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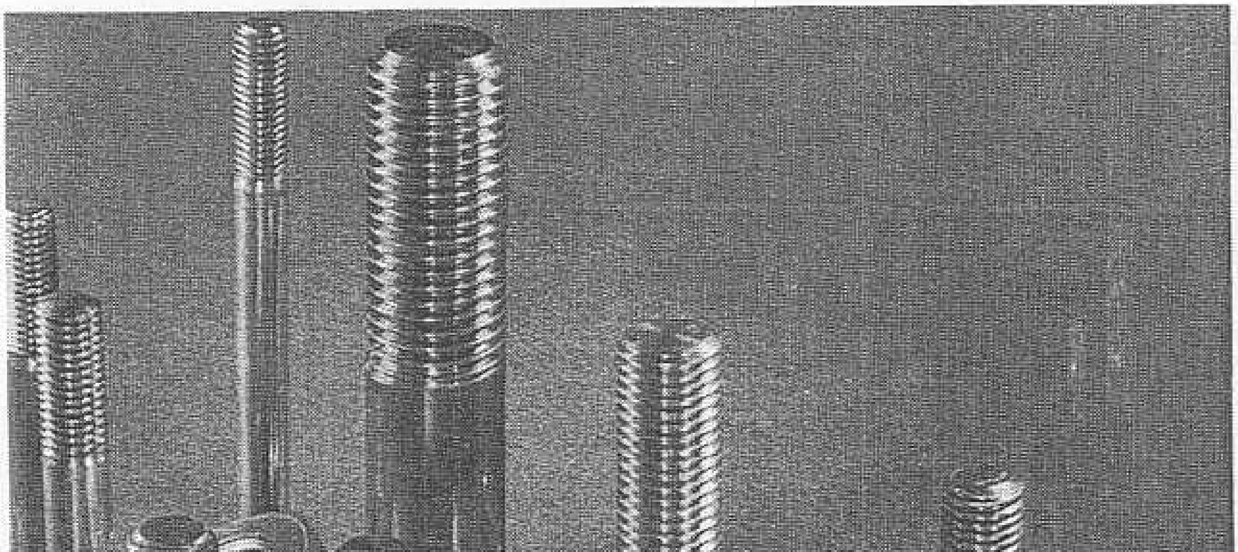
Screws, Fasteners, etc

Chapter 8



A collection of threaded fasteners. (Courtesy of Clark Craft Fasteners)

CHAPTER 8

SCREWS, FASTENERS, CONNECTIONS

1. Thread geometry
2. Bolt strength
3. Mechanics of Screws
4. Preloaded joints ... AXIAL LOAD
   - Preload forces
   - Total forces
   - Bolt & joint stiffness
   - Static failure
   - Fatigue failure

5 BOLTED & Riveted joints in SHEAR

- Failure modes

- Eccentric loading

6 MISC. Components

- keys, pins, retaining rings

- ~~set screws~~

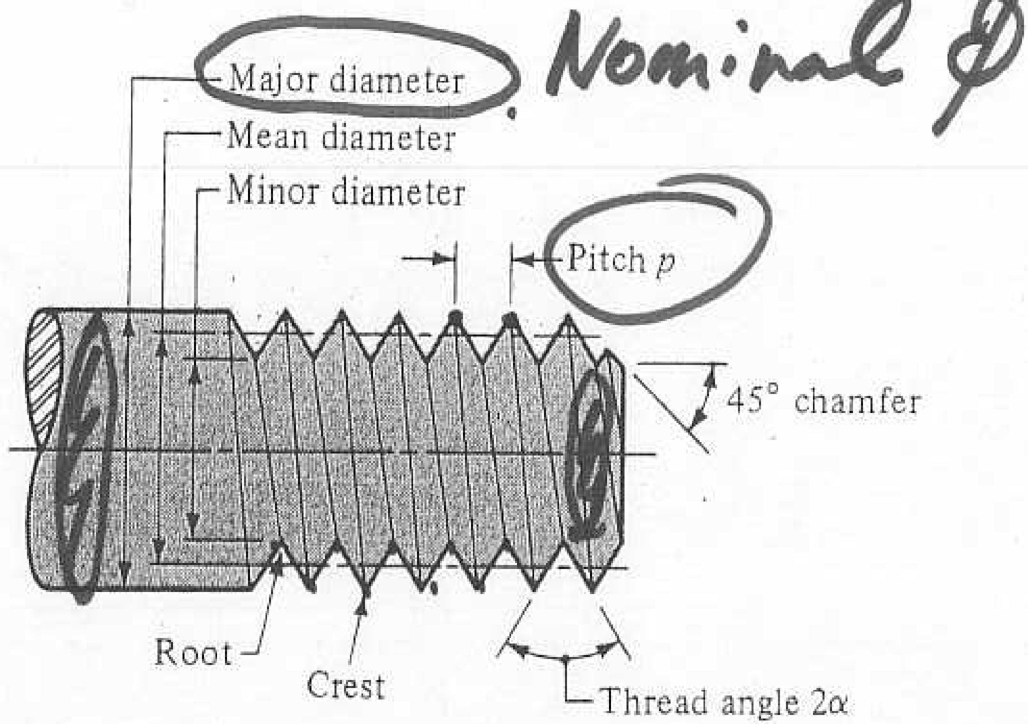


FIGURE 8-1

Terminology of screw threads. Sharp vee threads shown for clarity; the crests and roots are actually flattened or rounded during the forming operation.

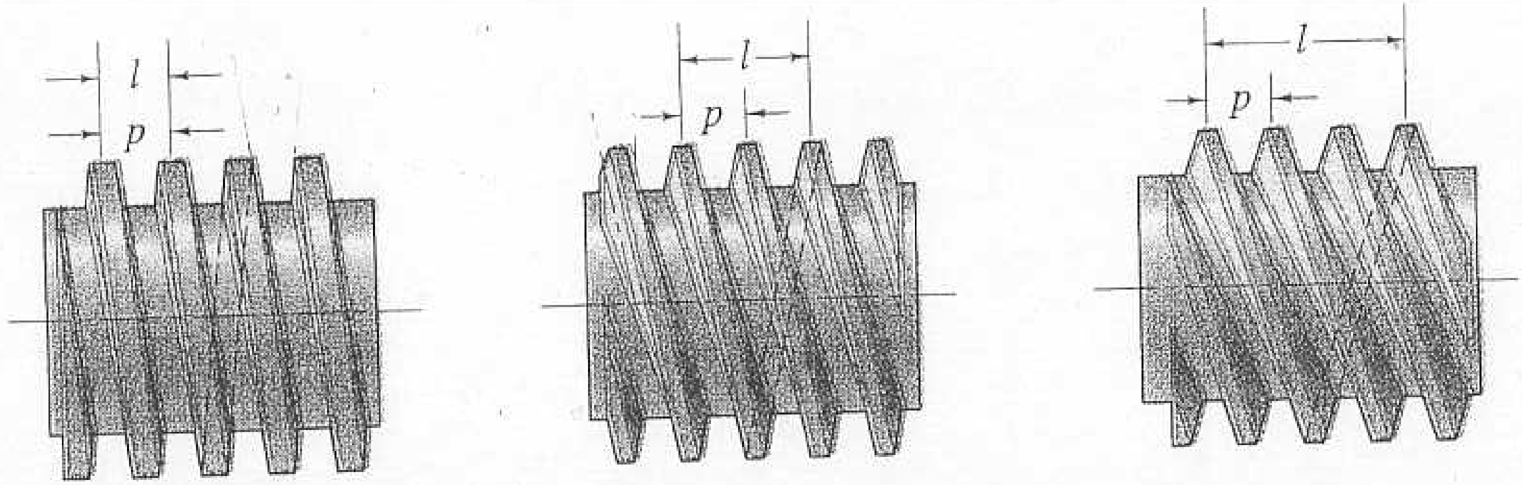
Some terminology

- pitch, p
- major (nominal) diameter, D, d

  Internal Thread (nut) → D

  External Thread (bolt) → d

- lead, l ... distance moved per turn
- Threads are normally right-handed
- American National (Unified) & Metric Threads



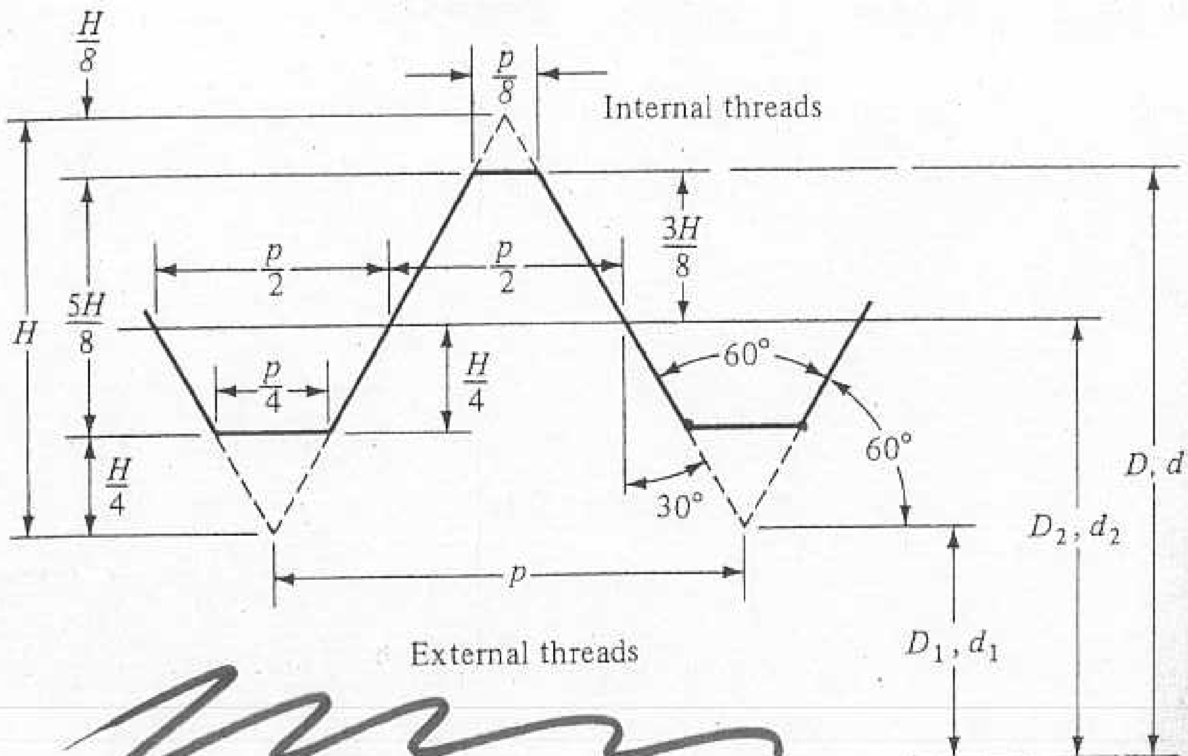
(a) (b) (c)

