ME 305
Machine Design II

CH # 8

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

Department of Mechanical Engineering
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Two rectangular metal pieces, the aim is to join them

How this can be done?

How Many methods are there to join them?

Which one is the most suitable?

- Permanent
  - Revit
  - Weld
  - Seam (in Sheets)
  - Bonding (chemical)
- Non-permanent
  - Threaded fasteners ***(Screws & bolts)***
  - Keys, Pins and cotters
  - Snap
  - Shrink
  - etc.

- Function of the fastener
- Operating environment of the fastener
- Type of loading on the fastener in service
- Thickness of materials to be joined
- Type of materials to be joined
- Configuration of the joint to be fastened

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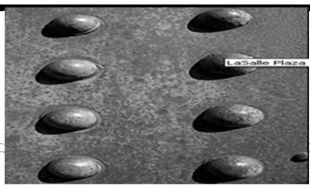
Mechanical Fasteners

Mechanical fasteners are frequently grouped as listed below:

- Keys and Pins
- *Threaded fasteners*
- Rivets
- Blind fasteners
- Adhesives
- Spring retainers
- Locking devices
- Special purpose fasteners



One of the Key Target of current designer for manufacturer is to reduce the number of fasteners

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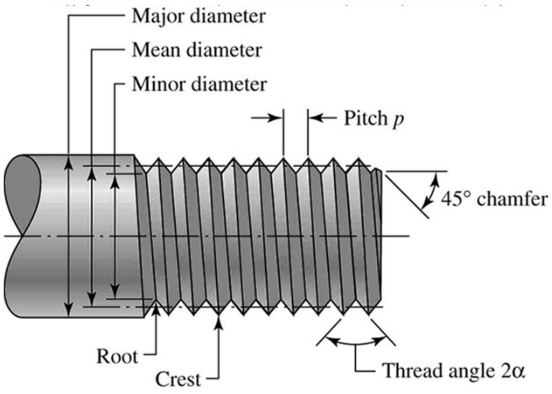
Boeing 747 requires 2.5 Millions fasteners

Each one Can cost a lot

Will add a lot to the over all cost

8-1 Thread Standards and Definitions



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Terminology of screw threads

- The *lead l* is the distance the nut moves parallel to the screw axis when the nut is given one turn. For a single thread, the lead is the same as the pitch
- A *multiple-threaded* product is one having two or more threads cut besides each other; a *double-threaded screw* had a *lead* equal to *twice the pitch*
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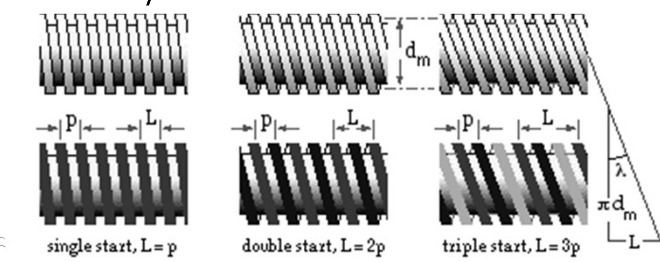

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Left and Right handed threads

Left Handed

Right Handed

L is written on the head

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Coarse Threads (UNC)

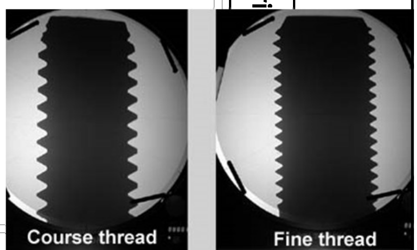
Coarse thread series UNC/UNRC. The most commonly used thread system used in the majority of screws, bolts, and nuts. It is used for producing threads in low strength materials such as cast iron, mild steel, and softer copper alloys, aluminum etc. The coarse thread is also used for rapid assembly or disassembly.

Fine Threads (UNF)

Fine thread series “UNF”. This is used for applications that require a higher tensile strength than the coarse thread series and where a thin wall is required.

Extra Fine (UNEF)

This is used when the length of engagement is smaller than the fine-thread series. It is also applicable in all applications where the fine thread can be used.



Terminology

- A bolt is available in a size as;

  M10 x 1.5-6g

- ***M*** stands for Metric, 10 is bolt nominal (major) diameter in mm, 1.5 is the pitch in mm, 6g is the tolerance class (external thread or g is capitalized if internal thread)
- ***MJ*** represents *external* thread has an *increased root radius* (shallower root relative to external M thread profile to reduce stress concentration at the root)

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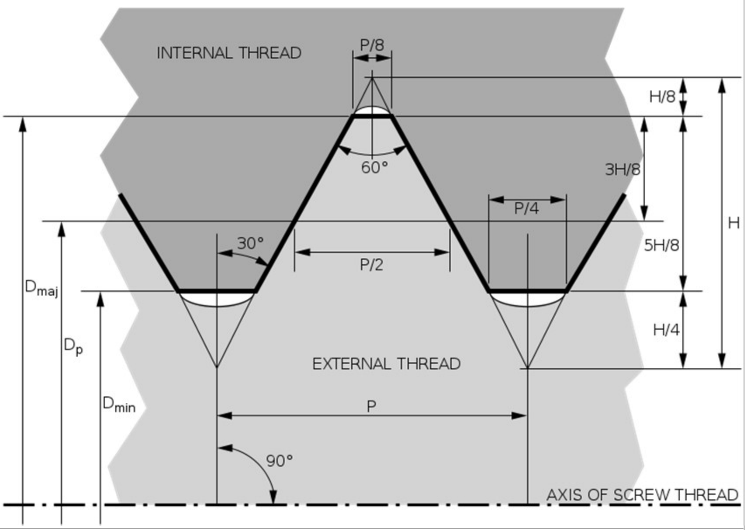
Basic thread profile for M ,MJ and Unified threads



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Diameters and areas of coarse-pitch and find pitch Metric threads

All dimensions are in mm

	Coarse-Pitch Series			Fine-Pitch Series		
Nominal Major Diameter d	Pitch p	Tensile-Stress Area A_t	Minor-Diameter Area A_r	Pitch p	Tensile-Stress Area A_t	Minor-Diameter Area A_r
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

* The equations and data used to develop this table have been obtained from ANSI B1.1-1974 and B18.3.1-1978. The minor diameter was found from the equation $d_r = d - 1.226\ 869p$, and the pitch diameter from $d_m = d - 0.649\ 519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

Diameters and area of Unified Screw Threads

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All dimensions are in inch

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

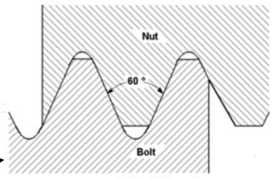
* This table was compiled from ANSI B1.1-1974. The minor diameter was found from the equation $d_r = d - 1.299\ 038p$, and the pitch diameter from $d_m = d - 0.649\ 519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

