5.3 Threaded Fasteners

Figure (5–9) 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. 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

$$L_T = \begin{cases} 2d + \frac{1}{4}\text{ in} & L \leq 6\text{ in} \\ 2d + \frac{1}{2}\text{ in} & L > 6\text{ in} \end{cases} \qquad 5\text{-}11$$

and for metric bolts is

$$L_T = \begin{cases} 2d + 6 & L \leq 125 \quad d \leq 48 \\ 2d + 12 & 125 < L \leq 200 \\ 2d + 25 & L > 200 \end{cases} \qquad 5\text{-}12$$

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.

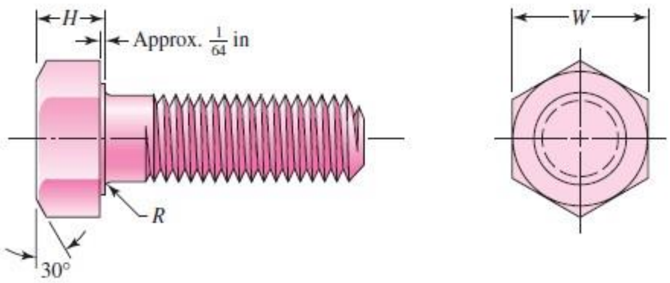


Figure (5–9)

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.

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

The head of a hexagon-head cap screw is slightly thinner than that of a hexagon- head bolt. 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. (5–10).

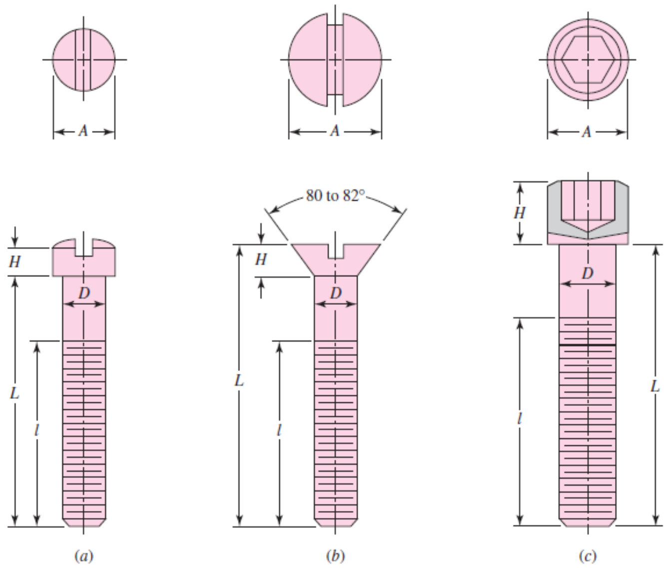


Figure (5–10)

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. (5–9), as well as a variety of other head styles. This illustration uses one of the conventional methods of representing threads

A variety of machine-screw head styles are shown in Fig. (5–11). Inch-series machine screws are generally available in sizes from No. 0 to about 3/8 in. Several styles of hexagonal nuts are illustrated in Fig. (5–12). 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.

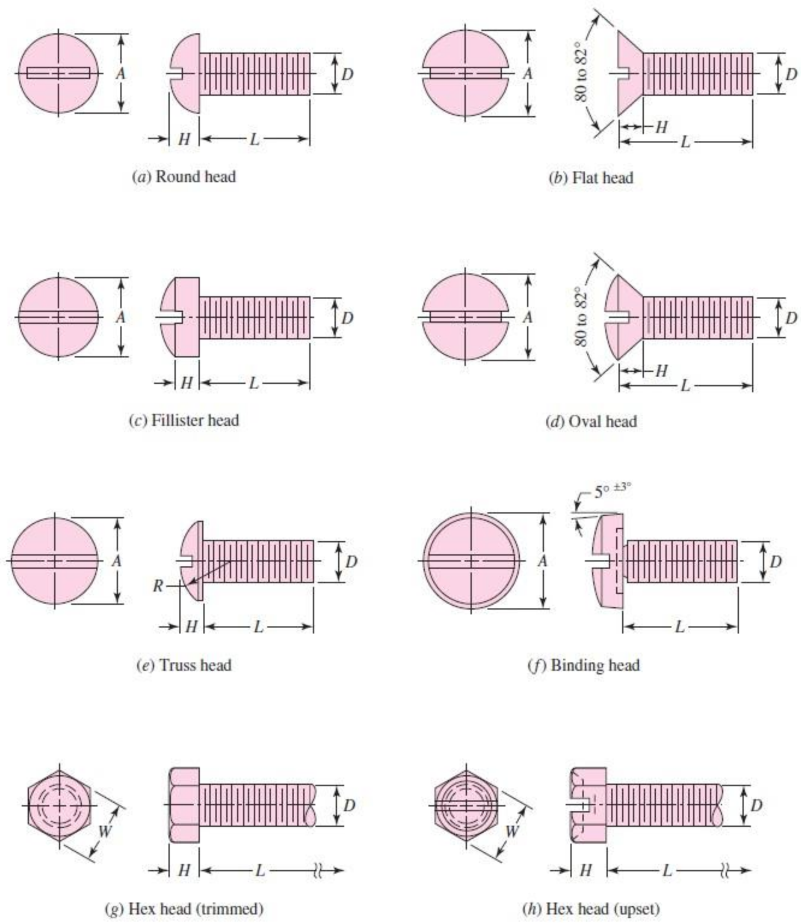


Figure (5–11)
Types of heads used on machine screws

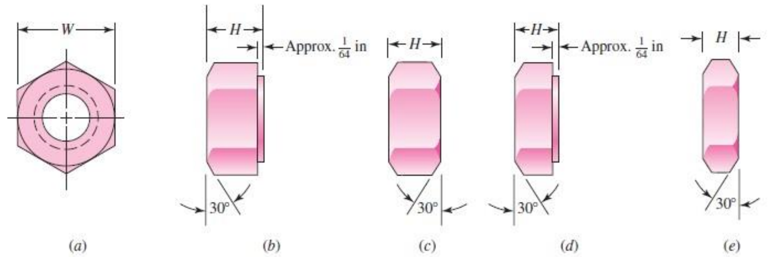


Figure (5–12)
Hexagonal nuts: (*a*) end view, general; (*b*) washer-faced regular nut; (*c*) regular nut chamfered on both sides; (*d*) jam nut with washer face; (*e*) jam nut chamfered on both sides.

