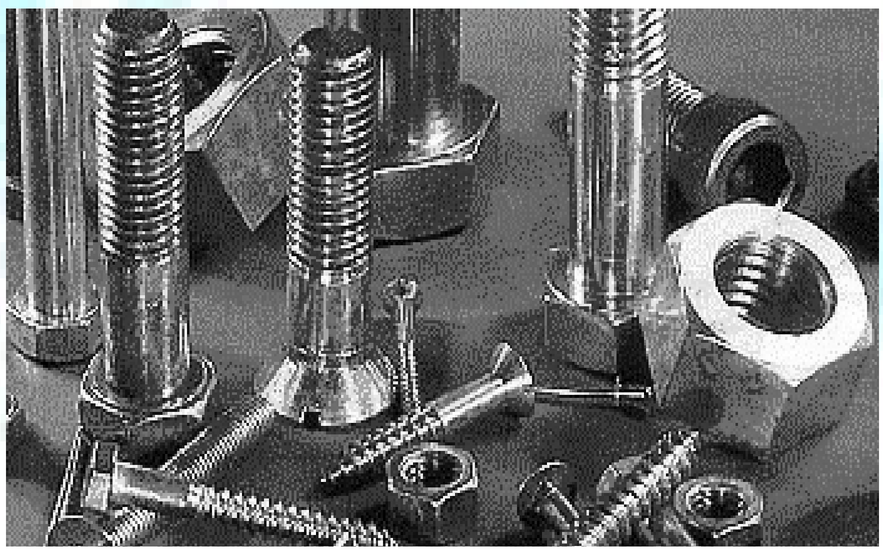
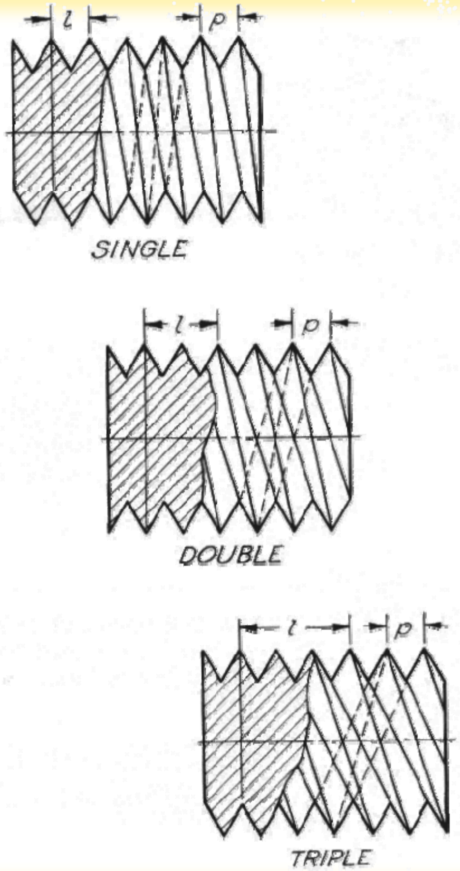
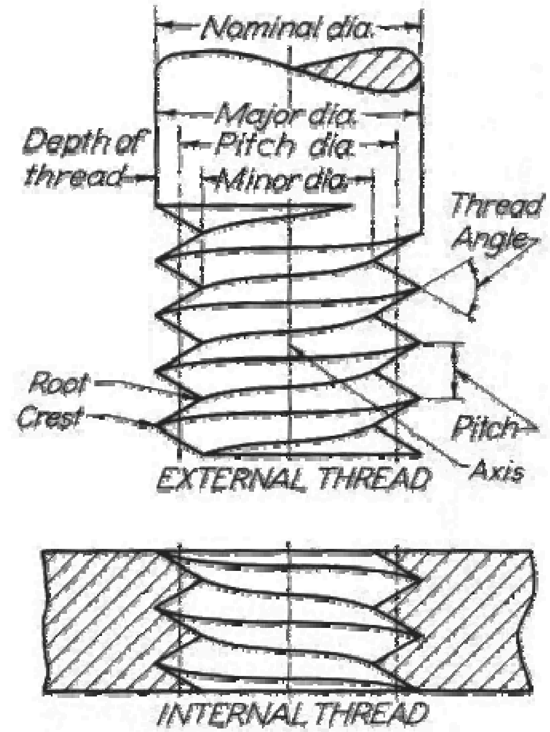
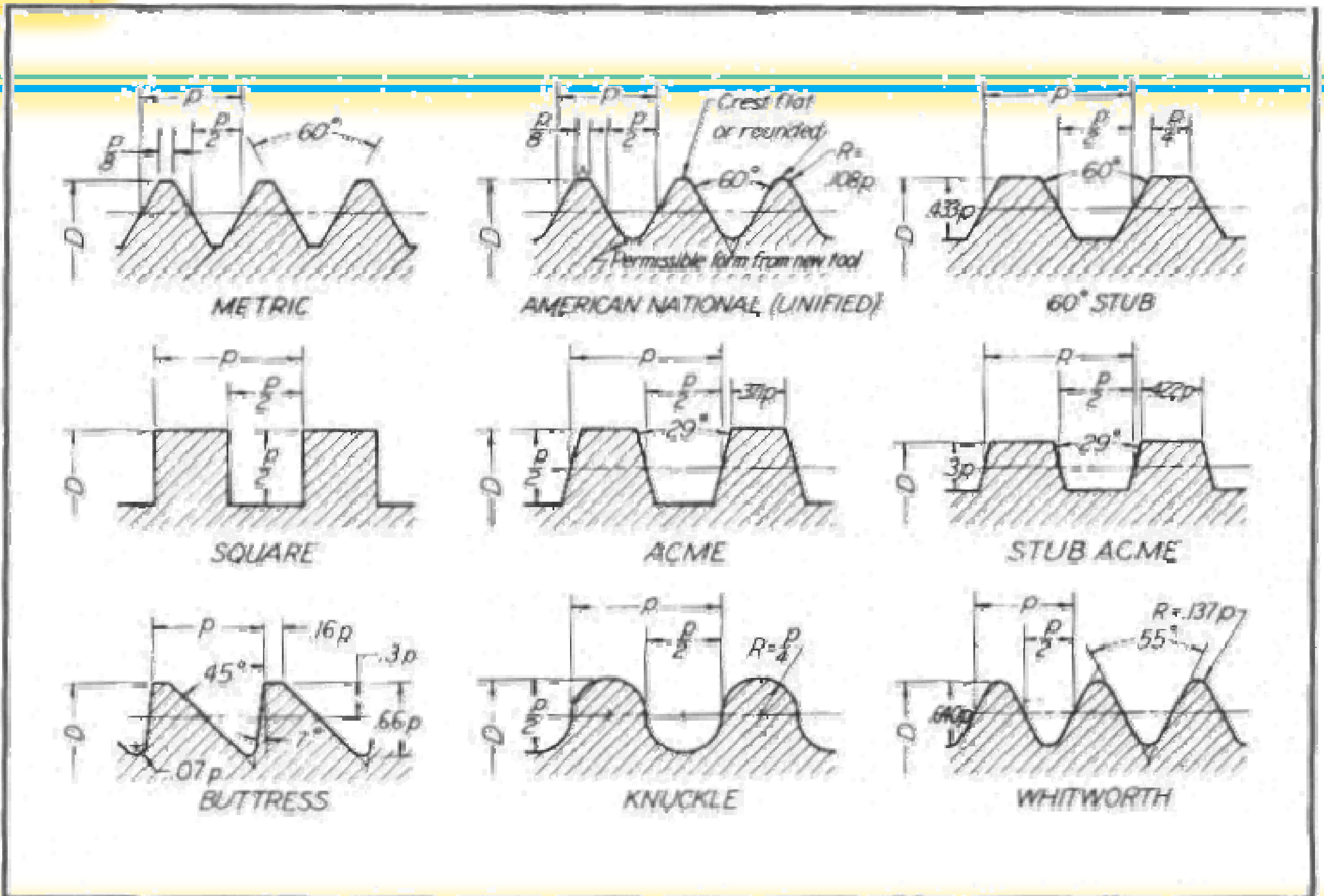
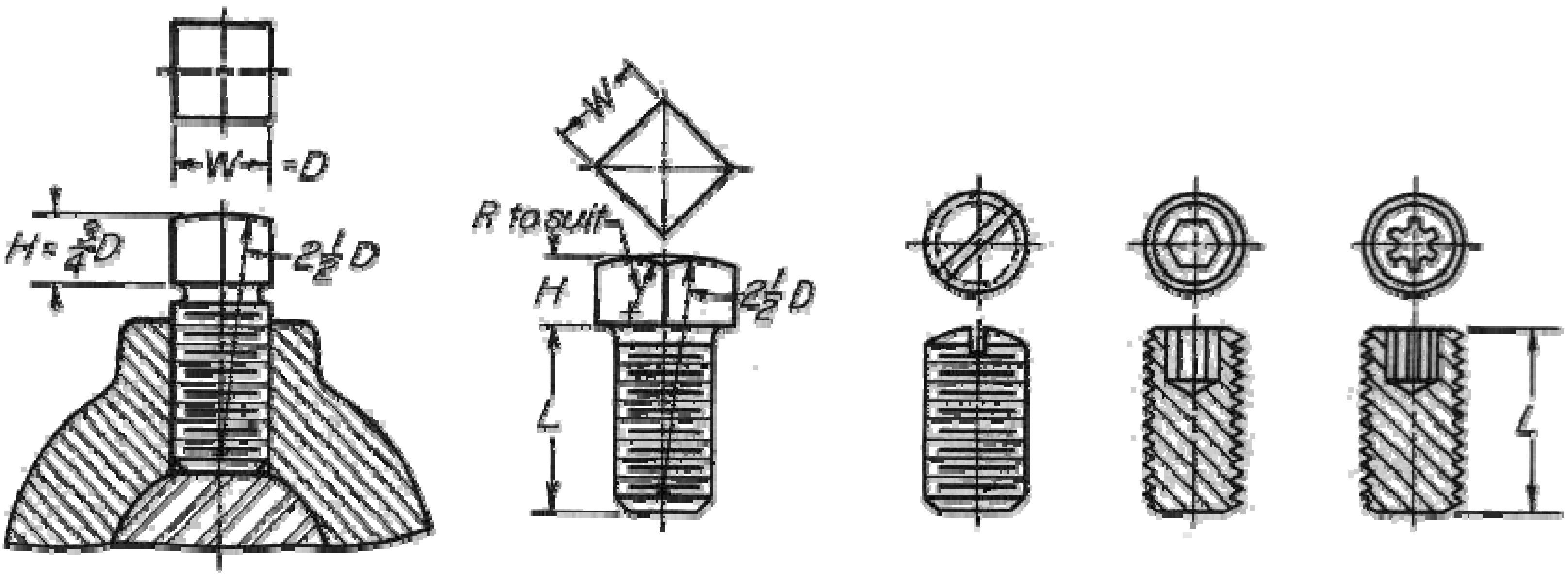
Screws, Fasteners, and the Design of Nonpermanent Joints

