جامعة بيرزيت
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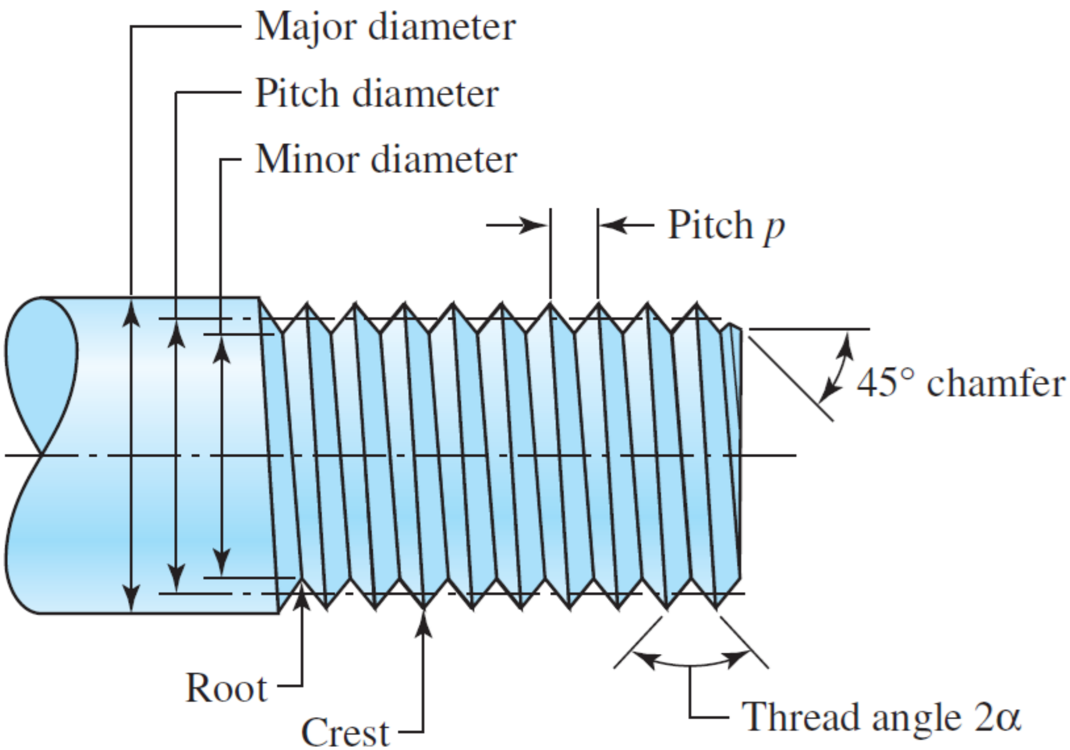
Introduction

Introduction

8.1 Thread Standards and Definitions

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.



8.1 Thread Standards and Definitions

Table 8–1

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

Nominal Major Diameter d mm	Coarse-Pitch Series: Pitch p mm	Coarse-Pitch Series: Tensile-Stress Area A_t mm²	Coarse-Pitch Series: Minor-Diameter Area A_r mm²	Fine-Pitch Series: Pitch p mm	Fine-Pitch Series: Tensile-Stress Area A_t mm²	Fine-Pitch Series: Minor-Diameter Area A_r mm²
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
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

8.1 Thread Standards and Definitions

Table 8–2

Diameters and Area of Unified Screw Threads UNC and UNF*

Size Designation	Nominal Major Diameter in	Coarse Series—UNC: Threads per Inch N	Coarse Series—UNC: Tensile-Stress Area A_t in²	Coarse Series—UNC: Minor-Diameter Area A_r in²	Fine Series—UNF: Threads per Inch N	Fine Series—UNF: Tensile-Stress Area A_t in²	Fine Series—UNF: Minor-Diameter Area A_r in²
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	0.373	0.351
$\frac{7}{8}$	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
$1\frac{1}{4}$	1.2500	7	0.969	0.890	12	1.073	1.024
$1\frac{1}{2}$	1.5000	6	1.405	1.294	12	1.581	1.521

8.1 Thread Standards and Definitions

Figure 8–3

(*a*) Square thread;
(*b*) Acme thread.

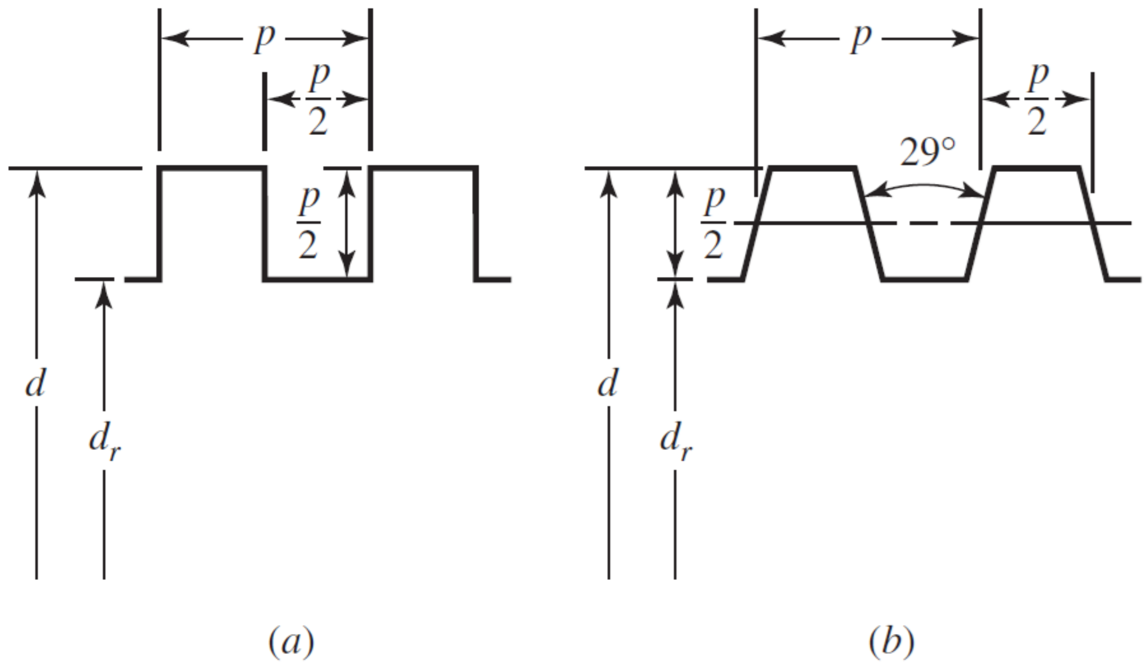


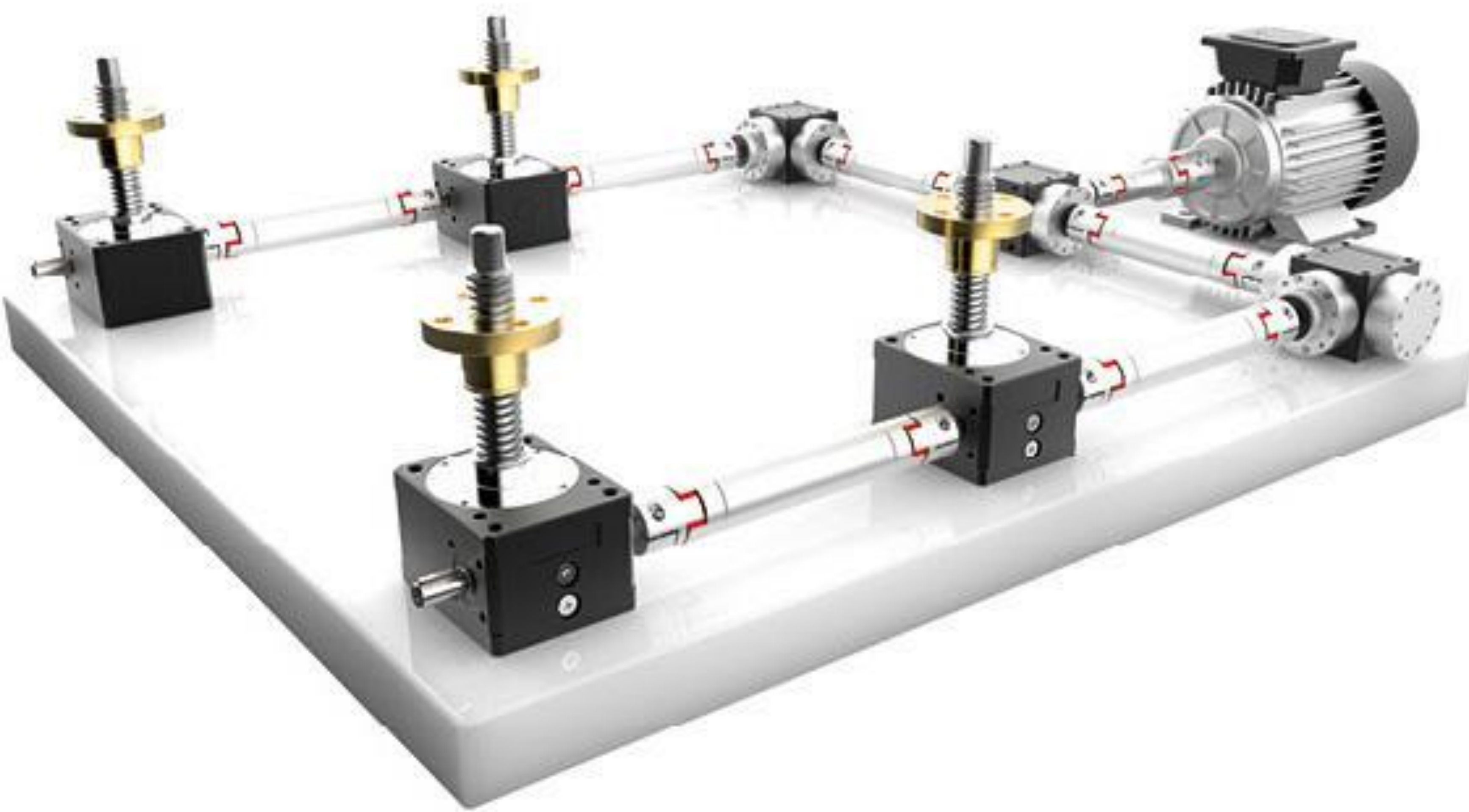
Table 8–3

Preferred Pitches for Acme Threads

d, in	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3
p, in	$\frac{1}{16}$	$\frac{1}{14}$	$\frac{1}{12}$	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{6}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$

8.2 Mechanics of Power Screws

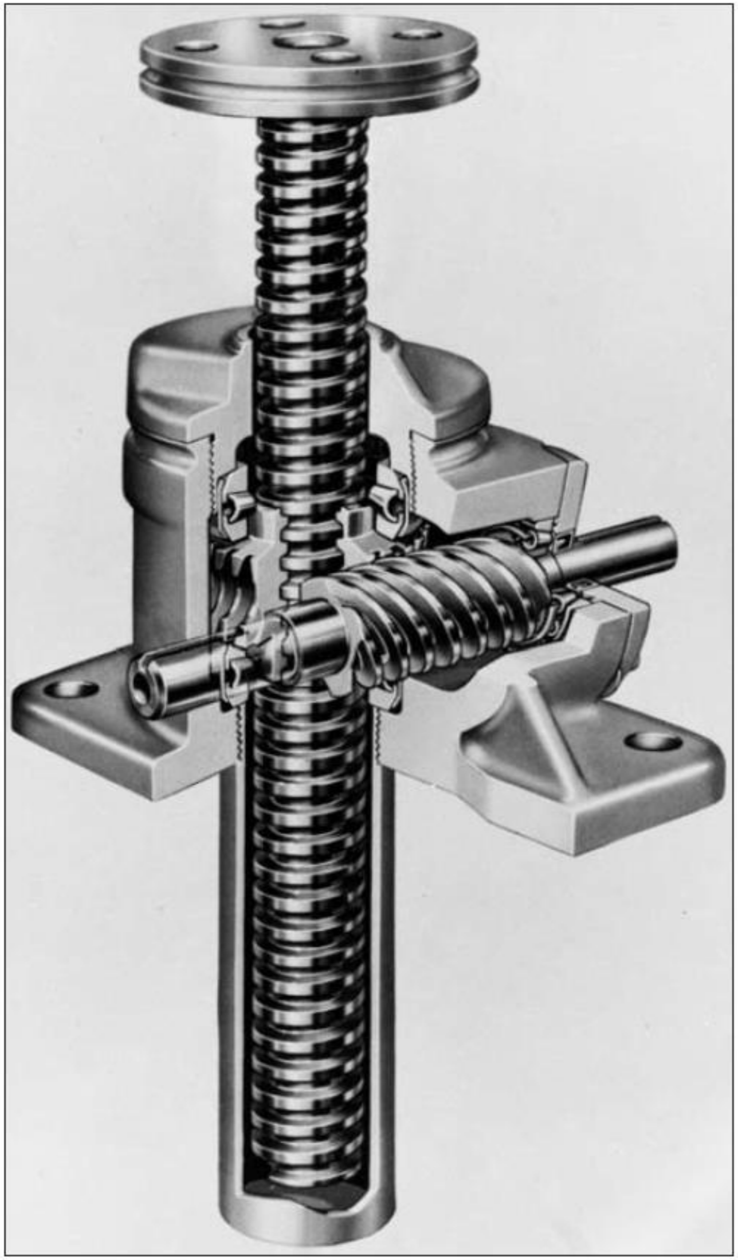
8.2 Mechanics of Power Screws



8.2 Mechanics of Power Screws

Figure 8–4

The Joyce worm-gear screw jack. *(Courtesy Joyce-Dayton Corp., Dayton, Ohio.)*



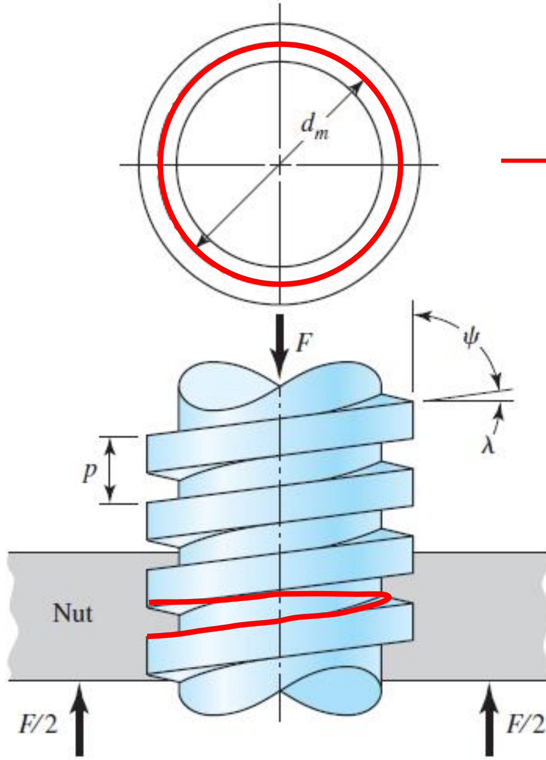
8.2 Mechanics of Power Screws

https://www.youtube.com/watch?v=i-Z4hz_KX0M

8.2 Mechanics of Power Screws

Figure 8–5

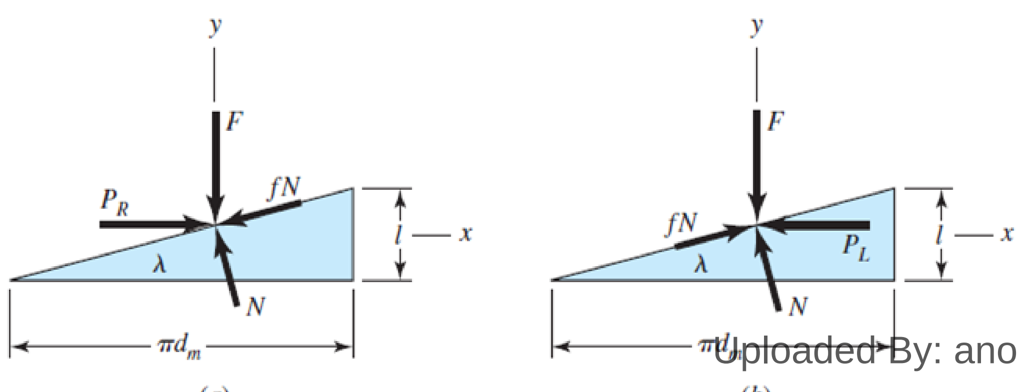
Portion of a power screw.



$\pi\, d_m$

Figure 8–6

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



8.2 Mechanics of Power Screws

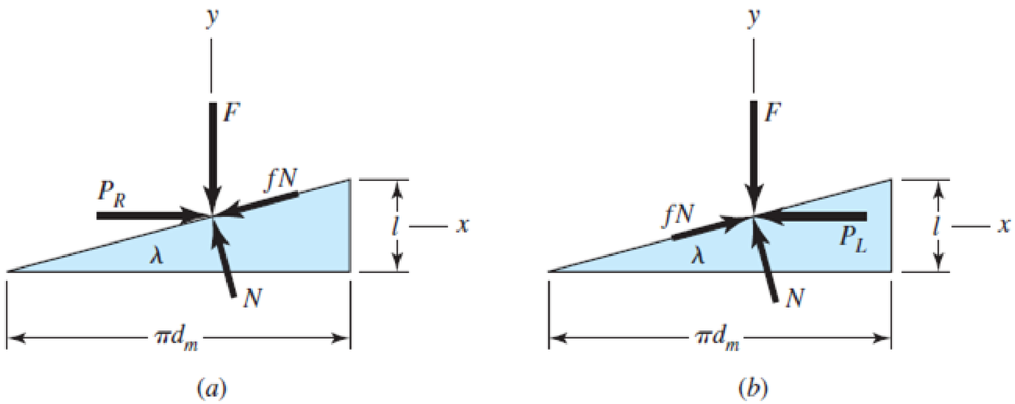


Figure 8–6

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

for raising the load, we have

$$\sum F_x = P_R - N \sin\lambda - fN\cos\lambda = 0$$

$$\sum F_y = -F - fN\sin\lambda + N\cos\lambda = 0$$

In a similar manner, for lowering the load, we have

$$\sum F_x = -P_L - N \sin\lambda + fN\cos\lambda = 0$$

$$\sum F_y = -F + fN\sin\lambda + N\cos\lambda = 0$$

8.2 Mechanics of Power Screws

Next, divide the numerator and the denominator of these equations by cos λ and use the relation $\tan\lambda = l/\pi d_m$ (Fig. 8–6). We then have, respectively,

$$P_R = \frac{F(\sin\lambda + f\cos\lambda)}{\cos\lambda - f\sin\lambda}$$

$$P_L = \frac{F(f\cos\lambda - \sin\lambda)}{\cos\lambda + f\sin\lambda}$$

$$P_R = \frac{F[(l/\pi d_m) + f]}{1 - (fl/\pi d_m)}$$

$$P_L = \frac{F[f - (l/\pi d_m)]}{1 + (fl/\pi d_m)}$$

8.2 Mechanics of Power Screws

x mean radius $d_m/2$,

$$P_R = \frac{F[(l/\pi d_m) + f]}{1 - (fl/\pi d_m)}$$

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

$$P_L = \frac{F[f - (l/\pi d_m)]}{1 + (fl/\pi d_m)}$$

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

8.2 Mechanics of Power Screws

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

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_L 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

$$\pi f d_m > l$$

Overhauling <> Self Locking

8.2 Mechanics of Power Screws

Power Screw Efficiency – Raising the load

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

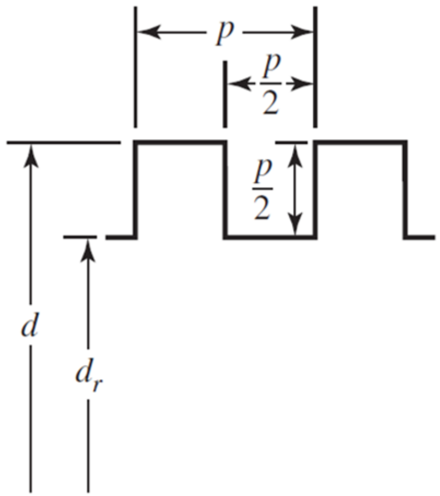
With Friction

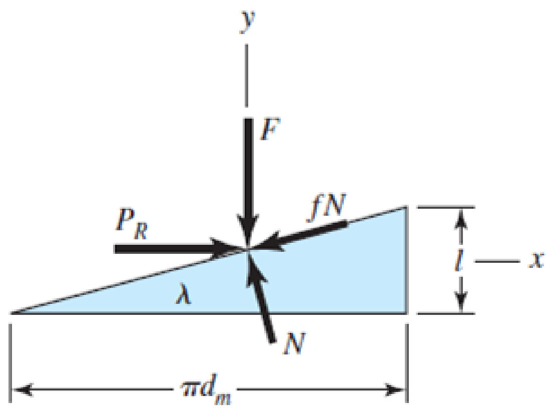
$$T_0 = \frac{Fl}{2\pi}$$

Without Friction $f = 0$

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

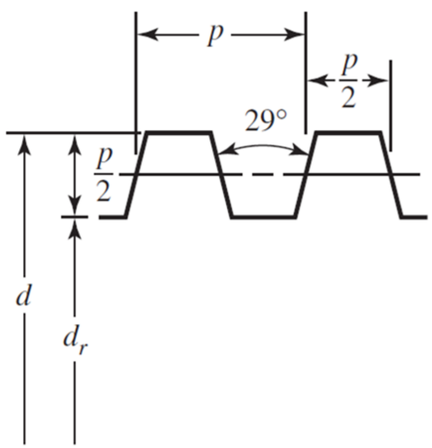
8.2 Mechanics of Power Screws

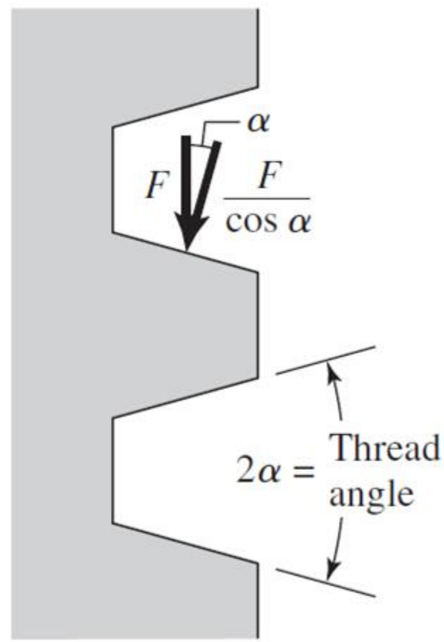




(a)

