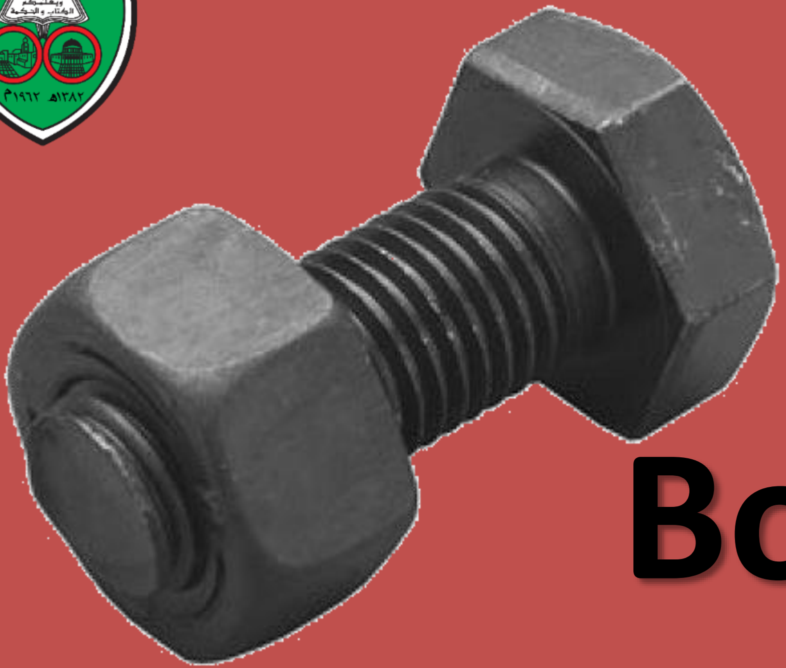
University of Jordan – School of Engineering – Mechanical Engineering Department

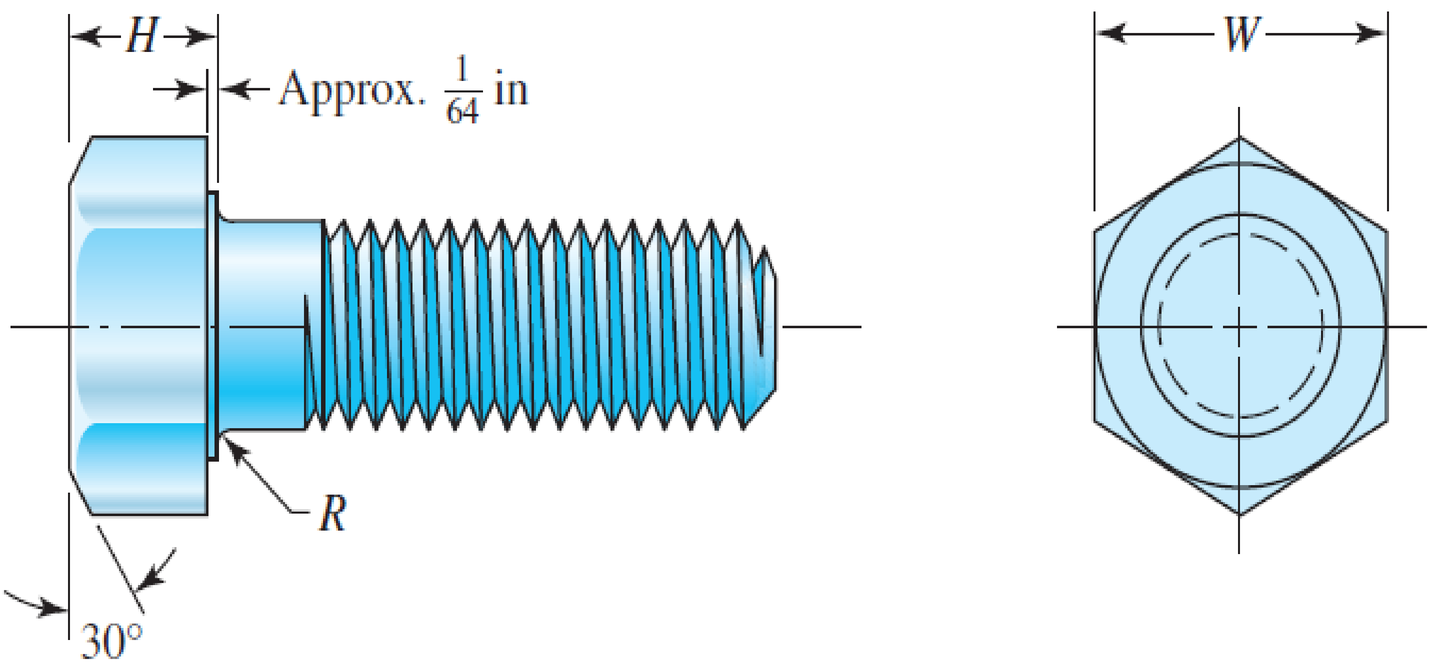


Bolt Strength

Spring 2022-2023 m.zgoul@ju.edu.jo Dr Moudar Zgoul

Threaded Fasteners

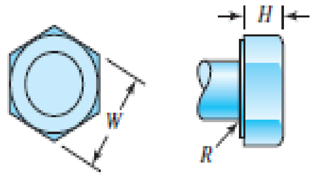
Points of stress concentration are at the fillet, at the start of the threads (runout), and at the thread-root fillet in the plane of the nut when it is present.



Threaded Fasteners

Table A-29

Dimensions of Square and Hexagonal Bolts



Nominal Size, in	Square W	Square H	Regular Hexagonal W	Regular Hexagonal H	Regular Hexagonal R_{min}	Heavy Hexagonal W	Heavy Hexagonal H	Heavy Hexagonal R_{min}	Structural Hexagonal W	Structural Hexagonal H	Structural Hexagonal R_{min}
$\frac{1}{4}$	$\frac{3}{8}$	$\frac{11}{64}$	$\frac{7}{16}$	$\frac{11}{64}$	0.01						
$\frac{5}{16}$	$\frac{1}{2}$	$\frac{13}{64}$	$\frac{1}{2}$	$\frac{7}{32}$	0.01						
$\frac{3}{8}$	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{9}{16}$	$\frac{1}{4}$	0.01						
$\frac{7}{16}$	$\frac{5}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{19}{64}$	0.01						
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{21}{64}$	$\frac{3}{4}$	$\frac{11}{32}$	0.01	$\frac{7}{8}$	$\frac{11}{32}$	0.01	$\frac{7}{8}$	$\frac{5}{16}$	0.009
Nominal Size, mm											
M5	8	3.58	8	3.58	0.2						
M6			10	4.38	0.3						
M8			13	5.68	0.4						
M10			16	6.85	0.4						

Threaded Fasteners

The diameter of the washer face is the same as the width across the flats of the hexagon. The thread length of inch-series bolts, where *d* is the nominal diameter, is

