ threaded length

$l_d = L - L_T = 80 - 26 = 54\,mm$

$l_t = l_p - l_d = 60 - 54 = 6\,mm$

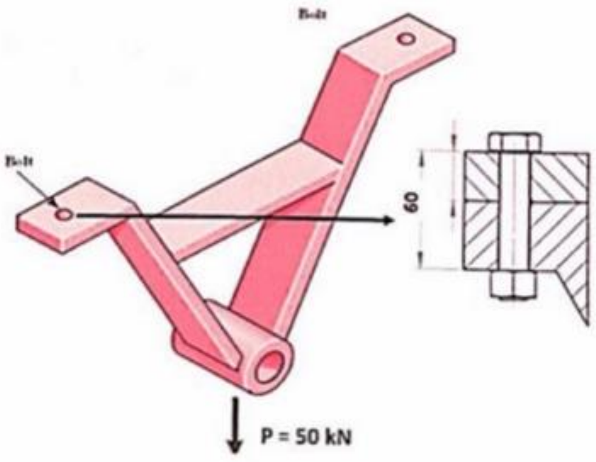
$A_d = \frac{\pi}{4}(10)^2 = 78.5\,mm^2$

$A_t = 58\,mm^2 \rightarrow$ (table 8-1)

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{78.5 \times 58 \times 207 \times 10^3}{78.5 \times 6 + 58 \times 54} = 261579.5\,\frac{N}{mm} = 0.261\,\frac{MN}{mm}$$

Question 3 (17 PTS)

The upside-down steel frame shown in the figure below is to be bolted to steel beams on the ceiling of a machine room using ISO grade 5.8 bolts. This frame is to support 50 kN radial load. The grip length is 60 mm. The bolts are size M10 × 1.5 coarse pitch. (use E = 207 Gpa for steel)



Q3.3. Calculate the stiffness of the joint (3 pts)

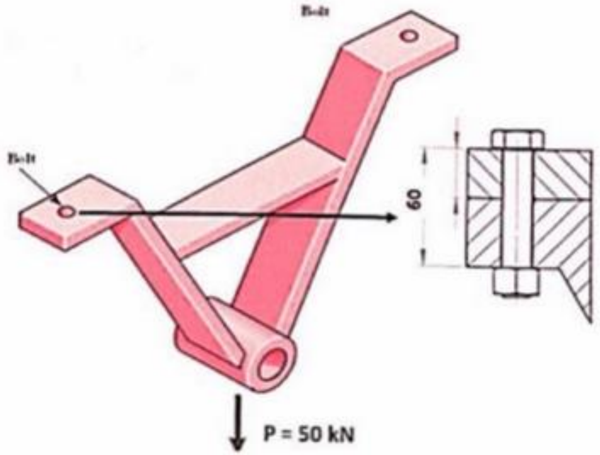
$$\boxed{k_m = \frac{0.5774\,\pi\,E\,d}{2\,\ln\left(5\,\dfrac{0.5774\,l + 0.5\,d}{0.5774\,l + 2.5\,d}\right)}}$$

$$K_m = \frac{0.5774\,\pi * 207 * 10^3 * 10}{2\ln\left(5\,\dfrac{0.5774 * 60 + 0.5 * 10}{0.5774 * 60 + 2.5 * 10}\right)} = 1562462\ \frac{N}{mm}$$

$$= 1.56\ \frac{MN}{mm}$$

Question 3 (17 PTS)

The upside-down steel frame shown in the figure below is to be bolted to steel beams on the ceiling of a machine room using ISO grade 5.8 bolts. This frame is to support 50 kN radial load. The grip length is 60 mm. The bolts are size M10 × 1.5 coarse pitch. (use E = 207 Gpa for steel)