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



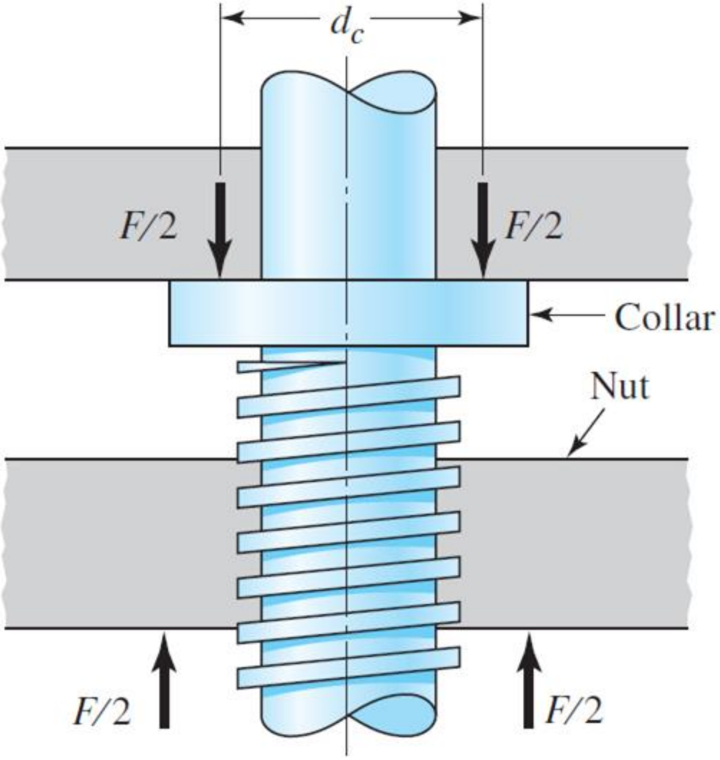


$$T_R = \frac{Fd_m}{2}\left(\frac{l + \pi f d_m \sec\alpha}{\pi d_m - f l \sec\alpha}\right)$$

8.2 Mechanics of Power Screws

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. The right figure shows a typical thrust collar in which the load is assumed to be concentrated at the mean collar diameter *dc*. If *fc* is the coefficient of collar friction, the torque required is

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



(b)

8.2 Mechanics of Power Screws

Table 8–5

Coefficients of Friction f for Threaded Pairs

Source: H. A. Rothbart and T. H. Brown, Jr., *Mechanical Design Handbook,* 2nd ed., McGraw-Hill, New York, 2006.

Screw Material	Nut Material			
	Steel	Bronze	Brass	Cast Iron
Steel, dry	0.15–0.25	0.15–0.23	0.15–0.19	0.15–0.25
Steel, machine oil	0.11–0.17	0.10–0.16	0.10–0.15	0.11–0.17
Bronze	0.08–0.12	0.04–0.06	—	0.06–0.09

Table 8–6

Thrust-Collar Friction Coefficients

Source: H. A. Rothbart and T. H. Brown, Jr., *Mechanical Design Handbook,* 2nd ed., McGraw-Hill, New York, 2006.

Combination	Running	Starting
Soft steel on cast iron	0.12	0.17
Hard steel on cast iron	0.09	0.15
Soft steel on bronze	0.08	0.10
Hard steel on bronze	0.06	0.08

8.2 Mechanics of Power Screws - Stresses

The maximum nominal shear stress τ in torsion of the screw body can be expressed as

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

The axial stress σ in the body of the screw due to load F is

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

in the absence of column action. For a short column the J. B. Johnson buckling formula is given by Eq. (4–43), which is

$$\left(\frac{F}{A}\right)_{\text{crit}} = S_y - \left(\frac{S_y}{2\pi}\frac{l}{k}\right)^2 \frac{1}{CE} \tag{8–9}$$

8.2 Mechanics of Power Screws - Stresses

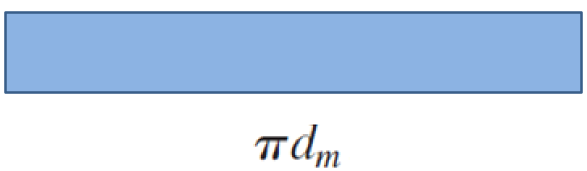
Nominal thread stresses in power screws can be related to thread parameters as follows. The bearing stress in Fig. 8–8, σ_B, is

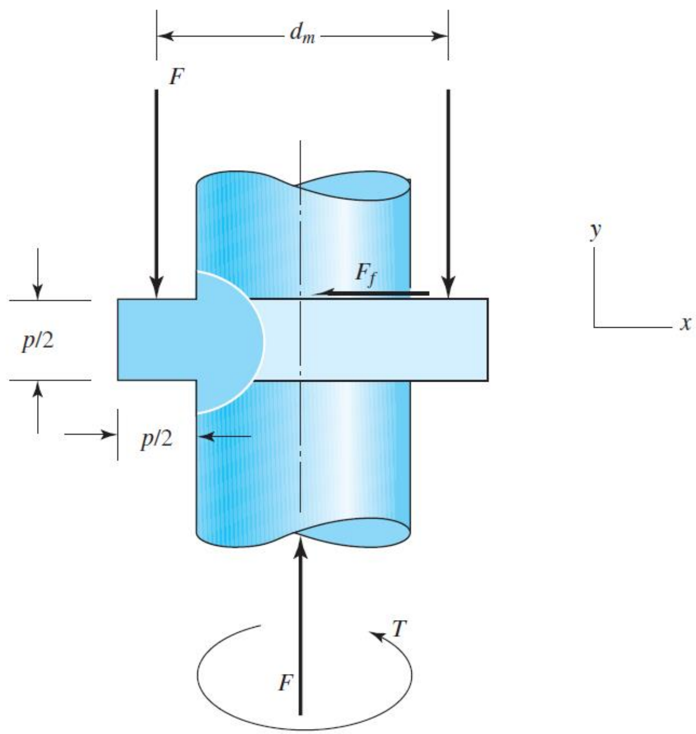
$$\sigma_B = -\frac{F}{\pi d_m n_t p/2} = -\frac{2F}{\pi d_m n_t p} \qquad (8\text{–}10)$$

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.



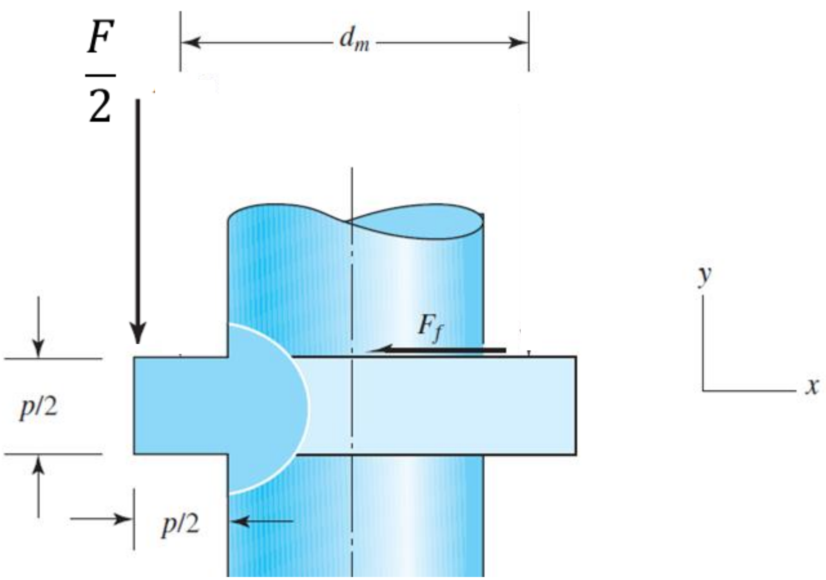


8.2 Mechanics of Power Screws - Stresses

The bending stress at the root of the thread σ_b is found from

$$Z = \frac{I}{c} = \frac{(\pi d_r n_t)(p/2)^2}{6} = \frac{\pi}{24} d_r n_t p^2 \qquad M = \frac{Fp}{4} \qquad M = \frac{F}{2}\frac{p}{2}$$

$$\sigma_b = \frac{M}{Z} = \frac{Fp}{4}\frac{24}{\pi d_r n_t p^2} = \frac{6F}{\pi d_r n_t p} \qquad (8\text{–}11)$$

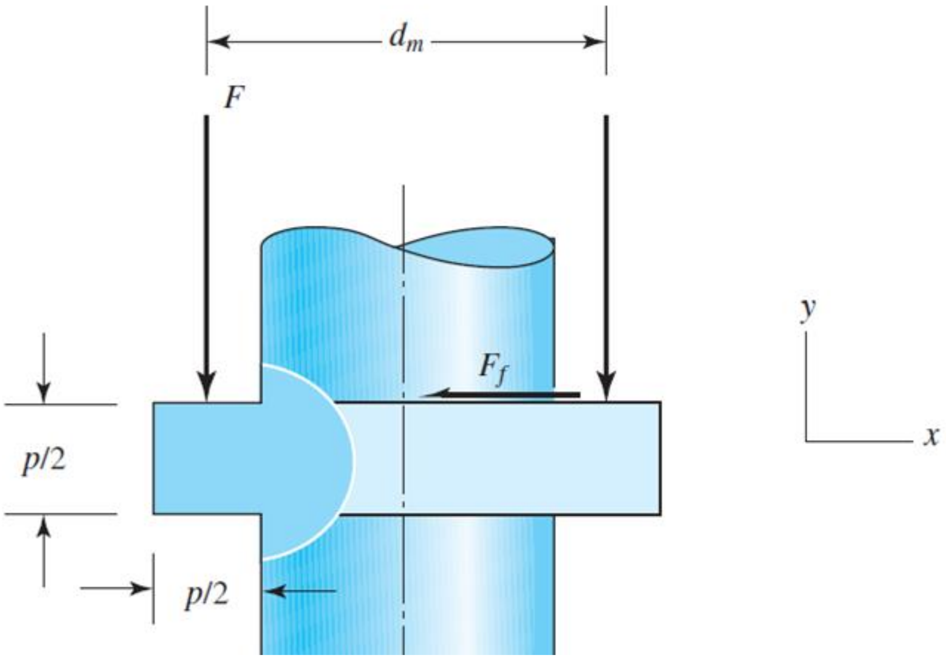


8.2 Mechanics of Power Screws - Stresses

The transverse shear stress τ at the center of the root of the thread due to load F is

$$\tau = \frac{3V}{2A} = \frac{3}{2}\frac{F}{\pi d_r n_t p/2} = \frac{3F}{\pi d_r n_t p} \tag{8–12}$$

and at the top of the root it is zero.



8.2 Mechanics of Power Screws

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.

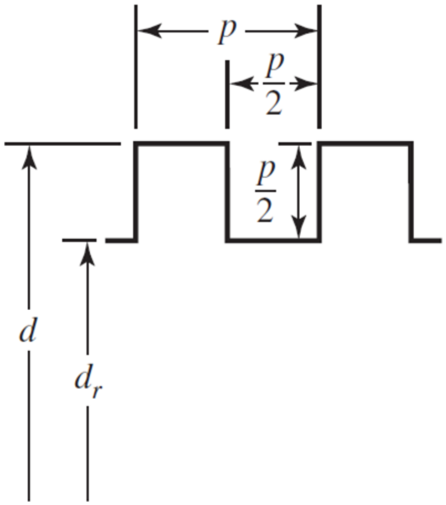
(*d*) Find the body stresses, torsional and compressive.

(*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.



8.2 Mechanics of Power Screws

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(*a*) Find the thread depth, thread width, pitch diameter, minor diameter, and lead.

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

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

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

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

8.2 Mechanics of Power Screws

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.

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

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

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

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

8.2 Mechanics of Power Screws

(*g*) Determine the von Mises stress at the root of the thread.

(*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

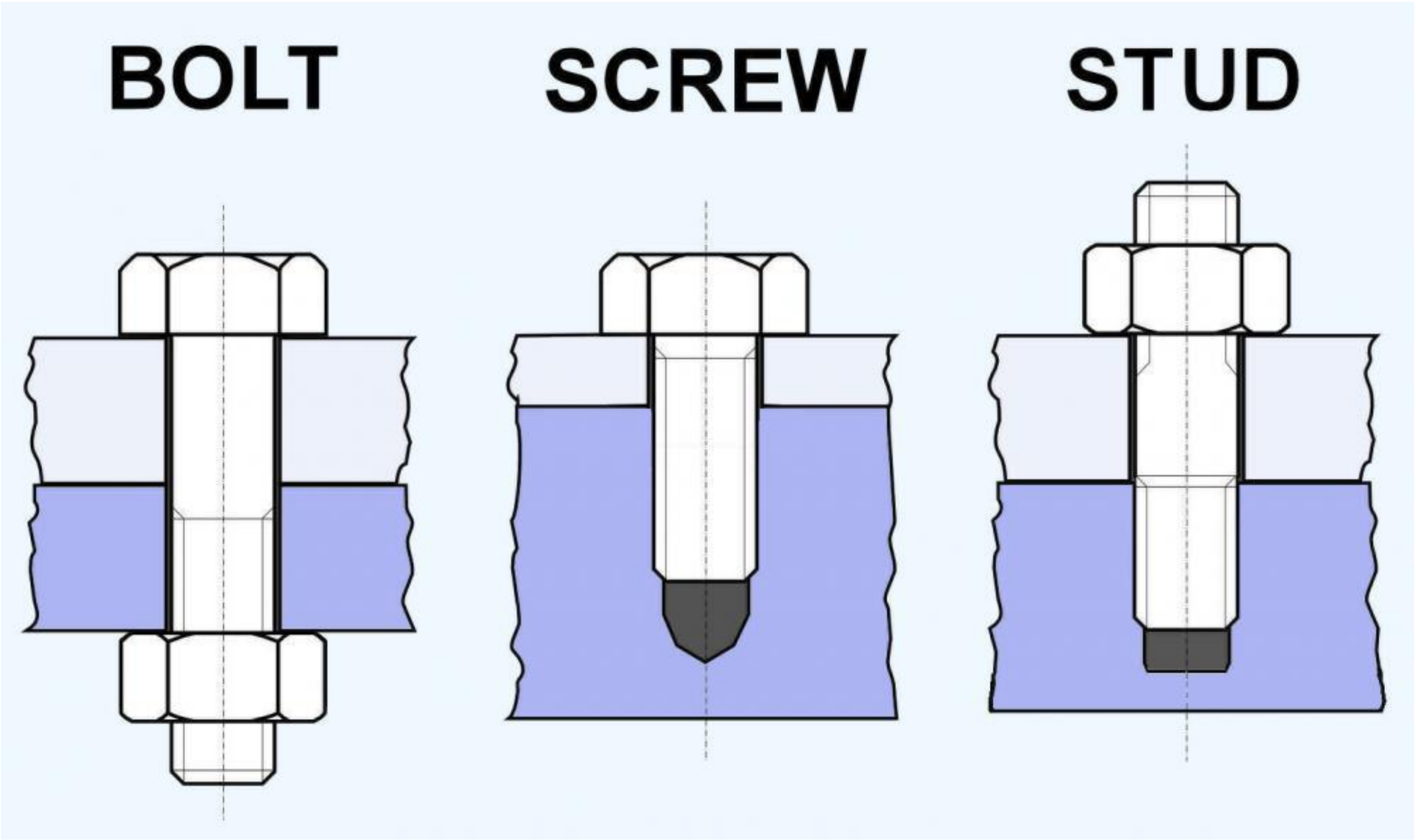
$$\sigma_x = 41.5 \text{ MPa} \qquad \tau_{xy} = 0$$
$$\sigma_y = -10.39 \text{ MPa} \qquad \tau_{yz} = 6.07 \text{ MPa}$$
$$\sigma_z = 0 \qquad \tau_{zx} = 0$$

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

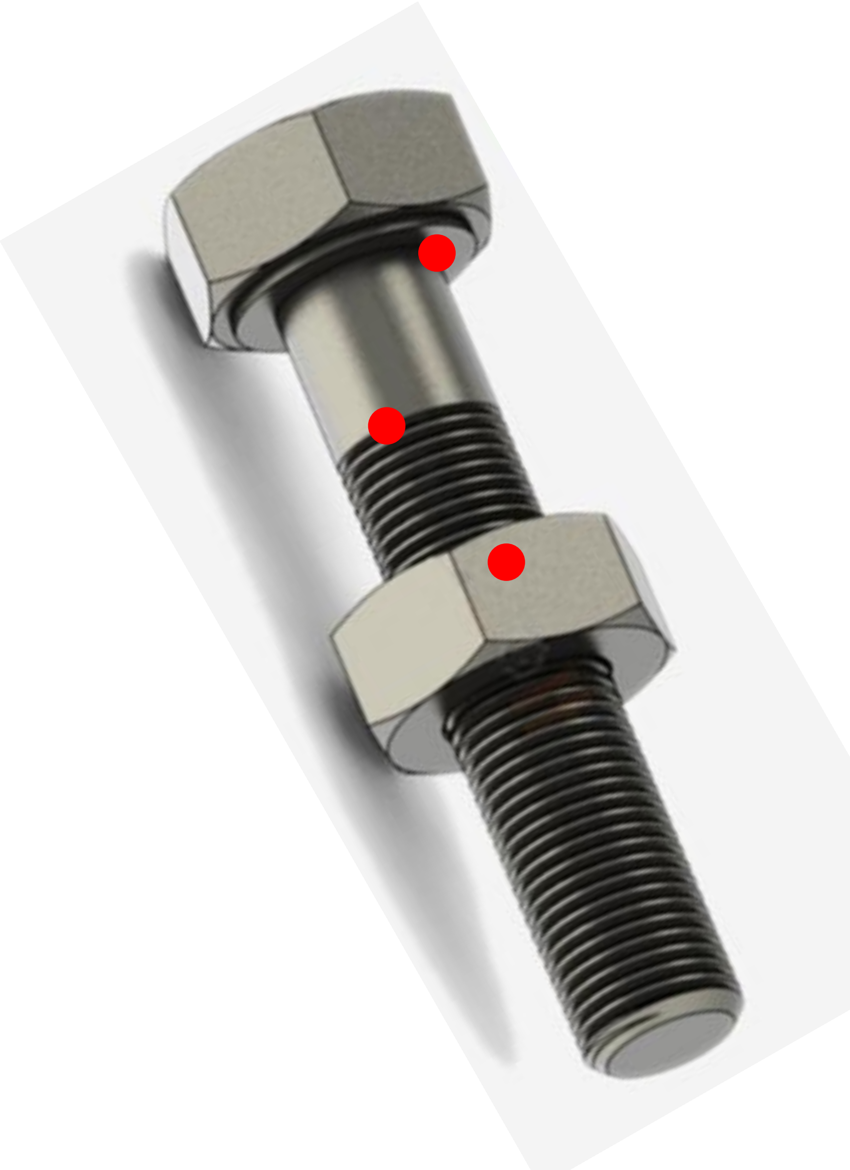
For the von Mises stress, Eq. (5–14) of Sec. 5–5 can be written as

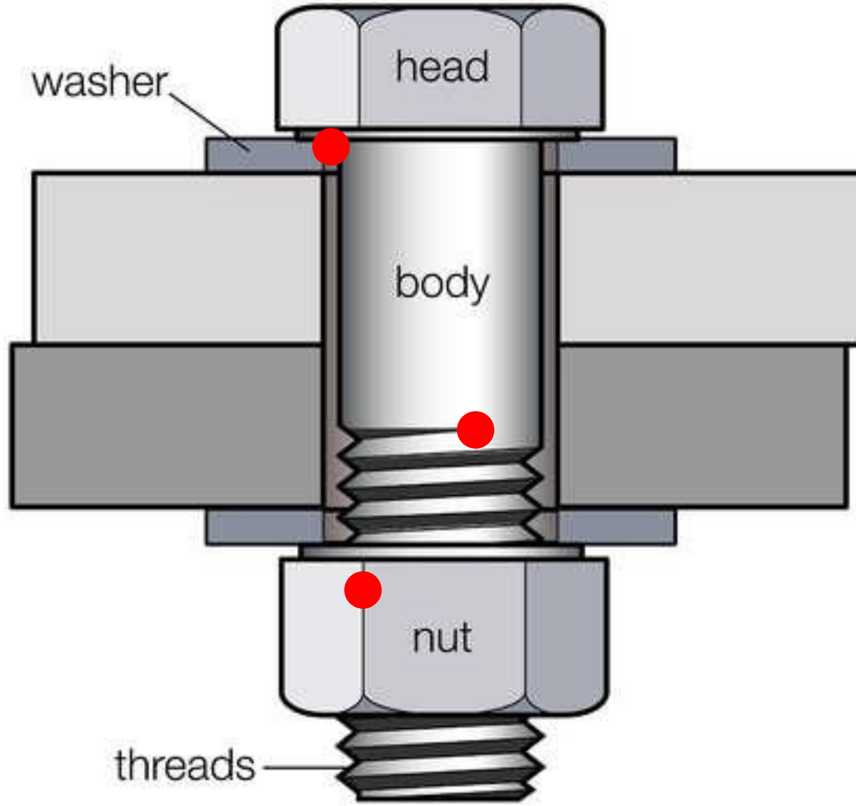
$$\sigma' = \frac{1}{\sqrt{2}}\{(41.5 - 0)^2 + [0 - (-10.39)]^2 + (-10.39 - 41.5)^2 + 6(6.07)^2\}^{1/2}$$
$$= 48.7 \text{ MPa}$$