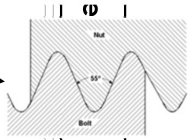
Tensile Stress Area A_t

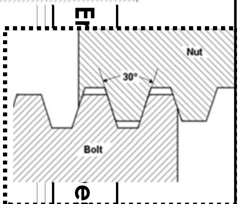
- A great many tensile tests of threaded rods have shown that an unthreaded rod having a diameter equal to the mean of the pitch and minor diameter will have the same tensile strength as the threaded rod
- The area of this unthreaded rod is called the tensile-stress area A_t

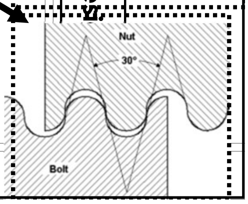
Most common Type of threads

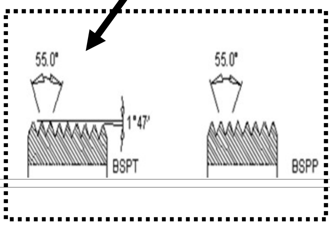
The most common threads are:
- ISO Thread (metric)
- Whitworth Thread
- Trapezoidal Thread
- Knuckle Thread
- Buttress Thread
- Pipe threads

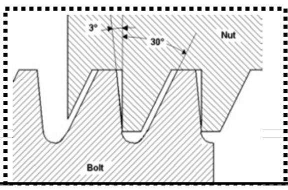








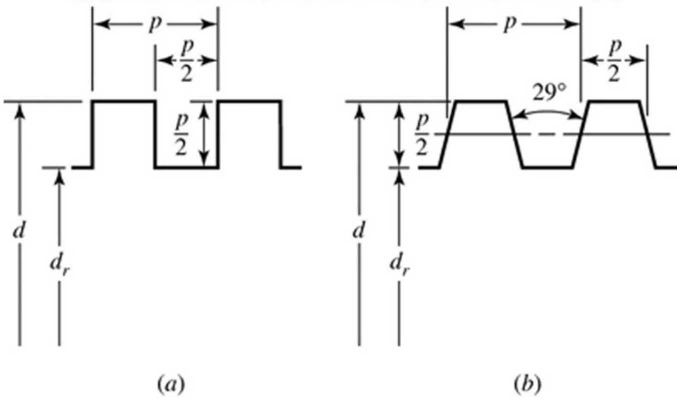




Threads for power transmission
(Power Screws)

Power Screws

- Square (a) and Acme (b) threads are used on screws when power is to be transmitted



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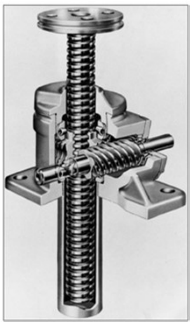
Power Screws

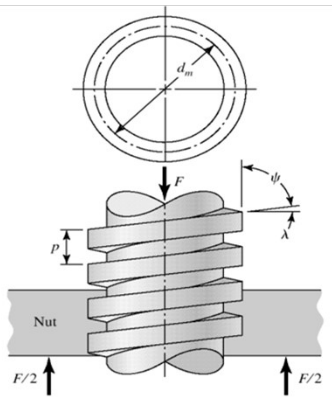
Power screws are used to convert rotary motion to linear motion of the meshing member along the screw axis. These screws are used to lift weights



Lathes, Screw Jack etc

For application that require power transmission, the Acme and square threads are used.

8-2 Mechanics of power screw



Mean diameter, d_m

Pitch, p

Lead, l

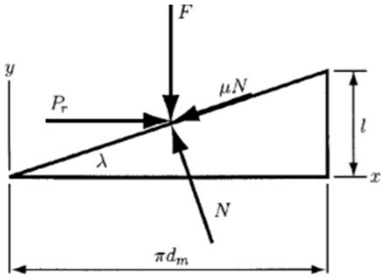
Lead angle, λ

Helix angle, ψ

Loaded by axial compressive force F

The screw is loaded by an axial compressive force F (Figs. 1.7 and 1.8). The force diagram for lifting the load is shown in Fig. 1.8a (the force P_r acts to the right). The force diagram for lowering the load is shown in Fig. 1.8b (the force P_l acts to the left). The friction force is

8-2 Mechanics of power screw…



: 1.8 (a)

The equilibrium of forces for raising the load gives

$$\sum F_x = P_r - N\sin\lambda - \mu N\cos\lambda = 0$$
$$\sum F_y = F + \mu N\sin\lambda - N\cos\lambda = 0.$$

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8-2 Mechanics of power screw…

and dividing the equations by $\cos\lambda$, one may obtain

$$P_r = \frac{F[(l/\pi d_m) + \mu]}{1 - (\mu l/\tau d_m)}$$

The torque required to overcome the thread friction and to raise the load is

$$T_r = P_r\frac{d_m}{2} = \frac{Fd_m}{2}\left(\frac{l + \pi\mu d_m}{\pi d_m - \mu l}\right).$$

The torque required to lower the load (and to overcome a part of the friction) is

$$T_l = \frac{Fd_m}{2}\left(\frac{\pi\mu d_m - l}{\pi d_m + \mu l}\right).$$

μ Is the same as f

ME-305 Mechanic

Self Locking of power screws

This is the torque required to overcome a part of the friction in lowering the load. It may turn out, in specific instances where the lead is large or the friction is low, that the load will lower itself by causing the screw to spin without any external effort. In such cases, the torque T from Eq. (8–2) will be negative or zero. When a positive torque is obtained from this equation, the screw is said to be *self-locking*. Thus the condition for self-locking is

$$\pi f d_m > l$$

Now divide both sides of this inequality by πd_m. Recognizing that $l/\pi d_m = \tan\lambda$, we get

$$f > \tan\lambda \tag{8–3}$$

This relation states that self-locking is obtained whenever the coefficient of thread friction is equal to or greater than the tangent of the thread lead angle.

Efficiency of Power Screw

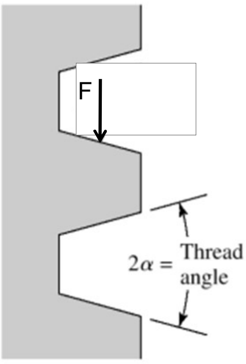
An expression for efficiency is also useful in the evaluation of power screws. If we let $f = 0$ in Eq. (8–1), we obtain

$$T_0 = \frac{Fl}{2\pi} \tag{g}$$

which, since thread friction has been eliminated, is the torque required only to raise the load. The efficiency is therefore

$$e = \frac{T_0}{T} = \frac{Fl}{2\pi T} \tag{8–4}$$

For ACME Threads

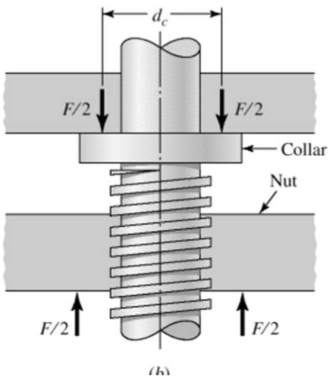


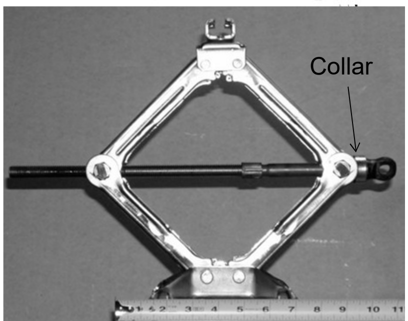
$$T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m \sec\alpha}{\pi d_m - f l \sec\alpha}\right)$$

remember that it is an approximation because the effect of the lead angle has been neglected.