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

and for metric bolts is

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

Threaded Fasteners

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

Threaded Fasteners

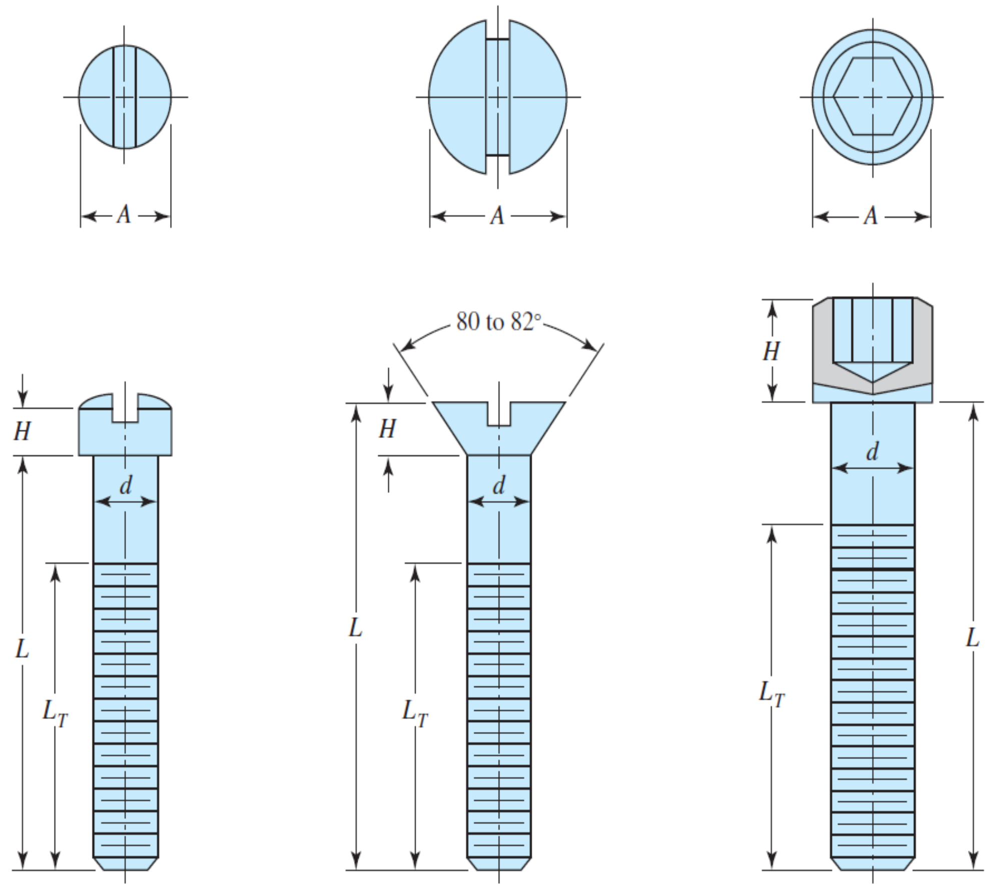
- The purpose of a bolt is to clamp two or more parts together.
- The clamping load stretches or elongates the bolt; the load is obtained by twisting the nut until the bolt has elongated almost to the elastic limit.
- If the nut does not loosen, this bolt tension remains as the preload or clamping force.
- When tightening, the mechanic should, if possible, hold the bolt head stationary and twist the nut; in this way the bolt shank will not feel the thread-friction torque.

Threaded Fasteners

The head of a hexagon-head cap screw is slightly thinner than that of a hexagonhead bolt.

Dimensions of hexagon-head cap screws are listed in Table A–30.

Hexagonhead cap screws are used in the same applications as bolts and also in applicatior in which one of the clamped members is threaded.



Threaded Fasteners

Table A–30

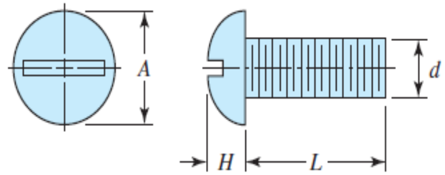
Dimensions of Hexagonal Cap Screws and Heavy Hexagonal Screws (W = Width across Flats; H = Height of Head; See Figure in Table A–29)

Nominal Size, in	Minimum Fillet Radius	Type of Screw Cap W	Type of Screw Heavy W	Height H
$\frac{1}{4}$	0.015	$\frac{7}{16}$		$\frac{5}{32}$
$\frac{5}{16}$	0.015	$\frac{1}{2}$		$\frac{13}{64}$
$\frac{3}{8}$	0.015	$\frac{9}{16}$		$\frac{15}{64}$
$\frac{7}{16}$	0.015	$\frac{5}{8}$		$\frac{9}{32}$
$\frac{1}{2}$	0.015	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{5}{16}$

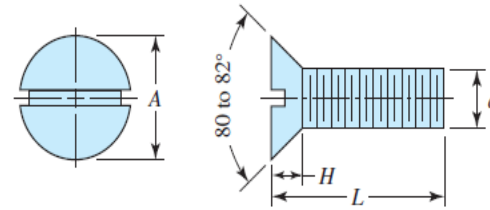
Nominal Size, mm				
M5	0.2	8		3.65
M6	0.3	10		4.15
M8	0.4	13		5.50
M10	0.4	16		6.63
M12	0.6	18	21	7.76
M14	0.6	21	24	9.09