8.3 Threaded Fasteners

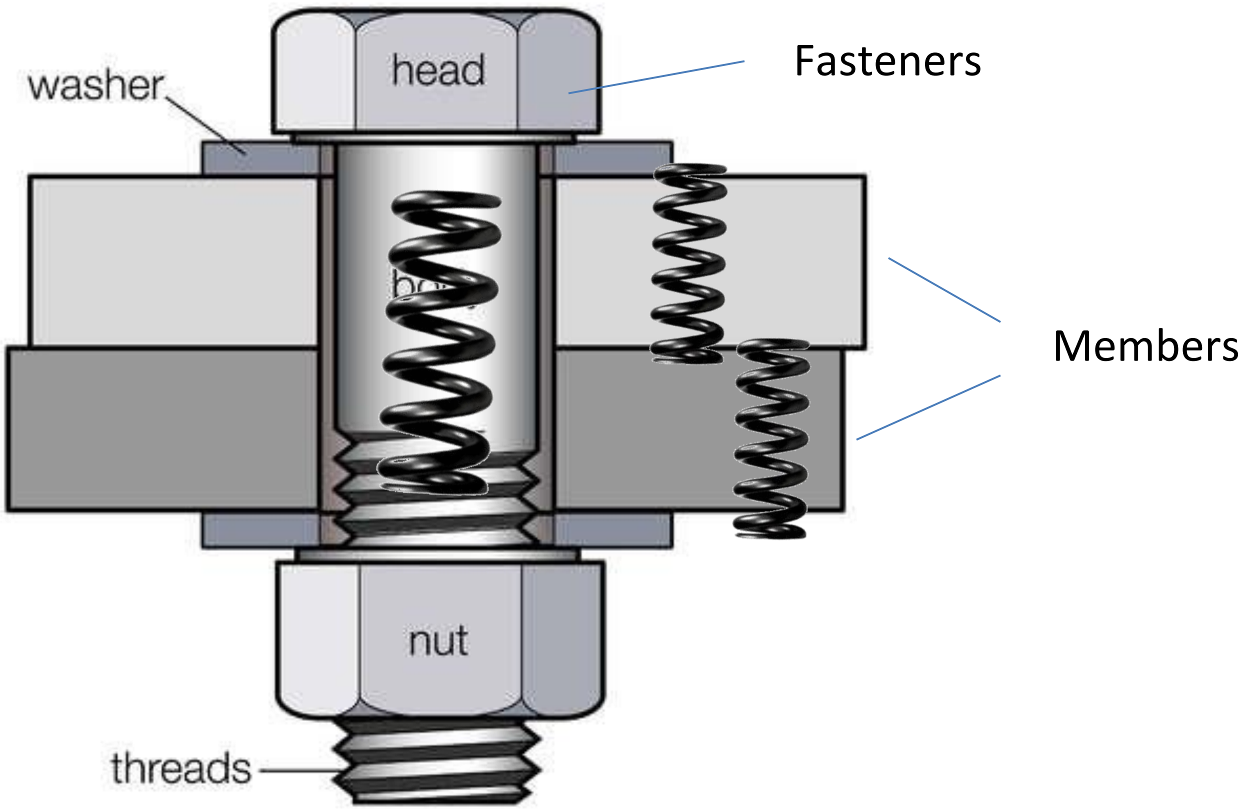


8.3 Threaded Fasteners





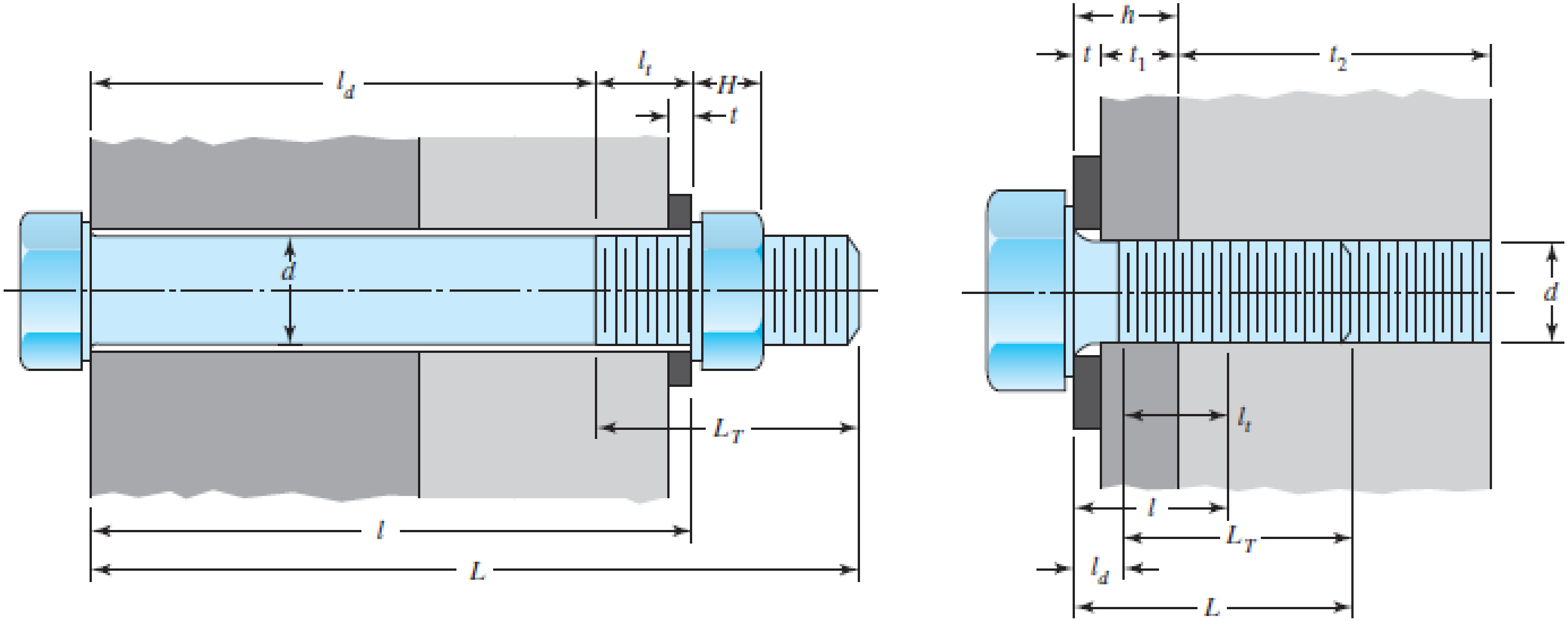
8.3 Threaded Fasteners



8.4 Joints – Fasteners Stiffness

8.5 Joints – Member Stiffness

8.4 Joints – Fasteners Stiffness



(a)

(b)

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

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

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

Grip length:

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

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

8.4 Joints – Fasteners Stiffness – Table A-32

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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
$\frac{7}{16}$ W	0.438	0.500	1.250	0.083
$\frac{1}{2}$ N	0.500	0.531	1.062	0.095
$\frac{1}{2}$ W	0.500	0.562	1.375	0.109
$\frac{9}{16}$ N	0.562	0.594	1.156	0.095
$\frac{9}{16}$ W	0.562	0.625	1.469	0.109
$\frac{5}{8}$ N	0.625	0.656	1.312	0.095
$\frac{5}{8}$ W	0.625	0.688	1.750	0.134
$\frac{3}{4}$ N	0.750	0.812	1.469	0.134
$\frac{3}{4}$ W	0.750	0.812	2.000	0.148

8.4 Joints – Fasteners Stiffness – Table A-33

Table A-33

Dimensions of Metric Plain Washers (All Dimensions in Millimeters)

Washer Size*	Minimum ID	Maximum OD	Maximum Thickness	Washer Size*	Minimum ID	Maximum OD	Maximum Thickness
1.6 N	1.95	4.00	0.70	10 N	10.85	20.00	2.30
1.6 R	1.95	5.00	0.70	10 R	10.85	28.00	2.80
1.6 W	1.95	6.00	0.90	10 W	10.85	39.00	3.50
2 N	2.50	5.00	0.90	12 N	13.30	25.40	2.80
2 R	2.50	6.00	0.90	12 R	13.30	34.00	3.50
2 W	2.50	8.00	0.90	12 W	13.30	44.00	3.50
2.5 N	3.00	6.00	0.90	14 N	15.25	28.00	2.80
2.5 R	3.00	8.00	0.90	14 R	15.25	39.00	3.50
2.5 W	3.00	10.00	1.20	14 W	15.25	50.00	4.00
3 N	3.50	7.00	0.90	16 N	17.25	32.00	3.50
3 R	3.50	10.00	1.20	16 R	17.25	44.00	4.00
3 W	3.50	12.00	1.40	16 W	17.25	56.00	4.60
3.5 N	4.00	9.00	1.20	20 N	21.80	39.00	4.00
3.5 R	4.00	10.00	1.40	20 R	21.80	50.00	4.60
3.5 W	4.00	15.00	1.75	20 W	21.80	66.00	5.10
4 N	4.70	10.00	1.20	24 N	25.60	44.00	4.60
4 R	4.70	12.00	1.40	24 R	25.60	56.00	5.10
4 W	4.70	16.00	2.30	24 W	25.60	72.00	5.60
5 N	5.50	11.00	1.40	30 N	32.40	56.00	5.10
5 R	5.50	15.00	1.75	30 R	32.40	72.00	5.60
5 W	5.50	20.00	2.30	30 W	32.40	90.00	6.40
6 N	6.65	13.00	1.75	36 N	38.30	66.00	5.60
6 R	6.65	18.80	1.75	36 R	38.30	90.00	6.40
6 W	6.65	25.40	2.30	36 W	38.30	110.00	8.50
8 N	8.90	18.80	2.30				
8 R	8.90	25.40	2.30				
8 W	8.90	32.00	2.80				

N = narrow; R = regular; W = wide.
*Same as screw or bolt size.

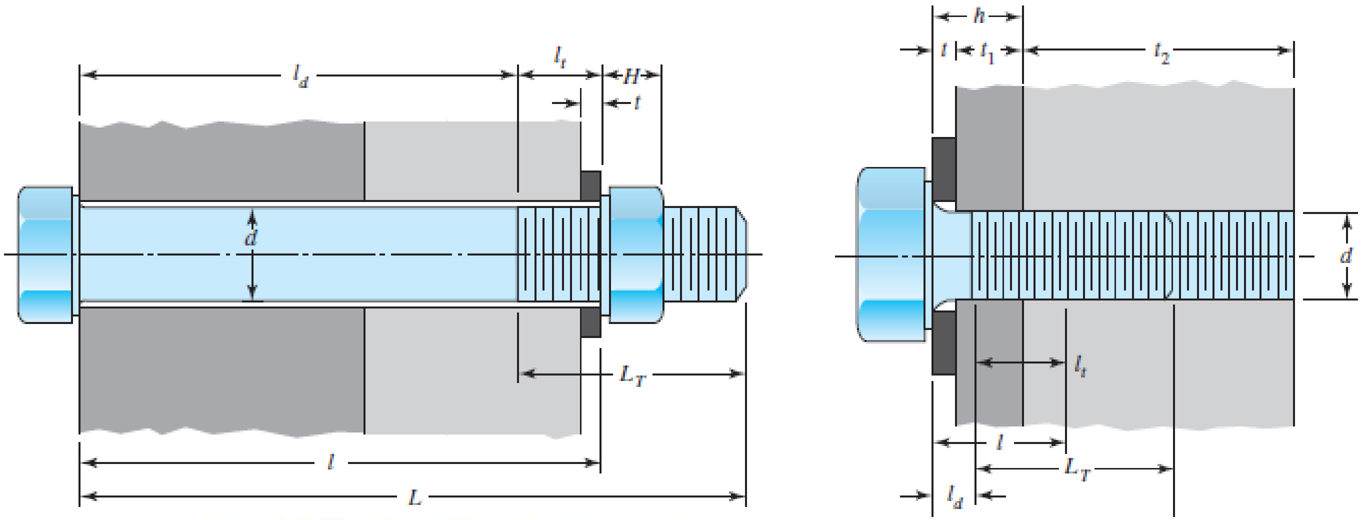
8.4 Joints – Fasteners Stiffness – Table A-31

Table A–31

Dimensions of Hexagonal Nuts

Nominal Size, in	Width W	Height H Regular Hexagonal	Thick or Slotted	JAM
$\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{32}$	$\frac{9}{32}$	$\frac{5}{32}$
$\frac{5}{16}$	$\frac{1}{2}$	$\frac{17}{64}$	$\frac{21}{64}$	$\frac{3}{16}$
$\frac{3}{8}$	$\frac{9}{16}$	$\frac{21}{64}$	$\frac{13}{32}$	$\frac{7}{32}$
$\frac{7}{16}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{29}{64}$	$\frac{1}{4}$
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{5}{16}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{31}{64}$	$\frac{39}{64}$	$\frac{5}{16}$
$\frac{5}{8}$	$\frac{15}{16}$	$\frac{35}{64}$	$\frac{23}{32}$	$\frac{3}{8}$
$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{41}{64}$	$\frac{13}{16}$	$\frac{27}{64}$
$\frac{7}{8}$	$1\frac{5}{16}$	$\frac{3}{4}$	$\frac{29}{32}$	$\frac{31}{64}$
1	$1\frac{1}{2}$	$\frac{55}{64}$	1	$\frac{35}{64}$
$1\frac{1}{8}$	$1\frac{11}{16}$	$\frac{31}{32}$	$1\frac{5}{32}$	$\frac{39}{64}$
$1\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$\frac{23}{32}$
$1\frac{3}{8}$	$2\frac{1}{16}$	$1\frac{11}{64}$	$1\frac{3}{8}$	$\frac{25}{32}$
$1\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{9}{32}$	$1\frac{1}{2}$	$\frac{27}{32}$
Nominal Size, mm				
M5	8	4.7	5.1	2.7
M6	10	5.2	5.7	3.2
M8	13	6.8	7.5	4.0
M10	16	8.4	9.3	5.0
M12	18	10.8	12.0	6.0
M14	21	12.8	14.1	7.0
M16	24	14.8	16.4	8.0
M20	30	18.0	20.3	10.0
M24	36	21.5	23.9	12.0
M30	46	25.6	28.6	15.0
M36	55	31.0	34.7	18.0

8.4 Joints – Fasteners Stiffness



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

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

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

Threaded length L_T: Inch series:

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

Metric series:

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

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

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

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

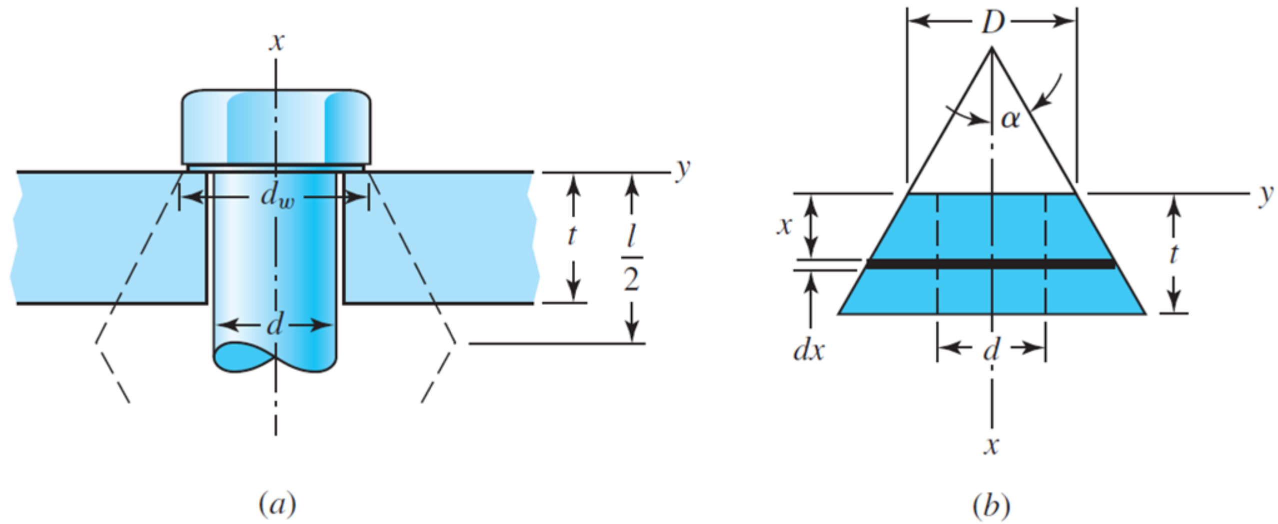
Area of threaded portion: A_t from Table 8–1 or 8–2

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

8.5 Joints – Member Stiffness

Figure 8–15

Compression of a member with the equivalent elastic properties represented by a frustum of a hollow cone. Here, l represents the grip length.



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

8.5 Joints – Member Stiffness

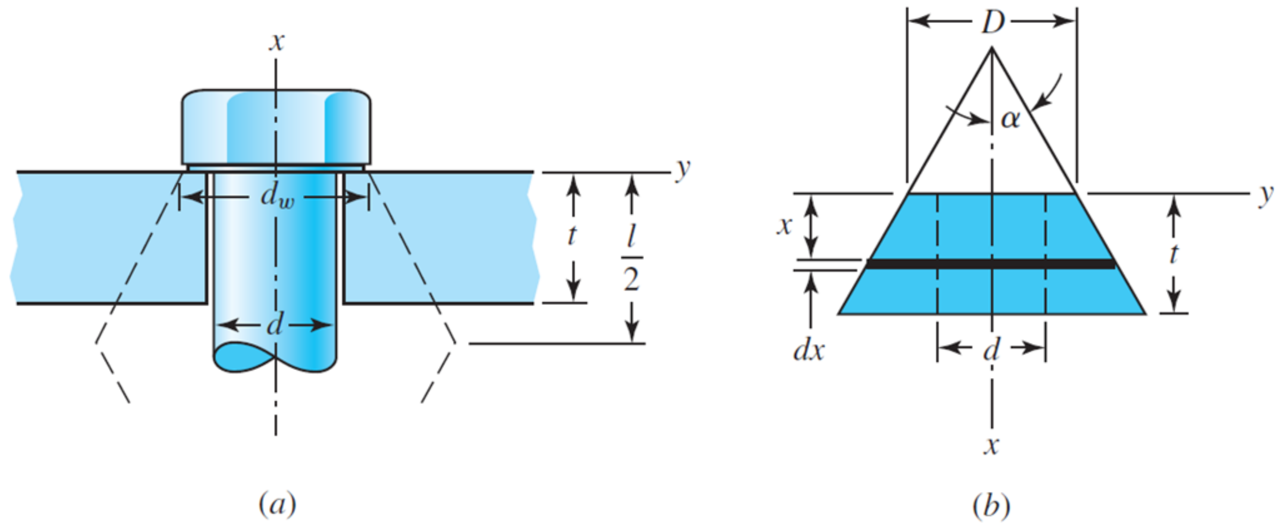


Figure 8–15

Compression of a member with the equivalent elastic properties represented by a frustum of a hollow cone. Here, l represents the grip length.

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

8.5 Joints – Member Stiffness

EXAMPLE 8–2

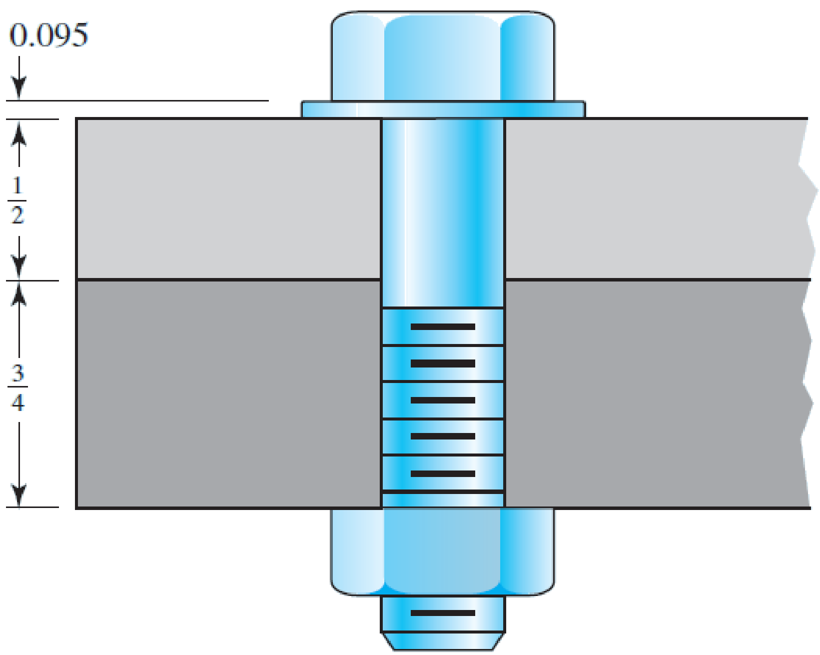
As shown in Fig. 8–17*a*, two plates are clamped by washer-faced $\frac{1}{2}$ in-20 UNF × $1\frac{1}{2}$ in SAE grade 5 bolts each with a standard $\frac{1}{2}$ N steel plain washer.