Effect of Collar Loads

Usually a third component of torque must be applied in power-screw applications. When the screw is loaded axially, a thrust or collar bearing must be employed between the rotating and stationary members in order to carry the axial component.





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

Stresses in Power Screws

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- Maximum Nominal Shear Stress in the body

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

- The Axial Stress in the body

$$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2}$$

- For Buckling of the screw, use J. B. Johnson formula

$$\left(\frac{F}{A}\right)_{\text{crit}} = S_y - \left(\frac{S_y}{2\pi}\frac{l}{k}\right)^2 \frac{1}{CE}$$

- The Bearing Stress

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

Stresses in Power Screws…

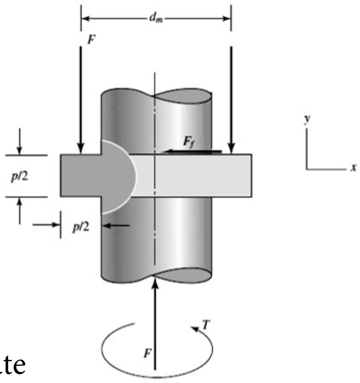
ME-3

- Bending Stress at the thread root

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

- Transverse Shear at the center of the thread root

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



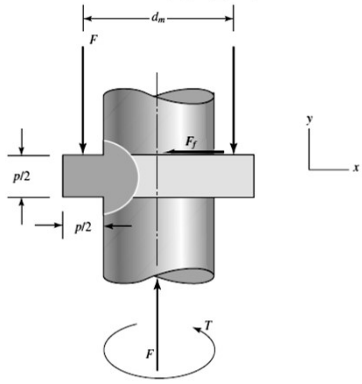
- The von-Mises stress for tri-axial stress state

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

sign II

Stresses in Power Screws…

Where
- σ_x = bending stress
- σ_y = axial
- $\sigma_z = 0$
- $\tau_{xy} = 0$
- τ_{yz} = Torsional stress
- $\tau_{xz} = 0$



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.

Example 8.1…

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

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

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

Answer

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

Example 8.1…

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

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

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

Answer

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

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

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

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

Answer

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

The axial nominal normal stress σ is

Answer

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

Example 8.1…

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

Example 8.1…

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

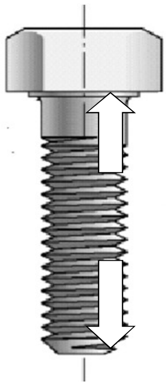
Answer
$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} = \frac{41.5-(-13.18)}{2} = 27.3 \text{ MPa}$$

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

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

Ordering the principal stresses gives $\sigma_1, \sigma_2, \sigma_3 = 41.5, 2.79, -13.18$ MPa.

8-3 Threaded fasteners



Purpose of the bolt is to clamp two are more parts together

Clamping Force/Preload

The clamping load stretches or elongates the bolt; the load is obtained by twisting the nut until the bolt has elongated almost to the elastic limit. If the nut does not loosen, this bolt tension remains as the preload or clamping force.

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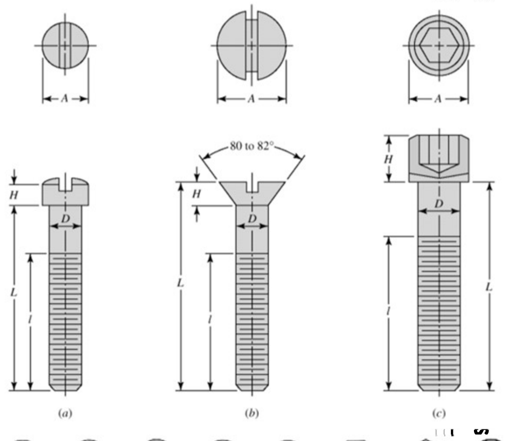
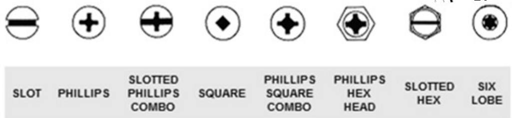
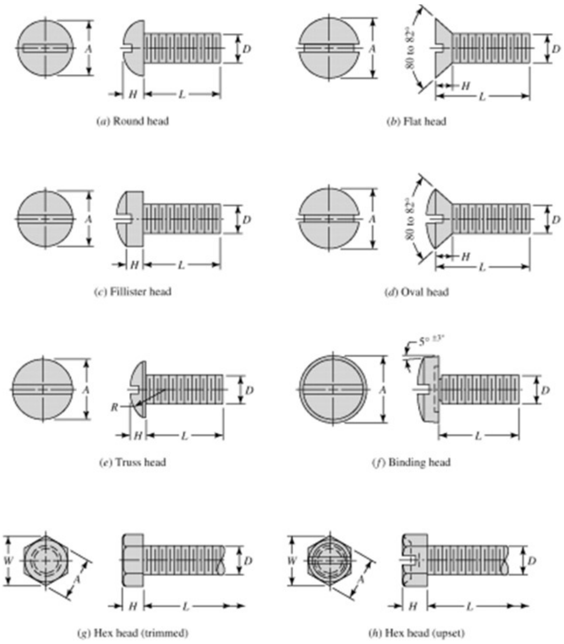
Typical cap-screw heads: (*a*) fillister head; (*b*) flat head; (*c*) hexagonal socket head. Cap screws are also manufactured with hexagonal heads similar to the one shown in Fig. 8–9, as well as a variety of other head styles. This illustration uses one of the conventional methods of representing threads.



