Fasteners (Advanced)

0.10 for the collar, which has a frictional diameter of 5 in. Find the maximum combined normal and shear stresses in the screw.

7-10. A $1\frac{1}{4}''$–6 square-thread power screw is required to raise and lower a load of 10,000 lbf. The frictional coefficients are 0.13 for the threads and for the collar. The collar has a frictional diameter of 2 in. Determine the values of both steady and varying components of shear stress in the screw.

7-11. A power screw is loaded eccentrically so that it is required to lift a load of 12,000 lbf, placing the screw in tension and also producing a constant bending moment of 7500 lbf-in. on the screw. The threads are $2''$–4 single-square. The coefficient of thread friction is 0.14. The screw is supported by tapered roller bearings so that the collar friction is negligible. The material is AISI 1042 carbon steel, heat-treated to a yield strength of 90 kpsi. Calculate the maximum von Mises stress, basing stress calculations on the average of the mean and root diameters.

7-4 THREADED FASTENERS

Fasteners are named according to how they are intended to be used rather than how they are actually employed in specific instances. If this basic fact is remembered, it will not be difficult to distinguish between a *screw* and a *bolt.*

If a product is designed so that its primary purpose is assembly into a tapped hole, it is a *screw.* Thus a screw is tightened by exerting torque on the *head.*

If a product is designed so that it is intended to be used with a nut, it is a *bolt.* A bolt is tightened by exerting torque on the *nut.*

A *stud* resembles a threaded rod; one end assembles into a tapped hole, the other end receives a nut.

It is the intent, rather than the actual use, which determines the name of a product. Thus it may be desirable on various occasions to use a drill through two sheets of steel, say, and join them using a machine screw and nut.

Space does not permit a complete tabulation of the dimensions of a large variety of threaded products, but Tables A-21 to A-24 show some of the sizes of bolts, screws, and nuts. Figure 7-9 also shows a variety of head forms available on standard *machine screws.* A machine screw is a small screw; though shown in Table A-12 in sizes up to $\frac{3}{4}$ in., it is usually available only in sizes of $\frac{3}{8}$ in. and under.

Cap screws are employed in sizes from $\frac{1}{4}$ in. up to and including $1\frac{1}{2}$ in., as shown in Table A-22. Figures 7-10 and 7-11*a* show four of the head forms most commonly used with cap screws. The heads in Fig. 7-10 require no wrench clearance.

There is no difference between a hexagon-head cap screw and a *finished* hexagon-head bolt, as an examination of Fig. 7-11*a* and *b* and Tables A-22 and A-23 will show. Of course, bolts are also made in *semifinished* and in *unfinished* forms which may be dimensionally different. The following characteristics should be noted from the figure: the chamfer on the head and on the body, the length of threads $2D + \frac{1}{4}$ in., the washer face under the head, and the fillet radius.

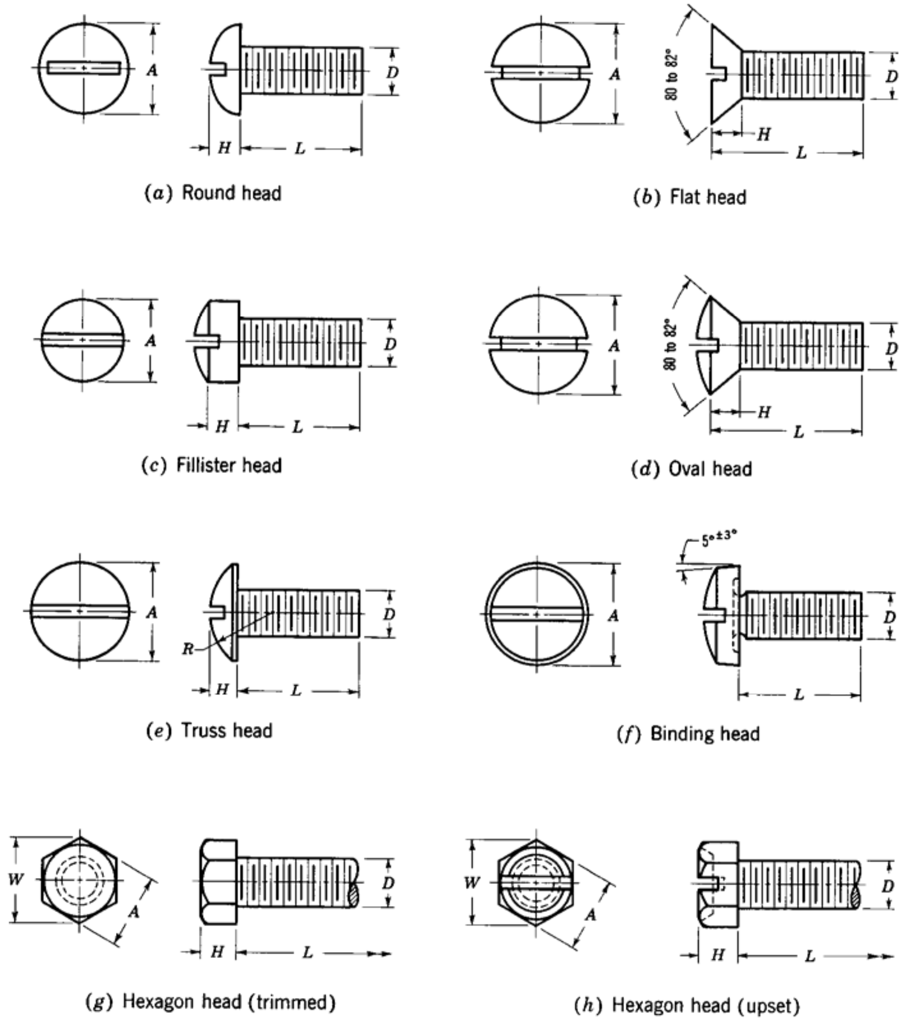


Fig. 7-9 Types of heads used on machine screws.