(*a*) Determine the member spring rate k_m if the top plate is steel and the bottom plate is gray cast iron.

(*b*) Using the method of conical frusta, determine the member spring rate k_m if both plates are steel.

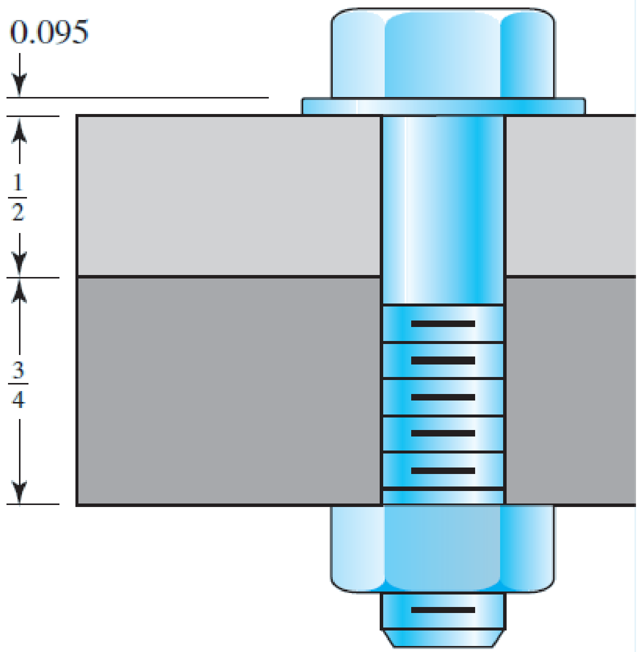
(*c*) Using Eq. (8–23), determine the member spring rate k_m if both plates are steel. Compare the results with part (*b*).

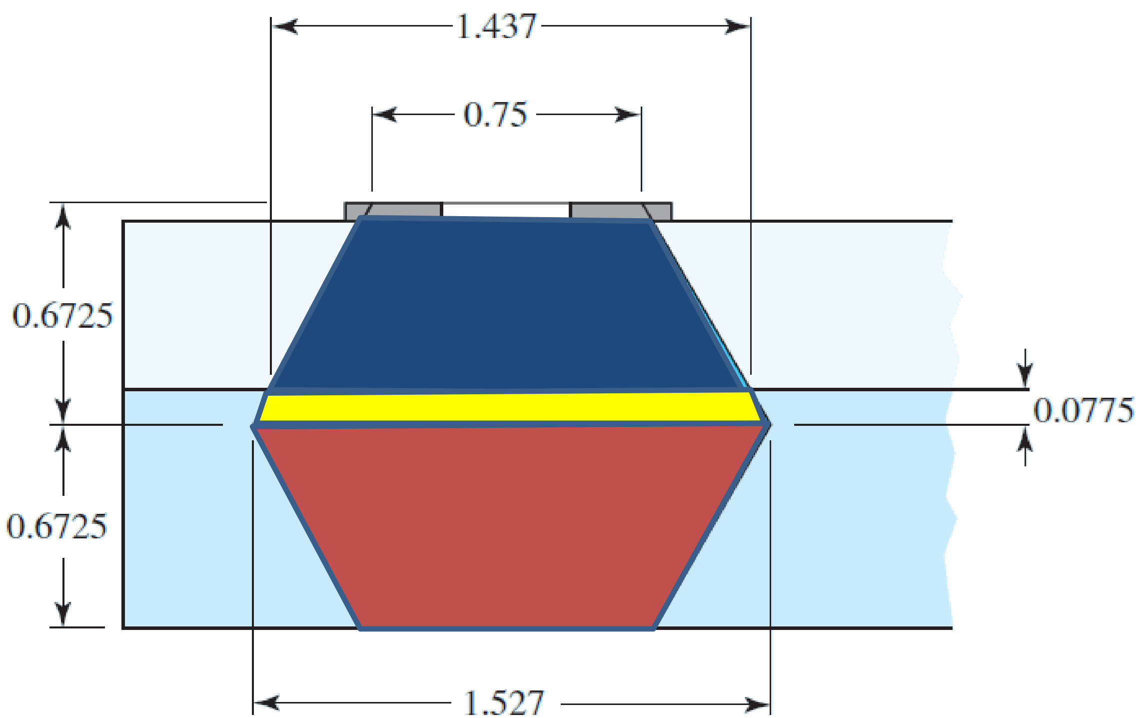
(*d*) Determine the bolt spring rate k_b.



(*a*)

8.5 Joints – Member Stiffness

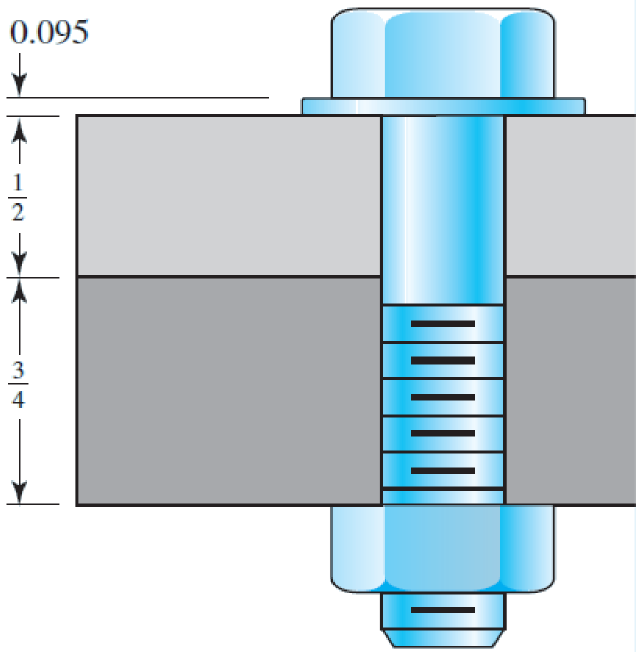




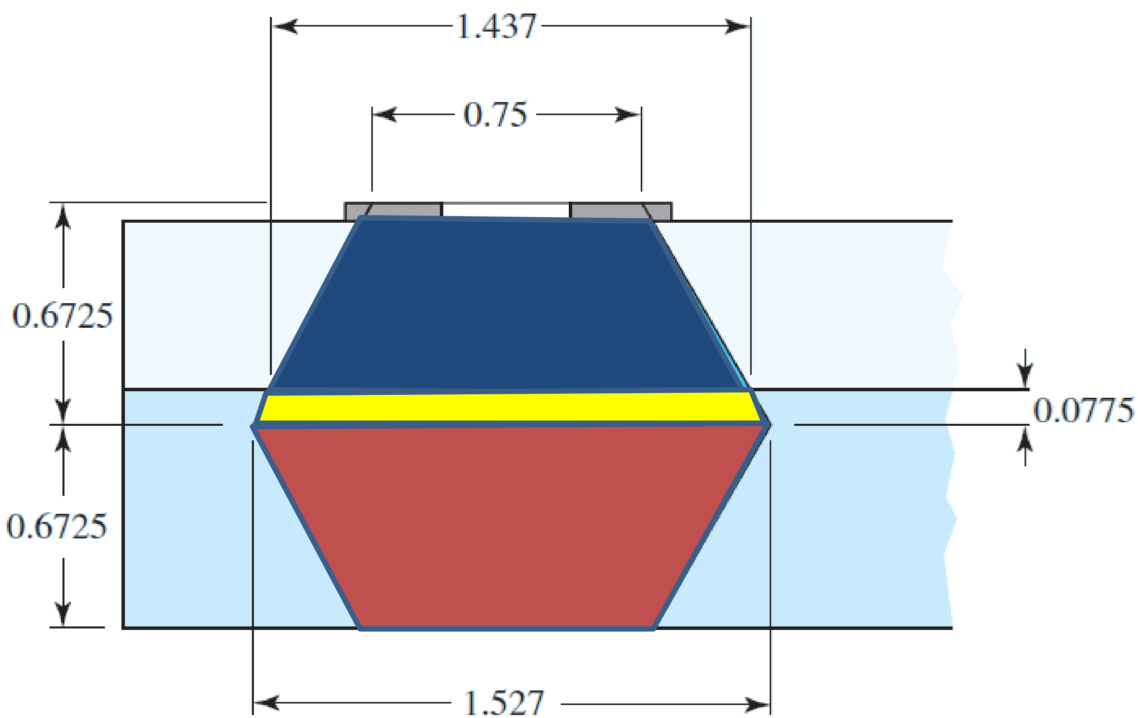
The fruste extend halfway into the joint the distance

$$\frac{1}{2}(0.5 + 0.75 + 0.095) = 0.6725 \text{ in}$$

8.5 Joints – Member Stiffness

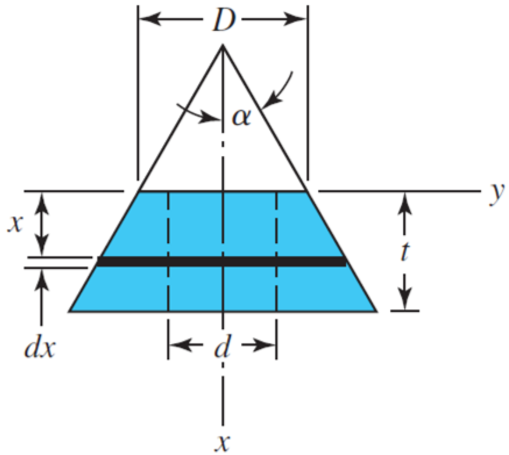


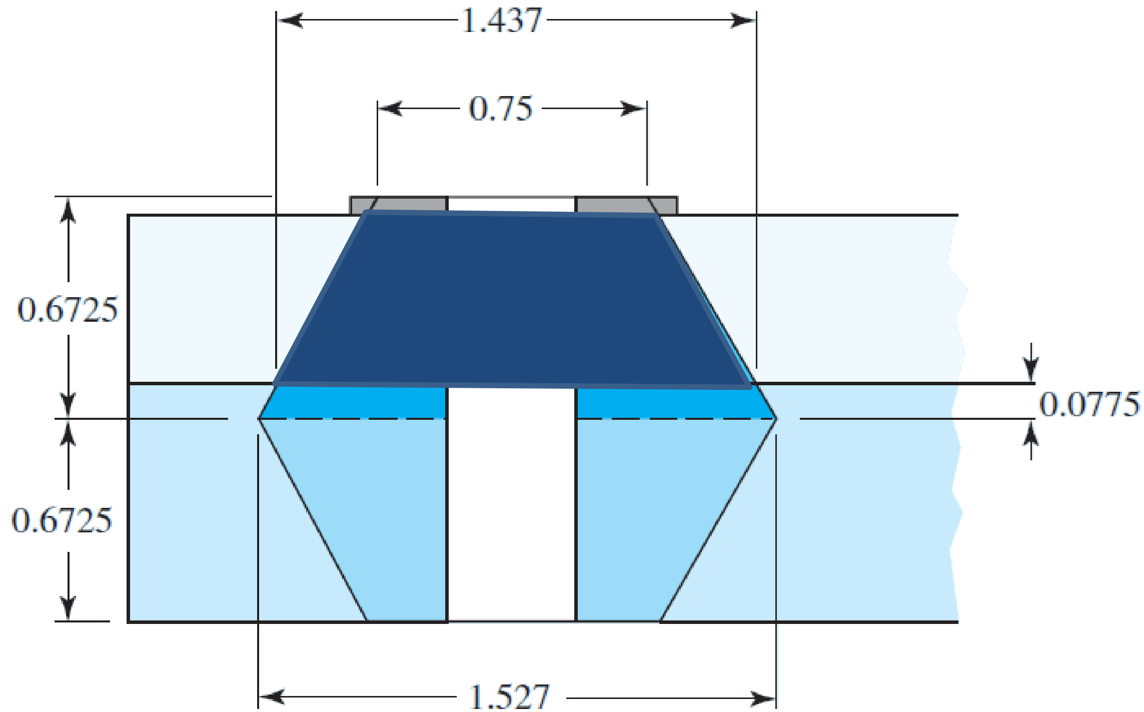
(a)



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

8.5 Joints – Member Stiffness





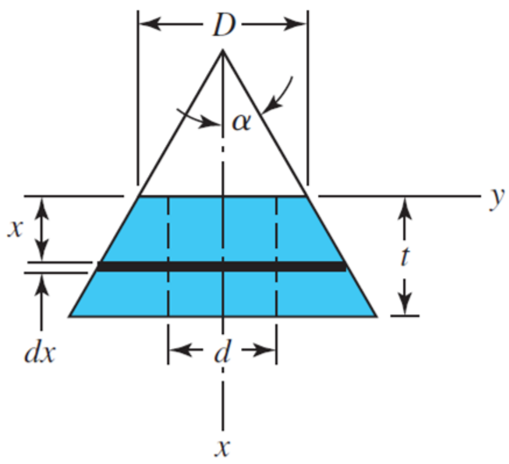
$$k = \frac{0.5774\pi E d}{\ln \dfrac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

t = 0.6725 – 0.0775 = 0.595 in

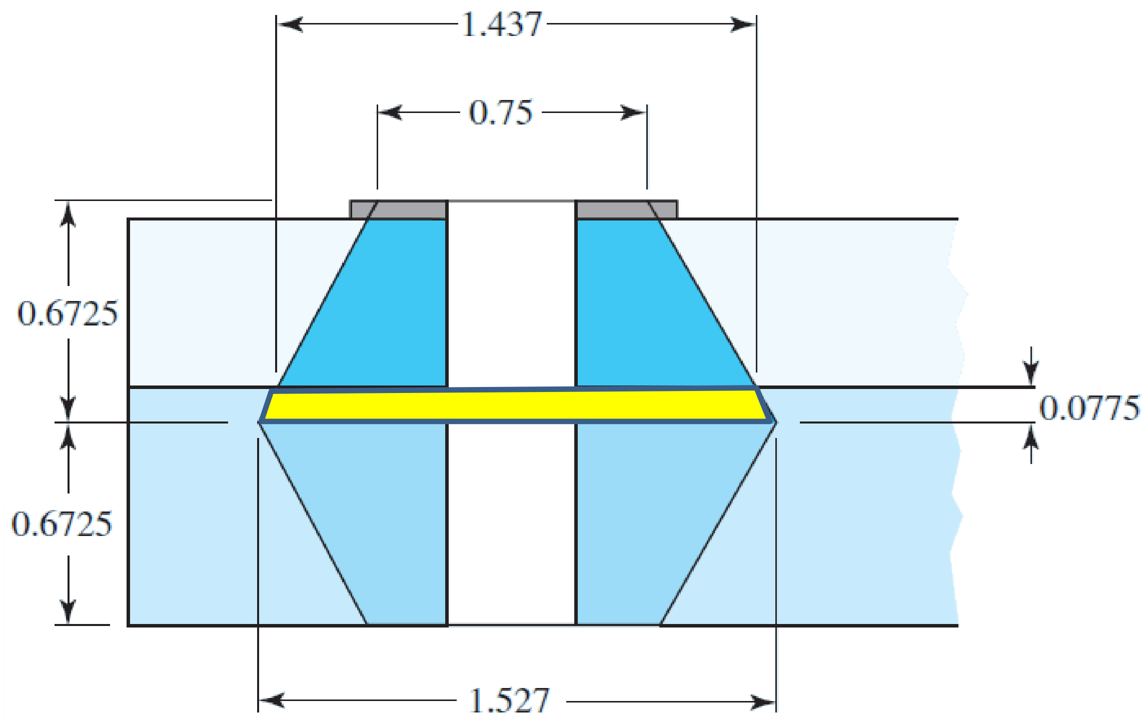
D = 0.75 in

$$k_1 = \frac{0.5774\pi(30)(10^6)0.5}{\ln\left\{\dfrac{[1.155(0.595) + 0.75 - 0.5](0.75 + 0.5)}{[1.155(0.595) + 0.75 + 0.5](0.75 - 0.5)}\right\}} = 30.80(10^6)\ \text{lbf/in}$$

8.5 Joints – Member Stiffness



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

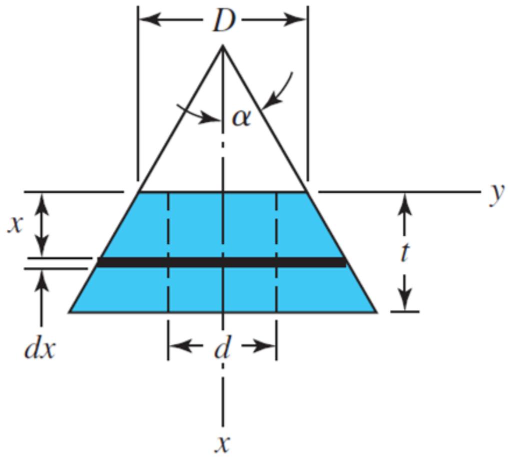


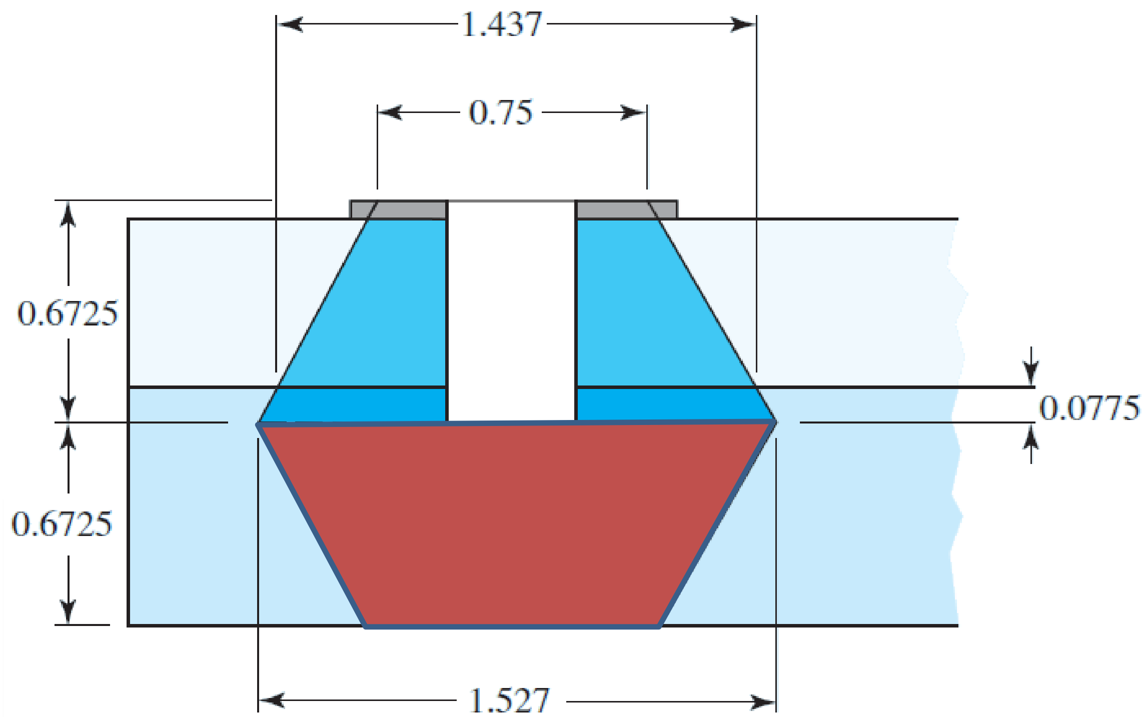
t = 0.0775 in

D = 0.75 + 2(0.595) tan30 = 1.437 in

$$k_2 = \frac{0.5774\pi(14.5)(10^6)0.5}{\ln\left\{\dfrac{[1.155(0.0775) + 1.437 - 0.5](1.437 + 0.5)}{[1.155(0.0775) + 1.437 + 0.5](1.437 - 0.5)}\right\}} = 285.5(10^6)\ \text{lbf/in}$$

