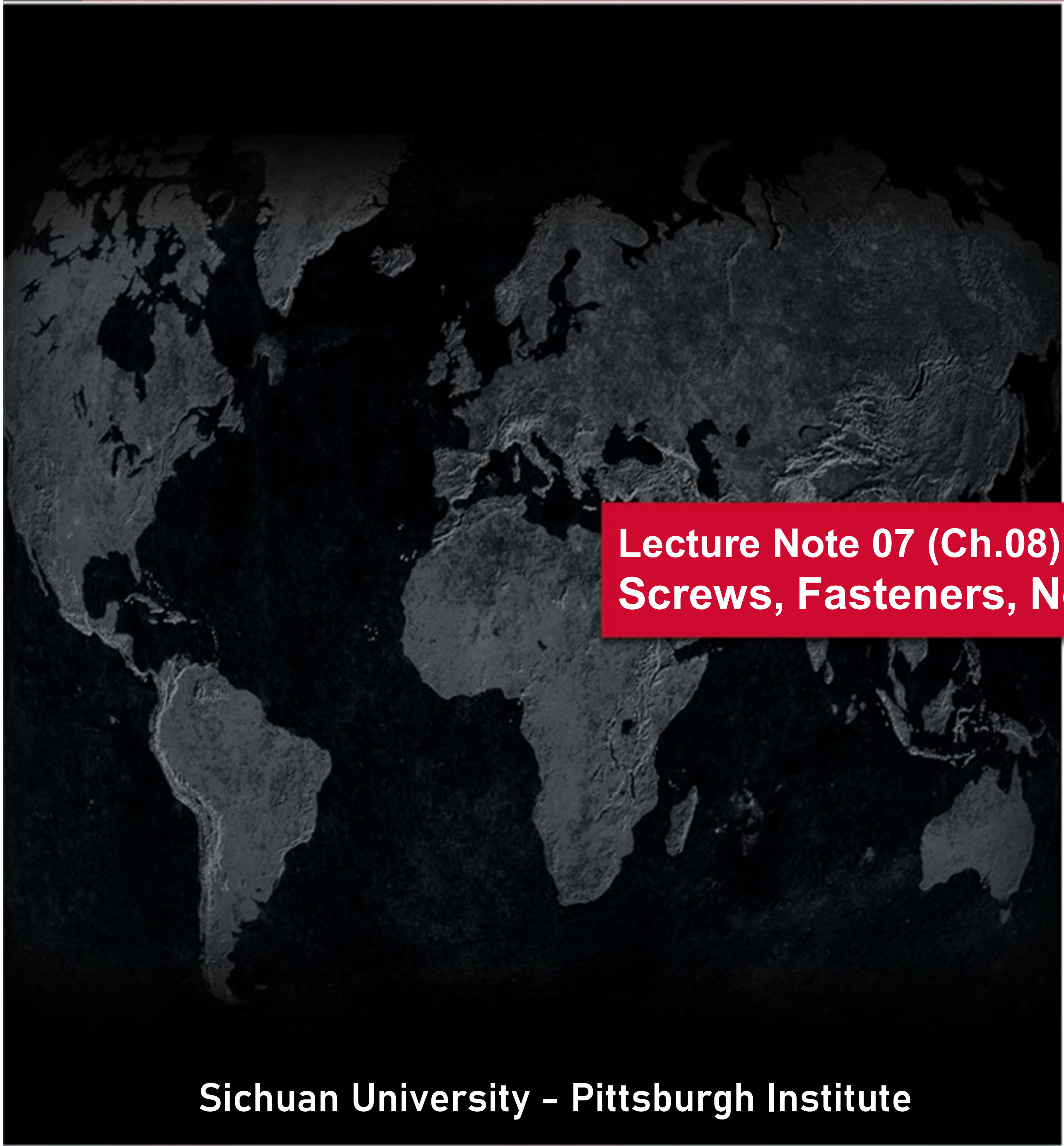


Lecture Note 07 (Ch.08)
Screws, Fasteners, Nonpermanent Joints

Instructor: Ping C. Sui, Ph.D.
ME 1029 Mechanical Design 2

Fall 2021

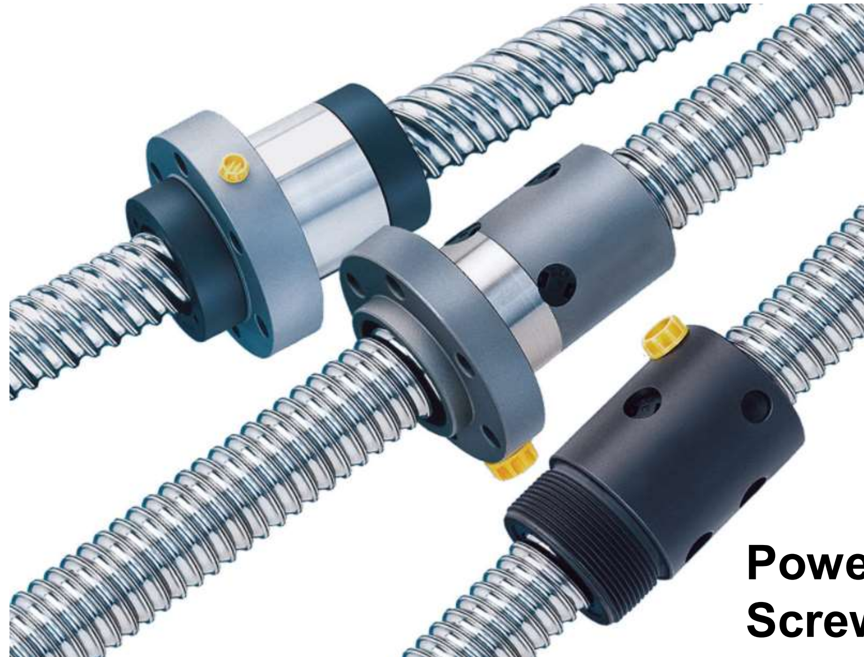
Topics Covered

- 8–1 Thread Standards and Definitions
- 8–2 The Mechanics of Power Screws
- 8–3 Threaded Fasteners
- 8–4 Joints-Fastener Stiffness
- 8–5 Joints Member Stiffness
- 8–6 Bolt Strength
- 8–7 Tension Joints-The External Load
- 8–8 Relating Bolt Torque to Bolt Tension
- 8–9 Statically Loaded Tension Joint with Preload
- 8–11 Fatigue Loading of Tension Joints
- 8–10 Gasketed Joints
- 8–12 Bolted and Riveted Joints Loaded in Shear

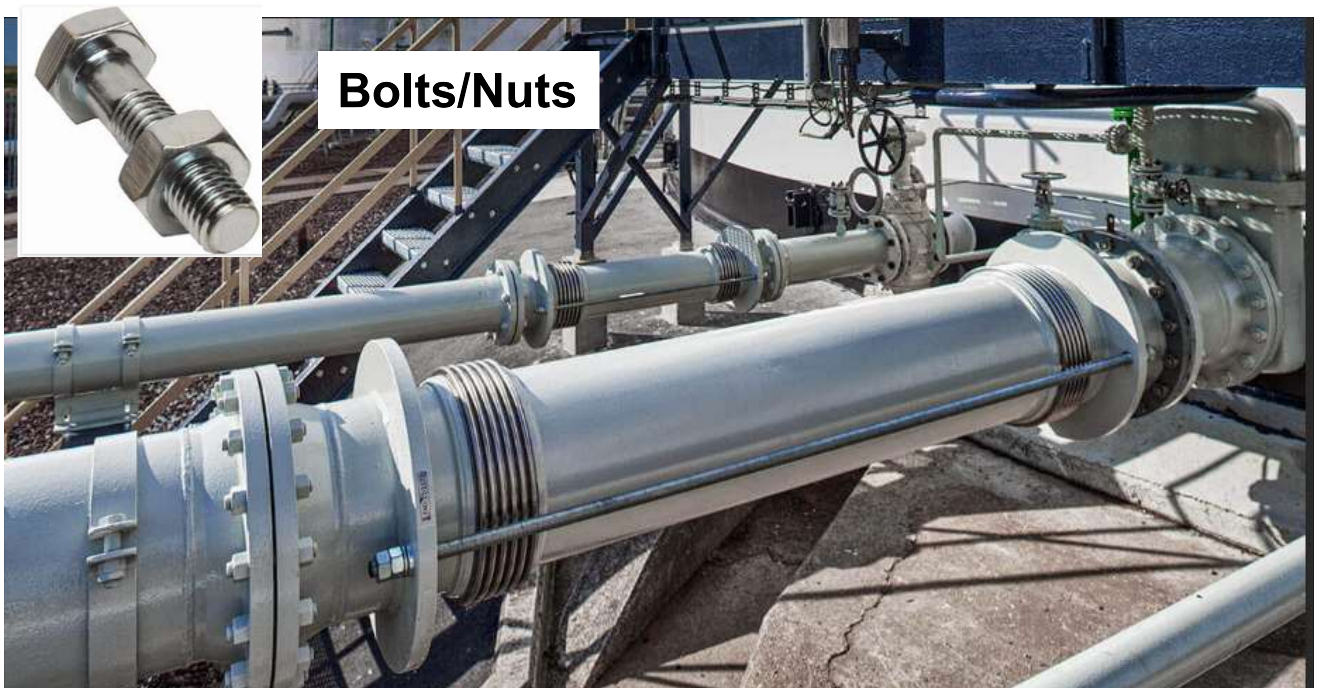
Introduction

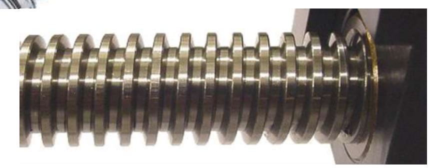
- **Nonpermanent** Joints/ Threaded Fasteners
  - Screws
  - Bolts/Nuts
  - Rivets

- Different names but similar working principle: change **angular** motion to **linear** motion
  - to transmit power, or
  - to develop large forces for position locking, liquid/gas sealing, etc.



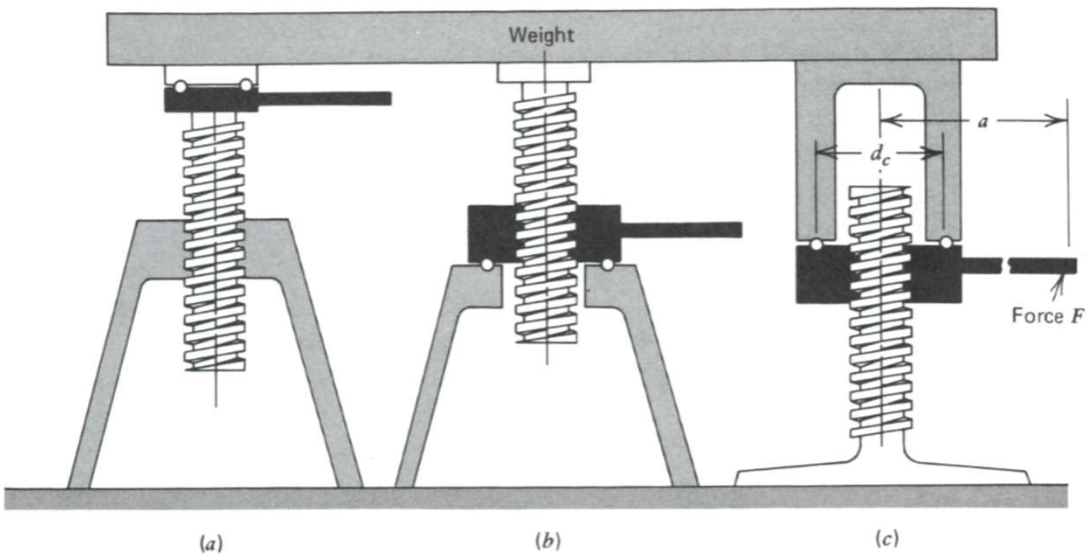
Power Screws





Power Screws

- A power screw is a device used in machinery to change angular motion into linear motion, and, usually, to <u>transmit power</u>.



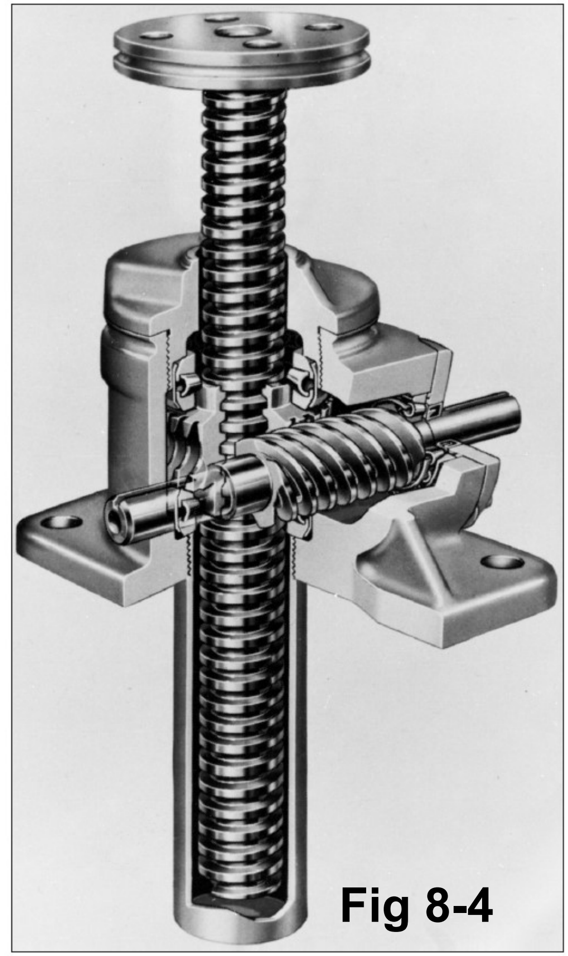
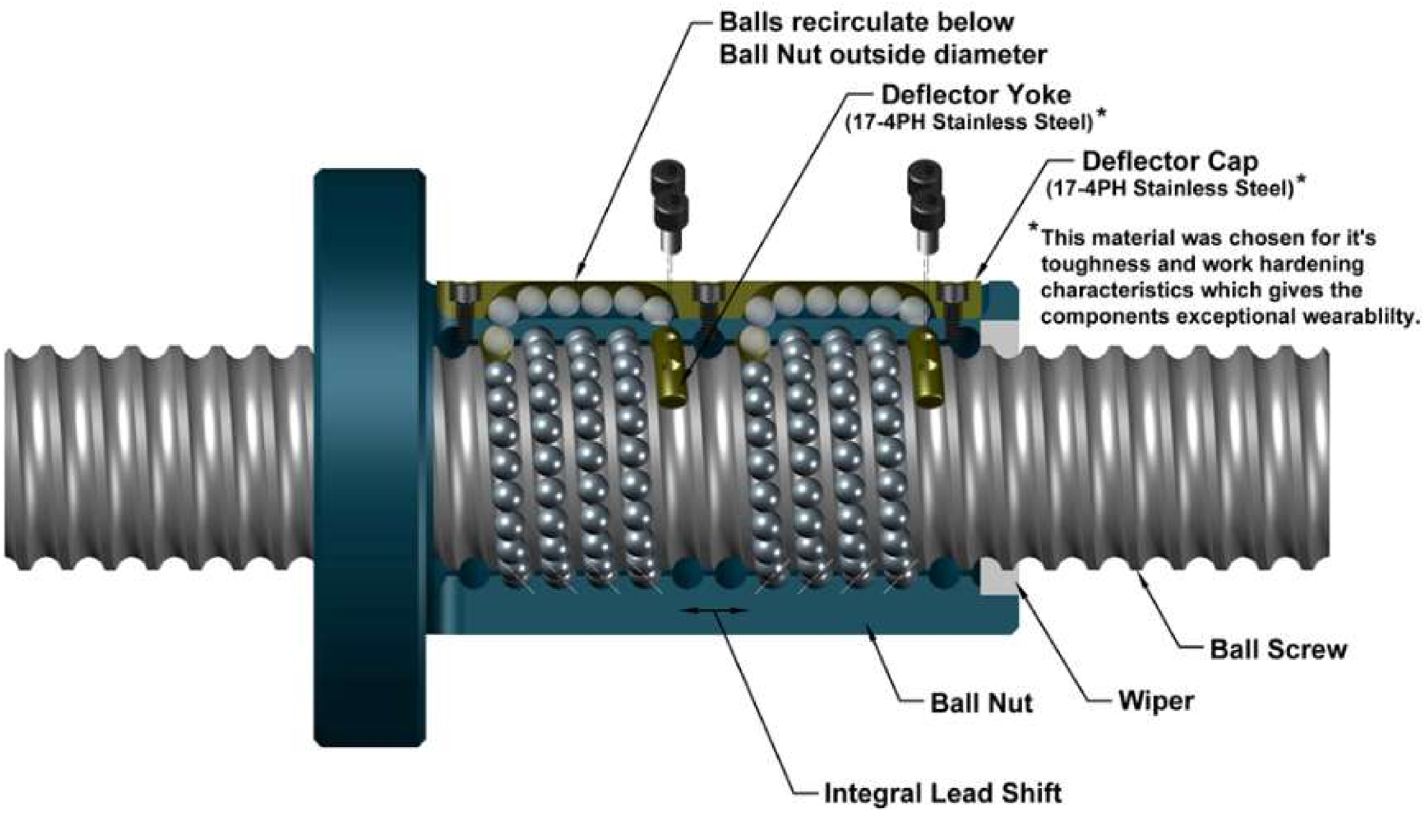
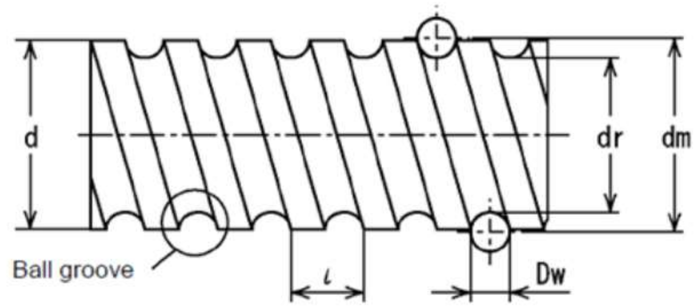


Fig 8-4

High Precision Application: Ball Screws



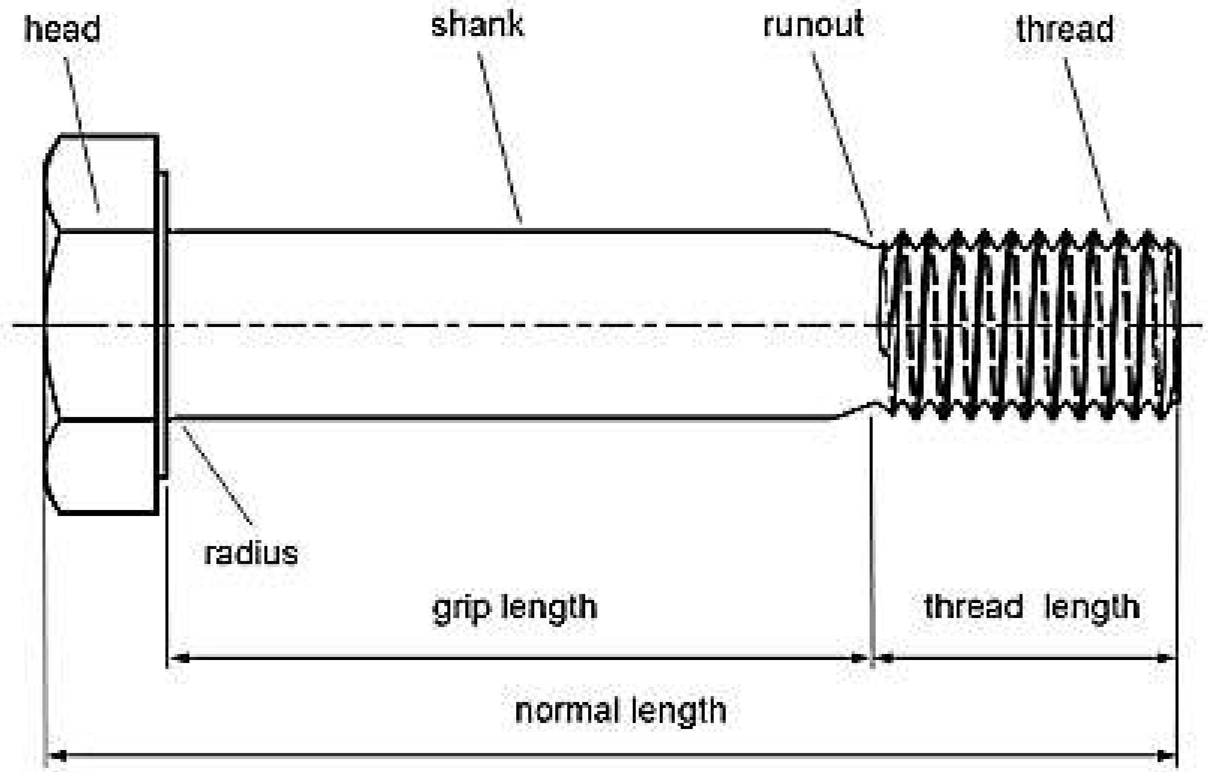


d：螺杆轴直径（简称直径）
dm：球的圆直径
dr：螺杆轴根部直径
l：主导
Dw：球直径

8-1 Thread Standards and Definitions:
Geometry and Terminology of External Screw Thread

Thread Details

- All threads are made according to the right-hand rule unless otherwise noted. That is, if the bolt is turned clockwise, the bolt advances toward the nut.



