



Shigley's Mechanical Engineering Design

Ninth Edition

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SHIGLEY'S MECHANICAL ENGINEERING DESIGN, NINTH EDITION

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Brief Contents

Preface xv

Part 1	Basics	2
1	Introduction to Mechanical Engineering Design	3
2	Materials	31
3	Load and Stress Analysis	71
4	Deflection and Stiffness	147
Part 2	Failure Prevention	212
5	Failures Resulting from Static Loading	213
6	Fatigue Failure Resulting from Variable Loading	265
Part 3	Design of Mechanical Elements	358
7	Shafts and Shaft Components	359
8	Screws, Fasteners, and the Design of Nonpermanent Joints	409
9	Welding, Bonding, and the Design of Permanent Joints	475
10	Mechanical Springs	517
11	Rolling-Contact Bearings	569
12	Lubrication and Journal Bearings	617
13	Gears—General	673
14	Spur and Helical Gears	733
15	Bevel and Worm Gears	785
16	Clutches, Brakes, Couplings, and Flywheels	825
17	Flexible Mechanical Elements	879
18	Power Transmission Case Study	933

Part 4 Analysis Tools 952**19 Finite-Element Analysis 953****20 Statistical Considerations 977****Appendixes****A Useful Tables 1003****B Answers to Selected Problems 1059***Index* 1065

8

Screws, Fasteners, and the Design of Nonpermanent Joints

Chapter Outline

8-1	Thread Standards and Definitions	410
8-2	The Mechanics of Power Screws	414
8-3	Threaded Fasteners	422
8-4	Joints—Fastener Stiffness	424
8-5	Joints—Member Stiffness	427
8-6	Bolt Strength	432
8-7	Tension Joints—The External Load	435
8-8	Relating Bolt Torque to Bolt Tension	437
8-9	Statically Loaded Tension Joint with Preload	440
8-10	Gasketed Joints	444
8-11	Fatigue Loading of Tension Joints	444
8-12	Bolted and Riveted Joints Loaded in Shear	451

The helical-thread screw was undoubtedly an extremely important mechanical invention. It is the basis of power screws, which 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.

This book presupposes a knowledge of the elementary methods of fastening. Typical methods of fastening or joining parts use such devices as bolts, nuts, cap screws, setscrews, rivets, spring retainers, locking devices, pins, keys, welds, and adhesives. Studies in engineering graphics and in metal processes often include instruction on various joining methods, and the curiosity of any person interested in mechanical engineering naturally results in the acquisition of a good background knowledge of fastening methods. Contrary to first impressions, the subject is one of the most interesting in the entire field of mechanical design.

One of the key targets of current design for manufacture is to reduce the number of fasteners. However, there will always be a need for fasteners to facilitate disassembly for whatever purposes. For example, jumbo jets such as Boeing's 747 require as many as 2.5 million fasteners, some of which cost several dollars apiece. To keep costs down, aircraft manufacturers, and their subcontractors, constantly review new fastener designs, installation techniques, and tooling.

The number of innovations in the fastener field over any period you might care to mention has been tremendous. An overwhelming variety of fasteners are available for the designer's selection. Serious designers generally keep specific notebooks on fasteners alone. Methods of joining parts are extremely important in the engineering of a quality design, and it is necessary to have a thorough understanding of the performance of fasteners and joints under all conditions of use and design.

8-1 Thread Standards and Definitions

The terminology of screw threads, illustrated in Fig. 8-1, is explained as follows:

The *pitch* is the distance between adjacent thread forms measured parallel to the thread axis. The pitch in U.S. units is the reciprocal of the number of thread forms per inch N .

The *major diameter* d is the largest diameter of a screw thread.

The *minor* (or root) *diameter* d_r is the smallest diameter of a screw thread.

The pitch diameter d_p is a theoretical diameter between the major and minor diameters.

The *lead* l , not shown, is the distance the nut moves parallel to the screw axis when the nut is given one turn. For a single thread, as in Fig. 8-1, the lead is the same as the pitch.

A *multiple-threaded* product is one having two or more threads cut beside each other (imagine two or more strings wound side by side around a pencil). Standardized products such as screws, bolts, and nuts all have single threads; a *double-threaded* screw has a lead equal to twice the pitch, a *triple-threaded* screw has a lead equal to 3 times the pitch, and so on.

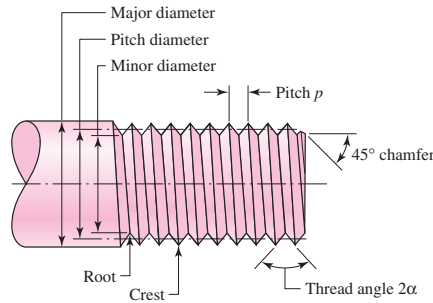
All threads are made according to the *right-hand rule* unless otherwise noted. That is, if the bolt is turned clockwise, the bolt advances toward the nut.

The *American National (Unified)* thread standard has been approved in this country and in Great Britain for use on all standard threaded products. The thread angle is 60° and the crests of the thread may be either flat or rounded.

Figure 8-2 shows the thread geometry of the metric M and MJ profiles. The M profile replaces the inch class and is the basic ISO 68 profile with 60° symmetric

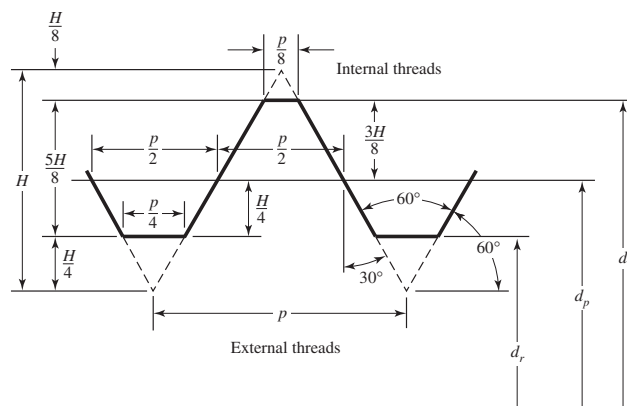
Figure 8-1

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

**Figure 8-2**

Basic profile for metric M and MJ threads.

d = major diameter
 d_r = minor diameter
 d_p = pitch diameter
 p = pitch
 $H = \frac{\sqrt{3}}{2} p$



threads. The MJ profile has a rounded fillet at the root of the external thread and a larger minor diameter of both the internal and external threads. This profile is especially useful where high fatigue strength is required.

Tables 8-1 and 8-2 will be useful in specifying and designing threaded parts. Note that the thread size is specified by giving the pitch p for metric sizes and by giving the number of threads per inch N for the Unified sizes. The screw sizes in Table 8-2 with diameter under $\frac{1}{4}$ in are numbered or gauge sizes. The second column in Table 8-2 shows that a No. 8 screw has a nominal major diameter of 0.1640 in.

A great many tensile tests of threaded rods have shown that an unthreaded rod having a diameter equal to the mean of the pitch diameter and minor diameter will have the same tensile strength as the threaded rod. The area of this unthreaded rod is called the tensile-stress area A_t of the threaded rod; values of A_t are listed in both tables.

Two major Unified thread series are in common use: UN and UNR. The difference between these is simply that a root radius must be used in the UNR series. Because of reduced thread stress-concentration factors, UNR series threads have improved fatigue strengths. Unified threads are specified by stating the nominal major diameter, the number of threads per inch, and the thread series, for example, $\frac{5}{8}$ in-18 UNRF or 0.625 in-18 UNRF.

Metric threads are specified by writing the diameter and pitch in millimeters, in that order. Thus, M12 $\times$ 1.75 is a thread having a nominal major diameter of 12 mm and a pitch of 1.75 mm. Note that the letter M, which precedes the diameter, is the clue to the metric designation.

Table 8-1

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

Nominal Major Diameter d mm	Coarse-Pitch Series			Fine-Pitch Series		
	Pitch p mm	Tensile- Stress Area A_t mm ²	Minor- Diameter Area A_r mm ²	Pitch p mm	Tensile- Stress Area A_t mm ²	Minor- Diameter Area A_r mm ²
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

*The equations and data used to develop this table have been obtained from ANSI B1.1-1974 and B18.3.1-1978. The minor diameter was found from the equation $d_r = d - 1.226\ 869p$, and the pitch diameter from $d_p = d - 0.649\ 519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

Square and Acme threads, whose profiles are shown in Fig. 8-3*a* and *b*, respectively, are used on screws when power is to be transmitted. Table 8-3 lists the preferred pitches for inch-series Acme threads. However, other pitches can be and often are used, since the need for a standard for such threads is not great.

Modifications are frequently made to both Acme and square threads. For instance, the square thread is sometimes modified by cutting the space between the teeth so as to have an included thread angle of 10 to 15°. This is not difficult, since these threads are usually cut with a single-point tool anyhow; the modification retains most of the high efficiency inherent in square threads and makes the cutting simpler. Acme threads

Table 8-2

Diameters and Area of Unified Screw Threads UNC and UNF*

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

*This table was compiled from ANSI B1.1-1974. The minor diameter was found from the equation $d_r = d - 1.299\,038p$, and the pitch diameter from $d_p = d - 0.649\,519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

Figure 8-3

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

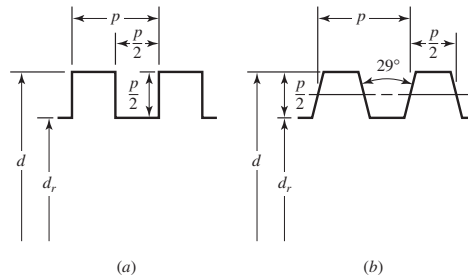


Table 8-3

Preferred Pitches for
Acme Threads

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

are sometimes modified to a stub form by making the teeth shorter. This results in a larger minor diameter and a somewhat stronger screw.

8-2 The Mechanics of Power Screws

A power screw is a device used in machinery to change angular motion into linear motion, and, usually, to transmit power. Familiar applications include the lead screws of lathes, and the screws for vises, presses, and jacks.

An application of power screws to a power-driven jack is shown in Fig. 8-4. You should be able to identify the worm, the worm gear, the screw, and the nut. Is the worm gear supported by one bearing or two?

Figure 8-4

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

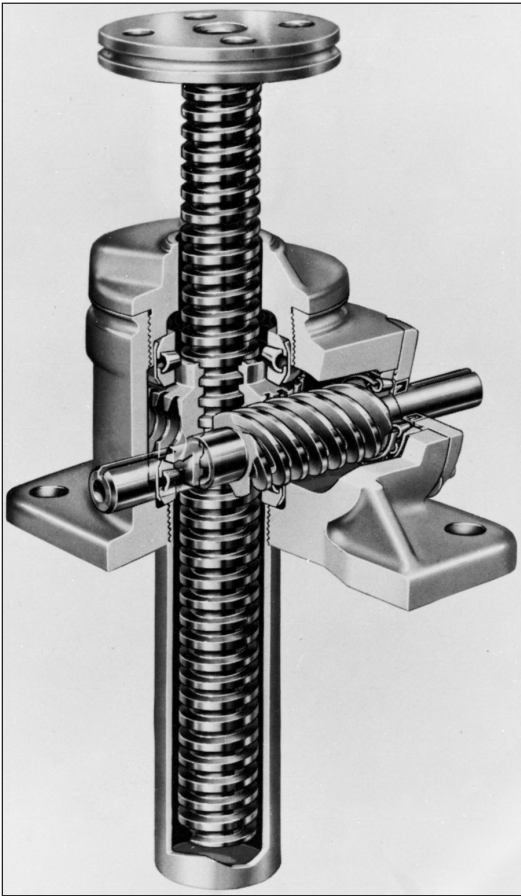
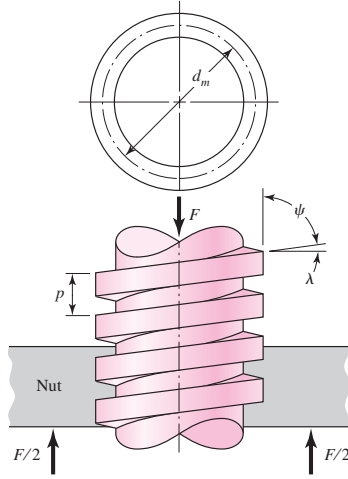
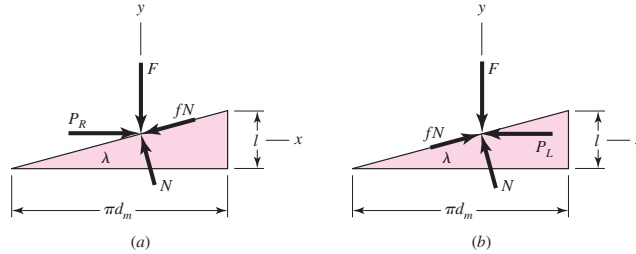


Figure 8-5

Portion of a power screw.

**Figure 8-6**

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



In Fig. 8-5 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.

First, imagine that a single thread of the screw is unrolled or developed (Fig. 8-6) for exactly a single turn. Then one edge of the thread will form the hypotenuse of a right triangle whose base is the circumference of the mean-thread-diameter circle and whose height is the lead. The angle λ , in Figs. 8-5 and 8-6, is the lead angle of the thread. We represent the summation of all the axial forces acting upon the normal thread area by F . To raise the load, a force P_R acts to the right (Fig. 8-6a), and to lower the load, P_L acts to the left (Fig. 8-6b). The friction force is the product of the coefficient of friction f with the normal force N , and acts to oppose the motion. The system is in equilibrium under the action of these forces, and hence, for raising the load, we have

$$\begin{aligned}\sum F_x &= P_R - N \sin \lambda - fN \cos \lambda = 0 \\ \sum F_y &= -F - fN \sin \lambda + N \cos \lambda = 0\end{aligned}\tag{a}$$

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

$$\begin{aligned}\sum F_x &= -P_L - N \sin \lambda + fN \cos \lambda = 0 \\ \sum F_y &= -F + fN \sin \lambda + N \cos \lambda = 0\end{aligned}\tag{b}$$

Since we are not interested in the normal force N , we eliminate it from each of these sets of equations and solve the result for P . For raising the load, this gives

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

and for lowering the load,

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

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

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

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

Finally, noting that the torque is the product of the force P and the mean radius $d_m/2$, for raising the load we can write

$$T_R = \frac{F d_m}{2} \left(\frac{l + \pi f d_m}{\pi d_m - fl} \right) \quad (8-1)$$

where T_R is the torque required for two purposes: to overcome thread friction and to raise the load.

The torque required to lower the load, from Eq. (f), is found to be

$$T_L = \frac{F d_m}{2} \left(\frac{\pi f d_m - l}{\pi d_m + fl} \right) \quad (8-2)$$

This is the torque required to overcome a part of the friction in lowering the load. It may turn out, in specific instances where the lead is large or the friction is low, that the load will lower itself by causing the screw to spin without any external effort. In such cases, the torque T_L from Eq. (8-2) will be negative or zero. When a positive torque is obtained from this equation, the screw is said to be *self-locking*. Thus the condition for self-locking is

$$\pi f d_m > l$$

Now divide both sides of this inequality by πd_m . Recognizing that $l/\pi d_m = \tan \lambda$, we get

$$f > \tan \lambda \quad (8-3)$$

This relation states that self-locking is obtained whenever the coefficient of thread friction is equal to or greater than the tangent of the thread lead angle.

An expression for efficiency is also useful in the evaluation of power screws. If we let $f = 0$ in Eq. (8-1), we obtain

$$T_0 = \frac{Fl}{2\pi} \quad (g)$$

which, since thread friction has been eliminated, is the torque required only to raise the load. The efficiency is therefore

$$e = \frac{T_0}{T_R} = \frac{Fl}{2\pi T_R} \quad (8-4)$$

The preceding equations have been developed for square threads where the normal thread loads are parallel to the axis of the screw. In the case of Acme or other threads, 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 (Fig. 8-7a) considered. The effect of the angle α is to increase the frictional force by the wedging action of the threads. Therefore the frictional terms in Eq. (8-1) must be divided by $\cos \alpha$. For raising the load, or for tightening a screw or bolt, this yields

$$T_R = \frac{Fd_m}{2} \left(\frac{l + \pi f d_m \sec \alpha}{\pi d_m - f l \sec \alpha} \right) \quad (8-5)$$

In using Eq. (8-5), remember that it is an approximation because the effect of the lead angle has been neglected.

For power screws, the Acme thread is not as efficient as the square thread, because of the additional friction due to the wedging action, but it is often preferred because it is easier to machine and permits the use of a split nut, which can be adjusted to take up for wear.