8.5 Joints – Member Stiffness



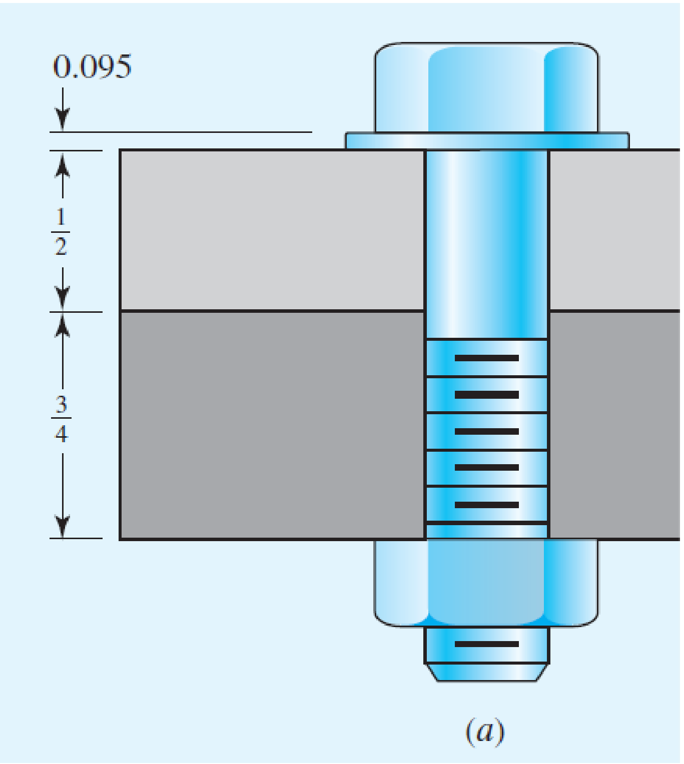


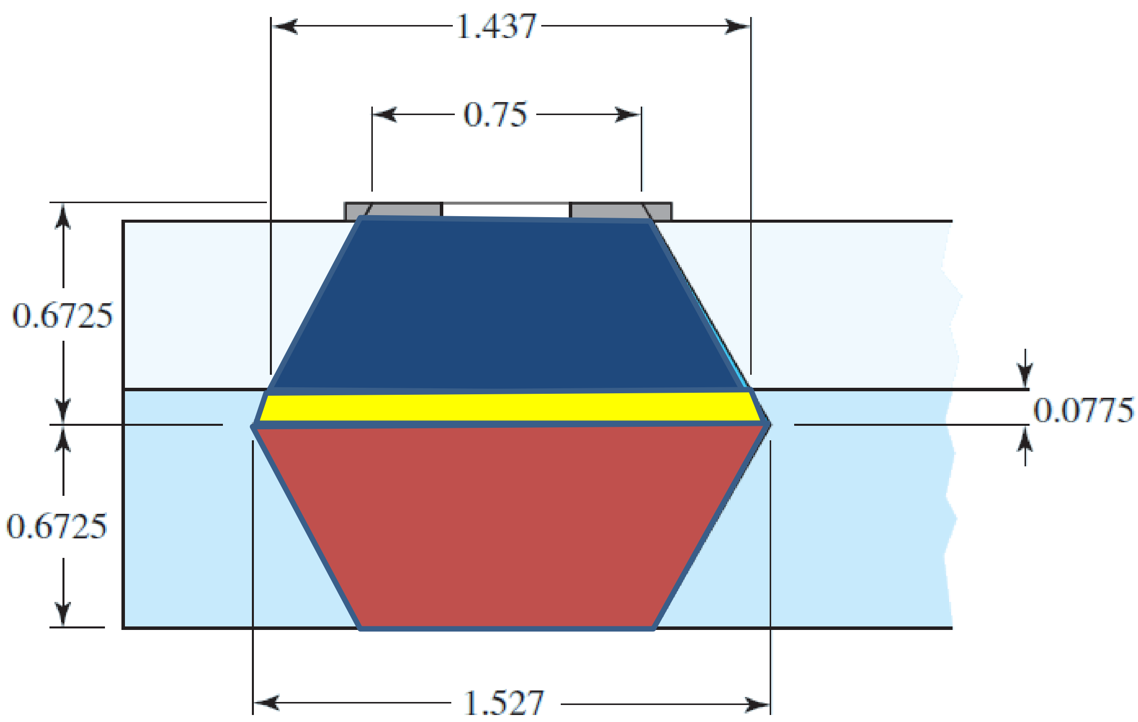
$$k = \frac{0.5774\pi Ed}{\ln\dfrac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

t = 0.6725 in
D = 0.75 in

$$k_3 = \frac{0.5774\pi(14.5)(10^6)0.5}{\ln\left\{\dfrac{[1.155(0.6725) + 0.75 - 0.5](0.75 + 0.5)}{[1.155(0.6725) + 0.75 + 0.5](0.75 - 0.5)}\right\}} = 14.15(10^6)\ \text{lbf/in}$$

8.5 Joints – Member Stiffness



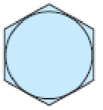
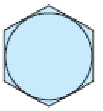
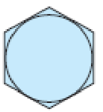
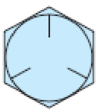
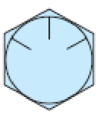
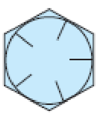
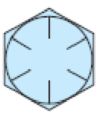
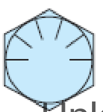


$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \cdots + \frac{1}{k_i}$$

8.6 Bolt Strength

Table 8–9

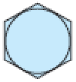
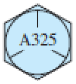
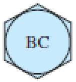
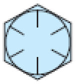
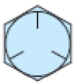
SAE Specifications for Steel Bolts

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	

8.6 Bolt Strength

Table 8–10

ASTM Specifications for Steel Bolts

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

8.6 Bolt Strength

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

8.6 Bolt Strength

8.7 Tension Joints – The External Load

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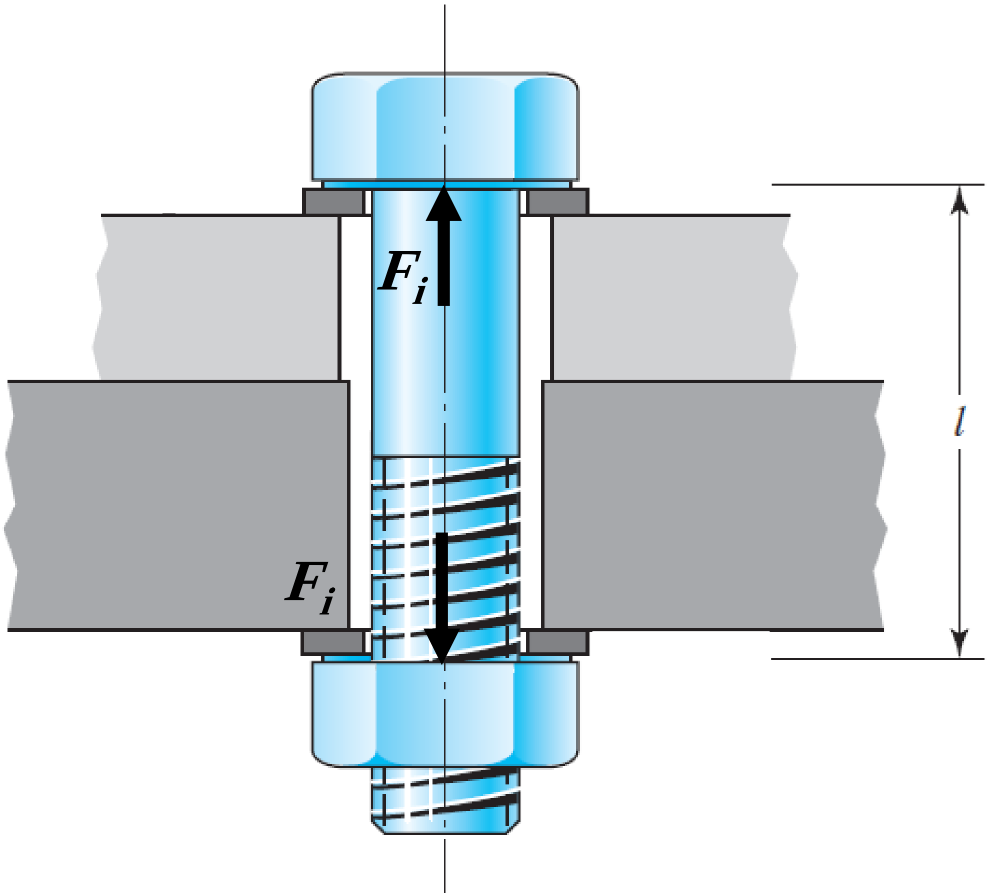
Figure 8–13

A bolted connection loaded in tension by the forces P. Note the use of two washers. Note how the threads extend into the body of the connection. This is usual and is desired. l is the grip of the connection.



P = external tensile load per bolt

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

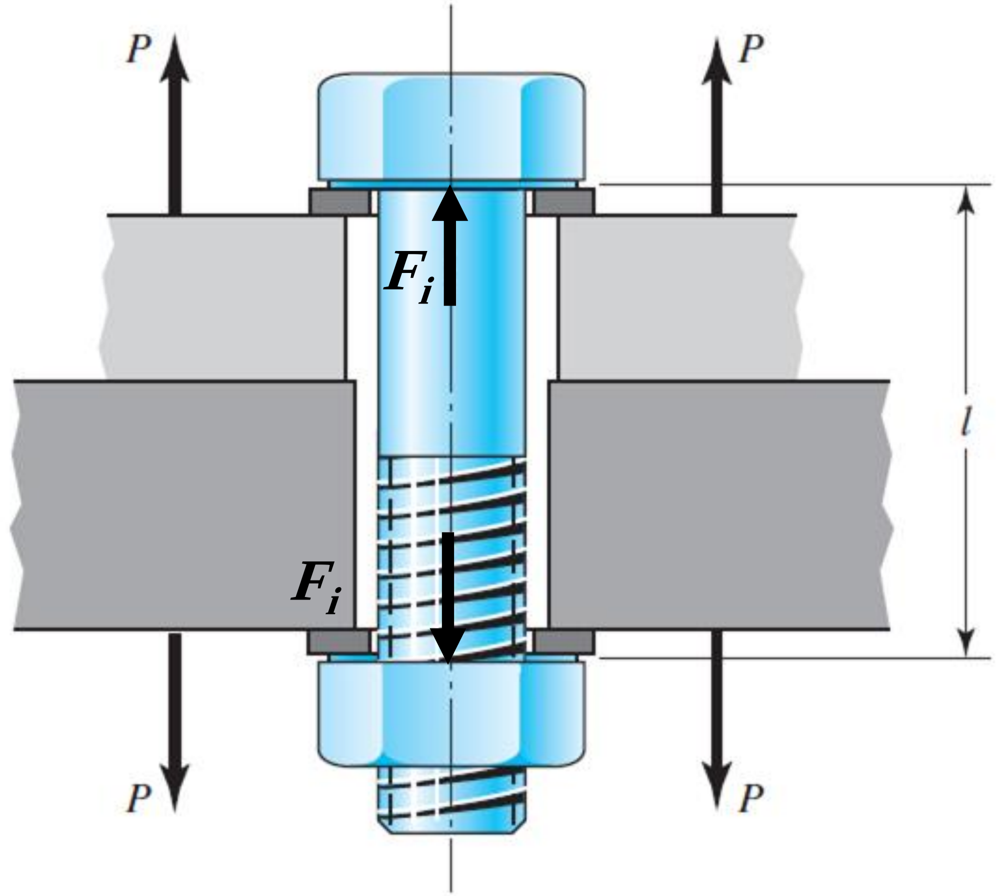
8.7 Tension Joints – The External Load

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

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

Since $P = P_b + P_m$, we have

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



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

Stiffness constant of the joint

8.7 Tension Joints – The External Load

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P = external tensile load per bolt

P_b = portion of P taken by bolt

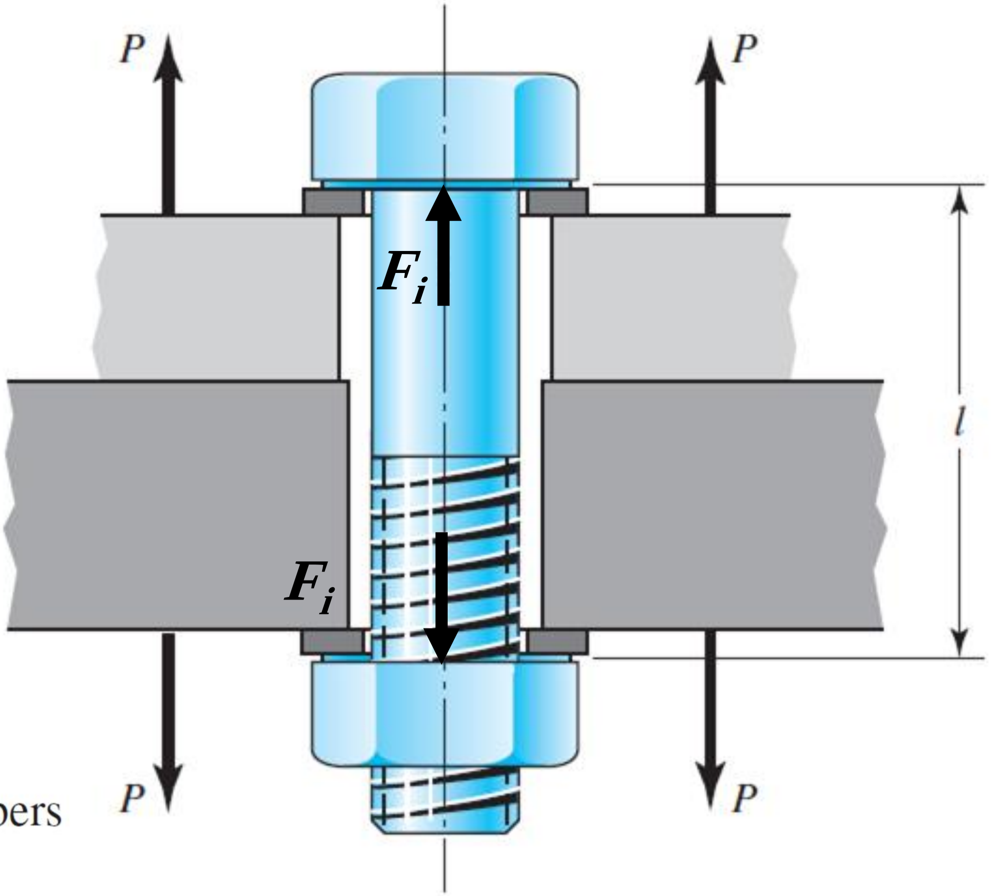
P_m = portion of P taken by members

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

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

$F_b = P_b + F_i$ = resultant bolt load

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



$$F_b = P_b + F_i = CP + F_i \qquad F_m < 0 \qquad (8\text{–}24)$$

?

$$F_m = P_m - F_i = (1 - C)P - F_i \qquad F_m < 0 \qquad (8\text{–}25)$$

8.8 Relating Bolt Torque to Bolt Tension

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- **Torque required to produce a given preload**

$$T = \frac{F_i d_m}{2}\left(\frac{\tan\lambda + f\sec\alpha}{1 - f\tan\lambda\sec\alpha}\right) + \frac{F_i f_c d_c}{2}$$

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

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

Torque Coefficient ▪

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

Table 8–15

Torque Factors K for Use with Eq. (8–27)

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

8.8 Relating Bolt Torque to Bolt Tension

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.

(a) The preload stress is

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

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

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

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

8.1 Thread Standards and Definitions

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Table 8–2

Diameters and Area of Unified Screw Threads UNC and UNF*

Size Designation	Nominal Major Diameter in	Coarse Series—UNC Threads per Inch N	Coarse Series—UNC Tensile-Stress Area A_t in²	Coarse Series—UNC Minor-Diameter Area A_r in²	Fine Series—UNF Threads per Inch N	Fine Series—UNF Tensile-Stress Area A_t in²	Fine Series—UNF Minor-Diameter Area A_r in²
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	$\boxed{0.373}$	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

8.8 Relating Bolt Torque to Bolt Tension

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 $f = f_c = 0.15$.

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

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

Table 8–15

Torque Factors K for Use with Eq. (8–27)

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

8.8 Relating Bolt Torque to Bolt Tension

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EXAMPLE 8–3 A $\frac{3}{4}$ in-16 UNF × $2\frac{1}{2}$ 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 $f = f_c = 0.15$.

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

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

8.1 Thread Standards and Definitions

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Table 8–2

Diameters and Area of Unified Screw Threads UNC and UNF*

		Coarse Series—UNC			Fine Series—UNF		
Size Designation	Nominal Major Diameter in	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2
0	0.0600				80	0.001 80	0.001 51
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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	$\boxed{0.351}$ d_r
$\frac{7}{8}$	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

8.8 Relating Bolt Torque to Bolt Tension

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 $f = f_c = 0.15$.

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

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

For $\alpha = 30^\circ$, Eq. (8–26) gives

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

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

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

8.9 Statically Loaded Tension Joint with Preload

1- Yielding Factor of Safety Against Static Stress

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

2- Load Factor of Safety Against Static Stress (Applied to load P against overloading)

$$\frac{Cn_L P + F_i}{A_t} = S_p \quad \Rightarrow \quad n_L = \frac{S_p A_t - F_i}{CP}$$

8.9 Statically Loaded Tension Joint with Preload

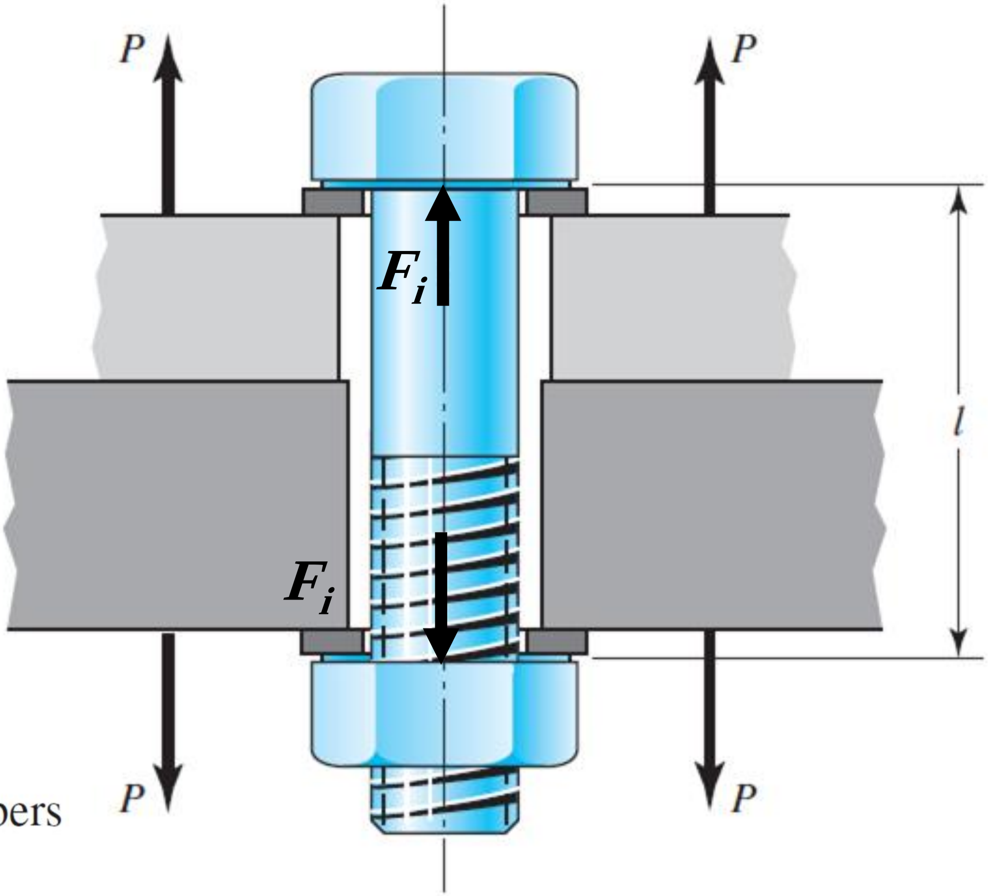
3- Factor of Safety Against Joint Separation

8.7 Tension Joints – The External Load

P = external tensile load per bolt

P_b = portion of P taken by bolt

P_m = portion of P taken by members



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

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

$F_b = P_b + F_i$ = resultant bolt load

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

$$F_b = P_b + F_i = CP + F_i \qquad F_m < 0 \qquad (8\text{–}24)$$

?

$$F_m = P_m - F_i = (1 - C)P - F_i \qquad F_m < 0 \qquad (8\text{–}25)$$

8.9 Statically Loaded Tension Joint with Preload

3- Factor of Safety Against Joint Separation

$$F_m = P_m - F_i = (1 - C)P - F_i \qquad F_m < 0 \qquad (8\text{–}25)$$

Let P_0 be the value of the external load that would cause joint separation.