Terminology – Thread Profile

- **Pitch** (p): distance between adjacent thread forms measured parallel to the thread axis
- **Major Diameter** (d): the largest diameter of a screw thread
- **Minor (or root) diameter** (d_r): the smallest diameter of a screw thread
- **Pitch diameter** (d_p): A theoretical diameter between the major and minor diameters (the diameter where the width of the thread and groove are equal)

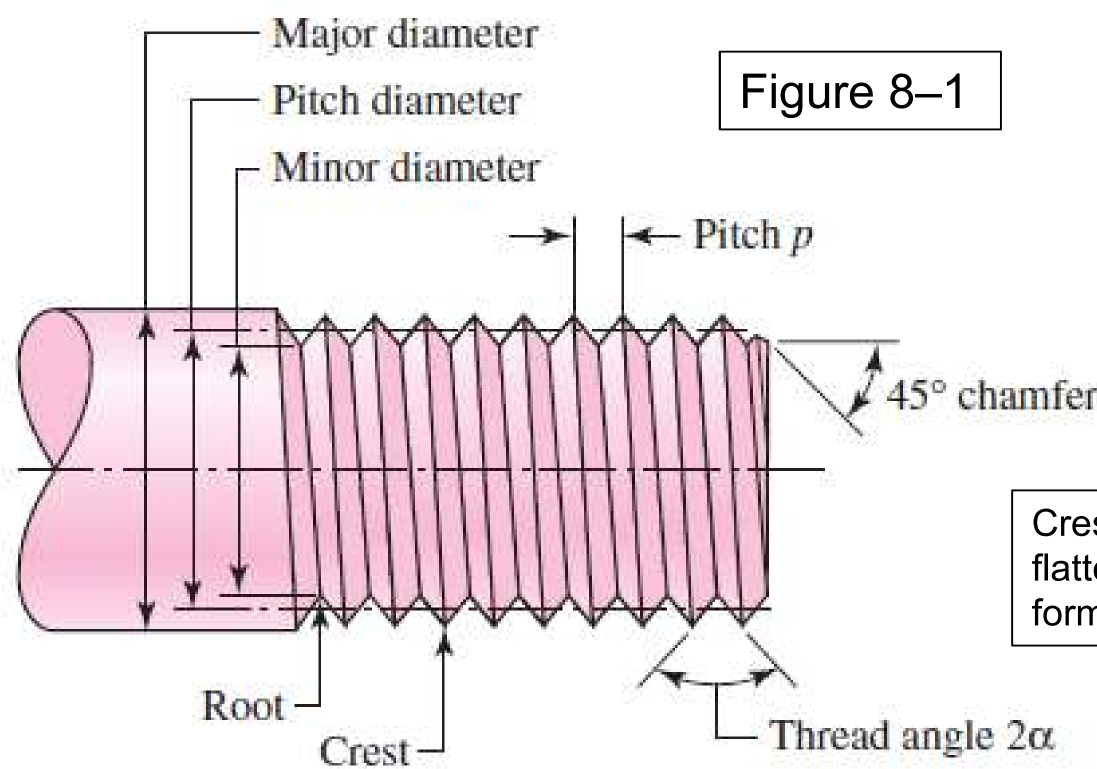


Figure 8–1

Crests and roots are actually flattened or rounded during the forming operation.

Terminology – Thread Profile

- **Lead** (l): axial distance the mating thread (or nut) advancing in one revolution.
- **Thread height** (h): radial distance between major diameter and minor diameter.
- **Thread angle** (α): angle between flanks of adjacent threads measured in an axial plane.
- **Lead angle** (λ): angle between perpendicular of the screw axis and rise of the thread.

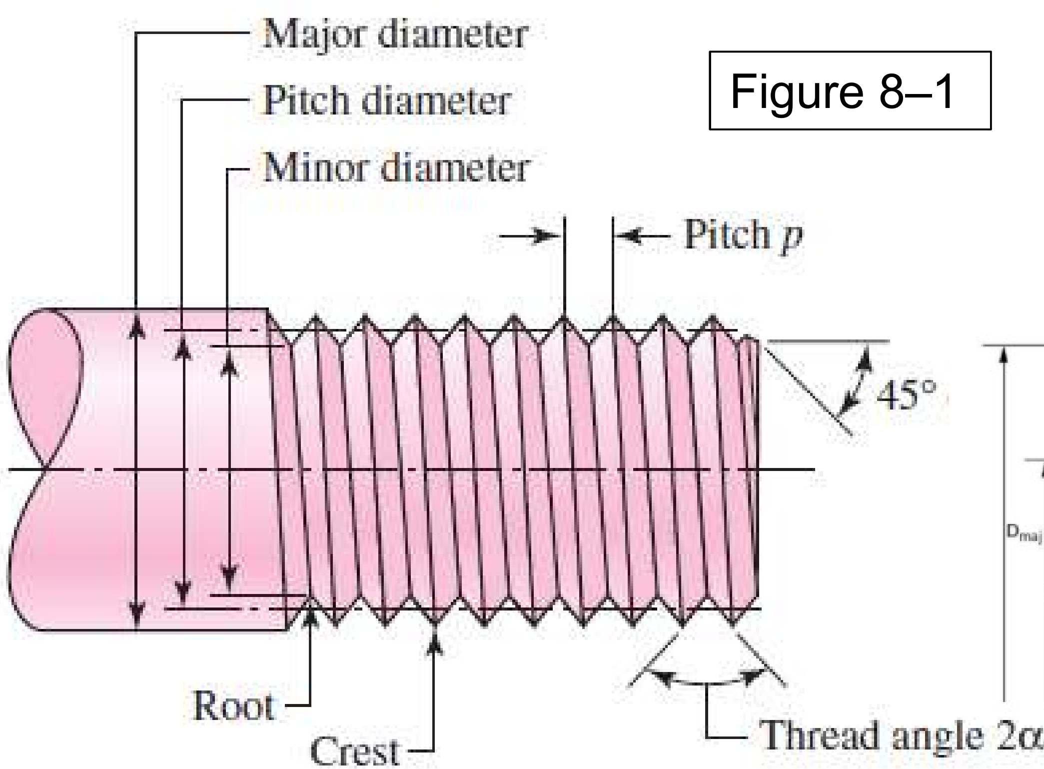
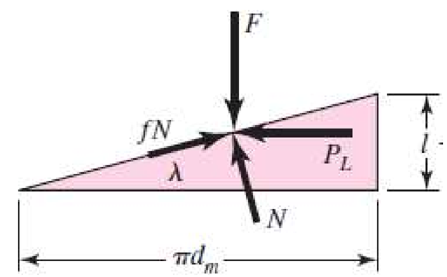
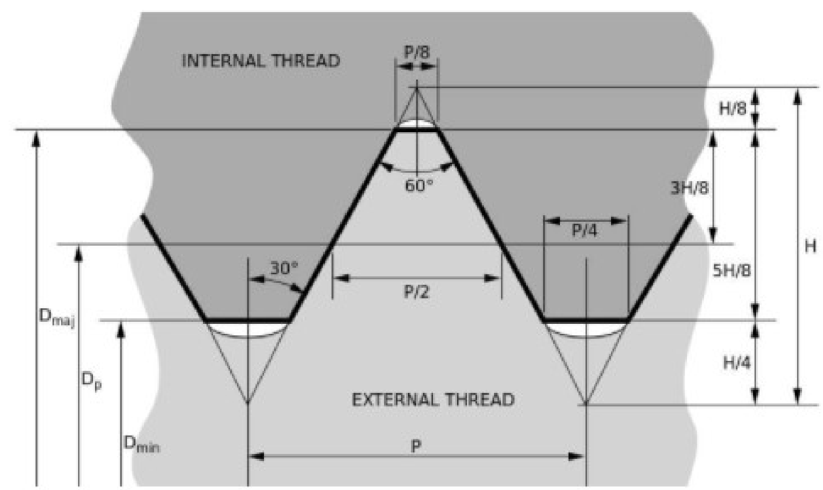


Figure 8–1

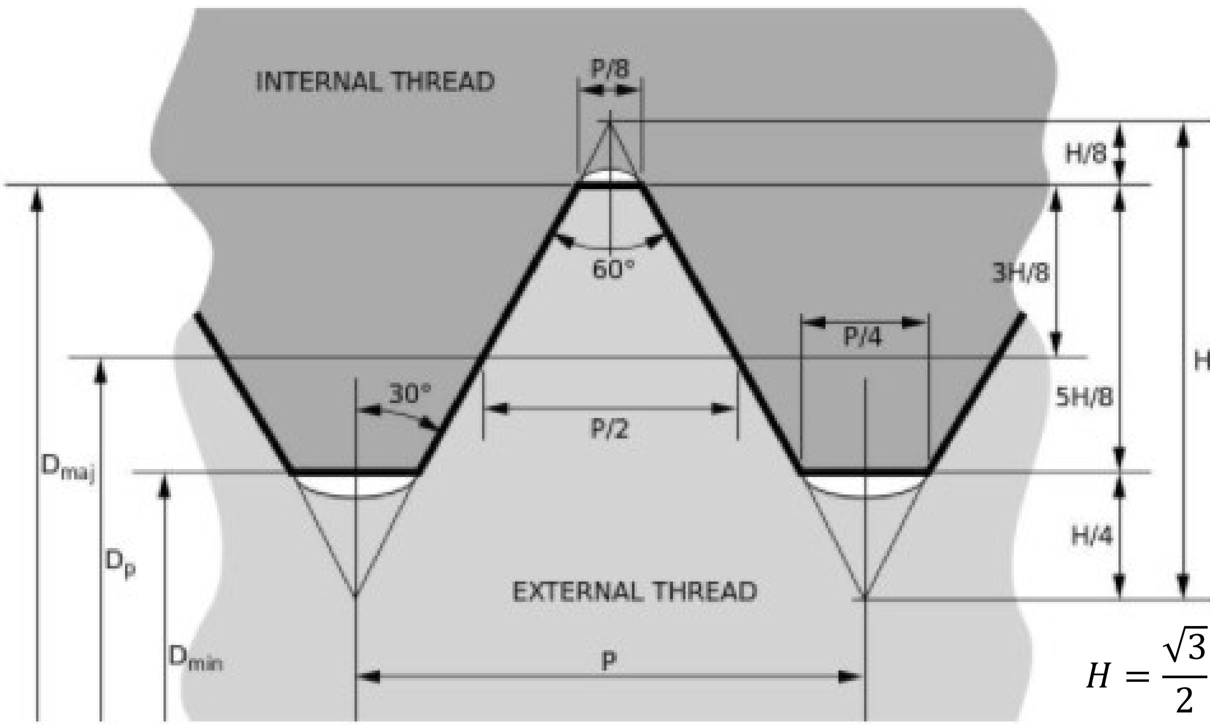




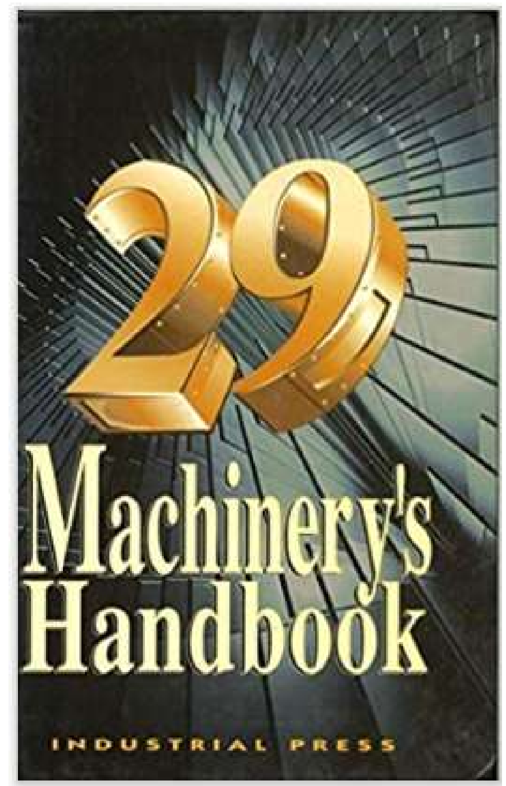
Unified Thread Profiles

American National (Unified) Thread Standard

- United States and Great Britain (Unit: inch)
- M and MJ profiles, etc. (Unit: mm)
- Thread angle is 60° and the crests of the thread may be either flat or rounded.

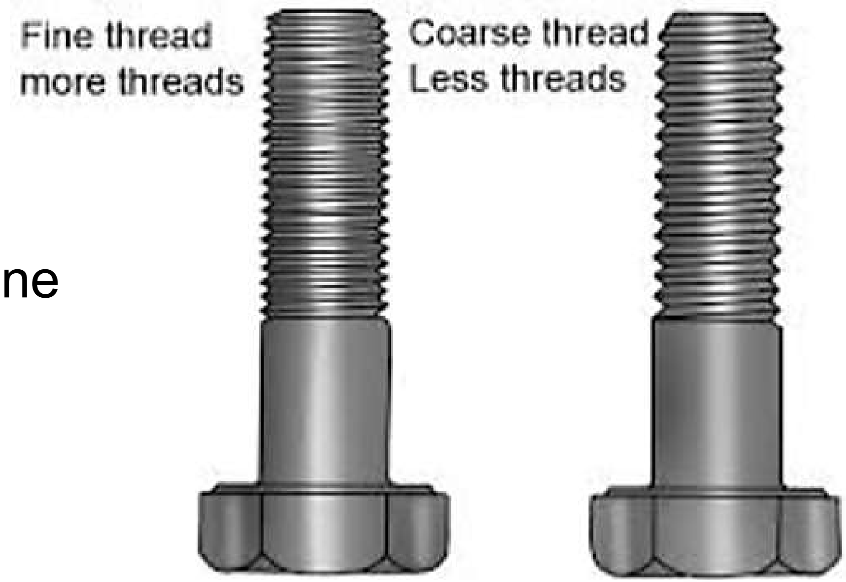


$$H = \frac{\sqrt{3}}{2}p$$



Unified Thread Series

- **UNC**: Coarse-Pitch Series
- **UNF**: Fine-Pitch Series
- **UNEF:** Extra-Fine-Pitch Series
- **UNRF**: Unified National Round Fine



- Unified threads are specified by stating the nominal major diameter, the number of threads per inch (TPI), and the thread series.
  – **5/8 in-18 UNRF** (0.625 in-18 UNRF): nominal major diameter of 0.625 in and 18 threads per inch
- Metric threads are specified by writing the nominal major diameter and pitch in millimeters, in that order.
  – **M12X1.75**: nominal major diameter of 12 mm and pitch of 1.75 mm.

Unified Screw Threads UNC and UNF (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

Table 8-1

ANSI B1.1-1974 and B18.3.1-1978

Minor Diameter: $d_r = d - 1.226869p$

Pitch Diameter: $d_p = d - 0.649519p$

$$Tensile\ Stress\ Area = \frac{\pi}{4}\left(\frac{d_r + d_p}{2}\right)^2$$

Unified Screw Threads UNC and UNF

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

ANSI B1.1-1974 and B18.3.1-1978

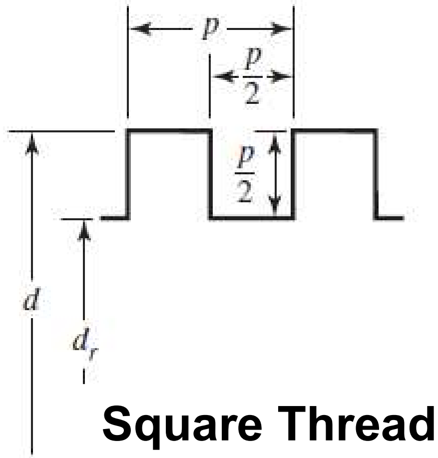
Minor Diameter: $d_r = d - 1.299038p$

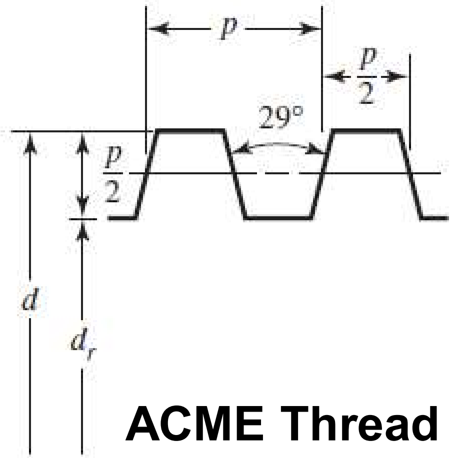
Pitch Diameter: $d_p = d - 0.649519p$

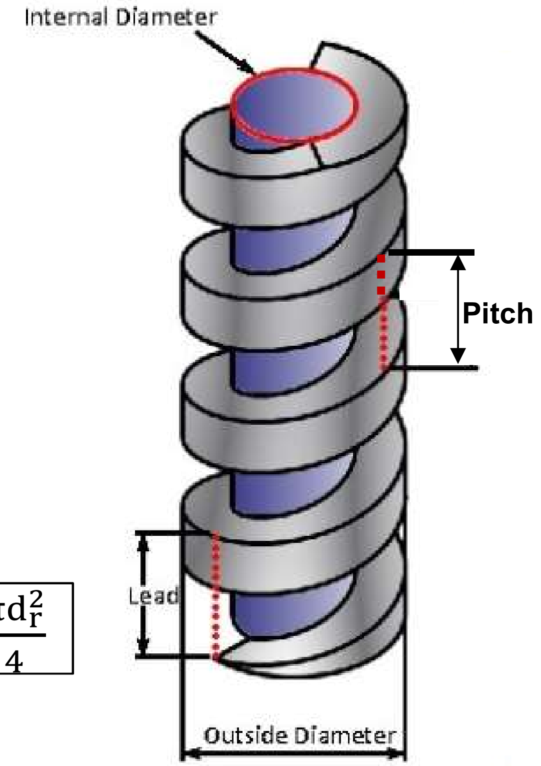
$$Tensile\ Stress\ Area = \frac{\pi}{4}\left(\frac{d_r + d_p}{2}\right)^2$$

Power Screws - Square Thread and ACME Thread

- Square and Acme threads are used on screws when power is to be transmitted.