7-5 PRELOADING OF BOLTS

When a connection is desired which can be disassembled without destructive methods and which is strong enough to resist both external tensile loads and shear loads, or a combination of these, then the simple bolted joint using hardened washers is a good solution. Such a joint is illustrated in Fig. 7-12, in which the bolt has first been tightened to produce an initial tensile preload F_i, after which the external tensile load P and the external shear load F_s are applied. The effect of the preload is to place the parts in compression for better resistance to the external tensile load and to create friction between the parts to resist the shear load.

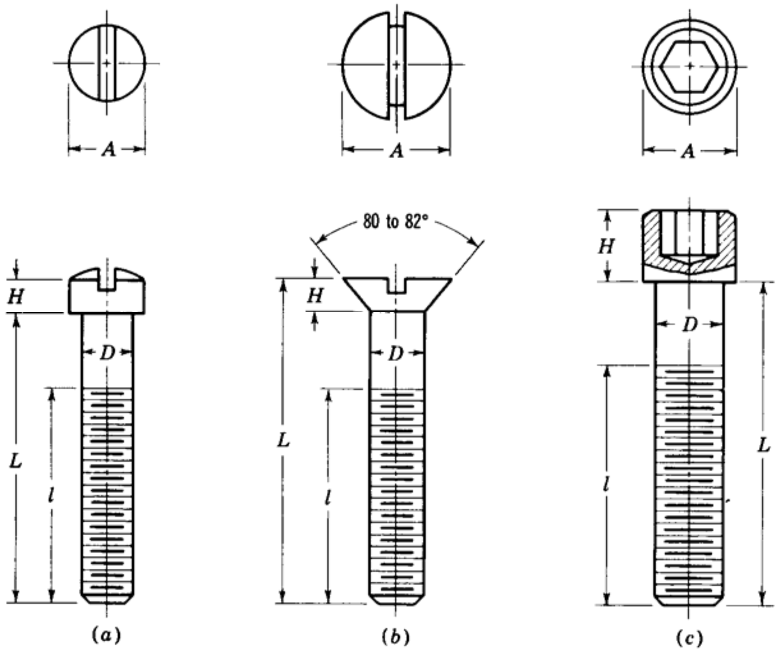


Fig. 7.10 Cap-screw heads; (a) Fillister head, (b) flat head, (c) hexagon socket head.

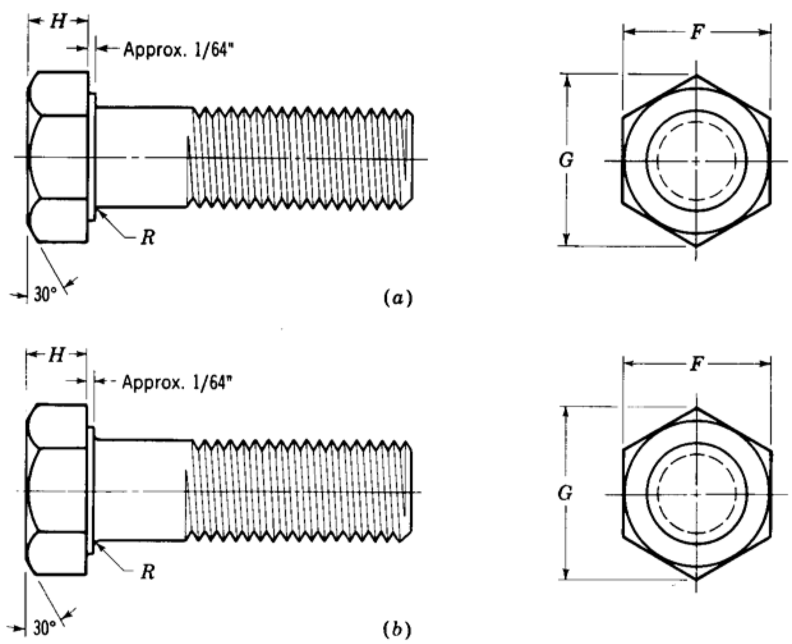


Fig. 7-11 (a) Hexagon-head cap screw; (b) hexagon-head bolt (finished).

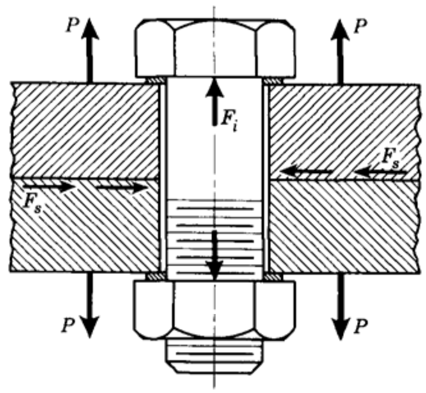


Fig. 7-12 A bolted connection.

The shear load does not affect the final bolt tension, and we shall neglect this load for the time being in order to study the effect of the external tensile load on the compression of the parts and the resultant bolt tension.

The *spring constant*, or *stiffness constant*, of an elastic member such as a bolt, as we learned in Chap. 3, is the ratio of the force applied to the member to the deflection produced by that force. The deflection of a bar in simple tension or compression was found to be

$$\delta = \frac{Fl}{AE} \tag{a}$$

where δ = deflection, in.
F = force, lbf
A = area, in.^{2}
E = modulus of elasticity, psi

Therefore the stiffness constant is

$$k = \frac{F}{\delta} = \frac{AE}{l} \tag{b}$$

and the units of k are lbf/in.