At separation, $F_m = 0$ in Eq. (8–25), and so

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

$$n_0 = \frac{P_0}{P} = \frac{F_i}{P(1 - C)}$$

8.9 Statically Loaded Tension Joint with Preload

$$F_i = \begin{cases} 0.75F_p & \text{for nonpermanent connections, reused fasteners} \\ 0.90F_p & \text{for permanent connections} \end{cases} \qquad (8\text{–}31)$$

where F_p is the proof load, obtained from the equation

$$F_p = A_t S_p \qquad (8\text{–}32)$$

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

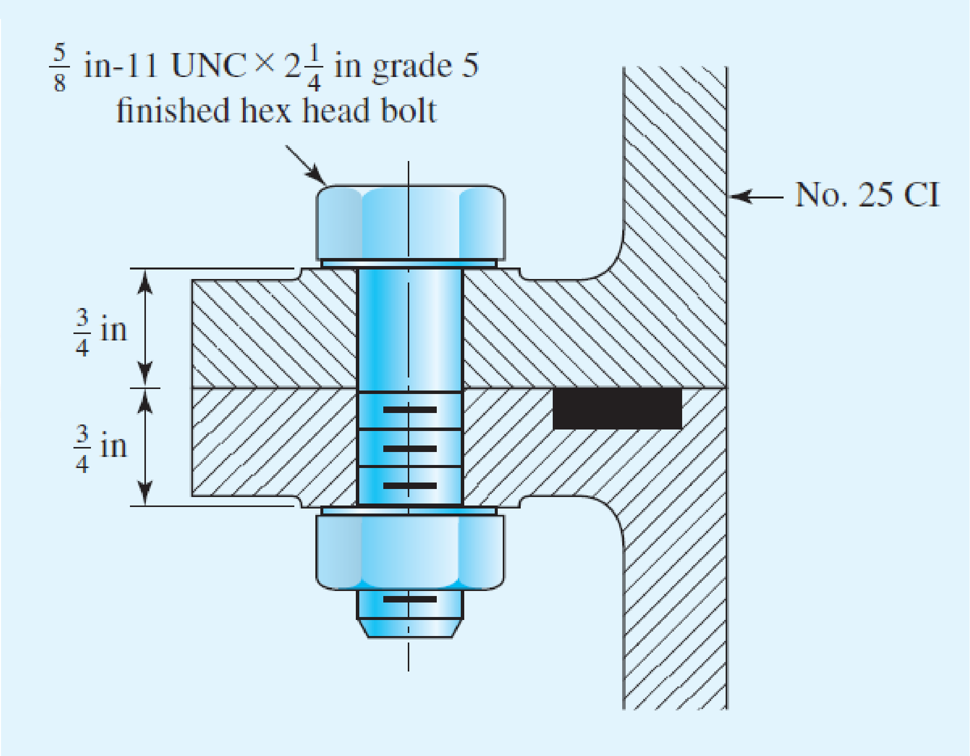
8.9 Statically Loaded Tension Joint with Preload

EXAMPLE 8–4 Figure 8–19 is a cross section of a grade 25 cast-iron pressure vessel. A total of N bolts are to be used to resist a separating force of 36 kip.

(*a*) Determine k_b, k_m, and C.

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

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



8.9 Statically Loaded Tension Joint with Preload

EXAMPLE 8–4 Figure 8–19 is a cross section of a grade 25 cast-iron pressure vessel. A total of N bolts are to be used to resist a separating force of 36 kip.

(*a*) Determine k_b, k_m, and C.

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

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

(b)

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

$$F_i = \begin{cases} 0.75F_p & \text{for nonpermanent connections, reused fasteners} \\ 0.90F_p & \text{for permanent connections} \end{cases} \qquad (8\text{–}31)$$

where F_p is the proof load, obtained from the equation

$$F_p = A_t S_p \qquad (8\text{–}32)$$

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

8.9 Statically Loaded Tension Joint with Preload

EXAMPLE 8–4 Figure 8–19 is a cross section of a grade 25 cast-iron pressure vessel. A total of N bolts are to be used to resist a separating force of 36 kip.

(*a*) Determine k_b, k_m, and C.

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

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

(b)

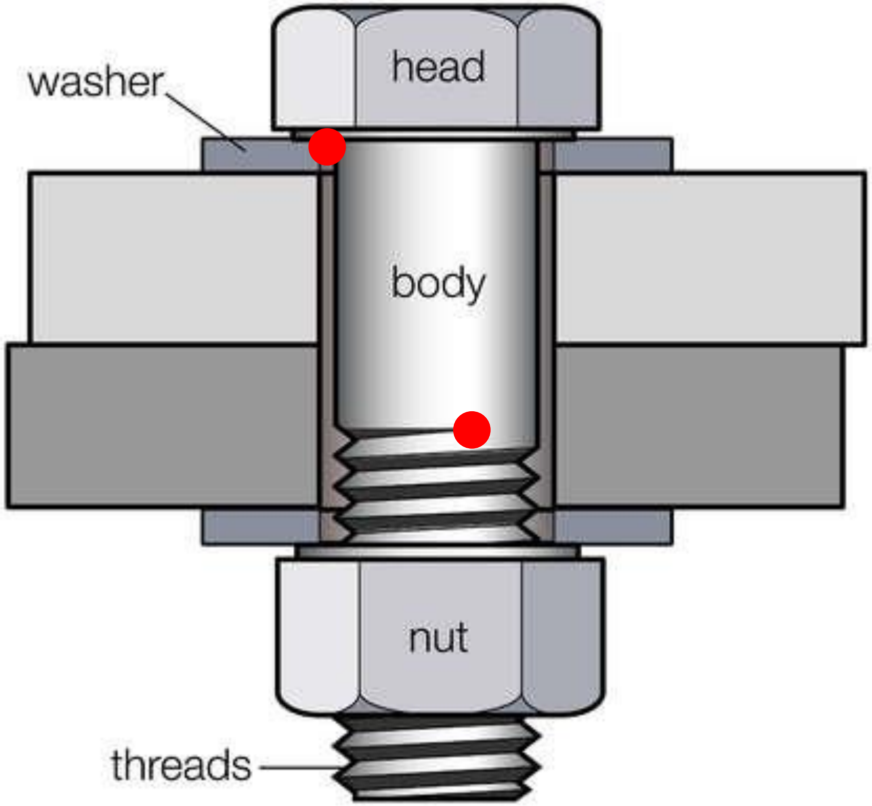
$$n_L = \frac{S_p A_t - F_i}{CP}$$

$$n_L = \frac{S_p A_t - F_i}{C(P_{\text{total}}/N)}$$

$$N = \frac{C n_L P_{\text{total}}}{S_p A_t - F_i} = \frac{0.368(2)(36)}{85(0.226) - 14.4} = 5.52$$

8.11 Fatigue Loading of Tension Joints

8.3 Threaded Fasteners



8.11 Fatigue Loading of Tension Joints

Table 8–16

Fatigue Stress-Concentration Factors K_f for Threaded Elements

SAE Grade	Metric Grade	Rolled Threads	Cut Threads	Fillet
0 to 2	3.6 to 5.8	2.2	2.8	2.1
4 to 8	6.6 to 10.9	3.0	3.8	2.3

Table 8–17

Fully Corrected Endurance Strengths for Bolts and Screws with Rolled Threads*

Grade or Class	Size Range	Endurance Strength
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

8.11 Fatigue Loading of Tension Joints

$$F_{b\min} = CP_{\min} + F_i$$

$$F_{b\max} = CP_{\max} + F_i$$

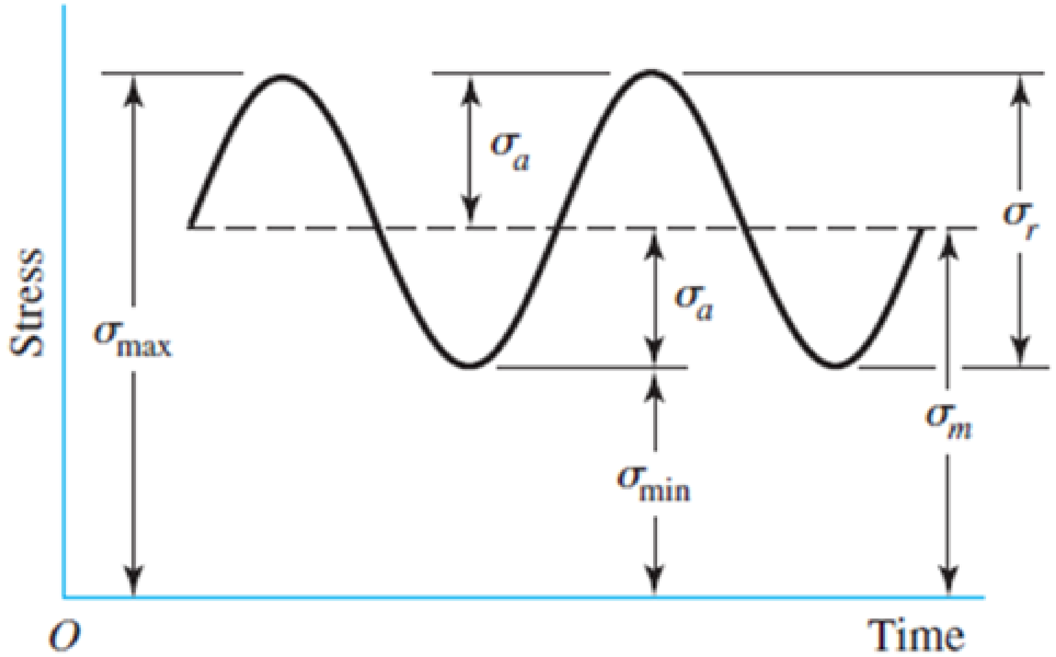
$$\sigma_a = \frac{(F_{b\max} - F_{b\min})/2}{A_t} = \frac{(CP_{\max} + F_i) - (CP_{\min} + F_i)}{2A_t}$$

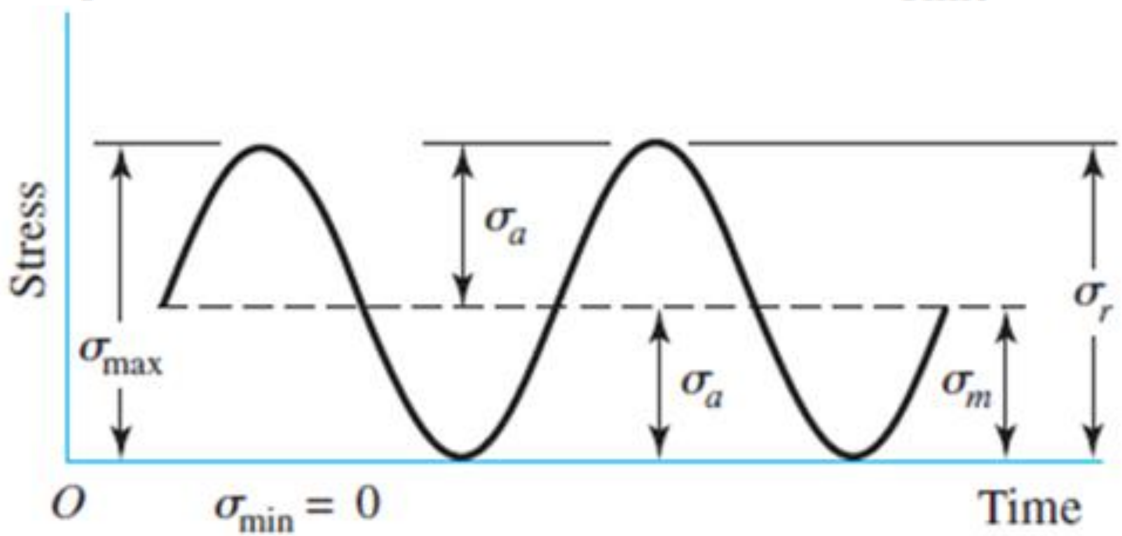
$$\sigma_a = \frac{C(P_{\max} - P_{\min})}{2A_t} \qquad (8\text{–}35)$$

$$\sigma_m = \frac{(F_{b\max} + F_{b\min})/2}{A_t} = \frac{(CP_{\max} + F_i) + (CP_{\min} + F_i)}{2A_t}$$

$$\sigma_m = \frac{C(P_{\max} + P_{\min})}{2A_t} + \frac{F_i}{A_t} \qquad (8\text{–}36)$$

8.11 Fatigue Loading of Tension Joints

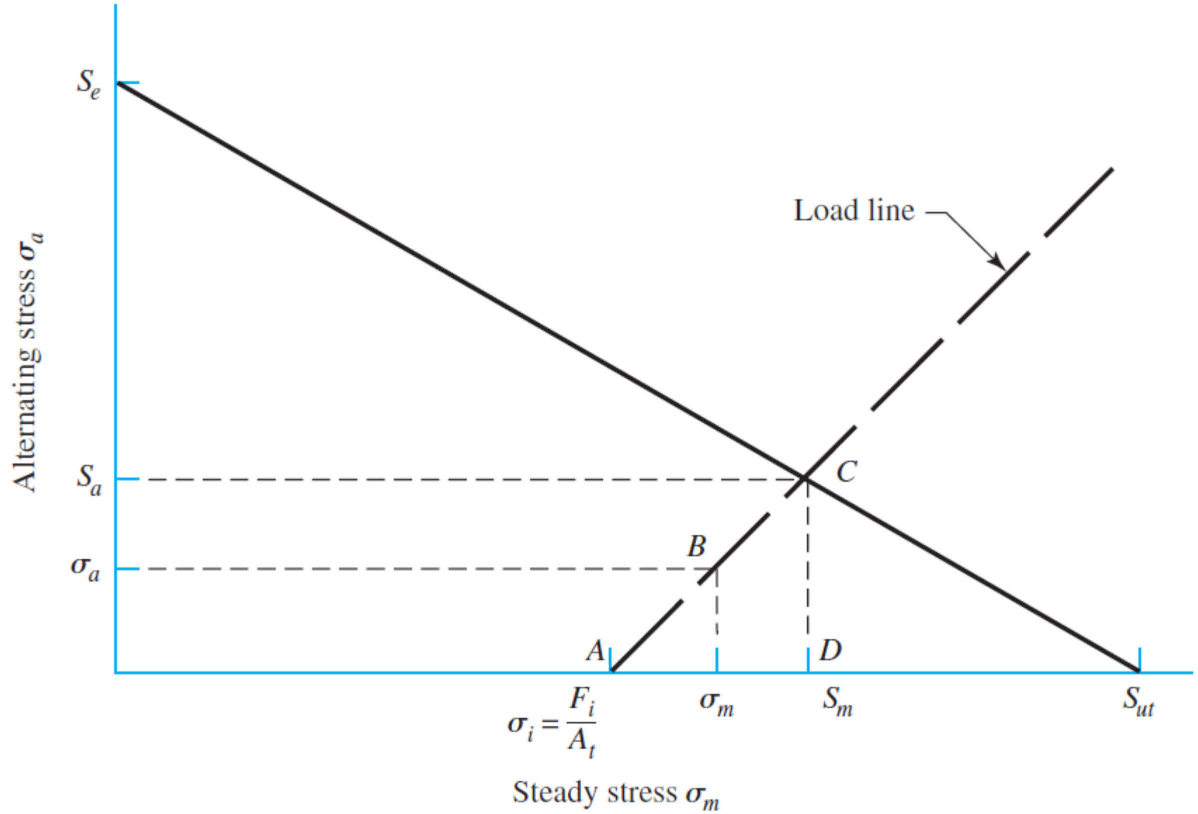




8.11 Fatigue Loading of Tension Joints

Figure 8–20

Designer's fatigue diagram showing a Goodman failure line and a commonly used load line for a constant preload and a fluctuating load.



$$\frac{S_a}{S_e} + \frac{S_m}{S_{ut}} = 1 \qquad (8\text{–}42)$$

Gerber:

$$\frac{S_a}{S_e} + \left(\frac{S_m}{S_{ut}}\right)^2 = 1 \qquad (8\text{–}43)$$

8.11 Fatigue Loading of Tension Joints

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Goodman:

$$n_f = \frac{S_e(S_{ut} - \sigma_i)}{\sigma_a(S_{ut} + S_e)} \tag{8–45}$$

Gerber:

$$n_f = \frac{1}{2\sigma_a S_e}\left[S_{ut}\sqrt{S_{ut}^2 + 4S_e(S_e + \sigma_i)} - S_{ut}^2 - 2\sigma_i S_e\right] \tag{8–46}$$

Yielding factor of safety

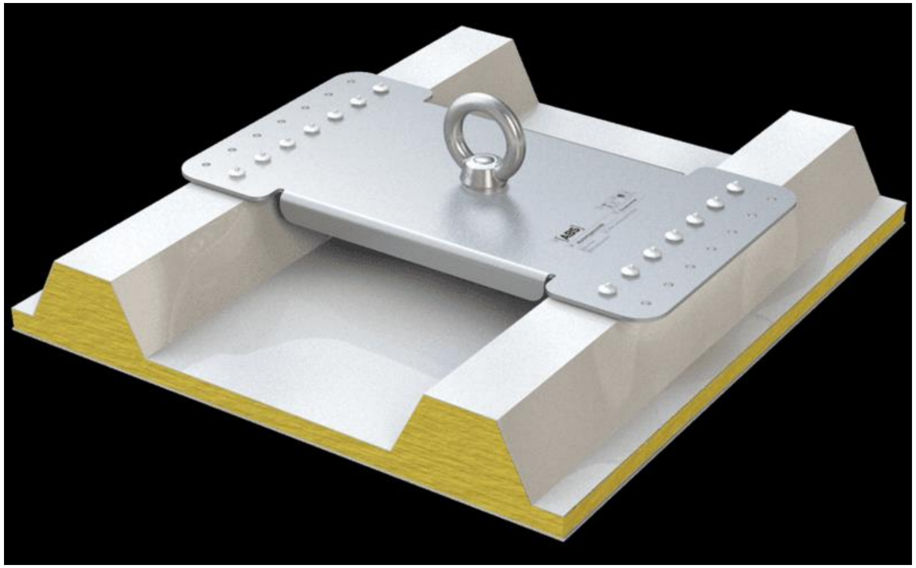
$$n_p = \frac{S_p}{\sigma_m + \sigma_a}$$

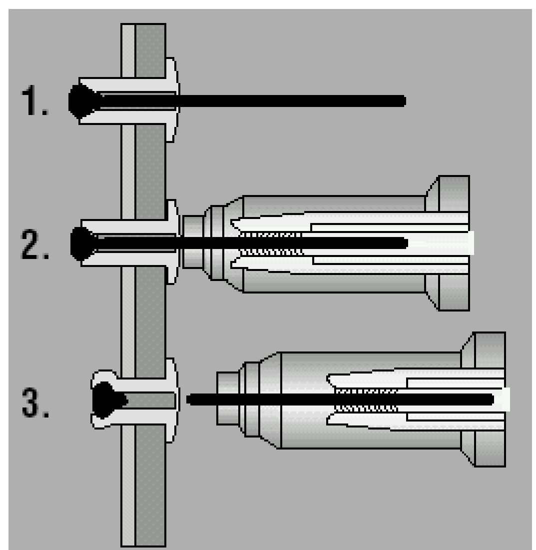
EXAMPLE 8–5 Figure 8–21 shows a connection using cap screws. The joint is subjected to a fluctuating force whose maximum value is 5 kip per screw. The required data are: cap screw, 5/8 in-11 UNC, SAE 5; hardened-steel washer, $t_w = \frac{1}{16}$ in thick; steel cover plate, $t_1 = \frac{5}{8}$ in, $E_s = 30$ 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.

8.12 Bolted and Riveted Joints Loaded in Shear



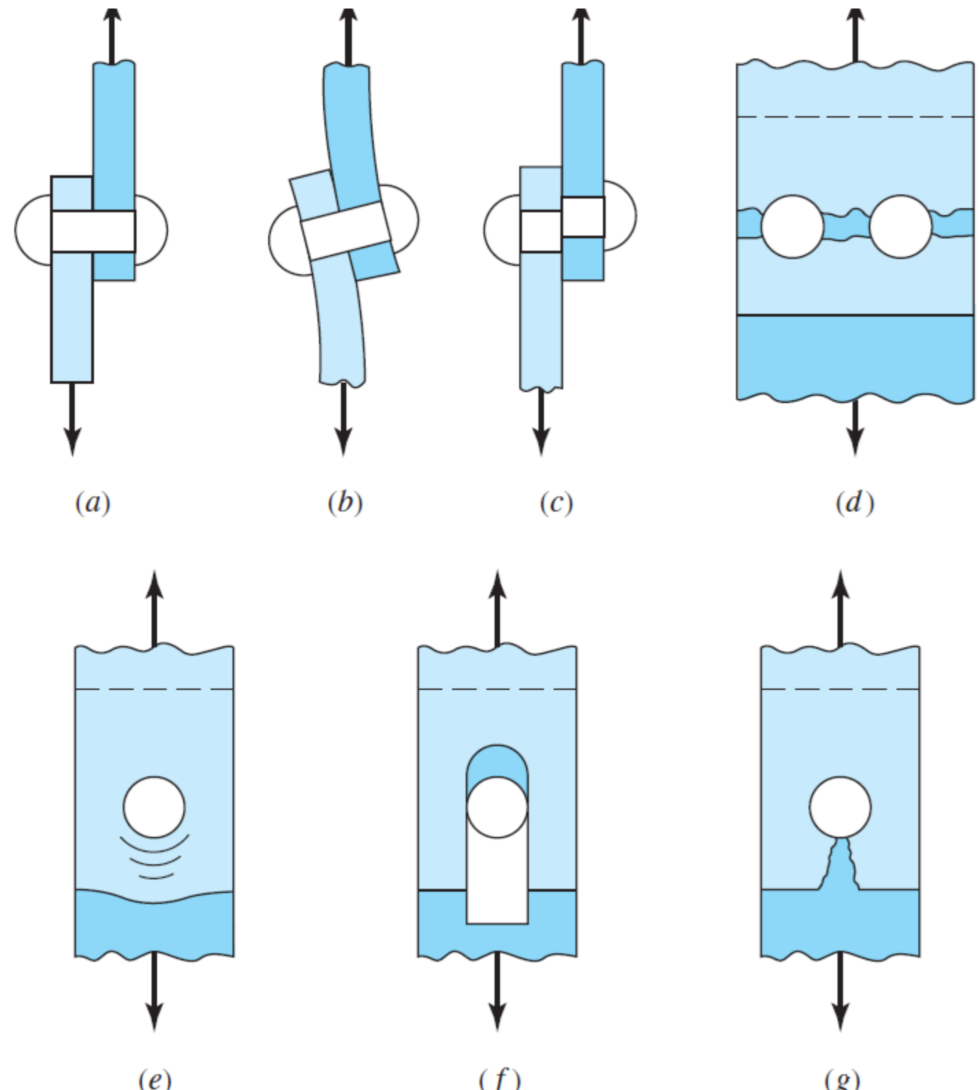


8.12 Bolted and Riveted Joints Loaded in Shear

Riveted and bolted joints loaded in shear are treated exactly alike in design and analysis.