common cap-screw head styles

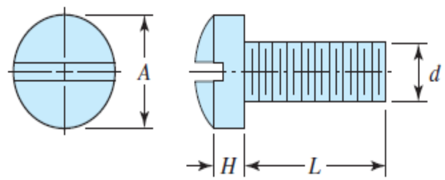
Threaded Fasteners



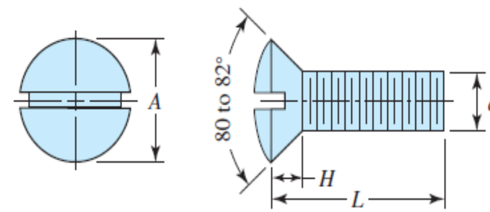
(*a*) Round head



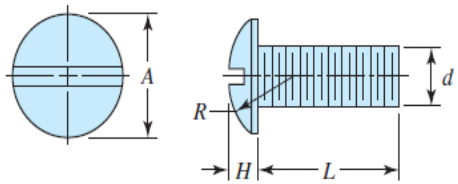
(*b*) Flat head



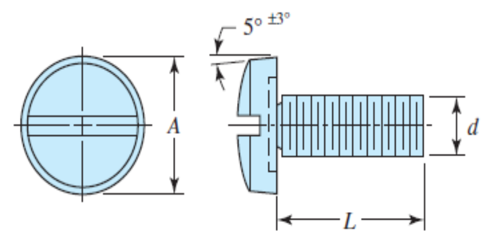
(*c*) Fillister head



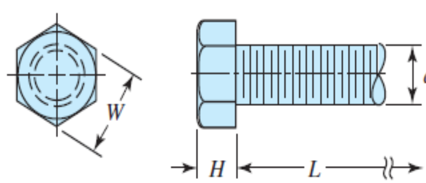
(*d*) Oval head



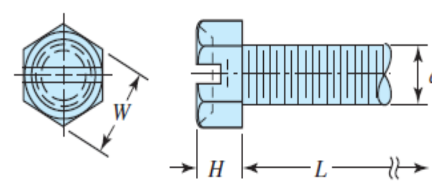
(*e*) Truss head



(*f*) Binding head

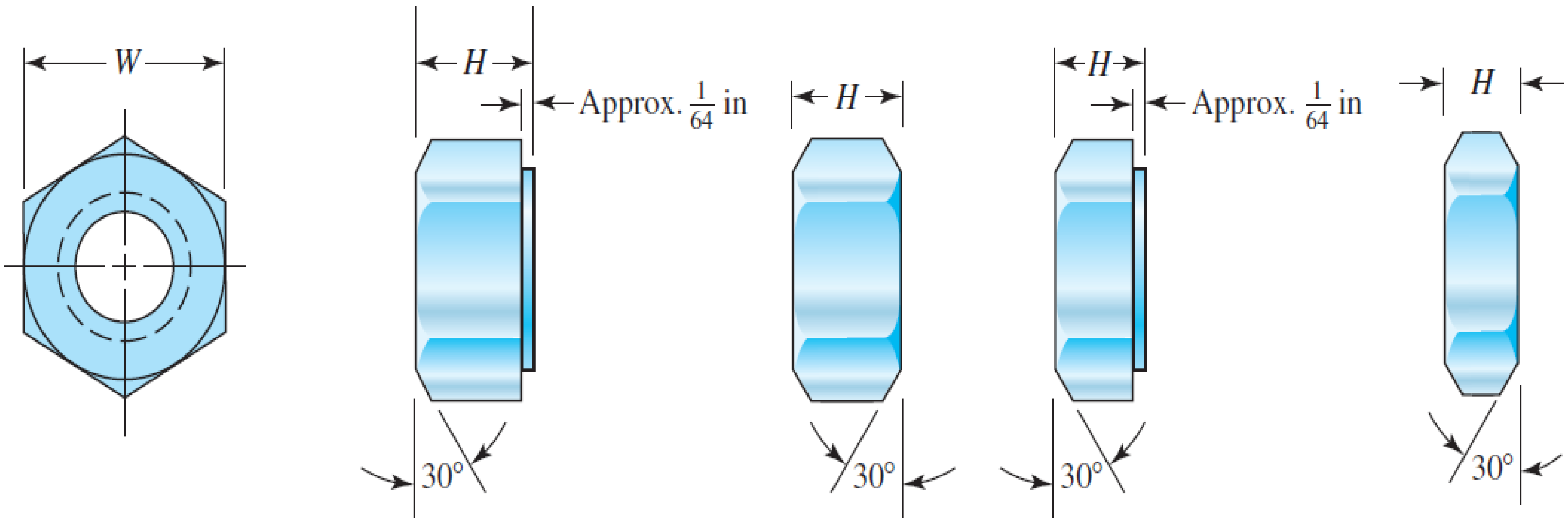


(*g*) Hex head (trimmed)



(*h*) Hex head (upset)

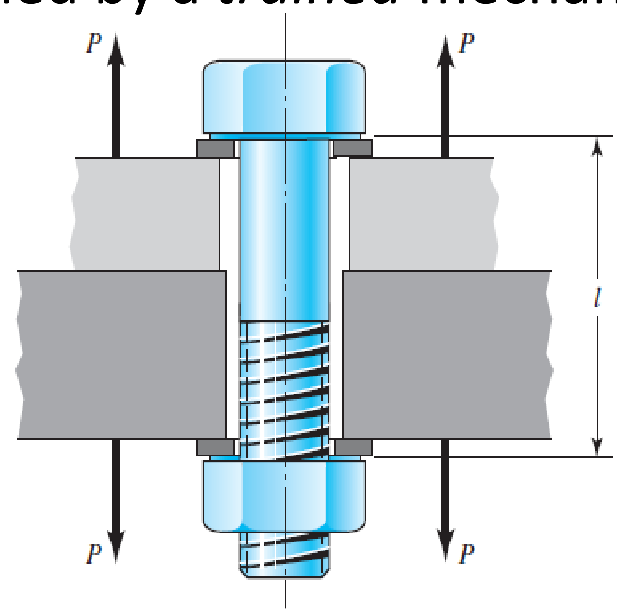
Threaded Fasteners



Joints—Fastener Stiffness

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, then the simple bolted joint using hardened-steel washers is a good solution.

Such a joint can also be dangerous unless it is properly designed and assembled by a *trained* mechanic.



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

Joints—Fastener Stiffness

The purpose of the bolt is to clamp the two, or more, parts together. Twisting the nut stretches the bolt to produce the clamping force.

This clamping force is called the *pretension* or *bolt preload*. It exists in the connection after the nut has been properly tightened no matter whether the external tensile load P is exerted or not.

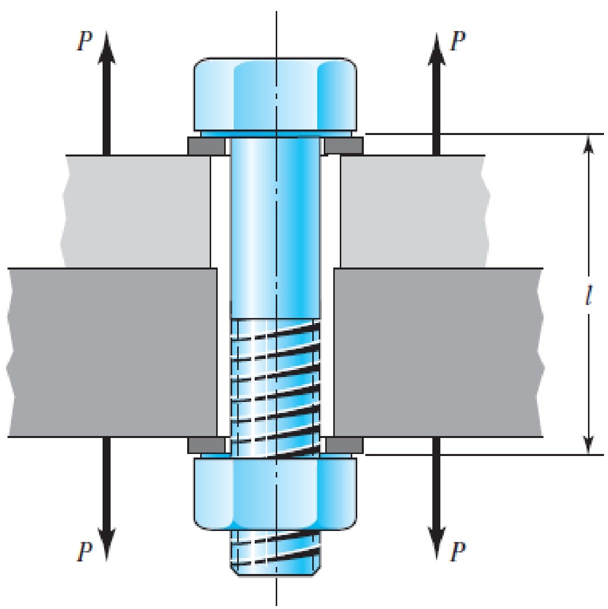
Of course, since the members are being clamped together, the clamping force that produces tension in the bolt induces compression in the members.

Joints—Fastener Stiffness

The bolt is an elastic member and thus it has a spring rate.

For the bolt it is the ratio between the force applied to the member and the deflection produced by that force.

Thus, the stiffness constant of a fastener in any bolted connection can be found.



The *grip l* of a connection is the total thickness of the clamped material.