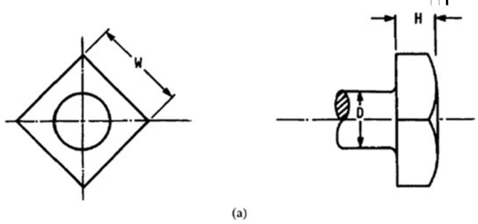




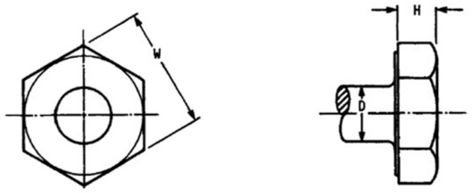
ME-305 Mechanical Engineering Design II

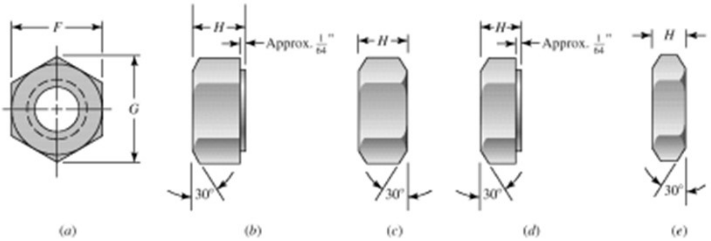
Bolts Head

Square Bolt Head



Hexagonal Bolt Head





Thread length (L_T)

- Inch system (D is the nominal major diameter)

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

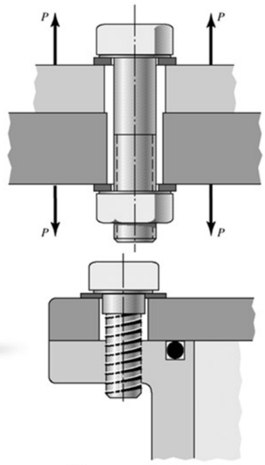
- Metric system $D \le 48$

$$L_T = \begin{cases} 2D + 6 & L \le 125 \\ 2D + 12 & 125 < L \le 200 \\ 2D + 25 & L > 200 \end{cases}$$

- The ideal bolt is one in which only one or two threads project from the nut
- Washer must always be used to avoid stress concentration.

8-4 Joints-Fastener stiffness

- When a non-permanent connection is required, bolted joint is the best choice
- Bolt pre-tension/pre-load when the nut is properly tightened
- Tension in the bolt and compression in the members
- Studs are also used



8-4 Joints-Fastener stiffness…

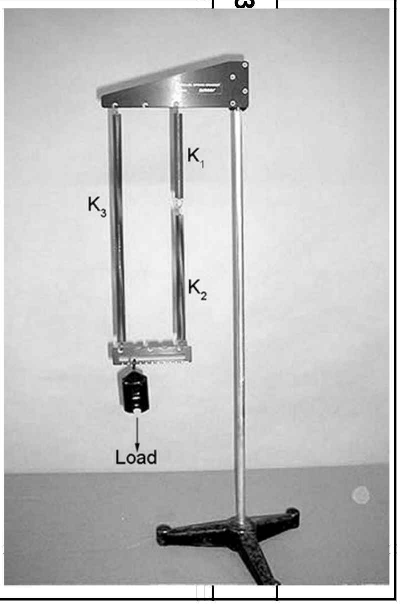
ME-305 Mechanical Engineering Design II

- Bolted joints should
  - Work without destruction
  - Resist external Tensile loads
  - Moment loads
  - Shear loads
  - Combination
- The bolt stiffness can be calculated by considering it to be fully elastic
- Stiffness?? $k = F/y$
- The stiffness/spring rate of a bolt "k" can be determined using the approach of "springs" connected in series or parallel.

8-4 Joints-Fastener stiffness…

ME-3

- Spring rate for Springs in series
$$\frac{1}{k_s} = \frac{1}{k_1} + \frac{1}{k_2}$$
- Overall stiffness "k" of the given figure
$$k = k_3 + k_s$$



8-4 Joints-Fastener stiffness…

- The bolt consists tow portions, threaded and unthreaded
- Both the portions are considered connected in series , then

$$k_b = \frac{k_t k_d}{k_t + k_d}$$

- Also the spring rate is given by (equation 4-4)

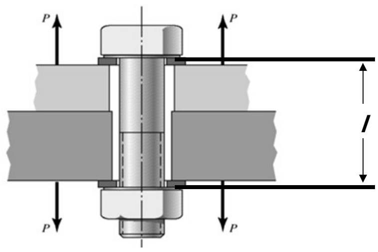
$$k = \frac{AE}{l}$$

- Then

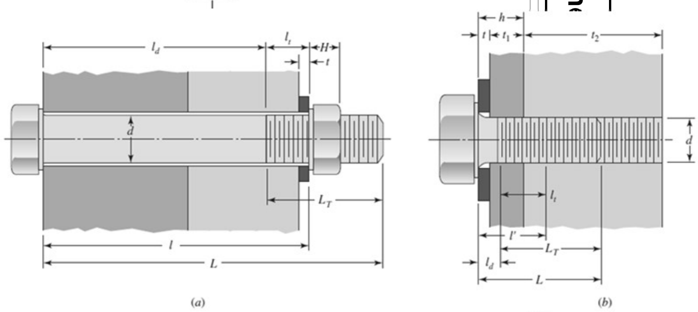
$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}$$

To determine unknowns for *k_b*

Use ***Table 8-7*** to understand different parameters and calculate bolt stiffness "k_b"



$$l_d = L - L_T$$
$$l_t = l - l_d$$



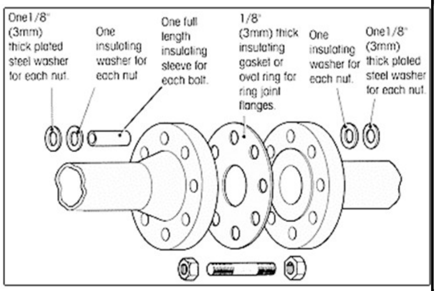
8-5 Joints-Member Stiffness

ME-3

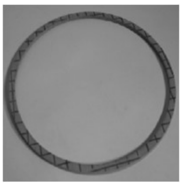
Stiffness of the members in the Clamp Zone

They act like compressive springs in Series

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \cdots + \frac{1}{k_i}$$



If one of the member is a soft gasket so its stiffness is very small to other members hence other can be neglected for all practical purposes and only gasket stiffness will be used


8-5 Joints-Member Stiffness

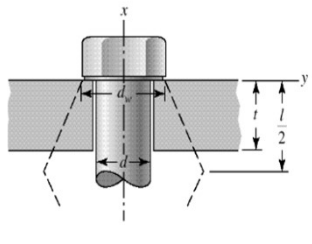
ME-305 Mechanical Engineering II

- If there is No Gasket then it is very difficult to find the stiffness except by experimentation.
- The difficulty is mainly because of the compressive spreads out between the bolt head and the Nut and hence the area is not uniform

Ito[2] has used ultrasonic techniques to determine the pressure distribution at the member interface. The results show that the pressure stays high out to about 1.5 bolt radii. The pressure, however, falls off farther away from the bolt. Thus Ito suggests the use of Rotscher's pressure-cone method for stiffness calculations with a variable cone angle. This method is quite complicated, and so here we choose to use a simpler approach using a fixed cone angle.

ME-3

Compression of a member with the equivalent elastic properties represented by a frustum of a hollow cone.