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. Figure 8-7b shows a typical thrust collar in which the load is assumed to be concentrated at the mean collar diameter d_c . If f_c is the coefficient of collar friction, the torque required is

$$T_c = \frac{F f_c d_c}{2} \quad (8-6)$$

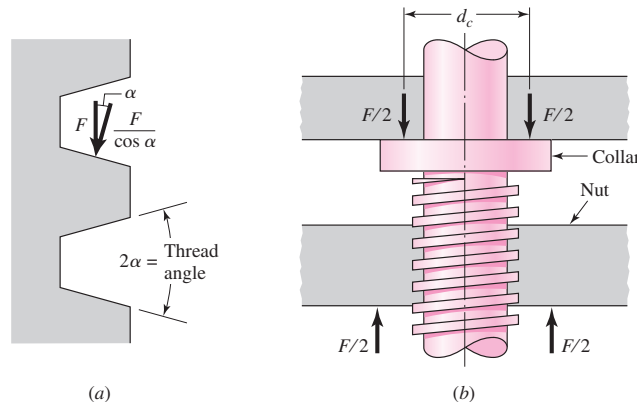
For large collars, the torque should probably be computed in a manner similar to that employed for disk clutches.

Nominal body stresses in power screws can be related to thread parameters as follows. The maximum nominal shear stress τ in torsion of the screw body can be expressed as

$$\tau = \frac{16T}{\pi d_p^3} \quad (8-7)$$

Figure 8-7

(a) Normal thread force is increased because of angle α ;
(b) thrust collar has frictional diameter d_c .



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

$$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2} \quad (8-8)$$

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

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

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

$$\sigma_B = -\frac{F}{\pi d_m n_l p/2} = -\frac{2F}{\pi d_m n_l p} \quad (8-10)$$

where n_t is the number of engaged threads. The bending stress at the root of the thread σ_b is found from

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

so

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

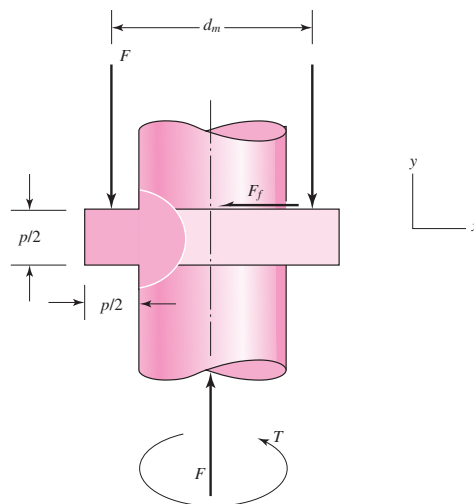
The transverse shear stress τ at the center of the root of the thread due to load F is

$$\tau = \frac{3V}{2A} = \frac{3}{2} \frac{F}{\pi d_r n_t p/2} = \frac{3F}{\pi d_r n_t p} \quad (8-12)$$

and at the top of the root it is zero. The von Mises stress σ' at the top of the root “plane” is found by first identifying the orthogonal normal stresses and the shear stresses. From

Figure 8-8

Geometry of square thread useful in finding bending and transverse shear stresses at the thread root.



the coordinate system of Fig. 8–8, we note

$$\begin{aligned}\sigma_x &= \frac{6F}{\pi d_r n_l p} & \tau_{xy} &= 0 \\ \sigma_y &= -\frac{4F}{\pi d_r^2} & \tau_{yz} &= \frac{16T}{\pi d_r^3} \\ \sigma_z &= 0 & \tau_{zx} &= 0\end{aligned}$$

then use Eq. (5–14) of Sec. 5–5.

The screw-thread form is complicated from an analysis viewpoint. Remember the origin of the tensile-stress area A_t , which comes from experiment. A power screw lifting a load is in compression and its thread pitch is *shortened* by elastic deformation. Its engaging nut is in tension and its thread pitch is *lengthened*. The engaged threads cannot share the load equally. Some experiments show that the first engaged thread carries 0.38 of the load, the second 0.25, the third 0.18, and the seventh is free of load. In estimating thread stresses by the equations above, substituting $0.38F$ for F and setting n_l to 1 will give the largest level of stresses in the thread-nut combination.

EXAMPLE 8–1

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

- Find the thread depth, thread width, pitch diameter, minor diameter, and lead.
- Find the torque required to raise and lower the load.
- Find the efficiency during lifting the load.
- Find the body stresses, torsional and compressive.
- Find the bearing stress.
- Find the thread bending stress at the root of the thread.
- Determine the von Mises stress at the root of the thread.
- Determine the maximum shear stress at the root of the thread.

Solution

- From Fig. 8–3a 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

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

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

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

$$\begin{aligned}T_R &= \frac{F d_m}{2} \left(\frac{l + \pi f d_m}{\pi d_m - f l} \right) + \frac{F f_c d_c}{2} \\ &= \frac{6.4(30)}{2} \left[\frac{8 + \pi(0.08)(30)}{\pi(30) - 0.08(8)} \right] + \frac{6.4(0.08)40}{2}\end{aligned}$$

Answer

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

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

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

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

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

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

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

Answer

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

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

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

(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

$$\begin{aligned} \sigma_x &= 41.5 \text{ MPa} & \tau_{xy} &= 0 \\ \sigma_y &= -10.39 \text{ MPa} & \tau_{yz} &= 6.07 \text{ MPa} \\ \sigma_z &= 0 & \tau_{zx} &= 0 \end{aligned}$$

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

Answer

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

$$= 48.7 \text{ MPa}$$

Alternatively, you can determine the principal stresses and then use Eq. (5–12) to find the von Mises stress. This would prove helpful in evaluating $\tau_{\max}$ as well. The principal stresses can be found from Eq. (3–15); however, sketch the stress element and note that there are no shear stresses on the x face. This means that σ_x is a principal stress. The remaining stresses can be transformed by using the plane stress equation, Eq. (3–13). Thus, the remaining principal stresses are

$$\frac{-10.39}{2} \pm \sqrt{\left(\frac{-10.39}{2}\right)^2 + 6.07^2} = 2.79, -13.18 \text{ MPa}$$

Ordering the principal stresses gives $\sigma_1, \sigma_2, \sigma_3 = 41.5, 2.79, -13.18$ MPa. Substituting these into Eq. (5–12) yields

Answer

$$\sigma' = \left\{ \frac{[41.5 - 2.79]^2 + [2.79 - (-13.18)]^2 + [-13.18 - 41.5]^2}{2} \right\}^{1/2}$$

$$= 48.7 \text{ MPa}$$

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

Answer

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

Table 8–4

Screw Bearing

Pressure p_b

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

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

Ham and Ryan¹ showed that the coefficient of friction in screw threads is independent of axial load, practically independent of speed, decreases with heavier lubricants, shows little variation with combinations of materials, and is best for steel on bronze. Sliding coefficients of friction in power screws are about 0.10–0.15.

Table 8–4 shows safe bearing pressures on threads, to protect the moving surfaces from abnormal wear. Table 8–5 shows the coefficients of sliding friction for

¹Ham and Ryan, *An Experimental Investigation of the Friction of Screw-threads*, Bulletin 247, University of Illinois Experiment Station, Champaign-Urbana, Ill., June 7, 1932.

Table 8-5

Coefficients of Friction f
for Threaded Pairs

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

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

Table 8-6

Thrust-Collar Friction
Coefficients

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

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

common material pairs. Table 8-6 shows coefficients of starting and running friction for common material pairs.

8-3 Threaded Fasteners

As you study the sections on threaded fasteners and their use, be alert to the stochastic and deterministic viewpoints. In most cases the threat is from overproof loading of fasteners, and this is best addressed by statistical methods. The threat from fatigue is lower, and deterministic methods can be adequate.

Figure 8-9 is a drawing of a standard hexagon-head bolt. 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. See Table A-29 for dimensions. 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

$$L_T = \begin{cases} 2d + \frac{1}{4} \text{ in} & L \leq 6 \text{ in} \\ 2d + \frac{1}{2} \text{ in} & L > 6 \text{ in} \end{cases} \quad (8-13)$$

and for metric bolts is

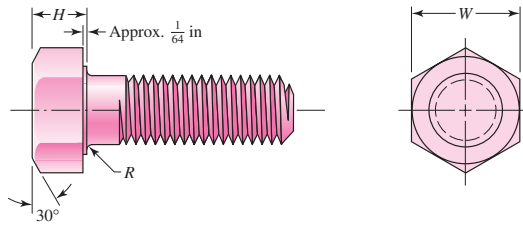
$$L_T = \begin{cases} 2d + 6 & L \leq 125 & d \leq 48 \\ 2d + 12 & 125 < L \leq 200 \\ 2d + 25 & L > 200 \end{cases} \quad (8-14)$$

where the dimensions are in millimeters. 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.

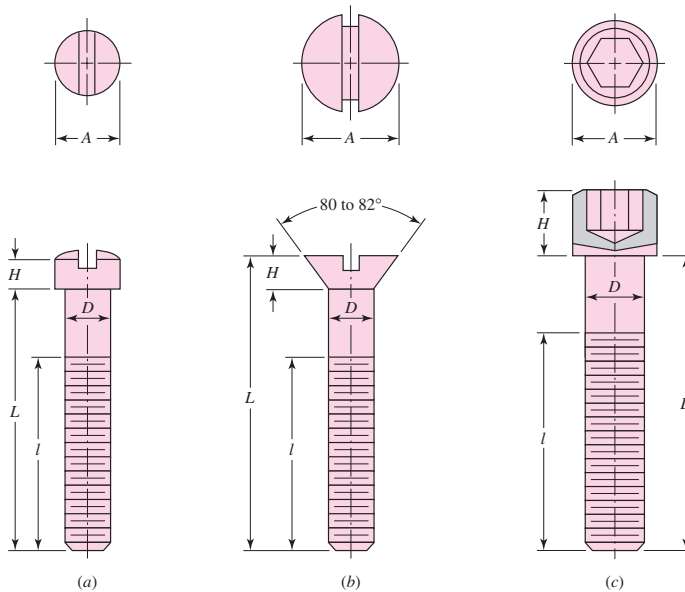
The purpose of a bolt is to clamp two or more parts together. The clamping load stretches or elongates the bolt; the load is obtained by twisting the nut until the bolt

Figure 8-9

Hexagon-head bolt; note the washer face, the fillet under the head, the start of threads, and the chamfer on both ends. Bolt lengths are always measured from below the head.

**Figure 8-10**

Typical cap-screw heads: (a) fillister head; (b) flat head; (c) hexagonal socket head. Cap screws are also manufactured with hexagonal heads similar to the one shown in Fig. 8-9, as well as a variety of other head styles. This illustration uses one of the conventional methods of representing threads.



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.

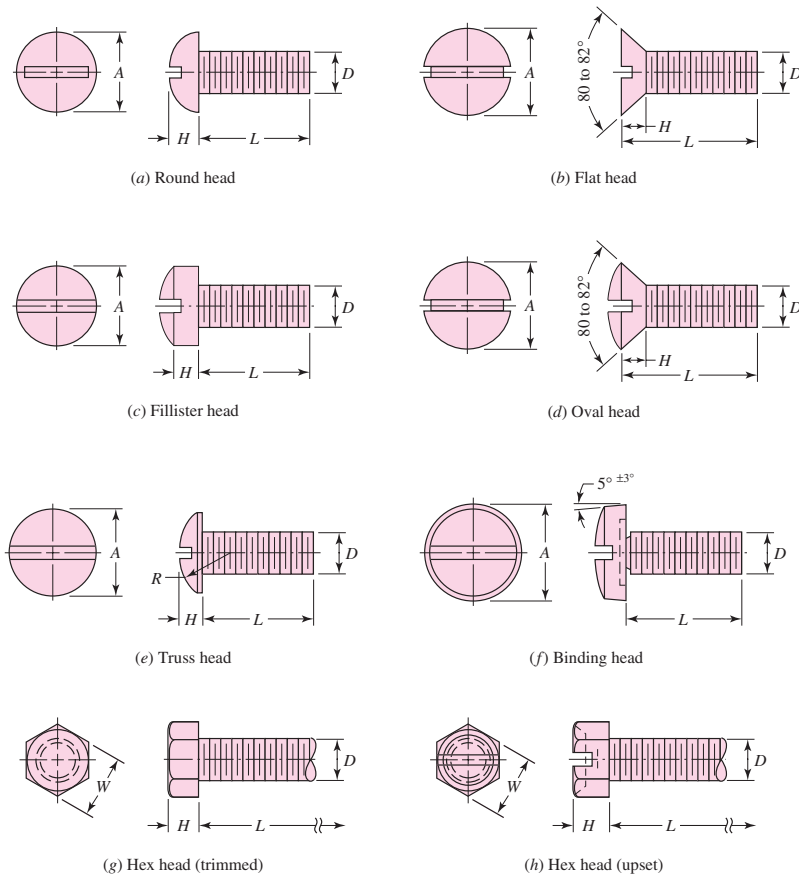
The head of a hexagon-head cap screw is slightly thinner than that of a hexagon-head bolt. Dimensions of hexagon-head cap screws are listed in Table A-30. Hexagon-head cap screws are used in the same applications as bolts and also in applications in which one of the clamped members is threaded. Three other common cap-screw head styles are shown in Fig. 8-10.

A variety of machine-screw head styles are shown in Fig. 8-11. Inch-series machine screws are generally available in sizes from No. 0 to about $\frac{3}{8}$ in.

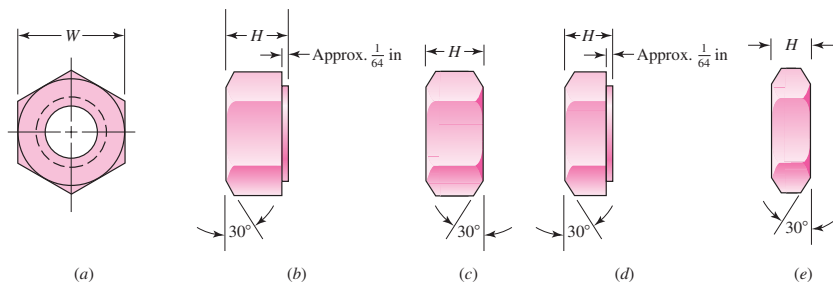
Several styles of hexagonal nuts are illustrated in Fig. 8-12; their dimensions are given in Table A-31. The material of the nut must be selected carefully to match that of the bolt. During tightening, the first thread of the nut tends to take the entire load; but yielding occurs, with some strengthening 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.

Figure 8-11

Types of heads used on machine screws.


Figure 8-12

Hexagonal nuts: (a) end view, general; (b) washer-faced regular nut; (c) regular nut chamfered on both sides; (d) jam nut with washer face; (e) jam nut chamfered on both sides.

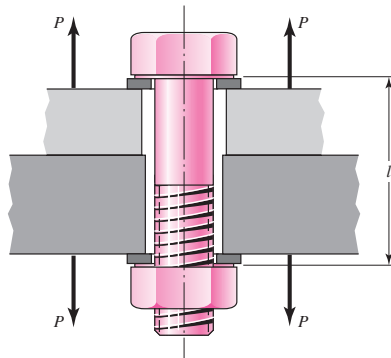


8-4 Joints—Fastener Stiffness

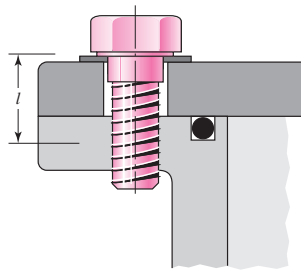
When a connection is desired that can be disassembled without destructive methods and that is strong enough to resist external tensile loads, moment loads, and shear loads, or a combination of these, then the simple bolted joint using hardened-steel washers is a good solution. Such a joint can also be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

Figure 8-13

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

**Figure 8-14**

Section of cylindrical pressure vessel. Hexagon-head cap screws are used to fasten the cylinder head to the body. Note the use of an O-ring seal. l is the effective grip of the connection (see Table 8-7).



A section through a tension-loaded bolted joint is illustrated in Fig. 8-13. Notice the clearance space provided by the bolt holes. Notice, too, how the bolt threads extend into the body of the connection.

As noted previously, 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.

Figure 8-14 shows another tension-loaded connection. This joint uses cap screws threaded into one of the members. An alternative approach to this problem (of not using a nut) would be to use studs. A stud is a rod threaded on both ends. The stud is screwed into the lower member first; then the top member is positioned and fastened down with hardened washers and nuts. The studs are regarded as permanent, and so the joint can be disassembled merely by removing the nut and washer. Thus the threaded part of the lower member is not damaged by reusing the threads.

