ME303 Introduction to Mechanical Design

Lecture 11
Non-permanent Joints (Screws, Fasteners)

Song Chaoyang

Assistant Professor

Department of Mechanical and Energy Engineering

songcy@sustech.edu.cn

SUSTech
Southern University of Science and Technology

Agenda

Week 10, Wednesday, Nov 11, 2019

- Thread Standards and Definitions
- The Mechanics of Power Screws
- Threaded Fasteners
- Joints—Fastener/Member Stiffness
- Tension Joints—The External Load
- Relating Bolt Torque to Bolt Tension
- Statically Loaded Tension Joint with Preload

Introduction

The helical-thread screw was undoubtably an extremely important mechanical invention.

- One of the key targets of current design for manufacture
  - To reduce the number of fasteners.
  - However, there will always be a need for fasteners to facilitate disassembly for whatever purposes.
- It is the basis of power screws,
  - Change angular motion to linear motion
  - To transmit power or to develop large forces (presses, jacks, etc.), and
  - Threaded fasteners, an important element in nonpermanent joints.

2.5 million fasteners
Several $/piece

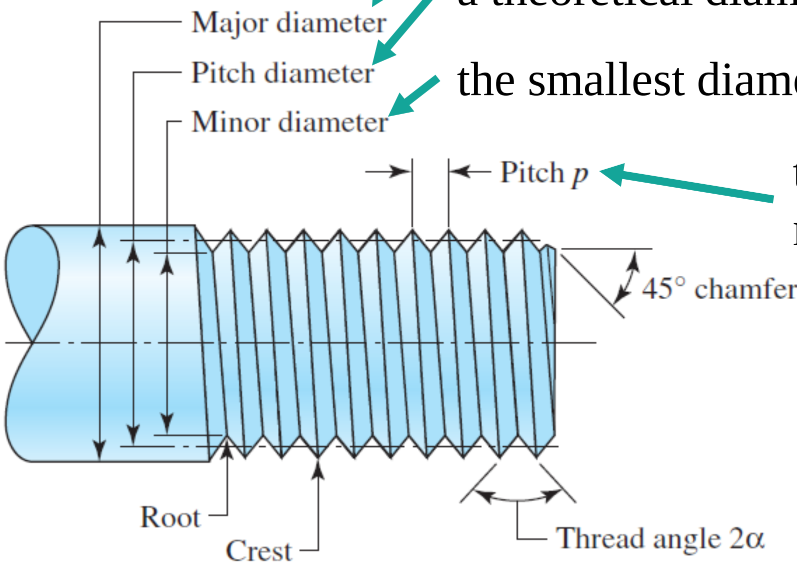
- Typical methods of fastening or joining parts
  - bolts, nuts, cap screws, setscrews, rivets, spring retainers, locking devices, pins, keys, welds, and adhesives.

Thread Standards and Definitions

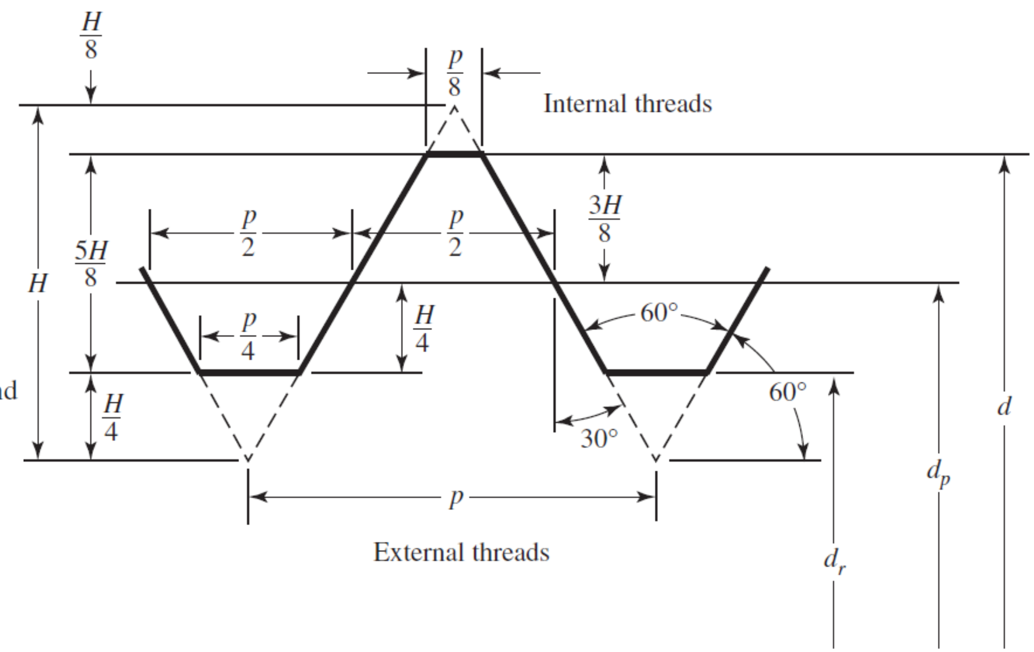
the largest diameter of a screw thread

a theoretical diameter between the major and minor diameters

the smallest diameter of a screw thread



the distance between adjacent thread forms measured parallel to the thread axis



M12X1.75: A thread having a nominal major diameter of 12 mm and a pitch of 1.75 mm

Basic profile for metric M and MJ threads.