In finding the stiffness of a bolt, A is the area based on the nominal or major diameter because the effect of the threads is neglected. The grip l is the total thickness of the parts which have been fastened together. Note that this is somewhat less than the length of the bolt.

There may be more than two members included in the grip of the bolt. These act like compressive springs in series, and hence the the total spring rate of the *members* is

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

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 spreads out between the bolt head and nut and hence the area is not uniform. In many cases the geometry is such that this area can be determined. When it cannot, a safe approach might be to use a hollow cylinder for the members having a hole the same size as the bolt and an outside diameter three times the bolt diameter. If we use this assumption, and also assume that all materials included in the grip are the same, then the stiffness of the members from Eq. (*b*) is

$$k_m = \frac{2\pi d^2 E}{l} \tag{7-10}$$

Also, the stiffness of the bolt, from Eq. (*b*), is

$$k_b = \frac{\pi d^2 E}{4l} \tag{7-11}$$

In Eqs. (7-10) and (7-11) d is the nominal bolt diameter. Thus, if the bolt and members have the same modulus of elasticity, the members are eight times as stiff as the bolt with this assumption.

Let us now visualize a tension-loaded bolted connection. We use the following nomenclature:

P = total external load on bolted assembly
F_i = preload on bolt due to tightening and in existence before P is applied
P_b = portion of P taken by bolt
P_m = portion of P taken by members
F_b = resultant bolt load
F_m = resultant load on members

When the external load P is applied to the preloaded assembly, there is a change in the deformation of the bolt and also in the deformation of the connected members. The bolt, initially in tension, gets longer. This *increase* in deformation of the bolt is

$$\Delta\delta_b = \frac{P_b}{k_b} \tag{d}$$

The connected members have initial compression due to the preload. When the external load is applied, this compression will *decrease*. The decrease in deformation of the members is

$$\Delta\delta_m = \frac{P_m}{k_m} \tag{e}$$

On the assumption that the members have not separated, the increase in deformation of the bolt must equal the decrease in deformation of the members, and

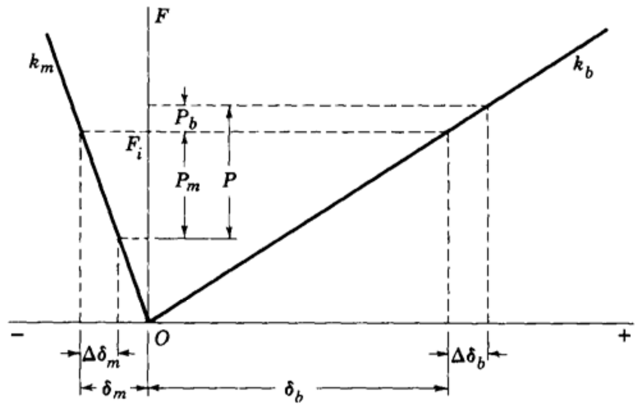


Fig. 7-13

consequently

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

Since $P = P_b + P_m$, we have

$$P_b = \frac{k_b P}{k_b + k_m} \tag{g}$$

Therefore the resultant load on the bolt is

$$F_b = P_b + F_i = \frac{k_b P}{k_b + k_m} + F_i \tag{7-12}$$

In the same manner, the resultant compression of the connected members is found to be

$$F_m = \frac{k_m P}{k_b + k_m} - F_i \tag{7-13}$$

Equations (7-12) and (7-13) hold only as long as some of the initial compression remains in the members. If the external force is large enough to remove this compression completely, the members will separate and the entire load will be carried by the bolt.

Figure 7-13 is a plot of the force-deflection characteristics and shows what is happening. The line k_m is the stiffness of the members; any force, such as the preload F_i, will cause a compressive deformation δ_m in the members. The same force will cause a tensile deformation δ_b in the bolt. When an external load is applied, δ_m is reduced by the amount $\Delta\delta_m$ and δ_b is increased by the same amount $\Delta\delta_b = \Delta\delta_m$. Thus the load on the bolt increases and the load in the members decreases.

The following example is used to illustrate the meaning of Eqs. (7-12) and (7-13). In Fig. 7-14*a* the fish scale with the 150-lbf weight is analogous to a bolt

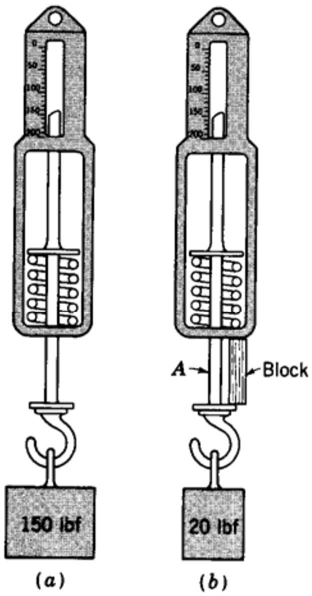


Fig. 7-14 Fish-scale analogy of a bolted joint.

tightened to a tensile preload of 150 lbf. Then in Fig. 7-14*b* a block is forced in position as shown and the 150-lbf weight removed and replaced with a 20-lbf weight. Figure 7-14*b* now represents a bolted assembly having a bolt preload of 150 lbf and an external load of 20 lbf. Adding the 20-lbf weight does not increase the tension in shank *A*, which represents the bolt. If the tension were greater than 150 lbf, the scale would read more than 150 lbf and the block would fall out. This is an extreme example, since the stiffness constant of the block k_m is a great deal more than that of the scale k_b, but it does illustrate the advantages to be gained by proper preloading. The following example is more realistic.

Example 7-2 In Fig. 7-12, let $k_m = 8k_b$. If the preload is $F_i = 1000$ lbf and the external load is $P = 1100$ lbf, what is the resultant tension in the bolt and the compression in the members?