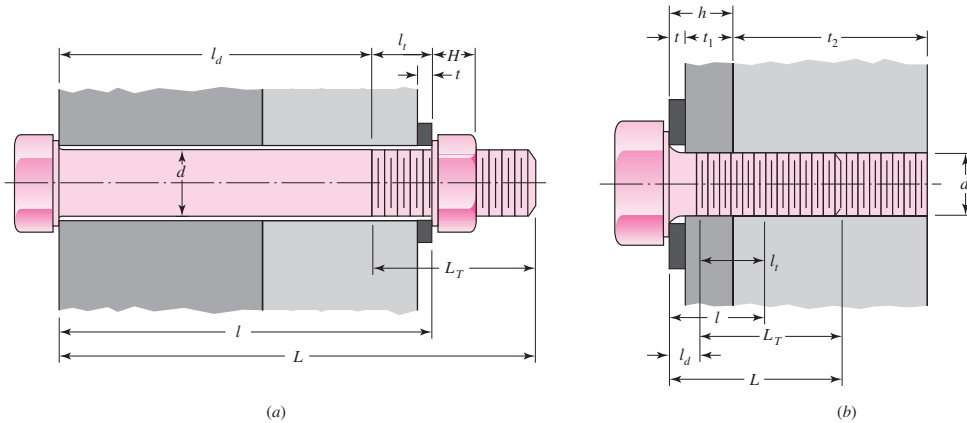
The *spring rate* is a limit as expressed in Eq. (4-1). For an elastic member such as a bolt, as we learned in Eq. (4-2), it is the ratio between the force applied to the member and the deflection produced by that force. We can use Eq. (4-4) and the results of Prob. 4-1 to find the stiffness constant of a fastener in any bolted connection.

The *grip* l of a connection is the total thickness of the clamped material. In Fig. 8-13 the grip is the sum of the thicknesses of both members and both washers. In Fig. 8-14 the effective grip is given in Table 8-7.

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

Table 8-7

Suggested Procedure for Finding Fastener Stiffness



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

*Bolts and cap screws may not be available in all the preferred lengths listed in Table A-17. Large fasteners may not be available in fractional inches or in millimeter lengths ending in a nonzero digit. Check with your bolt supplier for availability.

threaded portion. Thus the stiffness constant of the bolt is equivalent to the stiffnesses of two springs in series. Using the results of Prob. 4–1, we find

$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2} \quad \text{or} \quad k = \frac{k_1 k_2}{k_1 + k_2} \quad (8-15)$$

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

$$k_t = \frac{A_t E}{l_t} \quad k_d = \frac{A_d E}{l_d} \quad (8-16)$$

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

Substituting these stiffnesses in Eq. (8–15) gives

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} \quad (8-17)$$

where k_b is the estimated effective stiffness of the bolt or cap screw in the clamped zone. For short fasteners, the one in Fig. 8–14, for example, the unthreaded area is small and so the first of the expressions in Eq. (8–16) can be used to find k_b . For long fasteners, the threaded area is relatively small, and so the second expression in Eq. (8–16) can be used. Table 8–7 is useful.

8–5 Joints—Member Stiffness

In the previous section, we determined the stiffness of the fastener in the clamped zone. In this section, we wish to study the stiffnesses of the members in the clamped zone. Both of these stiffnesses must be known in order to learn what happens when the assembled connection is subjected to an external tensile loading.

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

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \cdots + \frac{1}{k_i} \quad (8-18)$$

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.

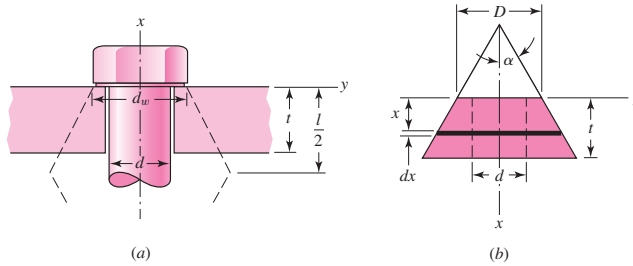
If there is no gasket, the stiffness of the members is rather difficult to obtain, except by experimentation, because the compression region spreads out between the bolt head and the nut and hence the area is not uniform. There are, however, some cases in which this area can be determined.

Ito² has used ultrasonic techniques to determine the pressure distribution at the member interface. The results show that the pressure stays high out to about 1.5 bolt radii.

²Y. Ito, J. Toyoda, and S. Nagata, “Interface Pressure Distribution in a Bolt-Flange Assembly,” ASME paper no. 77-WA/DE-11, 1977.

Figure 8-15

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



The pressure, however, falls off farther away from the bolt. Thus Ito suggests the use of Rotscher's pressure-cone method for stiffness calculations with a variable cone angle. This method is quite complicated, and so here we choose to use a simpler approach using a fixed cone angle.

Figure 8-15 illustrates the general cone geometry using a half-apex angle α . An angle $\alpha = 45^\circ$ has been used, but Little³ reports that this overestimates the clamping stiffness. When loading is restricted to a washer-face annulus (hardened steel, cast iron, or aluminum), the proper apex angle is smaller. Osgood⁴ reports a range of $25^\circ \leq \alpha \leq 33^\circ$ for most combinations. In this book we shall use $\alpha = 30^\circ$ except in cases in which the material is insufficient to allow the frusta to exist.

Referring now to Fig. 8-15b, the contraction of an element of the cone of thickness dx subjected to a compressive force P is, from Eq. (4-3),

$$d\delta = \frac{P dx}{EA} \quad (a)$$

The area of the element is

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

Substituting this in Eq. (a) and integrating gives a total contraction of

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

Using a table of integrals, we find the result to be

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

Thus the spring rate or stiffness of this frustum is

$$k = \frac{P}{\delta} = \frac{\pi E d \tan \alpha}{\ln \frac{(2t \tan \alpha + D - d)(D + d)}{(2t \tan \alpha + D + d)(D - d)}} \quad (8-19)$$

³R. E. Little, "Bolted Joints: How Much Give?" *Machine Design*, Nov. 9, 1967.

⁴C. C. Osgood, "Saving Weight on Bolted Joints," *Machine Design*, Oct. 25, 1979.

With $\alpha = 30^\circ$, this becomes

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

Equation (8-20), or (8-19), must be solved separately for each frustum in the joint. Then individual stiffnesses are assembled to obtain k_m using Eq. (8-18).

If the members of the joint have the same Young's modulus E with symmetrical frusta back to back, then they act as two identical springs in series. From Eq. (8-18) we learn that $k_m = k/2$. Using the grip as $l = 2t$ and d_w as the diameter of the washer face, from Eq. (8-19) we find the spring rate of the members to be

$$k_m = \frac{\pi Ed \tan \alpha}{2 \ln \frac{(l \tan \alpha + d_w - d)(d_w + d)}{(l \tan \alpha + d_w + d)(d_w - d)}} \quad (8-21)$$

The diameter of the washer face is about 50 percent greater than the fastener diameter for standard hexagon-head bolts and cap screws. Thus we can simplify Eq. (8-21) by letting $d_w = 1.5d$. If we also use $\alpha = 30^\circ$, then Eq. (8-21) can be written as

$$k_m = \frac{0.5774\pi Ed}{2 \ln \left(5 \frac{0.5774l + 0.5d}{0.5774l + 2.5d} \right)} \quad (8-22)$$

It is easy to program the numbered equations in this section, and you should do so. The time spent in programming will save many hours of formula plugging.

To see how good Eq. (8-21) is, solve it for k_m/Ed :

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

Earlier in the section use of $\alpha = 30^\circ$ was recommended for hardened steel, cast iron, or aluminum members. Wileman, Choudury, and Green⁵ conducted a finite element study of this problem. The results, which are depicted in Fig. 8-16, agree with the $\alpha = 30^\circ$ recommendation, coinciding exactly at the aspect ratio $d/l = 0.4$. Additionally, they offered an exponential curve-fit of the form

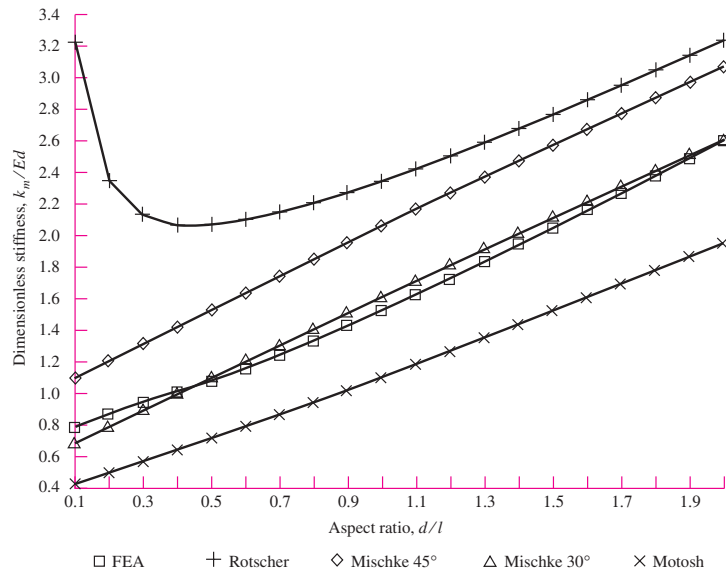
$$\frac{k_m}{Ed} = A \exp(Bd/l) \quad (8-23)$$

with constants A and B defined in Table 8-8. Equation (8-23) offers a simple calculation for member stiffness k_m . However, it is very important to note that the *entire joint* must be made up of the *same material*. For departure from these conditions, Eq. (8-20) remains the basis for approaching the problem.

⁵J. 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.

Figure 8-16

The dimensionless plot of stiffness versus aspect ratio of the members of a bolted joint, showing the relative accuracy of methods of Rotscher, Mischke, and Motosh, compared to a finite-element analysis (FEA) conducted by Wileman, Choudury, and Green.

**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

EXAMPLE 8-2

As shown in Fig. 8-17a, two plates are clamped by washer-faced $\frac{1}{2}$ in-20 UNF $\times$ $1\frac{1}{2}$ 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 .

Solution

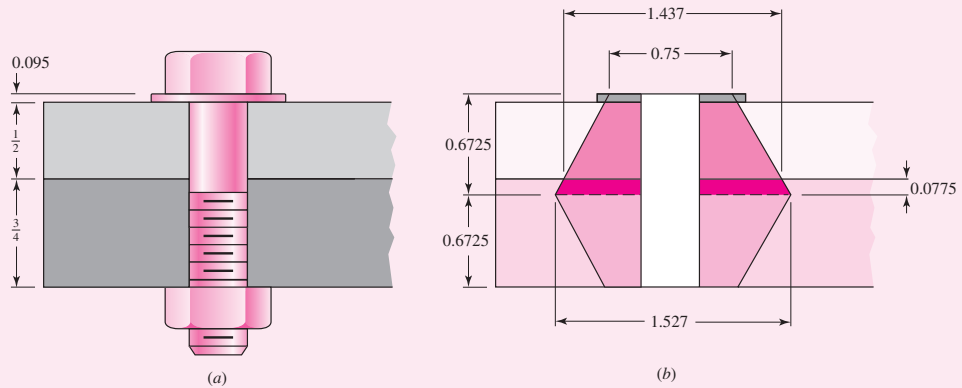
From Table A-32, the thickness of a standard $\frac{1}{2}$ N plain washer is 0.095 in.

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

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

Figure 8-17

Dimensions in inches.



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 $0.75 + 2(0.595) \tan 30^\circ = 1.437$ in. The outer diameter at the midpoint of the entire joint is $0.75 + 2(0.6725) \tan 30^\circ = 1.527$ in. Using Eq. (8-20), the spring rate of the steel is

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

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

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

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

(c) From Table 8–8, $A = 0.78715$, $B = 0.62873$. Equation (8–23) gives

Answer $k_m = 30(10^6)(0.5)(0.78715) \exp[0.62873(0.5)/1.345] = 14.92(10^6) \text{ lbf/in}$

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 $A_d = (\pi/4)(0.5)^2 = 0.1963 \text{ in}^2$. From Table 8–2, the tensile-stress area is $A_t = 0.1599 \text{ in}^2$. From Eq. (8–17)

Answer
$$k_b = \frac{0.1963(0.1599)30(10^6)}{0.1963(1.095) + 0.1599(0.25)} = 3.69(10^6) \text{ lbf/in}$$

8–6 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 and corresponds to 0.0001-in permanent set in the fastener (first measurable deviation from elastic behavior). Tables 8–9, 8–10, and 8–11 provide *minimum* strength specifications for steel bolts. The values of the mean proof strength, the mean tensile strength, and the corresponding standard deviations are not part of the specification codes, so it is the designer's responsibility to obtain these values, perhaps by laboratory testing, if designing to a reliability specification.

The SAE specifications are found in Table 8–9. The bolt grades are numbered according to the tensile strengths, with decimals used for variations at the same strength level. Bolts and screws are available in all grades listed. Studs are available in grades 1, 2, 4, 5, 8, and 8.1. Grade 8.1 is not listed.

ASTM specifications are listed in Table 8–10. ASTM threads are shorter because ASTM deals mostly with structures; structural connections are generally loaded in shear, and the decreased thread length provides more shank area.

Specifications for metric fasteners are given in Table 8–11.

It is worth noting that all specification-grade bolts made in this country bear a manufacturer's mark or logo, in addition to the grade marking, on the bolt head. Such marks confirm that the bolt meets or exceeds specifications. If such marks are missing, the bolt may be imported; for imported bolts there is no obligation to meet specifications.

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

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	

*Minimum strengths are strengths exceeded by 99 percent of fasteners.

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 $\leq 15^\circ$). Inattention to this requirement has led to a record of 15 percent fastener fatigue failure under the head, 20 percent at thread runout, and 65 percent where the designer is focusing attention. It does little good to concentrate on the plane of the nut washer face if it is not the weakest location.

Nuts are graded so that they can be mated with their corresponding grade of bolt. The purpose of the nut is to have its threads deflect to distribute the load of the bolt more evenly to the nut. The nut's properties are controlled in order to accomplish this. The grade of the nut should be the grade of the bolt.

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 $1\frac{1}{8}$ – $1\frac{1}{2}$	85	120	92	Medium carbon, Q&T	
		74	105	81		
A325, type 2	$\frac{1}{2}$ –1 $1\frac{1}{8}$ – $1\frac{1}{2}$	85	120	92	Low-carbon, martensite, Q&T	
		74	105	81		
A325, type 3	$\frac{1}{2}$ –1 $1\frac{1}{8}$ – $1\frac{1}{2}$	85	120	92	Weathering steel, Q&T	
		74	105	81		
A354, grade BC	$\frac{1}{4}$ – $2\frac{1}{2}$ $2\frac{3}{4}$ –4	105	125	109	Alloy steel, Q&T	
		95	115	99		
A354, grade BD	$\frac{1}{4}$ –4	120	150	130	Alloy steel, Q&T	
A449	$\frac{1}{4}$ –1	85	120	92	Medium-carbon, Q&T	
	$1\frac{1}{8}$ – $1\frac{1}{2}$	74	105	81		
	$1\frac{3}{4}$ –3	55	90	58		
A490, type 1	$\frac{1}{2}$ – $1\frac{1}{2}$	120	150	130	Alloy steel, Q&T	
A490, type 3	$\frac{1}{2}$ – $1\frac{1}{2}$	120	150	130	Weathering steel, Q&T	

*Minimum strengths are strengths exceeded by 99 percent of fasteners.

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	
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	

*The thread length for bolts and cap screws is

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

where L is the bolt length. The thread length for structural bolts is slightly shorter than given above.[†]Minimum strengths are strengths exceeded by 99 percent of fasteners.

8-7 Tension Joints—The External Load

Let us now consider what happens when an external tensile load P , as in Fig. 8-13, 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:

$$F_i = \text{preload}$$

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

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

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

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

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

or

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

Since $P = P_b + P_m$, we have

$$P_b = \frac{k_b P}{k_b + k_m} = C P \quad (d)$$

and

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

where

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

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

$$F_b = P_b + F_i = C P + F_i \quad F_m < 0 \quad (8-24)$$

and the resultant load on the connected members is

$$F_m = P_m - F_i = (1 - C)P - F_i \quad F_m < 0 \quad (8-25)$$

Of course, these results are valid only as long as some clamping load remains in the members; this is indicated by the qualifier in the equations.

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

Table 8-12

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

Bolt Grip, in	Stiffnesses, M lbf/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

washers. The ratios C and $1 - C$ are the coefficients of P in Eqs. (8-24) and (8-25), respectively. They describe the proportion of the external load taken by the bolt and by the members, respectively. 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.

8-8 Relating Bolt Torque to Bolt Tension

Having learned that a high preload is very desirable in important bolted connections, we must next consider means of ensuring that the preload is actually developed when the parts are assembled.

If the overall length of the bolt can actually be measured with a micrometer when it is assembled, the bolt elongation due to the preload F_i can be computed using the formula $\delta = F_i l / (AE)$. Then the nut is simply tightened until the bolt elongates through the distance δ . This ensures that the desired preload has been attained.