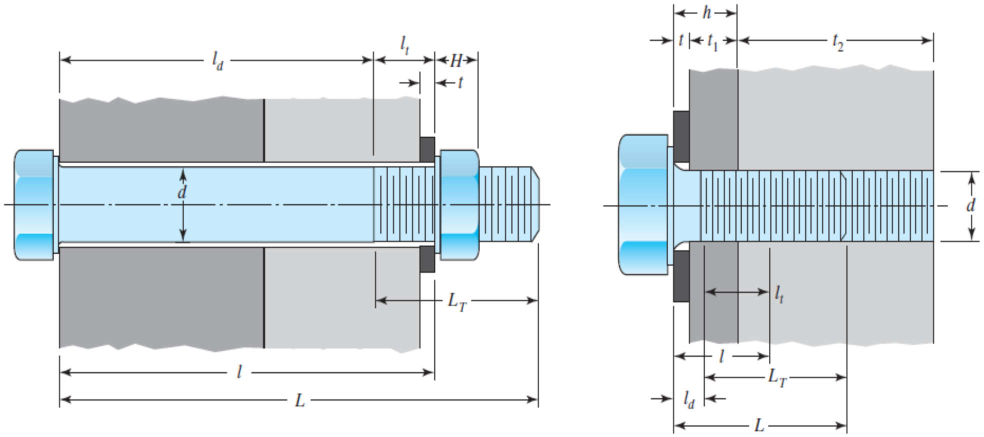
the grip is the sum of the thicknesses of both members and both washers.

The effective grip is given in Table 8–7.

Joints—Fastener Stiffness

Table 8–7

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*):

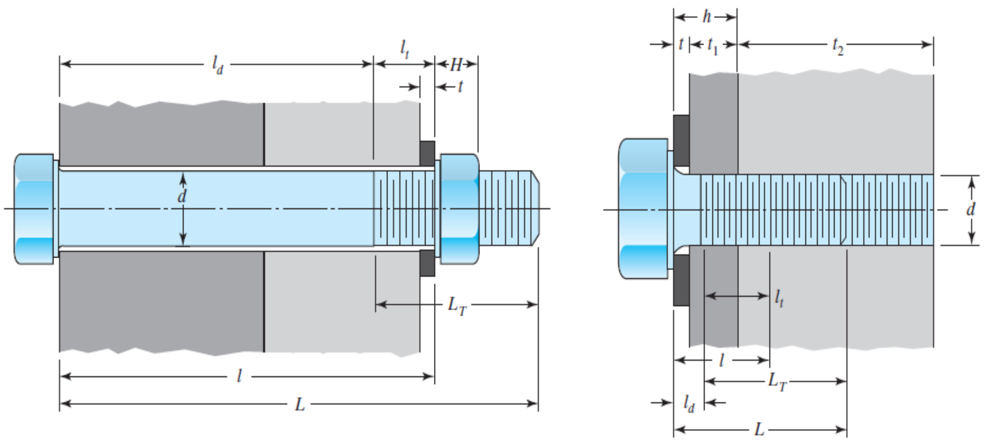
For Fig. (*a*): $L > l + H$

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

Joints—Fastener Stiffness

Table 8–7

Suggested Procedure for Finding Fastener Stiffness



Threaded length L_T: Inch series:

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

Metric series:

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

Length of unthreaded portion in grip:	$l_d = L - L_T$
Length of threaded portion in grip:	$l_t = l - l_d$
Area of unthreaded portion:	$A_d = \pi d^2/4$
Area of threaded portion:	A_t from Table 8–1 or 8–2
Fastener stiffness:	$k_b = \dfrac{A_d A_t E}{A_d l_t + A_t l_d}$

Joints—Fastener Stiffness

The stiffness of the portion of a bolt or screw within the clamped zone will generally consist of two parts, that of the unthreaded shank portion and that of the threaded portion.

Thus the stiffness constant of the bolt is equivalent to the stiffnesses of two springs in series.

$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2} \quad \text{or} \quad k = \frac{k_1 k_2}{k_1 + k_2}$$

Joints—Fastener Stiffness

For two springs in series. The spring rates of the threaded and unthreaded portions of the bolt in the clamped zone are, respectively,

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

where 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

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

where k_b is the estimated effective stiffness of the bolt or cap screw in the clamped zone.

Joints—Fastener Stiffness

For short fasteners, the unthreaded area is small: $$k_b = \frac{A_t E}{l_t}$$

For long fasteners, the threaded area is relatively small, and so:

$$k_b = \frac{A_d E}{l_d}$$

Joints—Member Stiffness

In order to understand what happens when an assembled connection is subjected to an external tensile loading, we need to know the stiffnesses of the members in the clamped zone.

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

All together these act like compressive springs in series, and hence the total spring rate of the members is

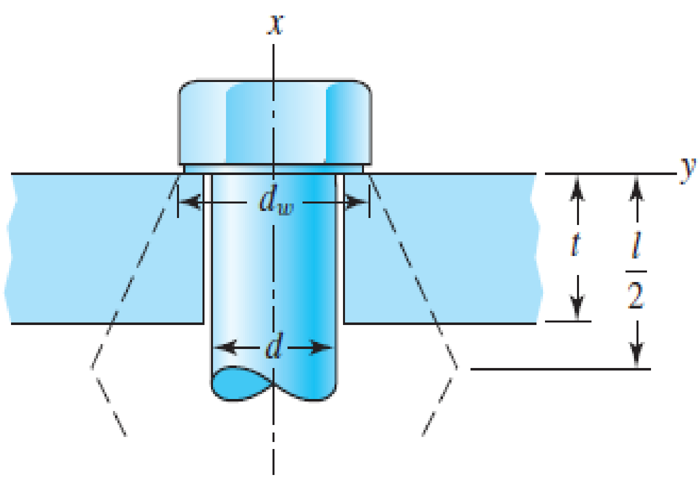
$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \cdots + \frac{1}{k_i}$$

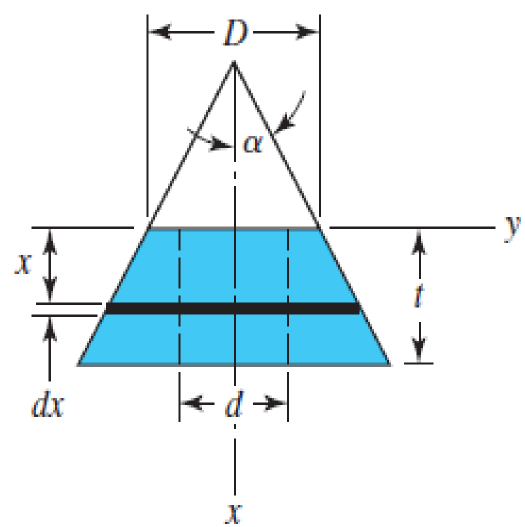
If one of the members is a soft gasket, its stiffness relative to the other members is usually so small that for all practical purposes the others can be neglected and only the gasket stiffness used.