Q3.4. Determine the factors of safety guarding against yielding, overload, and joint separation. (4.5 pts)

$$F_i = 0{,}75 S_p A_t = 0{,}75 * 380 * 58$$
$$= 16530 \text{ N}$$

$$C = \frac{k_b}{k_b + k_m} = \frac{0{,}261}{0{,}261 + 1{,}56} = 0{,}143$$

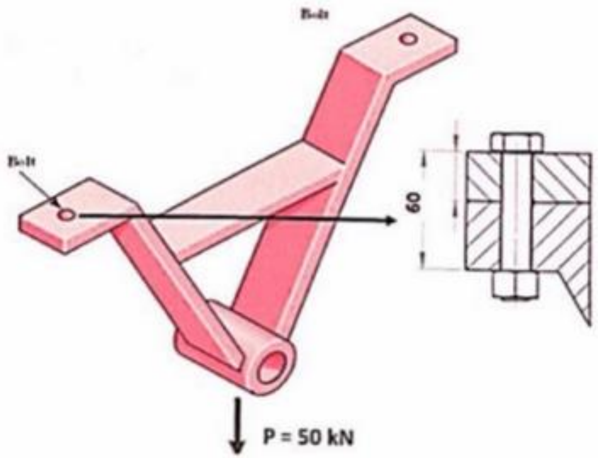
$$n_p = \frac{S_p A_t}{CP + F_i}$$
$$= \frac{380 * 58}{0{,}143 * \frac{50*10^3}{2} + 16530}$$
$$= 1{,}09$$

$$n_L = \frac{S_p A_t - F_i}{CP}$$
$$= \frac{380 * 58 - 16530}{0{,}143 * \frac{50*10^3}{2}}$$
$$= 1{,}54$$

$$\boxed{n_o = \frac{F_i}{P(1 - C)}}$$
$$= \frac{16530}{\frac{50*10^3}{2}(1 - 0{,}143)}$$
$$= 0{,}77$$

Question 3 (17 PTS)

The upside-down steel frame shown in the figure below is to be bolted to steel beams on the ceiling of a machine room using ISO grade 5.8 bolts. This frame is to support 50 kN radial load. The grip length is 60 mm. The bolts are size M10 × 1.5 coarse pitch. (use E = 207 Gpa for steel)



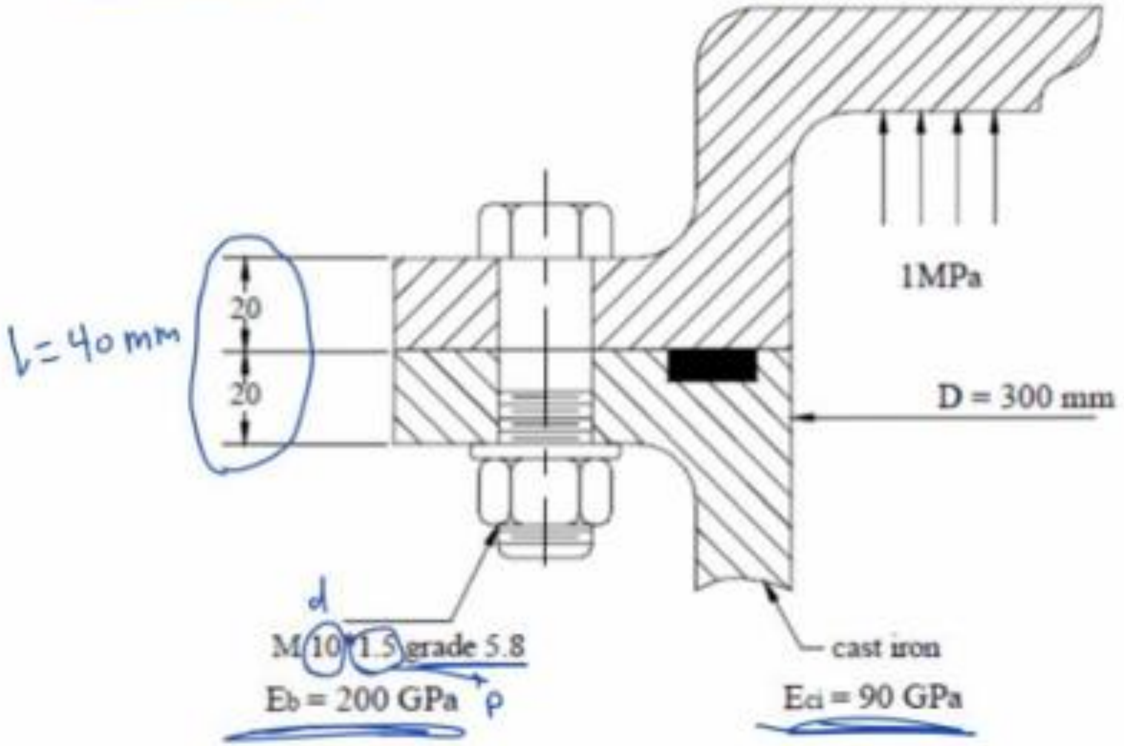
Q3.4. Conclude on the selected bolt and propose any modifications (2.5 pts)

Selection of this bolt is not safe

as $n_o < 1$

The head of a 300 mm diameter cylinder vessel will be designed. The internal pressure in the cylinder is Ptot = 1 MPa. The bolts have initial tightening load of 12 kN. The used washer is of 2.8 mm thickness.