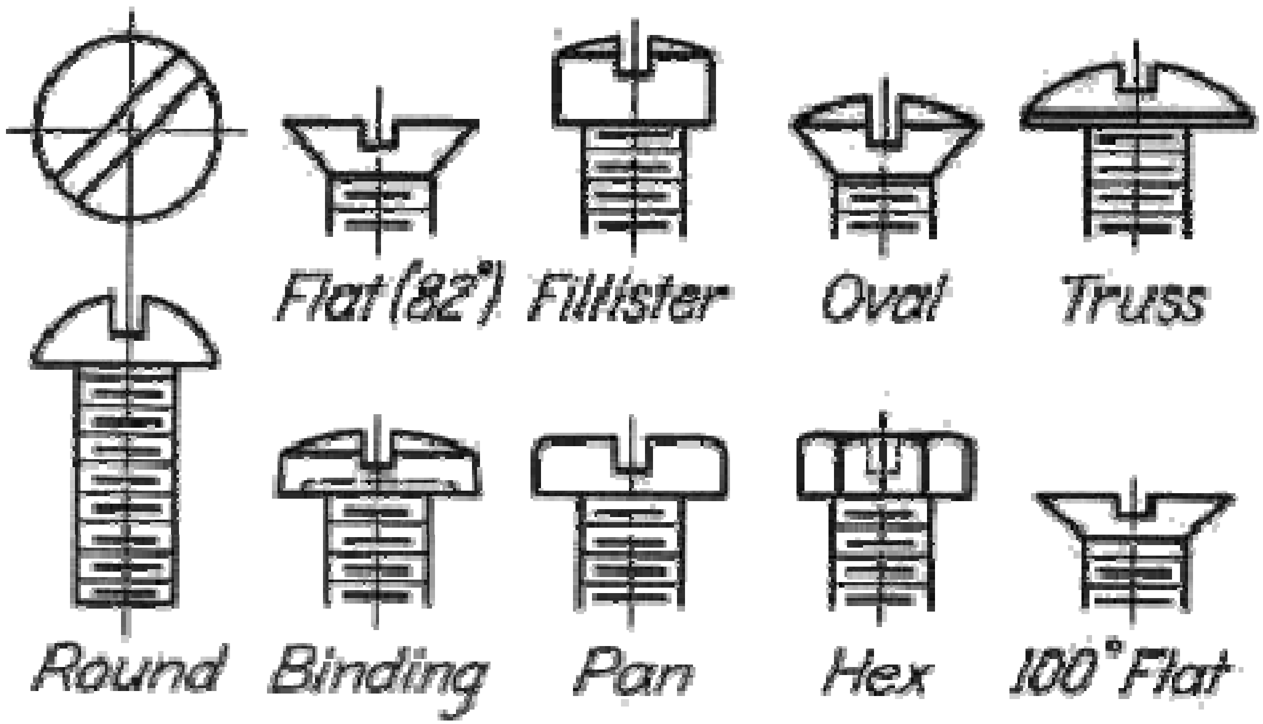



l
p
SINGLE
l
p
DOUBLE
l
p
TRIPLE


Nominal dia.
Major dia.
Pitch dia.
Minor dia.
Depth of thread
Thread Angle
Root
Crest
Pitch
Axis
EXTERNAL THREAD
INTERNAL THREAD


METRIC
AMERICAN NATIONAL (UNIFIED)
60° STUB
SQUARE
ACME
STUB ACME
BUTTRESS
KNUCKLE
WHITWORTH
Crest flat
or rounded
Permissible form from new tool
60°
29°
45°
55°
.433p
.3p
.16p
.66p
.07p
R = .137p


W
= D
H = $\frac{3}{4}$D
$2\frac{1}{2}$D
W
R to suit
H
$2\frac{1}{2}$D
L
L


Flat (82°)
Fillister
Oval
Truss
Round
Binding
Pan
Hex
100° Flat

Chapter Outline

LECTURE 35

8-3 Strength Constraints
8-4 Joints-Fasteners Stiffness
8-5 Joints-Member Stiffness
8-6 Bolt Strength

Example-3

A single-threaded 20 mm power screw is 20 mm in diameter with a pitch of 5 mm. A vertical load on the screw reaches a maximum of 3 kN. The coefficients of friction are 0.06 for the collar and 0.09 for the threads. The frictional diameter of the collar is 45 mm. Find the overall efficiency and the torque to "raise" and "lower" the load.

Example-3

Given

l = 5 mm

d = 20 mm

Pitch, p = 5 mm

$d_m = (d - p/2) = 20 - 2.5 = 17.5$ mm

Load, F = 3 kN

Frictional coefficient for the thread, $f = 0.09$

Frictional coefficient for the collar, $f_c = 0.06$

Frictional diameter of the collar, D_c = 45 mm

Example-3 (Cont.'d)

Solution

The torque required to raise the load is:

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

$$T_R = \frac{3(17.5)}{2}\left(\frac{5 + \pi(0.09)(17.5)}{\pi(17.5) - (0.09)(5)}\right) + \frac{3(0.06)(45)}{2}$$

$$= 4.78902 + 4.05 = 8.84 \text{ N.m}$$

The torque required to lower the load is:

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

$$T_L = \frac{3(17.5)}{2}\left(\frac{\pi(0.09)(17.5) - 5}{\pi(17.5) + (0.09)(5)}\right) + \frac{3(0.06)(45)}{2}$$

$$= -0.02462 + 4.05 = 4.025 \text{ N.m}$$

Example-2 (Cont.'d)

Since T_L is positive, the thread is self -locking. The efficiency is:

$$e = \frac{T_0}{T_R} = \frac{Fl}{2\pi T_R}$$

$$e = \frac{3(5)}{2\pi(8.84)} = 0.27$$

Power Screws Stress Analysis

The following stresses should be checked on both the nut and the screw:

1. Shearing stress in screw body.

$$\tau = \frac{16T}{\pi d_r^3} \tag{8-7}$$

2. Axial stress in screw body

$$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2} \tag{8-8}$$

Power Screws Stress Analysis

3. Thread bearing stress

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

where n_t is the number of engaged threads.

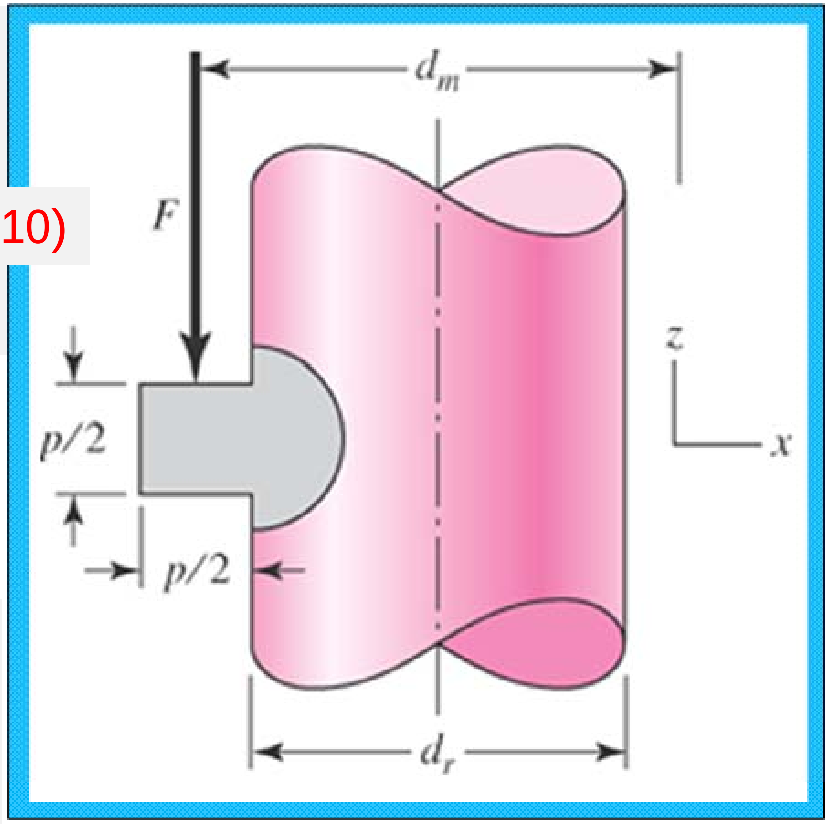
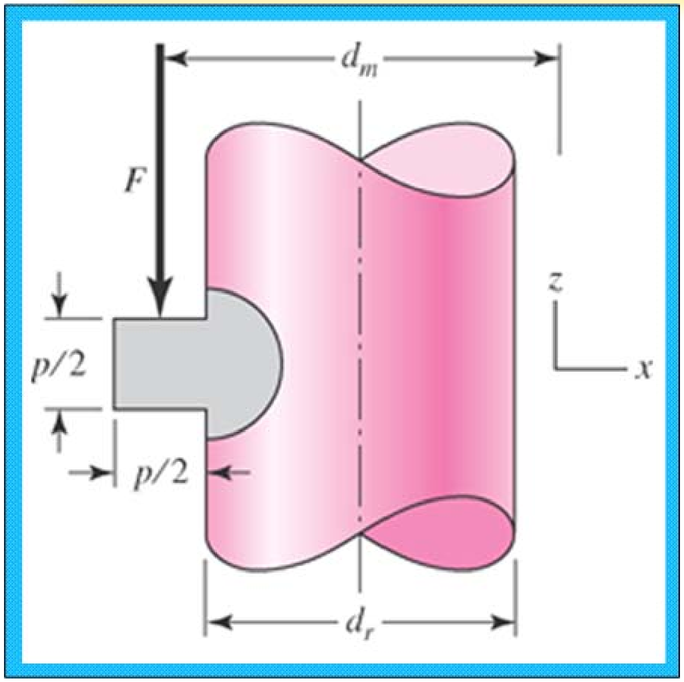


Figure 8-8
Geometry of square thread useful in finding bending and transverse shear stresses at the thread root

Power Screws Stress Analysis

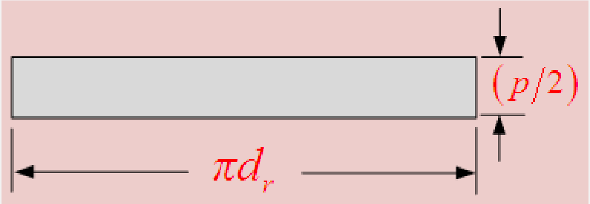


4. Thread bending stress

The bending stress at the root of the thread is given by

$$\frac{I}{C} = \frac{(1/12)(\pi d_r n_t)(p/2)^3}{(P/2)/2} = \frac{\pi}{24} d_r n_t p^2$$

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

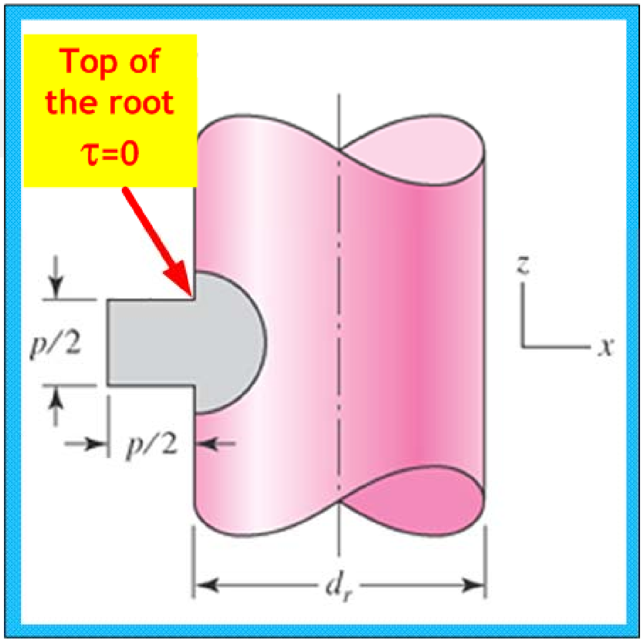


$$\sigma_b = \frac{M}{I/c} = \left(\frac{Fp}{4}\right)\frac{24}{\pi d_r n_t p^2} = \frac{6F}{\pi d_r n_t p} \qquad \text{(8-11)}$$

Power Screws Stress Analysis

5. Transverse shear stress at the center of the thread root

$$\tau = \frac{3V}{2A} = \frac{3}{2}\frac{F}{\pi d_m n_t\, p/2} = \frac{3F}{\pi d_m n_t\, p} \quad (8\text{-}12)$$



Notice that the transverse shear stress at **the top of the root** is zero

$$\tau = 0$$

Power Screws Stress Analysis

The state of stress at top of root "plane" is

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

$$\sigma_y = 0 \qquad \tau_{yz} = \frac{16T}{\pi d_r^3}$$

$$\sigma_z = -\frac{4F}{\pi d_r^2} \qquad \tau_{zx} = 0$$

- Von-Mises Stress at top of root plane is calculated using Eq. (6-14) of Sec. (6-5) and failure criteria applied (see example 8-1).