Joints—Member Stiffness

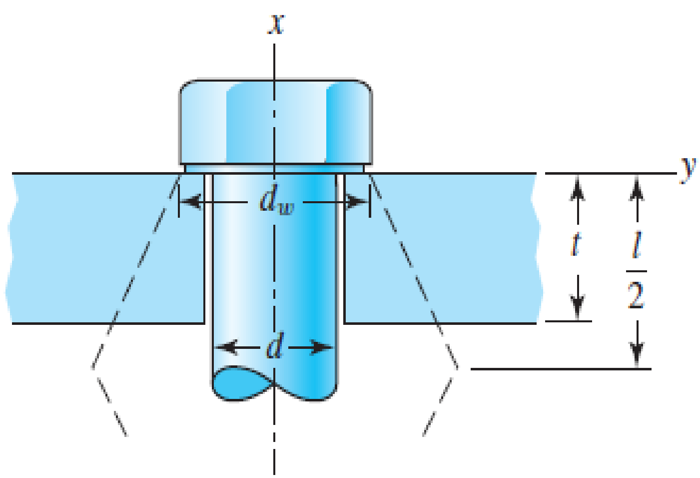
If there is no gasket, the stiffness of the members is rather difficult to obtain, except by experimentation, because the compression region spreads out between the bolt head and the nut and hence the area is not uniform.

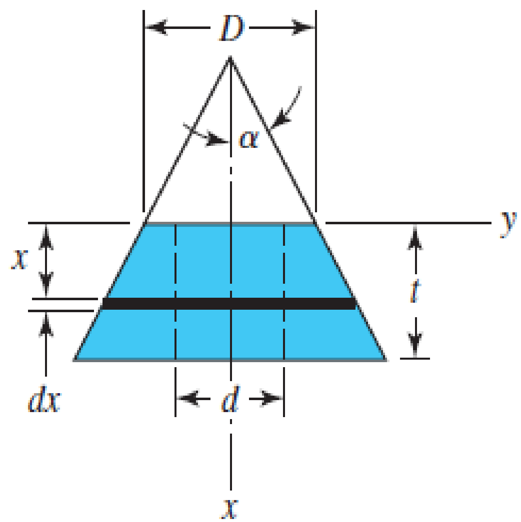
There are, however, some cases in which this area can be determined.





Joints—Member Stiffness





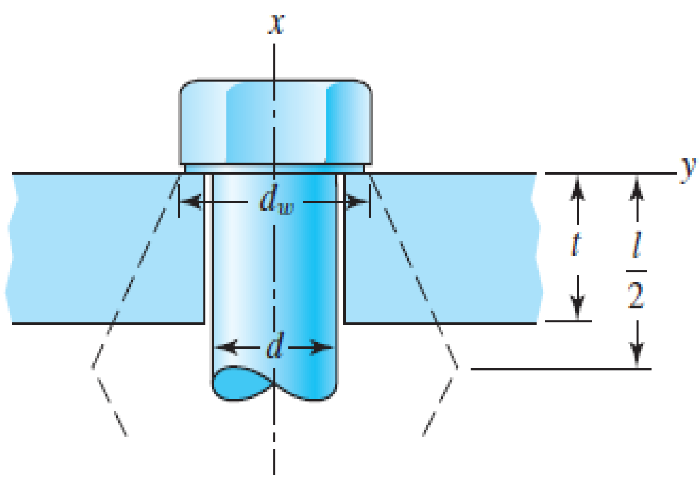
$$d\delta = \frac{P\,dx}{EA}$$

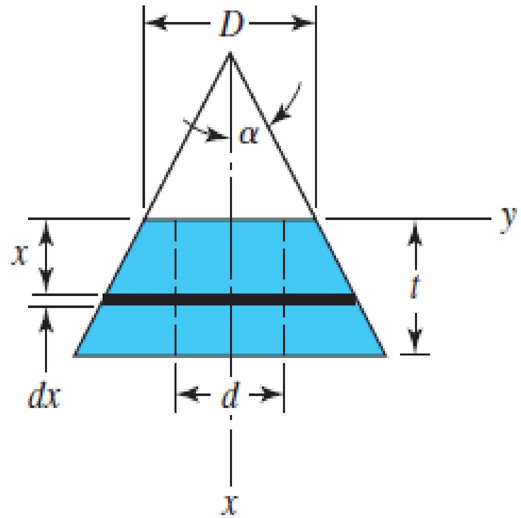
$$A = \pi(r_o^2 - r_i^2) = \pi\left[\left(x\tan\alpha + \frac{D}{2}\right)^2 - \left(\frac{d}{2}\right)^2\right]$$

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

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

Joints—Member Stiffness





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

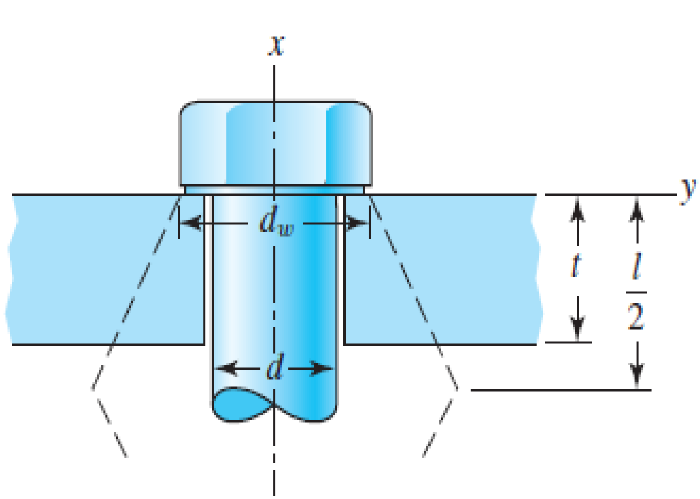
$$\delta = \frac{P}{\pi E d \tan \alpha} \ln \frac{(2t \tan \alpha + D - d)(D + d)}{(2t \tan \alpha + D + d)(D - d)}$$

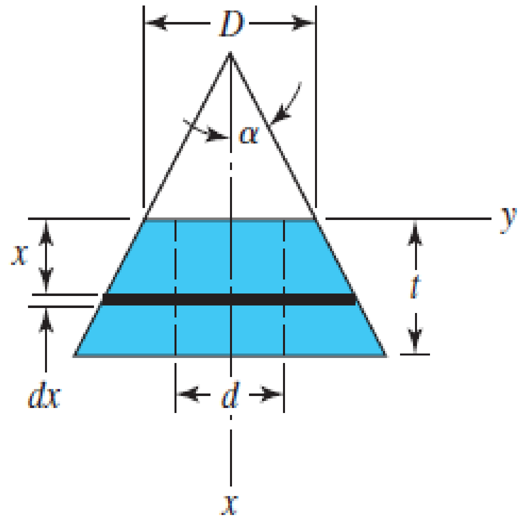
$$k = \frac{P}{\delta} = \frac{\pi E d \tan \alpha}{\ln \dfrac{(2t \tan \alpha + D - d)(D + d)}{(2t \tan \alpha + D + d)(D - d)}}$$

When α = 30°, this becomes

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

Joints—Member Stiffness





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

$k_m = k/2$. Using the grip as $l = 2t$ and d_w as the diameter of the washer face,

$$k_m = \frac{\pi E d \tan\alpha}{2\ln\dfrac{(l\tan\alpha + d_w - d)(d_w + d)}{(l\tan\alpha + d_w + d)(d_w - d)}}$$

$$\frac{k_m}{Ed} = \frac{\pi\tan\alpha}{2\ln\left[\dfrac{(l\tan\alpha + d_w - d)(d_w + d)}{(l\tan\alpha + d_w + d)(d_w - d)}\right]}$$

Joints—Member Stiffness

$$\frac{k_m}{Ed} = A \exp(Bd/l)$$

Table 8–8

Stiffness Parameters of Various Member Materials[†]

[†]*Source:* J. Wileman, M. Choudury, and I. Green, "Computation of Member Stiffness in Bolted Connections," *Trans. ASME, J. Mech. Design,* vol. 113, December 1991, pp. 432–437.