d = major diameter

d_r = minor diameter

d_p = pitch diameter

p = pitch

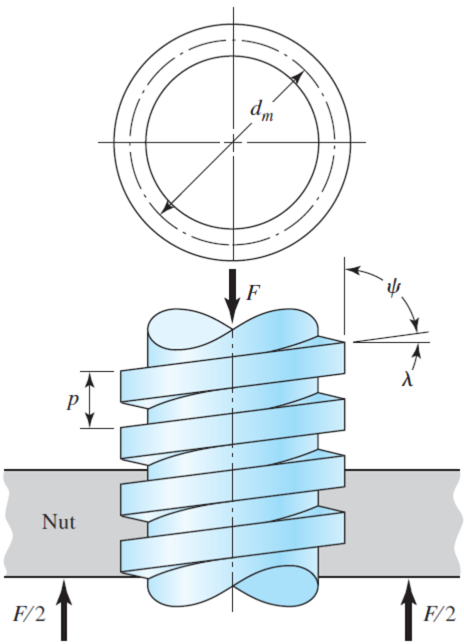
$H = \frac{\sqrt{3}}{2}p$

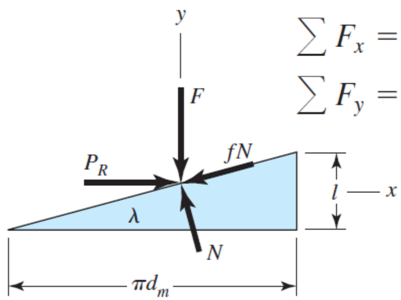
AncoraSIR.com

The Mechanics of Power Screws

A power screw is a device used in machinery to change angular motion into linear motion to transmit power.

- A square-threaded power screw with single thread having a mean diameter d_m, a pitch p, a lead angle λ, and a helix angle ψ is loaded by the axial compressive force F.
  - We wish to find an expression for the torque required to raise this load, and another expression for the torque required to lower the load.



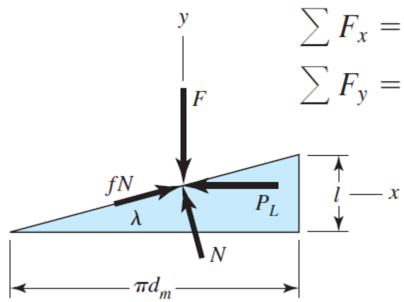


(a)

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

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

Raising the load

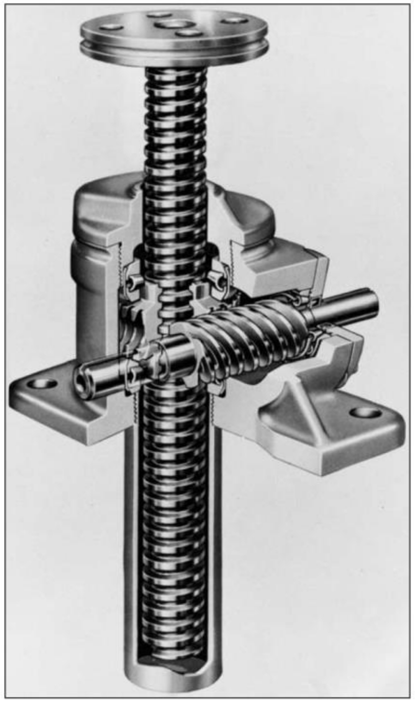


(b)

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

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

Lowering the load



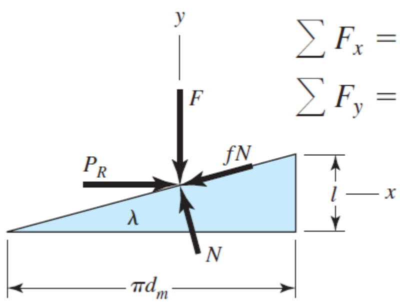
The Joyce worm-gear screw jack.

AncoraSIR.com

SUSTech Southern University of Science and Technology

Self-locking

When the lead is small or the friction is high, that the load will lock itself by holding the screw from spin without any external effort.



(a)

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

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

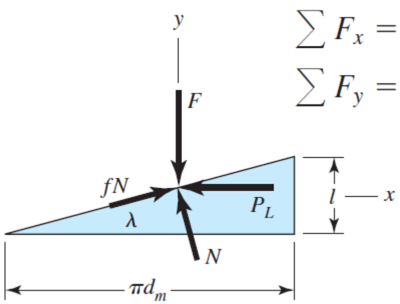
Raising the load

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

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

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

Torque



(b)

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

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

Lowering the load

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

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

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

Self-locking happens with the Torque is positive

$$\pi f d_m > l \qquad l/\pi d_m = \tan\lambda$$

$$f > \tan\lambda$$

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

Efficiency =
No Friction Torque / With Friction Torque

AncoraSIR.com

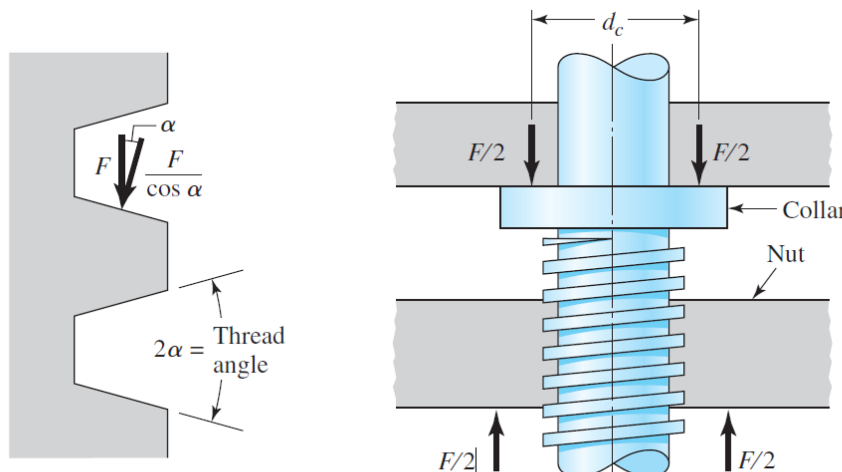
Acme Threads

the Acme thread is not as efficient as the square thread, but easier to machine

- the normal thread load is inclined to the axis because of the thread angle 2α and the lead angle λ.
  - Since lead angles are small, this inclination can be neglected
  - and only the effect of the thread angle considered
  - The effect of the angle α is to increase the frictional force by the wedging action of the threads
- For raising the load, or for tightening a screw or bolt, this yields

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

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



(a) (b)