Solution From Eq. (7-12) the resultant bolt tension is

$$F_b = \frac{k_b P}{k_b + k_m} + F_i = \frac{k_b(1100)}{k_b + 8k_b} + 1000 = 1122 \text{ lbf} \qquad \textit{Ans.}$$

The resultant compression of the members, from Eq. (7-13), is

$$F_m = \frac{k_m P}{k_b + k_m} - F_i = \frac{8k_b(1100)}{k_b + 8k_b} - 1000 = -22 \text{ lbf} \qquad \textit{Ans.}$$

This example shows that the proportion (11 percent) of the load taken by the bolt is small and that it depends on the relative stiffness. The members are still in compression, and hence there is no separation of the parts even though the external load is greater than the preload.

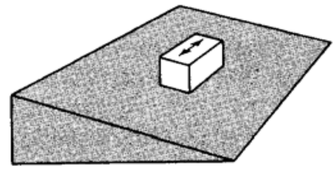


Fig. 7-15 Why a nut loosens.

The importance of preloading of bolts cannot be overestimated. A high preload improves both the fatigue resistance of a bolted connection and the locking effect. To see why this is true, imagine an external tensile load which varies from 0 to P. If the bolts are preloaded, only about 10 percent of this load will cause a fluctuating bolt stress. Thus we will be operating with a very small σ_a/σ_m slope on the modified Goodman fatigue diagram.

To see why preloading of bolts improves the locking effect, visualize (Fig. 7-15) an inclined plane representing a screw thread, and on this plane we place a block, which is analogous to the nut. Now if the block is wiggled back and forth, it will eventually work its way down the plane. This is analogous to the loosening of a nut. To retain a tight nut, the resultant bolt tension should vary as little as possible; in other words, σ_a should be very small compared with σ_m.

7-6 TORQUE REQUIREMENTS

Having learned that a high preload is very desirable in important bolted connections, we must next consider means of assuring 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 simply tighten the nut until the bolt elongates through the distance δ. This assures that the desired preload has been attained.

The elongation of a screw cannot usually be measured because the threaded end may be 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 which 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 man using an ordinary wrench. Having attained the snug-tight condition, all additional turning develops useful tension in the bolt. The turn-of-the-nut method requires that one compute the fractional number of turns necessary to develop the required preload from the snug-tight condition. For example, for heavy hexagon structural bolts the turn-of-the-nut specification states that the nut should be turned a minimum of 180 deg 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.

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. (7-5) and (7-6).

$$T = \frac{F_i d_m}{2}\left(\frac{l + \pi\mu d_m \sec\alpha}{\pi d_m - \mu l \sec\alpha}\right) + \frac{F_i \mu_c d_c}{2} \qquad (a)$$

Since $\tan\psi = l/\pi d_m$, we divide the numerator and denominator of the first term by πd_m and get

$$T = \frac{F_i d_m}{2}\left(\frac{\tan\psi + \mu\sec\alpha}{1 - \mu\tan\psi\sec\alpha}\right) + \frac{F_i \mu_c d_c}{2} \qquad (b)$$

Examination of Table A-24 shows that the diameter of the washer face of a hex 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 $d_c = 1.25d$. Equation (*b*) can now be arranged to give

$$T = \left[\left(\frac{d_m}{2d}\right)\left(\frac{\tan\psi + \mu\sec\alpha}{1 - \mu\tan\psi\sec\alpha}\right) + 0.625\mu_c\right]F_i d \qquad (c)$$

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

$$K = \left(\frac{d_m}{2d}\right)\left(\frac{\tan\psi + \mu\sec\alpha}{1 - \mu\tan\psi\sec\alpha}\right) + 0.625\mu_c \qquad (7\text{-}14)$$

Equation (*c*) now can be written

$$T = KF_i d \qquad (7\text{-}15)$$

The coefficients of thread and collar friction for bolts, screws, and nuts range from about 0.12 to 0.20, depending upon thread finish, accuracy, and degree of lubrication. On the average, both μ and μ_c are about 0.15. The interesting thing about Eq. (7-14) is that $K \approx 0.20$ for $\mu = \mu_c = 0.15$, no matter what size bolts are employed and no matter whether the threads are coarse or fine.[1] Thus Eq. (7-15) is more convenient as

$$T = 0.20F_i d \qquad (7\text{-}16)$$

In this form it is very simple to compute the wrench torque T needed to create a desired preload F_i when the size d of the fastener is known.

Blake and Kurtz have published results of numerous tests of the torquing of bolts.[2] By subjecting their data to a statistical analysis we can learn something

[1] See Table 7-3, p. 246, in the original (1963) edition of this book for a complete listing of these values.

[2] J. C. Blake and H. J. Kurtz, The Uncertainties of Measuring Fastener Preload, *Machine Design*, vol. 37, pp. 128–131, Sept. 30, 1965.

Table 7-3 The distribution of preload for 20 tests of unlubricated bolts, size ½ in.–20 UNF, torqued to 800 lbf-in.

F_i, *kips**	f	fF_i	fF_i^2
5.3	1	5.3	28.1
6.2	1	6.2	38.5
6.3	1	6.3	39.7
6.6	1	6.6	43.5
6.8	1	6.8	46.2
6.9	1	6.9	47.6
7.4	1	7.4	54.8
7.6	3	22.8	173.4
7.8	1	7.8	60.8
8.0	2	16.0	128.0
8.4	1	8.4	70.6
8.5	2	17.0	144.5
8.8	1	8.8	77.4
9.0	1	9.0	81.0
9.1	1	9.1	82.8
9.6	1	9.6	92.2
Total	20	154.0	1209.1

$$\bar{F}_i = \frac{\sum fF_i}{N} = \frac{154.0}{20} = 7.7 \text{ kips}$$

$$S_F^2 = \frac{\sum fF_i^2 - \frac{(\sum fF_i)^2}{N}}{N-1}$$

$$= \frac{1209.1 - \frac{(154)^2}{20}}{19} = 1.226$$

$$S_F = \sqrt{1.226} = 1.107 \text{ kips}$$

* 1 kip = 1000 lbf.