Major Diameter $= d$

Minor Diameter $d_r = d - p$

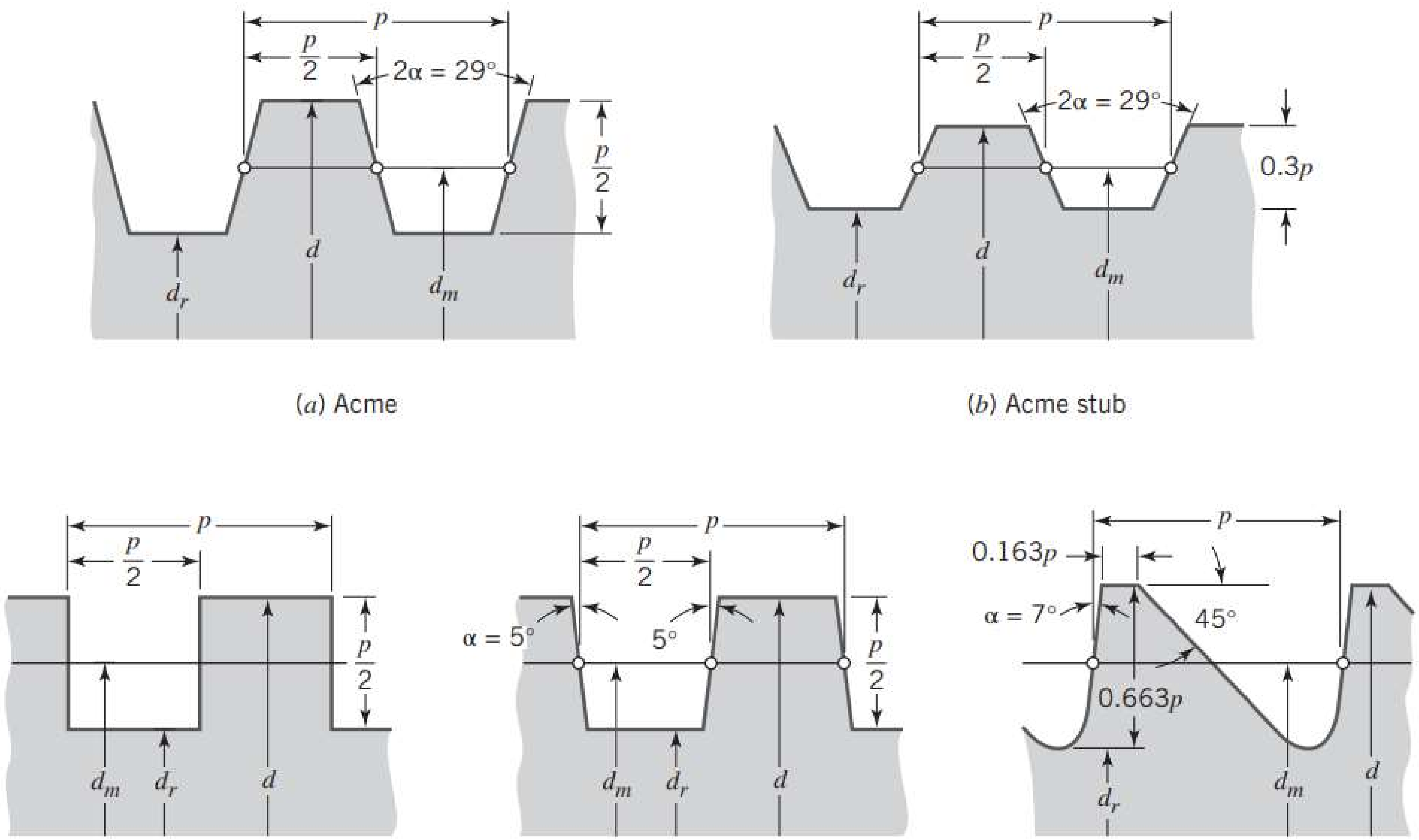
Mean Diameter $d_m = d - \frac{p}{2}$

$$\text{Tensile Stress Area} = \frac{\pi d_r^2}{4}$$

Table 8-3 Preferred Pitches for ACME Thread

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

Alternative Power Screw Profiles



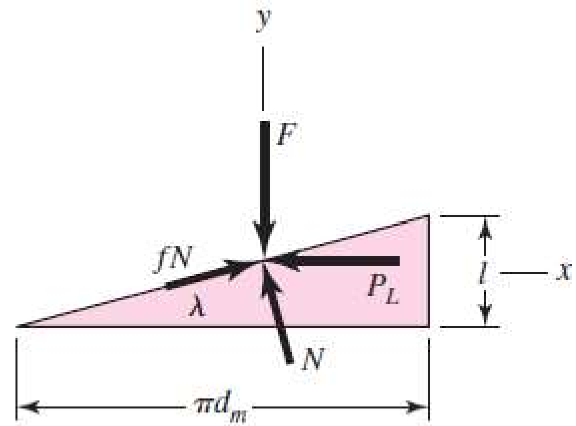
(a) Acme (b) Acme stub

(c) Square (d) Modified square (e) Buttress

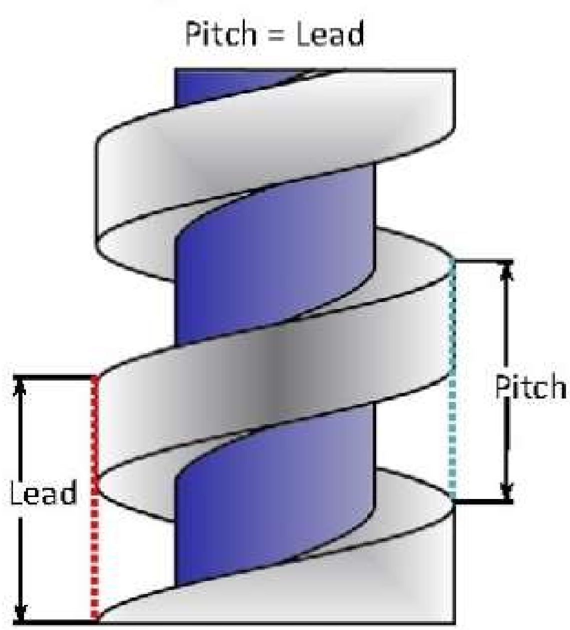
Relationship between Lead vs. Pitch

- For l as the lead, and λ as the lead angle, and p as the pitch

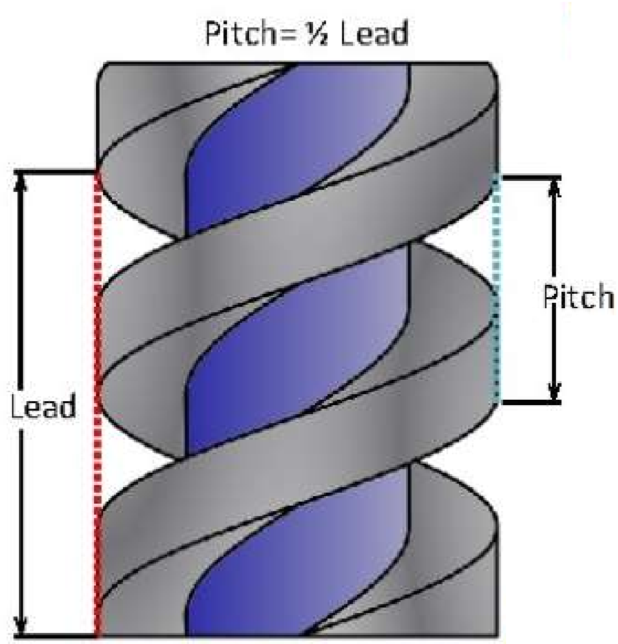
$$\tan \lambda = \frac{l}{\pi d_m}$$



- Single Thread: $l = p$
- Double Thread: $l = 2p$



Single Thread



Double Thread

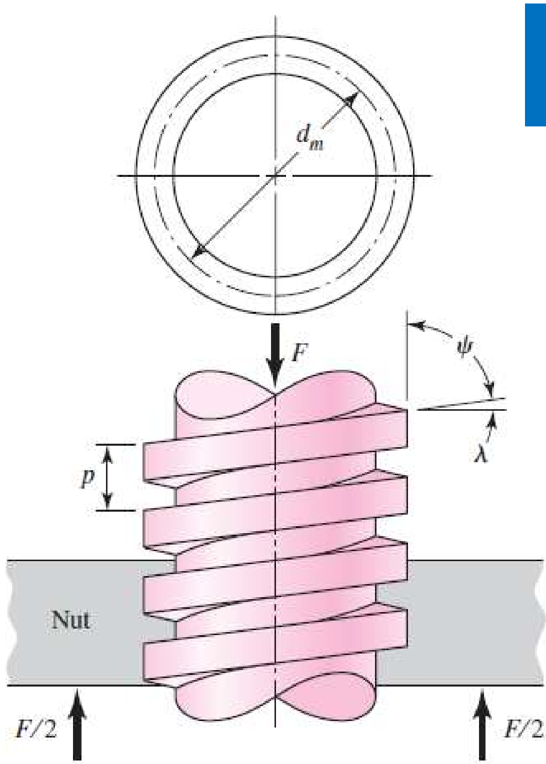
8-2 The Mechanics of Power Screws

Objectives:

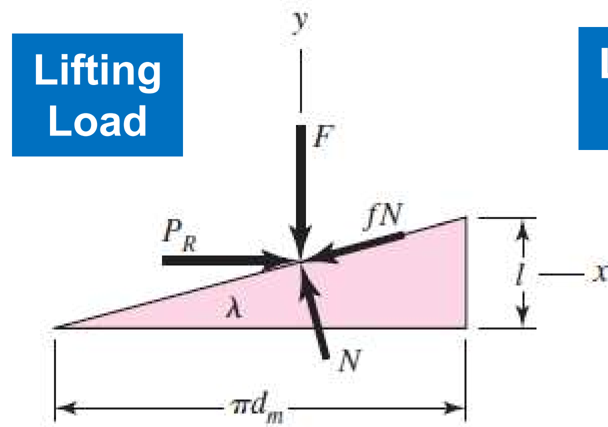
- Estimate needed torque to raise the load; and
- Evaluate structural integrity of the threads.

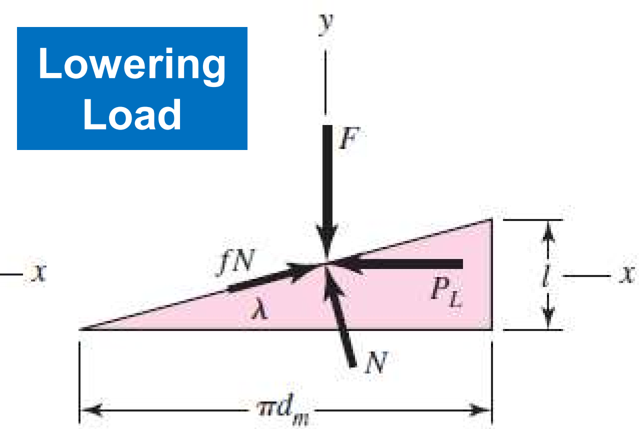
Mechanics of A Power Screw (Square Thread Form)

- Torque required to <u>raise</u> this load (tightening the bolt/screw), and
- Torque required to <u>lower</u> the load (loosening the bolt/screw)



ψ: *helix angle*
λ: *lead angle*



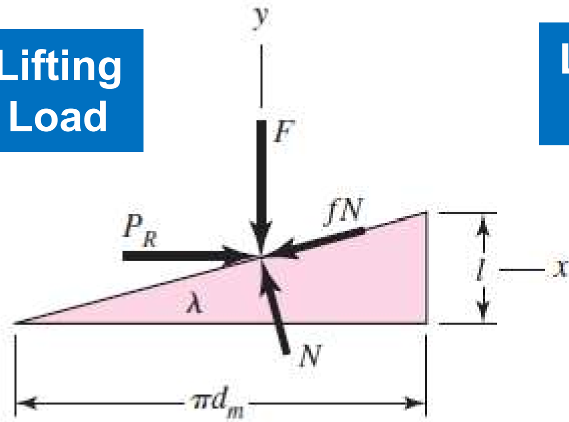


P_R: Force applied on thread to raise the load.
P_L: Force applied on thread to lower the load.

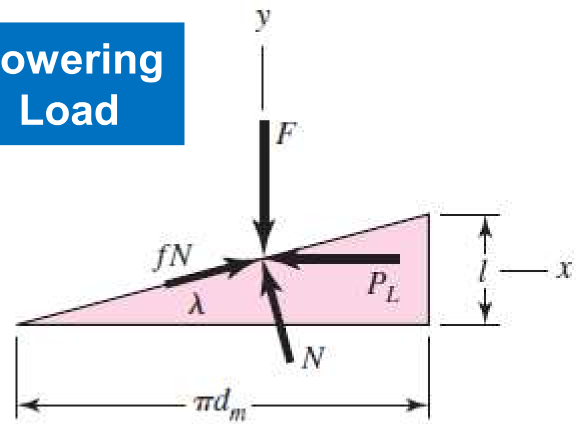
Table 8-4 Coefficient of Friction, f

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

Lifting Load



Lowering Load



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

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

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

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

By cancelling normal force N,

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

Divide both numerator and denominator by $\cos\lambda$ and use $\tan\lambda = \dfrac{l}{\pi d_m}$

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

Multiply by the mean radius $d_m/2$ to get torque

$$\boxed{T_R = \frac{F d_m}{2}\frac{\pi f d_m + l}{\pi d_m - fl}} \qquad \boxed{T_L = \frac{F d_m}{2}\frac{\pi f d_m - l}{\pi d_m + fl}}$$

Power Screw Torque Requirement

- Torque required to overcome thread friction and to raise the load

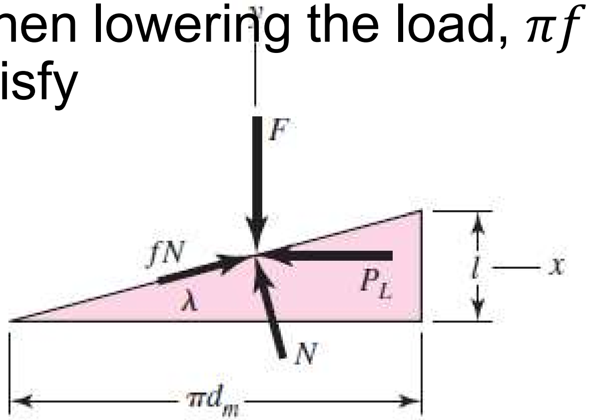
$$T_R = \frac{F d_m}{2} \frac{\pi f d_m + l}{\pi d_m - f l}$$

- Torque required to overcome thread friction and to lower the load

$$T_L = \frac{F d_m}{2} \frac{\pi f d_m - l}{\pi d_m + f l}$$

- Self-Locking: Prevent screw from slipping when lowering load
  - For self-locking condition when lowering the load, $\pi f d_m > l$, the following condition must satisfy

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



Power Screw Torque Efficiency

- Let COF $f = 0$, torque on frictionless surface:

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

  – since thread friction has been eliminated, this is the torque required only to raise the load.

- Define torque efficiency as

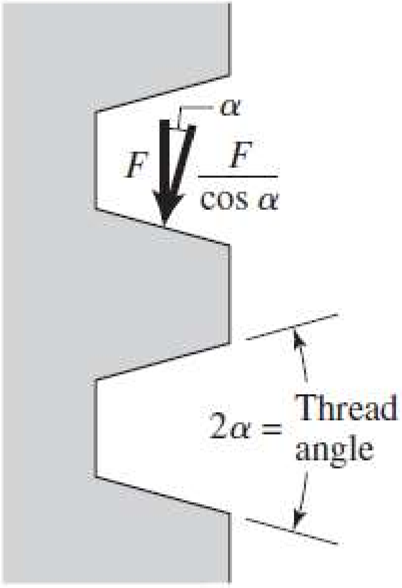
$$e = \frac{T_0}{T_R} = \frac{Fl}{2\pi T_R}$$

  Is T_0 smaller or greater than T_R?

2nd Torque Contribution Term (Wedge Friction)

- For non-square threads, like ACME, additional friction is introduced by the thread angle. This is known as the "Wedge" effect.
- Effect of thread angle is simplified by neglecting the compound effect of lead angle on thread angle. In this way, normal force component on thread should be represented by $\frac{N}{\cos\alpha}$

Fig 8-7a



- The frictional torque for raising the load is:

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

- Note that for ACME Thread: $2\alpha = 29°$

3rd Torque Contribution Term (Collar/Shoulder Friction)

- In case a collar or shoulder is used to support the axial load or exert preload on joints, this friction also needs to be overcome in order to tighten or loosen the joint.

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

- Henceforth, the total torque for tightening a joint is

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

Fig 8-7b

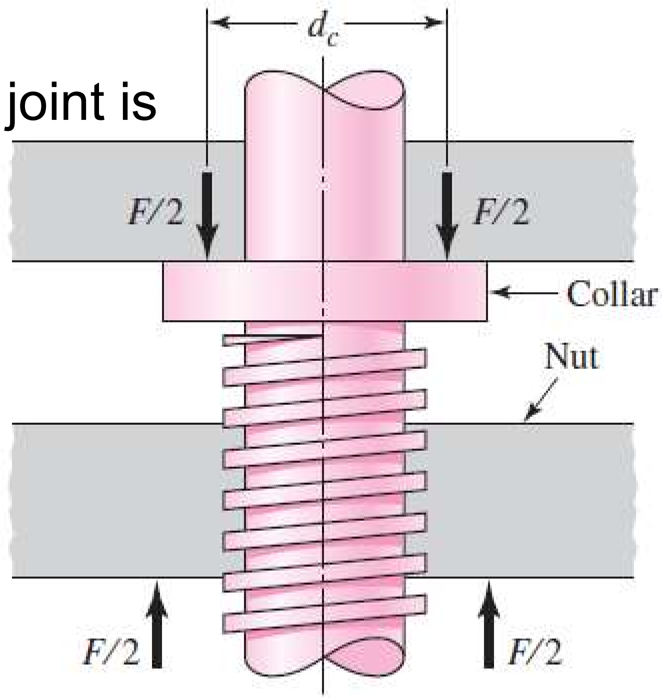
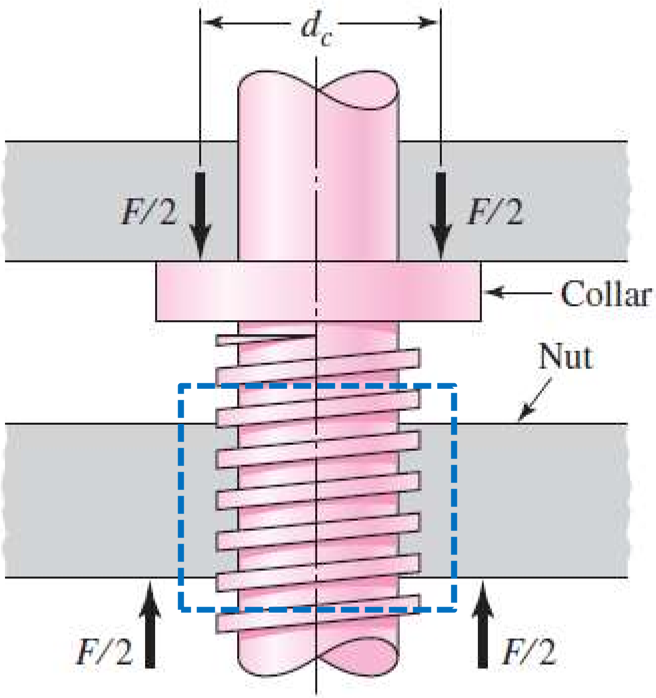


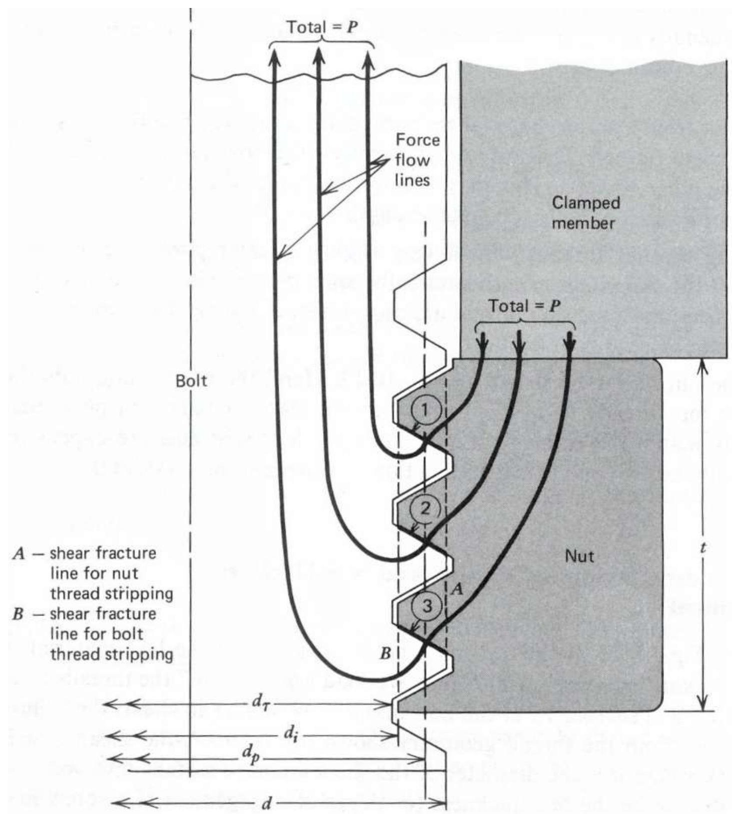
Table 8-6 Coefficient of Friction, fc

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

Load Sharing Among Power Screw Threads

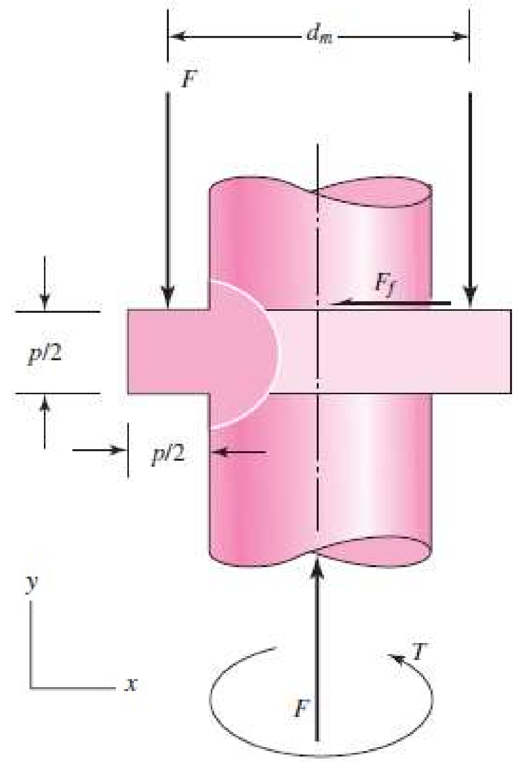
- The engaged threads don't share the load equally.
- Experiments showed that the first engaged thread (FET) carries 0.38 of the load, the second 0.25, the third 0.18, and the seventh is free of load.
- Use 0.38F and $n_t = 1$ for the maximum stress on power screw thread form





Static Thread Stresses

Fig 8-8



<u>On Thread Body</u>

- Torsional stress during tightening $\tau = \dfrac{16\,T}{\pi d_r^3}$
- Axial stress $\sigma = \dfrac{F}{A_t}$
  - Threaded Fastener: See Table 8-1, 8-2 for A_t
  - Power Screw: $A_t = \dfrac{\pi d_r^2}{4}$

Static Thread Stresses

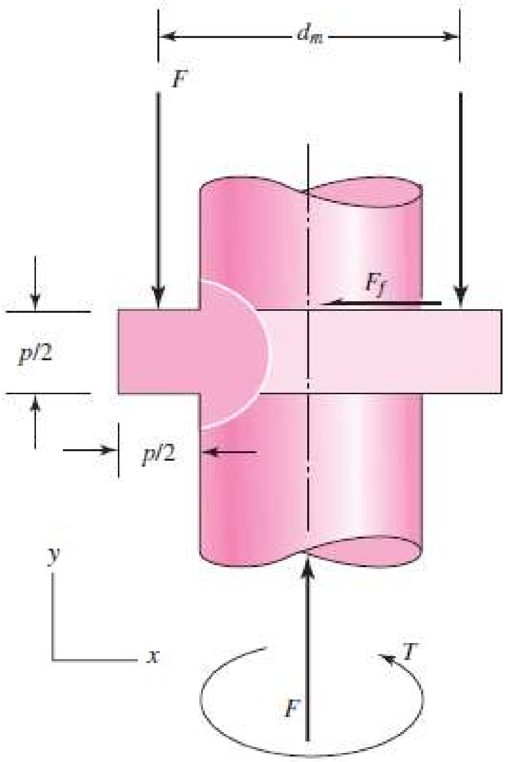
On Thread Flank Face

- Bearing stress: $\sigma_B = -\frac{F}{\pi d_m n_t (p/2)} = -\frac{2F}{\pi d_m n_t p}$
  - n_t: number of engaged threads