Single lead — $l = p$

double — $l = 2p$

triple — $l = 3p$

NUT



BOLT

FIGURE 8-2

ısic thread profile for metric M and MJ threads. D (d) = basic jor diameter of internal (exter- thread; D_1 (d_1) = basic minor diameter of internal (external) ıd; D_2 (d_2) = basic pitch diam- ɜr of internal (external) thread; p = pitch; $H = 0.5(3)^{1/2}\,p$.

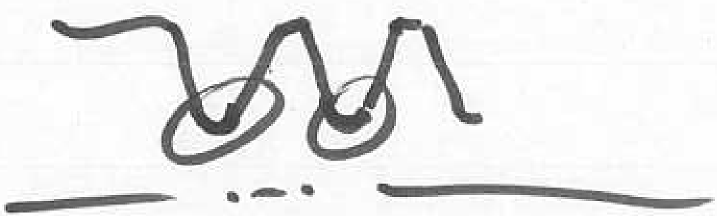


TABLE 8-2

Diameters and Area of Unified Screw Threads UNC and UNF*

e.g. 1/2" - 13UNC

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

*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.

ANSI Standards

NOMINAL MAJOR DIAMETER d	COARSE-PITCH SERIES PITCH p	TENSILE-STRESS AREA A_t	MINOR-DIAMETER AREA A_r	FINE-PITCH SERIES PITCH p	TENSILE-STRESS AREA A_t	MINOR-DIAMETER AREA A_r
mm 1.6	mm 0.35	mm^2 1.27	mm^2 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

*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.

COARSE

FINE

Metric Threads

e.g. M12×1.75

ϕ

pitch

TENSILE AREA

A_t

164

165

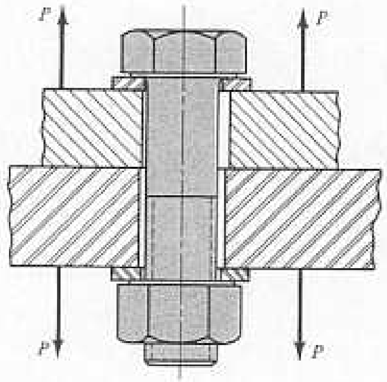


FIGURE 8-12

A bolted connection loaded in tension by the forces P. Note the use of two washers. A simplified conventional method is used here to represent the screw threads. Note how the threads extend into the body of the connection. This is usual and is desired.

Axial Loading

The *spring constant*, or *stiffness constant*, of an elastic member such as a bolt, as we learned in Chap. 3, is the ratio between the force applied to the member and the deflection produced by that force. We can use Eq. (3-4) and the results of Prob. 3-1 to find the stiffness constant of a fastener in any bolted connection.

The *grip* of a connection is the total thickness of the clamped material. In Fig. 8-12 the grip is the sum of the thicknesses of both members and both washers. In Fig. 8-13 the grip is the thickness of the top member plus that of the washer.

The stiffness of the portion of a bolt or screw within the clamped zone will generally consist of two parts, that of the unthreaded shank portion and that of the threaded portion. Thus the stiffness constant of the bolt is equivalent to the stiffnesses of two springs in series. Using the results of Prob. 3-1, we find

$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2} \qquad \text{or} \qquad k = \frac{k_1 k_2}{k_1 + k_2} \tag{8-9}$$

for two springs in series. From Eq. (3-4), the spring rates of the threaded and unthreaded portions of the bolt in the clamped zone are, respectively,

$$k_T = \frac{A_t E}{l_T} \qquad k_d = \frac{A_d E}{l_d} \tag{8-10}$$

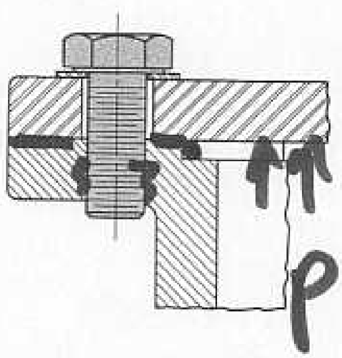


FIGURE 8-13

Section of a cylindrical pressure vessel. Hexagon-head cap screws are used to fasten the cylinder head to the body. Note the use of the O-ring seal.

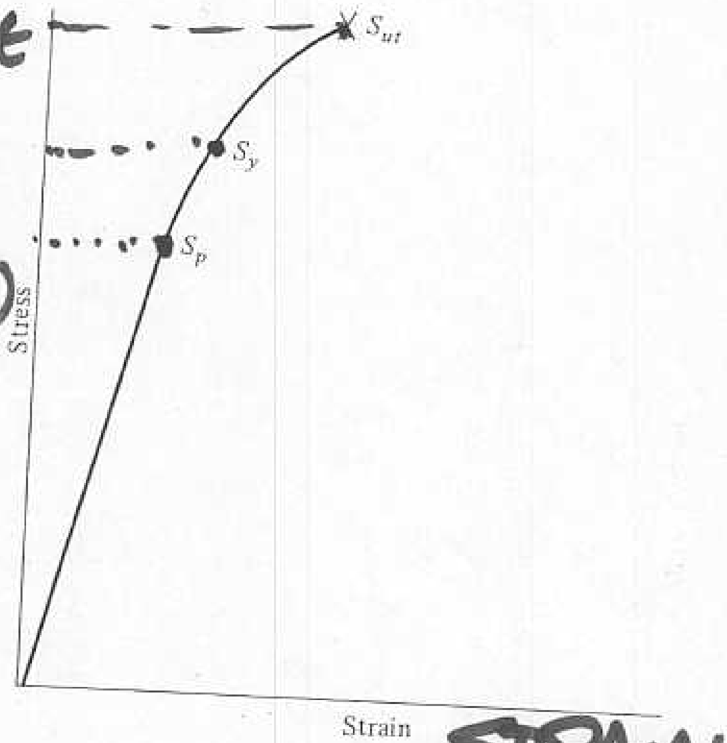
Axial Loading

STRESS

S_{ut}

S_y

S_p



STRAIN

FIGURE 8-15

Typical stress-strain diagram for bolt materials showing proof strength S_p, yield strength S_y, and tensile strength S_{ut}.

Ultimate Strength

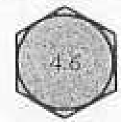
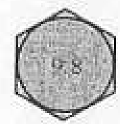
Yield Strength

Proof Strength ... use in place of yield strength for Bolts

TABLE 8-6

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

METRIC

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	
4.8	M1.6–M16	310	420	340	Low or medium carbon	
5.8	M5–M24	380	520	420	Low or medium carbon	
8.8	M16–M36	600	830	660	Medium carbon, Q&T	
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	

*The thread length for bolts and cap screws is

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

where L is the bolt length. The thread length for structural bolts is slightly shorter than given above.

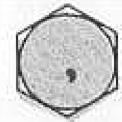
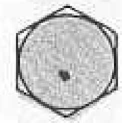
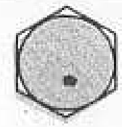
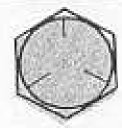
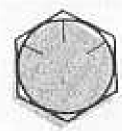
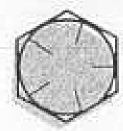
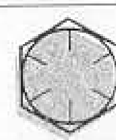
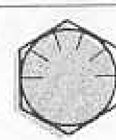
TABLE 8-4
SAE Specifications for Steel Bolts

STRENGTH OF BOLTS

SAE

GRADE

Don't use

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	

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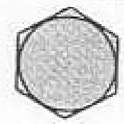
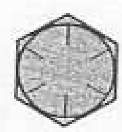
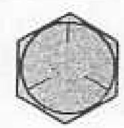
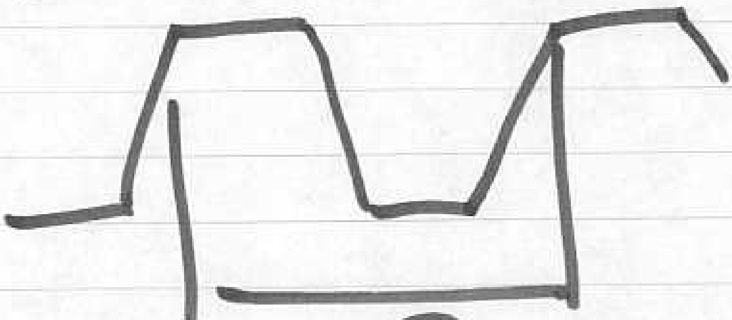
ASTM