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 ½ in.–20 UNF when torqued to 800 lbf-in. The statistical analyses of these two groups of bolts are displayed in Table 7-3 and 7-4.

We first note that both groups have about the same mean preload, 7700 lbf. The unlubricated bolts have a standard deviation of 1100 lbf, which is about 15 percent of the mean. The lubricated bolts have a standard deviation of 680 lbf, or about 9 percent of the mean, a substantial reduction. These deviations are quite large, though, and emphasize the necessity for quality-control procedures throughout the entire manufacturing and assembly process to assure uniformity.

Table 7-4 The distribution of preload for 10 tests of lubricated bolts, size ½ in.—20 UNF, torqued to 800 lbf-in.

F_i, *kips*	f	fF_i	fF_i^2
6.8	1	6.8	46.3
7.3	2	14.6	106.9
7.4	2	14.8	109.7
7.6	1	7.6	57.7
7.7	1	7.7	59.3
7.8	1	7.8	60.8
8.4	1	8.4	70.5
9.1	1	9.1	82.8
Total	10	76.8	594.0

$$\bar{F}_i = \frac{\sum fF_i}{N} = \frac{76.8}{10} = 7.68 \text{ kips}$$

$$S_F^2 = \frac{\sum fF_i^2 - \frac{(\sum fF_i)^2}{N}}{N-1}$$

$$= \frac{594.0 - \frac{(76.8)^2}{10}}{9} = 0.464$$

$$S_F = \sqrt{0.464} = 0.681 \text{ kips}$$

The means obtained from the two samples are nearly identical, 7700 lbf; using Eq. (7-15), we find, for both samples, $K = 0.208$, which is close to the recommended value.

7-7 BOLT STRENGTH AND PRELOAD SPECIFICATIONS

Now that we have methods of estimating bolt tension, it is appropriate to investigate the strength of bolts and relate this to the preload.

The Society of Automotive Engineers (SAE)[1] has for many years published material specifications for many threaded products. The designer, of course, is free to specify any material he chooses for bolts, or he may prefer to specify a bolt made to SAE requirements. The SAE specifications cover all externally threaded fasteners and include eight grades of steels. The specifications give the minimum strength required but include no requirements on the composition of the steels; thus the manufacturer has the freedom to choose a composition which will give these properties most economically. To conform to SAE specifications, a bolt must have the minimum properties, listed in Table 7-5.

[1] See Physical Requirements for Bolts, Capscrews, Studs, and Nuts, in the "SAE Handbook" for any year. These are published annually.

Table 7-5 SAE specifications for bolts, cap screws, and studs

Grade	Bolt size, diameter, in.	Proof strength, psi	Tensile strength, min, psi	Hardness		Material—heat-treatment
				Brinell	Rockwell	
0	1/4 to 1 1/2	. . .	. . .	. . .	. . .	No requirements.
1	1/4 to 1 1/2	. . .	55,000	207 max	95 B max	Commercial steel.
2	1/4 to 1/2	55,000	69,000	241 max	100 B max	This is intended to be a cold-headed product made from low-carbon steel; C 0.28 max, P 0.04 max, and S 0.05 max. Lengths over 6 in. may be hot-headed from medium carbon steel, C 0.55 max. Deviation from specified chemistry may be made by agreement between producer and consumer.
	Over 1/2 to 3/4	52,000	64,000			
	Over 3/4 to 1 1/2	28,000	55,000	207 max	. . .	Commercial steel.
3	1/4 to 1/2	85,000	110,000	207–269	95–104 B	Produced by the cold-heading process, up to and including 6 in. in length from medium-carbon steel; C 0.28 to 0.55, P 0.04 max, and S 0.05 max.
	Over 1/2 to 5/8	80,000	100,000			
5	1/4 to 3/4	85,000	120,000	241–302	23–32 C	Medium-carbon steel; C 0.28 to 0.55, P 0.04 max, and S 0.05 max. Quenched and tempered at a minimum temperature of 800 F.
	Over 3/4 to 1	78,000	115,000	235–302	22–32 C	
	Over 1 to 1 1/2	74,000	105,000	223–285	19–30 C	
6	1/4 to 5/8	110,000	140,000	285–331	30–36 C	Medium carbon steel; C 0.28 to 0.55, P 0.04 max, and S 0.05 max. Oil quenched and tempered at a minimum temperature of 800 F.
	Over 5/8 to 3/4	105,000	133,000	269–331	28–36 C	
7	1/4 to 1 1/2	105,000	133,000	269–321	28–34 C	Medium-carbon fine-grain alloy steel,* C 0.28 to 0.55, P 0.04 max, and S 0.05 max, providing sufficient hardenability to have a minimum oil-quenched hardness of 47 RC at the center of the threaded section one diameter from the end of the bolt. Oil quenched and tempered at a minimum temperature of 800 F. Roll-threaded after heat-treatment.
8	1/4 to 1 1/2	120,000	150,000	302–352	32–38 C	Medium-carbon fine-grain alloy steel,* C 0.28 to 0.55, P 0.04 max and S 0.05 max, providing sufficient hardenability to have a minimum oil-quenched hardness of 47 RC at the center of the threaded section one diameter from the end of the bolt. Oil-quenched and tempered at a minimum temperature of 800 F.

* Carbon steel may be used by agreement between producer and consumer. *Note:* Carbon range is for check analysis of product.