Table 8-4 Screw Bearing Pressure

Screw Material	Nut Material	Safe p_b, psi	Notes
Steel	Bronze	2500–3500	Low speed
Steel	Bronze	1600–2500	≤10 fpm
	Cast iron	1800–2500	≤8 fpm
Steel	Bronze	800–1400	20–40 fpm
	Cast iron	600–1000	20–40 fpm
Steel	Bronze	150–240	≥50 fpm

Fig 8-8



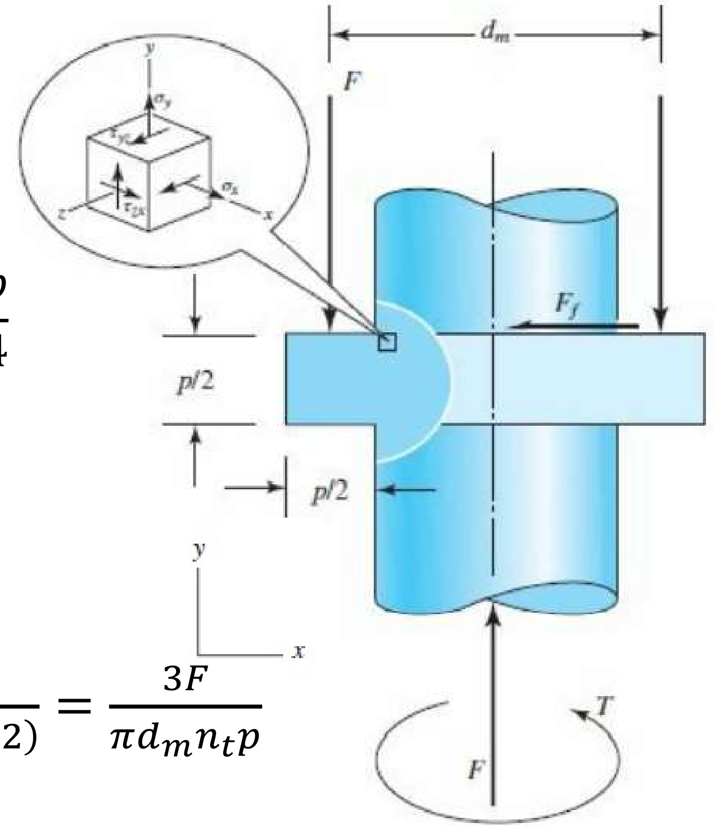
Static Thread Stresses

At Thread Root

- Bending Stress

$$I = \frac{\pi d_r n_t p^3}{96} \qquad M = \frac{Fp}{4} \qquad c = \frac{p/2}{2} = \frac{p}{4}$$

$$\sigma_b = \frac{Mc}{I} = \frac{6F}{\pi d_r n_t p}$$



- Transverse (or Punching) Shear
  - @Center of Root $\tau = \frac{3V}{2A} = \frac{3}{2}\frac{F}{\pi d_m n_t (p/2)} = \frac{3F}{\pi d_m n_t p}$

- Shear force acting at the root radius balances the torsion, T.

$$\tau = \frac{T}{\left(\pi d_r n_t \frac{p}{2}\right)\frac{d_r}{2}} = \frac{4T}{\pi d_r^2 n_t p}$$

Failure Assessment for Static Power Screw Threads

- Component Stresses at Thread Root:

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

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

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

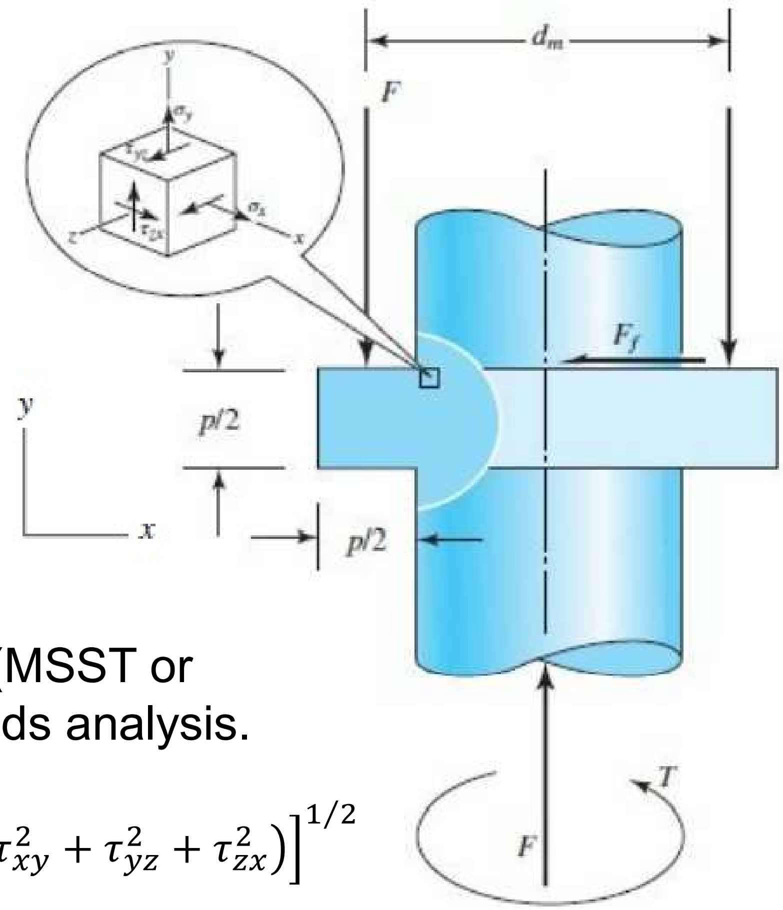


Fig 8-8

- Typical static failure assessment methods (MSST or DET) can be applied for power screw threads analysis.

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

Example 8-1

A square-thread power screw has a major diameter of 32 mm and a pitch of 4 mm with <u>double threads</u>, and it is to be used in an application similar to that in Fig. 8–4.

Given $f = f_c = 0.08$, $d_c = 40$ mm, and $F = 6.4$ kN per screw.

Find:

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

(b) torque required to raise and lower the load

(c) efficiency during lifting the load

(d) body stresses, torsional and compressive

(e) bearing stress

(f) thread bending stress at the root of the thread

(g) von Mises stress at the root of the thread

(h) maximum shear stress at the root of the thread.

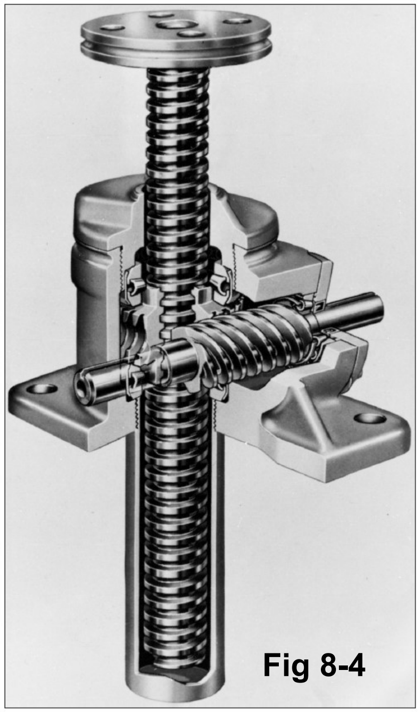


Fig 8-4

Example 8-1 (Cont'd)

Solution:

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

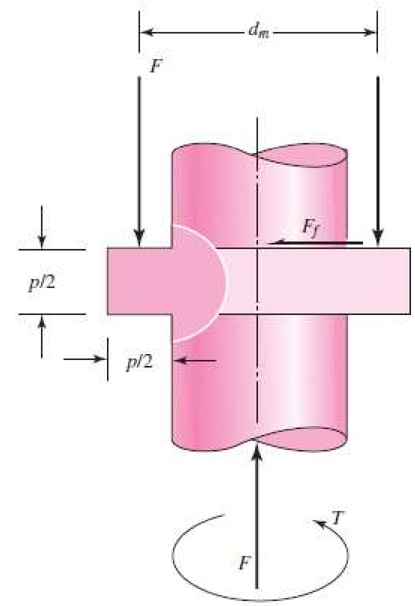
Major diameter d=32mm; Pitch p=4mm;

Minor diameter d_r=d-p=32-4=28mm

Mean diameter d_m=d-p/2=32-4/2=30mm

Lead l=np=2(4)=8mm

Thread width=$\frac{p}{2}=2mm$; Thread depth=$\frac{p}{2}=2mm$



(b) Find the torque required to raise and lower the load

Torque required to raise the load

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

$$=\frac{6400\cdot 30}{2}\left(\frac{8+\pi\cdot 0.08\cdot 30}{\pi 30-0.08\cdot 8}\right)+\frac{6400\cdot 0.08\cdot 40}{2}=26180\ N\cdot mm=26.18\ N\cdot m$$

Torque required to lower the load

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

$$=\frac{6400\cdot 30}{2}\left(\frac{\pi\cdot 0.08\cdot 30-8}{\pi 30+0.08\cdot 8}\right)+\frac{6400\cdot 0.08\cdot 40}{2}=9770\ N\cdot mm=9.77\ N\cdot m$$

Example 8-1 (Cont'd)

Solution:

(c) Find the efficiency during lifting the load

Overall efficiency $e = \frac{Fl}{2\pi T_R} = \frac{6400 \cdot 8}{2\pi \cdot 26180} = 0.31$

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

Thread body shear stress $\tau = \frac{16\,T_R}{\pi d_r^3} = \frac{16 \cdot 26180}{\pi \cdot 28^3} = 6.1\ MPa$

Power Screw $A_t = \frac{\pi d_r^2}{4} = \frac{\pi 2\ \ ^2}{4} = 616\ mm^2$

Axial nominal stress $\sigma = -\frac{6400}{616} = -10.39\ MPa$

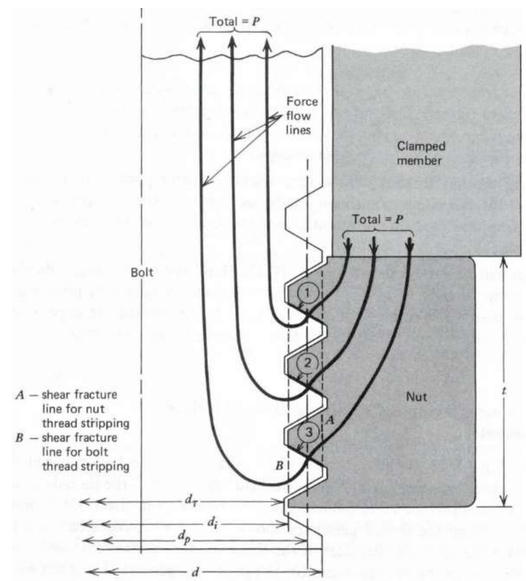
(e) Find the bearing stress

Assume one thread (n_t=1) carrying 0.38F

$$\sigma_B = -\frac{F}{\pi d_m n_t (p/2)} = \frac{0.38 \cdot 6400}{\pi \cdot 30 \cdot 1 \cdot 2} = -12.9\ MPa$$

(f) Find the thread bending stress at the root of the thread.

$$\sigma_b = \frac{Mc}{I} = \frac{6F}{\pi d_r n_t p} = \frac{6 \cdot 0.38 \cdot 6400}{\pi \cdot 28 \cdot 1 \cdot 4} = 41.5\ MPa$$



Example 8-1 (Cont'd)

Solution:

Tangential shear stress:

$$\tau_{zx} = -\frac{4T}{\pi d_r^2 n_t p} = -4.04\ MPa$$

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

σ_x = 41.5 MPa $\quad$ τ_{xy} = 0

σ_y = −10.39 MPa $\quad$ τ_{yz} = 6.07 MPa

σ_z = 0 $\quad$ τ_{zx} = -4.04 MPa

Von Mises Stress is:

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

$$\sigma_e = \frac{1}{\sqrt{2}}\left[\left(41.5 - (-10.39)\right)^2 + (-10.39 - 0)^2 + (0 - 41.5)^2 + 6(0 + 6.07^2 + 4.04^2)\right]^{1/2}$$

$$= 49.2\ MPa$$

8-3 Threaded Fasteners

Examples of Threaded Fasteners

- Hexagon-head bolt and nuts

Figure 8–9

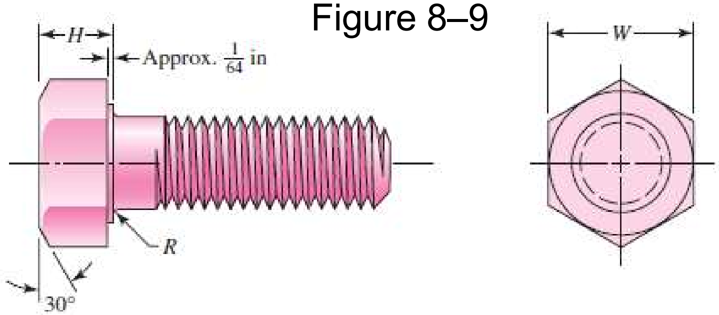
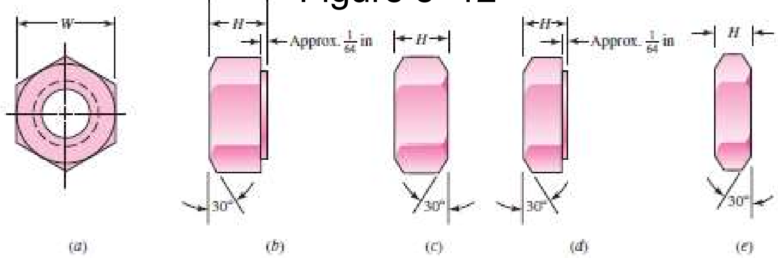


Figure 8–12



- Cap-screws
  - fillister head;
  - flat head;
  - hexagonal socket head.

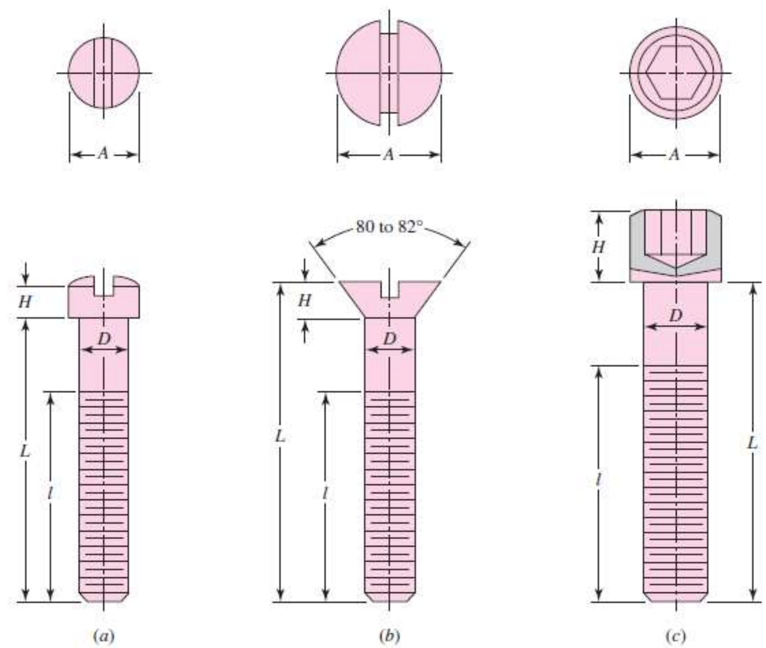


Figure 8–10

Unified Screw Threads UNC and UNF (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

Table 8-1

ANSI B1.1-1974 and B18.3.1-1978

Minor Diameter: $d_r = d - 1.226869p$

Pitch Diameter: $d_p = d - 0.649519p$

$$Tensile\ Stress\ Area = \frac{\pi}{4}\left(\frac{d_r + d_p}{2}\right)^2$$

Unified Screw Threads UNC and UNF

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

ANSI B1.1-1974 and B18.3.1-1978

Minor Diameter: $d_r = d - 1.299038p$

Pitch Diameter: $d_p = d - 0.649519p$

$$Tensile\ Stress\ Area = \frac{\pi}{4}\left(\frac{d_r + d_p}{2}\right)^2$$

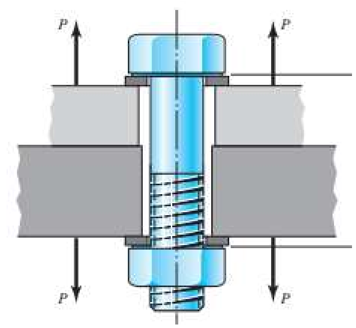
Bolt/Screw Geometric and Structural Information

Metric	USCS	Provided Information
Table 8-1	Table 8-2	Diameters, Areas, TPI of Coarse-Pitch and Fine- Pitch Threads
	Table 8–9	SAE Specifications for Steel Bolts (Min Proof/Tensile/Yield Strength)
	Table 8-10	ASTM Specifications for Steel Bolts (Min Proof/Tensile/Yield Strength)
Table 8-11		Steel Bolts, Screws, and Studs (Min Proof/Tensile/Yield Strength)
Table A–29	Table A–29	Bolt head dimensions of Square and Hexagonal Bolts
Table A–30	Table A–30	Screw head dimensions of Hexagonal Cap Screws and Heavy Hexagonal Screws
Table A–31	Table A–31	Dimensions of Hexagonal Nuts
Table A–33	Table A–32	Dimensions of Plain Washers
Table 8–4	Table 8–4	Screw Bearing Pressure P_b
Table 8–5	Table 8–5	Coefficients of Friction f for Threaded Pairs
Table 8–6	Table 8–6	Thrust-Collar Friction Coefficients
Table 8–7	Table 8–7	Procedure for Finding Fastener Stiffness
Table 8–8	Table 8–8	Simplified Member Stiffness Calculation: Parameters of Various Member Materials
	Table 8-12	Load Sharing based on Bolt and Member Stiffnesses.
Table 8-15	Table 8-15	Torque Factors K for Use with Eq 8-27
Table 8-16	Table 8-16	Fatigue Stress-Concentration Factors K_f for Threaded Elements
Table 8-17	Table 8-17 (P.445)	Fully Corrected Endurance Strengths for Bolts and Screws with Rolled Threads

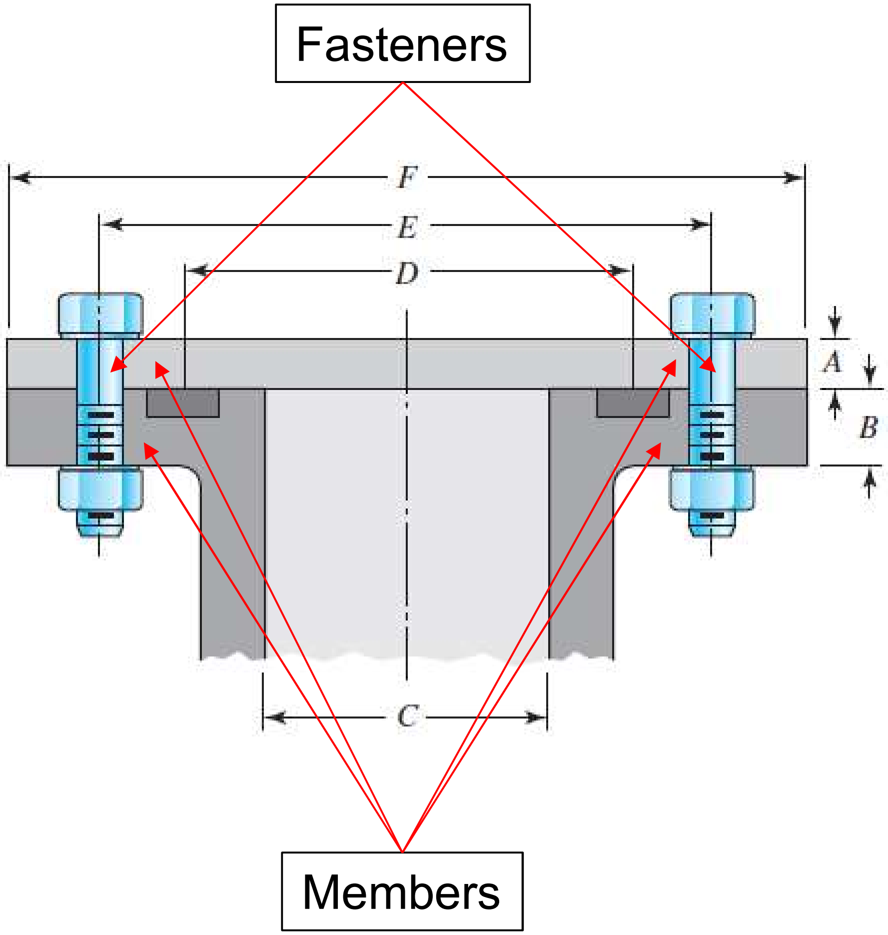
Structural Risks of Nonpermanent Joints

- In most cases the threat is from overproof loading of fasteners.
- The threat from fatigue is low and deterministic method is adequate.
- Washers must always be used under the bolt head
- During tightening, the first thread of the nut tends to take the entire load; but yielding occurs, which eventually further distributed the loading over about three nut threads. For this reason, <u>nuts should never be re-used</u>.