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

Bolt Strength

In the specification standards for bolts, the strength is specified by stating SAE or ASTM minimum quantities, the *minimum proof strength,* or *minimum proof load,* and the *minimum tensile strength.*

The *proof load* is the maximum load (force) that a bolt can withstand without acquiring a permanent set.

The *proof strength* is the quotient of the proof load and the tensile-stress area.

The proof strength thus corresponds roughly to the proportional limit.

Tables 8–9, 8–10, and 8–11 provide *minimum* strength specifications for steel bolts.

Bolt Strength

Bolts in fatigue axial loading fail at the fillet under the head, at the thread runout, and at the first thread engaged in the nut.

If the bolt has a standard shoulder under the head, it has a value of K_f from 2.1 to 2.3, *and* this shoulder fillet is protected from scratching or scoring by a washer.

If the thread runout has a 15° or less half-cone angle, the stress is higher at the first engaged thread in the nut.

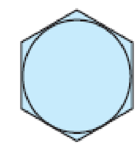
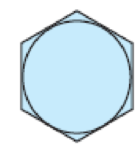
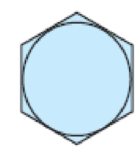
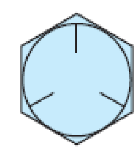
Bolts are sized by examining the loading at the plane of the washer face of the nut.

This is the weakest part of the bolt *if and only if* the conditions above are satisfied (washer protection of the shoulder fillet and thread runout $\leq 15^{o}$

Bolt Strength

Table 8–9

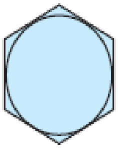
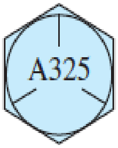
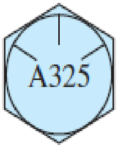
SAE Specifications for Steel Bolts

SAE Grade No.	Size Range Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
1	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low or medium carbon	
2	$\frac{1}{4}$–$\frac{3}{4}$	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		

Bolt Strength

Table 8–10

ASTM Specifications for Steel Bolts

ASTM Designation No.	Size Range, Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
A307	$\frac{1}{4}$–$1\frac{1}{2}$	33	60	36	Low carbon	
A325, type 1	$\frac{1}{2}$–1	85	120	92	Medium carbon, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
A325, type 2	$\frac{1}{2}$–1	85	120	92	Low-carbon, martensite, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		
A325, type 3	$\frac{1}{2}$–1	85	120	92	Weathering steel, Q&T	
	$1\frac{1}{8}$–$1\frac{1}{2}$	74	105	81		

Bolt Strength

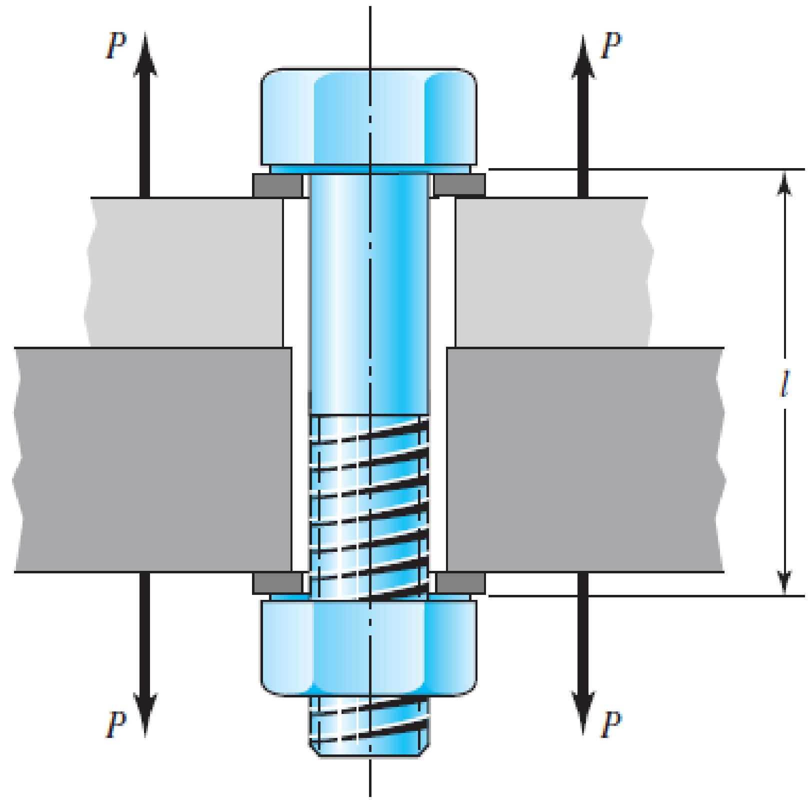
Table 8–11

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

Property Class	Size Range, Inclusive	Minimum Proof Strength,* MPa	Minimum Tensile Strength,* MPa	Minimum Yield Strength,* MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	
4.8	M1.6–M16	310	420	340	Low or medium carbon	
5.8	M5–M24	380	520	420	Low or medium carbon	
8.8	M16–M36	600	830	660	Medium carbon, Q&T	

Tension Joints—The External Load

What happens when an external tensile load *P* is applied to a bolted connection.