Table 7-6 Identification of bolt grades; head markings

Grades 0, 1, and 2, no marking
Grade 3: 2 radial dashes 180 deg apart
Grade 5: 3 radial dashes 120 deg apart
Grade 6: 4 radial dashes 90 deg apart
Grade 7: 5 radial dashes 72 deg apart
Grade 8: 6 radial dashes 60 deg apart

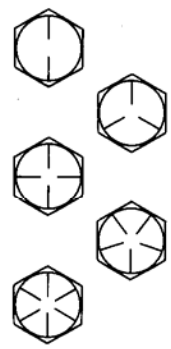


Table 7-6 shows how bolt grades are identified.

The terms proof load and proof strength appear frequently in fastener literature. The *proof load* of a bolt is the maximum tensile load a bolt can withstand without incurring a permanent set. The *proof strength* is the stress corresponding to the proof load based on the tensile-stress area (Table 7-1). Thus proof strength is roughly equivalent to yield strength.

The recommendations included here for preload apply only to ungasketed joints using high-quality bolt materials, such as SAE 3 or better. Such bolt materials will have a stress-strain diagram much like that of Fig. 7-16, in which there is no clearly defined yield point and a diagram which progresses smoothly upward until fracture. Under these conditions, *if the loads are static*, the *minimum* preload should be 90 percent of the proof load.

The recommendations for preload when fatigue is present are so important that we shall discuss this problem separately in the next section.

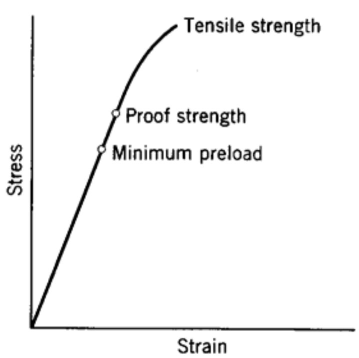


Fig. 7-16 Typical stress-strain diagram for good bolt materials.

There are two sound reasons for the 90 percent recommendation for statically loaded joints composed of high-quality bolt materials. Because the stress-strain diagram (Fig. 7-16) proceeds smoothly to fracture, the bolt will retain its load-carrying capacity no matter how high the pre-tension. That is, there is no loss of capacity due to gross plastic yielding. The second reason for the 90 percent recommendation is that the torsional stress disappears after tightening. A joint subjected to slight movements will cause flattening of high spots, paint, or dirt and will relieve the torsional friction. Thus, if such a bolt does not fail during tightening, there is a very good chance that it never will fail! In some cases it may be desirable to turn the nut backward an eighth of a turn or so after tightening to relieve this torsion, but not enough to decrease the bolt tension.

Example 7-3 Calculate the tightening torque for an SAE grade 6, ½″–13 NC bolt. Compute the tensile stress and the torsional stress, and show on a Mohr's circle diagram the reduction in principal stress due to the disappearance of the torsional stress.

Solution From Table 7-5, the proof strength is $S_P = 110{,}000$ psi. Also from Table 7-1, the tensile stress area is $A_t = 0.1419$ in.^{2}. Therefore the recommended preload is

$$F_i = 0.90 S_P A_t = (0.90)(110{,}000)(0.1419) = 14{,}050 \text{ lbf}$$

With $K = 0.20$, the tightening torque is

$$T = 0.20 F_i d = (0.20)(14{,}050)(0.500) = 1405 \text{ lbf-in.} \qquad \textit{Ans.}$$

The tensile stress is 90 percent of the proof strength, and so $\sigma_x = (0.90)(110{,}000) = 99{,}000$ psi.

The torque applied to the nut is used up in three ways. About 50 percent of it is used to overcome the friction between the bearing face of the nut and the member. About 40 percent of the applied torque is used to overcome thread friction, and the balance produces the bolt tension. The last two items are the only contributions to the torsion in the screw. From Table 7-1, $A_r = 0.1257$ in.^{2}, and so the root diameter is $d_r = \sqrt{4A_r/\pi} = \sqrt{(4)(0.1257)/\pi} = 0.400$ in. The torsion in the screw is $T = (0.50)(1405) = 702$ lbf-in., and so

$$\tau_{xy} = \frac{16T}{\pi d_r^3} = \frac{(16)(702)}{\pi(0.400)^3} = 56{,}000 \text{ psi}$$ (Eqn. 2-41)

Drawing Mohr's circle corresponding to $\sigma_x = 99{,}000$ psi and $\tau_{xy} = 56{,}000$ psi gives the solid circle of Fig. 7-17. If the principal stresses are calculated, they will be found to be $\sigma_1 = 124{,}200$ psi and $\sigma_2 = -25{,}200$ psi.

For fracture, the maximum-normal-stress theory applies. Thus, during tightening, the factor of safety is

$$n = \frac{S_u}{\sigma_1} = \frac{140{,}000}{124{,}200} = 1.12$$

or a 12 percent margin of safety. (Note that this is less than one standard deviation for unlubricated bolts!)

After the torsion disappears, the principal stress is σ_x and the factor of safety will be

$$n = \frac{S_u}{\sigma_x} = \frac{140{,}000}{99{,}000} = 1.41$$

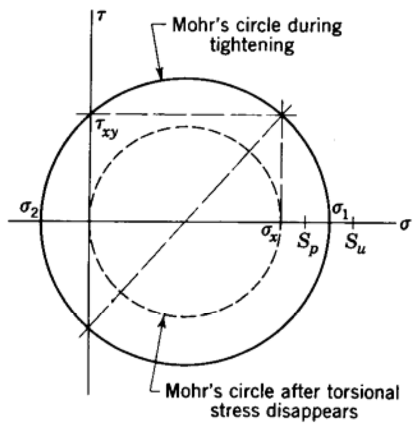


Fig. 7-17 Mohr's circle diagram for a bolt during tightening. Torquing of the nut produces shear stress τ_{xy}. The bolt tensile stress is σ_x. The proof strength is S_p, and the tensile strength S_u.