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

Power Screws Stress Analysis

- The engaged threads cannot share the load equally. Some experiments show that

  the first engaged thread carries 0.38 of the load

  the second engaged thread carries 0.25 of the load

  the third engaged thread carries 0.18 of the load

  the seventh engaged thread is free of load

- In estimating thread stresses by the equations above, substituting n_t to 1 will give the largest level of stresses in the thread-nut combination

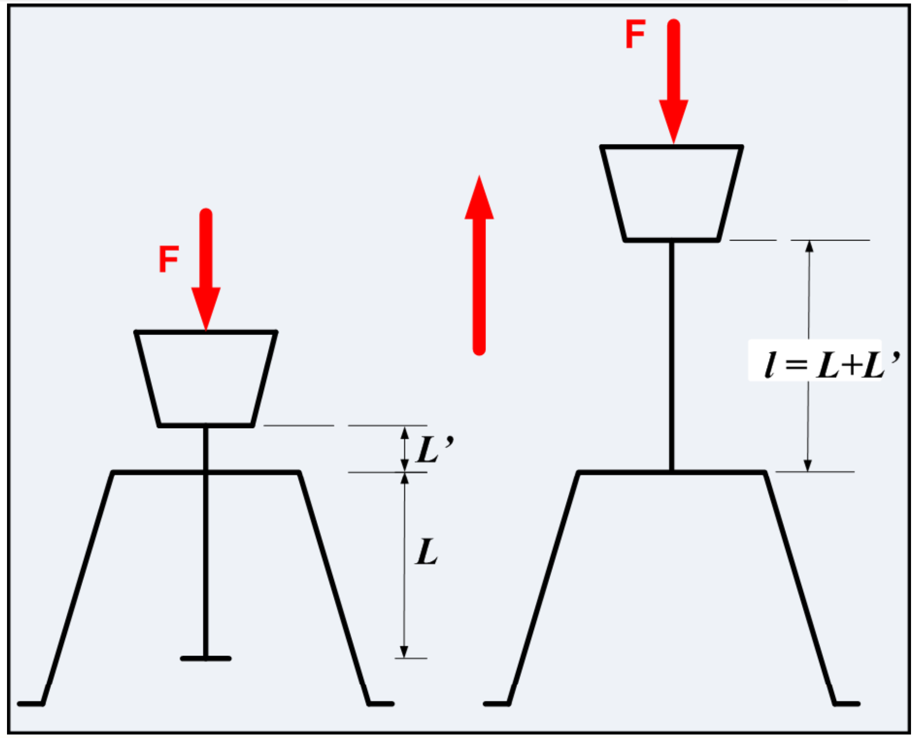
Power Screws Buckling

Assuming that the column (screw) is a Johnson column

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

where

$$l = L + L'$$



Example-3 (Example 8-1 in Textbook)

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 stresses bending at the root, shear at the root, and von Mises stress and maximum shear stress at the same location.

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

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

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

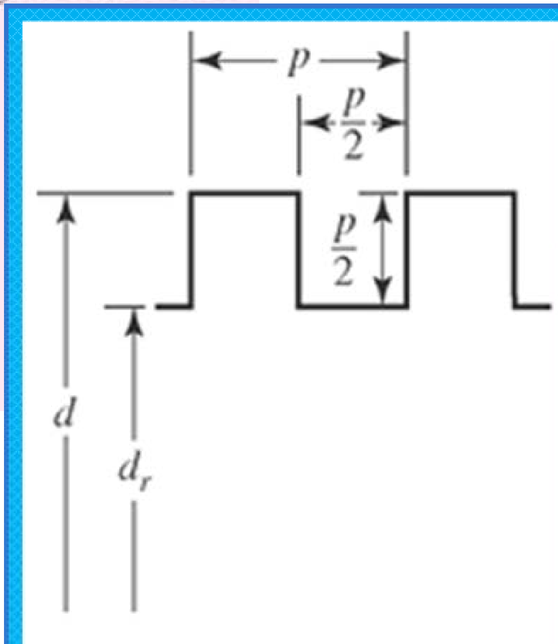
$$l = np = 2(4) = 8 \text{ mm}$$

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

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

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

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



(*a*)

Example-3 (Example 8-1 in Textbook)

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

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

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

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

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

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

Example-3 (Example 8-1 in Textbook)

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

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

The axial nominal normal stress σ is

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

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

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

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

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

Example-3 (Example 8-1 in Textbook)

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

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

$$\sigma_y = 0 \qquad \tau_{yz} = 6.07 \text{ MPa}$$

$$\sigma_z = -10.39 \text{ MPa} \qquad \tau_{zx} = 0$$

Equation (5–14) of Sec. 5–5 can be written as

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

$$= 48.7 \text{ MPa}$$

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 τ_{max} as well. The principal stresses can be found from Eq. (3–15); however, sketch the stress element and note that there are no shear stresses on the x face. This means that σ_x is a principal stress. The remaining stresses can be transformed by using the plane stress equation, Eq. (3–13). Thus, the remaining principal stresses are

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

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

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

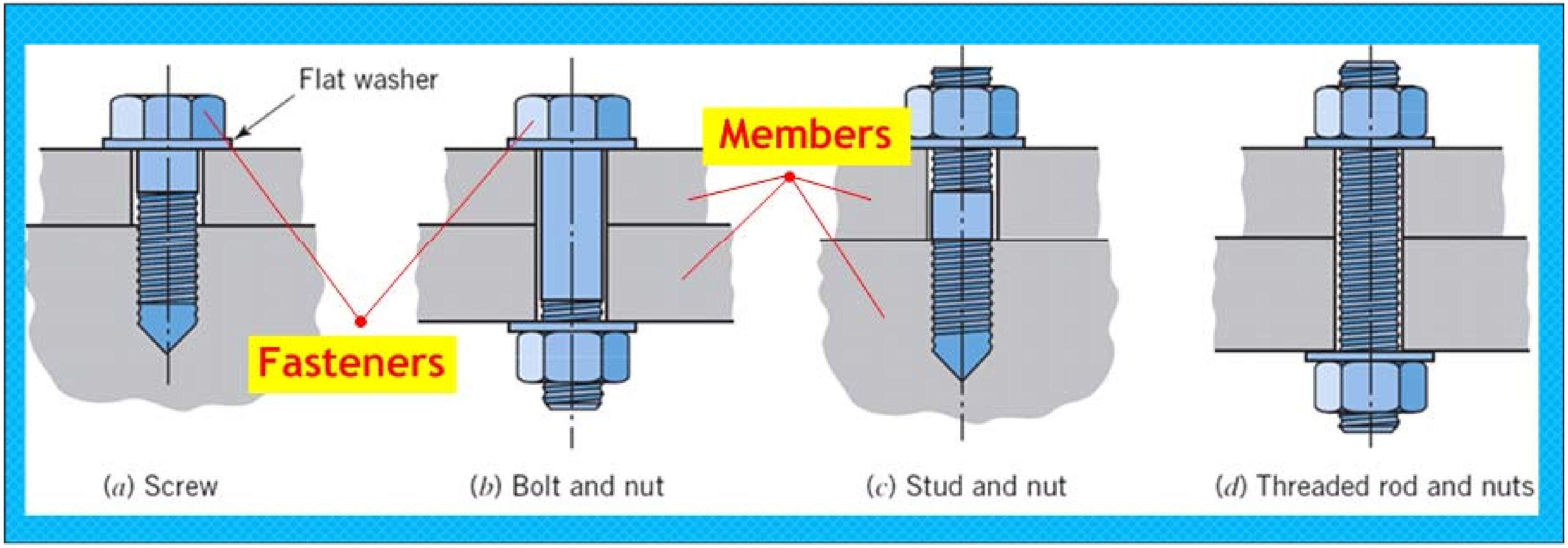
$$= 48.7 \text{ MPa}$$

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

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

Types of fasteners

Nonpermanent Joints



Three types of threaded fastener: (a) Screw (b)Bolt and nut; (c) Stud and nut, (d) Threaded rod and nuts

Threaded Fasteners

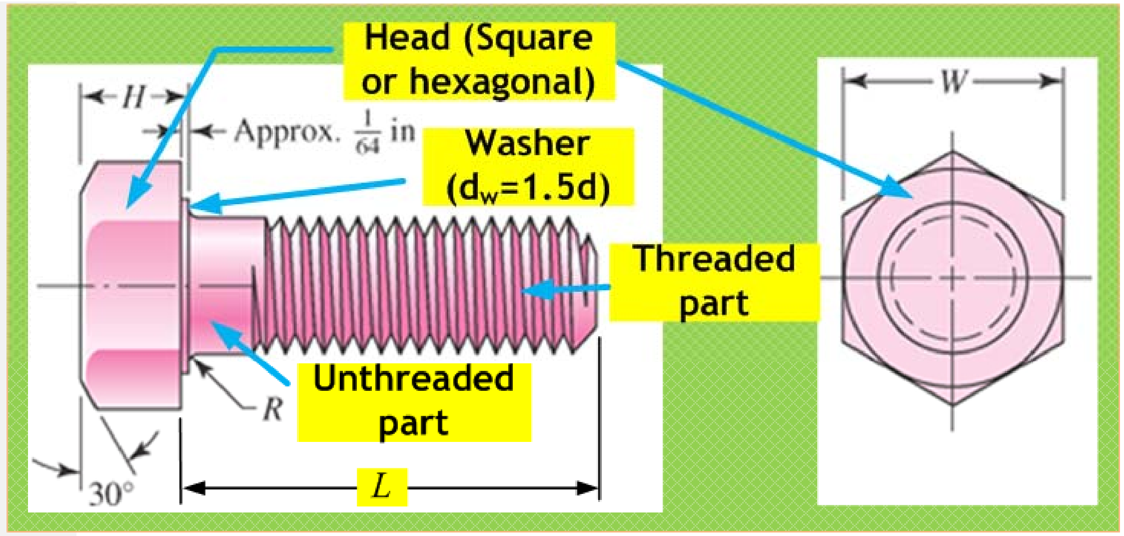
A- BOLTS:

Purpose:

to clamp two or more members together.

Parts:

(1) Head (Square or Hexagonal)
(2) Washer $(d_w=1.5d)$
(3) Threaded part
(4) Unthreaded part



Dimensions of square and hexagonal bolts are given in TABLE A-29

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Threaded Fasteners

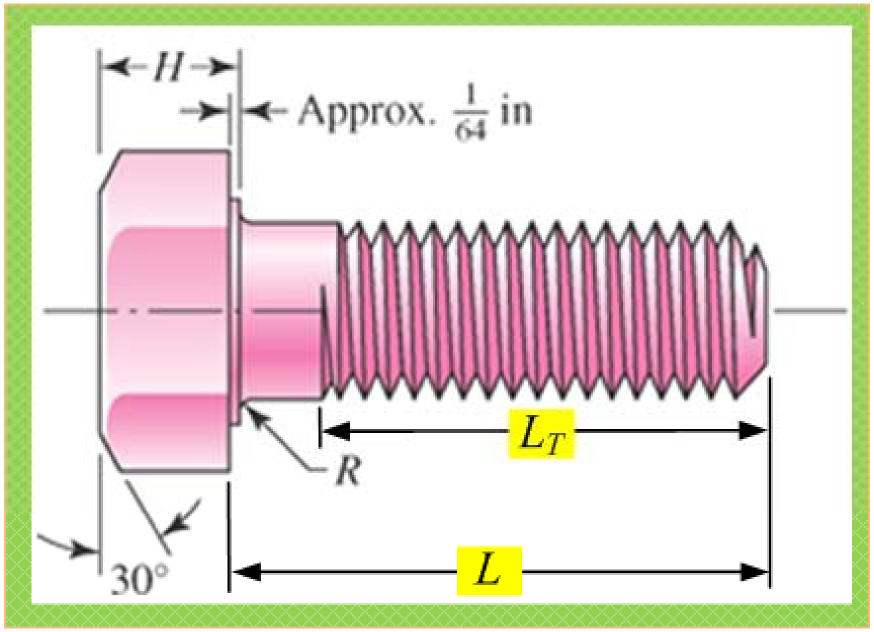
The diameter of the washer face is the same as the width across the flats of the hexagon. The thread length (L_T) is :

English

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

Metric (in mm)

$$L_T = \begin{cases} 2D + 6 & L \le 125 \quad D \le 48 \\ 2D + 12 & 125 < L \le 200 \\ 2D + 25 & L > 200 \end{cases}$$