Figure 8–23

Modes of failure in shear loading of a bolted or riveted connection: (*a*) shear loading; (*b*) bending of rivet; (*c*) shear of rivet; (*d*) tensile failure of members; (*e*) bearing of rivet on members or bearing of members on rivet; (*f*) shear tear-out; (*g*) tensile tear-out.



8.12 Bolted and Riveted Joints Loaded in Shear

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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*. Assume the bolt threads do not extend into the joint.

For the $F = 16$ kN load shown find

(*a*) The resultant load on each bolt

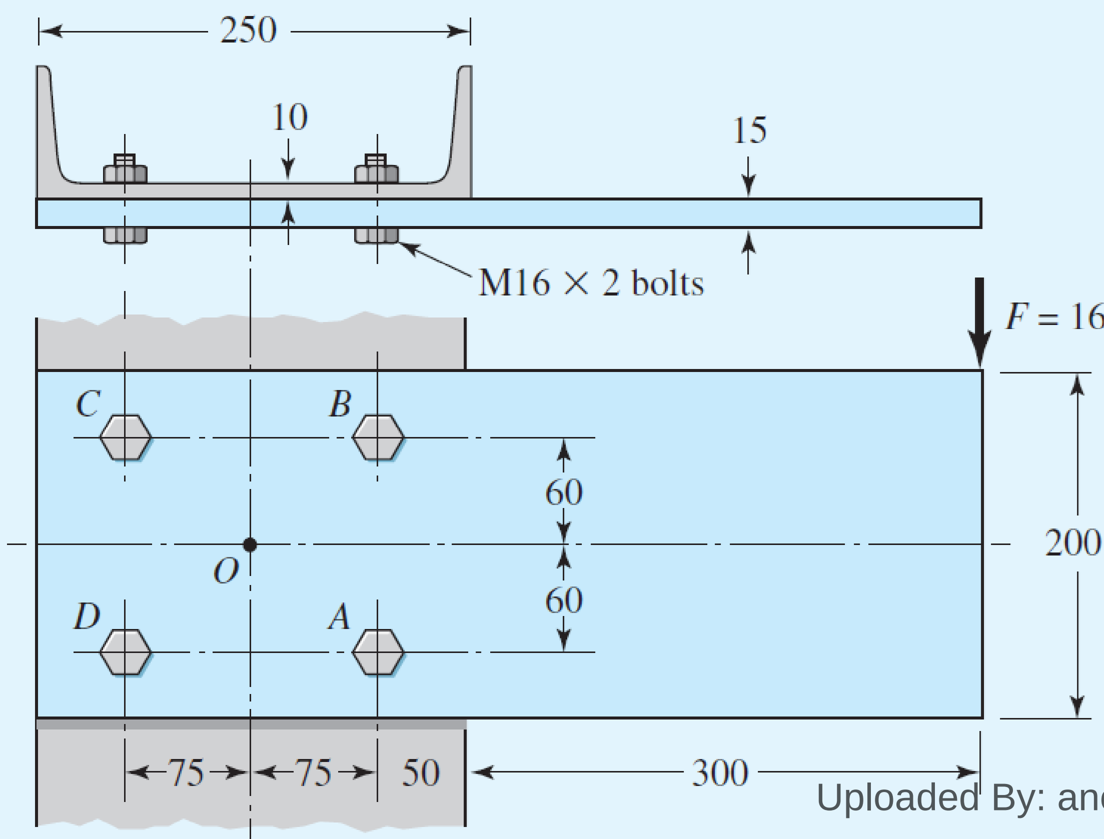
(*b*) The maximum shear stress in each bolt

(*c*) The maximum bearing stress

(*d*) The critical bending stress in the bar

Figure 8–28

Dimensions in millimeters.



8.12 Bolted and Riveted Joints Loaded in Shear

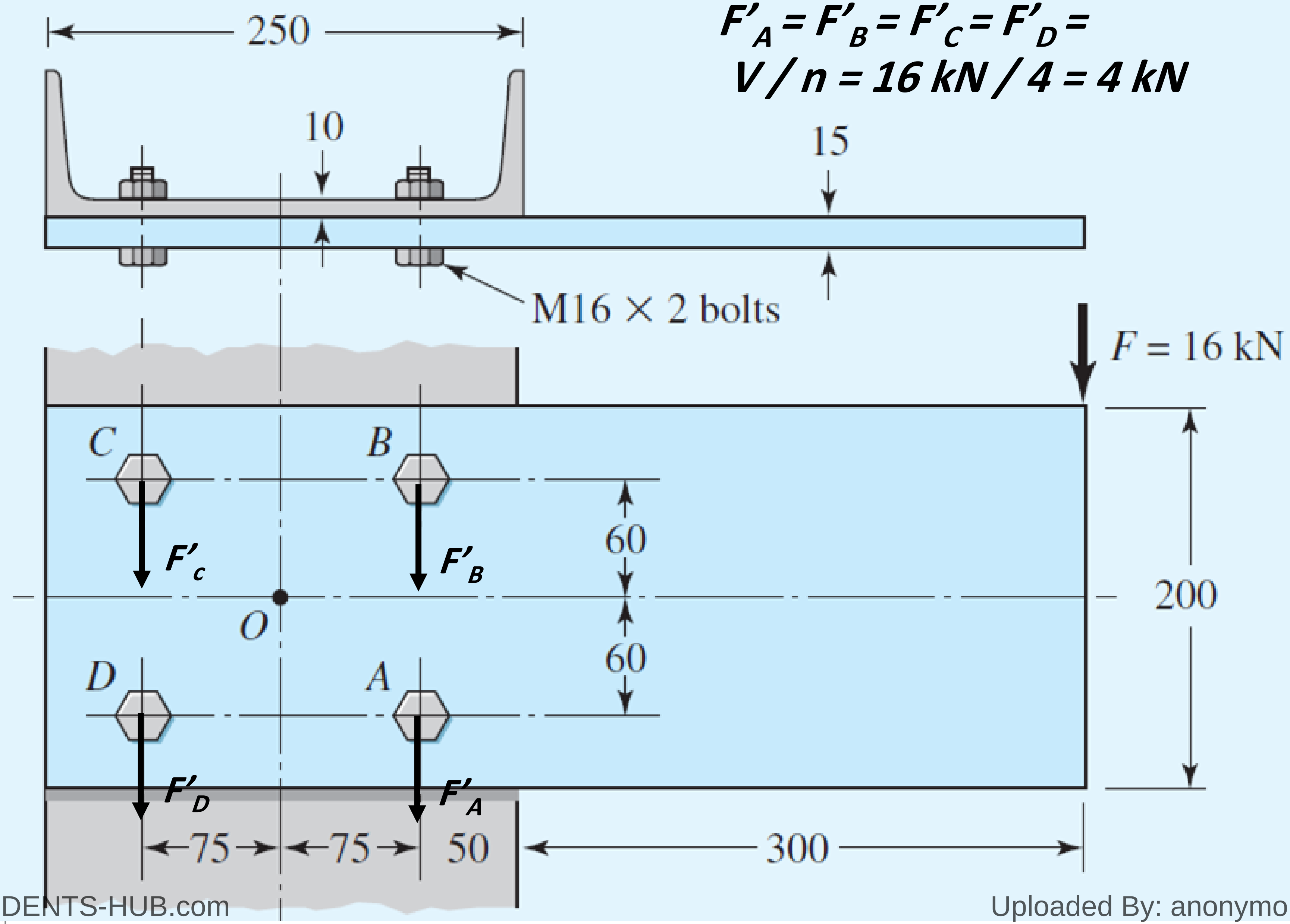
EXAMPLE 8–7

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*. Assume the bolt threads do not extend into the joint.

For the $F = 16$ kN load shown 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

8.12 Bolted and Riveted Joints Loaded in Shear

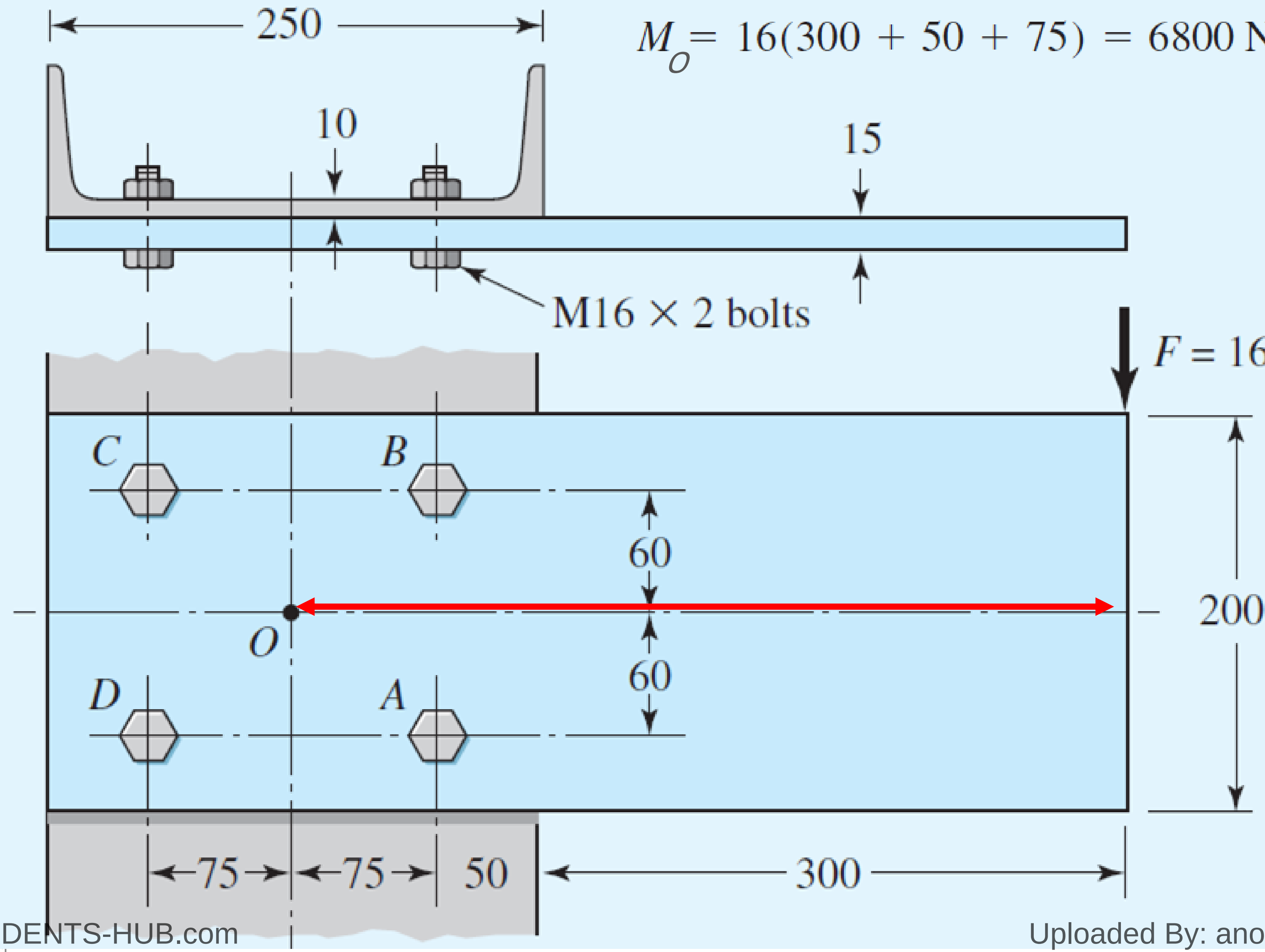


$F'_A = F'_B = F'_C = F'_D =$

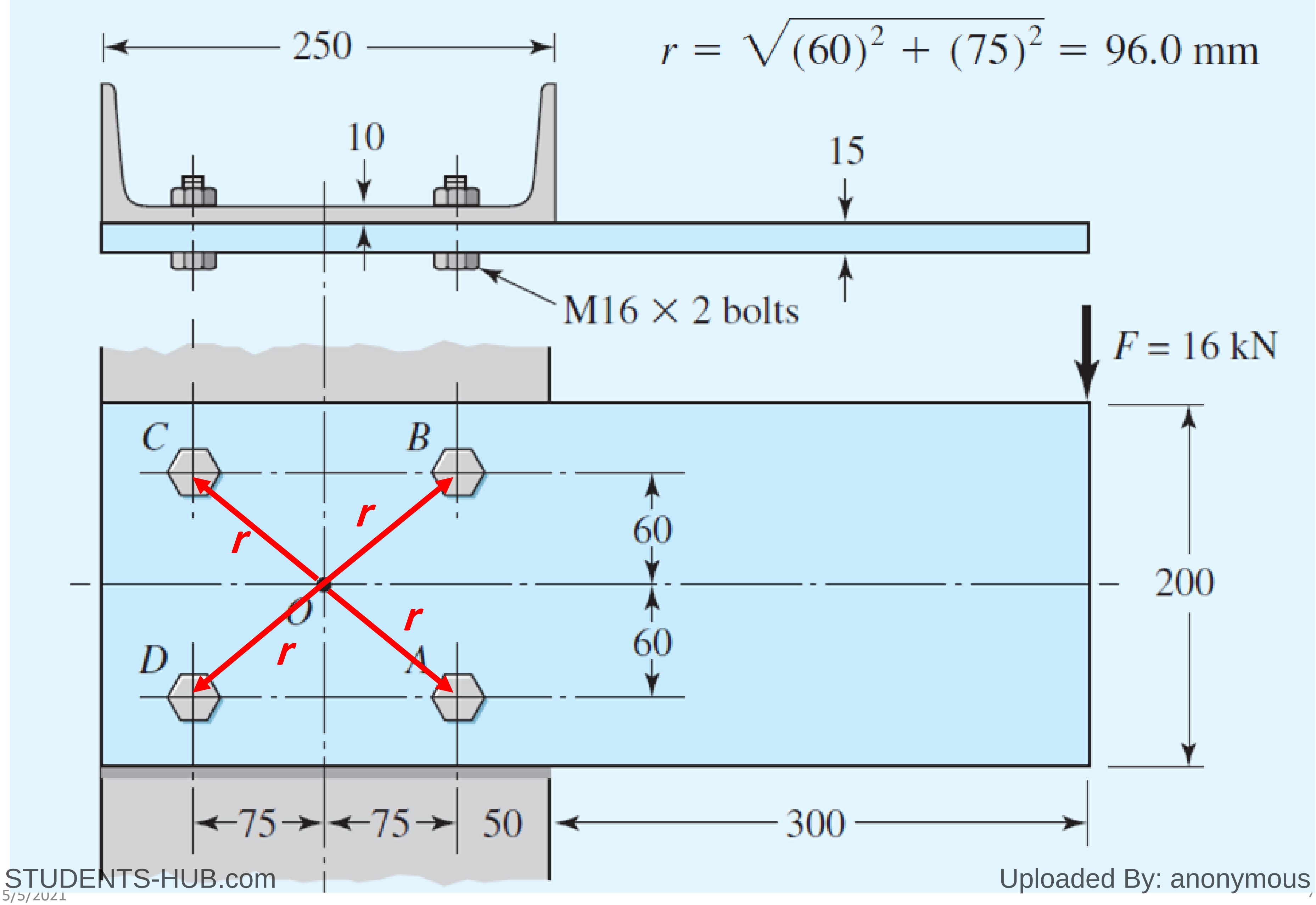
$V / n = 16\ kN / 4 = 4\ kN$

8.12 Bolted and Riveted Joints Loaded in Shear

$$M_O = 16(300 + 50 + 75) = 6800 \text{ N} \cdot \text{m}$$

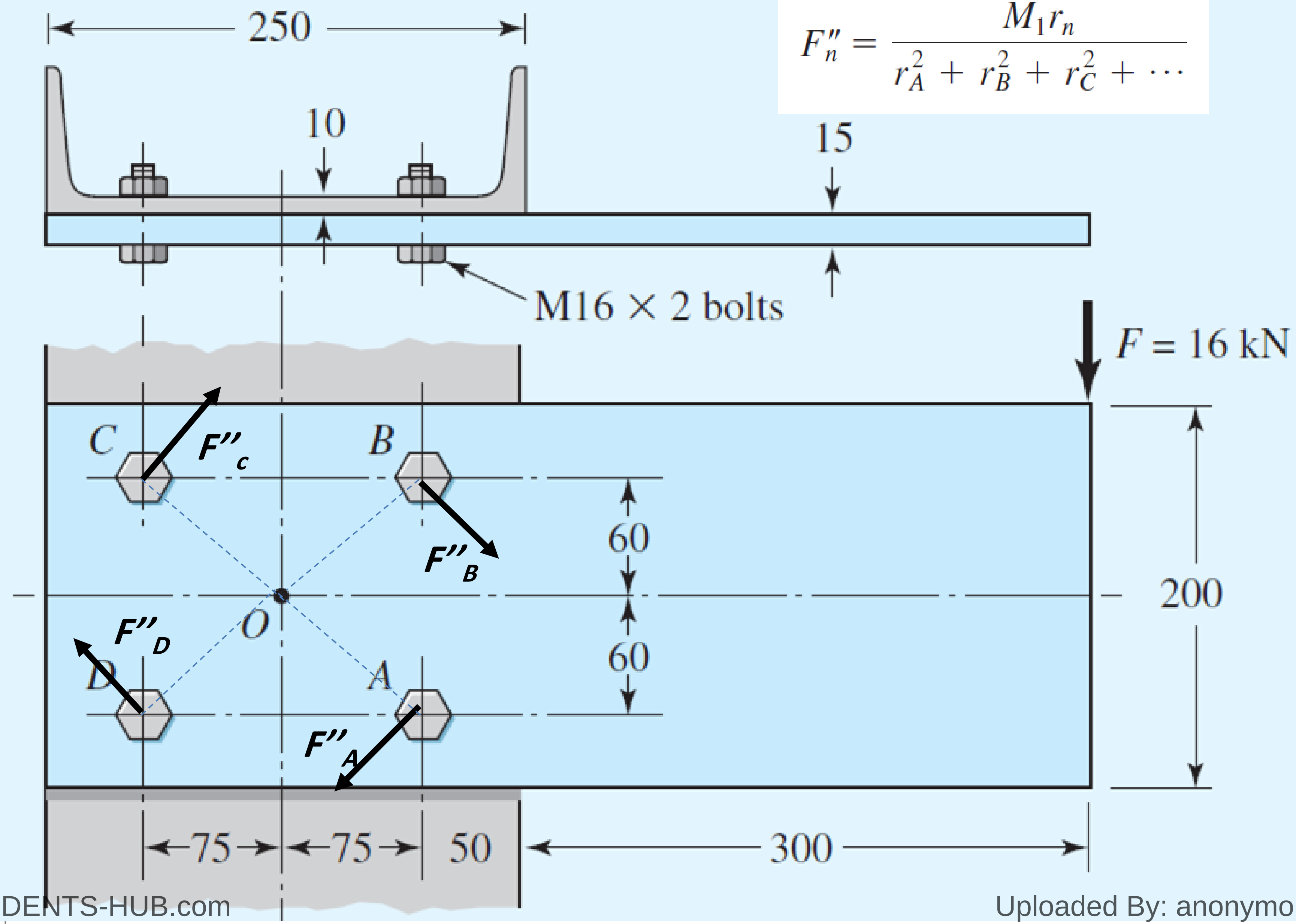


8.12 Bolted and Riveted Joints Loaded in Shear



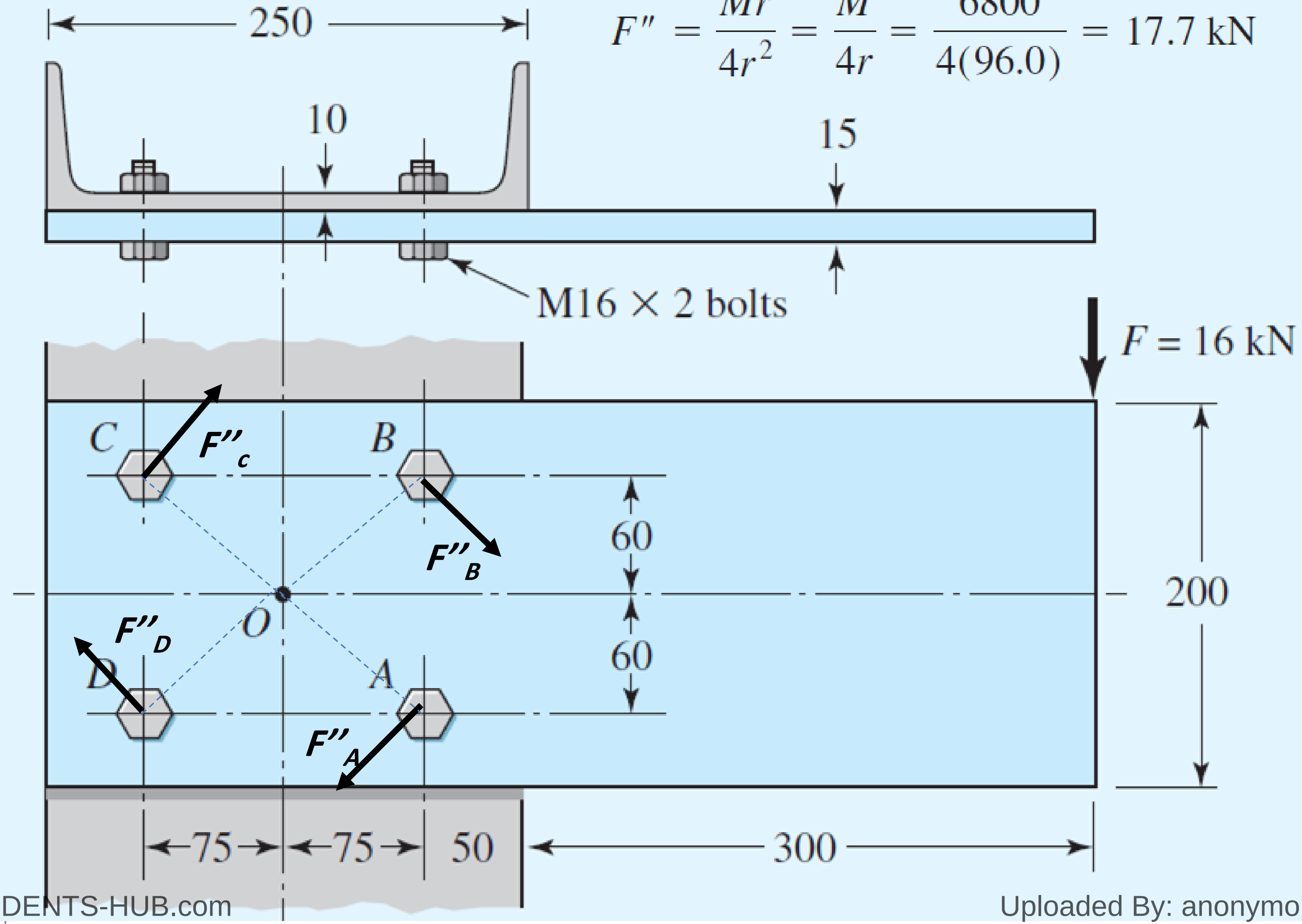
8.12 Bolted and Riveted Joints Loaded in Shear