TABLE 8-5
ASTM Specifications for Steel Bolts

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 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Medium carbon, Q&T	
A325, type 2	$\frac{1}{2}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Low-carbon martensite, Q&T	
A325, type 3	$\frac{1}{2}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$	85 74	120 105	92 81	Weathering steel, Q&T	
A354, grade BC					Alloy-steel, Q&T	
A354, grade BD	$\frac{1}{4}$–4	120	150	130	Alloy steel, Q&T	
A449	$\frac{1}{4}$–1 $1\frac{1}{8}$–$1\frac{1}{2}$ $1\frac{3}{4}$–3	85 74 55	120 105 90	92 81 58	Medium-carbon, Q&T	
A490, type 1	$\frac{1}{2}$–$1\frac{1}{2}$	120	150	130	Alloy steel, Q&T	
A490, type 3					Weathering steel, Q&T	

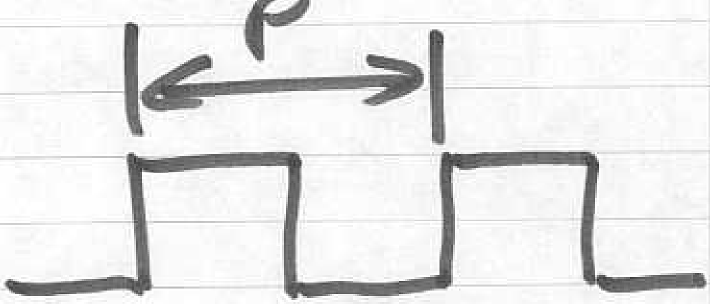
Other threads

Acme



P

Square



Mechanics of Threads... square thread

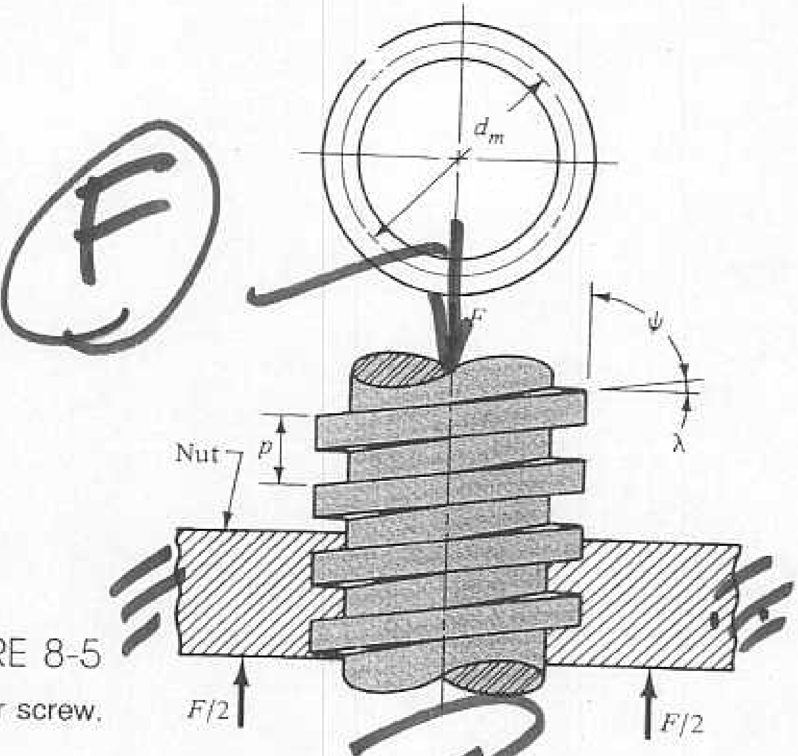


FIGURE 8-5
Portion of a power screw.

$$\frac{F}{2} + \frac{F}{2}$$

Advance screw against load F.

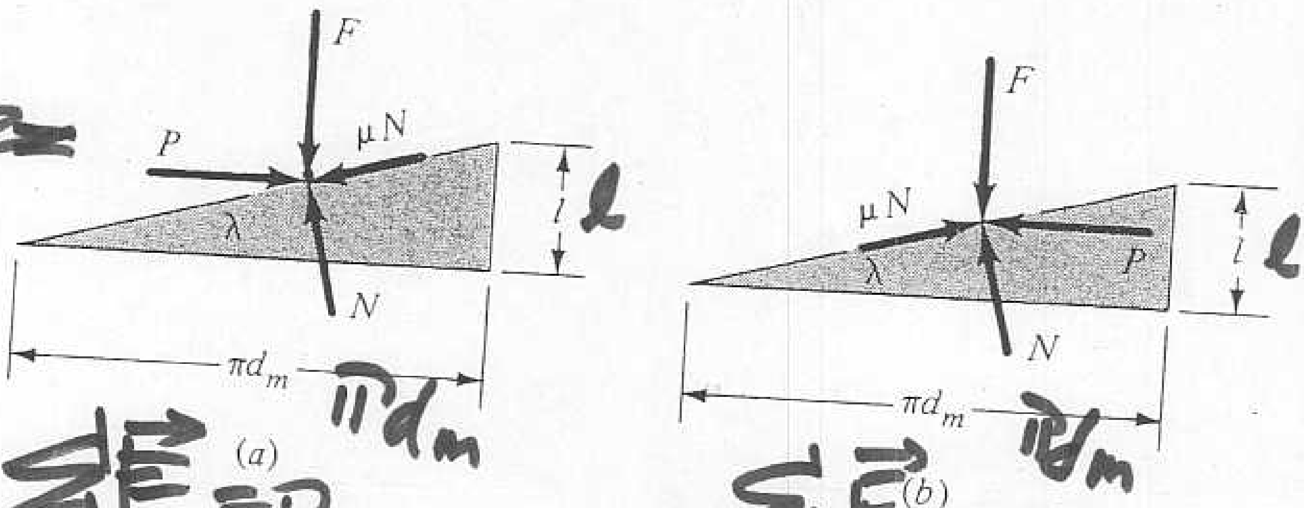
or

Retract

FIGURE 8-6

Force diagrams: (*a*) lifting the load; (*b*) lowering the load.

"Raising" "Lowering"



$\sum \vec{F} = 0$ $\sum \vec{F} = 0$

Mechanics of Screws

Torque to raise or lower $T = P\frac{d_m}{2}$ ← mean diam.

Raising Load

H ⇒ Horizontal
V ⇒ Vertical

$$\sum F_H = P - N\sin\lambda - \mu N\cos\lambda = 0$$

$$\sum F_V = F + \mu N\sin\lambda - N\cos\lambda = 0$$

Eliminate N & Solve for P:

$$P = \frac{F(\sin\lambda + \mu\cos\lambda)}{(\cos\lambda - \mu\sin\lambda)}$$

$$= \frac{F\left(\frac{\sin\lambda}{\cos\lambda} + \mu\right)}{\left(1 - \mu\frac{\sin\lambda}{\cos\lambda}\right)}$$

$$\frac{\sin\lambda}{\cos\lambda} = \tan\lambda = \frac{l \rightarrow \text{lead}}{\pi d_m \rightarrow \text{circumfer. (mean)}}$$

$$\therefore \quad P = F\left(\frac{l/\pi d_m + \mu}{1 - \mu l/\pi d_m}\right)$$

Lower Load

... same process leads to

$$P = F\,\frac{(\mu - l/\pi d_m)}{(1 + \mu l/\pi d_m)}$$

Torque req'd:

Raise	Lower
$T = \frac{F d_m}{2}(\quad)$	$T = \frac{F d_m}{2}(\quad)$
$= \frac{F d_m}{2}\left(\frac{l/\pi d_m + \mu}{1 - \mu l/\pi d_m}\right)$	$= \frac{F d_m}{2}\left(\frac{\mu - l/\pi d_m}{1 + \mu l/\pi d_m}\right)$

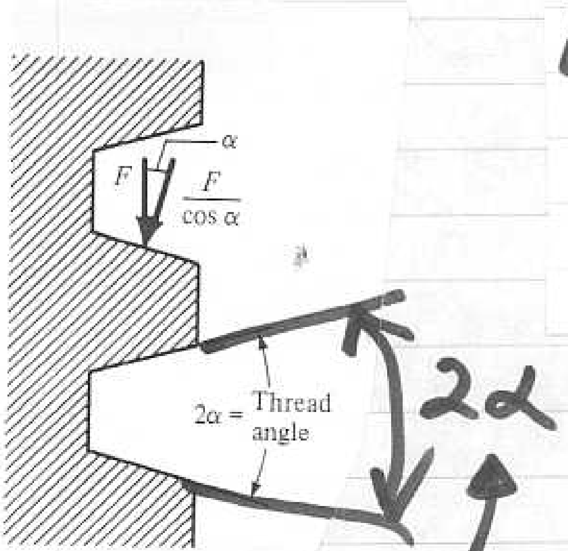
if $\mu > \dfrac{l}{\pi d_m}$

or $\mu \pi d_m > l$

∴ Lowering torque is $\underline{+ve}$

∴ Screw/Thread is "<u>SELF-LOCKING</u>"

<u>NON-SQUARE THREADS</u>



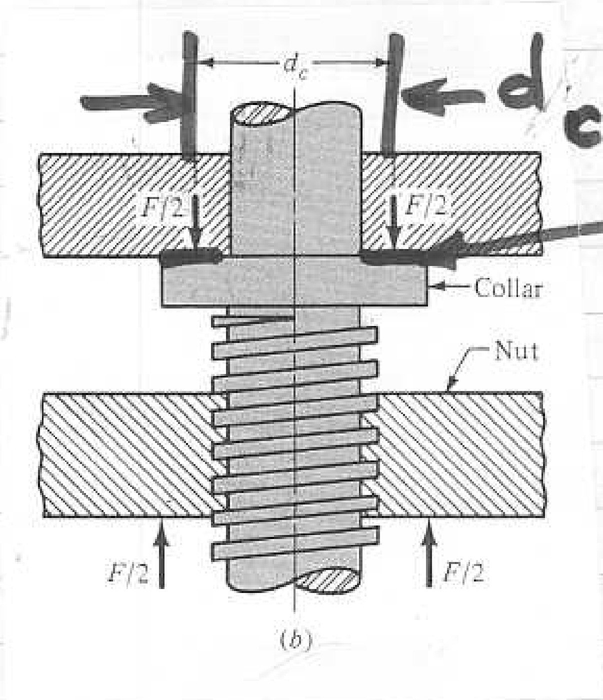
(a)