D **is the nominal diameter**

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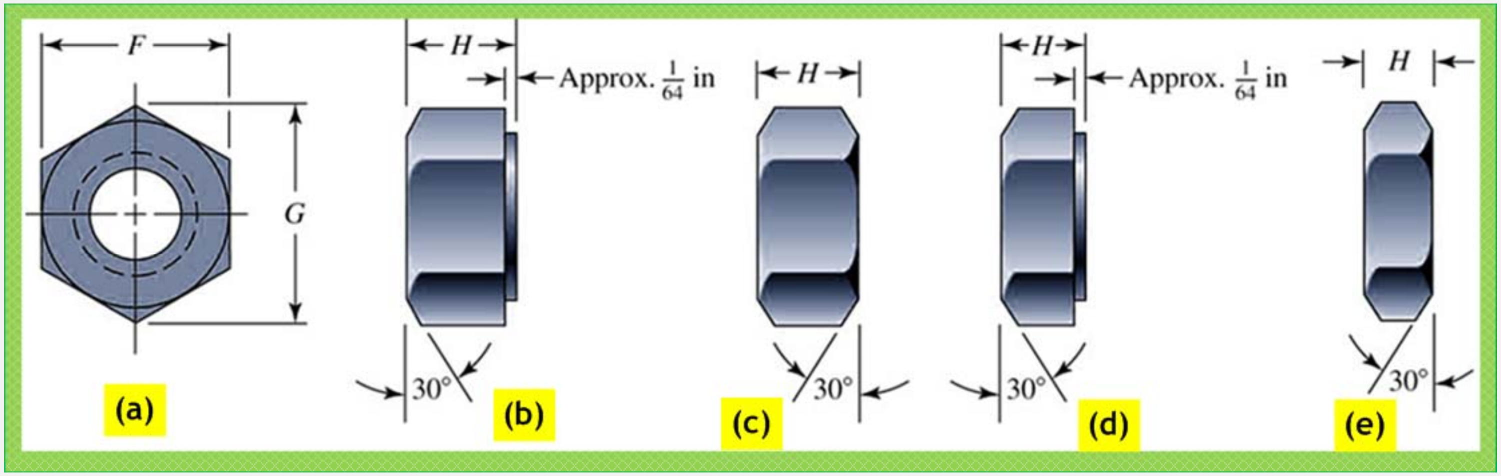
Threaded Fasteners

B- NUTS: Same material as that of a screw

Table A-31 gives dimensions of Hexagonal nuts



Good Practice:

1. Never re-use nuts;
2. Tightening should be done such that 1 or 2 threads come out of the nut;
3. Washers should always be used under bolt head to prevent burr stress concentration.

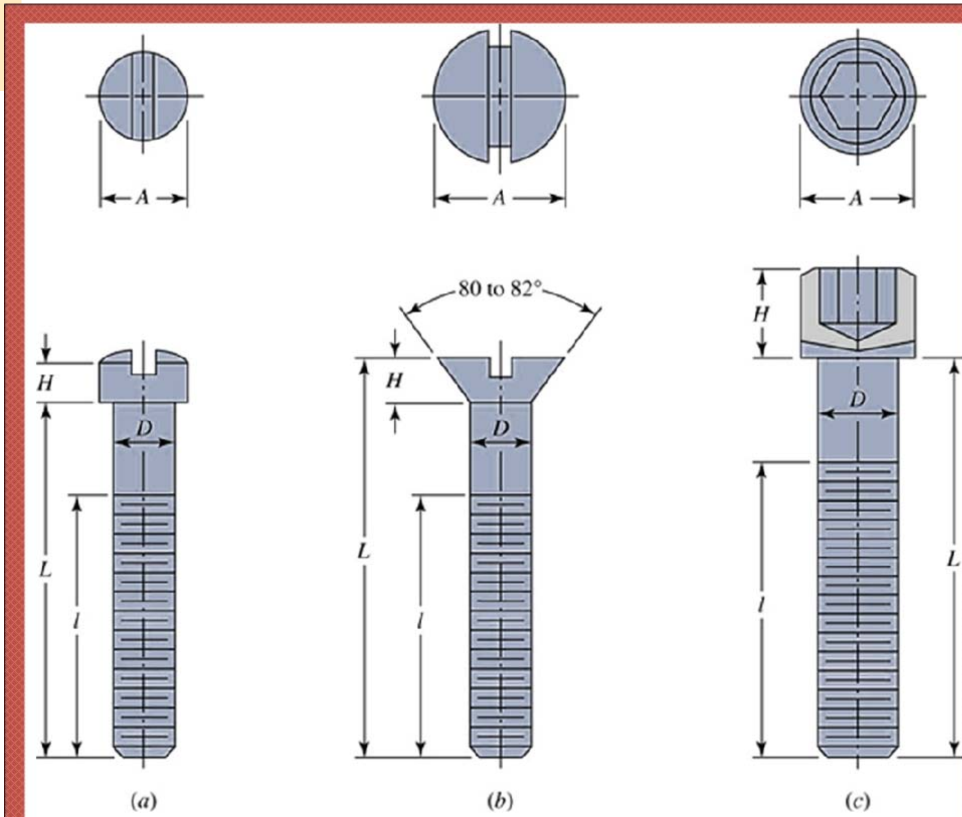
Threaded Fasteners

Common Cap screws

Used for clamping members same as bolt except that 1 member should be threaded.

The head of a hexagon-head cup screw H_{cap} is slightly thinner than that of a hexagonal head bot H_{bolt}.

Figure 8-10
Typical cap-screw heads: (a) fillister head; (b) flat head; (c) hexagonal socket head



Threaded Fasteners

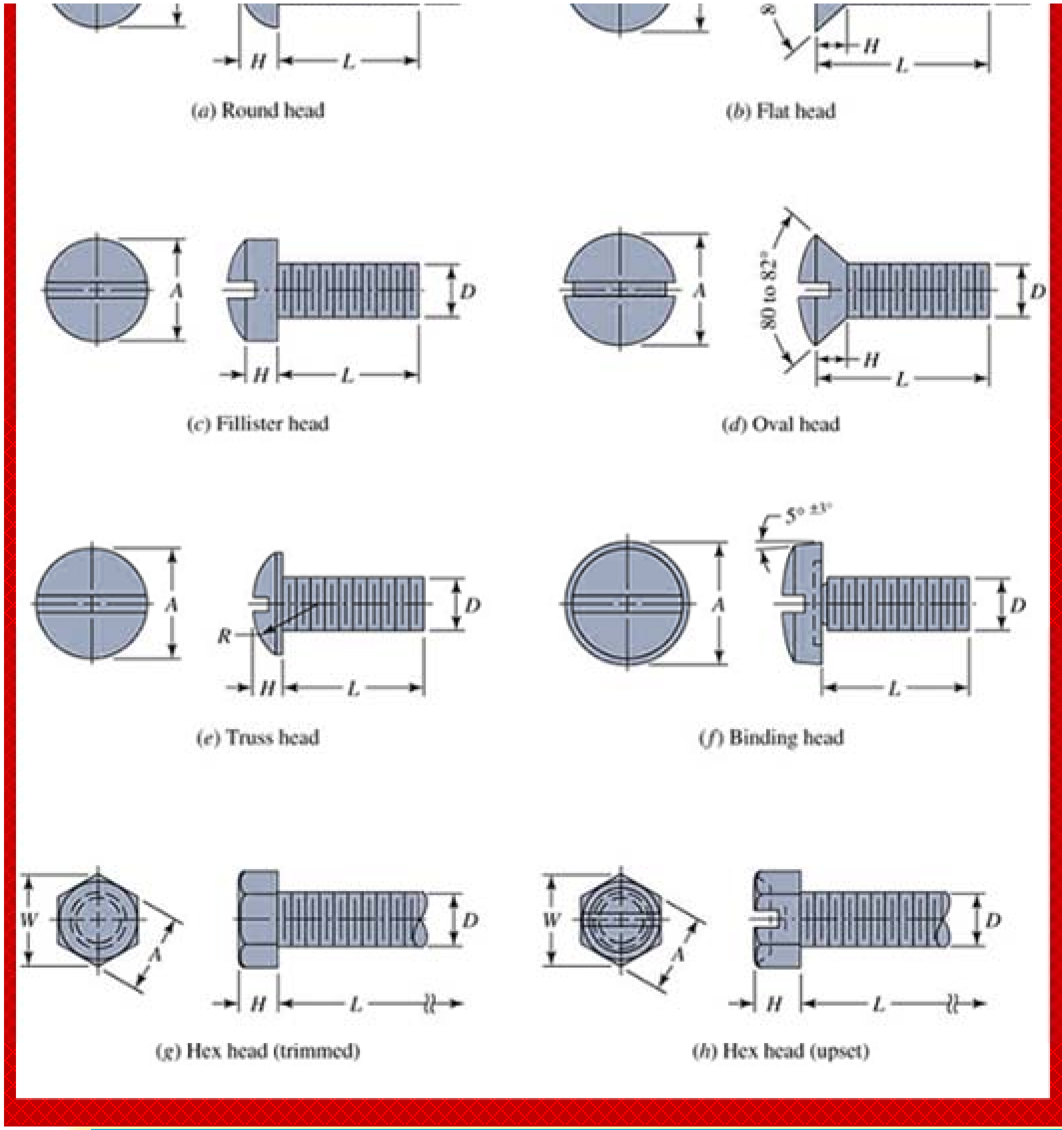


Figure 8.11:
Types of heads used in machine screws

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Joints: Fastener Stiffness

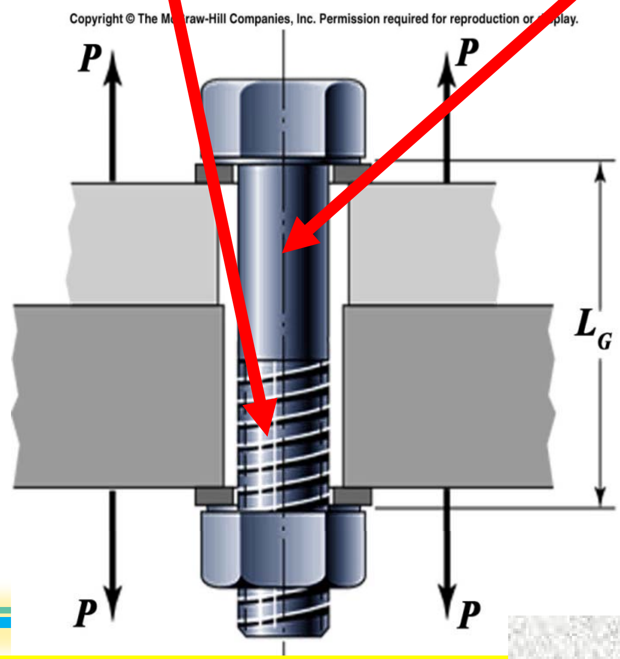
In joint under tension the **members are under compression and the bolt under tension**: k_b = equivalent spring constant of bolt composed of threaded (k_t) and unthreaded (k_d) parts acting as springs in series

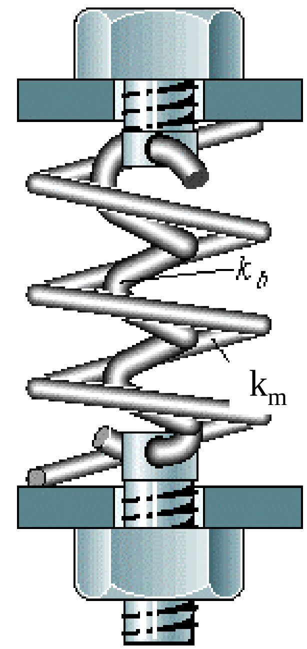
$$\frac{1}{k_b} = \frac{1}{k_d} + \frac{1}{k_t}$$