It is to be assumed, of course, that the clamping force, which we will call the *preload* F_i, has been correctly applied by tightening the nut *before P* is applied. The nomenclature used is:

Tension Joints—The External Load

The nomenclature used is:

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

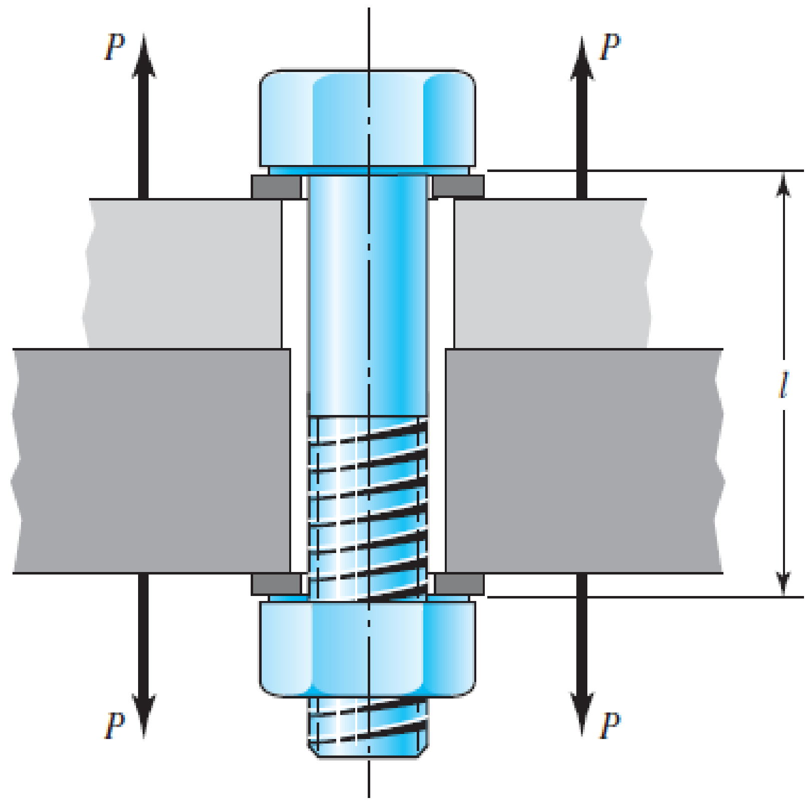
$F_b = P_b + F_i$ = resultant bolt load

$F_m = P_m - F_i$ = 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 $P = P_{total}/N$

Tension Joints—The External Load

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

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

Since $P = P_b + P_m$, we have

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

and

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

where $C = \dfrac{k_b}{k_b + k_m}$ is called the *stiffness constant of the joint.*

Tension Joints—The External Load

The resultant bolt load is: $F_b = P_b + F_i = CP + F_i \qquad F_m < 0$

and the resultant load on the connected members is:

$$F_m = P_m - F_i = (1 - C)P - F_i \qquad F_m < 0$$

Table 8–12 is included to provide some information on the relative values of the stiffnesses encountered.

The grip contains only two members, both of steel, and no washers.

The ratios *C* and 1 - *C* are the coefficients of *P*. They describe the proportion of the external load taken by the bolt and by the members, respectively.

Tension Joints—The External Load

In all cases, the members take over 80 percent of the external load.

Think how important this is when fatigue loading is present. Note also that making the grip longer causes the members to take an even greater percentage of the external load.

Table 8–12

Computation of Bolt and Member Stiffnesses. Steel members clamped using a $\frac{1}{2}$ in-13 NC steel bolt.

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

	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

Relating Bolt Torque to Bolt Tension

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

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

Table 8–13

Distribution of Preload F_i for 20 Tests of Unlubricated Bolts Torqued to 90 N · m

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.

Table 8–14

Distribution of Preload F_i for 10 Tests of Lubricated Bolts Torqued to 90 N · m

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.

Relating Bolt Torque to Bolt Tension

The diameter of the washer face of a hexagonal nut is the same as the width across flats and equal to 1.5 times the nominal size.

Therefore the mean collar diameter is $d_c = (d + 1.5d)/\,2 = 1.25d$.

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

Or $$T = KF_i d$$

where

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

Relating Bolt Torque to Bolt Tension

Table 8–15

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

Bolt Condition	K
Nonplated, black finish	0.30
Zinc-plated	0.20
Lubricated	0.18
Cadmium-plated	0.16
With Bowman Anti-Seize	0.12
With Bowman-Grip nuts	0.09

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

Statically Loaded Tension Joint with Preload

The tensile stress in the bolt can be found as

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

Thus, the yielding factor of safety guarding against the static stress exceeding the proof strength is

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

Or

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

Since it is common to load a bolt close to the proof strength, the yielding factor of safety is often not much greater than unity.

Statically Loaded Tension Joint with Preload

Another indicator of yielding that is sometimes used is a *load factor,* which is applied only to the load *P* as a guard against overloading.

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

Solving for the load factor gives:

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

It is also essential for a safe joint that the external load be smaller than that needed to cause the joint to separate.

Statically Loaded Tension Joint with Preload

If separation does occur, then the entire external load will be imposed on the bolt.

Let P_0 be the value of the external load that would cause joint separation. At separation, $F_m = 0$:

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

Let the factor of safety against joint separation be

$$n_0 = \frac{P_0}{P}$$

Substituting $P_0 = n_0 P$

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

Statically Loaded Tension Joint with Preload

it is recommended for both static and fatigue loading that the following be used for preload:

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

where F_p is the proof load, obtained from:

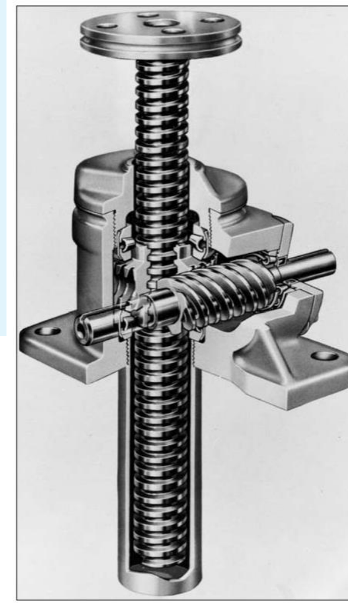
$$F_p = A_t S_p$$

Here S_p is the proof strength obtained from Tables 8–9 to 8–11. For other materials, an approximate value is $S_p = 0.85S_y$.

Examples

A square-thread power screw has a major diameter of 32 mm and a pitch of 4 mm with double threads, and it is to be used in an application similar to that in Fig. 8–4. The given data include $f = f_c = 0.08$, $d_c = 40$ mm, and $F = 6.4$ kN per screw.

(*a*) Find the thread depth, thread width, pitch diameter, minor diameter, and lead.
(*b*) Find the torque required to raise and lower the load.
(*c*) Find the efficiency during lifting the load.
(*d*) Find the body stresses, torsional and compressive.
(*e*) Find the bearing stress.
(*f*) Find the thread bending stress at the root of the thread.
(*g*) Determine the von Mises stress at the root of the thread.
(*h*) Determine the maximum shear stress at the root of the thread.



Examples

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

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

Answer

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

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

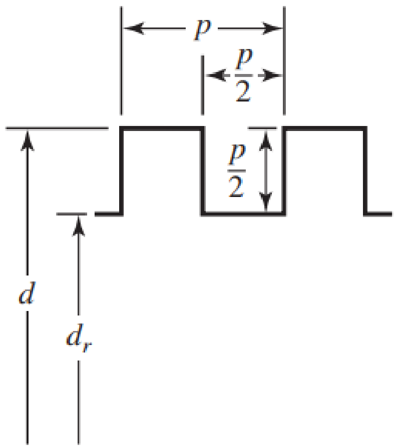
(*b*) Using Eqs. (8–1) and (8–6), the torque required to turn the screw against the load is

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

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

Answer

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



Examples

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

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

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

Answer

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

The minus sign in the first term indicates that the screw alone is not self-locking and would rotate under the action of the load except for the fact that the collar friction is present and must be overcome, too. Thus the torque required to rotate the screw "with" the load is less than is necessary to overcome collar friction alone.