(a)

the general cone geometry using a half-apex angle α. An angle $\alpha = 45°$ has been used, but Little[3] reports that this overestimates the clamping stiffness. When loading is restricted to a washer-face annulus (hardened steel, cast iron, or aluminum), the proper apex angle is smaller.

Osgood[4] reports a range of $25° \leq \alpha \leq 33°$

According to the Text, $\alpha = 30^o$ for hardened steel, cast iron and Aluminum Members

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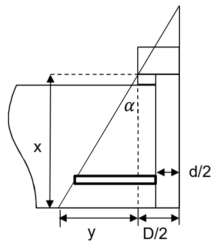
the elongation of an element of the cone of thickness dx subjected to a tensile force P is,

$$d\delta = \frac{P\,dx}{EA} \qquad \text{(a)}$$

Area of the Element



$$A = \pi\left(r_o^2 - r_i^2\right) = \pi\left[\left(x\tan\alpha + \frac{D}{2}\right)^2 - \left(\frac{d}{2}\right)^2\right]$$

Substituting this in Eq. (*a*) and integrating the left side gives the elongation as

$$\delta = \frac{P}{\pi E}\int_0^t \frac{dx}{[x\tan\alpha + (D+d)/2][x\tan\alpha + (D-d)/2]}$$

From table of integrals

$$\int \frac{1}{(ax+b)(ax+c)}dx = \frac{\ln(b+ax) - \ln(c+ax)}{a(c-b)}$$

Using a table of integrals, we find the result to be

$$\delta = \frac{P}{\pi E d\tan\alpha}\ln\frac{(2t\tan\alpha + D - d)(D+d)}{(2t\tan\alpha + D + d)(D-d)}$$

$$k_m = \frac{P}{\delta} = \frac{\pi E d \tan\alpha}{ln\dfrac{(2t\tan\alpha + D - d)(D+d)}{(2t\tan\alpha + D + d)(D-d)}}$$

ME-3

- With α =30º, equation becomes;

$$k = \frac{0.5774\pi Ed}{\ln \frac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

- With $t = l/2$ and $D = 1.5d$

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

- Choudury and Green developed a curve for Km/Ed using FEM simulation and concluded the equation;

$$\frac{k_m}{Ed} = A \exp(Bd/l)$$

Table 8–8

Stiffness Parameters of Various Member Materials

Source: J. Wileman, M. Choudury, and I. Green, "Computation of Member

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

Example 8-2

ME-305 Mec

As shown in Fig. 8–17*a*, two plates are clamped by washer-faced $\frac{1}{2}$ in-20 UNF × $1\frac{1}{2}$ in SAE grade 5 bolts each with a standard $\frac{1}{2}$ N ...el plain washer.
(*a*) Determine the member spring rate k_m if th... ...ate is steel and the bottom plate is gray cast iron.
(*b*) Using the method of conical f... plates are steel.
(*c*) Using Eq. (8–23), determine t... Compare the results with part (*b*).
(*d*) Determine the bolt spring rate

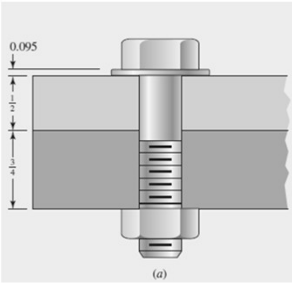


Table A–32

Basic Dimensions of American Standard Plain Washers (All Dimensions in Inches)

Fastener Size	Washer Size	Diameter ID	Diameter OD	Thickness
#6	0.138	0.156	0.375	0.049
#8	0.164	0.188	0.438	0.049
#10	0.190	0.219	0.500	0.049
#12	0.216	0.250	0.562	0.065
$\frac{1}{4}$ N	0.250	0.281	0.625	0.065
$\frac{1}{4}$ W	0.250	0.312	0.734	0.065
$\frac{5}{16}$ N	0.312	0.344	0.688	0.065
$\frac{5}{16}$ W	0.312	0.375	0.875	0.083
$\frac{3}{8}$ N	0.375	0.406	0.812	0.065
$\frac{3}{8}$ W	0.375	0.438	1.000	0.083
$\frac{7}{16}$ N	0.438	0.469	0.922	0.065
		0.500	1.250	0.083
		0.531	1.062	0.095

$$k = \frac{0.5774\pi Ed}{\ln \frac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

8-6 Bolt Strength (Read)

ME-305 Mechanical Engineering Design II

American Society for Testing and Materials

In the specification standards for bolts, the strength is specified by stating ASTM minimum quantities, the *minimum proof strength*, or *minimum proof load*, and the *minimum tensile strength*.

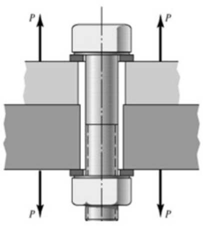
The *proof load* is the maximum load (force) that a bolt can withstand without acquiring a permanent set.

The *proof strength* is the quotient of the proof load and the tensile-stress area.

Proof strength corresponds roughly to the proportional limit and corresponds to 0.0001 in permanent set in the fastener (First measurable deviation from elastic behaviour)

8-7 Tension Joints-The External Load

ME-305 Me



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

F_i = preload

P = external tensile load

P_b = portion of P taken by bolt

P_m = portion of P taken by members

$F_b = P_b + F_i$ = resultant bolt load

$F_m = P_m - F_i$ = resultant load on members

C = fraction of external load P carried by bolt

$1 - C$ = fraction of external load P carried by members

N = Number of bolts in the joint

If N bolts equally share the total external load, then

$$P = P_{\text{total}}/N$$

8-7 Tension Joints-The External Load

ME-3

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

or

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

Since $P = P_b + P_m$, we have

$$P_b = \frac{k_b P}{k_b + k_m} \quad =CP$$

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

C **is Called stiffness constant of the joint**

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

Therefore the resultant bolt load is

$$F_b = P_b + F_i = \frac{k_b P}{k_b + k_m} + F_i = CP + F_i \qquad F_m < 0$$

8-7 Tension Joints-The External Load

ME-3

The Resultant load on the connected members is

$$F_m = P_m - F_i = \frac{k_m P}{k_b + k_m} - F_i = (1 - C)P - F_i \qquad F_m < 0$$

Table 8–12

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

	Stiffnesses, Mlbf/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

Members take over 80% of the external load

The initial tension "F_i"

ME-3

- The initial tension is given as;

$$F_i = \begin{cases} 0.75F_p & \text{(Non permanent)} \\ 0.90F_p & \text{(Permanent)} \end{cases}$$

- Where F_p is the proof load obtained from

$$F_p = A_t S_p$$

- A_t is the tensile stress area obtained from Tables 8-1 and 8-2, S_p is the minimum proof strength obtained from Tables 8-9 to 8-11
- If root dia. Is known then nominal diameter "d" is determined as

$$d = 1.25dr$$

Table 8–11

Metric Mechanical-Property Classes for Steel Bolts, Screws, and Studs*

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

*The thread length for bolts and cap screws is

n II

Bolt Tightening Techniques…

- Torque Wrench
- Pneumatic-impact wrenching
- Turn-of-the-nut method
  - The snug-tight condition is the tightness by a few impacts of an impact wrench, or the full effort of a person using an ordinary wrench
  - all additional turning after snug-tight condition develops useful tension in the bolt
  - The turn-of-the-nut method requires that the fractional number of turns necessary to develop a required preload from the snug tight condition is computed
  - For example for heavy hexagonal structural bolts, the turn-of-nut specification states that the nut should be turned a minimum of 180° from the snug-tight condition under optimum conditions.