F_i



Q1.1. Find the stiffness of the bolt (3 pts).

Q1.2 Find the stiffness of the members (2.5pts).

Q1.3. Determine the number of bolts (N) for a load safety factor of 2 (3 pts).

Q1.4. With the found N, check the separation (1.5 pts)

(The required data could be found in the tables below)

Table A-17 $\longrightarrow$ Nut thickness (H) = 8,4 mm

L = 40 + 8,4 + 2,8 + 3 = 54,2 mm

Fastener length

l_p = 40 + 2,8 = 42,8 mm

grib length

$L_T = 2d + 6 = 2 * 10 + 6 = 26$ mm

$l_d = L - L_T = 54,2 - 26 = 28,2$ mm

$l_t = l_p - l_d = 42,8 - 28,2 = 14,6$ mm

$A_d = \frac{\pi}{4}(10)^2 = 78,53$ mm²

$A_t \xrightarrow[8-1]{\text{table}} 58$ mm²

$$K_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = 327\,427 \frac{N}{mm}$$

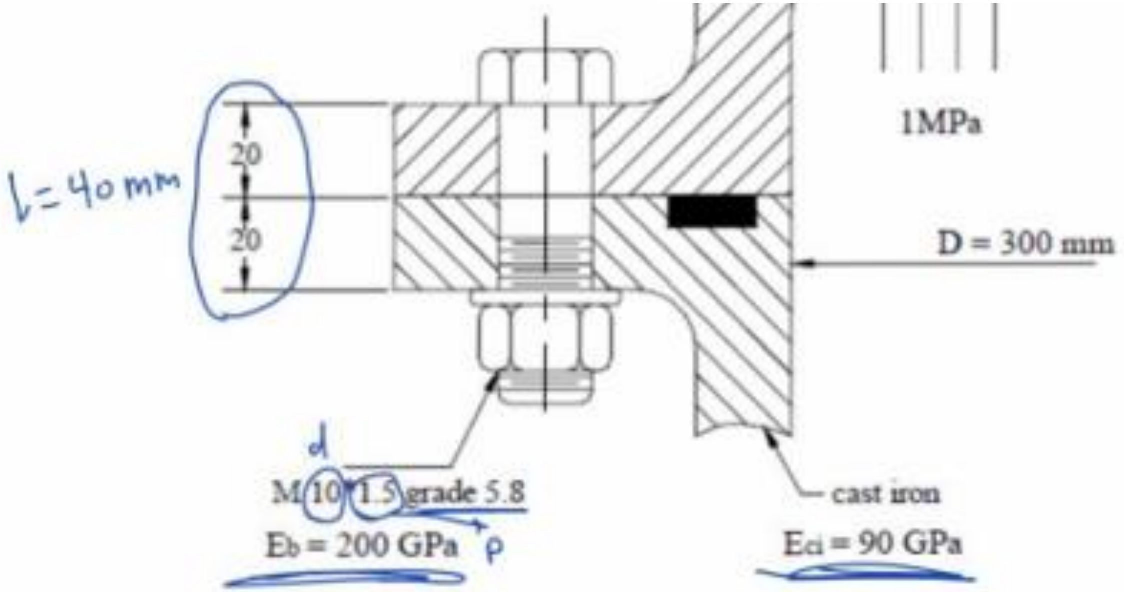
$$= 0,327 \frac{MN}{mm}$$

The head of a 300 mm diameter cylinder vessel will be designed. The internal pressure in the cylinder is Ptot = 1 MPa. The bolts have initial tightening load of 12 kN. The used washer is of 2.8 mm thickness.