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

AncoraSIR.com

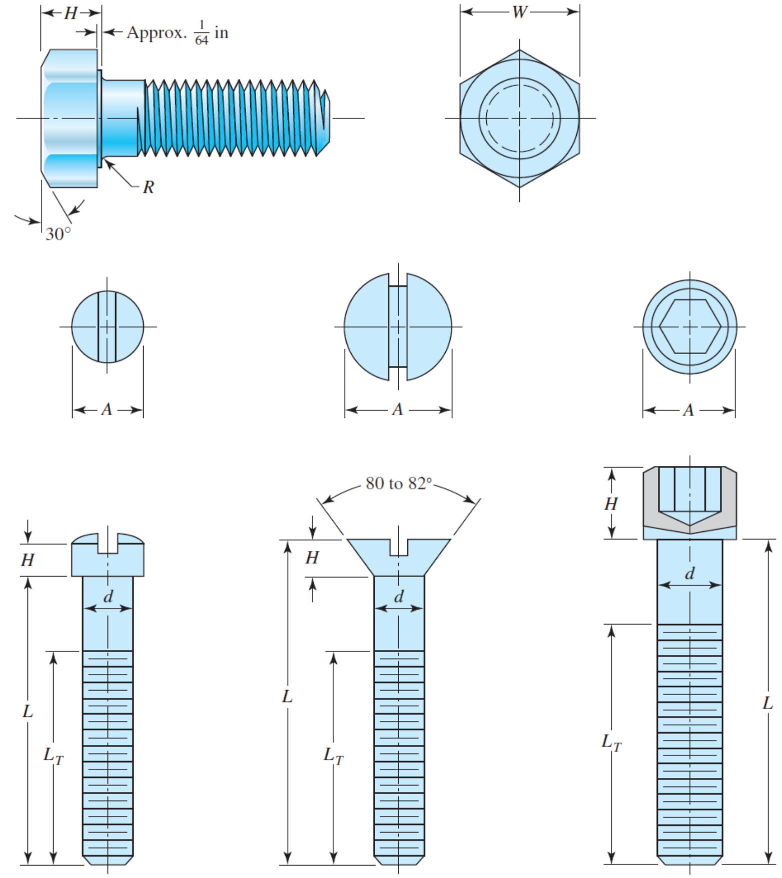
Threaded Fasteners

The ideal bolt length is one in which only one or two threads project from the nut after it is tightened

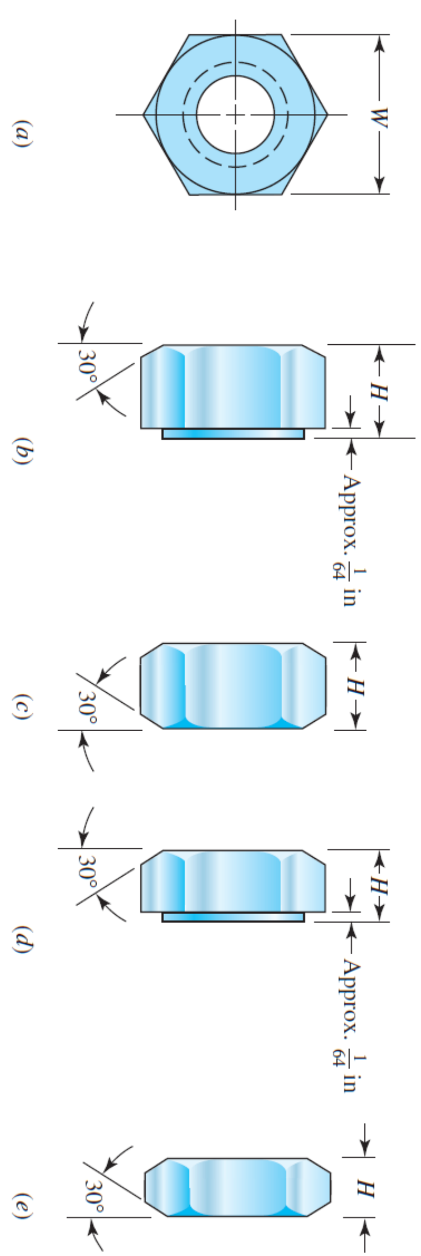
$$L_T = \begin{cases} 2d + \frac{1}{4}\text{ in} & L \le 6\text{ in} \\ 2d + \frac{1}{2}\text{ in} & L > 6\text{ in} \end{cases}$$

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

- Bolt holes may have burrs or sharp edges after drilling.
    - These could bite into the fillet and increase stress concentration.
    - Therefore, **washers** must always be used under the bolt head to prevent this.
    - They should be of hardened steel and loaded onto the bolt so that the rounded edge of the stamped hole faces the washer face of the bolt.
    - **Sometimes it is necessary to use washers under the nut too.**
- The material of the nut must be selected carefully to match that of the bolt.
    - During tightening, the first thread of the nut tends to take the entire load; but yielding occurs, with some strengthening due to the cold work that takes place, and the load is eventually divided over about three nut threads.
    - **For this reason you should never reuse nuts; in fact, it can be dangerous to do so.**