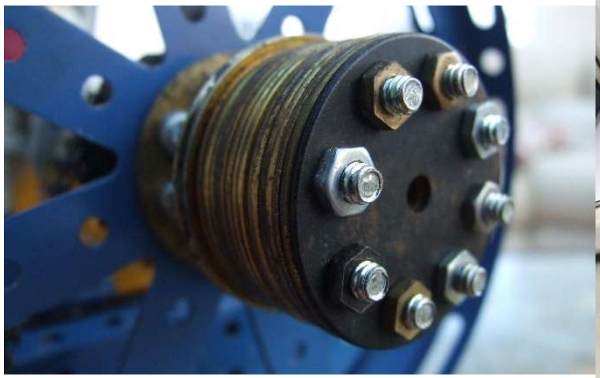



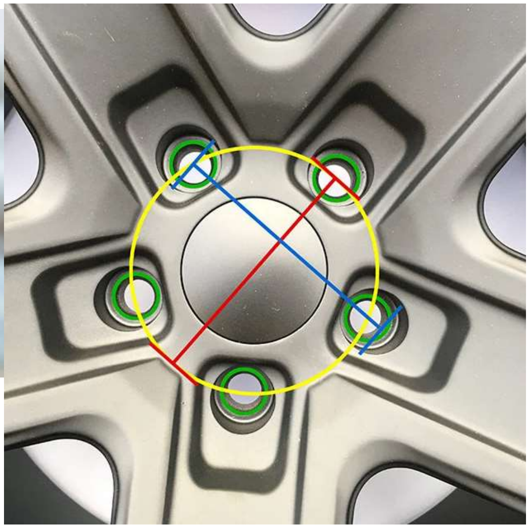
8-4 Joints: Fastener Stiffness
8-5 Joints: Member Stiffness
Final joint stiffness is the combination of fastener and member.

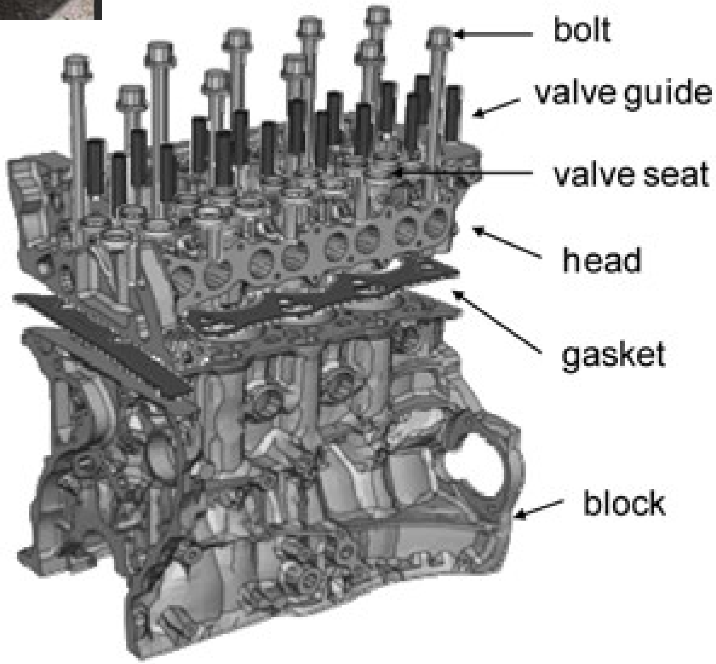


Joint = Fastener + Members

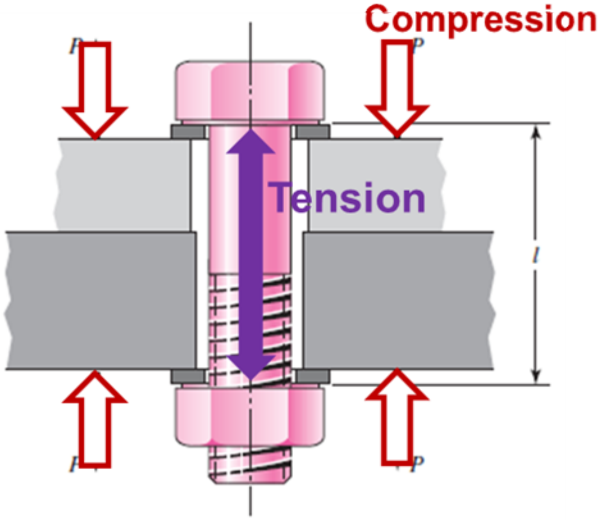
Q: How many joints needed?

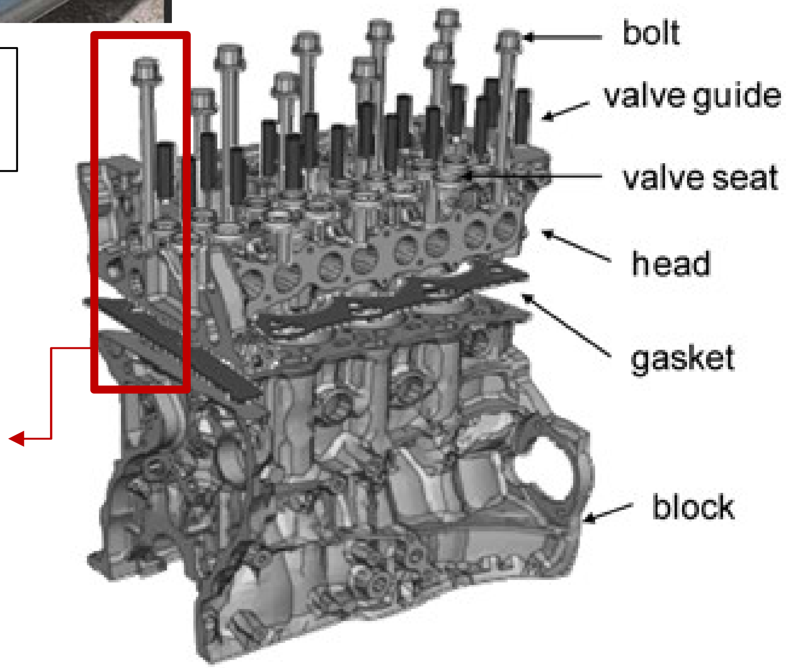


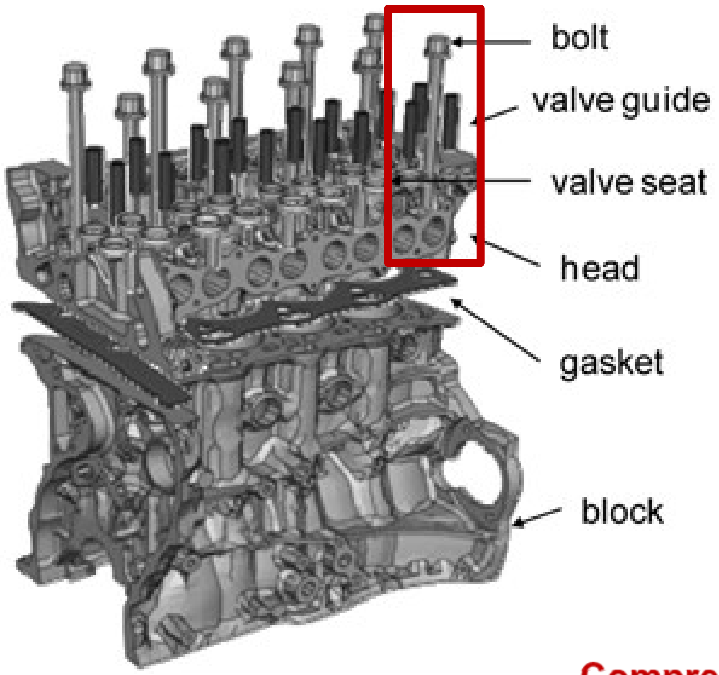




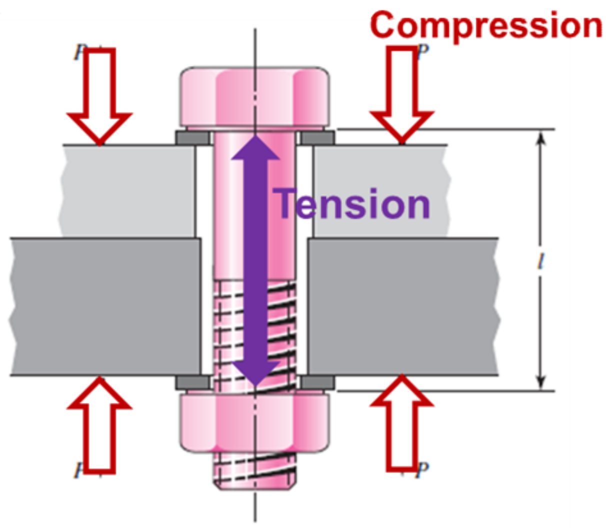
Q: What are the tension/compression acting on each joint?

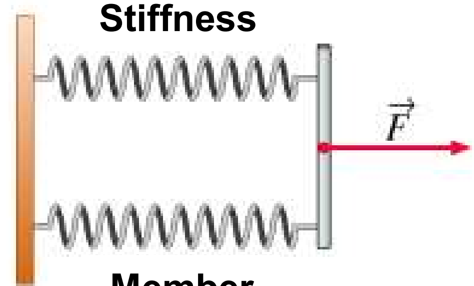






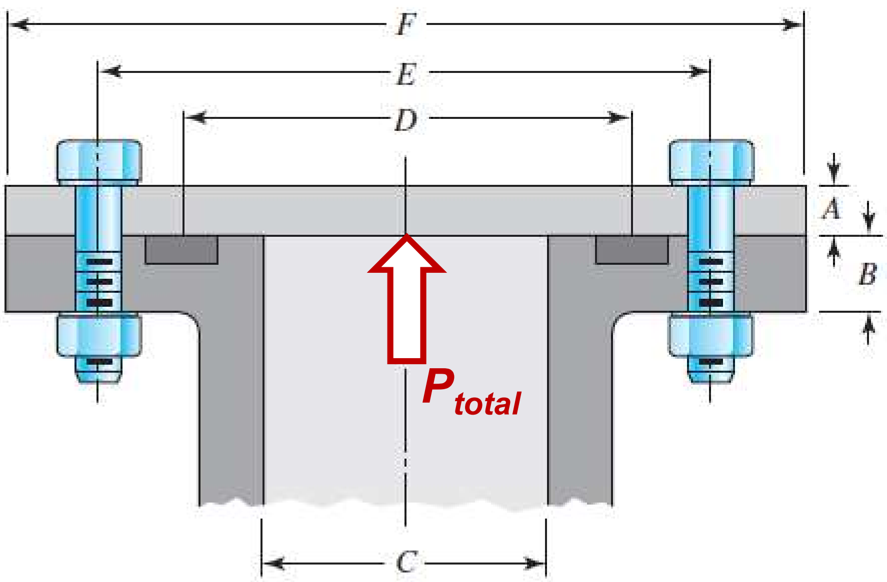
Questions:
How many joints are needed?
What are the tension/compression acting on each joint?

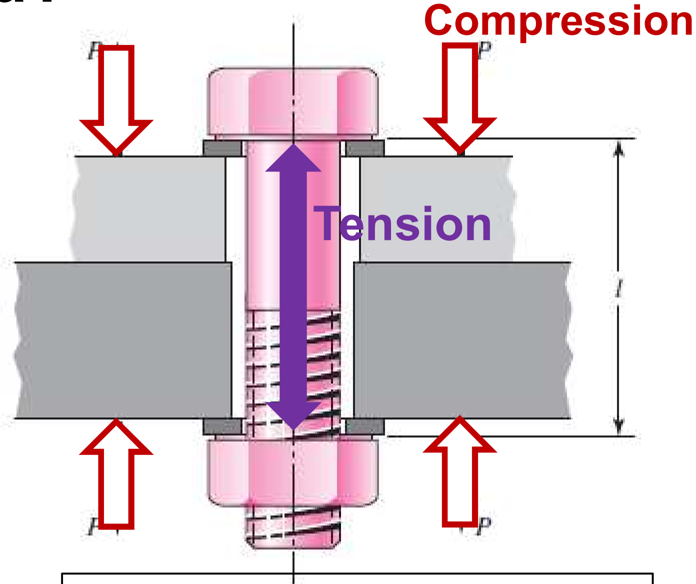




$$\frac{1}{K_{joint}} = \frac{1}{K_{fastener}} + \frac{1}{K_{member}}$$

Why Joints Have to be Tightened?

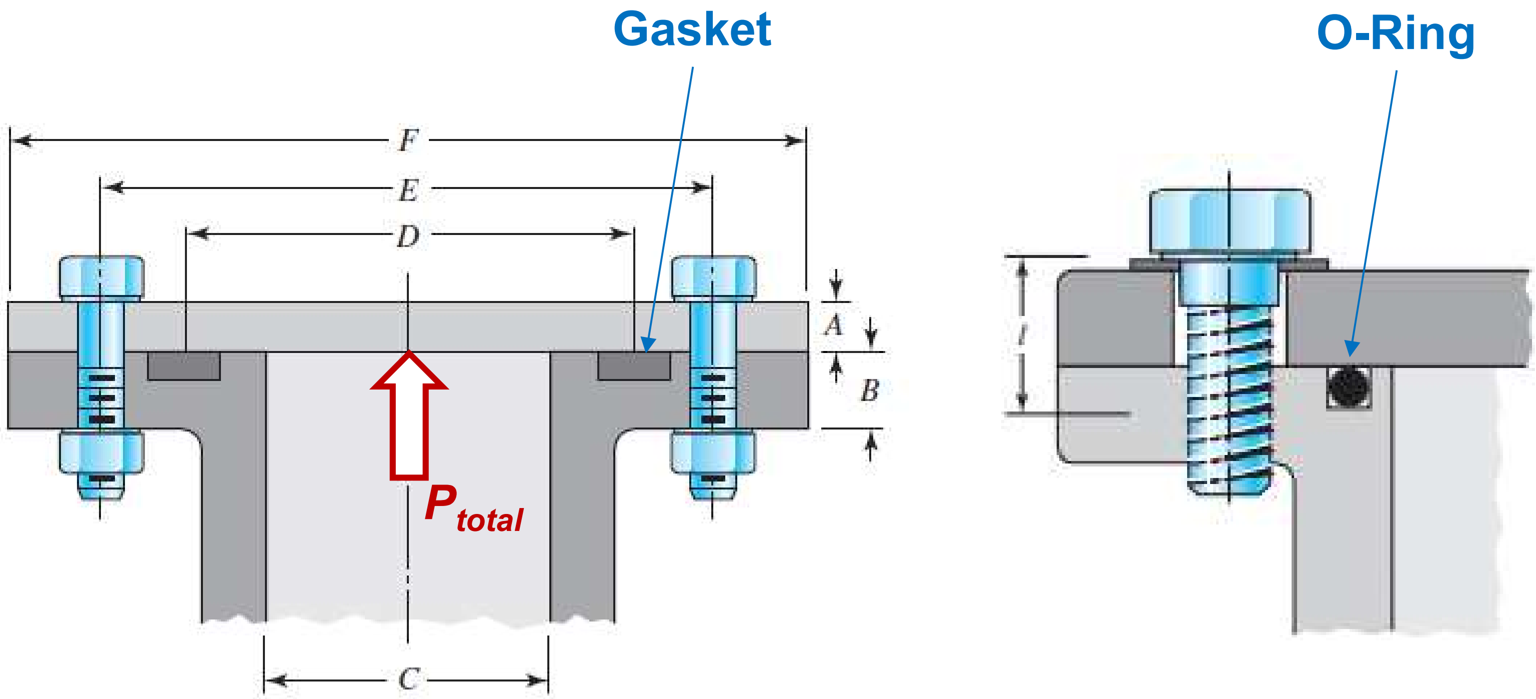


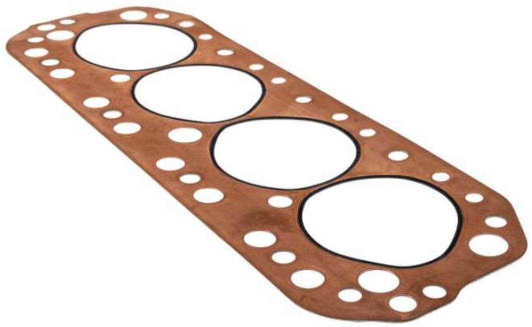


Bolt: Tension
Members: Compression

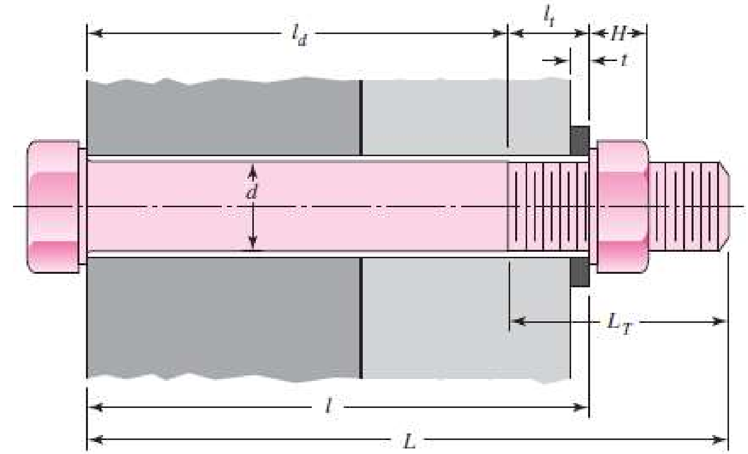
- Purpose of a joint is to clamp two, or more, parts together.
- Clamping force is produced by twisting the nut stretches the bolt to produce the clamping force.
- This clamping force is called the **pretension** or **bolt preload**.

Gaskets or O-Ring Seals for Leakage Prevention





Fastener Stiffness Calculation



- Fastener Length: L
- Thread Length: L_T
- $\boxed{\text{Grip Length: } l}$
- Length of Unthreaded Portion in Grip: $l_d = L - L_T$
- Length of Threaded Portion in Grip: $l_t = l - l_d$
- Stress Area of Unthreaded Portion: $A_d = \pi d^2/4$
- Stress Area of Threaded Portion: A_t (Table 8-1 & 8-2)
- Stiffness of Unthreaded Portion: $k_d = \frac{A_d E}{l_d}$
- Stiffness of Threaded Portion: $k_t = \frac{A_t E}{l_t}$
- Fastener Effective Stiffness k_b

$$k_b = \frac{1}{\frac{1}{k_d} + \frac{1}{k_t}} = \frac{k_d k_t}{k_d + k_t} = \frac{A_d A_t E}{A_d l_t + A_t l_d}$$

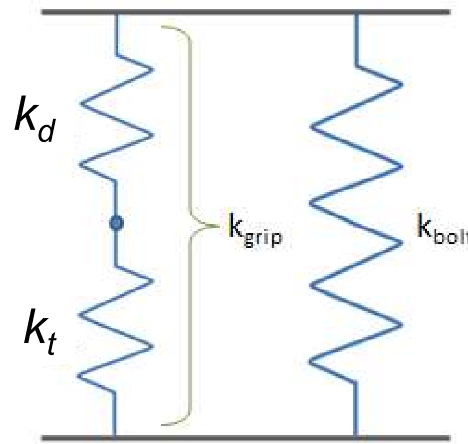


Table 8-7 Procedure for Finding Fastener Stiffness

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

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

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:

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

Metric series:

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

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

Fig. (a)

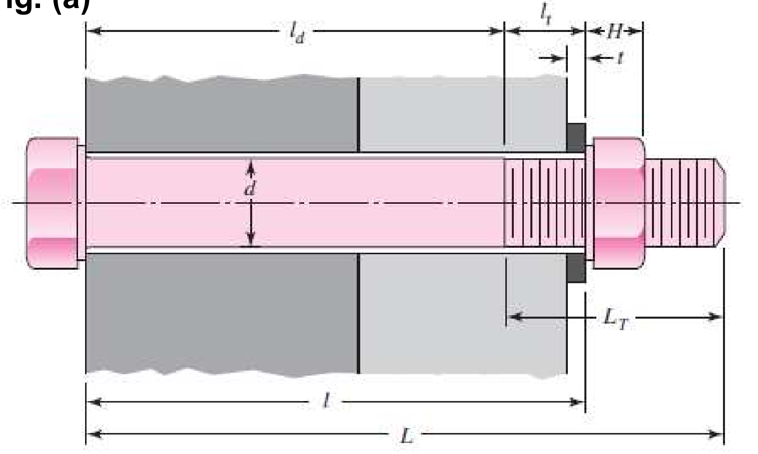
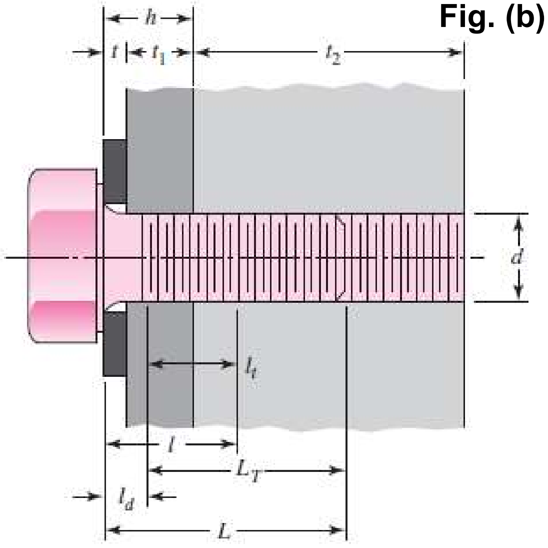


Fig. (b)



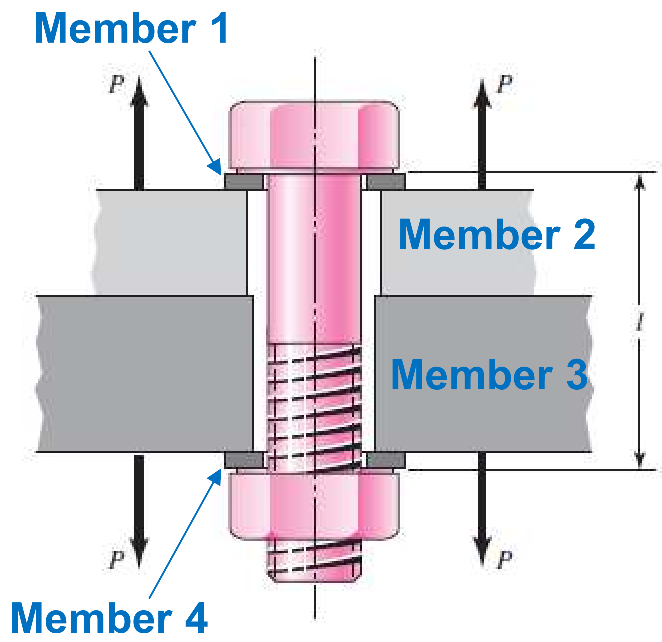
8-5 Joints: Member Stiffness

Member Stiffness Calculation

- Multiple members can be included in the grip of the fastener. All together these act like compressive springs in series.
- Total Member Stiffness

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

- Experimental technique was used to determine the pressure distribution at member interface. Its **effective zone** is approximated by the <u>pressure cone method</u>.