$$\boxed{n_o = \frac{F_i}{P(1-C)}}$$

$$n_L = \frac{S_p A_t - F_i}{CP}$$

$$k_m = \frac{0.5774\,\pi E d}{2\ln\left(5\,\frac{0.5774\,l + 0.5\,d}{0.5774\,l + 2.5\,d}\right)}$$



$$K_M = \frac{0{,}5774\ \pi * 90*10^3 * 10}{2\ln\left(5 * \frac{0{,}5774*42{,}8+0{,}5*10}{0{,}5774*42{,}8+2{,}5*10}\right)} = 745629\ \frac{N}{mm} = 0{,}749\ \frac{MN}{mm}$$

$$C = \frac{0{,}327}{0{,}327+0{,}745} = 0{,}3$$

table 8.11 $\longrightarrow$ $S_p = 380\ MPa$, $f_i = 12*10^3 N$

$$P_{total} = \frac{\pi}{4}(300)^2 * 1 = 70685{,}8\ N$$

Q1.1. Find the stiffness of the bolt (3 pts).

Q1.2 Find the stiffness of the members (2.5pts).

Q1.3. Determine the number of bolts (N) for a load safety factor of 2 (3 pts).

Q1.4. With the found N, check the separation (1.5 pts)

(The required data could be found in the tables below)

$$n_L = \frac{S_p A_t - f_i}{C\ \frac{P_{total}}{N}}$$

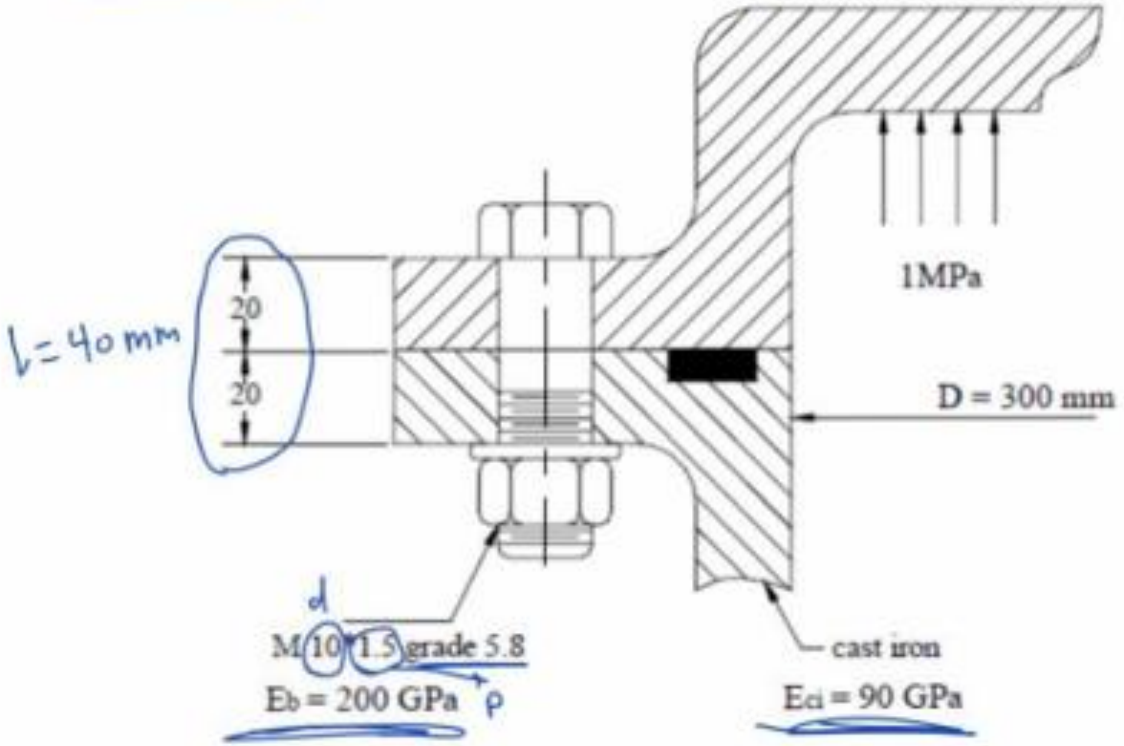
$$2 = \frac{380*58 - 12000}{0{,}3 * \frac{70685{,}8}{N}}$$

$N = 4{,}22$

$\boxed{N = 5\ \text{bolts}}$

The head of a 300 mm diameter cylinder vessel will be designed. The internal pressure in the cylinder is Ptot = 1 MPa. The bolts have initial tightening load of 12 kN. The used washer is of 2.8 mm thickness.