THREAD ∡

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

RAISING TORQUE

THERE IS ALSO FRICTION AT THE COLLAR/WASHER



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

Typically, $d_c = \frac{5d}{4}$

$\therefore$ Total torque is:

$$T = F_i d \left[\frac{\tan\lambda + \mu \sec\alpha}{1 - \mu \tan\lambda \sec\alpha} + \frac{5}{8}\mu_c \right] \left[\frac{d_m}{2d}\right] \quad \ldots 8\text{-}19$$

F_i is initial load on a screw or bolt

∴ Normally can set K so That

$$\underline{\underline{T = K F_i d}}$$

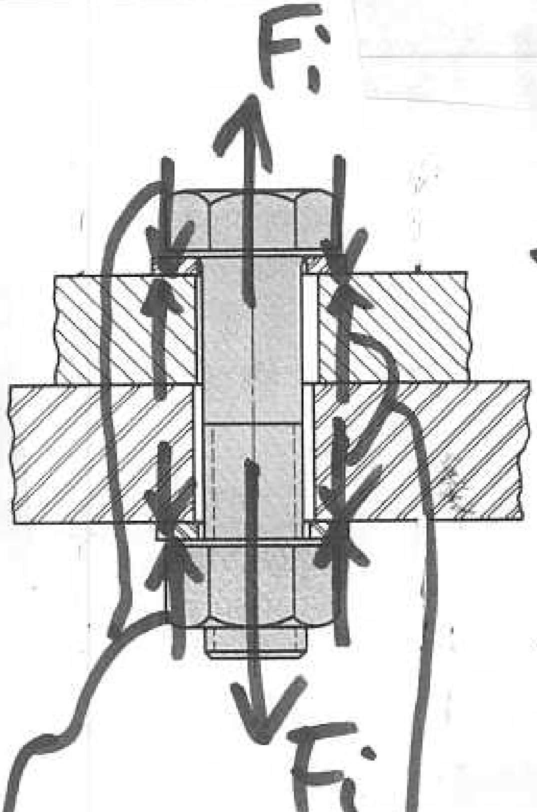
← d is nominal major dia.

∴ see table 8-10

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 Bowma-Grip nuts	0.09

- Std 60° Thread angle
- Various surface treatments
- WHAT SHOULD F_i be??

UNDERSTANDING LOAD & PRELOAD



- Tighten bolt to preload F_i
- Bolt in tension
- mat'l or grip in compression

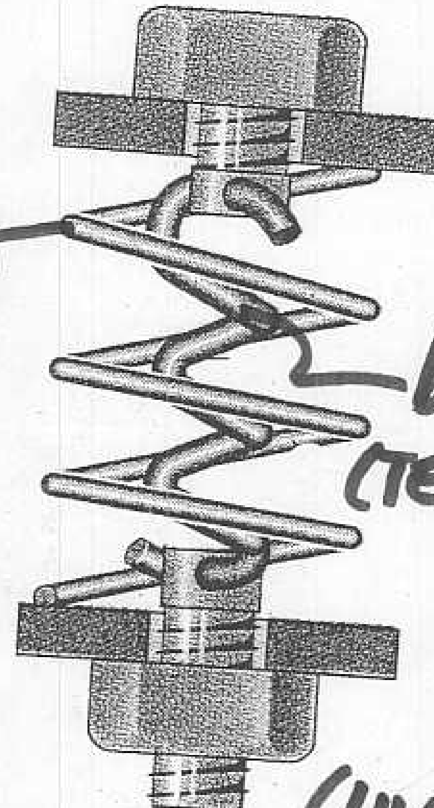
bolt pressing against mtl

mtl pressing against bolt

NO EXTERNAL LOAD

... YET

MATL & BOLT STIFFNESS



Material or Grip (COMPRESSION)

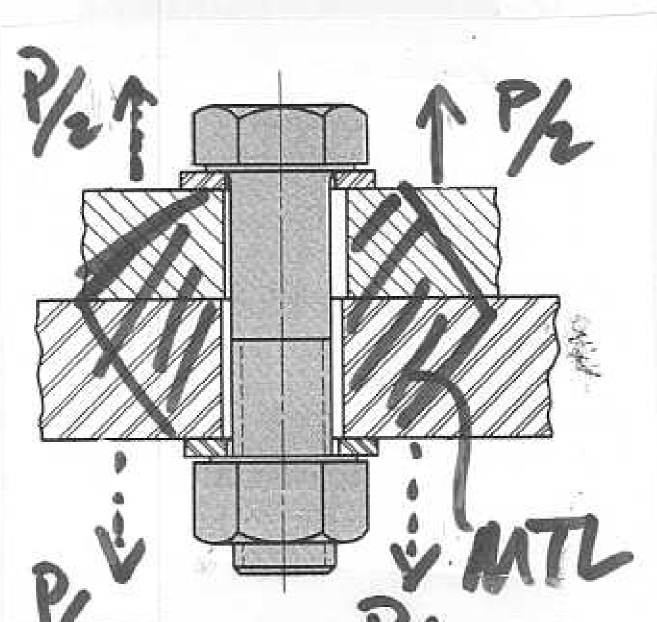
k_m

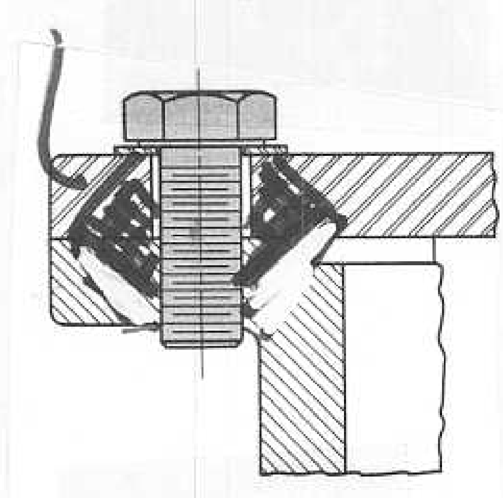
bolt (TENSION)

k_b : (LIKE ROD)

External load, P

MTL

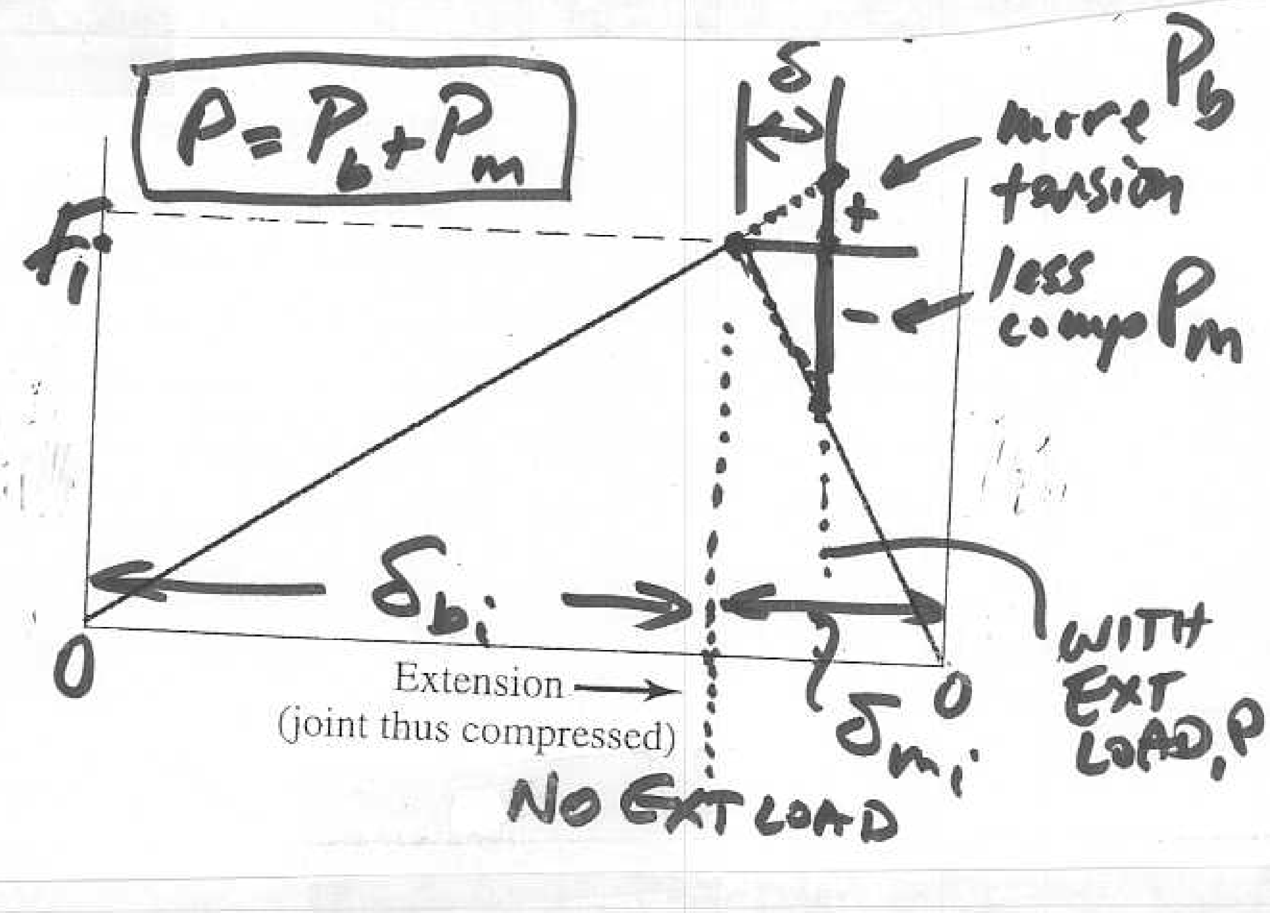




$\Longrightarrow$ k_b & k_m will be Given (in ARE 311)

When external load, P is applied .... Tension:

1. Bolt gets more tension
2. Material sees less compression.



$F_i \rightarrow$ initial preload

$P \rightarrow$ external load

$$F_b = F_i + P_b$$

$$F_m = P_m - F_i$$

$$P_m = k_m \delta \qquad P_b = k_b \delta$$

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

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

But $P = P_m + P_b$

& $P_m = P - P_b$

$\therefore P_b = \frac{k_b}{k_m} P_m = \frac{k_b}{k_m}(P - P_b)$

and $P_b = \frac{k_b P}{k_b + k_m}$

also $P_m = \frac{k_m}{k_b + k_m} P$

Note $k_m > k_b$ (usually)

Total load on bolt:

$F_b = F_i + P_b = \frac{k_b P}{k_b + k_m} + F_i$

$F_m = \frac{k_m P}{k_b + k_m} - F_i$ .... for contact $F_m < 0$ **

EXAMPLE

A 1/2" UNC SAE Grade 7 bolt grips two plates of steel. The grip is 2". The recommended preload is 13,000 lbs.