Member Stiffness Calculation (Pressure Cone Method)

Compression of an conical element with thickness dx is $d\delta = \frac{P\,dx}{EA}$

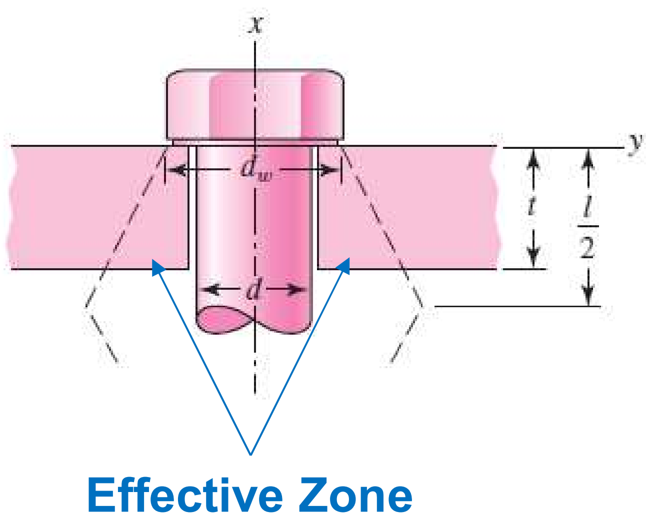
Area of the element $A = \pi\left(r_o^2 - r_i^2\right) = \pi\left[\left(x\tan\alpha + \frac{D}{2}\right)^2 - \left(\frac{d}{2}\right)^2\right]$

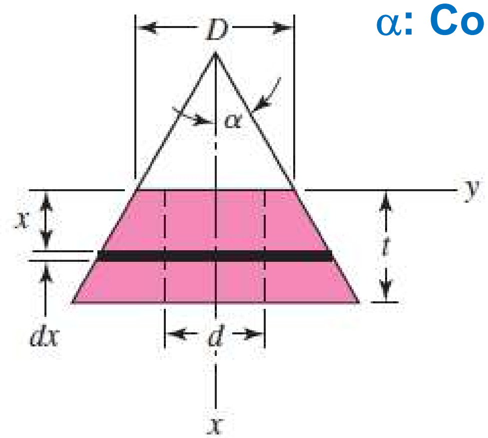
Frustum compression $\delta = \int_0^t d\delta = \frac{P}{\pi E d\ \tan\alpha}\ln\frac{(2t\tan\alpha+D-d)(D+d)}{(2t\tan\alpha+D+d)(D-d)}$

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

α: Cone Angle

Fig 8-15

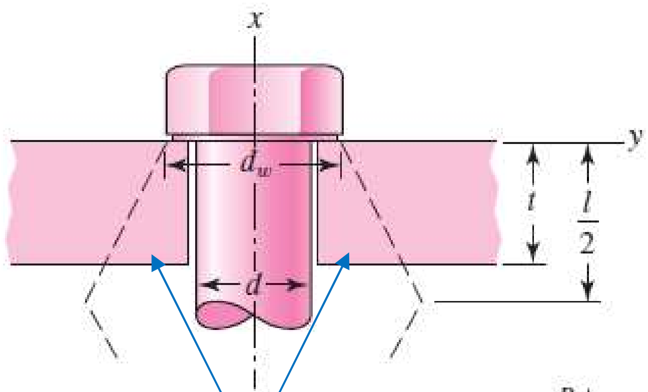




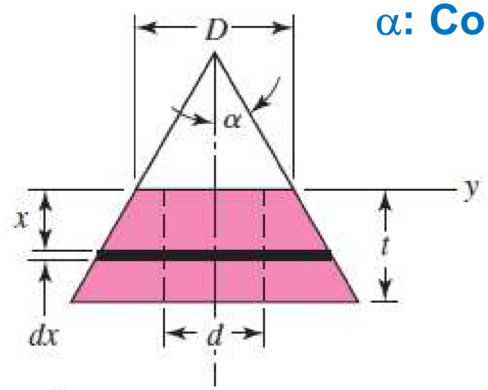
Member Stiffness Calculation (Pressure Cone Method)

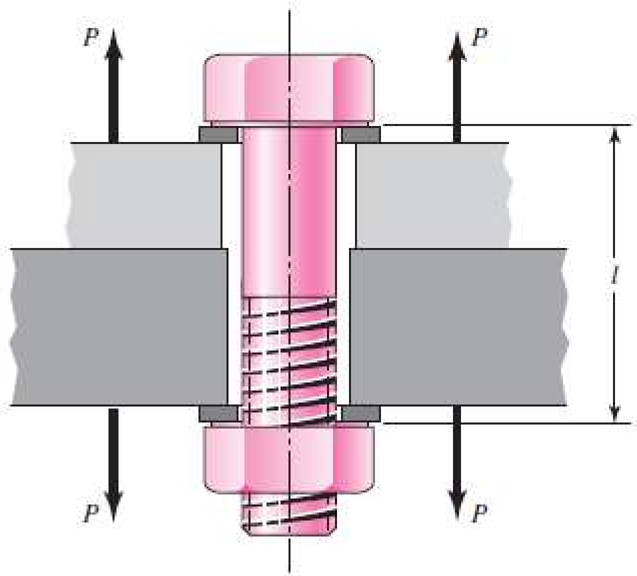
Frustum Stiffness $k = \frac{P}{\delta} = \frac{\pi E d \tan\alpha}{\ln\frac{(2t\tan\alpha + D - d)(D+d)}{(2t\tan\alpha + D + d)(D-d)}}$

α: Cone Angle



Effective Zone





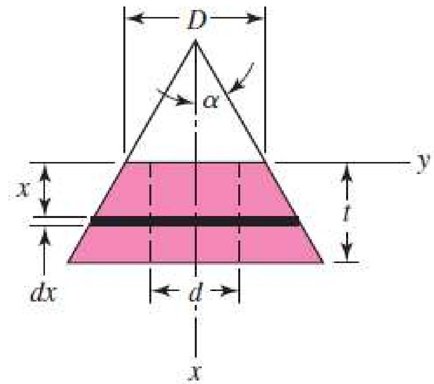
Questions:
D = ?
d = ?
t = ?

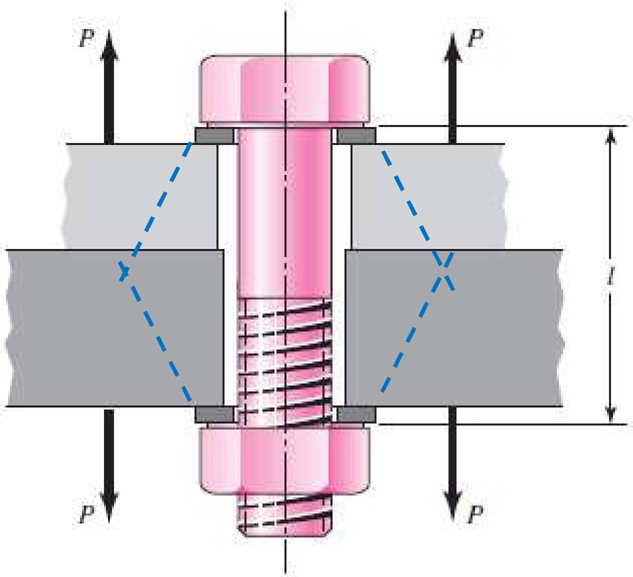
Member Stiffness Calculation (Pressure Cone Method)

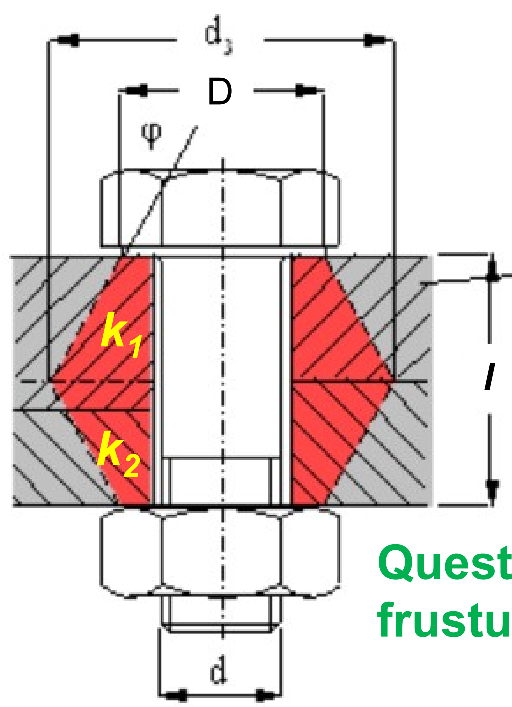
Frustum Stiffness $k = \frac{P}{\delta} = \frac{\pi E d \ \tan\alpha}{\ln\frac{(2t\tan\alpha + D - d)(D+d)}{(2t\tan\alpha + D + \ )(D-d)}}$

Must be solved separately for each frustum in the joint and assembled to obtain member stiffness k_m.

Total Member Stiffness: $\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \cdots + \frac{1}{k_i}$







Regardless of member thickness, depth of the frustum is always half of the grip length.

Question: How many frustums in this case?

Member Stiffness Calculation (Pressure Cone Method)

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

- For most combinations, 25°≤α≤33°
- In this book **α=30deg** was recommended for hardened steel, cast iron, or aluminum members

- Simplification #1: Joint members have the same Young's modulus E with symmetrical frusta back to back

$$k_m = \frac{\pi E d \ \tan\alpha}{\ln\frac{(l\tan\alpha \quad w-d)(d_w+d)}{(l\tan\alpha \quad w+d)(d_w-d)}}$$

k_m=k/2
l=2t
D=d_w=washer face diameter

- Simplification #2: For α=30deg and washer diameter d_w=1.5d, member stiffness k_m is reduced to

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

- Refer to P.427- 428 for more discussions of cone angle α

Member Stiffness Calculation (Simplification #3)

FEA was used to seek simplification in calculation procedures.

Assumptions:

- α=30deg
- washer diameter D=1.5d
- Ignore washer stiffness
- Two members with same thickness and use same material

Member Equivalent Stiffness $\frac{k_m}{Ed} = A\, exp\left(\frac{Bd}{l}\right)$

Table 8-8

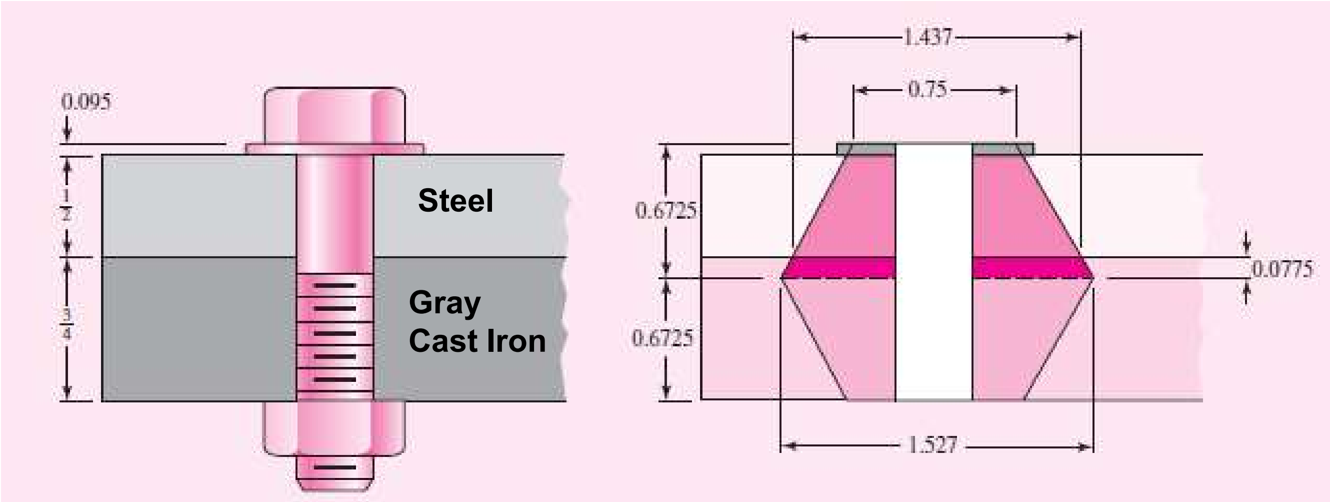
Material Used	Poisson Ratio	Elastic GPa	Modulus Mpsi	A	B
Steel	0.291	207	30.0	0.787 15	0.628 73
Aluminum	0.334	71	10.3	0.796 70	0.638 16
Copper	0.326	119	17.3	0.795 68	0.635 53
Gray cast iron	0.211	100	14.5	0.778 71	0.616 16
General expression				0.789 52	0.629 14

Example 8-2 (Bolt with Nut)

Given: Two plates are clamped by washer-faced ½ in-20 UNF × 1 ½ in SAE grade 5 bolts each with a standard steel plain washer.

Calculate:

- member spring rate k_m if the top plate is steel and the bottom plate is gray cast iron
- member spring rate k_m if both plates are steel and have equal thickness by method of conical frusta
- member spring rate k_m if both plates are steel using Eq. (8–23)
- bolt spring rate k_b

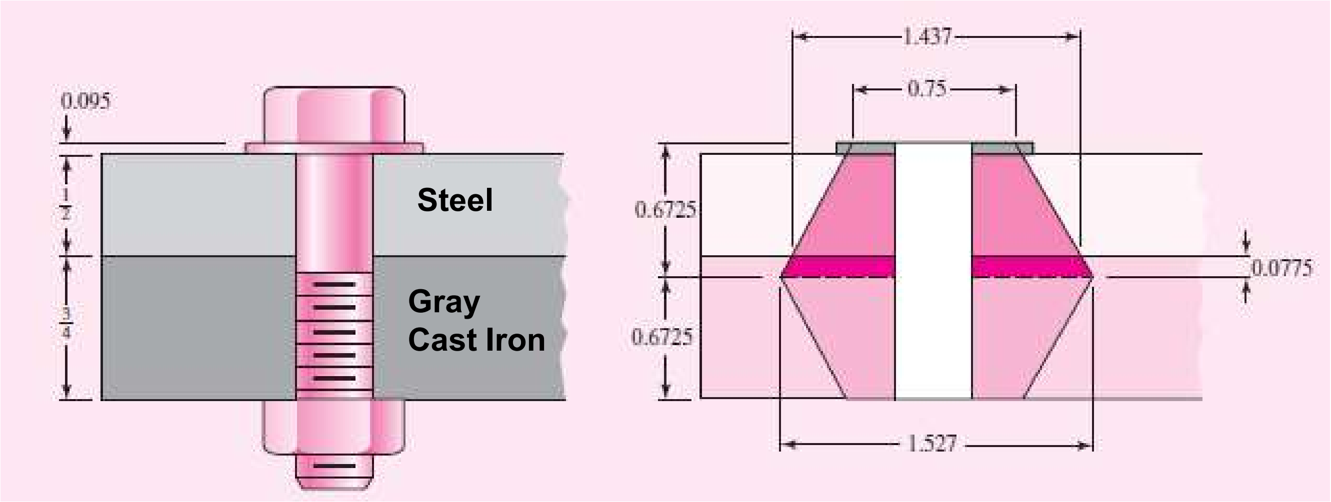


Example 8-2 (Cont'd)

Table A–32 ► thickness of a standard 1/2 N plain washer is 0.095 in.

As shown in figure, the frusta extend halfway into the joint.

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



Example 8-2 (Cont'd)

<u>Member Spring Rate</u>

Upper frustum thickness: Washer + Steel Plate=0.6725-0.0775=0.595"

$$k = \frac{\pi E d \ \tan\alpha}{\ln\frac{(2t\tan\alpha+D-d)(D+d)}{(2t\tan\alpha+D+d)(D-d)}} = \frac{\pi(30\cdot 10^6)\cdot 0.5\cdot 0.5774}{\ln\frac{(2\cdot 0.595\cdot 0.5774+0.75-0.5)(0.75+0.5)}{(2\cdot 0.595\cdot 0.5774+0.75+0.5)(0.75-0.5)}} = 30.8\cdot 10^6\ lbf/in$$

Upper cast iron frustum thickness=0.0775"

$$k = \frac{\pi E d \ \tan\alpha}{\ln\frac{(2t\tan\alpha+D-d)(D+d)}{(2t\tan\alpha+D+d)(D-d)}} = \frac{\pi(14.5\cdot 10^6)\cdot 0.5\cdot 0.5774}{\ln\frac{(2\cdot 0.0775\cdot 0.5774+1.437-0.5)(1.437+0.5)}{(2\cdot 0.0775\cdot 0.5774+1.437+0.5)(1.437-0.5)}} = 285.6\cdot 10^6\ lbf/in$$

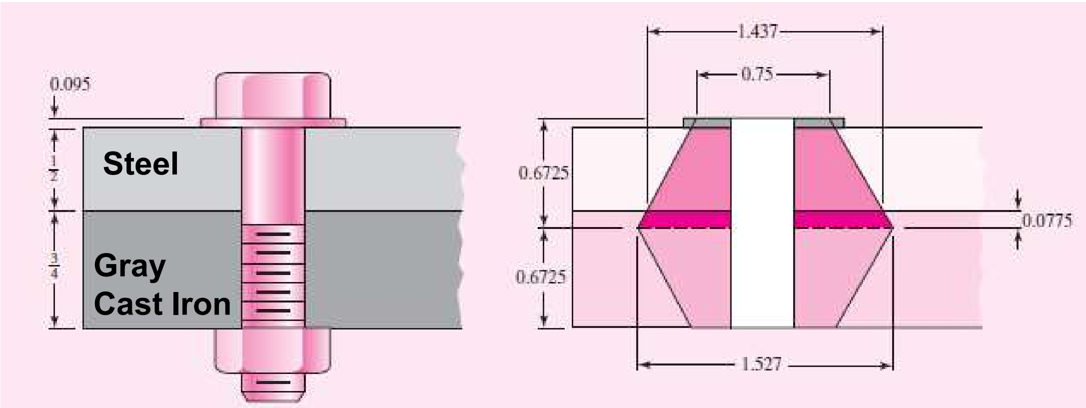
Lower cast iron frustum thickness=0.6725"

$$k = \frac{\pi E d \ \tan\alpha}{\ln\frac{(2t\tan\alpha+D-d)(D+d)}{(2t\tan\alpha+D+d)(D-d)}} = \frac{\pi(14.5\cdot 10^6)\cdot 0.5\cdot 0.5774}{\ln\frac{(2\cdot 0.6725\cdot 0.5774+0.75-0.5)(0.75+0.5)}{(2\cdot 0.6725\cdot 0.5774+0.75+0.5)(0.75-0.5)}} = 14.15\cdot 10^6\ lbf/in$$

Total frustum spring rate:

$$\frac{1}{k_m} = \frac{1}{30.8\cdot 10^6} + \frac{1}{285.6\cdot 10^6} + \frac{1}{14.15\cdot 10^6}$$

$$k_m = 9.378\cdot 10^6\ lbf/in$$



Example 8-2 (Cont'd)

Simplification #2: Member Spring Rate (Both are steel and equal thickness)