8-8 Relating Bolt Torque to Bolt Tension

ME-305 Mechanical Engineering Design II

- F_i is determined by tightening the bolt and measuring the elongation *"δ"*
- Some times not possible
- Although the coefficient of friction may vary widely, a good estimate of the torque required to produce a given preload can be obtained as;

$$T = \frac{F_i d_m}{2}\left(\frac{l + \pi f d_m \sec\alpha}{\pi d_m - f l \sec\alpha}\right) + \frac{F_i f_c d_c}{2}$$

- Divide by πd_m and put $d_c = 1.25d$

$$T = \left[\left(\frac{d_m}{2d}\right)\left(\frac{\tan\lambda + f\sec\alpha}{1 - f\tan\lambda\sec\alpha}\right) + 0.625 f_c\right] F_i d \quad \rightarrow 8\text{-}26$$

- or

$$T = K F_i d$$

8-8 Relating Bolt Torque to Bolt Tension

ME-305 Mechanical Engineering Design II

- Blake and Kurtz experimentally determined F_i (lubricated and un-lubricated) from which K ≈ 0.2

Table 8–13

Distribution of Preload F_i for 20 Tests of Unlubricated Bolts Torqued to 90 N · m

23.6,	27.6,	28.0,	29.4,	30.3,	30.7,	32.9,	33.8,	33.8,	33.8,
34.7,	35.6,	35.6,	37.4,	37.8,	37.8,	39.2,	40.0,	40.5,	42.7

Mean value $\bar{F}_i = 34.3$ kN. Standard deviation, $\hat{\sigma} = 4.91$ kN.

Table 8–14

Distribution of Preload F_i for 10 Tests of Lubricated Bolts Torqued to 90 N · m

30.3,	32.5,	32.5,	32.9,	32.9,	33.8,	34.3,	34.7,	37.4,	40.5

Mean value, $\bar{F}_i = 34.18$ kN. Standard deviation, $\hat{\sigma} = 2.88$ kN.

8-8 Relating Bolt Torque to Bolt Tension

Table 8–15

Torque Factors K for Use with Eq. $T = KF_i d$

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

$$T = KF_i d$$

$$\boxed{\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t}}$$ →8-27

The coefficient of friction depends upon the surface smoothness, accuracy, and degree of lubrication. On the average, both f and f_c are about 0.15. The interesting fact about Eq. (8–26) is that $K \approx 0.20$ for $f = f_c = 0.15$ no matter what size bolts are employed and no matter whether the threads are coarse or fine.

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 $f = f_c = 0.15$.

8-9 Statically Loaded Tension Joint with Preload

- Yield factor of safety (n_p) guarding against static loading is

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

Thus, the yielding factor of safety guarding against the static stress exceeding the proof strength is

$$n_p = \frac{S_p}{\sigma_b} = \frac{S_p}{(CP + F_i)/A_t} \qquad (b)$$

or

$$n_p = \frac{S_p A_t}{CP + F_i} \qquad (8\text{–}28)$$

8-9 Statically Loaded Tension Joint with Preload…

ME-305 Mechanical Engineering Design II

- The load factor (n_L) is

$$\frac{C n_L P + F_i}{A_t} = S_p \qquad (c)$$

Solving for the load factor gives

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

8-9 Statically Loaded Tension Joint with Preload…

- At joint separation, all load is on the bolt and is P_0, therefore F_m= 0 and

$$(1 - C)P_0 - F_i = 0$$

- Let the factor of safety against joint separation (n_o) is

$$n_0 = \frac{P_0}{P} \qquad (e)$$

Substituting $P_0 = n_0 P$ in Eq. (d), we find

$$n_0 = \frac{F_i}{P(1 - C)} \qquad (8\text{–}30)$$

ME-305 Mechanical Engineering Design II

8-11 Fatigue Loading of Tension Joints

ME-3

- The fatigue factor of safety is given by

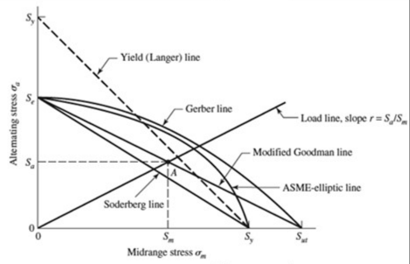
$$n_f = \frac{S_a}{\sigma_a} \rightarrow (1)$$

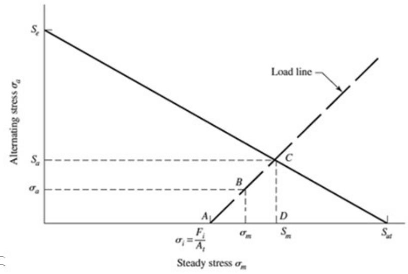
- Where S_a can be obtained on the intersection of the load line and the failure criteria
- The Load line is;

$$S_a = \frac{\sigma_a}{\sigma_m - \sigma_i}(S_m - \sigma_i) \rightarrow (2)$$

- The Goodman criteria is

$$S_a = S_e - \frac{S_e}{S_{ut}} S_m \rightarrow (3)$$





8-11 Fatigue Loading of Tension Joints…

- Equate (2) and (3) to get;

$$S_a = \frac{S_e \sigma_a (S_{ut} - \sigma_i)}{S_{ut}\sigma_a + S_e(\sigma_m - \sigma_i)}$$

- Where
  - S_e is the endurance strength from Table 8-17
  - and $\sigma_a = \frac{C(P_{\max} - P_{\min})}{2A_t}$
  - and $\sigma_m = \frac{C(P_{\max} + P_{\min})}{2A_t} + \frac{F_i}{A_t}$
  - and $\sigma_i = \frac{F_i}{A_t}$
- Put for S_a and σ_a in (1) to get

$$n_f = \frac{S_e(S_{ut} - \sigma_i)}{S_{ut}\sigma_a + S_e(\sigma_m - \sigma_i)}$$