1. What will be the load on the bolt and the grip (material) if an external load of 10,000 lbs is applied?

$$F_b = F_i + \frac{k_b P}{k_b + k_m}$$

From Table 8-7 $k_b = 2.57$ Mlbs/in
$k_m = 12.69$ Mlbs/in

$$F_b = F_i + \frac{2.57P}{15.26} = 13000 + .168(10000)$$

$F_b = 14,680 \text{ lb}$ i.e. $F_b \uparrow$ by 1680 lbs

$$F_m = \frac{k_m P}{k_m + k_b} - F_i$$

$$= \frac{12.69}{15.26}(10000) - 13,000$$

$$\boxed{F_m = 8320 - 13,000 = -4680 \text{ lbs}}$$

$F_m \downarrow$ by 8320 lbs

2. At what value of external load will the joint separate?

THIS WILL OCCUR WHEN $F_m = 0$

$$+\frac{k_m P}{k_m + k_b} - F_i = 0$$

$$P = \frac{(k_b + k_m) F_i}{k_m} = \frac{1 \times 13,000}{0.832}$$

$\boxed{@ \text{ Separation } P = 15,625}$ $\therefore$ more than F_i

STATIC Failure

$$F_b = \frac{k_b P}{k_b + k_m} + F_i = CP + F_i$$

Stress on bolt:

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

"Tensile Area"

Allowing for a factor of Safety on "P" $\Rightarrow nP$

$$\sigma_b = \frac{CnP}{A_t} + \frac{F_i}{A_t}$$

i.e. no F.S. on F_i

at "failure" $\sigma_b = S_p$

Solve for "n" Proof Strength

$$\boxed{n = \frac{S_p A_t - F_i}{CP}}$$

... Also .... The "Proof Load"

$F_p = S_p A_t$

... The recommended preload on bolted connections is

$F_i = 0.75\, F_p = 0.75\, S_p A_t$

.... "Reusable"

OR $F_i = 0.90\, F_p = 0.90\, S_p A_t$

... "Permanent"

CONT'D EXAMPLE

... $\frac{1}{2}$" UNC Grade 7 bolt

... $F_i = 13{,}000$ lbs

... What is the factor of safety for bolt failure at $P_1 = 10{,}000$ lbs and at $P_2 = 15{,}625$ lbs

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

$$n_1 = \frac{107\,\text{kpsi}\,(0.142\,\text{in}^2) - 13{,}000}{(0.168)(10{,}000)}$$

$\boxed{n_1 = 1.31}$ @ 10,000 lbs

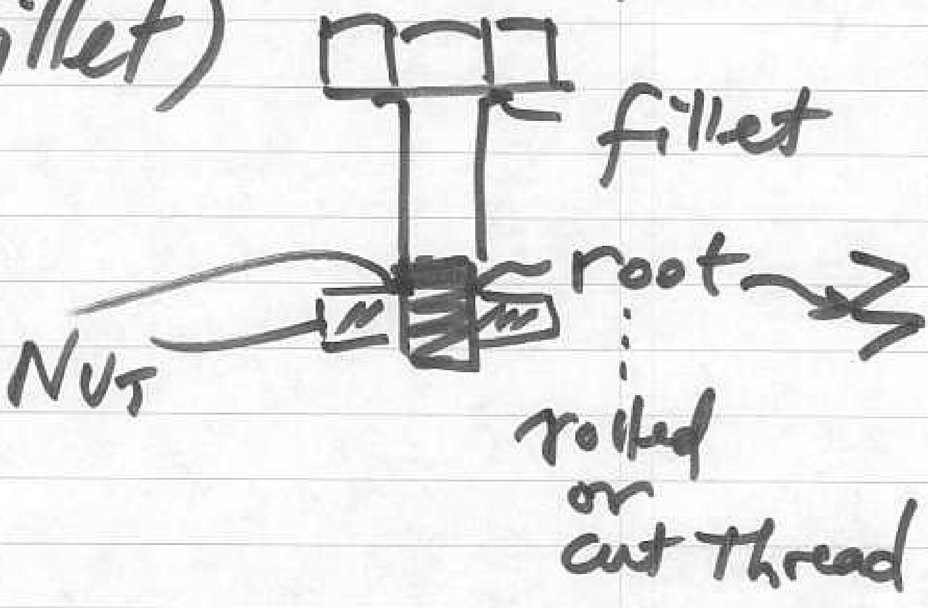
$$n_2 = \frac{107k(.142\ in^2) - 13k}{(.168)(15,625)}$$

$$\boxed{n_2 = 0.83}$$

joint (bolt) would fail due to "yielding" before separation.

Fatigue Failure

- Stress concentrations at roots of The Threads and under head of bolt (fillet)



$2.1 < K_f < 3.8$ Table 8-11

... machined finish.

- For our purposes just use Table 8-12
- Fully corrected Endurance Limits, S_e

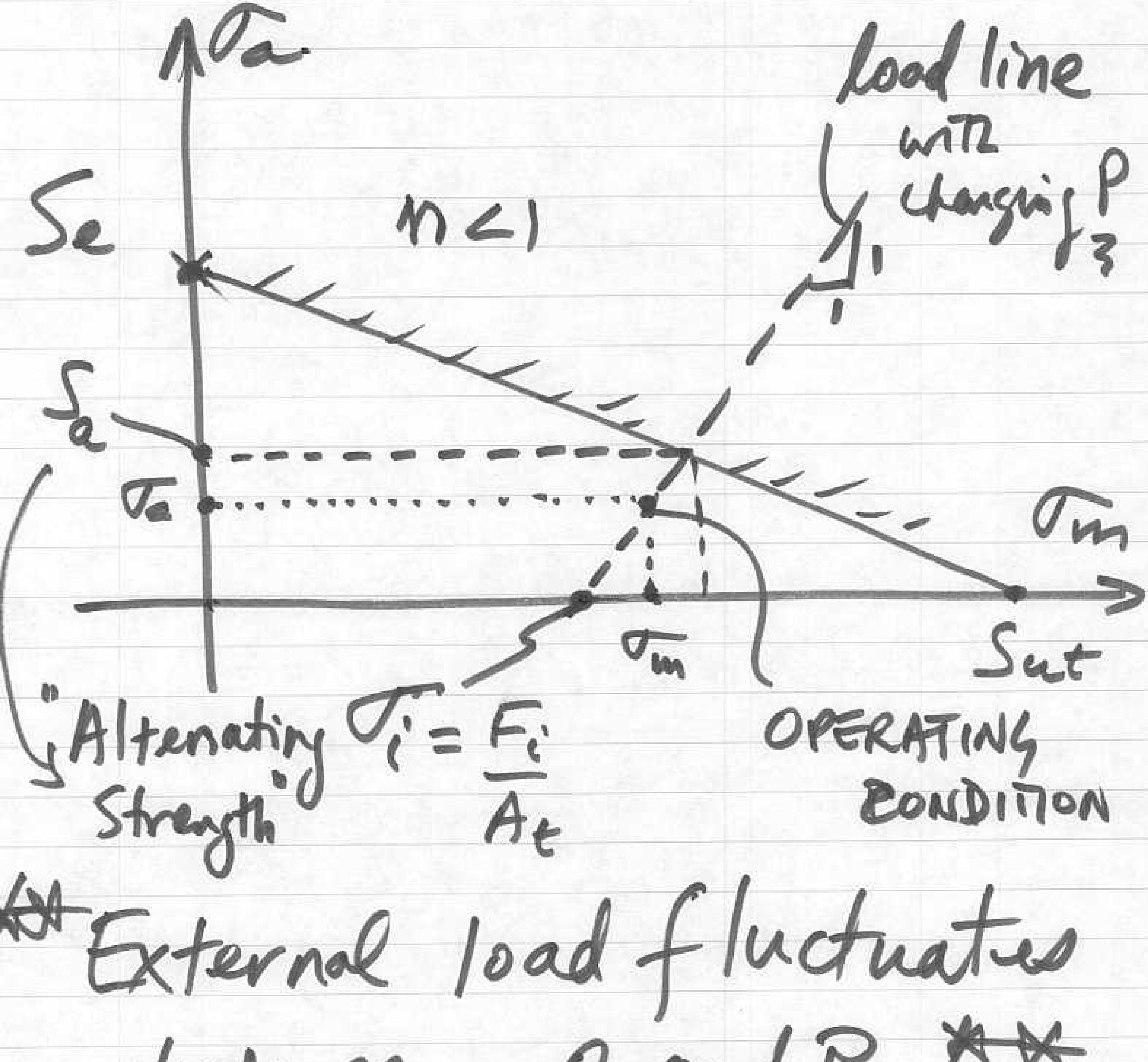
S_e

GRADE OR CLASS	SIZE RANGE	ENDURANCE LIMIT
SAE 5	$\frac{1}{4}$–1 in	18.6 kpsi
	$1\frac{1}{8}$–$1\frac{1}{2}$ in	16.3 kpsi
SAE 7	$\frac{1}{4}$–$1\frac{1}{2}$ in	20.6 kpsi
SAE 8	$\frac{1}{4}$–$1\frac{1}{2}$ in	23.2 kpsi
ISO 8.8	M16–M36	129 MPa
ISO 9.8	M1.6–M16	140 MPa
ISO 10.9	M5–M36	162 MPa
ISO 12.9	M1.6–M36	190 MPa