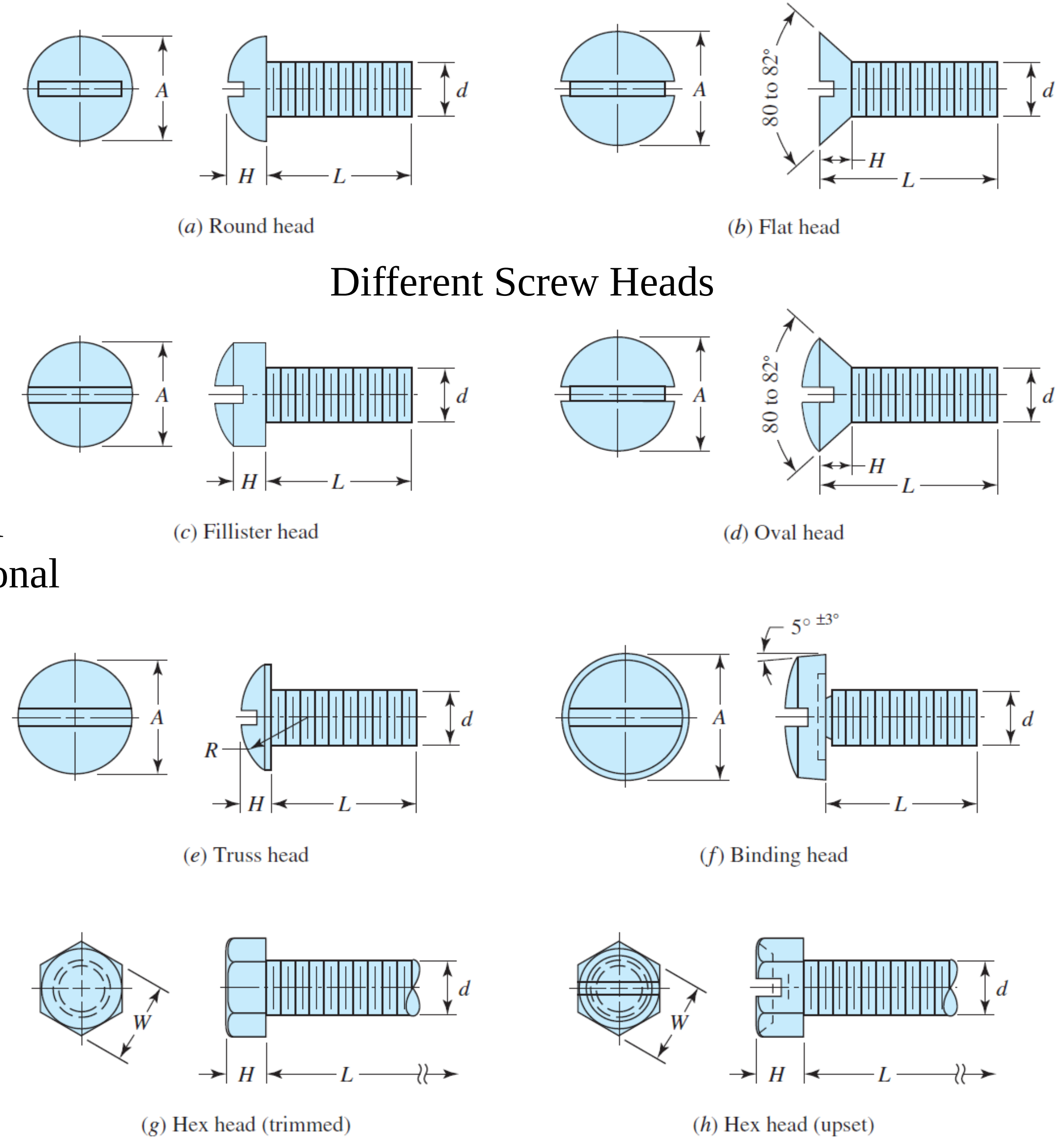


AncoraSIR.com



Typical Hexagonal Nuts

Different Screw Heads

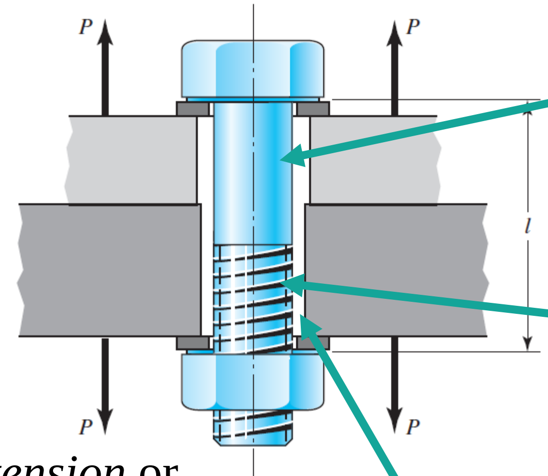


AncoraSIR.com

Joints—Fastener Stiffness

stiffness of the ***fastener*** in the clamped zone

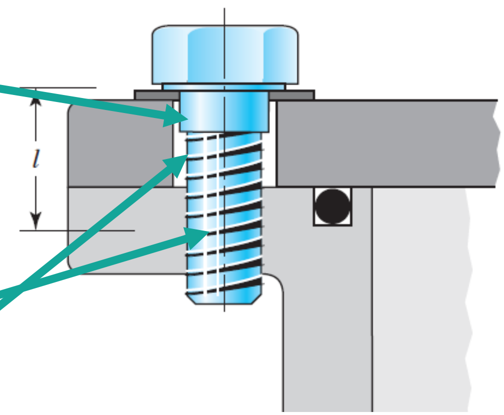
- When to use bolted joints with hardened-steel washers
  - When a connection is desired that can be disassembled without destructive methods and that is strong enough to resist external tensile loads, moment loads, and shear loads, or a combination of these
  - Such a joint can also be dangerous unless it is properly designed and assembled by a *trained* mechanic.



No thread section

$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2}$$

Threaded section



pretension or *bolt preload*

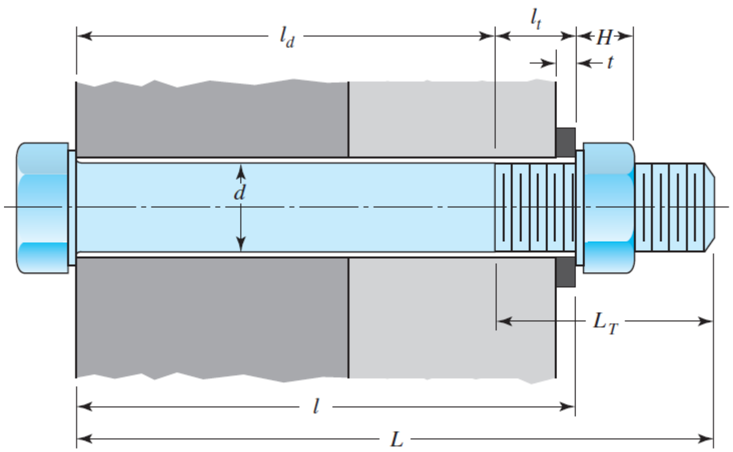
the clearance space provided by the bolt holes