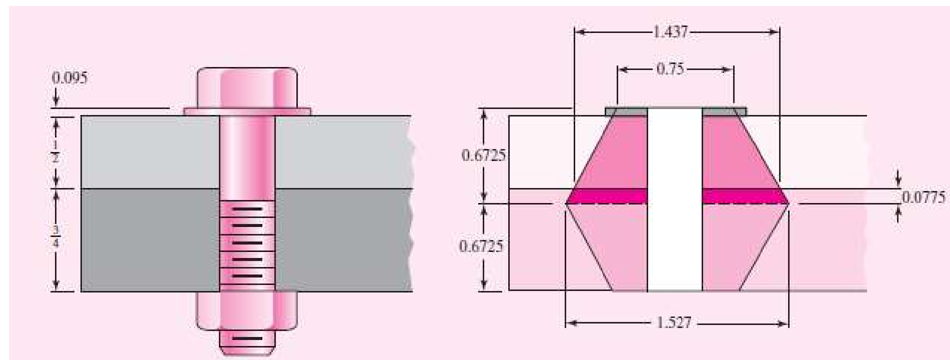
Frustum thickness $l = 2 * 0.6725 = 1.345\ in$

$$k_m = \frac{0.5774\ \pi E d}{2\ \ln\frac{5(0.5774\ l + 0.5d)}{(0.5774\ l + 2.5d)}} = \frac{0.5774\ \pi\left(30\cdot 10^6\right)0.5}{2\ \ln\frac{5(0.5774\cdot\ .345 + 0.5\cdot 0.5)}{(0.5774\cdot\ .345 + 2.5\cdot 0.5)}} = 14.64\cdot 10^6\ lbf/in$$

Simplification #3: Member Spring Rate (Both are steel and equal thickness)

$$\frac{k_m}{Ed} = A\ exp\left(\frac{Bd}{l}\right) = 0.78715\ exp\left(\frac{0.62873\quad .5}{1.345}\right)$$

$$k_m = 14.92\cdot 10^6\ lbf/in$$



Bolt Spring Rate (Table 8-7)

$$L_T = 2d + \frac{1}{4} = 2\cdot 0.5 + 0.25 = 1.25\ in$$

$$l_d = L - L_T = 1.5 - 1.25 = 0.25\ in$$

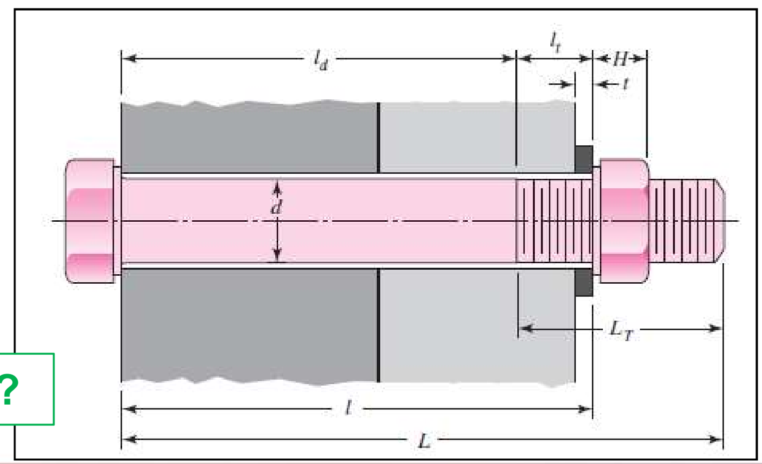
$$l_t = l - l_d = 1.345 - 0.25 = 1.095\ in$$

$$A_d = \frac{\pi\cdot 0.5^2}{4} = 0.1963\ in^2$$

$A_t = 0.1599\ in^2$ (Table 8-2)

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963\cdot 0.1599}{0.1963\cdot 1.095 + 0.1599\cdot 0.25}$$

$k_b = 3.69\cdot 10^6\ lbf/in$ (Generally $k_b < k_m$)



Question: Which should fail first?

Example 8-5 (Tapped Joint)

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.

- Cap screw, 5/8 in-11 UNC, SAE 5;
- Hardened-steel washer, t_w = 1/16 in thick;
- Steel cover plate, t_1 =5/8in, E_s = 30 Mpsi;
- Cast-iron base, t_2 = 5/8in, E_{ci} = 16 Mpsi.

Find k_b and k_m using the assumptions given in the caption of Fig. 8–21.

Solution:

Number of frustums in the joint = ??

- the upper two frusta are steel and
- the lower one is cast iron.

$$h = t_1 + t_w = \left(\frac{5}{8} + \frac{1}{16}\right) = \frac{11}{16} = 0.6875in$$

$$D_2 = 1.5d = 1.5 \cdot \frac{5}{8} = 0.9375\ in$$

Since t_2=d, effective grip

$$l = h + \frac{d}{2} = \left(\frac{11}{16} + \frac{1}{2}\frac{5}{8}\right) = 1\ in$$

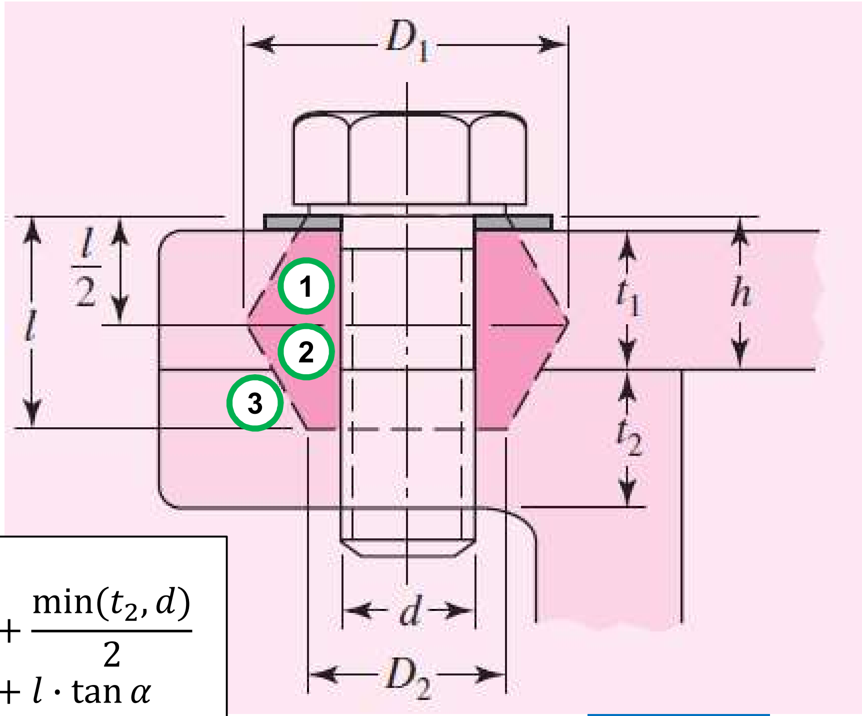


Table 8-7:

$$l = t_1 + t_w + \frac{\min(t_2, d)}{2}$$
$$D_1 = d_w + l \cdot \tan\alpha$$
$$D_2 = d_w = 1.5\ d$$

Example 8-5 (Cont'd)

Upper Frustum #1: $t = \frac{l}{2} = 0.5\ in,\ D = 0.9375\ in,\ d = \frac{5}{8}\ in,\ \alpha = 30deg,\ E = E_s$

$$k_1 := \frac{\pi \cdot E \cdot d \cdot \tan(\alpha)}{\ln\left(\frac{(2 \cdot t \cdot \tan(\alpha) + D - d) \cdot (D + d)}{(2 \cdot t \cdot \tan(\alpha) + D + d) \cdot (D - d)}\right)} = (46.46 \cdot 10^6)\ \frac{lbf}{in}$$

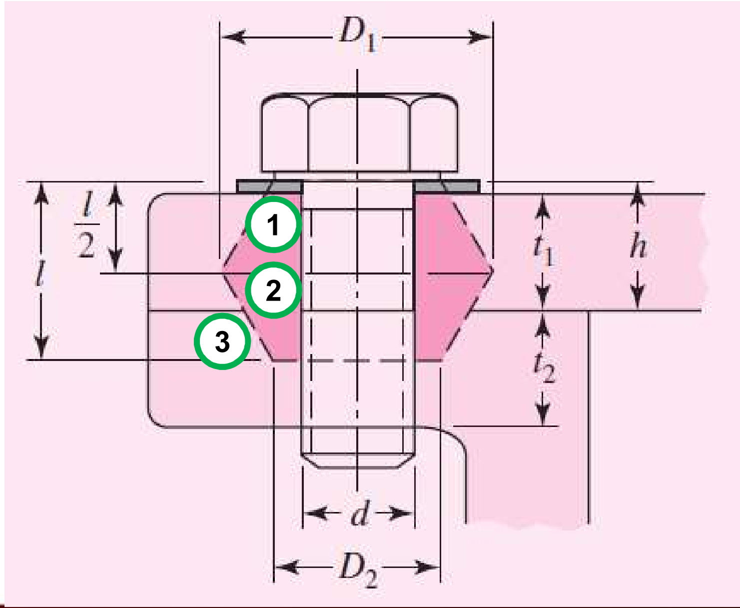
Frustum dia @Interface: $d_w + 2(l - h) \cdot \tan\alpha = 1.2983\ in$

Middle Frustum #2: $t = t_2 = h - \frac{l}{2} = 0.1875\ in,\ D = 1.2983\ in,\ d = \frac{5}{8}\ in,\ \alpha = 30deg,\ E = E_s$

$$k_2 := \frac{\pi \cdot E \cdot d \cdot \tan(\alpha)}{\ln\left(\frac{(2 \cdot t \cdot \tan(\alpha) + D - d) \cdot (D + d)}{(2 \cdot t \cdot \tan(\alpha) + D + d) \cdot (D - d)}\right)} = (197.58 \cdot 10^6)\ \frac{lbf}{in}$$

Bottom Frustum #3: $t = l - h = 0.3125\ in,\ D = 0.9375\ in,\ d = \frac{5}{8}\ in,\ \alpha = 30deg,\ E = E_{ci}$

$$k_3 := \frac{\pi \cdot E \cdot d \cdot \tan(\alpha)}{\ln\left(\frac{(2 \cdot t \cdot \tan(\alpha) + D - d) \cdot (D + d)}{(2 \cdot t \cdot \tan(\alpha) + D + d) \cdot (D - d)}\right)} = (32.397 \cdot 10^6)\ \frac{lbf}{in}$$



Total member stiffness:

$$k_m := \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3}\right)^{-1} = (17.406 \cdot 10^6)\ \frac{lbf}{in}$$

Example 8-5 (Cont'd)

Bolt Stiffness:

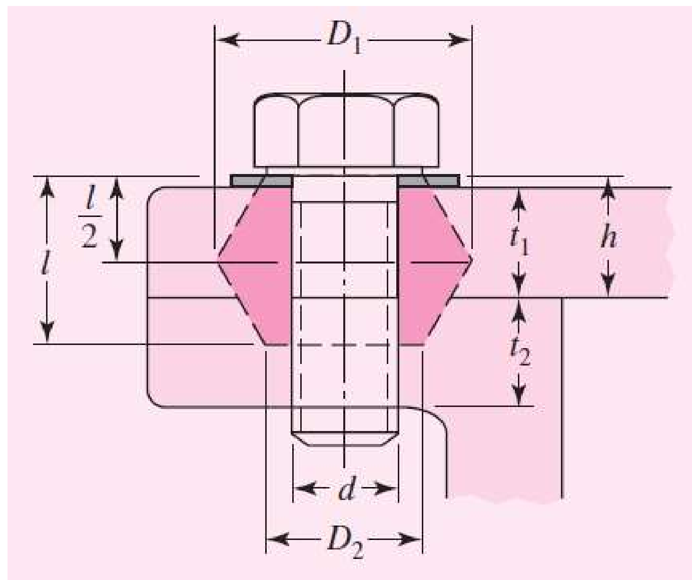
Cap screw is threaded all the way

From Table 8-2, stress area in the thread: $A_t = 0.226\ in^2$

Effective grip length: $l = h + \frac{d}{2} = \left(\frac{11}{16} + \frac{1}{2}\frac{5}{8}\right) = 1\ in$

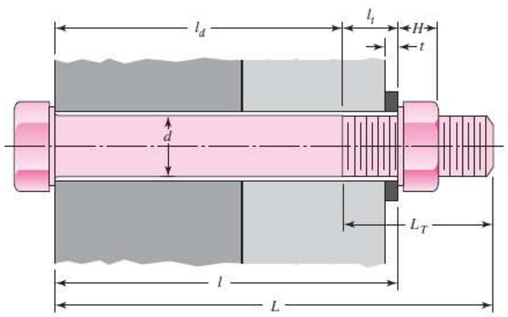
Hence, bolt stiffness is

$$k_b := \frac{A_t \cdot E}{l} = \left(6.78 \cdot 10^6\right) \frac{lbf}{in}$$



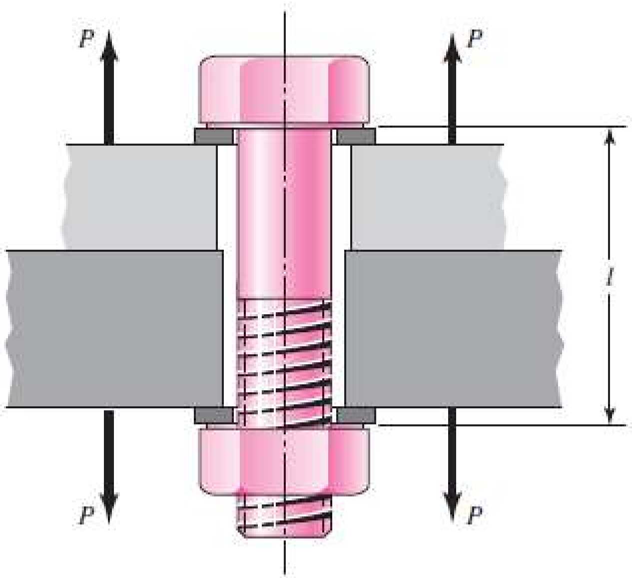
Summary of Pressure Cone Method

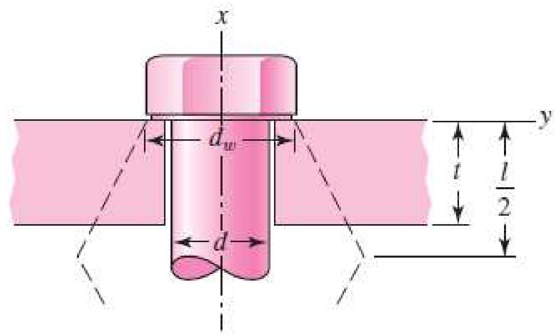
Joint Stiffness Calculation

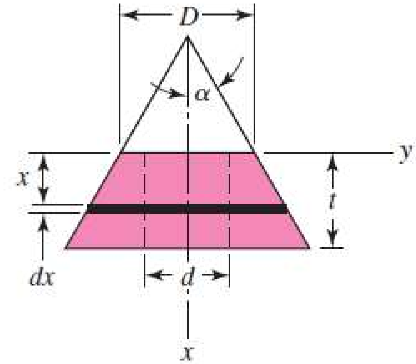


1. Bolt Stiffness k_b

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







2. Member Stiffness (Pressure Cone Method)

$$k = \frac{P}{\delta} = \frac{\pi E d \ \tan\alpha}{\ln\frac{(2t\tan\alpha + D - d)(D + d)}{(2t\tan\alpha + D + d)(D - d)}} \qquad \text{(Method \#1)}$$

Let joint members have the same material, k_m=k/2; l=2t; α=30deg and washer diameter d_w=1.5d

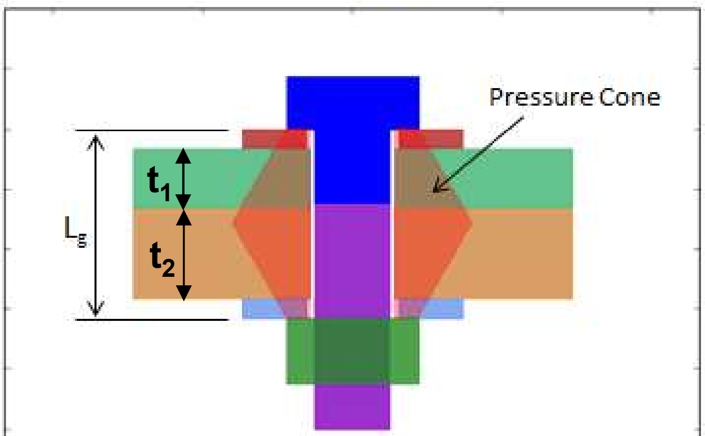
$$k_m = \frac{0.5774\,\pi E d}{2\ln\frac{5(0.5774\,l + 0.5d)}{(0.5774\,l + 2.5d)}} \qquad \text{(Method \#2)}$$

or $\frac{k_m}{Ed} = A\,exp\left(\frac{Bd}{l}\right)$ (Method #3)

Effective Grip Length

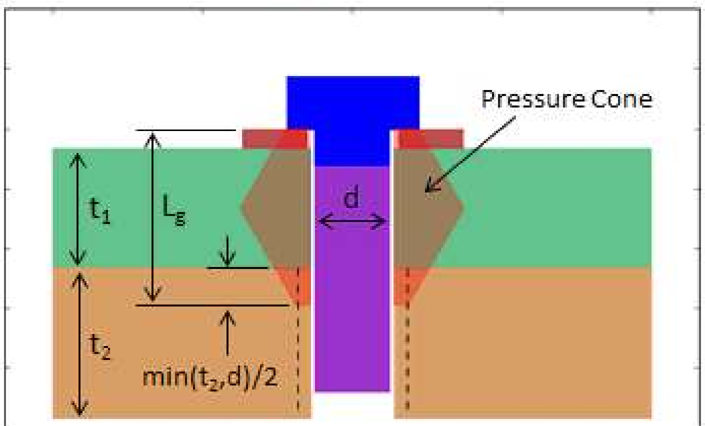
- Height of the pressure cone depends on the grip length, L_g, which is the combined thickness of the parts being clamped in the joint.
- Bolt with Nut
  - pressure cone starts under the head of the bolt and ends under the nut
  - frustum diameters in this case is determined using the diameters of the bearing faces
- Tapped Joint (Cap Screw)
  - pressure cone starts under the head of the bolt and ends in the threaded portion of the final plate.

Bolt With Nut



$$L_g = t_1 + t_2 + 2t_w$$

Tapped Joint



$$L_g = t_1 + t_w + \frac{min(t_2, d)}{2}$$

8-6 Bolt Strength

- Typically the bolt fails first.
  - It is the least expensive
  - It is the most easily replaced

Bolt Strength

- **Proof Load**: Maximum load that a bolt can withstand without acquiring a permanent set (force)
- **Proof Strength**: proportional limit and corresponds to 0.0001-in permanent set in the fastener (stress)

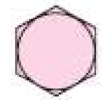
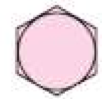
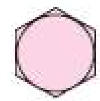
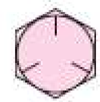
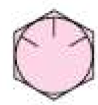
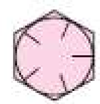
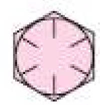
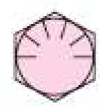
- Minimum Strength Specification of Steel Bolts
  - Table 8-9 (SAE), p.425
  - Table 8-10 (ASTM), p.426
  - Table 8-11 (Metric Bolts), p.427

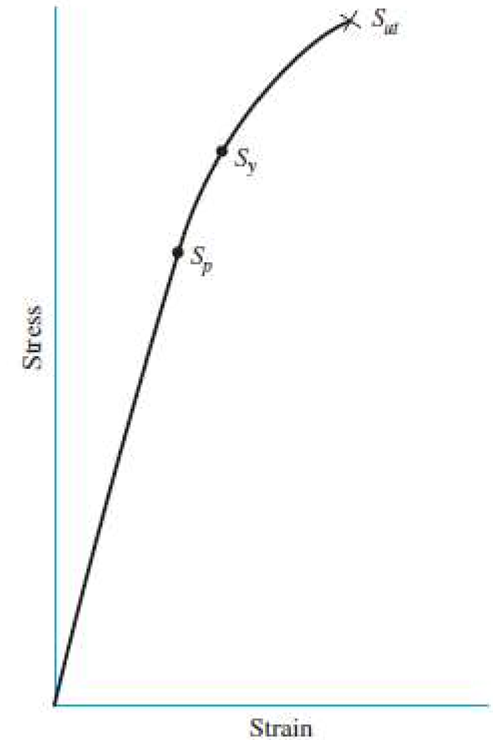
- The grade of the nut should be the same grade as the bolt.