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

$$k_d = \frac{A_d E}{l_d} \qquad k_t = \frac{A_t E}{l_t}$$

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

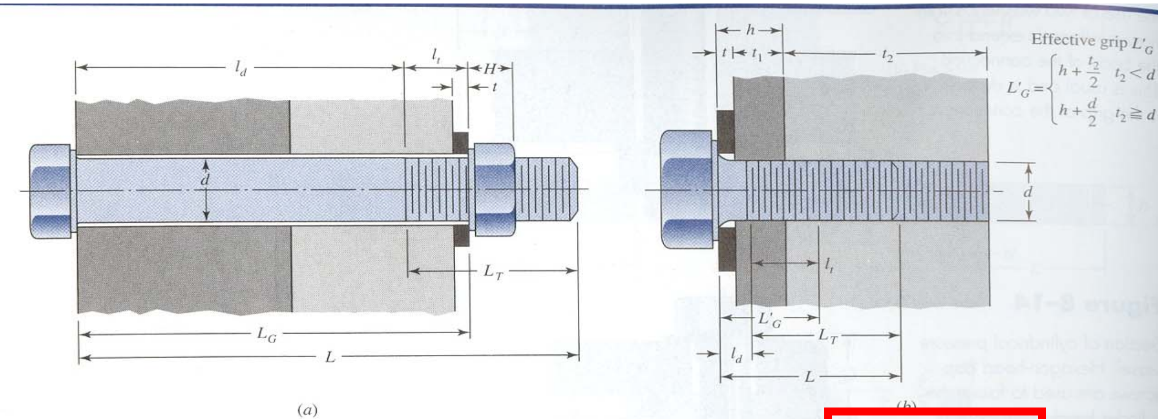




For short bolts $k_b = k_t$

Joints: Fastener Stiffness

To find different parameters use table 8-7



Given fastener diameter d and pitch p or number of threads

Grip is thickness L_G

Effective grip

$$L'_G = \begin{cases} h + t_2/2, & t_2 < d \\ h + d/2, & t_2 > d \end{cases}$$

Washer thickness from Table A–32 or A–33

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:

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

Fastener length: $L > L_G + H$

Round up using Table A–17*

Fastener length: $L > h + 1.5$

Length of useful unthreaded portion: $l_d = L - L_T$

Length of threaded portion: $l_t = L_G - l_d$

Length of useful unthreaded portion: $l_d = L - L_T$

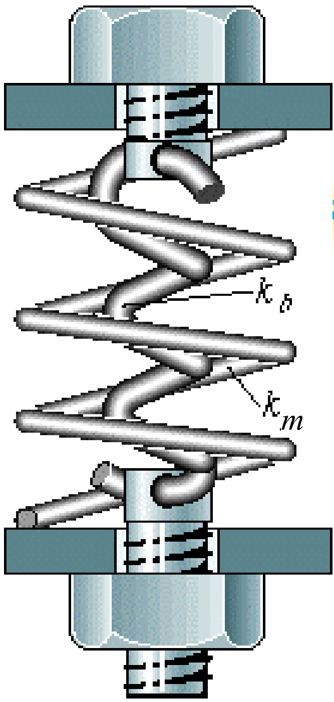
Length of useful threaded portion: $l_t = L'_G - l_d$

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

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

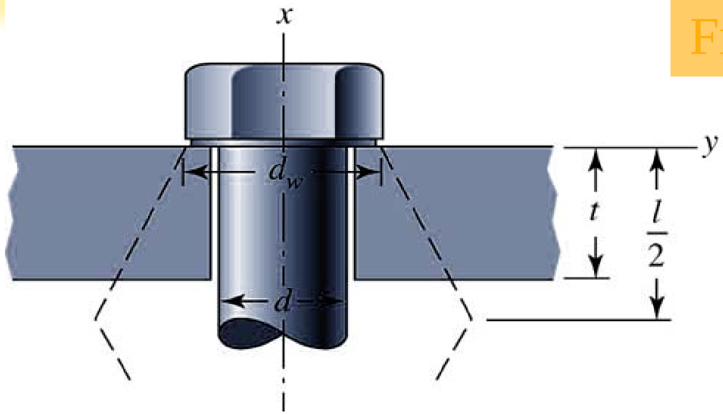
Fastener stiffness:

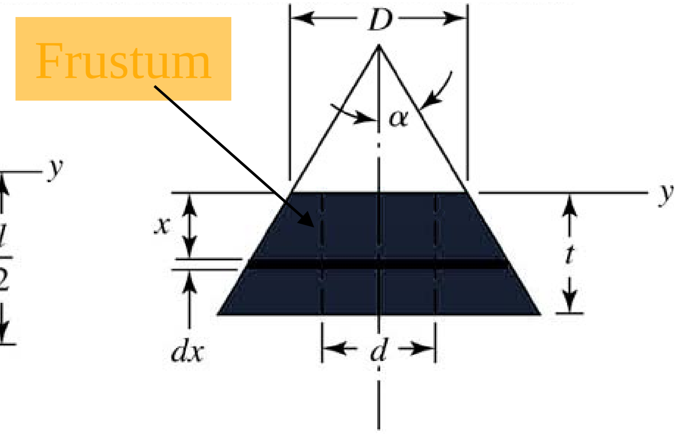
$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}$$



Joints: Member Stiffness

Members act as springs under compression





Compression stress distribution from experimental data

Equivalent spring constant k_m

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

$$d\delta = \frac{P\,dx}{EA}$$

Integrating from 0 to l

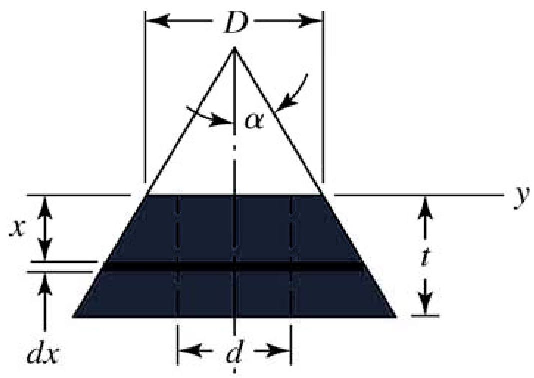
$$\delta = \frac{P}{\pi E d \tan\alpha} \ln \frac{(2t\tan\alpha + D - d)(D + d)}{(2t\tan\alpha + D + d)(D - d)}$$

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

Joints: Member Stiffness

For Members made of Aluminum, hardened steel and cast iron 25 <a< 33
For a= 30

$$k = \frac{0.5774\,\pi E d}{\ln \dfrac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}} \qquad (8-20)$$



If Members have same E with symmetrical frusta (*l*= 2*t*) they act as 2 identical springs $k_m = k/2$ For a= 30° and $D = d_w = 1.5d$

$$k_m = \frac{0.5774\,\pi E d}{2\ln\left(5\,\dfrac{0.5774\,l + 0.5d}{0.5774\,l + 2.5d}\right)} \qquad (8-22)$$

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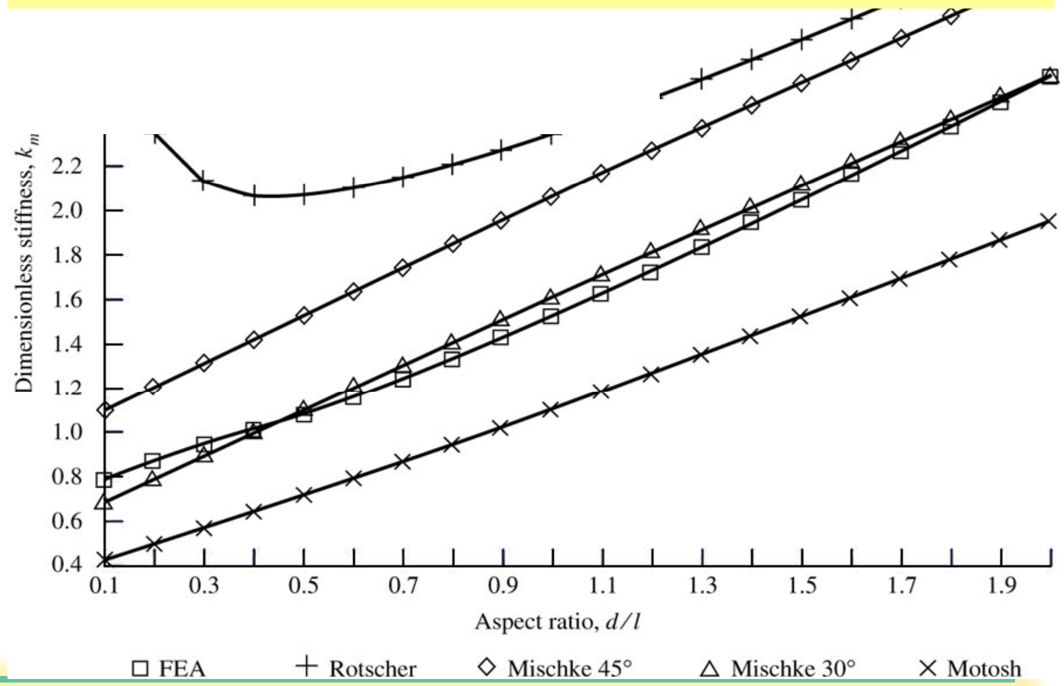
Joints: Member Stiffness
Other equations

From Finite element analysis results, A and B from table 8.8 for standard washer Faces and members Of same material

$$\frac{k_m}{Ed} = A \exp(Bd / l) \qquad (8-23)$$



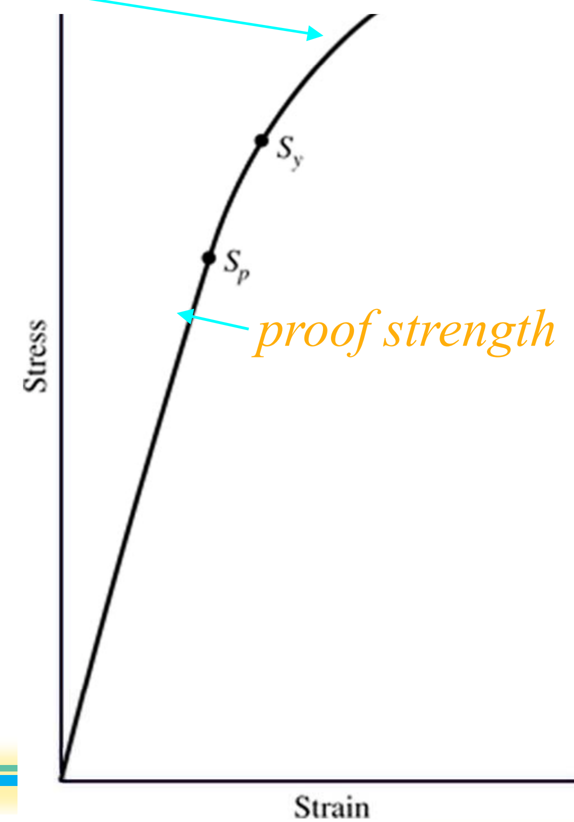
Bolt Strength

Bolt strength is specified by *minimum proof strength* S_p or *minimum proof load,* F_p and *minimum tensile strength, Sut*

The SAE specifications are given in Table 8-9 bolt grades are numbered according to *minimum tensile strength*

The ASTM Specs for steel bolts (structural) are in Table 8-10.

Metric Specs are in table 8-11.