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

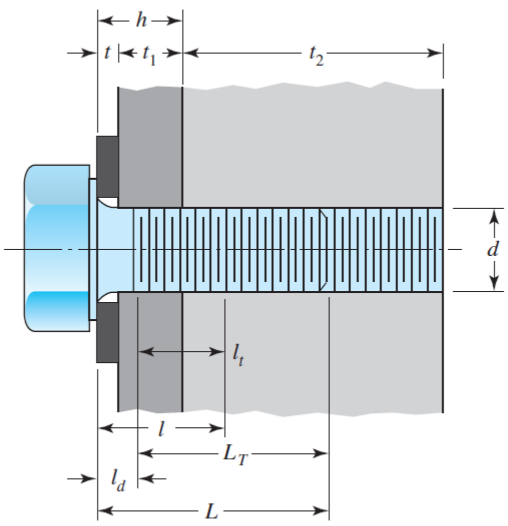
A_t = tensile-stress area (Tables 8–1, 8–2)
l_t = length of threaded portion of grip
A_d = major-diameter area of fastener
l_d = length of unthreaded portion in grip

AncoraSIR.com

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

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

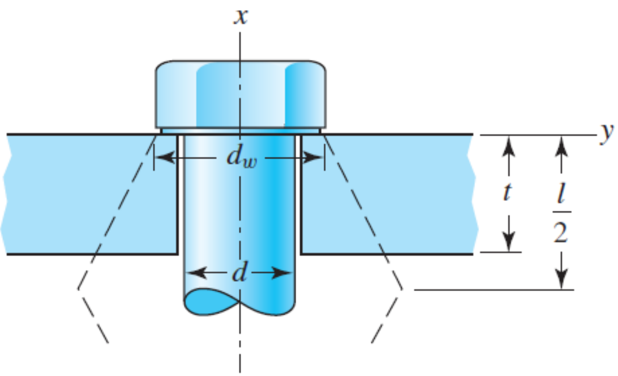
Fastener stiffness:
$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}$$

Joints—Member Stiffness

the stiffnesses of the ***members*** in the clamped zone

- There may be more than two members included in the grip of the fastener.

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



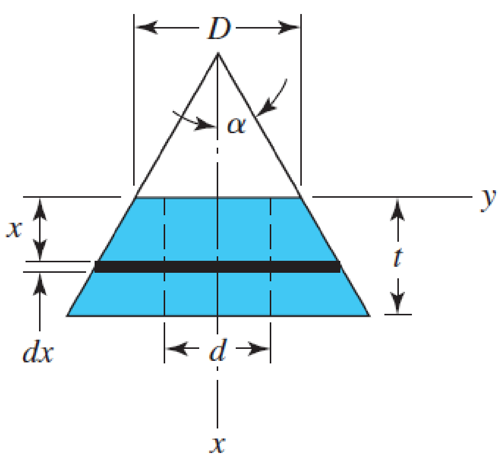
the contraction of an element of the cone of thickness d_x subjected to a compressive force P

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

The area of the element

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

$$= \pi\left(x \tan\alpha + \frac{D+d}{2}\right)\left(x \tan\alpha + \frac{D-d}{2}\right)$$



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

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

stiffness of this frustum

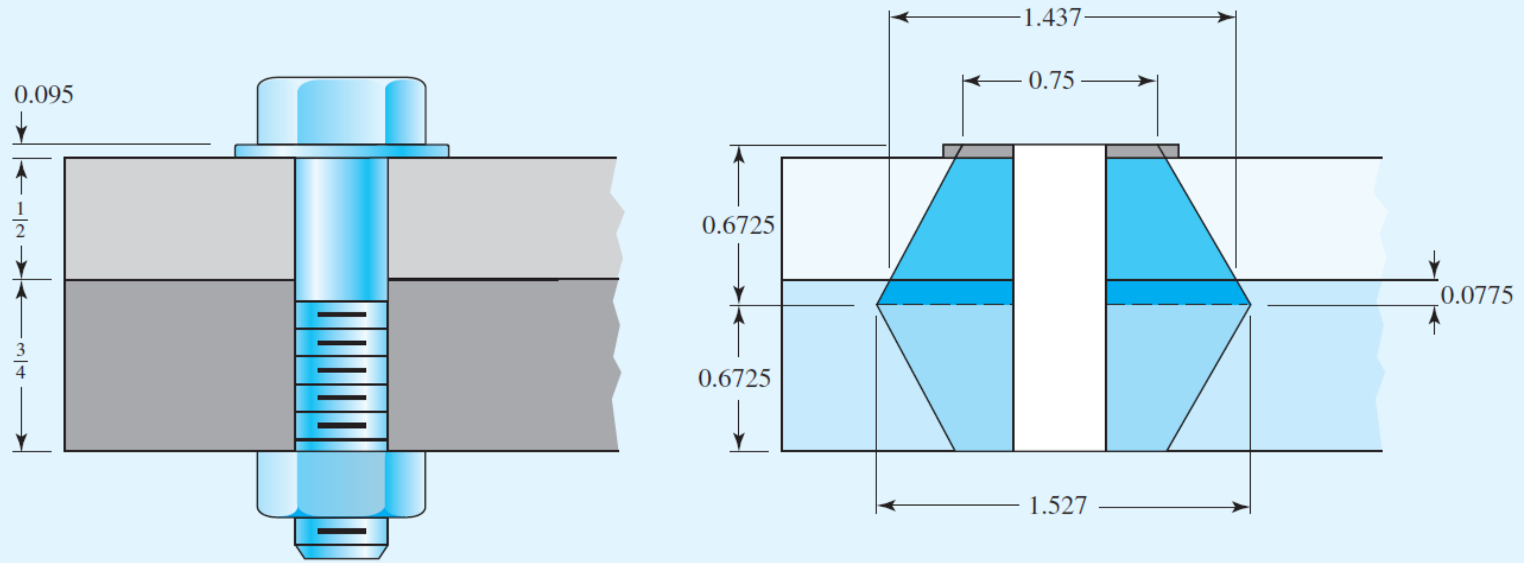
$$k = \frac{P}{\delta} = \frac{\pi E d \tan\alpha}{\ln\dfrac{(2t \tan\alpha + D - d)(D + d)}{(2t \tan\alpha + D + d)(D - d)}}$$

Example

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.