TABLE 8-12

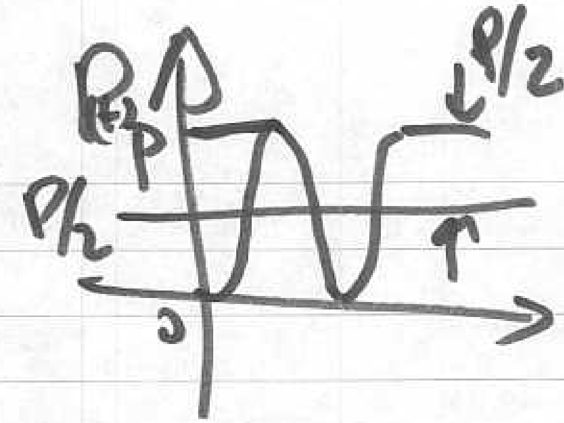
Fully Corrected Endurance Limits for Bolts and Screws with Rolled Threads

Use Goodman diagram for Fatigue



** External load fluctuates between 0 and P **

For Fatigue



$$\boxed{n = \frac{S_a}{\sigma_a}}$$

$$\sigma_a = \frac{CP}{2}\left(\frac{1}{A_t}\right) = \frac{CP}{2A_t}$$

$$\sigma_m = \frac{F_i}{A_t} + C\frac{P}{2}\cdot\left(\frac{1}{A_t}\right)$$

From geometry of Goodman diagram:

$$\boxed{S_a = \frac{S_{ut} - F_i/A_t}{1 + S_{ut}/S_e}}$$

FOR Finite Life use S_f @ N cycles

e.g. Returning to example

let P fluctuate from 0 to 10,000 lbs. Let S_e = 20.6 kpsi and Table 8-12 S_{ut} = 133 kpsi

$C = .168$

$$\sigma_a = C\,\frac{10{,}000}{2}\left(\frac{1}{0.142}\right)$$

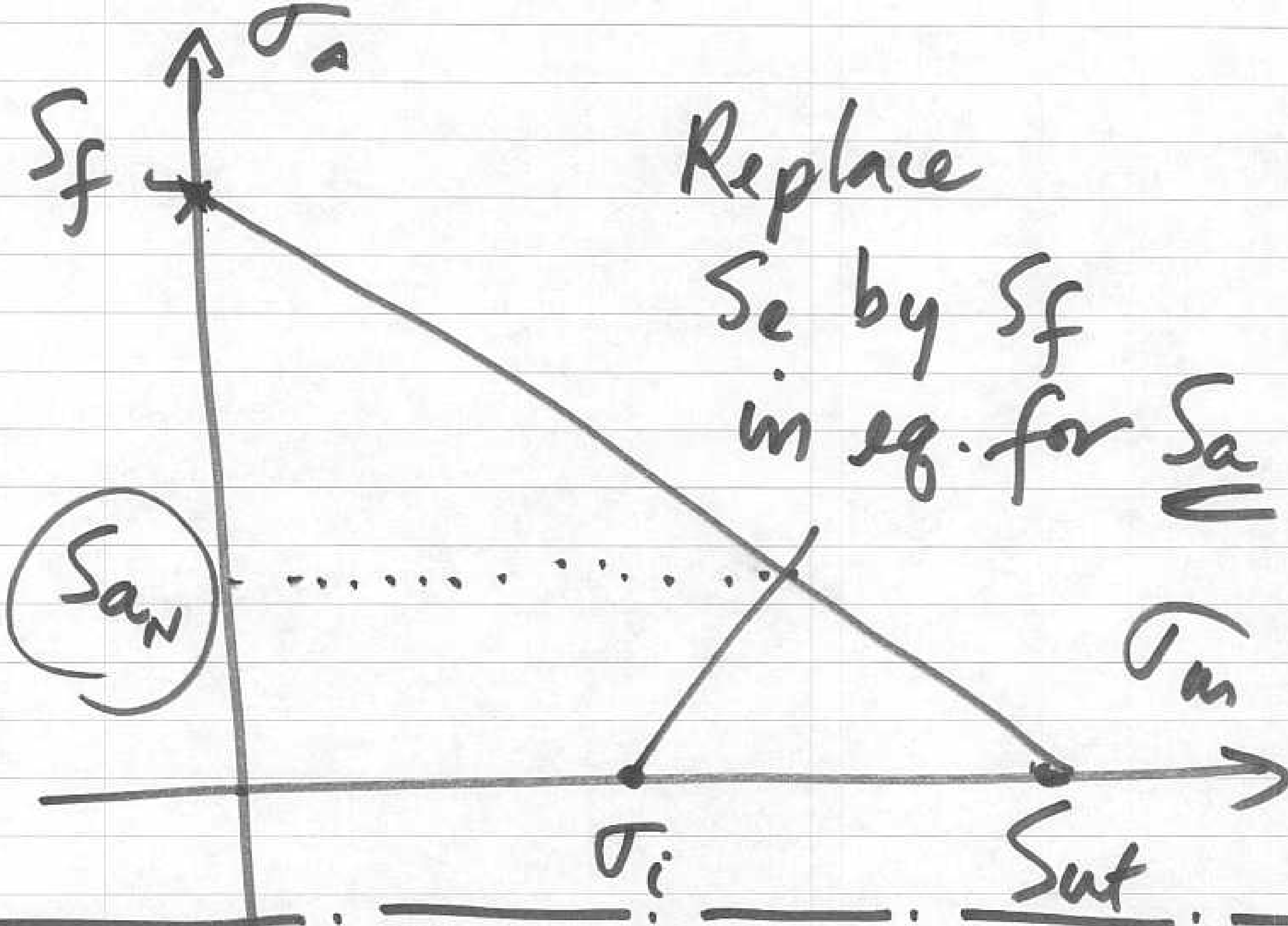
$$= C\,35.2\text{ kpsi} = 5.91\text{ kpsi}$$

$$S_a = \frac{133k - 13k/0.142}{1 + 133/20.6}$$

$$= 41.45/7.46 = 5.56\text{ kpsi}$$

$$\boxed{n = \frac{S_a}{\sigma_a} = \frac{0.94}{\text{Fails ??}}}$$

i.e. Finite Life



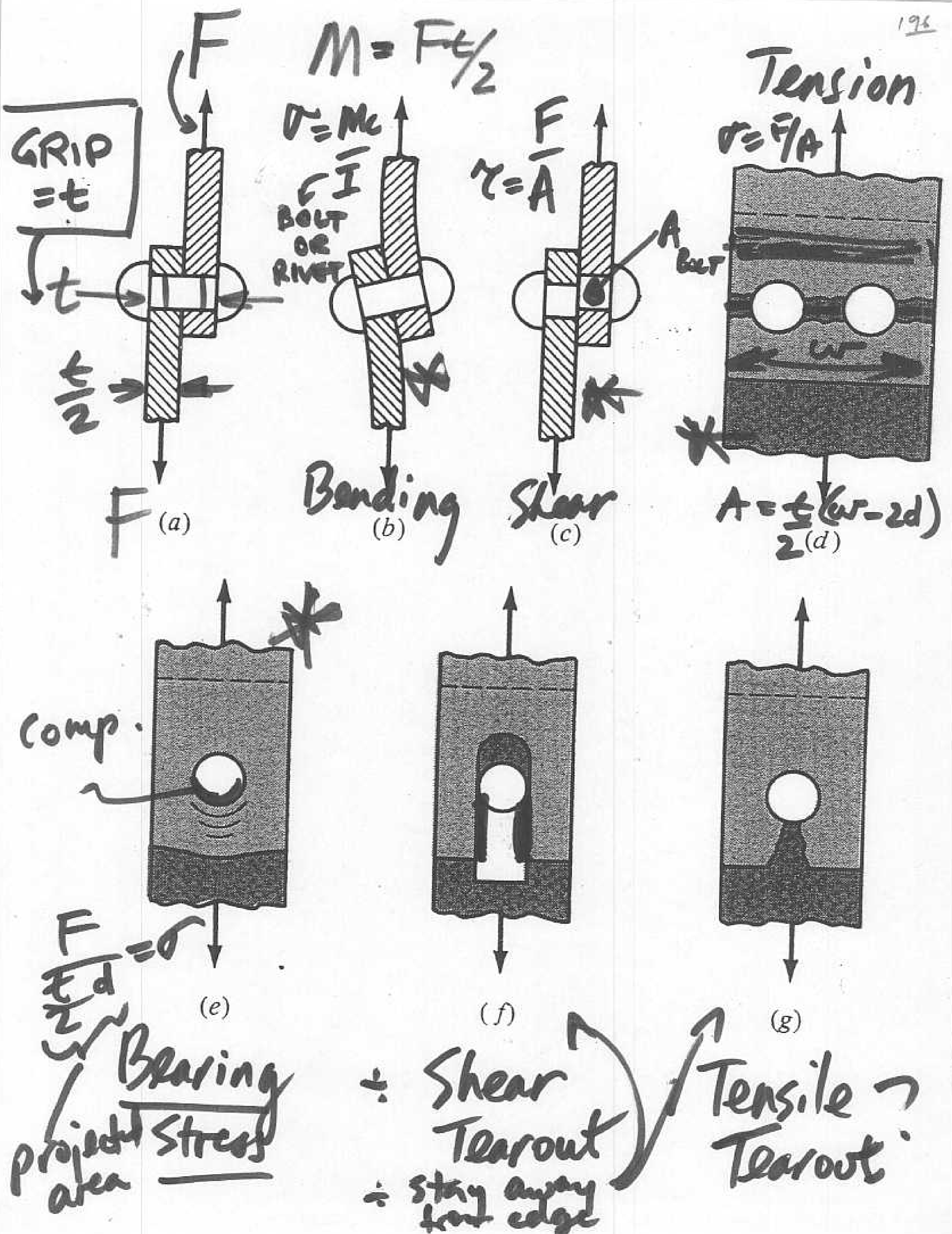
Replace S_e by S_f in eq. for $\underline{S_a}$

Exercise: find N if a safety factor of 1.3 is needed.