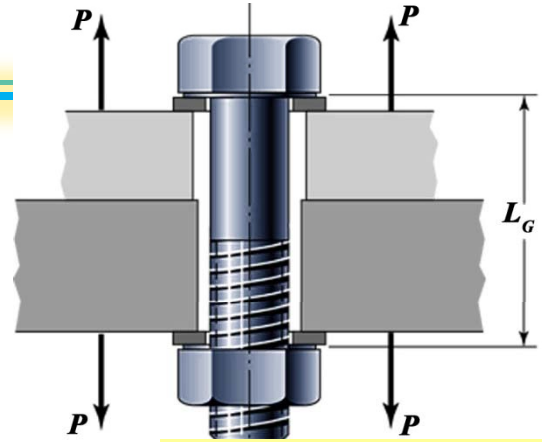


proof strength

If $\boldsymbol{S_p}$ not available use $\boldsymbol{S_p} = 0.85\, \boldsymbol{S_y}$

$\boldsymbol{F_p} = \boldsymbol{A_t}\, \boldsymbol{S_p}$

Tension Joints



Static Analysis

a) External Load

External Load P is shared by bolt and members

Equilibrium $$P = P_b + P_m \quad (1)$$

Compatibility $$\delta = \delta_b = \delta_m \quad (2)$$

Relation P-δ $$\delta = \frac{P_m}{k_m} = \frac{P_b}{k_b} \ \& \ P_m = \frac{P_b k_m}{k_b} \quad (3)$$

$$\left.\begin{aligned} P_b &= \frac{k_b}{k_b + k_m} P = CP \\ P_m &= \frac{k_m}{k_b + k_m} P = (1 - C)P \\ C &= \frac{k_b}{k_b + k_m} \end{aligned}\right\} (4)$$

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

For typical values of C see table 8-12

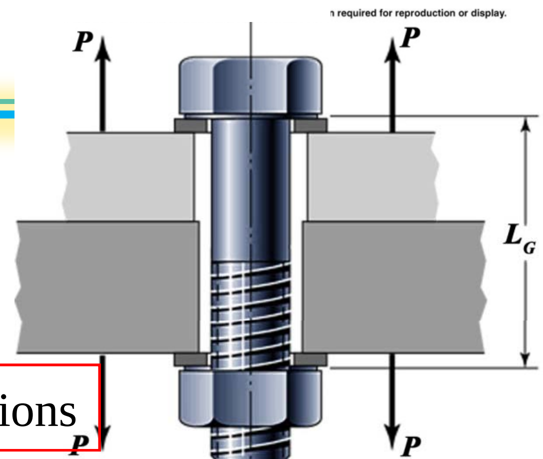
Most of external Load P is taken by members

Tension Joints

Static Analysis

b) Resultant Bolt & member loads: F_b & F_m

$$\left.\begin{aligned} F_b &= P_b + F_i = CP + F_i \\ F_m &= P_m - F_i = (1-C)P - F_i \end{aligned}\right\} F_m<0$$



F_i is preload; high preload is desirable in tension connections

c) Torque required to give preload F_i

F_i = 0.75 F_p *For re-use*
F_i = 0.90 F_p *For permanent joint*

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

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

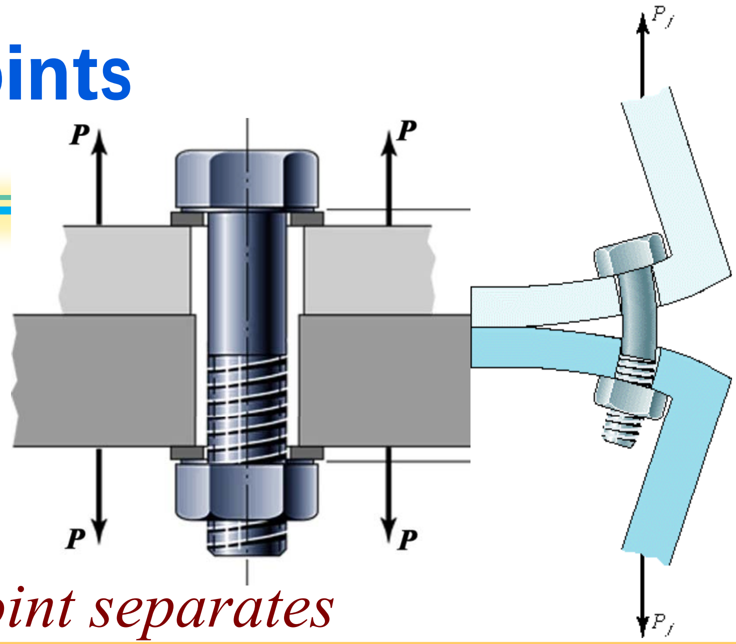
$$d_c = (d + d_w)/2 = (d + 1.5d)/2 = 1.25d$$

$$T = \left[\frac{d_m}{2d}\left(\frac{\tan\lambda + f\sec\alpha}{l - f\tan\lambda\sec\alpha}\right) + 0.625 f_c\right] F_i d$$

$$K = \frac{d_m}{2d}\left(\frac{\tan\lambda + f\sec\alpha}{l - f\tan\lambda\sec\alpha}\right) + 0.625 f_c \qquad T = KF_i d$$

K is torque coefficient
K values are given in table 8-15 (Average Value = 0.2)

Tension Joints



Static Analysis

d) Joint Safety

Failure of Joint occurs when:

1) Bolt yields

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

Failure starts $\boxed{\sigma_b = S_P}$

$$\frac{C(nP)}{A_t} + \frac{F_i}{A_t} = S_P$$

Load factor $\boxed{n = \frac{S_P A_t - F_i}{CP}}$

For N Bolts $n = \dfrac{S_P A_t - F_i}{CP/N}$

or 2) Joint separates

Let P_0 be external load causing separation F_m=0

$$\left.\begin{aligned} F_m &= (1-C)P - F_i \\ F_m &= 0 \end{aligned}\right\} \Rightarrow 0 = (1\text{-}C)\, P_0\text{-}F_i$$

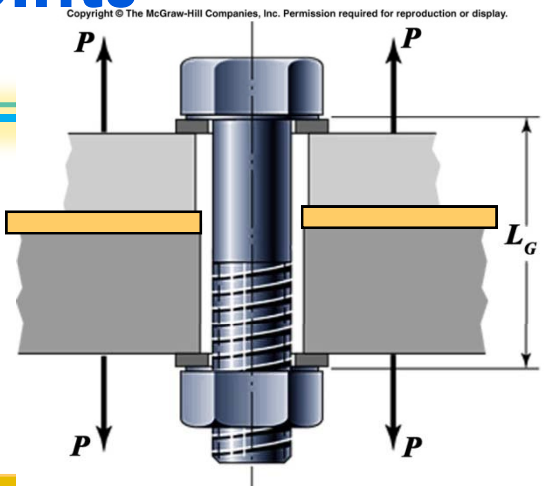
$$\boxed{n = \frac{F_i}{P(1-C)}}$$

For n bolts $n = \dfrac{F_i}{P/N(1-C)}$

Tension Joints

Static Analysis

e) Gasketed Joints



If a full gasket is present in joint
The gasket pressure p is:

$$p = -\frac{F_m}{A_g / N}$$

With load factor n

$$F_m = (1-C)nP - F_i$$

$$p = [F_i - nP(1-C)]\frac{N}{A_g}$$

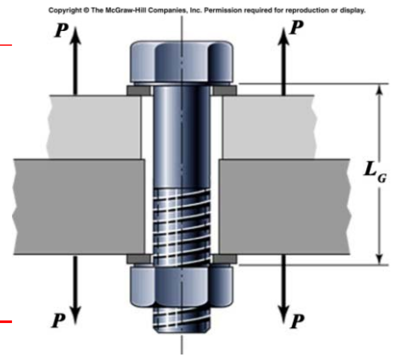
To maintain uniformity of pressure adjacent bolts should not be placed more than 6 nominal diameters apart on bolt circle. To maintain wrench clearance bolts should be placed at least 3 d apart

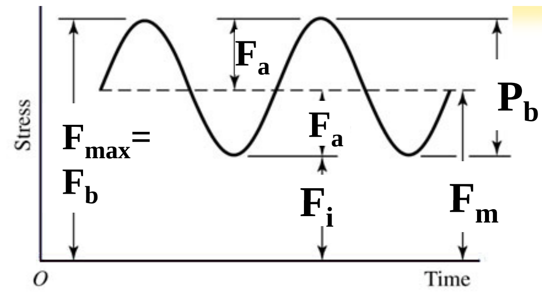
$$3 \le \frac{\pi D_b}{Nd} \le 6$$

D_b is the diameter of the bolt circle

Tension Joints

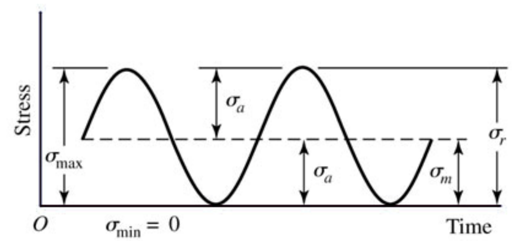
Fatigue **Analysis**





(d)

In general, bolted joints are subject to 0-$P_{max,}$ e.g pressure vessels, flanges, pipes, …



(b)

$$F_{ba} = \frac{F_b - F_i}{2} \Rightarrow \sigma_{ba} = \frac{F_b - F_i}{2A_t} = \frac{(CP + F_i) - F_i}{2A_t} = \frac{CP}{2A_t}$$

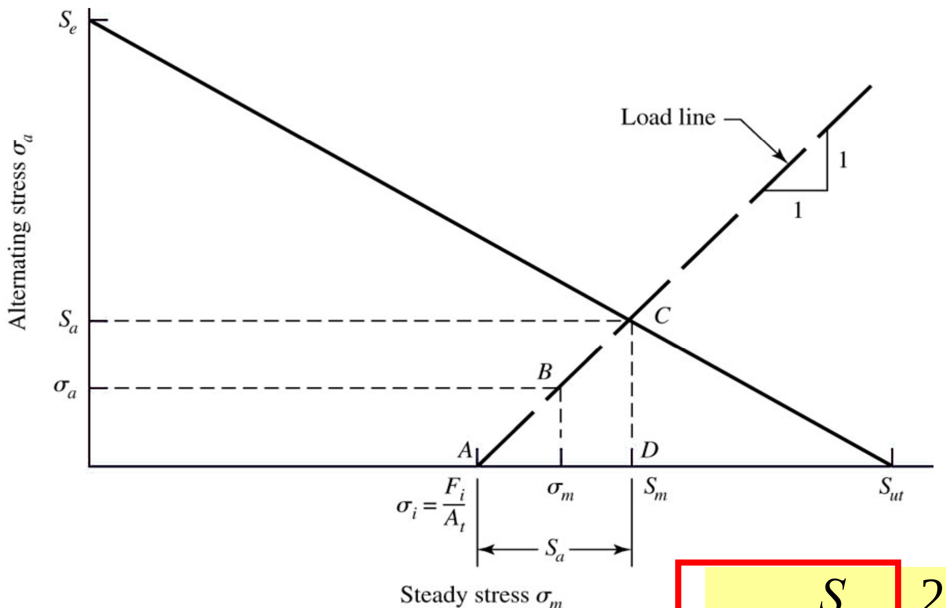
$$F_{bm} = \frac{F_b + F_i}{2} \Rightarrow \sigma_{bm} = \frac{F_b + F_i}{2A_t} = \frac{(CP + F_i) + F_i}{2A_t} = \frac{CP}{2A_t} + \frac{F_i}{A_t} = \sigma_{ba} + \sigma_i$$

(c) (f)

Tension Joints

Fatigue Analysis

High Preload is especially important in fatigue. σ_i *is a constant the load line at* F_i/A_t *has a unit slope, r=1.0*



To find s_a use Goodman

$$\frac{S_a}{S_e} + \frac{S_m}{S_u} = 1$$

for conservative assessment of n:

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

or Gerber Eq. 8.42
or ASME-elliptic. Eq. 8.43

Safety factor (using Goodman)

$$\boxed{n_f = \frac{S_a}{\sigma_i}} = \frac{2S_e(S_{ut}A_t - F_i)}{CP(S_{ut} + S_e)} \quad with \quad \sigma_i = \frac{F_i}{A_t}$$