From Table A–32, the thickness of a standard $\frac{1}{2}$ N plain washer is 0.095 in.

the frusta extend halfway into the joint the distance $\frac{1}{2}(0.5 + 0.75 + 0.095) = 0.6725$ in



(*a*) (*b*)

the spring rate of the steel

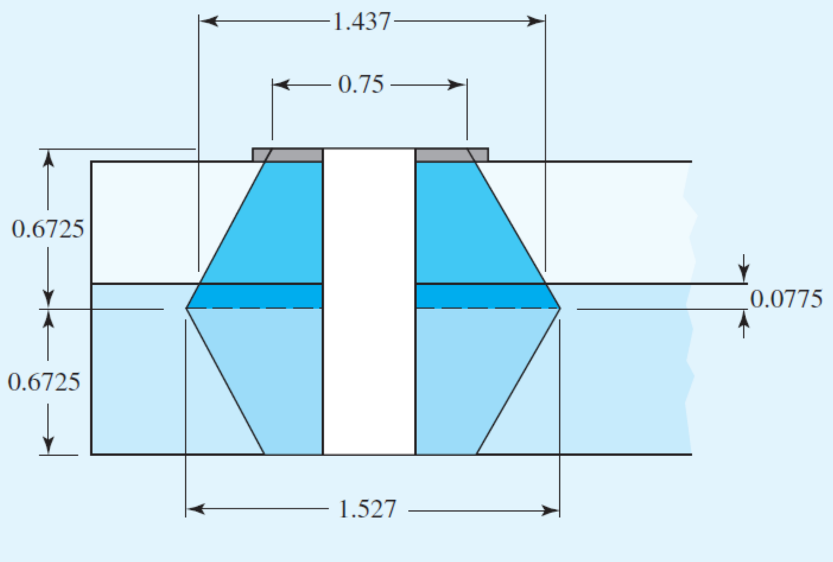
$$k_1 = \frac{0.5774\pi(30)(10^6)0.5}{\ln\left\{\frac{[1.155(0.595) + 0.75 - 0.5](0.75 + 0.5)}{[1.155(0.595) + 0.75 + 0.5](0.75 - 0.5)}\right\}} = 30.80(10^6)\ \text{lbf/in}$$

for the upper cast iron frustum

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

For the lower cast-iron frustum

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



(*b*)

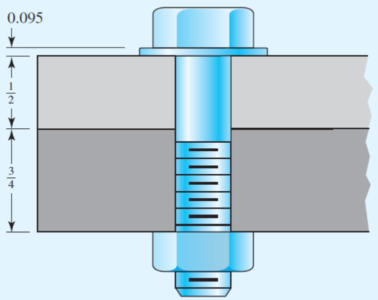
$$k = \frac{P}{\delta} = \frac{\pi E d \tan\alpha}{\ln\frac{(2t\tan\alpha + D - d)(D + d)}{(2t\tan\alpha + D + d)(D - d)}}$$

$$\frac{1}{k_m} = \frac{1}{30.80(10^6)} + \frac{1}{285.5(10^6)} + \frac{1}{14.15(10^6)}$$

$$k_m = 9.378\ (10^6)\ \text{lbf/in.}$$

SUSTech
Southern University of Science and Technology

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.
(*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*)

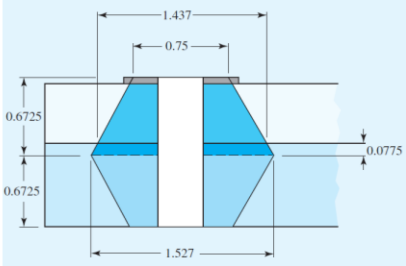
(*b*) If the entire joint is steel, Eq. (8–22) with $l = 2(0.6725) = 1.345$ in gives

$$k_m = \frac{0.5774\pi(30.0)(10^6)0.5}{2\ln\left\{5\left[\dfrac{0.5774(1.345) + 0.5(0.5)}{0.5774(1.345) + 2.5(0.5)}\right]\right\}} = 14.64(10^6)\ \text{lbf/in.}$$

(*c*) From Table 8–8, $A = 0.787\,15$, $B = 0.628\,73$. Equation (8–23) gives

$$k_m = 30(10^6)(0.5)(0.787\,15)\exp[0.628\,73(0.5)/1.345] = 14.92(10^6)\ \text{lbf/in}$$

For this case, the difference between the results for Eqs. (8–22) and (8–23) is less than 2 percent.



(*b*)

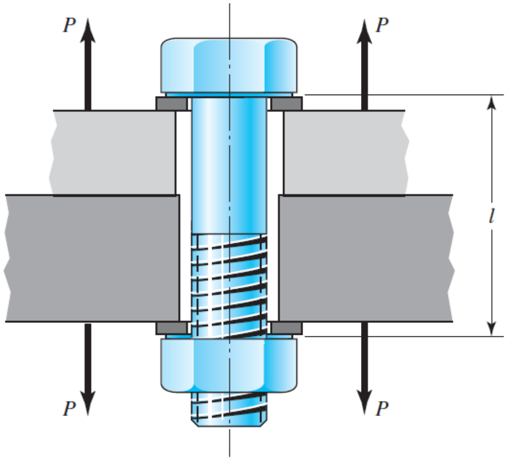
(*d*) Following the procedure of Table 8–7, the threaded length of a 0.5-in bolt is $L_T = 2(0.5) + 0.25 = 1.25$ in. The length of the unthreaded portion is $l_d = 1.5 - 1.25 = 0.25$ in. The length of the unthreaded portion in grip is $l_t = 1.345 - 0.25 = 1.095$ in. The major diameter area is $A_d = (\pi/4)(0.5^2) = 0.196\,3$ in^2. From Table 8–2, the tensile-stress area is $A_t = 0.159\,9$ in^2. From Eq. (8–17)

$$k_b = \frac{0.196\,3(0.159\,9)30(10^6)}{0.196\,3(1.095) + 0.159\,9(0.25)} = 3.69(10^6)\ \text{lbf/in}$$

SUSTech
Southern University of Science and Technology

Tension Joints—The External Load

Consider what happens when an external tensile load P is applied to a bolted connection.