You can now appreciate the statement: If a bolt does not fail during tightening, there is good reason to believe that it never will fail.

In the design of a tension-loaded bolted joint, factor of safety n is used in a somewhat different way than in Example 7-3. In Example 7-2 it was learned that an external load about equal to the preload would remove most of the compression in the members. Therefore, to get a first approximation in the determination of bolt size, define factor of safety as

$$n = \frac{F_i}{P} \tag{7-17}$$

Then the bolt size is determined such that its preload is larger than the external tensile load by the amount of the factor of safety. When high-quality bolt materials are used, when the members are stiff, and when the loads are static, this is about as far as we need go. If fatigue loads are present, a detailed analysis of the resulting design should be made using the approach of the next section.

7-8 FATIGUE RESISTANCE OF BOLTED JOINTS

Bolted joints subjected to fatigue loads may be analyzed directly using the methods of Chap. 6. Geometric stress-concentration factors for the fillet under

Table 7-7 Fatigue-strength reduction factors K_f for threaded elements

SAE steel	*Rolled threads*	*Cut threads*
Grade 0 to 3	2.2	2.8
Grade 5 to 8	3.0	3.8

the bolt head may be obtained from Table A-20-7. The fatigue-strength reduction factors for rolled or cut threads may be obtained from Table 7-7. The improvement due to rolling is so substantial that one should not even consider the use of a fastener having cut threads when fatigue failure is a possibility. A similar improvement can be obtained by cold-rolling the bolt-head fillet.

In a fatigue analysis bolts and screws are considered to have a machined finish.

Example 7-4 A tension-loaded bolted connection employing a confined gasket is shown in Fig. 7-18. This connection has been tentatively designed to resist a fluctuating load $P = 12{,}000$ lbf. Based on a reliability of 50 percent and recommended bolt preload, determine whether fatigue failure is likely to occur.

Solution From Table 7-5, we find the proof and tensile strengths to be $S_P = 80$ kpsi and $S_{ut} = 100$ kpsi. From Table 7-1 the tensile-stress area is $A_t = 0.226$ in.^{2}. Therefore the recommended preload is

$$F_i = (0.90)(0.226)(80{,}000) = 16{,}300 \text{ lbf}$$

From Eqs. (7-10) and (7-11), we find the stiffness, respectively, of the members and the bolt to be

$$k_m = \frac{2\pi d^2 E}{l} = \frac{2\pi(0.625)^2(12)(10)^6}{1.5} = 19.6(10)^6 \text{ lbf/in.}$$

$$k_b = \frac{\pi d^2 E}{4l} = \frac{\pi(0.625)^2(30)(10)^6}{(4)(1.5)} = 6.13(10)^6 \text{ lbf/in.}$$

where the grip is $l = 1\frac{1}{2}$ in. and $E = 12(10)^6$ psi for No. 25 cast iron. The maximum tension in the bolt is, from Eq. (7-12),

$$F_b = \frac{k_b P}{k_b + k_m} + F_i = \frac{(6.13)(12{,}000)}{6.13 + 19.6} + 16{,}300 = 19{,}160 \text{ lbf}$$

Therefore the maximum tensile stress is

$$\sigma_{max} = \frac{F_b}{A_t} = \frac{19{,}160}{0.226} = 84{,}000 \text{ psi}$$

The minimum stress is caused by the preload, which is 90 percent of the proof strength, or

$$\sigma_{min} = 0.90 S_P = (0.90)(80{,}000) = 72{,}000 \text{ psi}$$

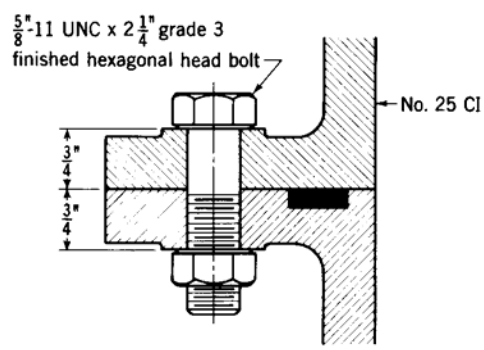


Fig. 7-18

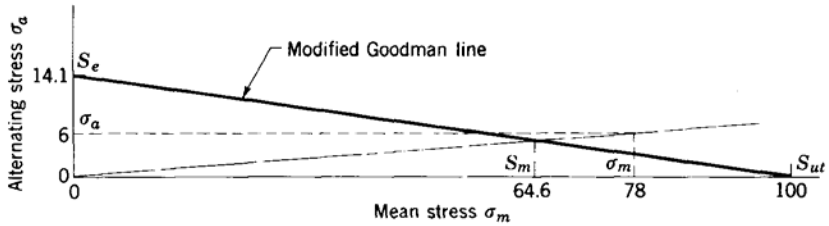


Fig. 7-19 Stresses, in kpsi.