Check for yielding also using proof strength: Eq. 8.48

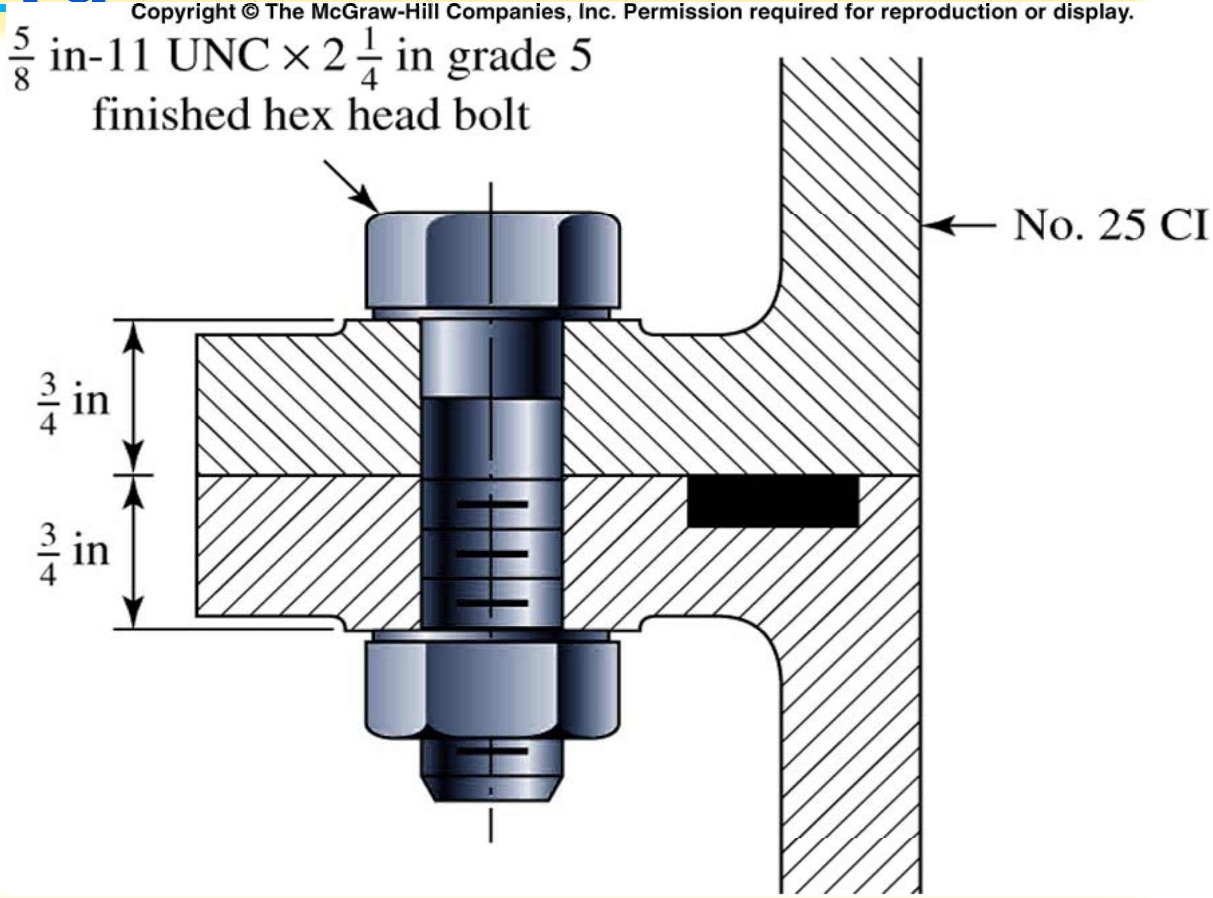
Tension Joints

Fatigue Analysis

In case of ***cut threads*** *use the method described in chapter 7 with* K_f *values of table 8-16.*

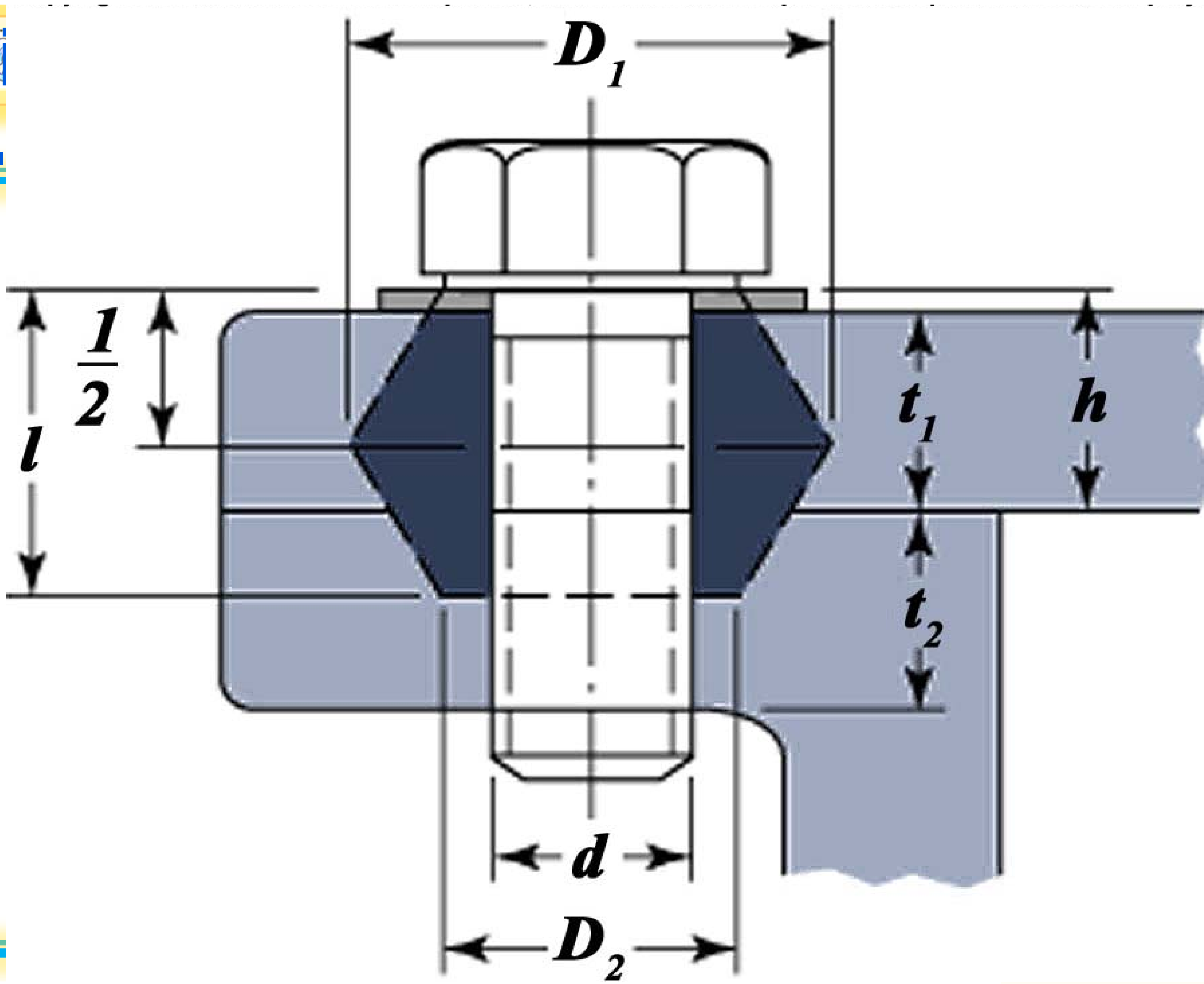
The fully corrected endurance limit for ***rolled threads*** *is given in table 8-17*

Fig. 8.19

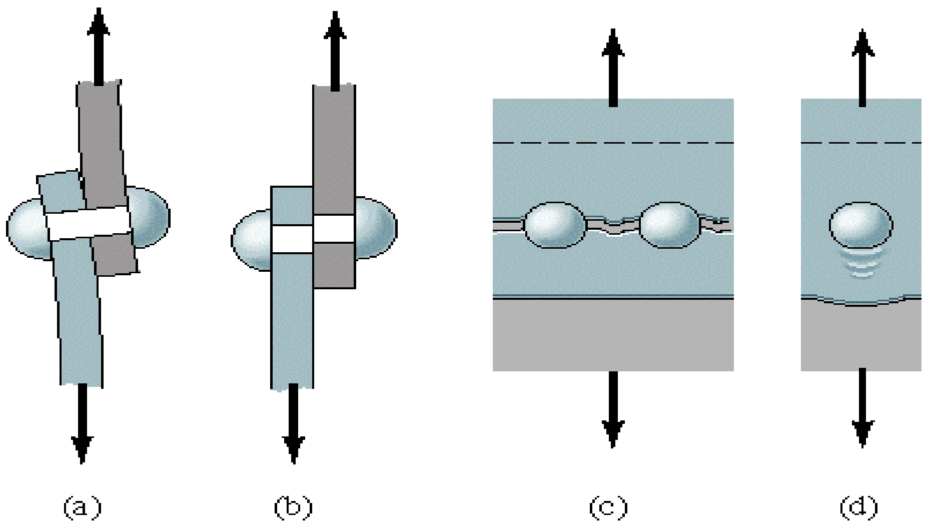

$\frac{5}{8}$ in-11 UNC × 2 $\frac{1}{4}$ in grade 5
finished hex head bolt
No. 25 CI
$\frac{3}{4}$ in
$\frac{3}{4}$ in

F

8.



Failure Modes of Riveted Fasteners Under Shear



Failure modes due to shear loading of riveted fasteners. (a) Bending of member; (b) shear of rivet; (c) tensile failure of member; (e) bearing of rivet on member or bearing of member on rivet.

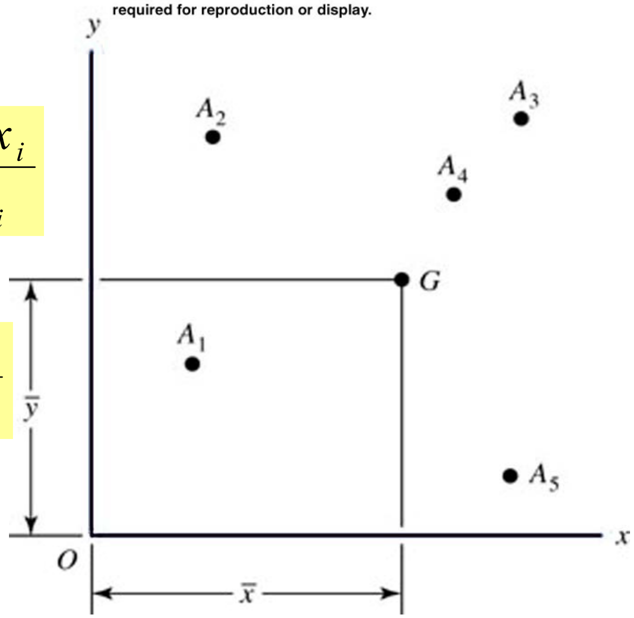
ME 307 Machine Design I

Shear Joints

Centroid of pins, rivets or bolts

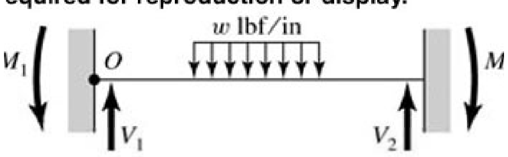
$$x' = \frac{A_1x_1 + A_2x_2 + A_3x_3 + A_4x_4 + A_5x_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\Sigma A_i x_i}{\Sigma A_i}$$

$$y' = \frac{A_1y_1 + A_2y_2 + A_3y_3 + A_4y_4 + A_5y_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\Sigma A_i y_i}{\Sigma A_i}$$



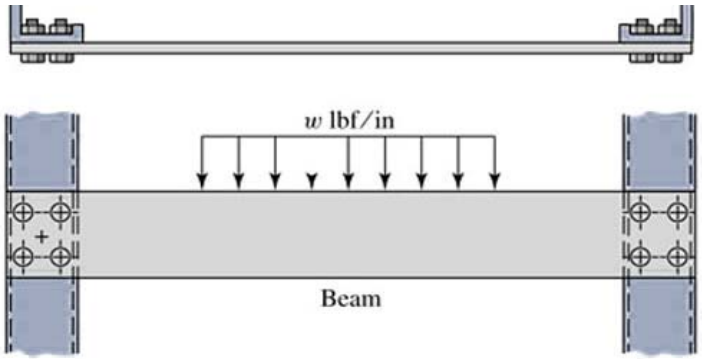
Shear Joints

Shear of pins, rivets bolts due to eccentric loading



(b)

V & M statically indeterminate problem
4 steps (assuming same diameter bolts, load shared equally)



(a)

1) Direct load F’

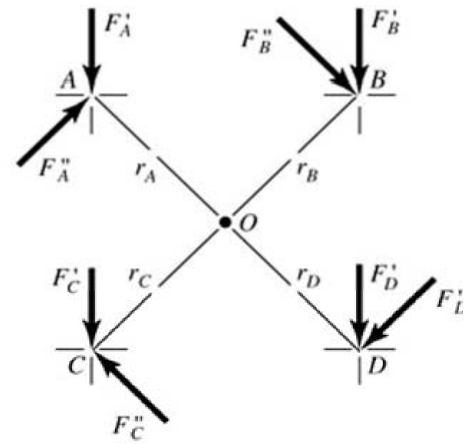
Primary Shear

$$F' = \frac{V}{N}$$

2) Centroid

$$x' = \frac{\Sigma A_i x_i}{\Sigma A_i}$$

$$y' = \frac{\Sigma A_i y_i}{\Sigma A_i}$$

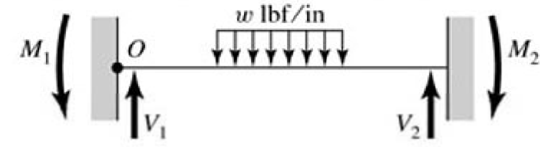


(c)