F_i = preload

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

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

C = fraction of external load P carried by bolt

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

N = Number of bolts in the joint

If N bolts equally share the total external load, then

$$P = P_{\text{total}}/N \qquad (a)$$

The load P is tension, and it causes the connection to stretch, or elongate, through some distance δ. We can relate this elongation to the stiffnesses by recalling that k is the force divided by the deflection. Thus

$$\delta = \frac{P_b}{k_b} \quad \text{and} \quad \delta = \frac{P_m}{k_m} \qquad (b)$$

or

$$P_m = P_b \frac{k_m}{k_b} \qquad (c)$$

Since $P = P_b + P_m$, we have

$$P_b = \frac{k_b P}{k_b + k_m} = CP \qquad (d)$$

and

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

where

$$C = \frac{k_b}{k_b + k_m} \qquad (f)$$

is called the *stiffness constant of the joint*. The resultant bolt load is

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

and the resultant load on the connected members is

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

Relating Bolt Torque to Bolt Tension

Ensure that the preload is actually developed when the parts are assembled

- How to ensure a desired preload F_i has been attained?
  - The bolt is tightened until a desired elongation δ is reached.
  - Usually difficult, even impractical to measure bolt elongation, i.e. blind hole.
  - Measure/Estimate the wrench torque become more practical

- Tools/Methods to ensure wrench torque
  - **Torque wrench**: a built-in dial that indicates the proper torque.
  - **Impact wrench**: the air pressure is adjusted so that the wrench stalls when the proper torque is obtained, or in some wrenches, the air automatically shuts off at the desired torque.
  - **Turn-of-the-nut**: reaching a snug-tight condition, manually turning with additional turns to develop useful tension in the bolt. Compute the fractional number of the turns for required preload. About a minimum of 180˚ from the snug-tight condition.

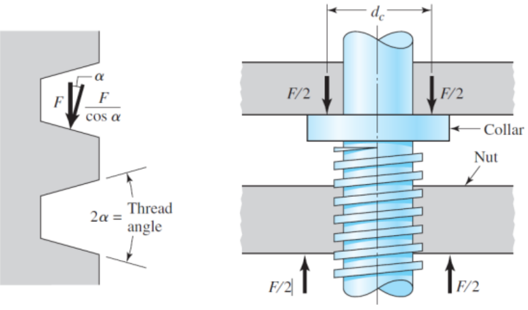
Bolt Torque Calculation/Estimation

- Total torque estimation

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

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

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



(a) (b)

$$\tan\lambda = l/\pi d_m$$

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

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

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

$$T = K F_i d$$

$K \approx 0.2$ for $f = f_c = 0.15$ no matter what size bolts are employed and no matter whether the threads are coarse or fine.

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

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

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

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

Distribution of Preload Fi for Lubricated/Unlubricated Bolts Torqued to 90 Nm

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

AncoraSIR.com

SUSTech Southern University of Science and Technology

Example

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.

From Table 8–2, $A_t = 0.373$ in^2.

- Preload stress: $$\sigma_i = \frac{F_i}{A_t} = \frac{25}{0.373} = 67.02 \text{ kpsi}$$
- Stiffness constant: $$C = \frac{k_b}{k_b + k_m} = \frac{6.5}{6.5 + 13.8} = 0.320$$
- Stress under the service load: $$\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t} = C\frac{P}{A_t} + \sigma_i = 72.17 \text{ kpsi}$$

From Table 8–9, the SAE minimum proof strength of the bolt is $S_p = 85$ kpsi. The preload and service load stresses are respectively 21 and 15 percent less than the proof strength.

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.
(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

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

(c) The minor diameter can be determined from the minor area in Table 8–2. Thus $d_r = \sqrt{4A_r/\pi} = \sqrt{4(0.351)/\pi} = 0.6685$ in. Thus, the mean diameter is $d_m = (0.75 + 0.6685)/2 = 0.7093$ in. The lead angle is

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

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).

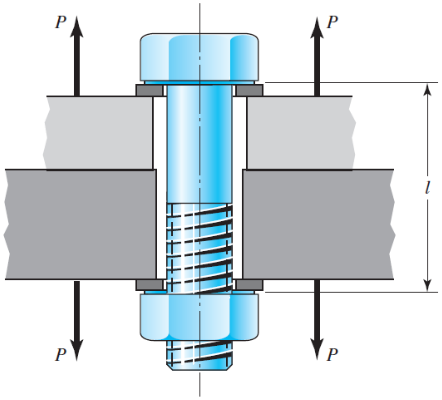
Statically Loaded Tension Joint with Preload

Safety Design

- The tensile stress in the bolt $\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t}$
- Thus, the yielding factor of safety guarding against the static stress exceeding the proof strength

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

- It is also essential for a safe joint that the external load be smaller than that needed to cause the joint to separate.
  - If so, then the entire external load will be imposed on the bolt.



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

$$\downarrow \quad F_m = 0$$

$$(1 - C)P_0 - F_i = 0 \longrightarrow n_0 = \frac{F_i}{P(1 - C)}$$

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

$$F_i = \begin{cases} 0.75F_p & \text{for nonpermanent connections, reused fasteners} \\ 0.90F_p & \text{for permanent connections} \end{cases}$$

AncoraSIR.com

SUSTech Southern University of Science and Technology

Example

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.