The elongation of a screw cannot usually be measured, because the threaded end is often in a blind hole. It is also impractical in many cases to measure bolt elongation. In such cases the wrench torque required to develop the specified preload must be estimated. Then torque wrenching, pneumatic-impact wrenching, or the turn-of-the-nut method may be used.

The torque wrench has a built-in dial that indicates the proper torque.

With impact wrenching, 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.

The turn-of-the-nut method requires that we first define the meaning of snug-tight. The *snug-tight* condition is the tightness attained by a few impacts of an impact wrench, or the full effort of a person using an ordinary wrench. When the snug-tight condition is attained, all additional turning develops useful tension in the bolt. The turn-of-the-nut method requires that you compute the fractional number of turns necessary to develop the required preload from the snug-tight condition. For example, for heavy hexagonal structural bolts, the turn-of-the-nut specification states that the nut should be turned a minimum of 180° from the snug-tight condition under optimum conditions. Note that this is also about the correct rotation for the wheel nuts of a passenger car. Problems 8-15 to 8-17 illustrate the method further.

Although the coefficients of friction may vary widely, we can obtain a good estimate of the torque required to produce a given preload by combining Eqs. (8-5) and (8-6):

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

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.

where d_m is the average of the major and minor diameters. Since $\tan \lambda = l/\pi d_m$, we divide the numerator and denominator of the first term by πd_m and get

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

The diameter of the washer face of a hexagonal nut is the same as the width across flats and equal to $1\frac{1}{2}$ times the nominal size. Therefore the mean collar diameter is $d_c = (d + 1.5d)/2 = 1.25d$. Equation (b) can now be arranged to give

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

We now define a *torque coefficient* K as the term in brackets, and so

$$K = \left(\frac{d_m}{2d} \right) \left(\frac{\tan \lambda + f \sec \alpha}{1 - f \tan \lambda \sec \alpha} \right) + 0.625 f_c \quad (8-26)$$

Equation (c) can now be written

$$T = K F_i d \quad (8-27)$$

The coefficient of friction depends upon the surface smoothness, accuracy, and degree of lubrication. On the average, both f and f_c are about 0.15. The interesting fact about Eq. (8-26) is that $K \doteq 0.20$ for $f = f_c = 0.15$ no matter what size bolts are employed and no matter whether the threads are coarse or fine.

Blake and Kurtz have published results of numerous tests of the torquing of bolts.⁶ By subjecting their data to a statistical analysis, we can learn something about the distribution of the torque coefficients and the resulting preload. Blake and Kurtz determined the preload in quantities of unlubricated and lubricated bolts of size $\frac{1}{2}$ in-20 UNF when torqued to 800 lbf · in. This corresponds roughly to an M12 × 1.25 bolt torqued to 90 N · m. The statistical analyses of these two groups of bolts, converted to SI units, are displayed in Tables 8-13 and 8-14.

We first note that both groups have about the same mean preload, 34 kN. The unlubricated bolts have a standard deviation of 4.9 kN and a COV of about 0.15. The lubricated bolts have a standard deviation of 3 kN and a COV of about 0.9.

The means obtained from the two samples are nearly identical, approximately 34 kN; using Eq. (8-27), we find, for both samples, $K = 0.208$.

Bowman Distribution, a large manufacturer of fasteners, recommends the values shown in Table 8-15. In this book we shall use these values and use $K = 0.2$ when the bolt condition is not stated.

⁶J. C. Blake and H. J. Kurtz, "The Uncertainties of Measuring Fastener Preload," *Machine Design*, vol. 37, Sept. 30, 1965, pp. 128-131.

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

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

EXAMPLE 8-3

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

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

Solution

From Table 8-2, $A_t = 0.373$ in².

(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

Answer
$$\begin{aligned} \sigma_b &= \frac{F_b}{A_t} = \frac{CP + F_i}{A_t} = C \frac{P}{A_t} + \sigma_i \\ &= 0.320 \frac{6}{0.373} + 67.02 = 72.17 \text{ kpsi} \end{aligned}$$

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.

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

Answer

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

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

8-9

Statically Loaded Tension Joint with Preload

Equations (8-24) and (8-25) represent the forces in a bolted joint with preload. The tensile stress in the bolt can be found as in Ex. 8-3 as

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t} \quad (a)$$

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

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

or

$$n_p = \frac{S_p A_t}{CP + F_i} \quad (8-28)$$

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. 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. Applying such a load factor to the load P in Eq. (a), and equating it to the proof strength gives

$$\frac{C n_L P + F_i}{A_t} = S_p \quad (c)$$

Solving for the load factor gives

$$n_L = \frac{S_p A_t - F_i}{CP} \quad (8-29)$$

It is also essential for a safe joint that the external load be smaller than that needed to cause the joint to separate. 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, $F_m = 0$ in Eq. (8-25), and so

$$(1 - C)P_0 - F_i = 0 \quad (d)$$

Let the factor of safety against joint separation be

$$n_0 = \frac{P_0}{P} \quad (e)$$

Substituting $P_0 = n_0 P$ in Eq. (d), we find

$$n_0 = \frac{F_i}{P(1 - C)} \quad (8-30)$$

as a load factor guarding against joint separation.

Figure 8-18 is the stress-strain diagram of a good-quality bolt material. Notice that there is no clearly defined yield point and that the diagram progresses smoothly up to fracture, which corresponds to the tensile strength. This means that no matter how much preload is given the bolt, it will retain its load-carrying capacity. This is what keeps the bolt tight and determines the joint strength. The pretension is the “muscle” of the joint, and its magnitude is determined by the bolt strength. If the full bolt strength is not used in developing the pretension, then money is wasted and the joint is weaker.

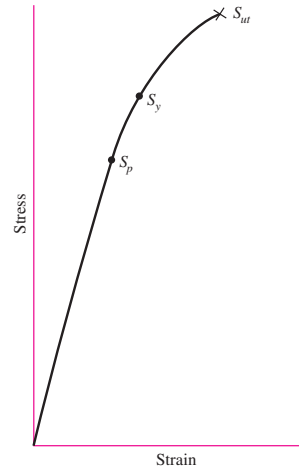
Good-quality bolts can be preloaded into the plastic range to develop more strength. Some of the bolt torque used in tightening produces torsion, which increases the principal tensile stress. However, this torsion is held only by the friction of the bolt head and nut; in time it relaxes and lowers the bolt tension slightly. Thus, as a rule, a bolt will either fracture during tightening, or not at all.

Above all, do not rely too much on wrench torque; it is not a good indicator of preload. Actual bolt elongation should be used whenever possible—especially with fatigue loading. In fact, if high reliability is a requirement of the design, then preload should always be determined by bolt elongation.

Russell, Burdsall & Ward Inc. (RB&W) recommendations for preload are 60 kpsi for SAE grade 5 bolts for nonpermanent connections, and that A325 bolts (equivalent to SAE grade 5) used in structural applications be tightened to proof load or beyond

Figure 8-18

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



(85 kpsi up to a diameter of 1 in).⁷ Bowman⁸ recommends a preload of 75 percent of proof load, which is about the same as the RB&W recommendations for reused bolts. In view of these guidelines, it is recommended for both static and fatigue loading that the following be used for preload:

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

where F_p is the proof load, obtained from the equation

$$F_p = A_t S_p \quad (8-32)$$

Here S_p is the proof strength obtained from Tables 8–9 to 8–11. For other materials, an approximate value is $S_p = 0.85S_y$. Be very careful not to use a soft material in a threaded fastener. For high-strength steel bolts used as structural steel connectors, if advanced tightening methods are used, tighten to yield.

You can see that the RB&W recommendations on preload are in line with what we have encountered in this chapter. The purposes of development were to give the reader the perspective to appreciate Eqs. (8–31) and a methodology with which to handle cases more specifically than the recommendations.

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 in part (b), determine the realized load factor for overload, the yielding factor of safety, and the load factor for joint separation.

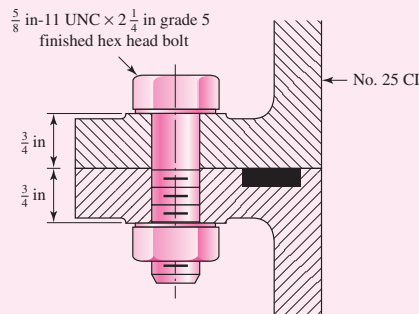
Solution

- The grip is $l = 1.50$ in. From Table A–31, the nut thickness is $\frac{35}{64}$ in. Adding two threads beyond the nut of $\frac{2}{11}$ in gives a bolt length of

$$L = \frac{35}{64} + 1.50 + \frac{2}{11} = 2.229 \text{ in}$$

From Table A–17 the next fraction size bolt is $L = 2\frac{1}{4}$ in. From Eq. (8–13), the thread length is $L_T = 2(0.625) + 0.25 = 1.50$ in. Thus, the length of the unthreaded portion

| Figure 8-19



⁷Russell, Burdsall & Ward Inc., *Helpful Hints for Fastener Design and Application*, Mentor, Ohio, 1965, p. 42.

⁸Bowman Distribution–Barnes Group, *Fastener Facts*, Cleveland, 1985, p. 90.

in the grip is $l_d = 2.25 - 1.50 = 0.75$ in. The threaded length in the grip is $l_t = l - l_d = 0.75$ in. From Table 8-2, $A_t = 0.226$ in². The major-diameter area is $A_d = \pi(0.625)^2/4 = 0.3068$ in². The bolt stiffness is then

Answer

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

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

Answer

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

If you are using Eq. (8-23), from Table 8-8, $A = 0.77871$ and $B = 0.61616$, and

$$\begin{aligned} k_m &= E d A \exp(Bd/l) \\ &= 14(0.625)(0.77871) \exp[0.61616(0.625)/1.5] \\ &= 8.81 \text{ Mlbf/in} \end{aligned}$$

which is only 1.6 percent lower than the previous result.

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

Answer

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

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

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

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

$$n_L = \frac{S_p A_t - F_i}{C(P_{\text{total}}/N)} \quad (1)$$

or

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

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

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

Answer

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

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

Answer

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

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

Answer

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

8-10 Gasketed Joints

If a full gasket is present in the joint, the gasket pressure p is found by dividing the force in the member by the gasket area per bolt. Thus, for N bolts,

$$p = \frac{F_m}{A_g/N} \quad (a)$$

With a load factor n , Eq. (8-25) can be written as

$$F_m = (1 - C)nP - F_i \quad (b)$$

Substituting this into Eq. (a) gives the gasket pressure as

$$p = [F_i - nP(1 - C)] \frac{N}{A_g} \quad (8-33)$$

In full-gasketed joints uniformity of pressure on the gasket is important. To maintain adequate uniformity of pressure adjacent bolts should not be placed more than six nominal diameters apart on the bolt circle. To maintain wrench clearance, bolts should be placed at least three diameters apart. A rough rule for bolt spacing around a bolt circle is

$$3 \leq \frac{\pi D_b}{Nd} \leq 6 \quad (8-34)$$

where D_b is the diameter of the bolt circle and N is the number of bolts.

8-11 Fatigue Loading of Tension Joints

Tension-loaded bolted joints subjected to fatigue action can be analyzed directly by the methods of Chap. 6. Table 8-16 lists average fatigue stress-concentration factors for the fillet under the bolt head and also at the beginning of the threads on the bolt shank. These are already corrected for notch sensitivity and for surface finish. Designers should be aware that situations may arise in which it would be advisable to investigate these factors more closely, since they are only average values. In fact, Peterson⁹ observes that the distribution of typical bolt failures is about 15 percent under the head, 20 percent at the end of the thread, and 65 percent in the thread at the nut face.

Use of rolled threads is the predominant method of thread-forming in screw fasteners, where Table 8-16 applies. In thread-rolling, the amount of cold work and strain-strengthening is unknown to the designer; therefore, fully corrected (including K_f) axial endurance strength is reported in Table 8-17. For cut threads, the methods of Chap. 6 are useful. Anticipate that the endurance strengths will be considerably lower.

Table 8-16

Fatigue Stress-Concentration Factors K_f for Threaded Elements

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

⁹W. D. Pilkey, *Peterson's Stress-Concentration Factors*, 2nd ed., John Wiley & Sons, New York, 1997, p. 387.

Table 8-17

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

Grade or Class	Size Range	Endurance Strength
SAE 5	$\frac{1}{4}$ –1 in	18.6 kpsi
	$1\frac{1}{8}$ – $1\frac{1}{2}$ in	16.3 kpsi
SAE 7	$\frac{1}{4}$ – $1\frac{1}{2}$ in	20.6 kpsi
SAE 8	$\frac{1}{4}$ – $1\frac{1}{2}$ in	23.2 kpsi
ISO 8.8	M16–M36	129 MPa
ISO 9.8	M1.6–M16	140 MPa
ISO 10.9	M5–M36	162 MPa
ISO 12.9	M1.6–M36	190 MPa

*Repeatedly applied, axial loading, fully corrected.

For a general case with a constant preload, and an external load on a per bolt basis fluctuating between $P_{\min}$ and $P_{\max}$, a bolt will experience fluctuating forces such that

$$F_{b\min} = CP_{\min} + F_i \quad (a)$$

$$F_{b\max} = CP_{\max} + F_i \quad (b)$$

The alternating stress experienced by a bolt is

$$\begin{aligned} \sigma_a &= \frac{(F_{b\max} - F_{b\min})/2}{A_t} = \frac{(CP_{\max} + F_i) - (CP_{\min} + F_i)}{2A_t} \\ \sigma_a &= \frac{C(P_{\max} - P_{\min})}{2A_t} \end{aligned} \quad (8-35)$$

The midrange stress experienced by a bolt is

$$\begin{aligned} \sigma_m &= \frac{(F_{b\max} + F_{b\min})/2}{A_t} = \frac{(CP_{\max} + F_i) + (CP_{\min} + F_i)}{2A_t} \\ \sigma_m &= \frac{C(P_{\max} + P_{\min})}{2A_t} + \frac{F_i}{A_t} \end{aligned} \quad (8-36)$$

A load line typically experienced by a bolt is shown in Fig. 8-20, where the stress starts from the preload stress and increases with a constant slope of $\sigma_a/(\sigma_m - \sigma_i)$. The Goodman failure line is also shown in Fig. 8-20. The fatigue factor of safety can be found by intersecting the load line and the Goodman line to find the intersection point (S_m , S_a). The load line is given by

$$\text{Load line:} \quad S_a = \frac{\sigma_a}{\sigma_m - \sigma_i}(S_m - \sigma_i) \quad (a)$$

The Goodman line, rearranging Eq. (6-40), p. 306, is

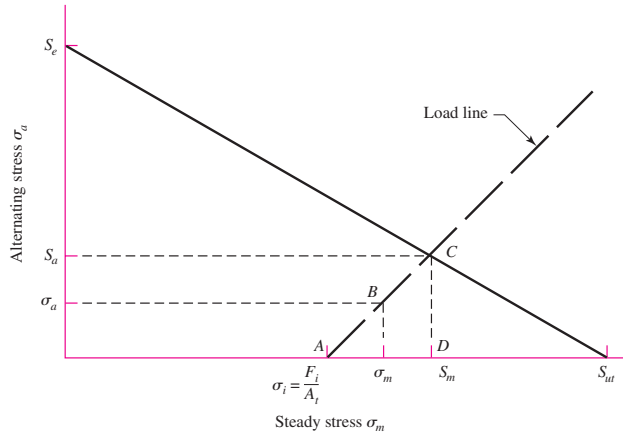
$$\text{Goodman line:} \quad S_a = S_e - \frac{S_e}{S_{ut}}S_m \quad (b)$$

Equating Eqs. (a) and (b), solving for S_m , then substituting S_m back into Eq. (b) yields

$$S_a = \frac{S_e\sigma_a(S_{ut} - \sigma_i)}{S_{ut}\sigma_a + S_e(\sigma_m - \sigma_i)} \quad (c)$$

Figure 8-20

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



The fatigue factor of safety is given by

$$n_f = \frac{S_a}{\sigma_a} \quad (8-37)$$

Substituting Eq. (c) into Eq. (8-37) gives

$$n_f = \frac{S_e(S_{ut} - \sigma_i)}{S_{ut}\sigma_a + S_e(\sigma_m - \sigma_i)} \quad (8-38)$$

The same approach can be used for the other failure curves, though the algebra is a bit more tedious to put in equation form such as Eq. (8-38). An easier approach would be to solve in stages numerically, first S_m , then S_a , and finally n_f .