Shear Joints

Shear of pins, rivets bolts due to eccentric loading

3) Moment load M



(b)

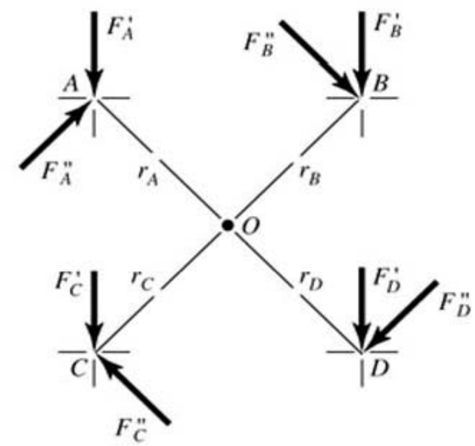
Secondary Load

$$M_1 = F_A{}'' r_A + F_B{}'' r_B + F_C{}'' r_C + F_D{}'' r_D$$

The force taken by each bolt is proportional to its radial distance from the centroid

$$\frac{F_A{}''}{r_A} = \frac{F_B{}''}{r_B} = \frac{F_C{}''}{r_C} = \frac{F_D{}''}{r_D}$$

$$F_n{}'' = \frac{M_1 r_n}{r_A^2 + r_B^2 + r_C^2 + \ldots}$$

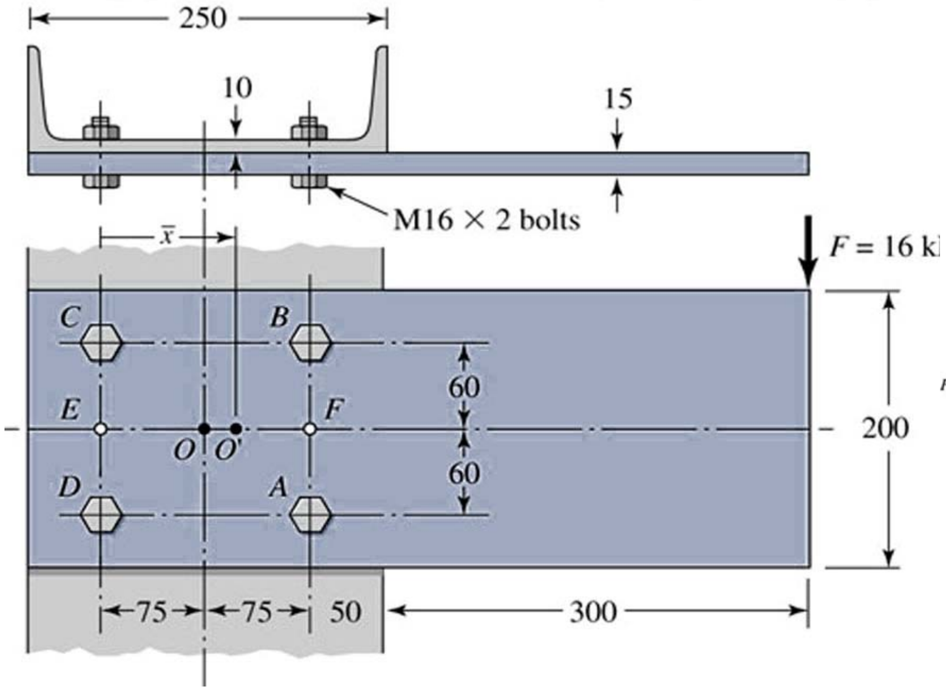


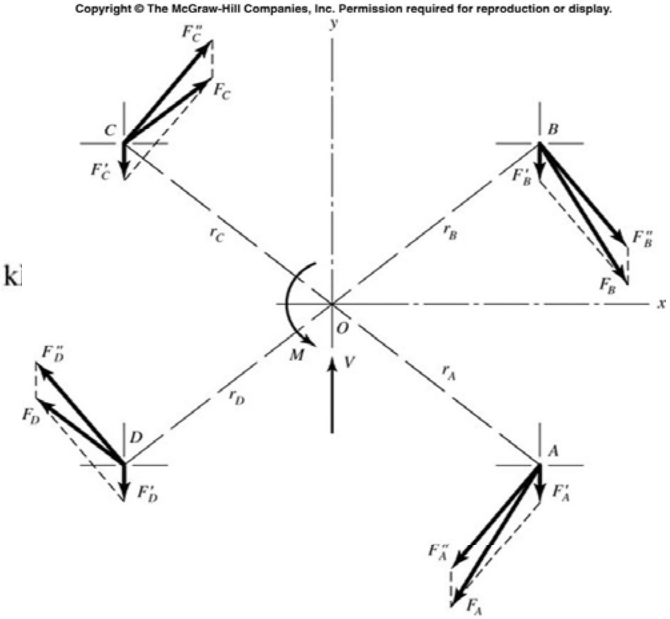
(c)

4) Add Vectorially the direct and moment loads

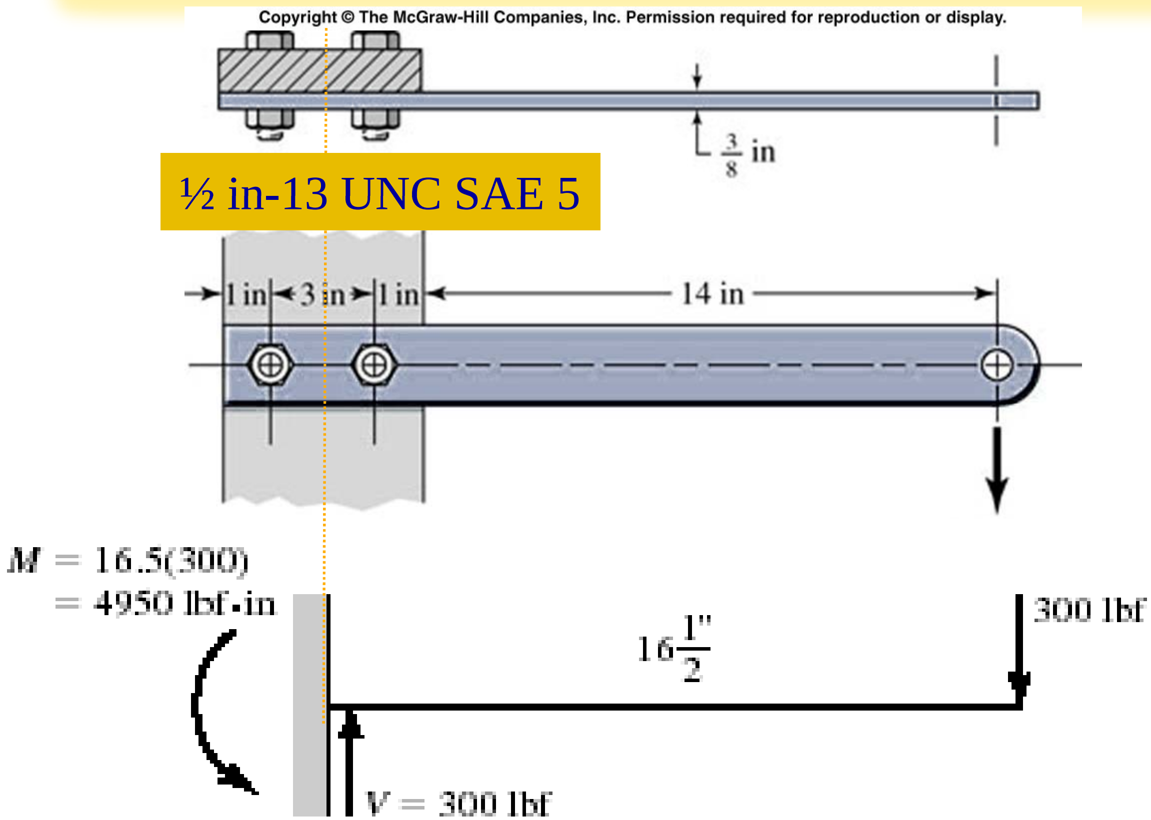
If bolts are not same size only bolts with max. R should be considered

See Examples 8.6 and 8.7





Problem 8-50



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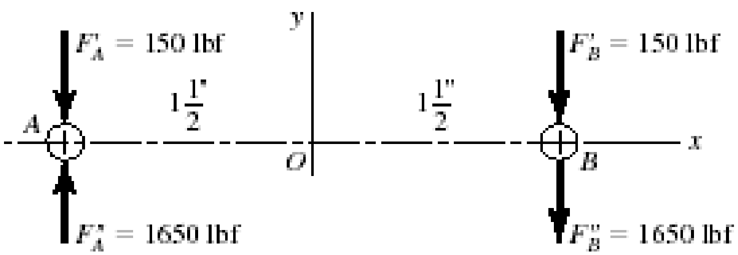
½ in-13 UNC SAE 5
3/8 in
1 in
3 in
1 in
14 in
M = 16.5(300)
= 4950 lbf·in
16 1"/2
300 lbf
V = 300 lbf

Problem 8-50

1) Direct load F'

$$F' = \frac{V}{N} = \frac{300}{2} = 150 \quad lb$$



2) Centroid is at O

3) Secondary Load

$$F_A'' = F_B'' = \frac{M_1 r_A}{r_A^2 + r_B^2} = \frac{4950 \quad x1.5}{1.5^2 + 1.5^2} = 1650 \quad lb$$

4)Add Vectorially the direct and moment loads

$$F_A = 1650 - 150 = 1500 \quad lb$$

$$F_B = -1650 - 150 = -1800 \quad lb$$

Shear of bolt:

$$A_s = \frac{\pi}{4}\boxed{(0.5)}^2 = 0.1963 \text{ in}^2$$

$$\tau = \frac{F}{A} = \frac{1800}{0.1963} = 9170 \text{ psi}$$

$$S_{sy} = 0.577\boxed{(92)} = 53.08 \text{ kpsi}$$

$$n = \frac{53.08}{9.17} = 5.79 \quad Ans.$$

Is As=Ad?

L_T=2d+1/4=1.25”

H_{nut}+2p+3/8=7/16+2(1/13)+3/8=0.97”<1.25”

So As=At = 0.1419in^2 and t_p=12.7 Kpsi;
n=4.2

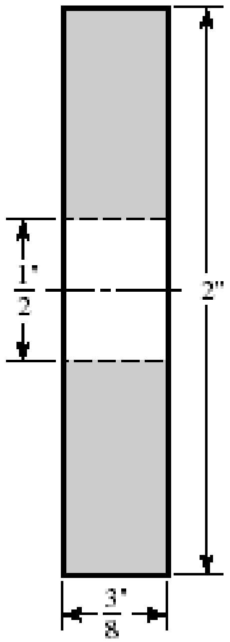
Problem 8-50

Bearing on bolt:

$$A_b = \frac{1}{2}\left(\frac{3}{8}\right) = 0.1875 \text{ in}^2$$

$$\sigma = -\frac{F}{A} = -\frac{1800}{0.1875} = -9600 \text{ psi}$$

$$n = \frac{92}{9.6} = 9.58 \quad Ans.$$



Bearing on members: $S_y = 54 \text{ kpsi}, n = \dfrac{54}{9.6} = 5.63 \quad Ans.$

Problem 8-50

Strength of members: Considering the right-hand bolt

$$M = 300(15) = 4500 \text{ lbf} \cdot \text{in}$$

$$I = \frac{0.375(2)^3}{12} - \frac{0.375(0.5)^3}{12} = 0.246 \text{ in}^4$$

$$\sigma = \frac{Mc}{I} = \frac{4500(1)}{0.246} = 18\,300 \text{ psi}$$

$$n = \frac{54(10)^3}{18\,300} = 2.95 \quad \textit{Ans.}$$