ME-305 Mechanical Engineering Design II

8-11 Fatigue Loading of Tension Joints…

- Some time P_{min} = 0 (like pressure vessel with gas, P_{max} = P, and no gas, P_{min} = 0) then;

$$\sigma_a = \frac{CP}{2A_t}$$

$$\sigma_m = \frac{CP}{2A_t} + \frac{F_i}{A_t}$$

$$\sigma_m = \sigma_a + \sigma_i$$

- The fatigue fos using Goodman

$$n_f = \frac{S_e(S_{ut} - \sigma_i)}{\sigma_a(S_{ut} + S_e)}$$

- The fatigue fos using Gerber

$$n_f = \frac{1}{2\sigma_a S_e}\left[S_{ut}\sqrt{S_{ut}^2 + 4S_e(S_e + \sigma_i)} - S_{ut}^2 - 2\sigma_i S_e\right]$$

ME-305 Mechanical Engineering Design II

8-11 Fatigue Loading of Tension Joints…

- The fatigue fos using ASME-elliptic

$$n_f = \frac{S_e}{\sigma_a(S_p^2 + S_e^2)}\left(S_p\sqrt{S_p^2 + S_e^2 - \sigma_i^2} - \sigma_i S_e\right)$$

- The yield fos

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

Example 8.5

ME-3

Figure 8–21 shows a connection using cap screws. The joint is subjected to a fluctuating force whose maximum value is 5 kip per screw. The required data are: cap screw, 5/8 in-11 NC, SAE 5; hardened-steel washer, $t_w = \frac{1}{16}$ in thick; steel cover plate, $t_1 = \frac{5}{8}$ in, $E_s = 30$ Mpsi; and cast-iron base, $t_2 = \frac{5}{8}$ in, $E_{ci} = 16$ Mpsi.
(*a*) Find k_b, k_m, and C using the assumptions given in the caption of Fig. 8–21.
(*b*) Find all factors of safety and explain what they mean.

Figure 8–21

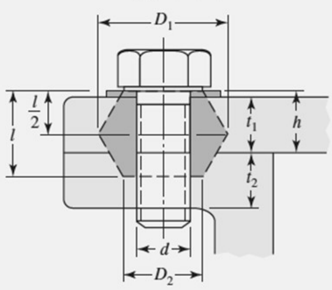
Pressure-cone frustum member model for a cap screw. For this model the significant sizes are

$$l = \begin{cases} h + t_2/2 & t_2 < d \\ h + d/2 & t_2 \ge d \end{cases}$$

$D_1 = d_w + l \tan\alpha = 1.5d + 0.577l$

$D_2 = d_w = 1.5d$

where l = effective grip. The solutions are for $\alpha = 30°$ and $d_w = 1.5d$.



Example 8.5 (Solution)

ME-3

(*a*) For the symbols of Figs. 8–15 and 8–21, $h = t_1 + t_w = 0.6875$ in, $l = h + d/2 = 1$ in, and $D_2 = 1.5d = 0.9375$ in. The joint is composed of three frusta; the upper two frusta are steel and the lower one is cast iron.

For the upper frustum: $t = l/2 = 0.5$ in, $D = 0.9375$ in, and $E = 30$ Mpsi. Using these values in Eq. (8–20) gives $k_1 = 46.46$ Mlbf/in.

For the middle frustum: $t = h - l/2 = 0.1875$ in and $D = 0.9375 + 2(l - h)\tan 30° = 1.298$ in. With these and $E_s = 30$ Mpsi, Eq. (8–20) gives $k_2 = 197.43$ Mlbf/in.

The lower frustum has $D = 0.9375$ in, $t = l - h = 0.3125$ in, and $E_{ci} = 16$ Mpsi. The same equation yields $k_3 = 32.39$ Mlbf/in.

Substituting these three stiffnesses into Eq. (8–18) gives $k_m = 17.40$ Mlbf/in. The cap screw is short and threaded all the way. Using $l = 1$ in for the grip and $A_t = 0.226\text{ in}^2$ from Table 8–2, we find the stiffness to be $k_b = A_t E/l = 6.78$ Mlbf/in. Thus the joint constant is

$$C = \frac{k_b}{k_b + k_m} = \frac{6.78}{6.78 + 17.40} = 0.280$$

n II

Example 8.5 (Solution)…

ME-3

(*b*) Equation (8–30) gives the preload as

$$F_i = 0.75F_p = 0.75A_t S_p = 0.75(0.226)(85) = 14.4 \text{ kip}$$

where from Table 8–9, $S_p = 85$ kpsi for an SAE grade 5 cap screw. Using Eq. (8–28), we obtain the load factor as the yielding factor of safety is

Answer
$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{85(0.226)}{0.280(5) + 14.4} = 1.22$$

This is the traditional factor of safety, which compares the maximum bolt stress to the proof strength.

Using Eq. (8–29),

Answer
$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{85(0.226) - 14.4}{0.280(5)} = 3.44$$

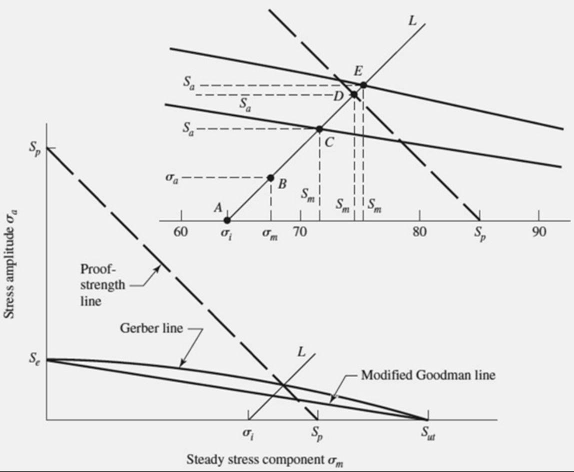
This factor is an indication of the overload on P that can be applied without exceeding the proof strength.

Next, using Eq. (8–30), we have

Answer
$$n_0 = \frac{F_i}{P(1 - C)} = \frac{14.4}{5(1 - 0.280)} = 4.00$$

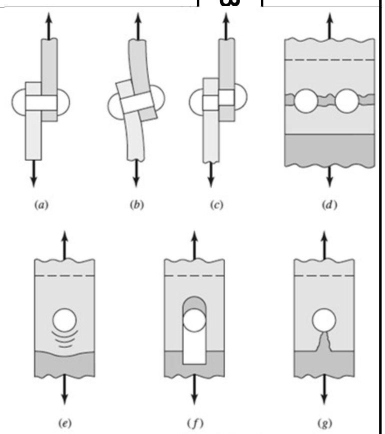
Example 8.5 (Solution)…

For the remaining factors, refer to Fig. 8–22. This diagram contains the modified Goodman line, the Gerber line, the proof-strength line, and the load line. The intersection



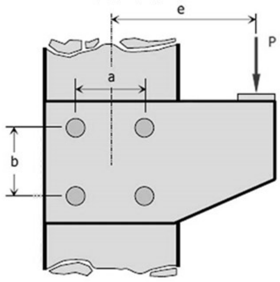
8-12 Bolted and Riveted joints loaded in shear