Min Proof Strength/Min Tensile Strength/Min Yield Strength

Table 8-9

SAE Specifications for Steel Bolts

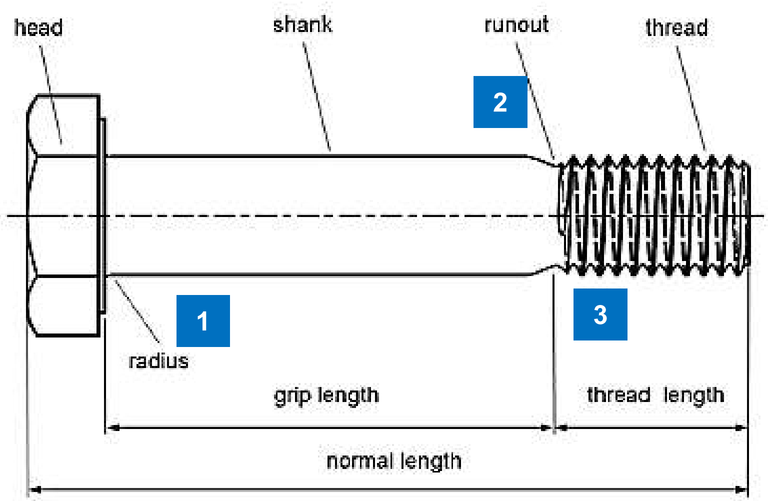
SAE Grade No.	Size Range Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
1	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low or medium carbon	
2	$\frac{1}{4}$–$\frac{3}{4}$	55	74	57	Low or medium carbon	
	$\frac{7}{8}$–$1\frac{1}{2}$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

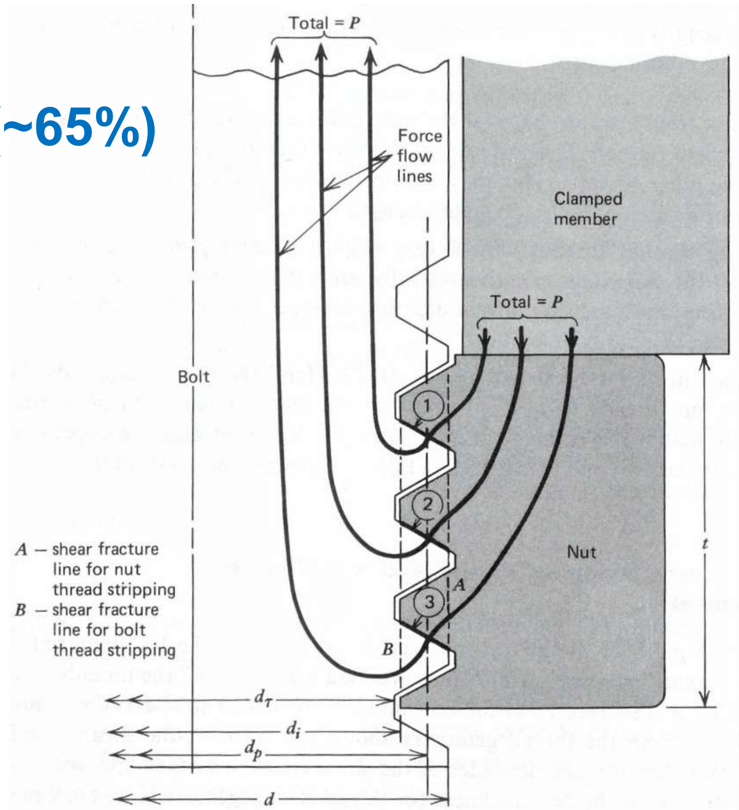


Proof strength is less than yield strength

Bolt Failures

- With washer protection of the shoulder fillet and thread runout ≤15°. Bolts in fatigue axial loading failed mostly at:
  1. Fillet under head (~15%)
  2. Thread runout (~20%)
  3. First engaged thread in the nut **(~65%)**

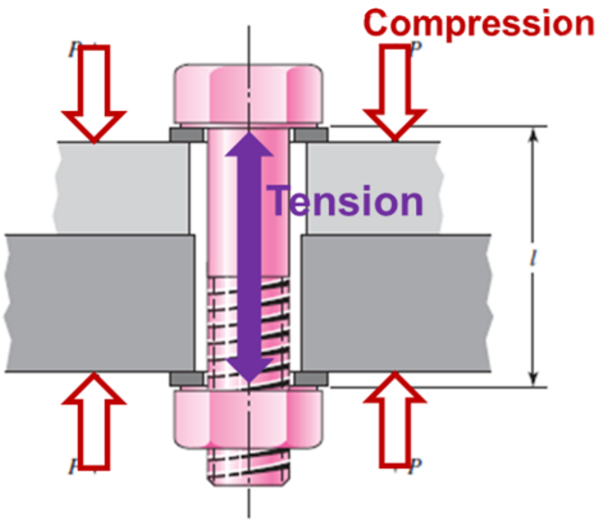




8-7 Tension Joints: The External Load

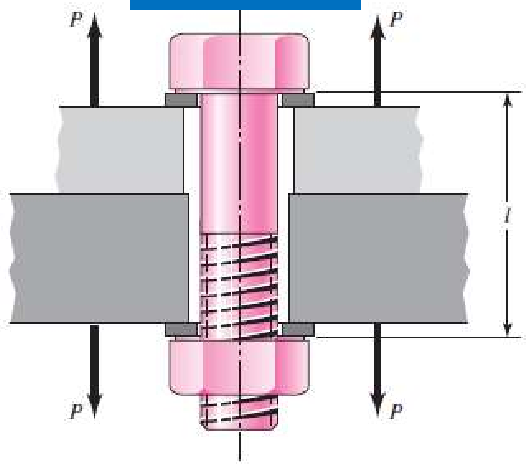
- Total load acting on each joint is the summation of preload and external load

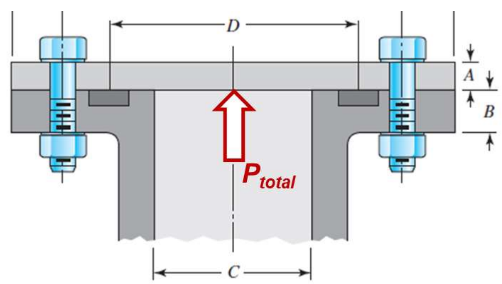
Preload



Bolt: Tension
Members: Compression

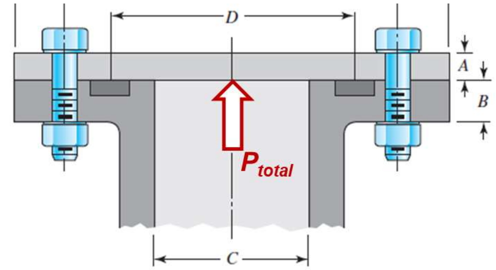
External Load





Total Joint Load

Load-Sharing Between Bolt and Members



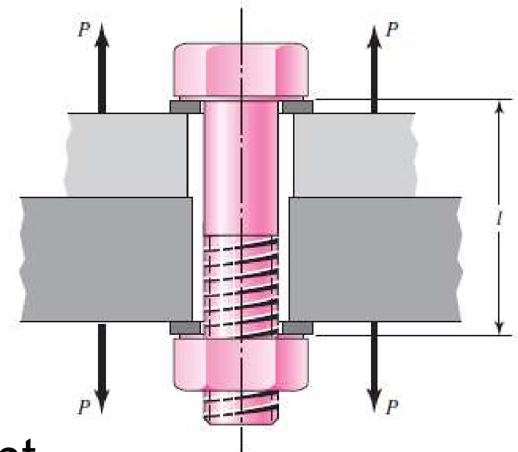
P_{total} = Total external tensile load applied to the joint

P = external <u>tensile</u> load per bolt

$$P = P_b + P_m = \frac{P_{total}}{N} = (k_b + k_m) \cdot \delta$$

P_b = portion of P taken by bolt

P_m = portion of P taken by members



Fraction of P carried by bolt $C = \frac{k_b}{k_b + k_m}$

$$P_b = CP \qquad P_m = (1 - C)P$$

C is also known as the **Stiffness Constant** of the Joint

Question: Why is the sign different?

F_i = bolt preload

Resultant Bolt Load $F_b = P_b \boxed{+} F_i = CP + F_i$

Resultant Member Load $F_m = P_m \boxed{-} F_i = (1 - C)P - F_i$

Load-Sharing Between Bolt and Members

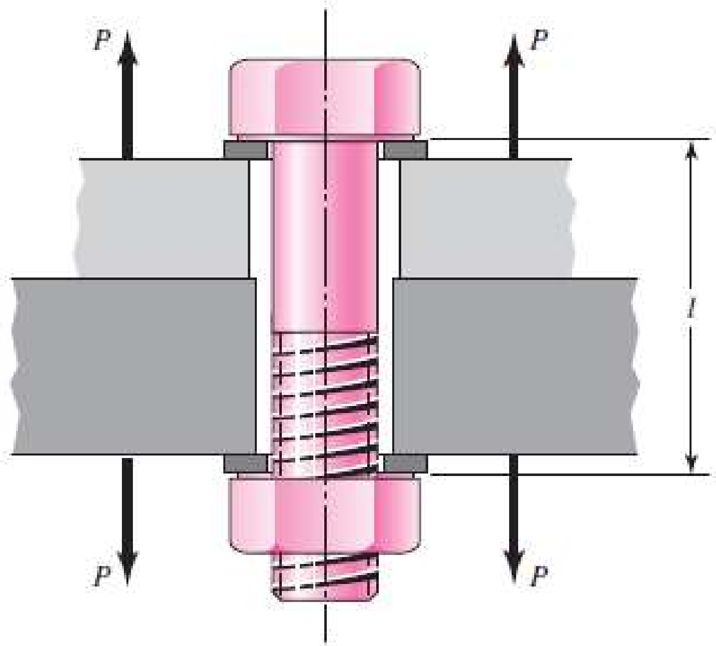
- Table 8–12: Steel members clamped using a $\frac{1}{2}\,\mathrm{in} - 13\ \mathrm{NC}$ bolt

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

- Noted that:
  - members take 80+ percent of the external load
  - making the bolt grip longer causes the members to take an even greater percentage of the external load. Why?

8-8 Relating Bolt Torque to Bolt Tension

Question: Preload is applied to any fastener. How to ensure the preload is developed and at its desired magnitude?



Bolt Preload Control

- In industrial applications, bolt preload is generated and controlled by
  - Torque wrench (most controllable and accurate)
  - Pneumatic-impact wrench
  - Turn-of-the-nut method (very time-consuming with wishy-washy outcome)

Relationship Between Bolt Preload and Torque

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

Given

- $\tan\lambda = \frac{l}{\pi d_m}$ $\quad T_R = \frac{F_i d_m}{2}\left(\frac{\tan\lambda + f\sec\alpha}{1 - f\tan\lambda\sec\alpha}\right) + \frac{F_i f_c d_c}{2}$
- Washer dia=$1.5d_m \sim 1.5d$ $\quad d_c = \frac{d + 1.5d}{2} = 1.25d$

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

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

- K is the **torque factor**
- K=0.2 is the most commonly used torque factor

Table 8-15

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

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.

Bolt stiffness k_b = 6.50 Mlbf/in and Joint Stiffness k_m = 13.8 Mlbf/in

- Determine the preload and service load stresses in the bolt. Compare these to the SAE minimum proof strength of the bolt.
- Specify the torque necessary to develop the preload, using Eq. (8–27).
- Specify the torque necessary to develop the preload, using Eq. (8–26) with f=fc = 0.15.

Solution:

For $\frac{3}{4}$ in-16 UNF × $2\frac{1}{2}$ in bolt, from Table 8-2, $A_t = 0.373\ in^2$

Preload stress in threaded area: $\frac{F_i}{A_t} = \frac{25}{0.373} = 67.02\ ksi$

Stiffness Constant $C = \frac{k_b}{k_b+k_m} = \frac{6.5}{6.5+13.8} = 0.32$

Stress under service load $\frac{F_b}{A_t} = \frac{CP+F_i}{A_t} = \frac{0.32 \cdot 6+25}{0.373} = 72.17\ ksi$

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 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	[illegible]	[illegible]
1	0.0730	64	0.002 63	0.002 18	72	[illegible]	[illegible]
2	0.0860	56	0.003 70	0.003 10	64	[illegible]	[illegible]
3	0.0990	48	0.004 87	0.004 06	56	[illegible]	[illegible]
4	0.1120	40	0.006 04	0.004 96	48	[illegible]	[illegible]
5	0.1250	40	0.007 96	0.006 72	44	[illegible]	[illegible]
6	0.1380	32	0.009 09	0.007 45	40	[illegible]	[illegible]
8	0.1640	32	0.014 0	0.011 96	36	[illegible]	[illegible]
10	0.1900	24	0.017 5	0.014 50	32	[illegible]	[illegible]
12	0.2160	24	0.024 2	0.020 6	28	[illegible]	[illegible]
$\frac{1}{4}$	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

ANSI B1.1-1974 and B18.3.1-1978

Minor Diameter: $d_r = d - 1.299038p$

Pitch Diameter: $d_p = d - 0.649519p$

$$Tensile\ Stress\ Area = \frac{\pi}{4}\left(\frac{d_r + d_p}{2}\right)^2$$

SCUPI

Example 8-3 (Cont'd)

For $\frac{3}{4}$ in-16 UNF × $2\frac{1}{2}$ in SAE grade 5 bolt, min proof strength S_p=85 ksi (Table 8-9)

Stress under service load $\frac{F_b}{A_t} = \frac{CP+F_i}{A_t} = \frac{0.32\cdot 6+25}{0.373} = 72.17\ ksi\ < S_p$

Torque for preload $T = KF_i d = 0.2 \cdot 25000 \cdot 0.75 = 3750\ in \cdot lbf$

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	$\frac{1}{4}$–$\frac{3}{4}$	55	74	57	Low or medium carbon	
	$\frac{7}{8}$–$1\frac{1}{2}$	33	60	36		
4	$\frac{1}{4}$–$1\frac{1}{2}$	65	115	100	Medium carbon, cold-drawn	
5	$\frac{1}{4}$–1	$\boxed{85}$	120	92	Medium carbon, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
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	

Use Eq 8-26 for preload calculation

Table 8-2, Minor Dia Area $A_r = 0.351\ in^2$

Minor Dia: $d_r = \sqrt{\frac{4A_r}{\pi}} = 0.6685\ in$

Mean Dia: $d_m = \frac{0.75+0.6685}{2} = 0.7093\ in$

For UNC/UNF thread, $\alpha = 30°$

$\tan\lambda = \frac{l}{\pi d_m} = \frac{1/1}{\pi\cdot 0.7093} = 0.0280 \quad \lambda = 1.6066°$

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

$$T_R = \left[\frac{0.7093}{2\cdot 0.75}\left(\frac{\tan 1.6066° + 0.15\cdot\sec 30°}{1 - 0.15\cdot\tan 1.6066°\sec 30°}\right) + 0.625\cdot 0.15\right] 25000\cdot 0.75 = 3551\ in\cdot lbf$$

$K = 0.189 < 0.2$

Risk Assessment
8-9 Statically Loaded Tension Joint with Preload

- Static applications
- Joint with preload
- External load exerts additional tensile force on bolt

Relationship Between Preload and Bolt Strength

1. Safety Factor Against Proofing Strength

Tensile stress in bolt $\sigma_b = \frac{F_b}{A_t} = \frac{CP+F_i}{A_t}$

Tensile load in bolt $n_p(CP + F_i) = n_p\sigma_b A_t = S_p A_t$

Yielding factor of safety against proof strength: $n_p = \frac{S_p}{\sigma_b}$

S_p is proof strength of the bolt material

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

Design Target: n_p= ~1 (it is typical to load the joint up to proof strength)

Relationship Between Preload and Bolt Strength

2. Safety Factor Against Overloading by External Load

Apply safety factor to external loading only: $C(n_L P) + F_i = S_p A_t = F_b$

Tensile stress in bolt up to proof strength $S_p = \frac{F_b}{A_t} = \frac{C(n_L P)+F_i}{A_t}$

$$n_L = \frac{S_p A_t - F_i}{C \cdot P}$$

Relationship Between Preload and Bolt Strength

3. Safety Factor Against Separation

Find limit of external load to prevent joint from separation.

P_0 : Minimum external load that causes joint separation

When separation occurs: $F_m = 0$ $\quad (1 - C)P_0 - F_i = 0$

Re-arrange the equation and express it as $P_0 = \frac{F_i}{1-C}$

Define factor of safety against joint separation $n_0 = \frac{P_0}{P}$

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

Relationship Between Preload and Bolt Strength

To further simplify the design calculations, several proof strength had been proposed for joint preload design.

A common practice for **both static and fatigue loading** is to use

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

, where F_p is the proof load and

$$F_p = S_p A_t$$

- Get S_p from Tables 8-9 to 8-11
- For other materials, use $S_p = 0.85S_y$

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.

- Determine k_b, k_m, and C.
- Find the number of bolts required for a load factor of 2 where the bolts may be reused when the joint is taken apart.
- With the number of bolts obtained above, determine the realized load factor for overload, the yielding factor of safety, and the load factor for joint separation.

Solution:

Grip $l = 1.5\ in$

From Table A–31, nut thickness = $\frac{35}{64} in$ (p.1063)

Bolt length $L = 1.5 + \frac{35}{64} + \frac{2}{11} = 2.229\ in$

Table A-17 (p.1043), the closest bolt length L = $2\frac{1}{4}$ in

Since $L < 6\ in$, $L_t = 2d + \frac{1}{4} = 2 \cdot 0.625 + 0.25 = 1.5\ in$

$l_d = 2.25 - 1.5 = 0.75\ in$

$l_t = l - l_d = 1.5 - 0.75 = 0.75\ in$

Table 8-2, $A_t = 0.226\ in^2$

Major Dia Area $A_d = \frac{\pi \cdot 0.625^2}{4} = 0.3068\ in^2$

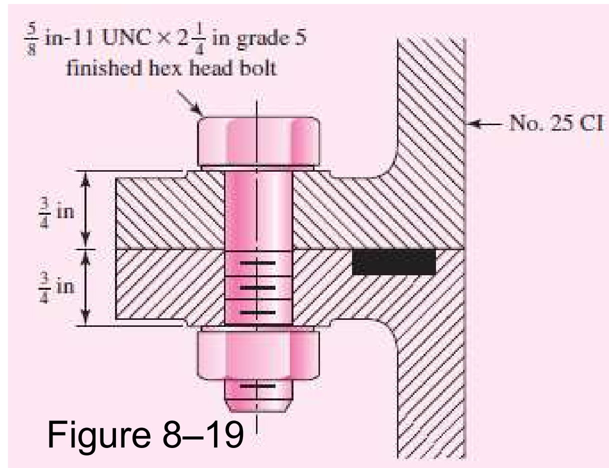
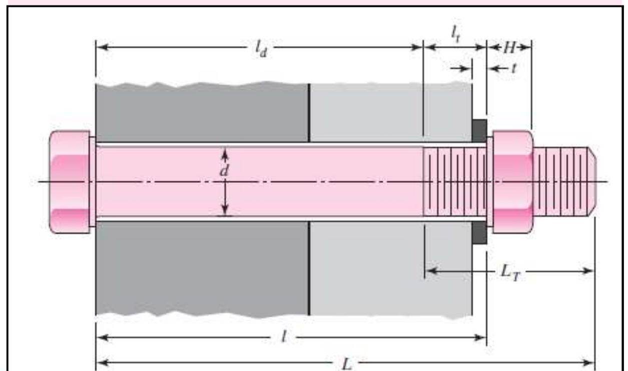


Figure 8–19



Example 8-4 (Cont'd)

Bolt Stiffness $k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.3068 \cdot 0.226 \cdot 30}{0.3068 \cdot 0.75 + 0.226 \cdot 0.75} = 5.21\ Mlbf/in$

Table A-24, No. 25 CI: E=14 Msi

Member Stiffness $k_m = \frac{0.5774\ \pi E d}{2 \ln \frac{5(0.5774\ l + 0.5d)}{(0.5774\ l + 2.5d)}} = \frac{0.5774\ \pi\left(14 \cdot 10^6\right) 0.62}{2 \ln \frac{5(0.5774 \cdot 1.5\ + 0.5 \cdot 0.625)}{(0.5774 \cdot 1.5\ + 2.5 \cdot 0.625)}} = 8.95\ Mlbf/in$

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

Table 8-9, $S_p = 85\ ksi$

Recommended preload

$F_i = 0.75\, S_p A_t = 0.75 \cdot 85 \cdot 0.226 = 14.4\ kip$

To resist overloading $n_L = \frac{S_p A_t - F_i}{C \cdot (P_{total}/N)} = 2$

Calculate N=5.52; so six bolts should be sufficient.

Using N=6;

Actual load factor $n_L = \frac{S_p A_t - F_i}{C \cdot P} = \frac{85 \cdot 0.226 - \quad .4}{0.368(36/\quad)} = 2.18$

SF against proof strength $n_p = \frac{S_p A_t}{CP + F_i} = \frac{85 \cdot 0.226}{0.368 \cdot 6 + 14.4} = 1.16$

SF against joint separation $n_0 = \frac{F_i}{P(1-C)} = \frac{14.4}{6(1-0.368)} = 3.80$

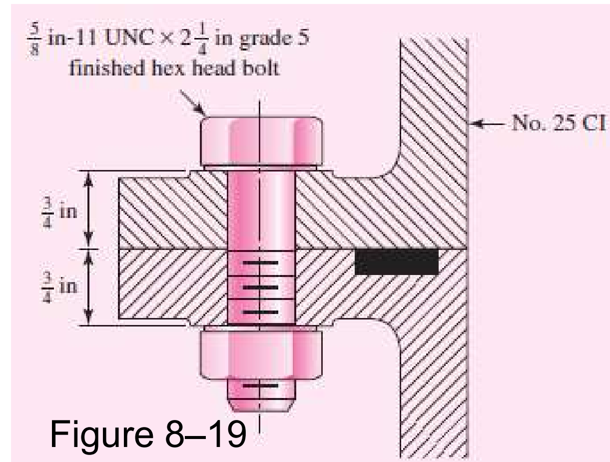


Figure 8–19