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



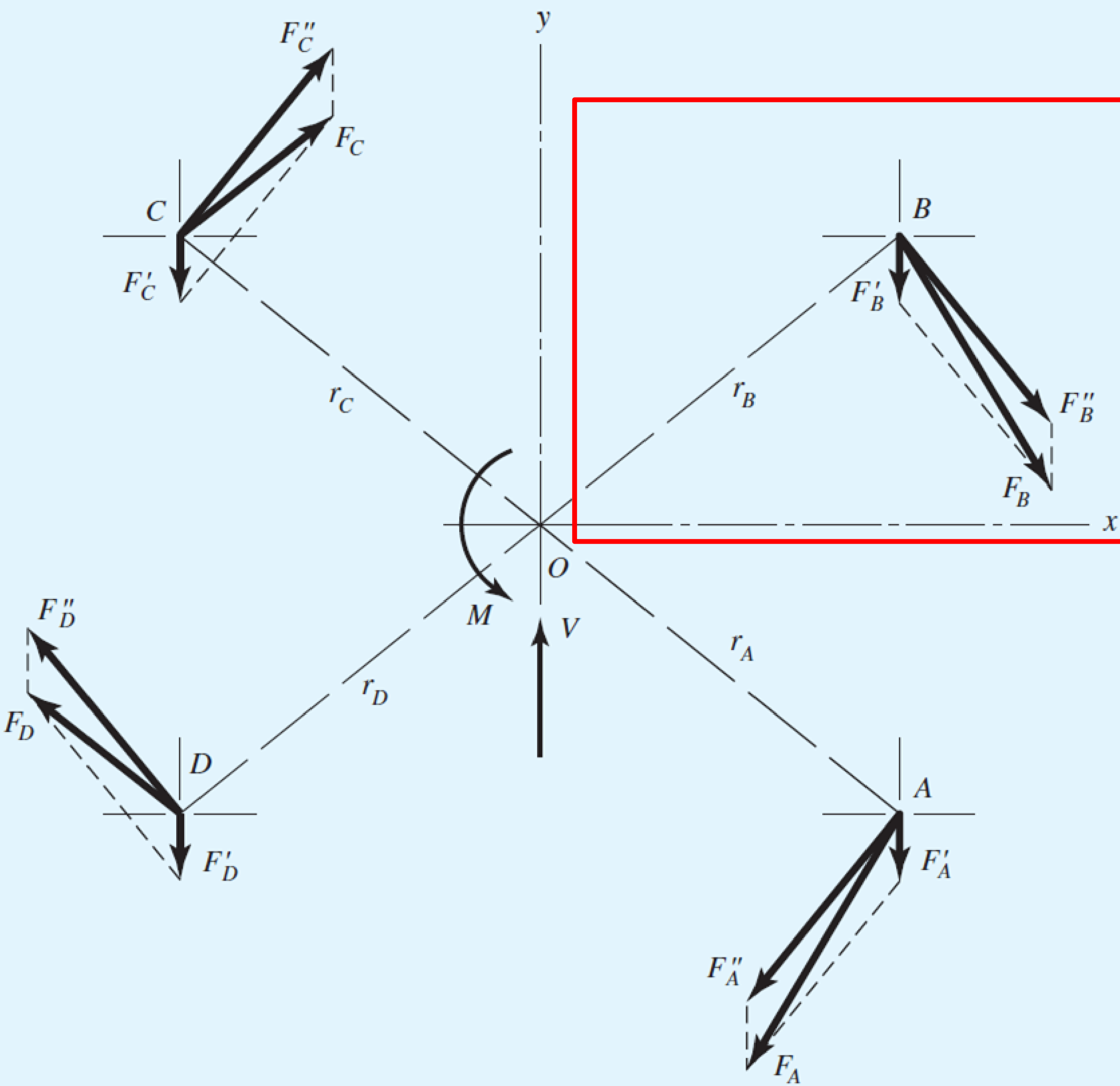
8.12 Bolted and Riveted Joints Loaded in Shear

$$F'' = \frac{Mr}{4r^2} = \frac{M}{4r} = \frac{6800}{4(96.0)} = 17.7 \text{ kN}$$

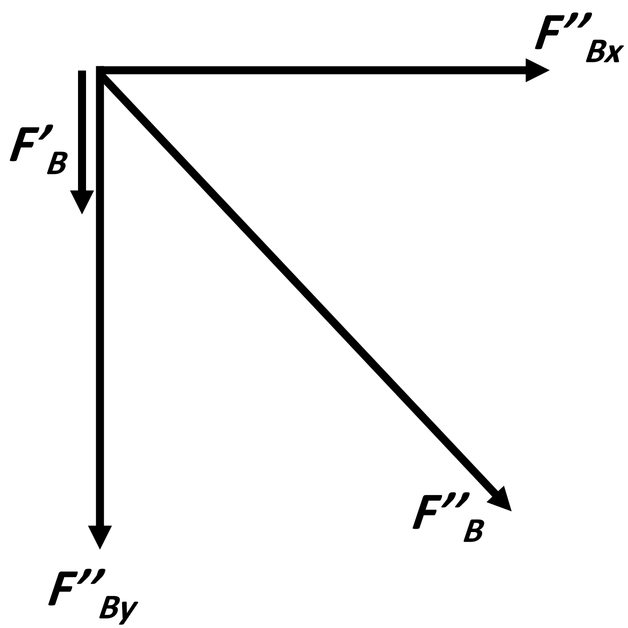


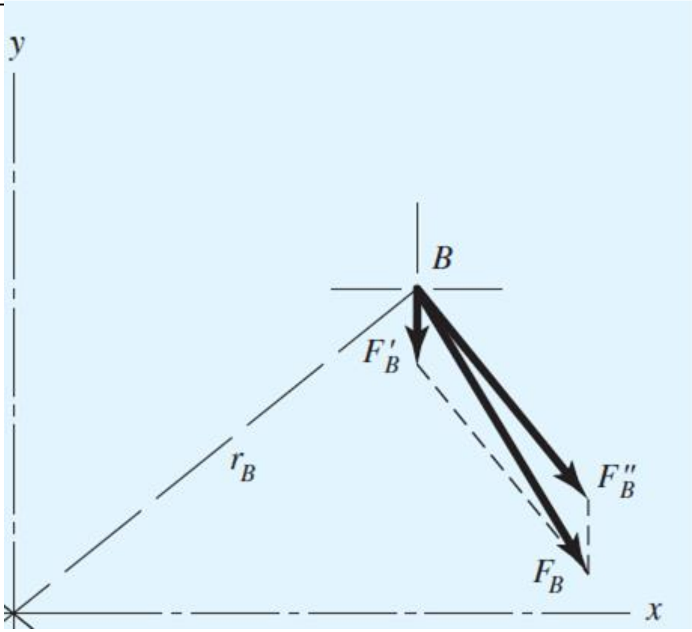
8.12 Bolted and Riveted Joints Loaded in Shear

Figure 8–29

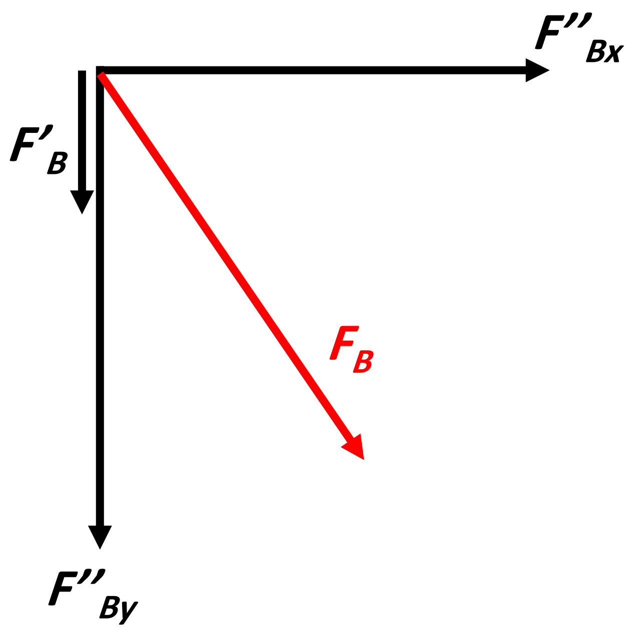


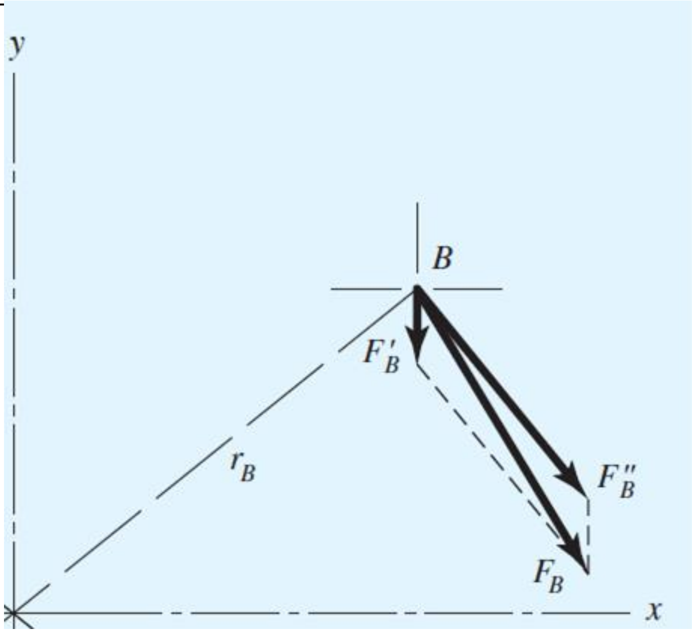
8.12 Bolted and Riveted Joints Loaded in Shear





8.12 Bolted and Riveted Joints Loaded in Shear





$$F_B = \sqrt{(F''_{By})^2 + (F''_{Bx} + F'_B)^2}$$

8.12 Bolted and Riveted Joints Loaded in Shear

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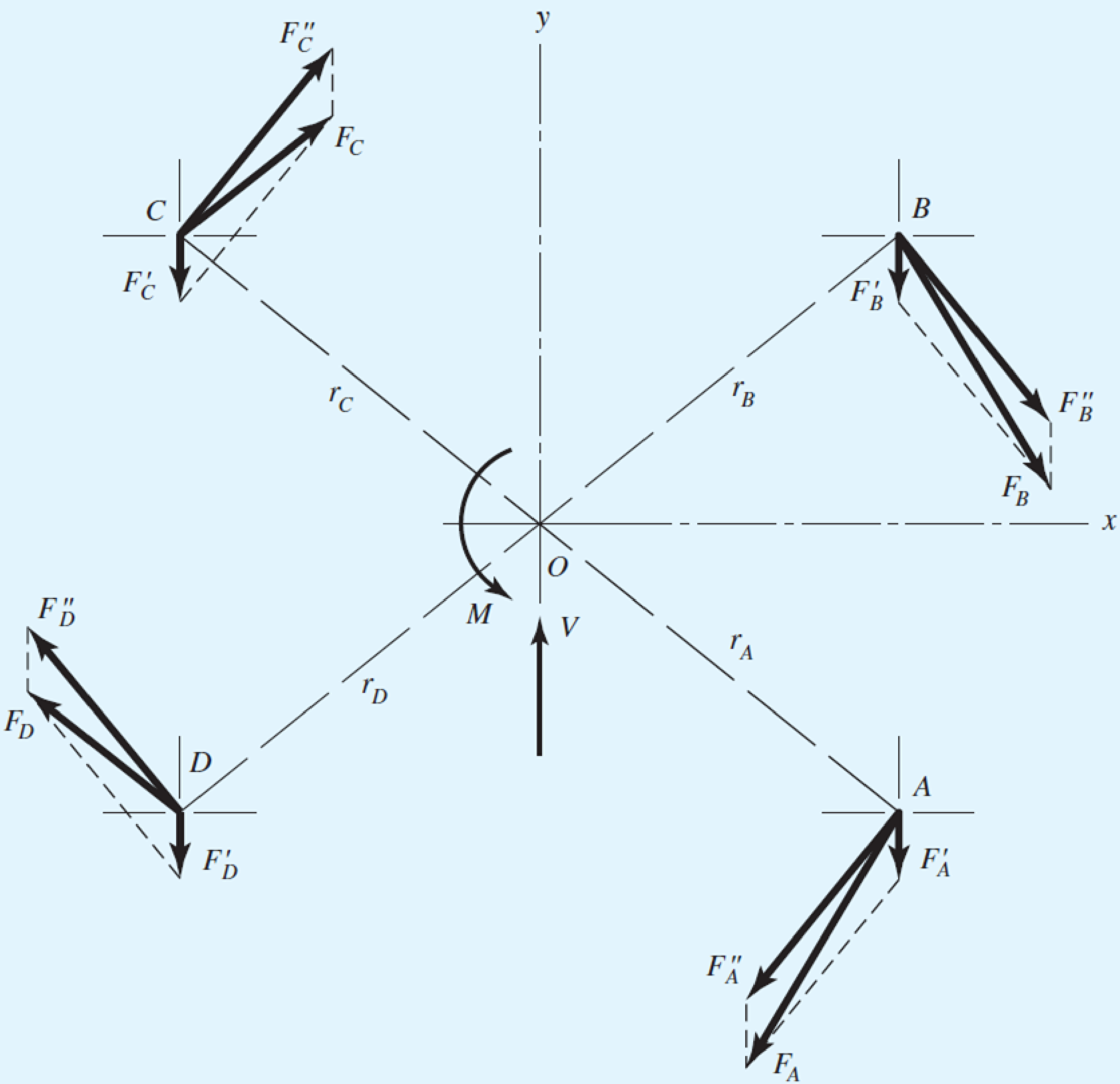
Figure 8–29



8.12 Bolted and Riveted Joints Loaded in Shear

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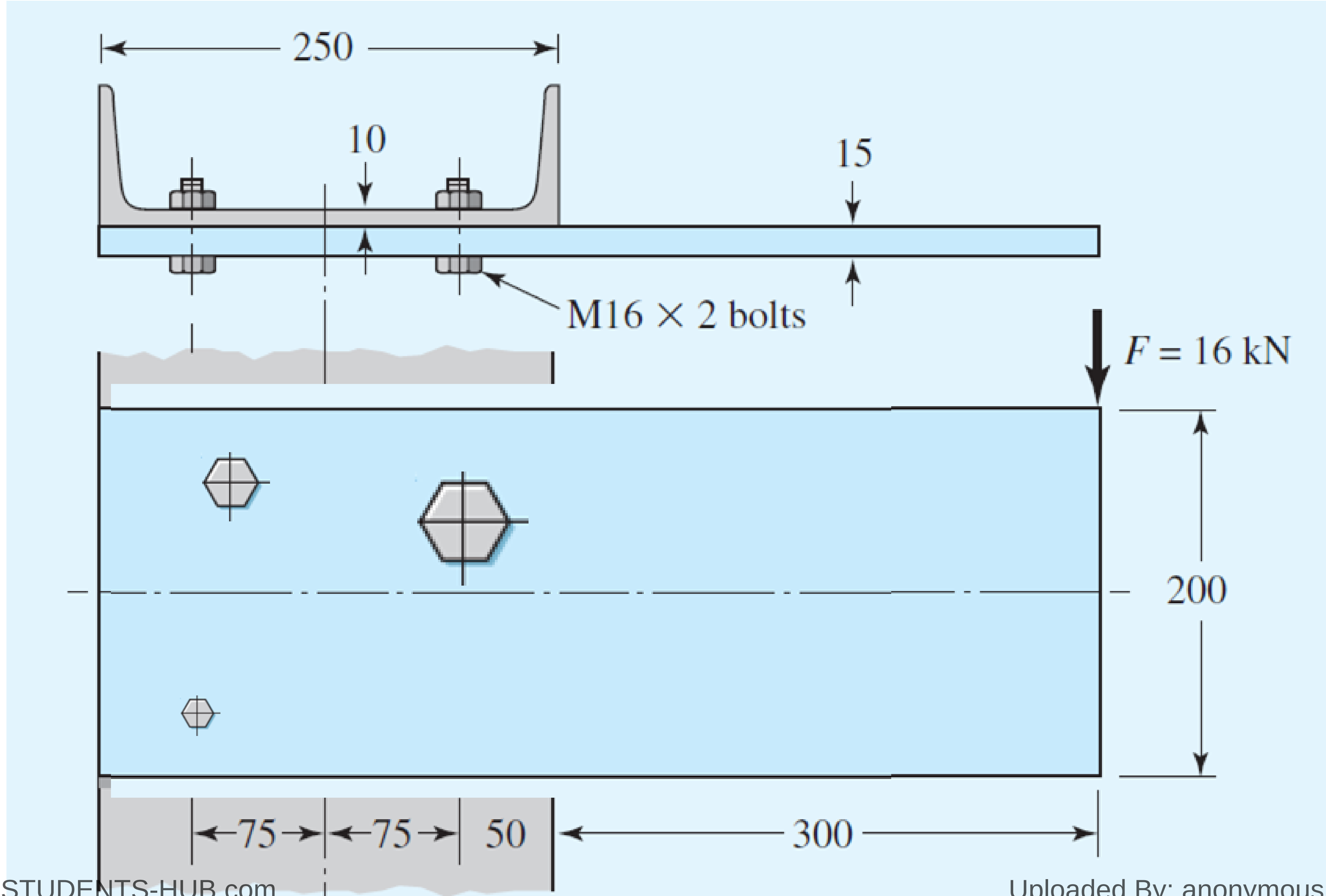
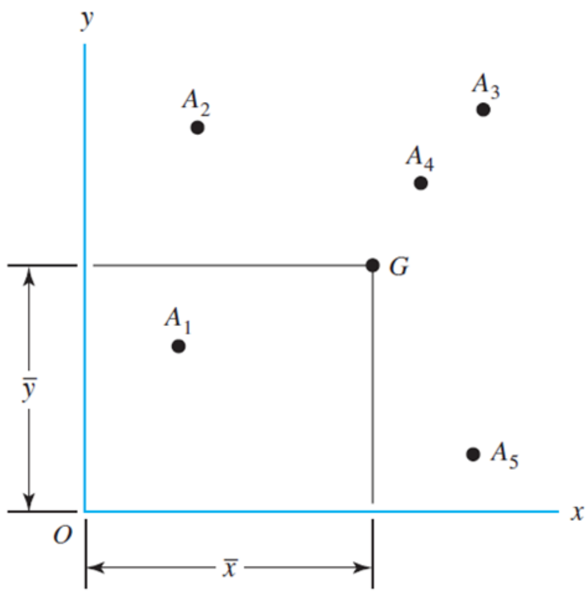


8.12 Bolted and Riveted Joints Loaded in Shear

$$\bar{x} = \frac{A_1x_1 + A_2x_2 + A_3x_3 + A_4x_4 + A_5x_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\sum_1^n A_i x_i}{\sum_1^n A_i}$$
$$\bar{y} = \frac{A_1y_1 + A_2y_2 + A_3y_3 + A_4y_4 + A_5y_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\sum_1^n A_i y_i}{\sum_1^n A_i} \tag{8–56}$$

Figure 8–26

Centroid of pins, rivets, or bolts.



8.12 Bolted and Riveted Joints Loaded in Shear