- Modes of failure in shear loading of a bolted or riveted connections are shown
- "a" is bolt under shear loading
- "b" is bending $\sigma = \frac{Mc}{I}$
- "c" is pure shear $\tau = \frac{F}{A}$
- "d" is tension $\sigma = \frac{F}{A}$
- "e" is crushing of rivet or plate
- "f" is tearing
- "g" is tensile tearing



8-12 Shear joints with Eccentric loading

- Joints shown in figure are widely used in structures
- The joint member is loaded eccentrically
- The bolts/rivets are in shear
- Let A_1, A_2, A_3 and A_4 be the cross-sectional areas of the bolts
- x and y are the coordinates of the bolts
- G is the centre of gravity of the group of bolts whose coordinates are given by

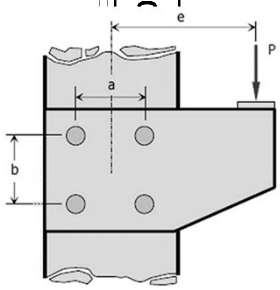


8-12 Eccentrically loaded bolted joints in shear…

$$\bar{x} = \frac{A_1x_1 + A_2x_2 + A_3x_3 + A_4x_4 + A_5x_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\sum_1^n A_i x_i}{\sum_1^n A_i}$$

$$\bar{y} = \frac{A_1y_1 + A_2y_2 + A_3y_3 + A_4y_4 + A_5y_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\sum_1^n A_i y_i}{\sum_1^n A_i}$$

- Where *x'* and *y'* are the coordinates of the *G* (i.e. find G using above eqns.)
- The external load *F* is at a distance *e* from the *G*.
- *Load on* each bolt F_i can be determined as shown on next slides



8-12 Eccentrically loaded bolted joints in shear…

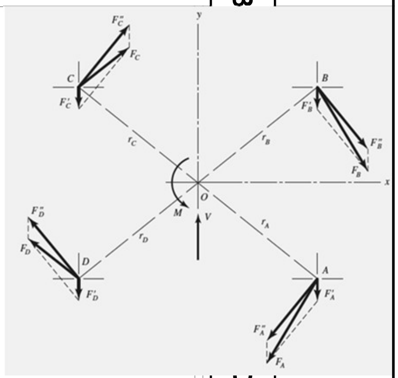
- Step 1 (Assume 4-bolts)
  - Determine the primary shear forces on each bolt i.e. *F'*

$$F'_A = F'_B = F'_C = F'_D = \frac{F}{\text{No. of bolts}}$$

- Step 2
  - Determine the secondary load *F"* by taking moment about the *G*

$$F''_A = \frac{Fer_A}{(r_A^2 + r_B^2 + r_C^2 + r_D^2)}$$

- Step 3
  - The primary and secondary loads are then added by vector addition method to determine load F_A to F_D



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8-12 Eccentrically loaded bolted joints in shear…

- Step 4
  - Choose the bolt which is subjected to maximum shear force
- Step 5
  - The size of bolt can be determined by

$$\tau = \frac{F}{A}$$

  - τ is maximum permissible shear stress.
  - F maximum shear force calculated in step 4.
  - $A = \frac{\pi}{4} d_r^2$ and $d=1.25d_r$
- Step 6
  - Choose standard size bolt from Table A-17

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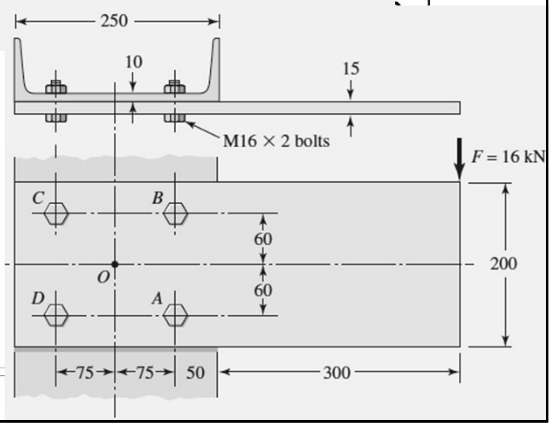
Example 8-7

Shown in Fig. 8–28 is a 15- by 200-mm rectangular steel bar cantilevered to a 250-mm steel channel using four tightly fitted bolts located at A, B, C, and D.

For a $F = 16$ kN load find
(a) The resultant load on each bolt
(b) The maximum shear stress in each bolt
(c) The maximum bearing stress
(d) The critical bending stress in the bar
e) Find the Factor of safety of the bolt(s) if the bolt material is of Property Class 4.6



Example 8-7…

(b) Bolts A and B are critical because they carry the largest shear load. Does this shear act on the threaded portion of the bolt, or on the unthreaded portion? The bolt length will be 25 mm plus the height of the nut plus about 2 mm for a washer. Table A–31 gives the nut height as 14.8 mm. Including two threads beyond the nut, this adds up to a length of 43.8 mm, and so a bolt 46 mm long will be needed. From Eq. (8–14) we compute the thread length as $L_T = 38$ mm. Thus the unthreaded portion of the bolt is $46 - 38 = 8$ mm long. This is less than the 15 mm for the plate in Fig. 8–28, and so the bolt will tend to shear across its minor diameter. Therefore the shear-stress area is $A_s = 144$ mm^2, and so the shear stress is

Table 8-1

$$\tau = \frac{F}{A_s} = -\frac{21.0(10)^3}{144} = 146 \text{ MPa}$$

(c) The channel is thinner than the bar, and so the largest bearing stress is due to the pressing of the bolt against the channel web. The bearing area is $A_b = td = 10(16) = 160$ mm^2. Thus the bearing stress is

$$\sigma = -\frac{F}{A_b} = -\frac{21.0(10)^3}{160} = -131 \text{ MPa}$$

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Example 8-7…

(*d*) The critical bending stress in the bar is assumed to occur in a section parallel to the y axis and through bolts A and B. At this section the bending moment is

$$M = 16(300 + 50) = 5600 \text{ N} \cdot \text{m}$$

The second moment of area through this section is obtained by the use of the transfer formula, as follows:

$$I = I_{\text{bar}} - 2(I_{\text{holes}} + \bar{d}^2 A)$$
$$= \frac{15(200)^3}{12} - 2\left[\frac{15(16)^3}{12} + (60)^2(15)(16)\right] = 8.26(10)^6 \text{ mm}^4$$

Then

$$\sigma = \frac{Mc}{I} = \frac{5600(100)}{8.26(10)^6}(10)^3 = 67.8 \text{ MPa}$$

e) Covered during Lecture

Sample Problems

- 8.4, 8.6, 8.7, 8.8, 8.9, 8.10
- 8.11, 8.12, 8.14, 8.15, 8.19,8.26
- 8.29, 8.30, 8.32, 8.33
- 8.48, 8.51, 8.52, 8.54
- 8.60, 8.67, 8.70, 8.75, 8.76

From

Shigley's Mechanical Engineering Design, 9th Ed.

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