Direct Loading of Bolted and Riveted Joints

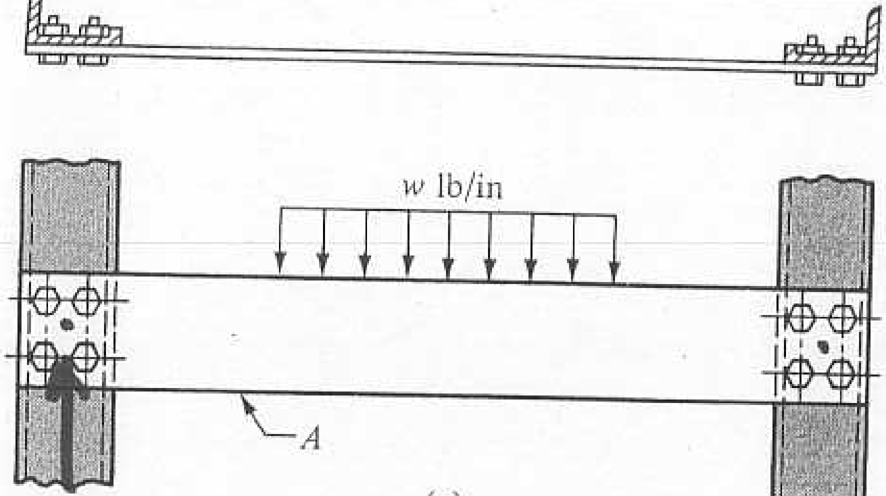
- Failure of bolts/rivets
  - bending* (b) 8-21
  - Shear* (c) 8-21
- Failure of materials/plates or beams joined
  - tension of members* (d) 8-21
  - bearing stresses/crushing* (e) 8-21
  - shear & tensile tear-out (g) & (f) 8-21 → place rivet/bolt 1½ diameters from edge

$\boxed{\text{*CALCULATE } n}$

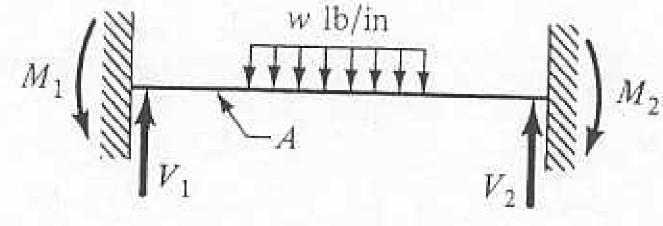


F
M = Ft/2
GRIP = t
t
t/2
F
(a)
σ = Mc/I
BOLT OR RIVET
Bending
(b)
τ = F/A
A BOLT
Shear
(c)
Tension
σ = F/A
w
A = t/2 (w − 2d)
(d)
Comp.
F / (t d/2) = σ
(e)
(f)
(g)
Bearing Stress
projected area
Shear Tearout
stay away from edge
Tensile Tearout

ECCENTRIC LOADING

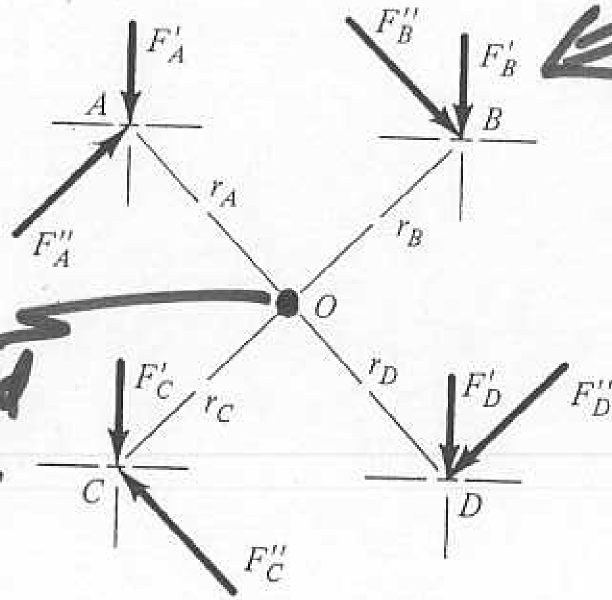


(a)



(b)

M & V from beam analysis



(c)

⟸ MAX FORCE ?

Centroid of joint

Primary Shear Force

$F' = \frac{V}{n}$ ⟸ Shear force @ centroid

n ← # of bolts

Secondary Shear Force F''

Due to "Applied Moment" M

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

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

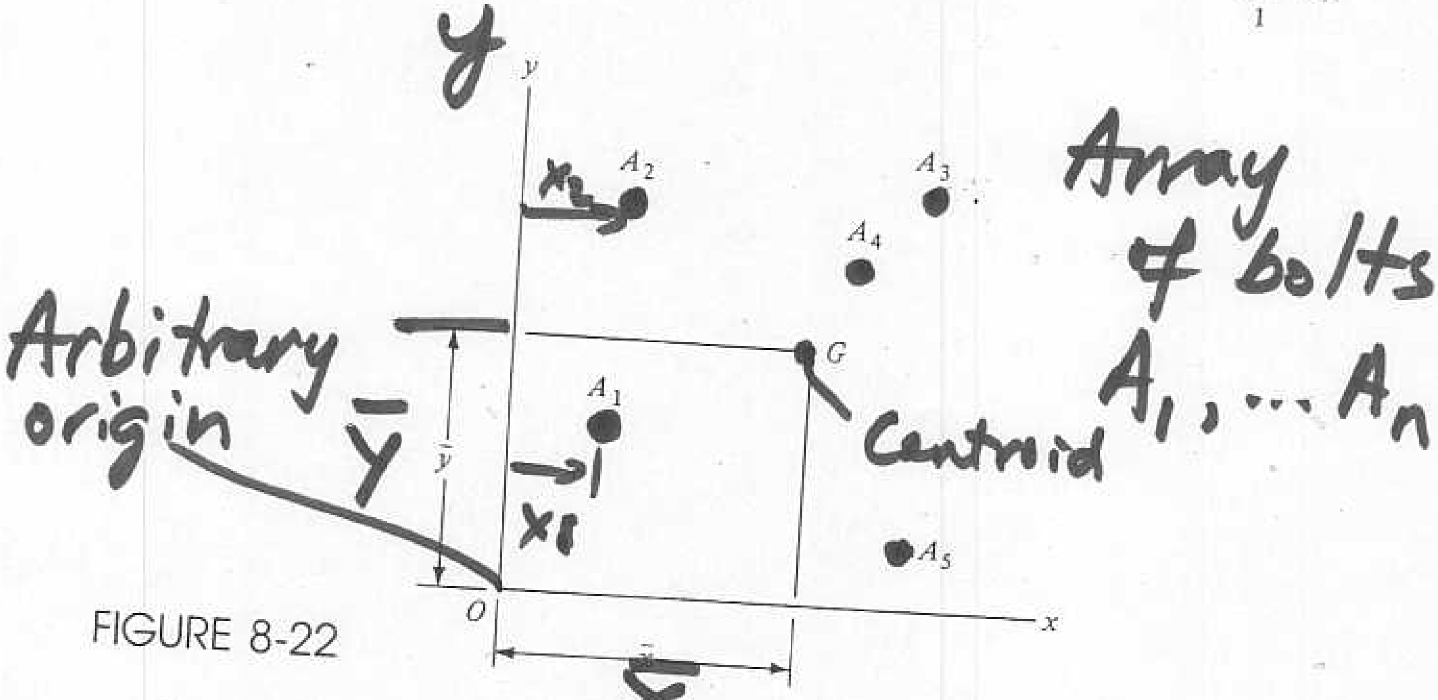


FIGURE 8-22

- Centroid is sometimes obvious from inspection or can be found from eq 8-42

- If all areas equal

$$\bar{x} = \frac{\sum x_i}{n} \quad \text{and} \quad \bar{y} = \frac{\sum y_i}{n}$$

Secondary Shear, F''

bolt loads } F''_A, F''_B, F''_C ... etc

distance of bolt from centroid } r_A, r_B, r_C ...etc

$$M = F''_A r_A + F''_B r_B + \ldots$$

Moment @ joint

ASSUME (not a bad assumption)

... The more distant (from centroid) bolts see bigger loads, i.e.

$$\frac{F''_A}{r_A} = \frac{F''_B}{r_B} = \frac{F''_C}{r_C} \ldots = \text{const}$$

Combining eqs for M & load sharing, we have

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

nth bolt

Combine $\vec{F}'$ and $\vec{F}''$ vectorially at each bolt or rivet.

i.e. $\frac{V}{n}$ & F_n''

direction of V

$\perp$ to $\vec{r}_n$

EXAMPLE 8-4

Shown in Fig. 8-24 is a 15- by 200-mm rectangular steel bar cantilevered to a 250-mm steel channel using four bolts. On the basis of the external load of 16 kN, find:

(*a*) The resultant load on each bolt

(*b*) The maximum bolt shear stress

c) bearing stress (max)

d) bending in bar (stress)