F_i



$$n_o = \frac{F_i}{P(1-C)}$$

$$n_o = \frac{F_i}{P(1-C)} = \frac{12*10^3}{\frac{70685,8}{5}(1-0,3)}$$

$$= \boxed{1,21}$$ safe

Q1.1. Find the stiffness of the bolt (3 pts).

Q1.2 Find the stiffness of the members (2.5pts).

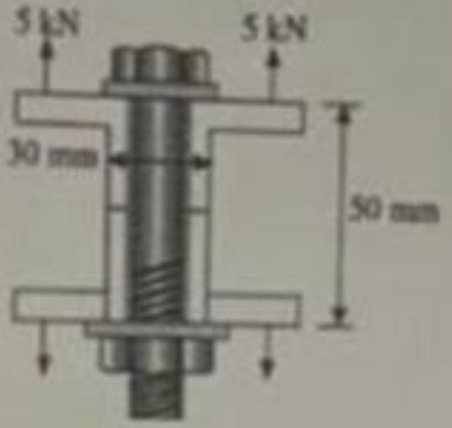
Q1.3. Determine the number of bolts (N) for a load safety factor of 2 (3 pts).

Q1.4. With the found N, check the separation (1.5 pts)

(The required data could be found in the tables below)

For the joint shown in figure below, and with the following assumptions:

- Both clamped parts are steel having E = 207 Gpa
- A steel bolt M16-type 4.8 class (coarse) is used
- External load applied to the bolt p = 10 KN



Determine:

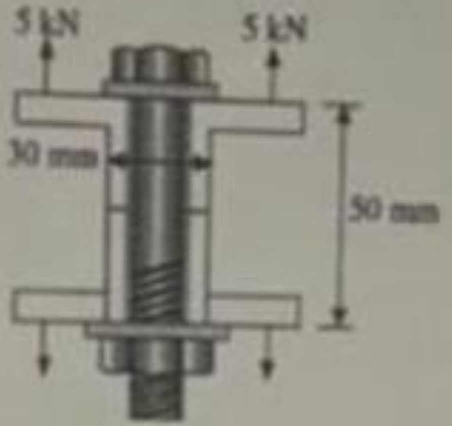
Q.1. The suitable bolt total length (neglect the washer's thickness), rounded to (1.5 pts).

$L_{\text{grib length}} = 50\,mm$, table A-17 $\rightarrow H = 14{,}8\,mm$

$L_{\text{fastener length}} = 50 + 14{,}8 + 2 * 2 = 68{,}8\,mm$

For the joint shown in figure below, and with the following assumptions:

- Both clamped parts are steel having E = 207 Gpa
- A steel bolt M16-type 4.8 class (coarse) is used
- External load applied to the bolt p = 10 KN



Q.2. The stiffness of the bolt (2.5 pts).

$$L_T = 2d + 6 = 2*16 + 6 = 38\ mm$$

$$L_d = L - L_T = 68{,}8 - 38 = 30{,}8\ mm$$

$$l_t = l - l_d = 50 - 30{,}8 = 19{,}2\ mm$$

$$A_d = \frac{\pi}{4}(16)^2 = 201{,}06\ mm^2$$

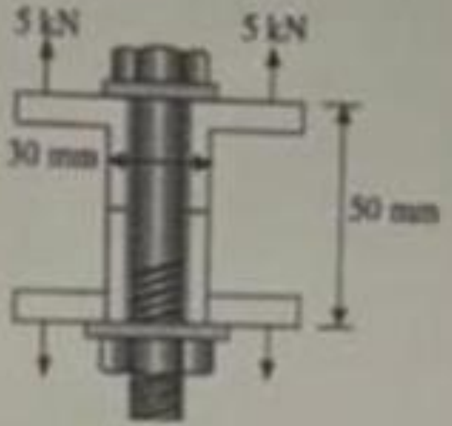
$$A_t \xrightarrow[8-1]{table} 157\ mm^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{201{,}06 * 157 * 207 * 10^3}{201{,}06 * 19{,}2 + 157 * 30{,}8} = 751412{,}72\ N/mm$$

$$0{,}751\ MN/mm$$

For the joint shown in figure below, and with the following assumptions:

- Both clamped parts are steel having E = 207 Gpa
- A steel bolt M16-type 4.8 class (coarse) is used
- External load applied to the bolt p = 10 KN



Q.3 The stiffness of the clamped material (1.5 pts)

$$\boxed{k_m = \frac{0.5774\,\pi\,E\,d}{2\,ln\left(5\,\dfrac{0.5774\,l + 0.5\,d}{0.5774\,l + 2.5\,d}\right)}}$$

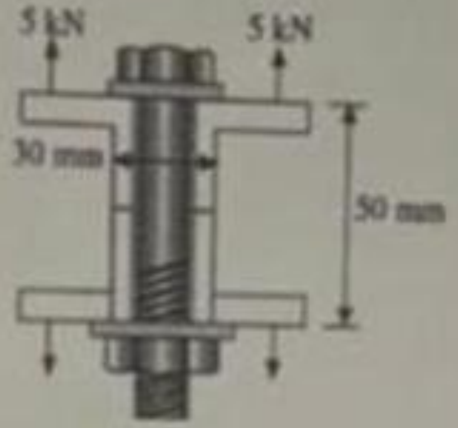
$$K_M = \frac{0{,}5774\,\pi * 207*10^3 * 16}{2\,ln\left(5 * \dfrac{0{,}5774*50 + 0{,}5*16}{0{,}5774*50 + 2{,}5*16}\right)}$$

$$= 3049299{,}7\ \frac{N}{MM}$$

$$= \boxed{3{,}04\ \frac{MN}{MM}}$$

For the joint shown in figure below, and with the following assumptions:

- Both clamped parts are steel having E = 207 Gpa
- A steel bolt M16-type 4.8 class (coarse) is used
- External load applied to the bolt p = 10 KN



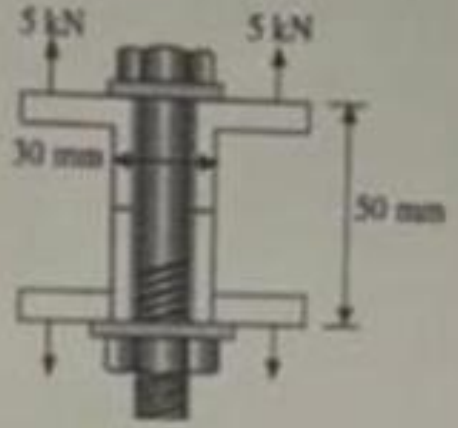
Q.4. The suitable preload force, F_i if the joint is subject to occasional disassembly (2 pts).

$S_p \xrightarrow[8.11]{table} 310\ MPa$, $A_t = 157\ mm^2$

$F_i = 0,75\, S_p A_t = 0,75 * 310 * 157 = 36502,5\ N$

For the joint shown in figure below, and with the following assumptions:

- Both clamped parts are steel having E = 207 Gpa
- A steel bolt M16-type 4.8 class (coarse) is used
- External load applied to the bolt p = 10 KN



Q.5. The factor of safety against yielding and separation (2.5 pts).

$$C = \frac{k_b}{k_b + k_M} = \frac{0,791}{0,791 + 3,04} = 0,198$$

$$\boxed{n_o = \frac{F_i}{P(1-C)}}$$

$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{310 * 157}{0,198 * 10 * 10^3 + 36502,5} = 1,26$$

$$n_o = \frac{36502,5}{10 * 10^3 (1 - 0,198)} = 4,55$$

Table 8–1

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

Nominal Major Diameter d mm	Coarse-Pitch Series			Fine-Pitch Series		
	Pitch p mm	Tensile-Stress Area A_t mm^2	Minor-Diameter Area A_r mm^2	Pitch p mm	Tensile-Stress Area A_t mm^2	Minor-Diameter Area A_r mm^2
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
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470
110				2	9180	9080

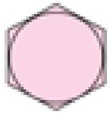
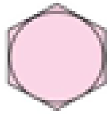
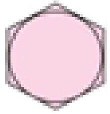
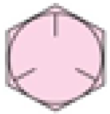
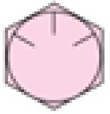
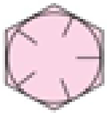
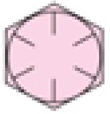
Table 8–2