Often, the type of fatigue loading encountered in the analysis of bolted joints is one in which the externally applied load fluctuates between zero and some maximum force P . This would be the situation in a pressure cylinder, for example, where a pressure either exists or does not exist. For such cases, Eqs. (8-35) and (8-36) can be simplified by setting $P_{\max} = P$ and $P_{\min} = 0$, resulting in

$$\sigma_a = \frac{CP}{2A_t} \quad (8-39)$$

$$\sigma_m = \frac{CP}{2A_t} + \frac{F_i}{A_t} \quad (8-40)$$

Note that Eq. (8-40) can be viewed as the sum of the alternating stress and the pre-load stress. If the preload is considered to be constant, the load line relationship between the alternating and midrange stresses can be treated as

$$\sigma_m = \sigma_a + \sigma_i \quad (8-41)$$

This load line has a slope of unity, and is a special case of the load line shown in Fig. 8-20. With the simplifications in the algebra, we can now proceed as before to obtain the fatigue factor of safety using each of the typical failure criteria, duplicated here from Eqs. (6-40), (6-41), and (6-42).

Goodman:

$$\frac{S_a}{S_e} + \frac{S_m}{S_{ut}} = 1 \quad (8-42)$$

Gerber:

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

ASME-elliptic:

$$\left(\frac{S_a}{S_e} \right)^2 + \left(\frac{S_m}{S_p} \right)^2 = 1 \quad (8-44)$$

Now if we intersect Eq. (8-41) and each of Eqs. (8-42) to (8-44) to solve for S_a , and apply Eq. (8-37), we obtain fatigue factors of safety for each failure criteria in a repeated loading situation.

Goodman:

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

Gerber:

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

ASME-elliptic:

$$n_f = \frac{S_e}{\sigma_a(S_p^2 + S_e^2)} \left(S_p \sqrt{S_p^2 + S_e^2 - \sigma_i^2} - \sigma_i S_e \right) \quad (8-47)$$

Note that Eqs. (8-45) to (8-47) are only applicable for repeated loads. Be sure to use K_f for both σ_a and σ_m . Otherwise, the slope of the load line will not remain 1 to 1.

If desired, σ_a from Eq. (8-39) and $\sigma_i = F_i/A_t$ can be directly substituted into any of Eqs. (8-45) to (8-47). If we do so for the Goodman criteria in Eq. (8-45), we obtain

$$n_f = \frac{2S_e(S_{ut}A_t - F_i)}{C P(S_{ut} + S_e)} \quad (8-48)$$

when preload F_i is present. With no preload, $C = 1$, $F_i = 0$, and Eq. (8-48) becomes

$$n_{f0} = \frac{2S_e S_{ut} A_t}{P(S_{ut} + S_e)} \quad (8-49)$$

Preload is beneficial for resisting fatigue when n_f/n_{f0} is greater than unity. For Goodman, Eqs. (8-48) and (8-49) with $n_f/n_{f0} \geq 1$ puts an upper bound on the preload F_i of

$$F_i \leq (1 - C)S_{ut}A_t \quad (8-50)$$

If this cannot be achieved, and n_f is unsatisfactory, use the Gerber or ASME-elliptic criterion to obtain a less conservative assessment. If the design is still not satisfactory, additional bolts and/or a different size bolt may be called for.

Bolts loosen, as they are friction devices, and cyclic loading and vibration as well as other effects allow the fasteners to lose tension with time. How does one fight loosening? Within strength limitations, the higher the preload the better. A rule of thumb is that preloads of 60 percent of proof load rarely loosen. If more is better, how much more? Well, not enough to create reused fasteners as a future threat. Alternatively, fastener-locking schemes can be employed.

After solving for the fatigue factor of safety, you should also check the possibility of yielding, using the proof strength

$$n_p = \frac{S_p}{\sigma_m + \sigma_a} \quad (8-51)$$

which is equivalent to Eq. (8-28).

EXAMPLE 8-5

Figure 8-21 shows a connection using cap screws. The joint is subjected to a fluctuating force whose maximum value is 5 kip per screw. The required data are: cap screw, 5/8 in-11 NC, SAE 5; hardened-steel washer, $t_w = \frac{1}{16}$ in thick; steel cover plate, $t_1 = \frac{5}{8}$ in, $E_s = 30$ Mpsi; and cast-iron base, $t_2 = \frac{5}{8}$ in, $E_{ci} = 16$ Mpsi.

- (a) Find k_b , k_m , and C using the assumptions given in the caption of Fig. 8-21.
(b) Find all factors of safety and explain what they mean.

Solution

(a) For the symbols of Figs. 8-15 and 8-21, $h = t_1 + t_w = 0.6875$ in, $l = h + d/2 = 1$ in, and $D_2 = 1.5d = 0.9375$ in. The joint is composed of three frusta; the upper two frusta are steel and the lower one is cast iron.

For the upper frustum: $t = l/2 = 0.5$ in, $D = 0.9375$ in, and $E = 30$ Mpsi. Using these values in Eq. (8-20) gives $k_1 = 46.46$ Mlbf/in.

For the middle frustum: $t = h - l/2 = 0.1875$ in and $D = 0.9375 + 2(l - h) \tan 30^\circ = 1.298$ in. With these and $E_s = 30$ Mpsi, Eq. (8-20) gives $k_2 = 197.43$ Mlbf/in.

The lower frustum has $D = 0.9375$ in, $t = l - h = 0.3125$ in, and $E_{ci} = 16$ Mpsi. The same equation yields $k_3 = 32.39$ Mlbf/in.

Substituting these three stiffnesses into Eq. (8-18) gives $k_m = 17.40$ Mlbf/in. The cap screw is short and threaded all the way. Using $l = 1$ in for the grip and $A_t = 0.226$ in² from Table 8-2, we find the stiffness to be $k_b = A_t E / l = 6.78$ Mlbf/in. Thus the joint constant is

Answer

$$C = \frac{k_b}{k_b + k_m} = \frac{6.78}{6.78 + 17.40} = 0.280$$

Figure 8-21

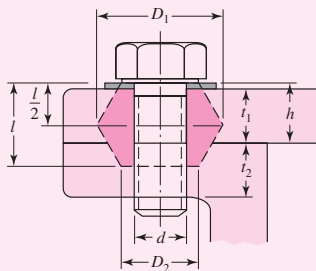
Pressure-cone frustum member model for a cap screw. For this model the significant sizes are

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

$$D_1 = d_w + l \tan \alpha = 1.5d + 0.577l$$

$$D_2 = d_w = 1.5d$$

where l = effective grip. The solutions are for $\alpha = 30^\circ$ and $d_w = 1.5d$.



(b) Equation (8–30) gives the preload as

$$F_i = 0.75F_p = 0.75A_t S_p = 0.75(0.226)(85) = 14.4 \text{ kip}$$

where from Table 8–9, $S_p = 85$ kpsi for an SAE grade 5 cap screw. Using Eq. (8–28), we obtain the load factor as the yielding factor of safety is

Answer
$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{85(0.226)}{0.280(5) + 14.4} = 1.22$$

This is the traditional factor of safety, which compares the maximum bolt stress to the proof strength.

Using Eq. (8–29),

Answer
$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{85(0.226) - 14.4}{0.280(5)} = 3.44$$

This factor is an indication of the overload on P that can be applied without exceeding the proof strength.

Next, using Eq. (8–30), we have

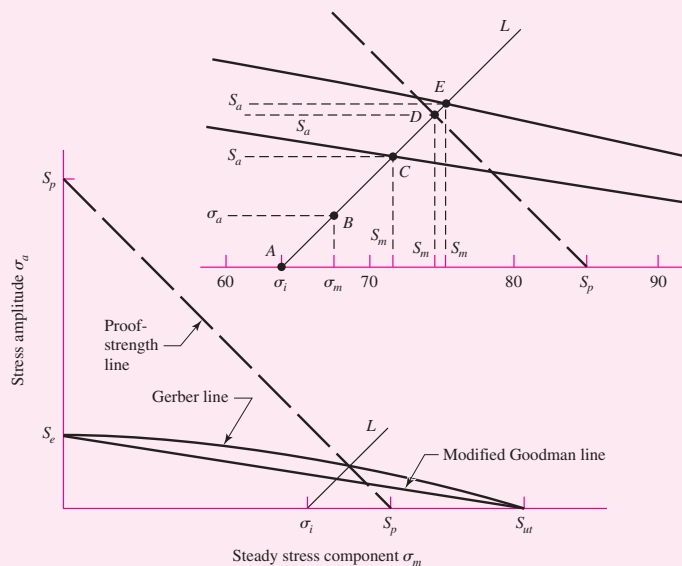
Answer
$$n_0 = \frac{F_i}{P(1 - C)} = \frac{14.4}{5(1 - 0.280)} = 4.00$$

If the force P gets too large, the joint will separate and the bolt will take the entire load. This factor guards against that event.

For the remaining factors, refer to Fig. 8–22. This diagram contains the modified Goodman line, the Gerber line, the proof-strength line, and the load line. The intersection

Figure 8–22

Designer's fatigue diagram for preloaded bolts, drawn to scale, showing the modified Goodman line, the Gerber line, and the Langer proof-strength line, with an exploded view of the area of interest. The strengths used are $S_p = 85$ kpsi, $S_e = 18.6$ kpsi, and $S_{ut} = 120$ kpsi. The coordinates are A , $\sigma_i = 63.72$ kpsi; B , $\sigma_a = 3.10$ kpsi, $\sigma_m = 66.82$ kpsi; C , $S_a = 7.55$ kpsi, $S_m = 71.29$ kpsi; D , $S_a = 10.64$ kpsi, $S_m = 74.36$ kpsi; E , $S_a = 11.32$ kpsi, $S_m = 75.04$ kpsi.



of the load line L with the respective failure lines at points C , D , and E defines a set of strengths S_a and S_m at each intersection. Point B represents the stress state σ_a , σ_m . Point A is the preload stress σ_i . Therefore the load line begins at A and makes an angle having a unit slope. This angle is 45° only when both stress axes have the same scale.

The factors of safety are found by dividing the distances AC , AD , and AE by the distance AB . Note that this is the same as dividing S_a for each theory by σ_a .

The quantities shown in the caption of Fig. 8–22 are obtained as follows:

Point A

$$\sigma_i = \frac{F_i}{A_t} = \frac{14.4}{0.226} = 63.72 \text{ kpsi}$$

Point B

$$\sigma_a = \frac{CP}{2A_t} = \frac{0.280(5)}{2(0.226)} = 3.10 \text{ kpsi}$$

$$\sigma_m = \sigma_a + \sigma_i = 3.10 + 63.72 = 66.82 \text{ kpsi}$$

Point C

This is the modified Goodman criteria. From Table 8–17, we find $S_e = 18.6$ kpsi. Then, using Eq. (8–45), the factor of safety is found to be

$$\text{Answer } n_f = \frac{S_e(S_{ut} - \sigma_i)}{\sigma_a(S_{ut} + S_e)} = \frac{18.6(120 - 63.72)}{3.10(120 + 18.6)} = 2.44$$

Point D

This is on the proof-strength line where

$$S_m + S_a = S_p \quad (1)$$

In addition, the horizontal projection of the load line AD is

$$S_m = \sigma_i + S_a \quad (2)$$

Solving Eqs. (1) and (2) simultaneously results in

$$S_a = \frac{S_p - \sigma_i}{2} = \frac{85 - 63.72}{2} = 10.64 \text{ kpsi}$$

The factor of safety resulting from this is

$$\text{Answer } n_p = \frac{S_a}{\sigma_a} = \frac{10.64}{3.10} = 3.43$$

which, of course, is identical to the result previously obtained by using Eq. (8–29).

A similar analysis of a fatigue diagram could have been done using yield strength instead of proof strength. Though the two strengths are somewhat related, proof strength is a much better and more positive indicator of a fully loaded bolt than is the yield strength. It is also worth remembering that proof-strength values are specified in design codes; yield strengths are not.

We found $n_f = 2.44$ on the basis of fatigue and the modified Goodman line, and $n_p = 3.43$ on the basis of proof strength. Thus the danger of failure is by fatigue, not by overproof loading. These two factors should always be compared to determine where the greatest danger lies.

Point E

For the Gerber criterion, from Eq. (8-46), the safety factor is

$$\begin{aligned} \text{Answer } n_f &= \frac{1}{2\sigma_a S_e} \left[S_{ut} \sqrt{S_{ut}^2 + 4S_e(S_e + \sigma_i)} - S_{ut}^2 - 2\sigma_i S_e \right] \\ &= \frac{1}{2(3.10)(18.6)} \left[120 \sqrt{120^2 + 4(18.6)(18.6 + 63.72)} - 120^2 - 2(63.72)(18.6) \right] \\ &= 3.65 \end{aligned}$$

which is greater than $n_p = 3.43$ and contradicts the conclusion earlier that the danger of failure is fatigue. Figure 8-22 clearly shows the conflict where point *D* lies between points *C* and *E*. Again, the conservative nature of the Goodman criterion explains the discrepancy and the designer must form his or her own conclusion.

8-12 Bolted and Riveted Joints Loaded in Shear¹⁰

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

Figure 8-23*a* shows a riveted connection loaded in shear. Let us now study the various means by which this connection might fail.

Figure 8-23*b* shows a failure by bending of the rivet or of the riveted members. The bending moment is approximately $M = Ft/2$, where F is the shearing force and t is the grip of the rivet, that is, the total thickness of the connected parts. The bending stress in the members or in the rivet is, neglecting stress concentration,

$$\sigma = \frac{M}{I/c} \quad (8-52)$$

where I/c is the section modulus for the weakest member or for the rivet or rivets, depending upon which stress is to be found. The calculation of the bending stress in this manner is an assumption, because we do not know exactly how the load is distributed to the rivet or the relative deformations of the rivet and the members. Although this equation can be used to determine the bending stress, it is seldom used in design; instead its effect is compensated for by an increase in the factor of safety.

In Fig. 8-23*c* failure of the rivet by pure shear is shown; the stress in the rivet is

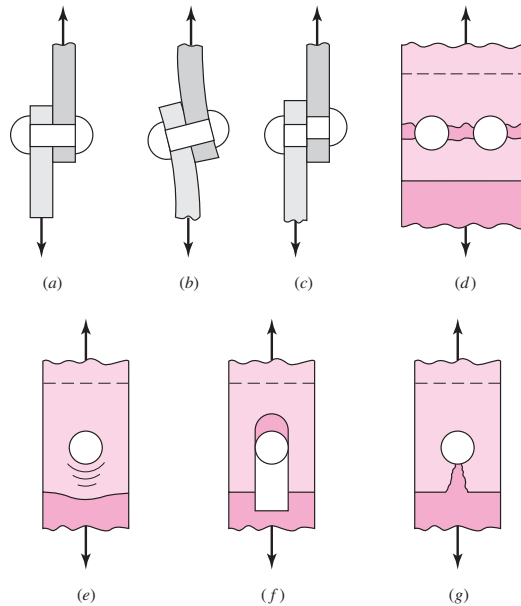
$$\tau = \frac{F}{A} \quad (8-53)$$

where A is the cross-sectional area of all the rivets in the group. It may be noted that it is standard practice in structural design to use the nominal diameter of the rivet rather than the diameter of the hole, even though a hot-driven rivet expands and nearly fills up the hole.

¹⁰The design of bolted and riveted connections for boilers, bridges, buildings, and other structures in which danger to human life is involved is strictly governed by various construction codes. When designing these structures, the engineer should refer to the *American Institute of Steel Construction Handbook*, the American Railway Engineering Association specifications, or the Boiler Construction Code of the American Society of Mechanical Engineers.

Figure 8-23

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



Rupture of one of the connected members or plates by pure tension is illustrated in Fig. 8-23d. The tensile stress is

$$\sigma = \frac{F}{A} \quad (8-54)$$

where A is the net area of the plate, that is, the area reduced by an amount equal to the area of all the rivet holes. For brittle materials and static loads and for either ductile or brittle materials loaded in fatigue, the stress-concentration effects must be included. It is true that the use of a bolt with an initial preload and, sometimes, a rivet will place the area around the hole in compression and thus tend to nullify the effects of stress concentration, but unless definite steps are taken to ensure that the preload does not relax, it is on the conservative side to design as if the full stress-concentration effect were present. The stress-concentration effects are not considered in structural design, because the loads are static and the materials ductile.

In calculating the area for Eq. (8-54), the designer should, of course, use the combination of rivet or bolt holes that gives the smallest area.

Figure 8-23e illustrates a failure by crushing of the rivet or plate. Calculation of this stress, which is usually called a *bearing stress*, is complicated by the distribution of the load on the cylindrical surface of the rivet. The exact values of the forces acting upon the rivet are unknown, and so it is customary to assume that the components of these forces are uniformly distributed over the projected contact area of the rivet. This gives for the stress

$$\sigma = \frac{F}{A} \quad (8-55)$$

where the projected area for a single rivet is $A = td$. Here, t is the thickness of the thinnest plate and d is the rivet or bolt diameter.