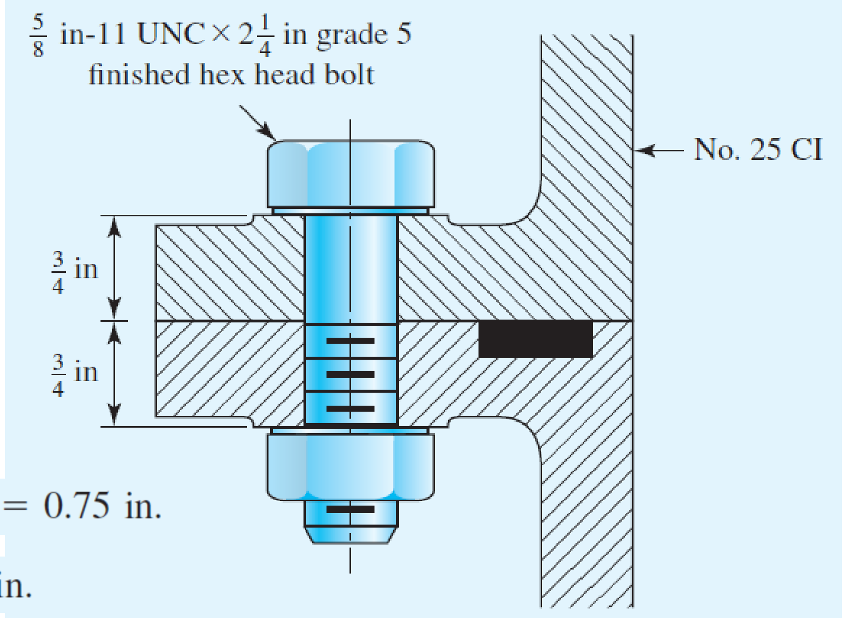
(*b*) Find the number of bolts required for a load factor of 2 where the bolts may be reused when the joint is taken apart.

(*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.

- The grip is $l = 1.5$ in.
- The nut thickness is $\frac{35}{64}$ in (Table A-31)
- Bolt length $L = \frac{35}{64} + 1.50 + \frac{2}{11} = 2.229$ in

Two extra threads beyond the nut

- The next fraction size bolt is $L = 2\frac{1}{4}$ in
- Thread length $L_T = 2(0.625) + 0.25 = 1.50$ in.
- Unthreaded length in the grip $l_d = 2.25 - 1.50 = 0.75$ in.
- Threaded length in the grip $l_t = l - l_d = 0.75$ in.



AncoraSIR.com

The bolt stiffness

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.3068(0.226)(30)}{0.3068(0.75) + 0.226(0.75)}$$
$$= 5.21 \text{ Mlbf/in}$$

From Table A–24, for no. 25 cast iron we will use $E = 14$ Mpsi. The stiffness of the members, from Eq. (8–22), is

$$k_m = \frac{0.5774\pi Ed}{2\ln\left(5\dfrac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)} = \frac{0.5774\pi(14)(0.625)}{2\ln\left[5\dfrac{0.5774(1.5) + 0.5(0.625)}{0.5774(1.5) + 2.5(0.625)}\right]}$$
$$= 8.95 \text{ Mlbf/in}$$

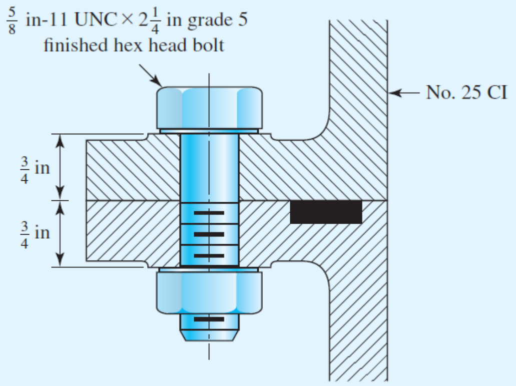
If you are using Eq. (8–23), from Table 8–8, $A = 0.778\ 71$ and $B = 0.616\ 16$, and

$$k_m = EdA\exp(Bd/l)$$
$$= 14(0.625)(0.778\ 71)\exp[0.616\ 16(0.625)/1.5]$$
$$= 8.81 \text{ Mlbf/in}$$

which is only 1.6 percent lower than the previous result.

From the first calculation for k_m, the stiffness constant C is

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



AncoraSIR.com

(*b*) From Table 8–9, $S_p = 85$ kpsi. Then, using Eqs. (8–31) and (8–32), we find the recommended preload to be

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

For N bolts, Eq. (8–29) can be written

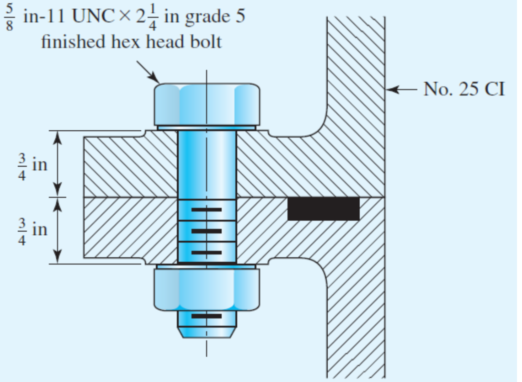
$$n_L = \frac{S_pA_t - F_i}{C(P_{\text{total}}/N)}$$

or

$$N = \frac{Cn_LP_{\text{total}}}{S_pA_t - F_i} = \frac{0.368(2)(36)}{85(0.226) - 14.4} = 5.52$$

Six bolts should be used to provide the specified load factor.

(*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.



(*c*) With six bolts, the load factor actually realized is

$$n_L = \frac{85(0.226) - 14.4}{0.368(36/6)} = 2.18$$

From Eq. (8–28), the yielding factor of safety is

$$n_p = \frac{S_pA_t}{C(P_{\text{total}}/N) + F_i} = \frac{85(0.226)}{0.368(36/6) + 14.4} = 1.16$$

From Eq. (8–30), the load factor guarding against joint separation is

$$n_0 = \frac{F_i}{(P_{\text{total}}/N)(1 - C)} = \frac{14.4}{(36/6)(1 - 0.368)} = 3.80$$

Next class

- **Lecture for Group 1+2**: Joint Components
- Wednesday 1400-1600, Nov 20
- Room 206, 2 Lychee Park

Thank you!

Song Chaoyang (songcy@sustech.edu.cn)

- Xiao Xiaochuan (xiaoxc@sustech.edu.cn)
- Yu Chengming (11930324@mail.sustech.edu.cn)
- Zhu Wenpei (11930368@mail.sustech.edu.cn)
- Guo Ning (11930729@mail.sustech.edu.cn)

AncoraSIR.com

SUSTech
Southern University of Science and Technology