Diameters and Area of Unified Screw Threads UNC and UNF*

		Coarse Series—UNC			Fine Series—UNF		
Size Designation	Nominal Major Diameter in	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2	Threads per Inch N	Tensile-Stress Area A_t in^2	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

SAE Specifications for Steel Bolts

Table 8–9

SAE Grade No.	Size Range Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
1	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low or medium carbon	
2	$\frac{1}{4}$–$\frac{3}{4}$ $\frac{7}{8}$–$1\frac{1}{2}$	55 33	74 60	57 36	Low or medium carbon	
4	$\frac{1}{4}$–$1\frac{1}{2}$	65	115	100	Medium carbon, cold-drawn	
5	$\frac{1}{4}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Medium carbon, Q&T	
5.2	$\frac{1}{4}$–1	85	120	92	Low-carbon martensite, Q&T	
7	$\frac{1}{4}$–$1\frac{1}{2}$	105	133	115	Medium-carbon alloy, Q&T	
8	$\frac{1}{4}$–$1\frac{1}{2}$	120	150	130	Medium-carbon alloy, Q&T	
8.2	$\frac{1}{4}$–1	120	150	130	Low-carbon martensite, Q&T	

*Minimum strengths are strengths exceeded by 99 percent of fasteners.

ASTM Specification for Steel Bolts

Table 8–10

ASTM Designation No.	Size Range, Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
A307	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low carbon	
A325, type 1	$\frac{1}{2}$–1	85	120	92	Medium carbon, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
A325, type 2	$\frac{1}{2}$–1	85	120	92	Low-carbon, martensite, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
A325, type 3	$\frac{1}{2}$–1	85	120	92	Weathering steel, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
A354, grade BC	$\frac{1}{4}$–$2\frac{1}{2}$	105	125	109	Alloy steel, Q&T	
	$2\frac{3}{4}$–4	95	115	99		
A354, grade BD	$\frac{1}{4}$–4	120	150	130	Alloy steel, Q&T	
A449	$\frac{1}{4}$–1	85	120	92	Medium-carbon, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
	$1\frac{3}{4}$–3	55	90	58		
A490, type 1	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Alloy steel, Q&T	
A490, type 3	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Weathering steel, Q&T	

Metric Mechanical-Property Classes for Steel Bolts

Property Class	Size Range, Inclusive	Minimum Proof Strength,† MPa	Minimum Tensile Strength,† MPa	Minimum Yield Strength,† MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	4.6
4.8	M1.6–M16	310	420	340	Low or medium carbon	4.8
5.8	M5–M24	380	520	420	Low or medium carbon	5.8
8.8	M16–M36	600	830	660	Medium carbon, Q&T	8.8
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	9.8
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	10.9
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	12.9

Table 8–11

*The thread length for bolts and cap screws is

Table A–31

Dimensions of Hexagonal Nuts

Nominal Size, in	Width W	Height H Regular Hexagonal	Thick or Slotted	JAM
$\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{32}$	$\frac{9}{32}$	$\frac{5}{32}$
$\frac{5}{16}$	$\frac{1}{2}$	$\frac{17}{64}$	$\frac{21}{64}$	$\frac{3}{16}$
$\frac{3}{8}$	$\frac{9}{16}$	$\frac{21}{64}$	$\frac{13}{32}$	$\frac{7}{32}$
$\frac{7}{16}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{29}{64}$	$\frac{1}{4}$
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{5}{16}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{31}{64}$	$\frac{39}{64}$	$\frac{5}{16}$
$\frac{5}{8}$	$\frac{15}{16}$	$\frac{35}{64}$	$\frac{23}{32}$	$\frac{3}{8}$
$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{41}{64}$	$\frac{13}{16}$	$\frac{27}{64}$
$\frac{7}{8}$	$1\frac{5}{16}$	$\frac{3}{4}$	$\frac{29}{32}$	$\frac{31}{64}$
1	$1\frac{1}{2}$	$\frac{55}{64}$	1	$\frac{35}{64}$
$1\frac{1}{8}$	$1\frac{11}{16}$	$\frac{31}{32}$	$1\frac{5}{32}$	$\frac{39}{64}$
$1\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$\frac{23}{32}$
$1\frac{3}{8}$	$2\frac{1}{16}$	$1\frac{11}{64}$	$1\frac{3}{8}$	$\frac{25}{32}$
$1\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{9}{32}$	$1\frac{1}{2}$	$\frac{27}{32}$