Edge shearing, or tearing, of the margin is shown in Fig. 8–23*f* and *g*, respectively. In structural practice this failure is avoided by spacing the rivets at least $1\frac{1}{2}$ diameters away from the edge. Bolted connections usually are spaced an even greater distance than this for satisfactory appearance, and hence this type of failure may usually be neglected.

In a rivet joint, the rivets all share the load in shear, bearing in the rivet, bearing in the member, and shear in the rivet. Other failures are participated in by only some of the joint. In a bolted joint, shear is taken by clamping friction, and bearing does not exist. When bolt preload is lost, one bolt begins to carry the shear and bearing until yielding slowly brings other fasteners in to share the shear and bearing. Finally, all participate, and this is the basis of most bolted-joint analysis if loss of bolt preload is complete. The usual analysis involves

- Bearing in the bolt (all bolts participate)
- Bearing in members (all holes participate)
- Shear of bolt (all bolts participate eventually)
- Distinguishing between thread and shank shear
- Edge shearing and tearing of member (edge bolts participate)
- Tensile yielding of member across bolt holes
- Checking member capacity

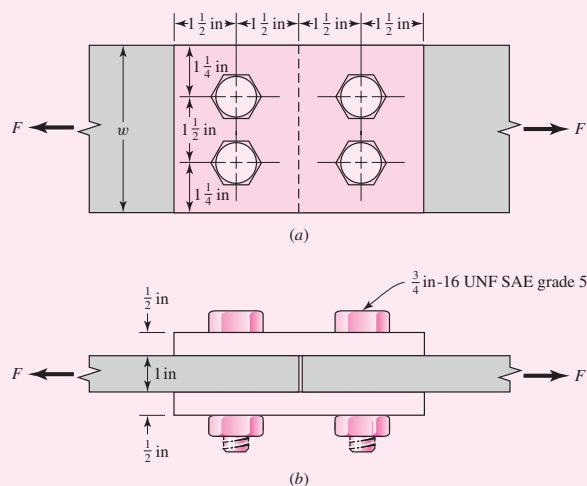
EXAMPLE 8–6

Two 1- by 4-in 1018 cold-rolled steel bars are butt-spliced with two $\frac{1}{2}$ - by 4-in 1018 cold-rolled splice plates using four $\frac{3}{4}$ -in-16 UNF grade 5 bolts as depicted in Fig. 8–24. For a design factor of $n_d = 1.5$ estimate the static load F that can be carried if the bolts lose preload.

Solution

From Table A–20, minimum strengths of $S_y = 54$ kpsi and $S_{ut} = 64$ kpsi are found for the members, and from Table 8–9 minimum strengths of $S_p = 85$ kpsi and $S_{ut} = 120$ kpsi for the bolts are found.

Figure 8–24



$F/2$ is transmitted by each of the splice plates, but since the areas of the splice plates are half those of the center bars, the stresses associated with the plates are the same. So for stresses associated with the plates, the force and areas used will be those of the center plates.

Bearing in bolts, all bolts loaded:

$$\sigma = \frac{F}{2td} = \frac{S_p}{n_d}$$

$$F = \frac{2tdS_p}{n_d} = \frac{2(1)(\frac{3}{4})85}{1.5} = 85 \text{ kip}$$

Bearing in members, all bolts active:

$$\sigma = \frac{F}{2td} = \frac{(S_y)_{\text{mem}}}{n_d}$$

$$F = \frac{2td(S_y)_{\text{mem}}}{n_d} = \frac{2(1)(\frac{3}{4})54}{1.5} = 54 \text{ kip}$$

Shear of bolt, all bolts active: If the bolt threads do not extend into the shear planes for four shanks:

$$\tau = \frac{F}{4\pi d^2/4} = 0.577 \frac{S_p}{n_d}$$

$$F = 0.577\pi d^2 \frac{S_p}{n_d} = 0.577\pi(0.75)^2 \frac{85}{1.5} = 57.8 \text{ kip}$$

If the bolt threads extend into a shear plane:

$$\tau = \frac{F}{4A_r} = 0.577 \frac{S_p}{n_d}$$

$$F = \frac{0.577(4)A_r S_p}{n_d} = \frac{0.577(4)(0.351)(85)}{1.5} = 45.9 \text{ kip}$$

Edge shearing of member at two margin bolts: From Fig. 8–25,

$$\tau = \frac{F}{4at} = \frac{0.577(S_y)_{\text{mem}}}{n_d}$$

$$F = \frac{4at(0.577(S_y)_{\text{mem}})}{n_d} = \frac{4(1.125)(1)(0.577)(54)}{1.5} = 93.5 \text{ kip}$$

Tensile yielding of members across bolt holes:

$$\sigma = \frac{F}{[4 - 2(\frac{3}{4})]t} = \frac{(S_y)_{\text{mem}}}{n_d}$$

$$F = \frac{[4 - 2(\frac{3}{4})]t(S_y)_{\text{mem}}}{n_d} = \frac{[4 - 2(\frac{3}{4})](1)(54)}{1.5} = 90 \text{ kip}$$

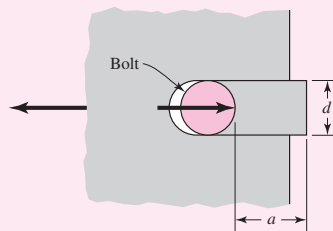
Member yield:

$$F = \frac{wt(S_y)_{\text{mem}}}{n_d} = \frac{4(1)54}{1.5} = 144 \text{ kip}$$

On the basis of bolt shear, the limiting value of the force is 45.9 kip, assuming the threads extend into a shear plane. However, it would be poor design to allow the threads to extend into a shear plane. So, assuming a *good* design based on bolt shear, the limiting value of the force is 57.8 kip. For the members, the bearing stress limits the load to 54 kip.

Figure 8-25

Edge shearing of member.



Shear Joints with Eccentric Loading

In the previous example, the load distributed equally to the bolts since the load acted along a line of symmetry of the fasteners. The analysis of a shear joint undergoing eccentric loading requires locating the center of relative motion between the two members. In Fig. 8-26 let A_1 to A_5 be the respective cross-sectional areas of a group of five pins, or hot-driven rivets, or tight-fitting shoulder bolts. Under this assumption the rotational pivot point lies at the centroid of the cross-sectional area pattern of the pins, rivets, or bolts. Using statics, we learn that the centroid G is located by the coordinates $\bar{x}$ and $\bar{y}$, where x_i and y_i are the distances to the i th area center:

$$\begin{aligned}\bar{x} &= \frac{A_1x_1 + A_2x_2 + A_3x_3 + A_4x_4 + A_5x_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\sum_1^n A_i x_i}{\sum_1^n A_i} \\ \bar{y} &= \frac{A_1y_1 + A_2y_2 + A_3y_3 + A_4y_4 + A_5y_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\sum_1^n A_i y_i}{\sum_1^n A_i}\end{aligned}\quad (8-56)$$

Figure 8-26

Centroid of pins, rivets, or bolts.

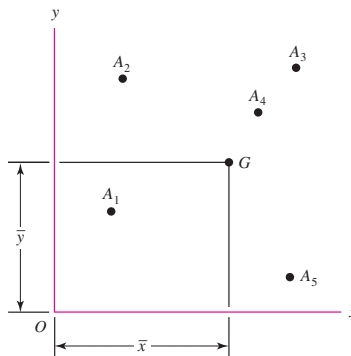
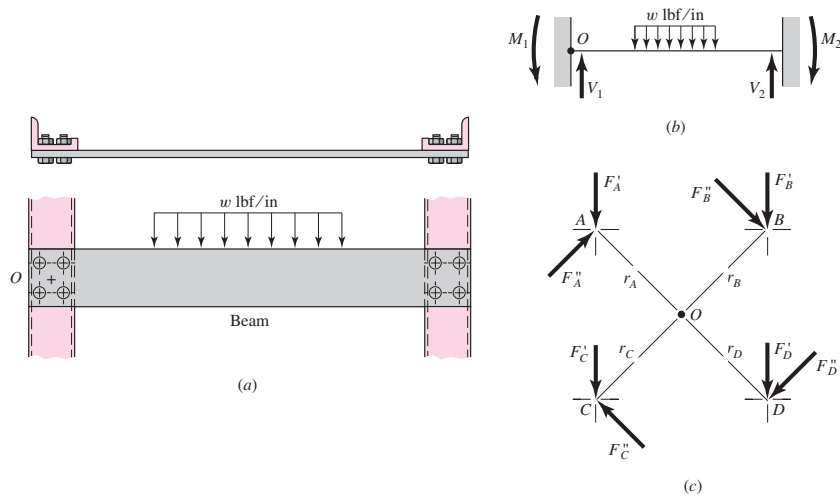


Figure 8-27

(a) Beam bolted at both ends with distributed load; (b) free-body diagram of beam; (c) enlarged view of bolt group centered at O showing primary and secondary resultant shear forces.



In many instances the centroid can be located by symmetry.

An example of eccentric loading of fasteners is shown in Fig. 8-27. This is a portion of a machine frame containing a beam subjected to the action of a bending load. In this case, the beam is fastened to vertical members at the ends with specially prepared load-sharing bolts. You will recognize the schematic representation in Fig. 8-27b as a statically indeterminate beam with both ends fixed and with moment and shear reactions at each end.

For convenience, the centers of the bolts at the left end of the beam are drawn to a larger scale in Fig. 8-27c. Point O represents the centroid of the group, and it is assumed in this example that all the bolts are of the same diameter. Note that the forces shown in Fig. 8-27c are the *resultant* forces acting on the pins with a net force and moment equal and opposite to the *reaction* loads V_1 and M_1 acting at O . The total load taken by each bolt will be calculated in three steps. In the first step the shear V_1 is divided equally among the bolts so that each bolt takes $F' = V_1/n$, where n refers to the number of bolts in the group and the force F' is called the *direct load*, or *primary shear*.

It is noted that an equal distribution of the direct load to the bolts assumes an absolutely rigid member. The arrangement of the bolts or the shape and size of the members sometimes justifies the use of another assumption as to the division of the load. The direct loads F'_n are shown as vectors on the loading diagram (Fig. 8-27c).

The *moment load*, or *secondary shear*, is the additional load on each bolt due to the moment M_1 . If r_A , r_B , r_C , etc., are the radial distances from the centroid to the center of each bolt, the moment and moment loads are related as follows:

$$M_1 = F'_A r_A + F'_B r_B + F'_C r_C + \cdots \quad (a)$$

where the F'' are the moment loads. The force taken by each bolt depends upon its radial distance from the centroid; that is, the bolt farthest from the centroid takes the greatest load, while the nearest bolt takes the smallest. We can therefore write

$$\frac{F''_A}{r_A} = \frac{F''_B}{r_B} = \frac{F''_C}{r_C} \quad (b)$$

$$F_n'' = \frac{M_1 r_n}{r_A^2 + r_B^2 + r_C^2 + \dots} \quad (8-57)$$

In the third step the direct and moment loads are added vectorially to obtain the resultant load on each bolt. Since all the bolts or rivets are usually the same size, only that bolt having the maximum load need be considered. When the maximum load is found, the strength may be determined by using the various methods already described.

Shown in Fig. 8-28 is a 15- by 200-mm rectangular steel bar cantilevered to a 250-mm steel channel using four tightly fitted bolts located at *A*, *B*, *C*, and *D*.

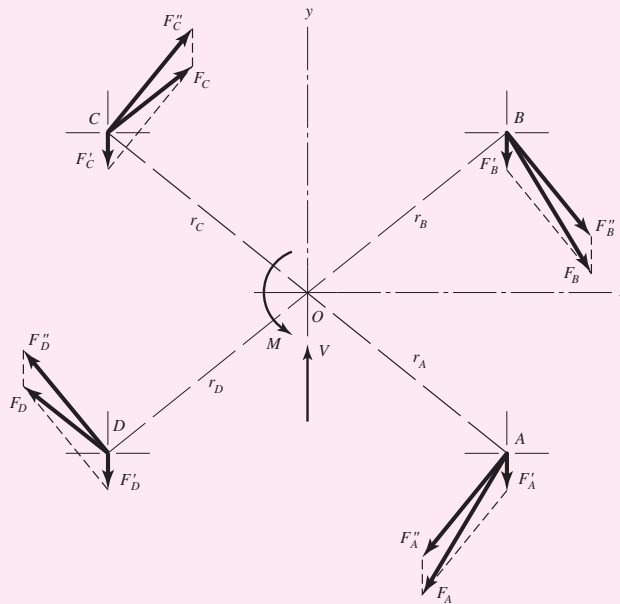
- The resultant load on each bolt
- The maximum shear stress in each bolt
- The maximum bearing stress
- The critical bending stress in the bar

(a) Point O , the centroid of the bolt group in Fig. 8–28, is found by symmetry. If a free-body diagram of the beam were constructed, the shear reaction V would pass through O and the moment reactions M would be about O . These reactions are

In Fig. 8-29, the bolt group has been drawn to a larger scale and the reactions are shown. The distance from the centroid to the center of each bolt is

Sandvik EX2015 Page 53
CMS Cepcor v. Sandvik IPR2024-01381

Figure 8-29



The primary shear load per bolt is

$$F' = \frac{V}{n} = \frac{16}{4} = 4 \text{ kN}$$

Since the secondary shear forces are equal, Eq. (8-57) becomes

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

The primary and secondary shear forces are plotted to scale in Fig. 8-29 and the resultants obtained by using the parallelogram rule. The magnitudes are found by measurement (or analysis) to be

Answer $F_A = F_B = 21.0 \text{ kN}$

Answer $F_C = F_D = 14.8 \text{ kN}$

(b) Bolts A and B are critical because they carry the largest shear load. Does this shear act on the threaded portion of the bolt, or on the unthreaded portion? The bolt length will be 25 mm plus the height of the nut plus about 2 mm for a washer. Table A-31 gives the nut height as 14.8 mm. Including two threads beyond the nut, this adds up to a length of 43.8 mm, and so a bolt 46 mm long will be needed. From Eq. (8-14) we compute the thread length as $L_T = 38 \text{ mm}$. Thus the unthreaded portion of the bolt is $46 - 38 = 8 \text{ mm}$ long. This is less than the 15 mm for the plate in Fig. 8-28, and so the bolt will tend to shear across its minor diameter. Therefore the shear-stress area is $A_s = 144 \text{ mm}^2$, and so the shear stress is

Answer
$$\tau = \frac{F}{A_s} = -\frac{21.0(10)^3}{144} = 146 \text{ MPa}$$

(c) The channel is thinner than the bar, and so the largest bearing stress is due to the pressing of the bolt against the channel web. The bearing area is $A_b = td = 10(16) = 160 \text{ mm}^2$. Thus the bearing stress is

Answer
$$\sigma = -\frac{F}{A_b} = -\frac{21.0(10)^3}{160} = -131 \text{ MPa}$$

(d) The critical bending stress in the bar is assumed to occur in a section parallel to the y axis and through bolts A and B . At this section the bending moment is

$$M = 16(300 + 50) = 5600 \text{ N} \cdot \text{m}$$

The second moment of area through this section is obtained by the use of the transfer formula, as follows:

$$\begin{aligned} I &= I_{\text{bar}} - 2(I_{\text{holes}} + \bar{d}^2 A) \\ &= \frac{15(200)^3}{12} - 2 \left[\frac{15(16)^3}{12} + (60)^2(15)(16) \right] = 8.26(10)^6 \text{ mm}^4 \end{aligned}$$

Then

Answer
$$\sigma = \frac{Mc}{I} = \frac{5600(100)}{8.26(10)^6} (10)^3 = 67.8 \text{ MPa}$$

PROBLEMS

- 8-1** A power screw is 25 mm in diameter and has a thread pitch of 5 mm.
 (a) Find the thread depth, the thread width, the mean and root diameters, and the lead, provided square threads are used.
 (b) Repeat part (a) for Acme threads.

- 8-2** Using the information in the footnote of Table 8-1, show that the tensile-stress area is

$$A_t = \frac{\pi}{4} (d - 0.938 \text{ } 194p)^2$$

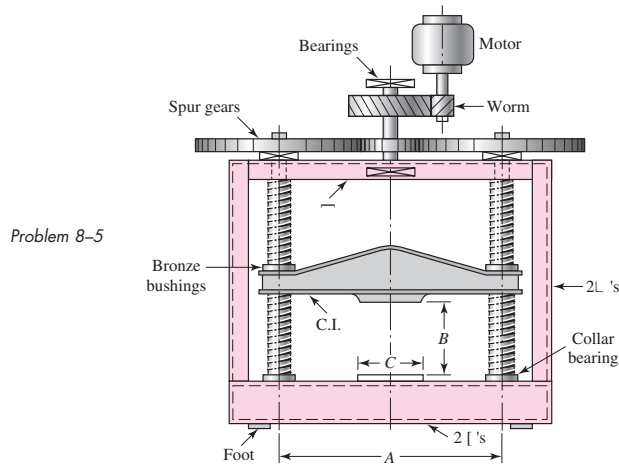
- 8-3** Show that for zero collar friction the efficiency of a square-thread screw is given by the equation

$$e = \tan \lambda \frac{1 - f \tan \lambda}{\tan \lambda + f}$$

Plot a curve of the efficiency for lead angles up to 45° . Use $f = 0.08$.