M16x2 bolts

all r's are 96 mm

- Centroid by observation
- Symmetry

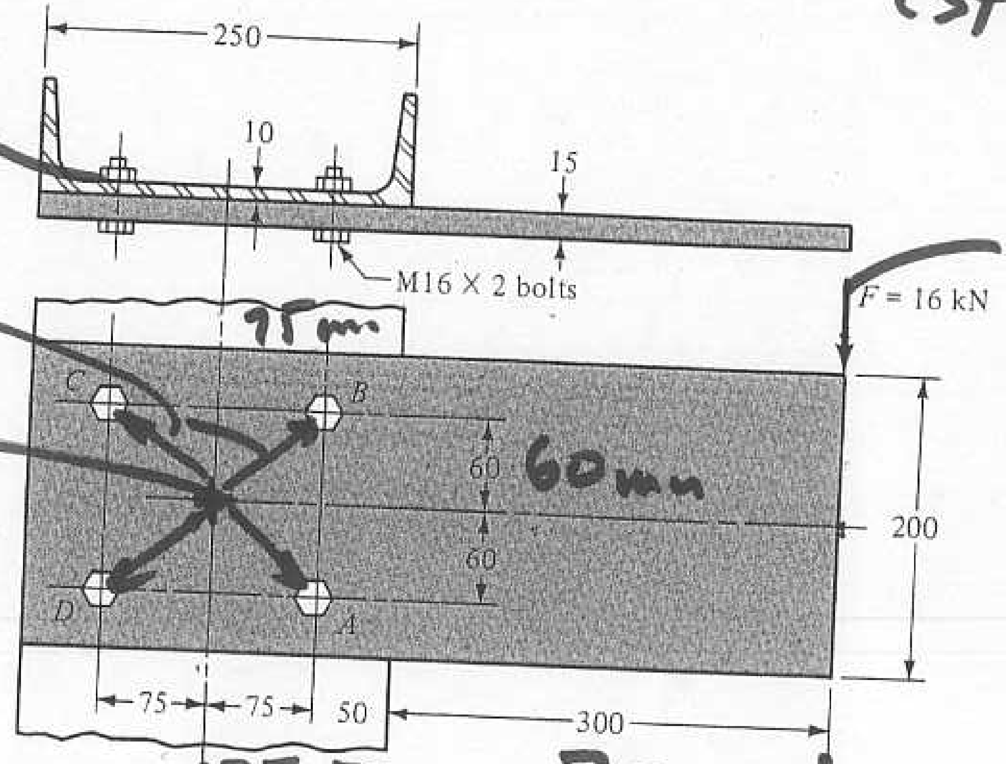


FIGURE 8-24

Dimensions in millimeters.

F=16kN

V = 16kN   M at centroid = .425F N-m

Primary Shear

$V = 16\,kN \qquad F' = \frac{V}{n} = \underline{4\,kN}$

Secondary Shear

$$F''_n = \frac{M r_n}{(r_A^2 + r_B^2 + r_C^2 + r_D^2)} = \frac{M r^{\cancel{2}}}{4 r^{\cancel{2}}}$$

since all r's are equal

$$F'' = \frac{M}{4r} \qquad M = (16k)(.425)$$

$$\underline{M = 6800\ N\text{-}m}$$

$$F'' = \frac{6800\ N\text{-}m}{4(96)\ mm} = \underline{\underline{17.7\,kN}}$$

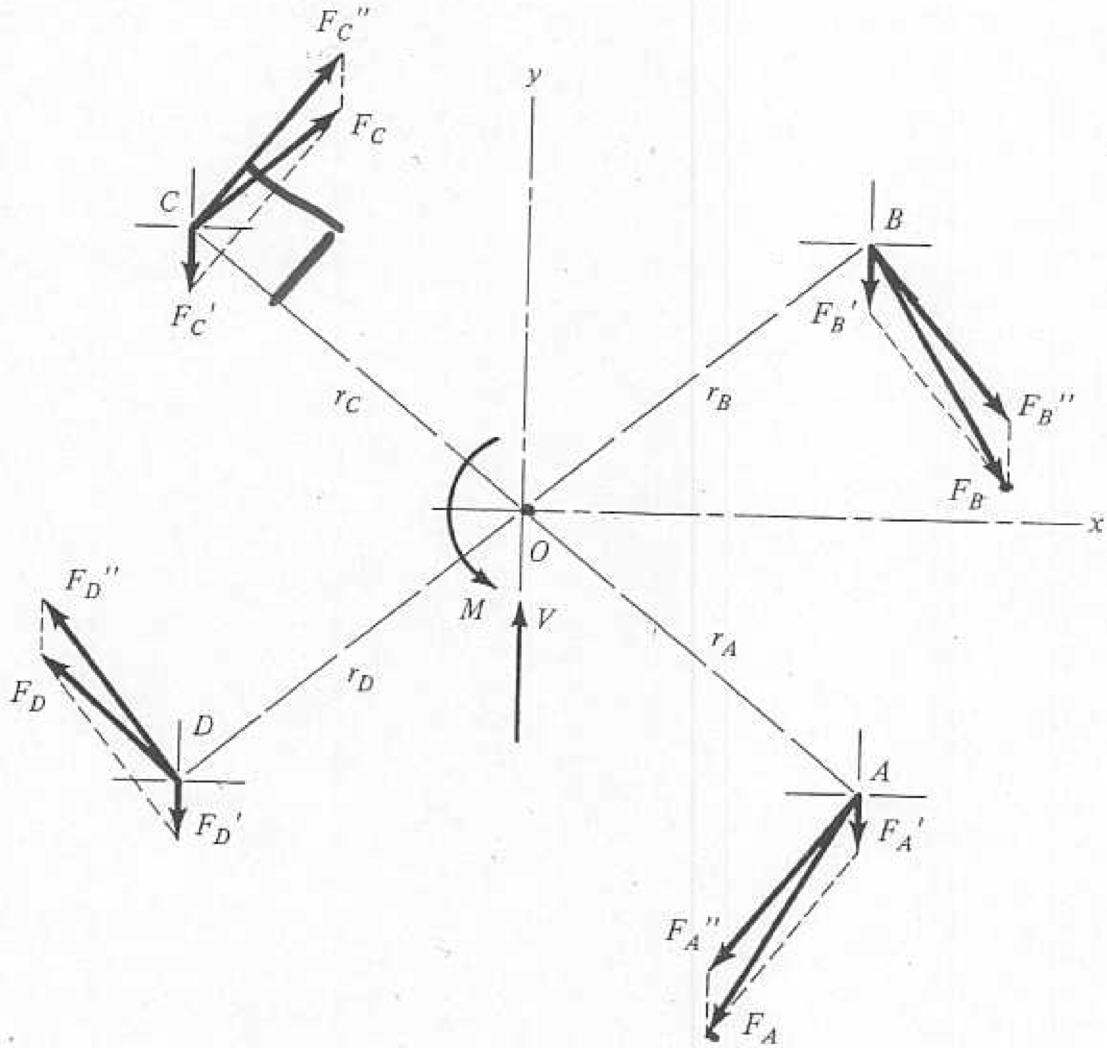


FIGURE 8-25

$F_A'' \perp r_A$, etc
F' acts vertically } APPLIED TO JOIN

<u>M & V</u> are reaction moment and shear

$F_A = F_B > F_C = F_D$

From geometry

a) $\vec{F}_A = \vec{F}_A' + \vec{F}_A''$

and $\boxed{F_A = 21\,kN = F_B}$

$\left(F_C = F_D = \underline{13.8\,kN} \right)$

---

b) $\tau_A = \dfrac{F_A}{A_s} = \dfrac{21\,kN}{201\,mm^2} = \underline{\underline{104\,MPa}}$

A_s → use major diameter

IF Bolt is Grade 5.8, $S_p = 380\,MPa$

BY MSST:

$n_s = \dfrac{S_p/2}{\tau} = \dfrac{S_{ssp}}{\tau} = \dfrac{190}{104} = \underline{\underline{1.82}}$

n_s ← Factor of safety against shear

c) Max bearing stress (@ A or B)

$$\sigma_b = \frac{F_A}{A_b}$$

$A_b = t\,d$ ⇒ Projected Area of bolt diam.

Thickness of thinner member

diameter (hole or bolt)

$A_b = (10\text{ mm})(16\text{ mm}) = \underline{160\text{ mm}^2}$

$$\boxed{\sigma_b = -\frac{21\text{ kN}}{160\text{ mm}^2} = \underline{-131\text{ MPa}} \quad \text{... Compression}}$$

$$\Rightarrow \underset{\text{S Factor}}{n_b} = \frac{S_y}{\sigma_b}$$

d) Critical bending of bar

- occurs across A-B
due to reduction in area

... - Assumption.

$$M_{AB} = 16\,kN\,(350\,mm)$$
$$= \underline{5600\ N\text{-}m}$$

$$\Rightarrow \boxed{\sigma_{bend} = \frac{Mc}{I}}$$

"Transfer"
60 mm

$$\Rightarrow I = I_{bar} - 2\left(I_{holes} + d^2A\right)$$

$$\underline{I = 8.26 \times 10^6\ mm^4}$$

$$\sigma_{bend} = \frac{(5800)(100\,mm)}{8.26 \times 10^6\ mm^4}$$

$$\boxed{\sigma_{bend} = 67.8\ MPa}$$

- SAFETY FACTORS
  FOR BEARING STRESS &
  BENDING OF BAR/BEAM
  DEPEND ON MTL AND
  ASSUMED FAILURE THEORY

- BUT MAT'LS ARE OFTEN
  WEAKER THAN BOLTS SO
  THIS COULD RESULT IN A
  LOWER SAFETY FACTOR
  THAN FOR THE BOLTS
  OR RIVETS.

Assignment #6

Due Mon. March 24

8-11 Use $\frac{k_b}{k_b + k_m} = C = 0.213$

8-21 Use $C = 0.213$

8-24

8-30

8-37 (Assume bracket pivots about lower edge)

8-39