(*c*) The overall efficiency in raising the load is

Answer

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

Examples

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

Answer

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

The axial nominal normal stress σ is

Answer

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

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

Answer

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

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

Answer

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

Examples

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

Answer

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

(*g*) The transverse shear at the extreme of the root cross section due to bending is zero. However, there is a circumferential shear stress at the extreme of the root cross section of the thread as shown in part (*d*) of 6.07 MPa. The three-dimensional stresses, after Fig. 8–8, noting the *y* coordinate is into the page, are

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

$$\sigma_y = -10.39 \text{ MPa} \qquad \tau_{yz} = 6.07 \text{ MPa}$$

$$\sigma_z = 0 \qquad \tau_{zx} = 0$$

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

Answer

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

$$= 48.7 \text{ MPa}$$

Examples

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

Answer

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

Examples

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

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

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

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

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

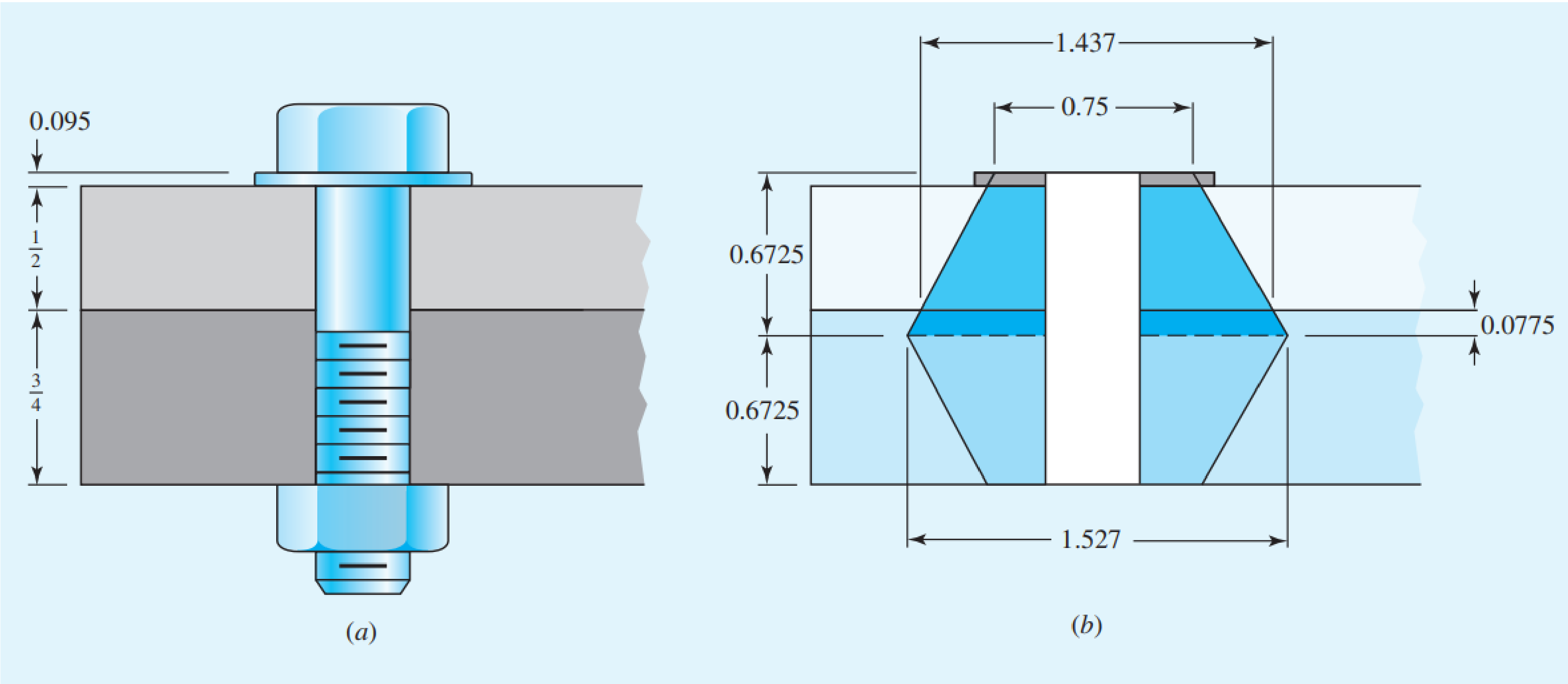
Examples

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

(*a*) As shown in Fig. 8–17*b*, the frusta extend halfway into the joint the distance

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

Examples



Examples

The distance between the joint line and the dotted frusta line is $0.6725 - 0.5 - 0.095 = 0.0775$ in. Thus, the top frusta consist of the steel washer, steel plate, and 0.0775 in of the cast iron. Since the washer and top plate are both steel with $E = 30(10^6)$ psi, they can be considered a single frustum of 0.595 in thick. The outer diameter of the frustum of the steel member at the joint interface is $0.75 + 2(0.595)\tan 30° = 1.437$ in. The outer diameter at the midpoint of the entire joint is $0.75 + 2(0.6725)\tan 30° = 1.527$ in. Using Eq. (8–20), the spring rate of the steel is

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

From Tables 8–8 or A–5, for gray cast iron, $E = 14.5$ Mpsi. Thus for the upper cast-iron frustum

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

Examples

For the lower cast-iron frustum

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

The three frusta are in series, so from Eq. (8–18)

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

Answer This results in $k_m = 9.378\ (10^6)$ lbf/in.

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

Answer

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

Examples

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

Answer

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

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

(*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 $A_d = (\pi/4)(0.5^2) = 0.196\ 3\ \text{in}^2$. From Table 8–2, the tensile-stress area is $A_t = 0.159\ 9\ \text{in}^2$. From Eq. (8–17)

Answer

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

Examples

A $\frac{3}{4}$ in-16 UNF $\times$ $2\frac{1}{2}$ in SAE grade 5 bolt is subjected to a load P of 6 kip in a tension joint. The initial bolt tension is $F_i = 25$ kip. The bolt and joint stiffnesses are $k_b = 6.50$ and $k_m = 13.8$ Mlbf/in, respectively.

(*a*) Determine the preload and service load stresses in the bolt. Compare these to the SAE minimum proof strength of the bolt.

(*b*) Specify the torque necessary to develop the preload, using Eq. (8–27).

(*c*) Specify the torque necessary to develop the preload, using Eq. (8–26) with $f = f_c = 0.15$.

Examples

Solution From Table 8–2, $A_t = 0.373 \text{ in}^2$.

(*a*) The preload stress is

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

The stiffness constant is

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

From Eq. (8–24), the stress under the service load is

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

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

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.

Examples

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

Answer

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

(*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

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

For $\alpha = 30°$, Eq. (8–26) gives

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

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

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