- 8-4** A single-threaded 25-mm power screw is 25 mm in diameter with a pitch of 5 mm. A vertical load on the screw reaches a maximum of 5 kN. The coefficients of friction are 0.06 for the collar and 0.09 for the threads. The frictional diameter of the collar is 45 mm. Find the overall efficiency and the torque to “raise” and “lower” the load.

- 8-5** The machine shown in the figure can be used for a tension test but not for a compression test. Why? Can both screws have the same hand?

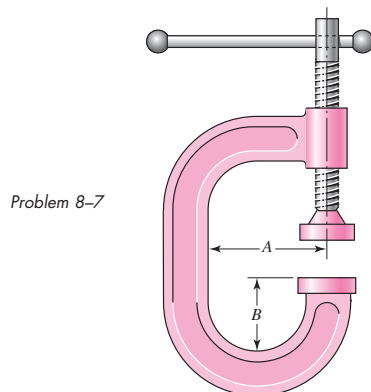


8-6 The press shown for Prob. 8-5 has a rated load of 5000 lbf. The twin screws have Acme threads, a diameter of 2 in, and a pitch of $\frac{1}{4}$ in. Coefficients of friction are 0.05 for the threads and 0.08 for the collar bearings. Collar diameters are 3.5 in. The gears have an efficiency of 95 percent and a speed ratio of 60:1. A slip clutch, on the motor shaft, prevents overloading. The full-load motor speed is 1720 rev/min.

- When the motor is turned on, how fast will the press head move?
- What should be the horsepower rating of the motor?

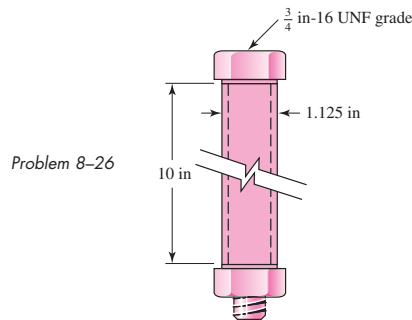
8-7 A screw clamp similar to the one shown in the figure has a handle with diameter $\frac{3}{8}$ in made of cold-drawn AISI 1006 steel. The overall length is 4.25 in. The screw is $\frac{3}{4}$ in-10 UNC and is 8 in long, overall. Distance A is 3 in. The clamp will accommodate parts up to 6 in high.

- What screw torque will cause the handle to bend permanently?
- What clamping force will the answer to part (a) cause if the collar friction is neglected and if the thread friction is 0.15?
- What clamping force will cause the screw to buckle?
- Are there any other stresses or possible failures to be checked?



- 8-8** The C clamp shown in the figure for Prob. 8-7 uses a $\frac{3}{4}$ -in-6 Acme thread. The frictional coefficients are 0.15 for the threads and for the collar. The collar, which in this case is the anvil striker's swivel joint, has a friction diameter of 1 in. Calculations are to be based on a maximum force of 8 lbf applied to the handle at a radius of $3\frac{1}{2}$ in from the screw centerline. Find the clamping force.
- 8-9** Find the power required to drive a 1.5-in power screw having double square threads with a pitch of $\frac{1}{4}$ in. The nut is to move at a velocity of 2 in/s and move a load of $F = 2.2$ kips. The frictional coefficients are 0.10 for the threads and 0.15 for the collar. The frictional diameter of the collar is 2.25 in.
- 8-10** A single square-thread power screw has an input power of 3 kW at a speed of 1 rev/s. The screw has a diameter of 40 mm and a pitch of 8 mm. The frictional coefficients are 0.14 for the threads and 0.09 for the collar, with a collar friction radius of 50 mm. Find the axial resisting load F and the combined efficiency of the screw and collar.
- 8-11** An M14 $\times$ 2 hex-head bolt with a nut is used to clamp together two 15-mm steel plates.
 (a) Determine a suitable length for the bolt, rounded up to the nearest 5 mm.
 (b) Determine the bolt stiffness.
 (c) Determine the stiffness of the members.
- 8-12** Repeat Prob. 8-11 with the addition of one 14R metric plain washer under the nut.
- 8-13** Repeat Prob. 8-11 with one of the plates having a threaded hole to eliminate the nut.
- 8-14** A 2-in steel plate and a 1-in cast-iron plate are compressed with one bolt and nut. The bolt is $\frac{1}{2}$ -in-13 UNC.
 (a) Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.
 (b) Determine the bolt stiffness.
 (c) Determine the stiffness of the members.
- 8-15** Repeat Prob. 8-14 with the addition of one $\frac{1}{2}$ -N American Standard plain washer under the head of the bolt, and another identical washer under the nut.
- 8-16** Repeat Prob. 8-14 with the cast-iron plate having a threaded hole to eliminate the nut.
- 8-17** Two identical aluminum plates are each 2 in thick, and are compressed with one bolt and nut. Washers are used under the head of the bolt and under the nut.
 Washer properties: steel; ID = 0.531 in; OD = 1.062 in; thickness = 0.095 in
 Nut properties: steel; height = $\frac{7}{16}$ in
 Bolt properties: $\frac{1}{2}$ -in-13 UNC grade 8
 Plate properties: aluminum; $E = 10.3$ Mpsi; $S_u = 47$ kpsi; $S_y = 25$ kpsi
 (a) Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.
 (b) Determine the bolt stiffness.
 (c) Determine the stiffness of the members.
- 8-18** Repeat Prob. 8-17 with no washer under the head of the bolt, and two washers stacked under the nut.
- 8-19** A 30-mm thick AISI 1020 steel plate is sandwiched between two 10-mm thick 2024-T3 aluminum plates and compressed with a bolt and nut with no washers. The bolt is M10 $\times$ 1.5, property class 5.8.
 (a) Determine a suitable length for the bolt, rounded up to the nearest 5 mm.
 (b) Determine the bolt stiffness.
 (c) Determine the stiffness of the members.

- 8-20** Repeat Prob. 8-19 with the bottom aluminum plate replaced by one that is 20 mm thick.
- 8-21** Repeat Prob. 8-19 with the bottom aluminum plate having a threaded hole to eliminate the nut.
- 8-22** Two 20-mm steel plates are to be clamped together with a bolt and nut. Specify a bolt to provide a joint constant C between 0.2 and 0.3.
- 8-23** A 2-in steel plate and a 1-in cast-iron plate are to be compressed with one bolt and nut. Specify a bolt to provide a joint constant C between 0.2 and 0.3.
- 8-24** An aluminum bracket with a $\frac{1}{2}$ -in thick flange is to be clamped to a steel column with a $\frac{3}{4}$ -in wall thickness. A cap screw passes through a hole in the bracket flange, and threads into a tapped hole through the column wall. Specify a cap screw to provide a joint constant C between 0.2 and 0.3.
- 8-25** An M14 $\times$ 2 hex-head bolt with a nut is used to clamp together two 20-mm steel plates. Compare the results of finding the overall member stiffness by use of Eqs. (8-20), (8-22), and (8-23).
- 8-26** A $\frac{3}{4}$ -in-16 UNF series SAE grade 5 bolt has a $\frac{3}{4}$ -in ID steel tube 10 in long, clamped between washer faces of bolt and nut by turning the nut snug and adding one-third of a turn. The tube OD is the washer-face diameter $d_w = 1.5d = 1.5(0.75) = 1.125$ in = OD.



- (a) Determine the bolt stiffness, the tube stiffness, and the joint constant C .
- (b) When the one-third turn-of-nut is applied, what is the initial tension F_i in the bolt?

- 8-27** From your experience with Prob. 8-26, generalize your solution to develop a turn-of-nut equation

$$N_t = \frac{\theta}{360^\circ} = \left(\frac{k_b + k_m}{k_b k_m} \right) F_i N$$

where N_t = turn of the nut from snug tight

θ = turn of the nut in degrees

N = number of thread/in ($1/p$ where p is pitch)

F_i = initial preload

k_b, k_m = spring rates of the bolt and members, respectively

Use this equation to find the relation between torque-wrench setting T and turn-of-nut N_t . ("Snug tight" means the joint has been tightened to perhaps half the intended preload to flatten asperities on the washer faces and the members. Then the nut is loosened and retightened finger tight, and the nut is rotated the number of degrees indicated by the equation. Properly done, the result is competitive with torque wrenching.)

8-28 RB&W¹¹ recommends turn-of-nut from snug fit to preload as follows: 1/3 turn for bolt grips of 1–4 diameters, 1/2 turn for bolt grips 4–8 diameters, and 2/3 turn for grips of 8–12 diameters. These recommendations are for structural steel fabrication (permanent joints), producing preloads of 100 percent of proof strength and beyond. Machinery fabricators with fatigue loadings and possible joint disassembly have much smaller turns-of-nut. The RB&W recommendation enters the nonlinear plastic deformation zone.

For Ex. 8-4, use Eq. (8-27) with $K = 0.2$ to estimate the torque necessary to establish the desired preload. Then, using the results from Prob. 8-27, determine the turn of the nut in degrees. How does this compare with the RB&W recommendations?

8-29 For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 3$ Mlbf/in and the stiffness of the members is $k_m = 12$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use $\frac{1}{2}$ in-13 UNC grade 8 bolts with rolled threads. Assume the bolts are preloaded to 75 percent of the proof load.

- Determine the yielding factor of safety.
- Determine the overload factor of safety.
- Determine the factor of safety based on joint separation.

8-30 For the bolted assembly of Prob. 8-29, it is desired to find the range of torque that a mechanic could apply to initially preload the bolts without expecting failure once the joint is loaded. Assume a torque coefficient of $K = 0.2$.

- Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts.
- Determine the minimum bolt preload that can be applied while avoiding joint separation.
- Determine the value of torque in units of lbf · ft that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b).

8-31 For a bolted assembly with eight bolts, the stiffness of each bolt is $k_b = 1.0$ MN/mm and the stiffness of the members is $k_m = 2.6$ MN/mm per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. Assume the external load is equally distributed to all the bolts. It has been determined to use M6 × 1 class 5.8 bolts with rolled threads.

- Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without exceeding the proof strength of the bolts.
- Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without causing the members to come out of compression.

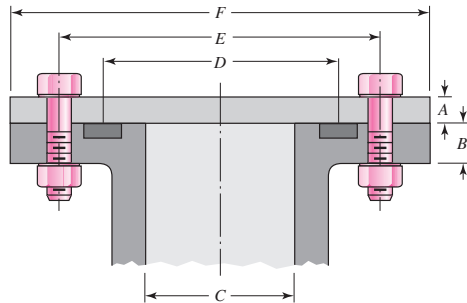
8-32 For a bolted assembly, the stiffness of each bolt is $k_b = 4$ Mlbf/in and the stiffness of the members is $k_m = 12$ Mlbf/in per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. A fluctuating external load is applied to the entire joint with $P_{\max} = 80$ kips and $P_{\min} = 20$ kips. Assume the load is equally distributed to all the bolts. It has been determined to use $\frac{1}{2}$ in-13 UNC grade 8 bolts with rolled threads.

- Determine the minimum number of bolts necessary to avoid yielding of the bolts.
- Determine the minimum number of bolts necessary to avoid joint separation.

8-33 to 8-36 The figure illustrates the connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific

¹¹Russell, Burdsall & Ward, Inc., Metal Forming Specialists, Mentor, Ohio.

problem assigned, select a suitable bolt length from the preferred sizes in Table A-17, then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 .



**8-37 to
8-40**

Repeat the requirements for the problem specified in the table if the bolts and nuts are replaced with cap screws that are threaded into tapped holes in the cast-iron cylinder.

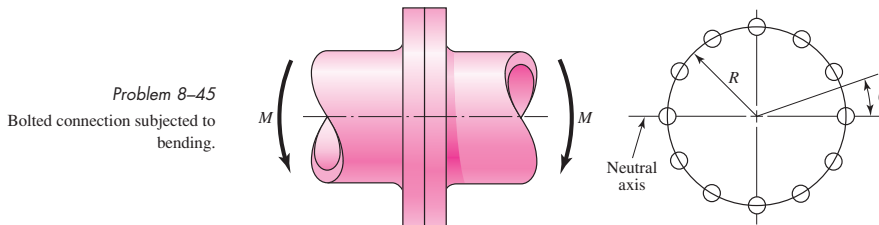
- Obtain a joint stiffness constant C between 0.2 and 0.3 to ensure most of the pressure load is carried by the members.
- The bolts may be reused, so the yielding factor of safety should be at least 1.1.
- The overload factor and the joint separation factor should allow for the pressure to exceed the expected pressure by 15 percent.

Problem Number	Originating Problem Number
8-41	8-33
8-42	8-34
8-43	8-35
8-44	8-36

8-45

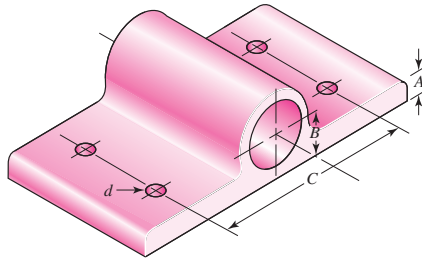
Bolts distributed about a bolt circle are often called upon to resist an external bending moment as shown in the figure. The external moment is 12 kip · in and the bolt circle has a diameter of 8 in. The neutral axis for bending is a diameter of the bolt circle. What needs to be determined is the most severe external load seen by a bolt in the assembly.

- View the effect of the bolts as placing a line load around the bolt circle whose intensity F'_b , in pounds per inch, varies linearly with the distance from the neutral axis according to the relation $F'_b = F'_{b,\max} R \sin \theta$. The load on any particular bolt can be viewed as the effect of the line load over the arc associated with the bolt. For example, there are 12 bolts shown in the figure. Thus each bolt load is assumed to be distributed on a 30° arc of the bolt circle. Under these conditions, what is the largest bolt load?
- View the largest load as the intensity $F'_{b,\max}$ multiplied by the arc length associated with each bolt and find the largest bolt load.
- Express the load on any bolt as $F = F_{\max} \sin \theta$, sum the moments due to all the bolts, and estimate the largest bolt load. Compare the results of these three approaches to decide how to attack such problems in the future.

**8-46**

The figure shows a cast-iron bearing block that is to be bolted to a steel ceiling joist and is to support a gravity load of 18 kN. Bolts used are M24 ISO 8.8 with coarse threads and with 4.6-mm-thick steel washers under the bolt head and nut. The joist flanges are 20 mm in thickness, and the dimension A , shown in the figure, is 20 mm. The modulus of elasticity of the bearing block is 135 GPa.

Problem 8-46



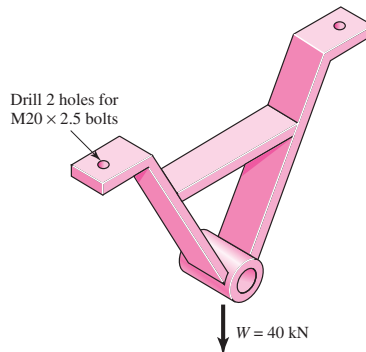
- Find the wrench torque required if the fasteners are lubricated during assembly and the joint is to be permanent.
- Determine the factors of safety guarding against yielding, overload, and joint separation.

8-47

The upside-down steel A frame shown in the figure is to be bolted to steel beams on the ceiling of a machine room using ISO grade 8.8 bolts. This frame is to support the 40-kN radial load as illustrated. The total bolt grip is 48 mm, which includes the thickness of the steel beam, the A-frame feet, and the steel washers used. The bolts are size M20 $\times$ 2.5.

- What tightening torque should be used if the connection is permanent and the fasteners are lubricated?
- Determine the factors of safety guarding against yielding, overload, and joint separation.

Problem 8-47

**8-48**

For the bolted assembly in Prob. 8-29, assume the external load is a repeated load. Determine the fatigue factor of safety for the bolts using the following failure criteria:

- Goodman.
- Gerber.
- ASME-elliptic.

8-49

For a bolted assembly with eight bolts, the stiffness of each bolt is $k_b = 1.0$ MN/mm and the stiffness of the members is $k_m = 2.6$ MN/mm per bolt. The bolts are preloaded to 75 percent of proof strength. Assume the external load is equally distributed to all the bolts. The bolts are M6 $\times$ 1 class 5.8 with rolled threads. A fluctuating external load is applied to the entire joint with $P_{\max} = 60$ kN and $P_{\min} = 20$ kN.

- Determine the yielding factor of safety.
- Determine the overload factor of safety.