Nominal Size, mm	Width W	Height H Regular Hexagonal	Thick or Slotted	JAM
M5	8	4.7	5.1	2.7
M6	10	5.2	5.7	3.2
M8	13	6.8	7.5	4.0
M10	16	8.4	9.3	5.0
M12	18	10.8	12.0	6.0
M14	21	12.8	14.1	7.0
M16	24	14.8	16.4	8.0
M20	30	18.0	20.3	10.0
M24	36	21.5	23.9	12.0
M30	46	25.6	28.6	15.0
M36	55	31.0	34.7	18.0

Table A–17

Preferred Sizes and Renard (R-Series) Numbers
(When a choice can be made, use one of these sizes; however, not all parts or items are available in all the sizes shown in the table.)

Fraction of Inches

$\frac{1}{64}$, $\frac{1}{32}$, $\frac{1}{16}$, $\frac{3}{32}$, $\frac{1}{8}$, $\frac{5}{32}$, $\frac{3}{16}$, $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, $\frac{7}{16}$, $\frac{1}{2}$, $\frac{9}{16}$, $\frac{5}{8}$, $\frac{11}{16}$, $\frac{3}{4}$, $\frac{7}{8}$, 1, $1\frac{1}{4}$, $1\frac{1}{2}$, $1\frac{3}{4}$, 2, $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, 3, $3\frac{1}{4}$, $3\frac{1}{2}$, $3\frac{3}{4}$, 4, $4\frac{1}{4}$, $4\frac{1}{2}$, $4\frac{3}{4}$, 5, $5\frac{1}{4}$, $5\frac{1}{2}$, $5\frac{3}{4}$, 6, $6\frac{1}{2}$, 7, $7\frac{1}{2}$, 8, $8\frac{1}{2}$, 9, $9\frac{1}{2}$, 10, $10\frac{1}{2}$, 11, $11\frac{1}{2}$, 12, $12\frac{1}{2}$, 13, $13\frac{1}{2}$, 14, $14\frac{1}{2}$, 15, $15\frac{1}{2}$, 16, $16\frac{1}{2}$, 17, $17\frac{1}{2}$, 18, $18\frac{1}{2}$, 19, $19\frac{1}{2}$, 20

Decimal Inches

0.010, 0.012, 0.016, 0.020, 0.025, 0.032, 0.040, 0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.24, 0.30, 0.40, 0.50, 0.60, 0.80, 1.00, 1.20, 1.40, 1.60, 1.80, 2.0, 2.4, 2.6, 2.8, 3.0, 3.2, 3.4, 3.6, 3.8, 4.0, 4.2, 4.4, 4.6, 4.8, 5.0, 5.2, 5.4, 5.6, 5.8, 6.0, 7.0, 7.5, 8.5, 9.0, 9.5, 10.0, 10.5, 11.0, 11.5, 12.0, 12.5, 13.0, 13.5, 14.0, 14.5, 15.0, 15.5, 16.0, 16.5, 17.0, 17.5, 18.0, 18.5, 19.0, 19.5, 20

Millimeters

0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.25, 0.30, 0.40, 0.50, 0.60, 0.70, 0.80, 0.90, 1.0, 1.1, 1.2, 1.4, 1.5, 1.6, 1.8, 2.0, 2.2, 2.5, 2.8, 3.0, 3.5, 4.0, 4.5, 5.0, 5.5, 6.0, 6.5, 7.0, 8.0, 9.0, 10, 11, 12, 14, 16, 18, 20, 22, 25, 28, 30, 32, 35, 40, 45, 50, 60, 80, 100, 120, 140, 160, 180, 200, 250, 300

Renard Numbers*

1st choice, R5: 1, 1.6, 2.5, 4, 6.3, 10

2d choice, R10: 1.25, 2, 3.15, 5, 8

3d choice, R20: 1.12, 1.4, 1.8, 2.24, 2.8, 3.55, 4.5, 5.6, 7.1, 9

4th choice, R40: 1.06, 1.18, 1.32, 1.5, 1.7, 1.9, 2.12, 2.36, 2.65, 3, 3.35, 3.75, 4.25, 4.75, 5.3, 6, 6.7, 7.5, 8.5, 9.5

*May be multiplied or divided by powers of 10.