Therefore the alternating and mean stress components are

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{84{,}000 - 72{,}000}{2} = 6000 \text{ psi}$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{84{,}000 + 72{,}000}{2} = 78{,}000 \text{ psi}$$

Next, we must determine the endurance limit of the bolt. First, using the methods of Chap. 6,

$$S'_e = 0.50S_{ut} = (0.50)(100{,}000) = 50{,}000 \text{ psi}$$

The surface-finish factor is $k_a = 0.73$, corresponding to a machined finish. The size factor is $k_b = 0.85$. From Table A-23 find the minimum fillet radius under the bolt head to be 0.021 in. Using the formulas in the stress-concentration chart of Table A-20-7, we find

$$\frac{r}{d} = \frac{0.021}{0.625} = 0.0336 \qquad \frac{D}{d} = \frac{0.9375}{0.625} = 1.5$$

where D corresponds to the diameter of the washer face (Table A-23). Then, from Table A-20-7, we find

$$K_t = 2.6$$

Using $q = 0.72$ from Chap. 6, we find the fatigue-strength reduction factor at the head of the bolt to be

$$K_f = 1 + 0.72(2.6 - 1) = 2.15$$

Therefore Table 7-7 governs, and we assume the bolt is weakest at the threads where $K_f = 2.2$. Hence

$$k_e = 1/2.2 = 0.454$$

We now find the endurance limit of the bolt to be

$$S_e = k_a k_b k_e S'_e = (0.73)(0.85)(0.454)(50{,}000) = 14{,}100 \text{ psi}$$

From the modified Goodman diagram of Fig. 7-19, plotted using the data of this example, we record

$$S_m = 64{,}600 \text{ psi}$$

which, when compared with the stress $\sigma_m = 78{,}000$ psi, indicates that a fatigue failure can be expected to occur before 10^6 cycles of load application.

7-9 SELECTION OF THE NUT

Imagine three annular rings, analogous to square threads on a screw, cut on a male member, and three corresponding grooves with clearances as shown in Fig.

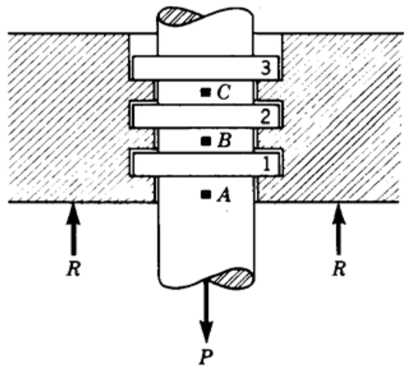


Fig. 7-20

7-20 on a female member or nut. Now apply a tension load P to the screw and let the nut react to this load as shown in the figure. If the load is assumed to be uniformly divided between the three threads, stress elements at A, B, and C on the screw will have tensile loads of $F_A = P$, $F_B = 2P/3$, and $F_C = P/3$. Corresponding stress elements in the nut, not shown, will have compressive loads $F_A = -P$, $F_B = -2P/3$, and $F_C = -P/3$. Now, with the screw in tension, the screw gets longer, and so threads 1, 2, and 3 will tend to move apart. However, the nut is in compression, and so the nut threads will tend to move closer together. But these actions prevent sharing of the loads as assumed in the beginning. We therefore conclude that the load will not be shared at all and that, instead, the first thread takes the entire force.

This tendency may be partially corrected by proportioning the nut so as to cause more deformation to exist at the bottom. Figure 7-21 shows two nut designs in which material has been removed from the lower portion of the nut in order to equalize the stress distribution.

In practice, conditions are not quite as severe as pictured, since yielding of the threads in the nut will permit the other threads to transfer some of the load. However, since such a tendency is present, it must be guarded against and knowledge of it used in selecting the nut.

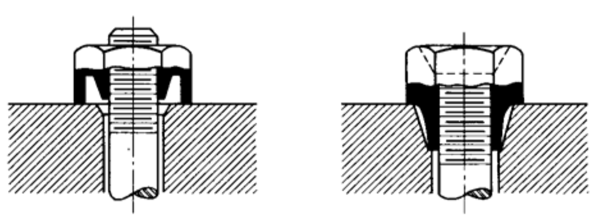


Fig. 7-21 Typical methods of distributing the thread load.

Another factor which acts to reduce the tendency of the bottom thread to take the entire load is that the wedging action of the threads tends to spread, or dilate, the nut.

These conditions point to the fact that, when preloading is desired, careful attention should be given to the nut material. Selecting a soft nut ensures plastic yielding, which will enable the nut threads to divide the load more evenly.

Nuts are tested by determining their stripping strength. The test is made by threading a nut on a hardened-steel mandrel and pulling it through the nut. The strength is the load divided by the mean thread area. Common nuts have a stripping strength of approximately 90,000 psi.

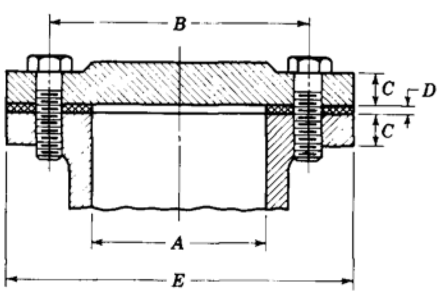
It is interesting to know that three full threads are all that are required to develop the full bolt strength.

Another factor which must be considered in the design of bolted joints is the maintenance of the initial preload. This load may be relaxed by yielding of the clamped material, by extrusion of paint or plating from the contact surfaces, or by a compression of rough places. Extra contact area may be provided by hardened washers. This is especially necessary if the bolted parts are relatively soft and the bolt head or nut does not provide sufficient bearing area.

PROBLEMS

7-12. Compute the proper tightening torque for a grade 5 bolt $\frac{1}{2}$ in.–13 UNC. What is the proper torque if fine threads are used?

7-13. The gasketed joint shown in the figure has $A = 4$ in., $B = 6$ in., $C = \frac{3}{4}$ in., $D = \frac{1}{8}$ in., and $E = 8$ in. Based on the assumption that the screws will create a uniform seal pressure when tightened, a pressure of at least 2200 psi must be obtained for a satisfactory seal. Using SAE grade 3 bolt material, determine the number and size of fine-thread cap screws required for effective sealing.



Prob. 7-13

7-14. The figure illustrates the connection of a cylinder head to a pressure vessel using 12 bolts and a confined gasket. The dimensions are $A = 4$ in., $B = 8$ in., $C = 10\frac{1}{2}$ in.,