- (c) Determine the factor of safety based on joint separation.
 (d) Determine the fatigue factor of safety using the Goodman criterion.

8-50

For the bolted assembly in Prob. 8-32, assume 10 bolts are used. Determine the fatigue factor of safety using the Goodman criterion.

**8-51 to
8-54**

For the pressure cylinder defined in the problem specified in the table, the gas pressure is cycled between zero and p_g . Determine the fatigue factor of safety for the bolts using the following failure criteria:

- (a) Goodman.
 (b) Gerber.
 (c) ASME-elliptic.

Problem Number	Originating Problem Number
8-51	8-33
8-52	8-34
8-53	8-35
8-54	8-36

**8-55 to
8-58**

For the pressure cylinder defined in the problem specified in the table, the gas pressure is cycled between p_g and $p_g/2$. Determine the fatigue factor of safety for the bolts using the Goodman criterion.

Problem Number	Originating Problem Number
8-55	8-33
8-56	8-34
8-57	8-35
8-58	8-36

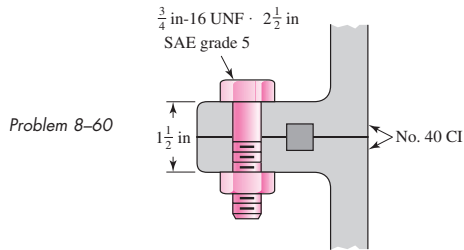
8-59

A 1-in-diameter hot-rolled AISI 1144 steel rod is hot-formed into an eyebolt similar to that shown in the figure for Prob. 3-122, with an inner 3-in-diameter eye. The threads are 1 in-12 UNF and are die-cut.

- (a) For a repeatedly applied load collinear with the thread axis, using the Gerber criterion, is fatigue failure more likely in the thread or in the eye?
 (b) What can be done to strengthen the bolt at the weaker location?
 (c) If the factor of safety guarding against a fatigue failure is $n_f = 2$, what repeatedly applied load can be applied to the eye?

8-60

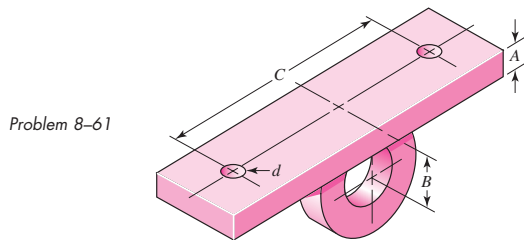
The section of the sealed joint shown in the figure is loaded by a force cycling between 4 and 6 kips. The members have $E = 16$ Mpsi. All bolts have been carefully preloaded to $F_i = 25$ kip each.



- Determine the yielding factor of safety.
- Determine the overload factor of safety.
- Determine the factor of safety based on joint separation.
- Determine the fatigue factor of safety using the Goodman criterion.

8-61

Suppose the welded steel bracket shown in the figure is bolted underneath a structural-steel ceiling beam to support a fluctuating vertical load imposed on it by a pin and yoke. The bolts are $\frac{1}{2}$ -in coarse-thread SAE grade 8, tightened to recommended preload for nonpermanent assembly. The stiffnesses have already been computed and are $k_b = 4$ Mlb/in and $k_m = 16$ Mlb/in.



- Assuming that the bolts, rather than the welds, govern the strength of this design, determine the safe repeated load that can be imposed on this assembly using the Goodman criterion with the load line in Fig. 8-20 and a fatigue design factor of 2.
- Compute the static load factors based on the load found in part (a).

8-62

Using the Gerber fatigue criterion and a fatigue-design factor of 2, determine the external repeated load P that a $1\frac{1}{4}$ -in SAE grade 5 coarse-thread bolt can take compared with that for a fine-thread bolt. The joint constants are $C = 0.30$ for coarse- and 0.32 for fine-thread bolts.

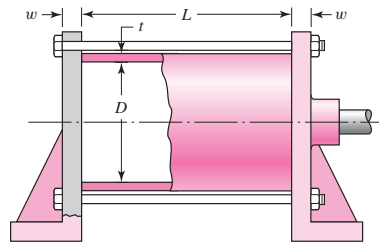
8-63

An M30 $\times$ 3.5 ISO 8.8 bolt is used in a joint at recommended preload, and the joint is subject to a repeated tensile fatigue load of $P = 65$ kN per bolt. The joint constant is $C = 0.28$. Find the static load factors and the factor of safety guarding against a fatigue failure based on the Gerber fatigue criterion.

8-64

The figure shows a fluid-pressure linear actuator (hydraulic cylinder) in which $D = 4$ in, $t = \frac{3}{8}$ in, $L = 12$ in, and $w = \frac{3}{4}$ in. Both brackets as well as the cylinder are of steel. The actuator has been designed for a working pressure of 2000 psi. Six $\frac{3}{8}$ -in SAE grade 5 coarse-thread bolts are used, tightened to 75 percent of proof load.

Problem 8-64



- Find the stiffnesses of the bolts and members, assuming that the entire cylinder is compressed uniformly and that the end brackets are perfectly rigid.
- Using the Gerber fatigue criterion, find the factor of safety guarding against a fatigue failure.
- What pressure would be required to cause total joint separation?

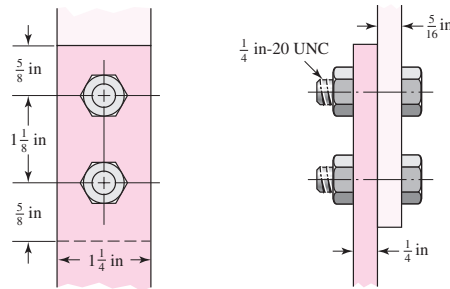
8-65

Using the Goodman fatigue criterion, repeat Prob. 8-64 with the working pressure cycling between 1200 psi and 2000 psi.

8-66

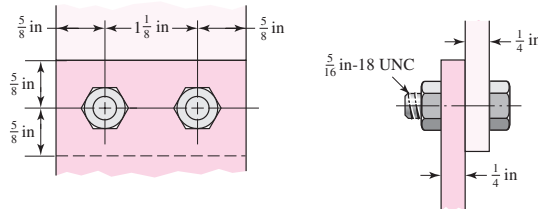
The figure shows a bolted lap joint that uses SAE grade 5 bolts. The members are made of cold-drawn AISI 1020 steel. Find the safe tensile shear load F that can be applied to this connection to provide a minimum factor of safety of 2 for the following failure modes: shear of bolts, bearing on bolts, bearing on members, and tension of members.

Problem 8-66

**8-67**

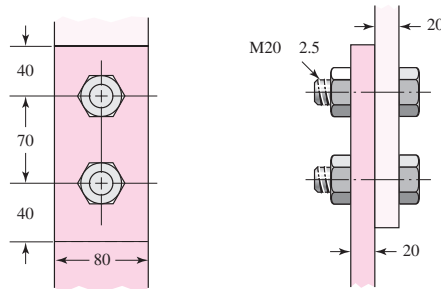
The bolted connection shown in the figure uses SAE grade 8 bolts. The members are hot-rolled AISI 1040 steel. A tensile shear load $F = 5000$ lbf is applied to the connection. Find the factor of safety for all possible modes of failure.

Problem 8-67

**8-68**

A bolted lap joint using ISO class 5.8 bolts and members made of cold-drawn SAE 1040 steel is shown in the figure. Find the tensile shear load F that can be applied to this connection to provide a minimum factor of safety of 2.5 for the following failure modes: shear of bolts, bearing on bolts, bearing on members, and tension of members.

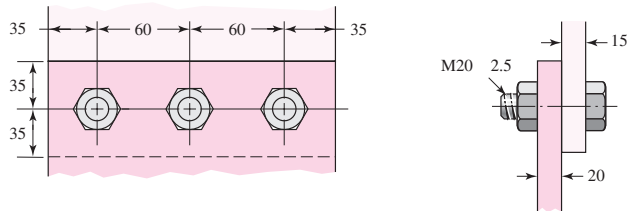
Problem 8-68
Dimensions in millimeters.



8-69

The bolted connection shown in the figure is subjected to a tensile shear load of 90 kN. The bolts are ISO class 5.8 and the material is cold-drawn AISI 1015 steel. Find the factor of safety of the connection for all possible modes of failure.

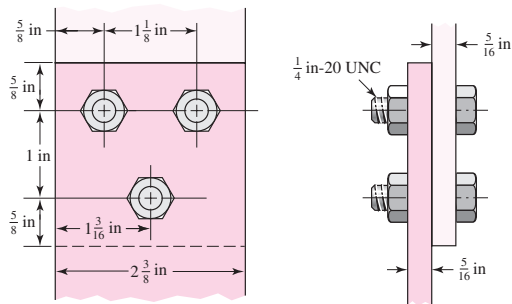
Problem 8-69
Dimensions in millimeters.



8-70

The figure shows a connection that employs three SAE grade 4 bolts. The tensile shear load on the joint is 5000 lbf. The members are cold-drawn bars of AISI 1020 steel. Find the factor of safety for each possible mode of failure.

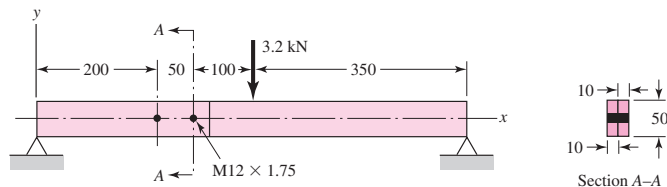
Problem 8-70



8-71

A beam is made up by bolting together two cold-drawn bars of AISI 1018 steel as a lap joint, as shown in the figure. The bolts used are ISO 5.8. Ignoring any twisting, determine the factor of safety of the connection.

Problem 8-71
Dimensions in millimeters.



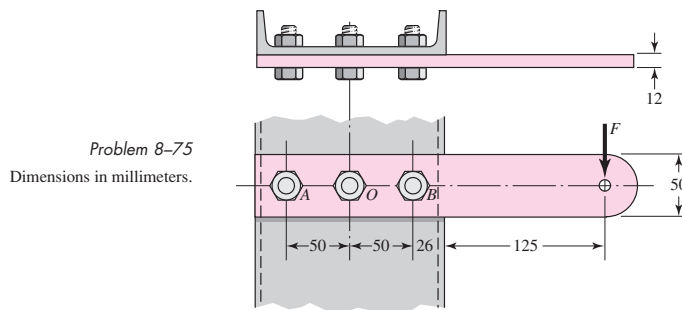
8-72 Standard design practice, as exhibited by the solutions to Probs. 8-66 to 8-70, is to assume that the bolts, or rivets, share the shear equally. For many situations, such an assumption may lead to an unsafe design. Consider the yoke bracket of Prob. 8-61, for example. Suppose this bracket is bolted to a wide-flange *column* with the centerline through the two bolts in the vertical direction. A vertical load through the yoke-pin hole at distance B from the column flange would place a shear load on the bolts as well as a tensile load. The tensile load comes about because the bracket tends to pry itself about the bottom corner, much like a claw hammer, exerting a large tensile load on the upper bolt. In addition, it is almost certain that both the spacing of the bolt holes and their diameters will be slightly different on the column flange from what they are on the yoke bracket. Thus, unless yielding occurs, only one of the bolts will take the shear load. The designer has no way of knowing which bolt this will be.

In this problem the bracket is 8 in long, $A = \frac{1}{2}$ in, $B = 3$ in, $C = 6$ in, and the column flange is $\frac{1}{2}$ in thick. The bolts are $\frac{1}{2}$ -in UNC SAE grade 4. The nuts are tightened to 75 percent of proof load. The vertical yoke-pin load is 2500 lbf. If the upper bolt takes all the shear load as well as the tensile load, how closely does the bolt stress approach the proof strength?

8-73 The bearing of Prob. 8-46 is bolted to a vertical surface and supports a horizontal shaft. The bolts used have coarse threads and are M20 ISO 5.8. The joint constant is $C = 0.25$, and the dimensions are $A = 20$ mm, $B = 50$ mm, and $C = 160$ mm. The bearing base is 240 mm long. The bearing load is 14 kN. If the bolts are tightened to 75 percent of proof load, will the bolt stress exceed the proof strength? Use worst-case loading, as discussed in Prob. 8-72.

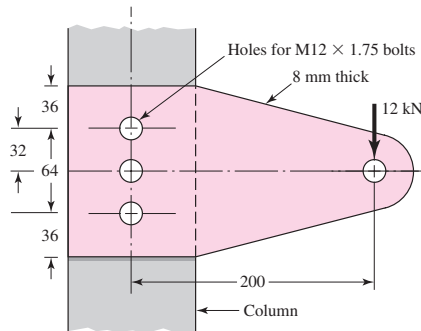
8-74 A split-ring clamp-type shaft collar such as is described in Prob. 5-67 must resist an axial load of 1000 lbf. Using a design factor of $n = 3$ and a coefficient of friction of 0.12, specify an SAE Grade 5 cap screw using fine threads. What wrench torque should be used if a lubricated screw is used?

8-75 A vertical channel 152×76 (see Table A-7) has a cantilever beam bolted to it as shown. The channel is hot-rolled AISI 1006 steel. The bar is of hot-rolled AISI 1015 steel. The shoulder bolts are M10 $\times$ 1.5 ISO 5.8. For a design factor of 2.0, find the safe force F that can be applied to the cantilever.

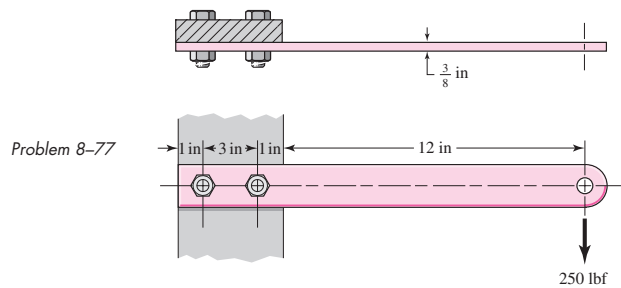


8-76 The cantilever bracket is bolted to a column with three M12 $\times$ 1.75 ISO 5.8 bolts. The bracket is made from AISI 1020 hot-rolled steel. Find the factors of safety for the following failure modes: shear of bolts, bearing of bolts, bearing of bracket, and bending of bracket.

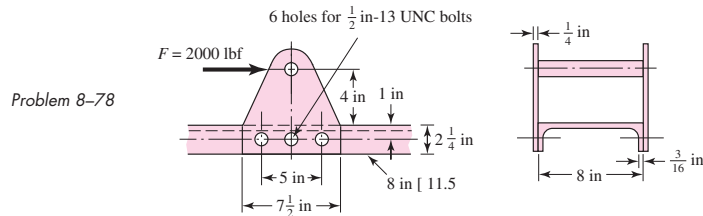
Problem 8-76
Dimensions in millimeters.



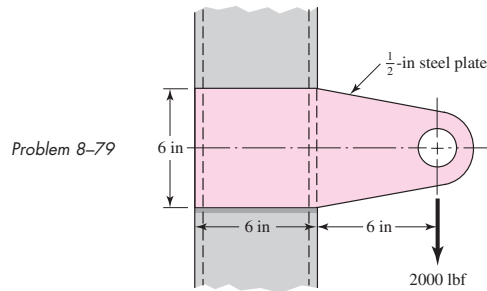
- 8-77** A $\frac{3}{8}$ - $\times$ 2-in AISI 1018 cold-drawn steel bar is cantilevered to support a static load of 250 lbf as illustrated. The bar is secured to the support using two $\frac{3}{8}$ in-16 UNC SAE grade 4 bolts. Find the factor of safety for the following modes of failure: shear of bolt, bearing on bolt, bearing on member, and strength of member.



- 8-78** The figure shows a welded fitting which has been tentatively designed to be bolted to a channel so as to transfer the 2000-lbf load into the channel. The channel and the two fitting plates are of hot-rolled stock having a minimum S_y of 42 kpsi. The fitting is to be bolted using six SAE grade 4 shoulder bolts. Check the strength of the design by computing the factor of safety for all possible modes of failure.



- 8-79** A cantilever is to be attached to the flat side of a 6-in, 13.0-lbf/in channel used as a column. The cantilever is to carry a load as shown in the figure. To a designer the choice of a bolt array is usually an a priori decision. Such decisions are made from a background of knowledge of the effectiveness of various patterns.



- (a) If two fasteners are used, should the array be arranged vertically, horizontally, or diagonally? How would you decide?
- (b) If three fasteners are used, should a linear or triangular array be used? For a triangular array, what should be the orientation of the triangle? How would you decide?

8-80

Using your experience with Prob. 8-79, specify an optimal bolt pattern for two bolts for the bracket in Prob. 8-79 and size the bolts.

8-81

Using your experience with Prob. 8-79, specify an optimal bolt pattern for three bolts for the bracket in Prob. 8-79 and size the bolts.

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