5.4 Joints–Fastener, Member Stiffness and Bolt Strength

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

A section through a tension-loaded bolted joint is illustrated in Fig. (5–13). **Notice the clearance space provided by the bolt holes. Notice, too, how the bolt threads extend into the body of the connection.**

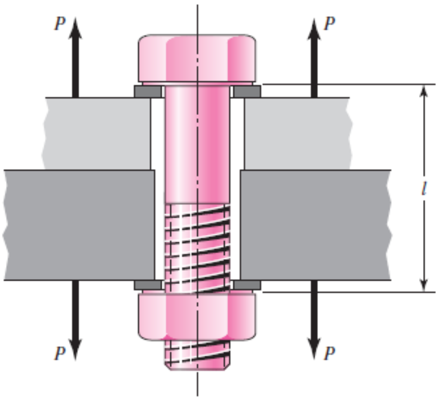


Figure (5–13)

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

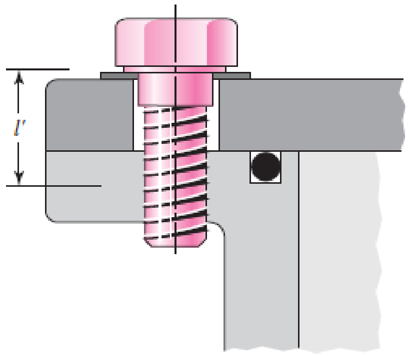


Figure (5–14)

Section of cylindrical pressure vessel. Hexaaggoonn--hhead 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

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 (5–14) 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.

Y. Ito 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. 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. Figure (5–15) illustrates the general cone geometry using a half-apex angle (α). C. C. Osgood reports a range of $25^o \leq \alpha \leq 33^o$ for most combinations.

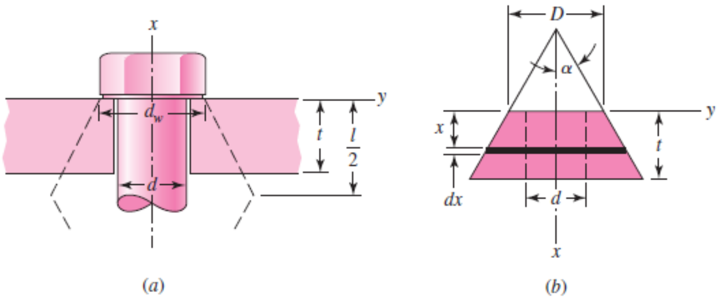


Figure (5–15)

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

In the specification standards for bolts, the strength is specified by stating ASTM minimum quantities, the *minimum proof strength* (S_p), 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). The value 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, before designing to a reliability specification. 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.

5.5 Bolted and Riveted Joints Loaded in Shear

Riveted and bolted joints loaded in shear are treated exactly alike in design and analysis. Figure (5–16*a*) shows a riveted connection loaded in shear. Figure (5–16*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,

$$\sigma = \frac{M}{I/c} \qquad \text{5-13}$$

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. (5–16*c*) failure of the rivet by pure shear is shown; the stress in the rivet is

$$\tau = \frac{F}{A} \qquad \text{5-14}$$

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.

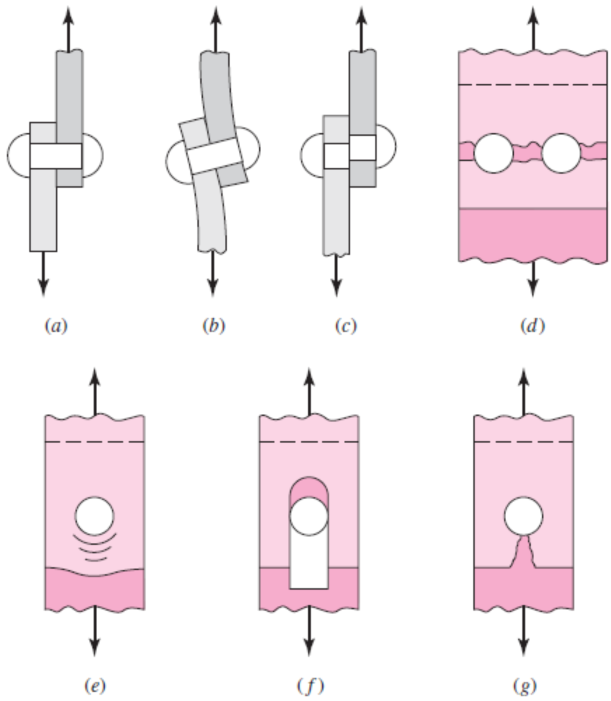


Figure (5–16)

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. (5–16*d*). The tensile stress is

$$\sigma = \frac{F}{A} \qquad \text{5-15}$$

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. (5–16), the designer should, of course, use the combination of rivet or bolt holes that gives the smallest area.

Figure (5–16*e*) 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

$$\sigma = -\frac{F}{A} \tag{5-16}$$

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.

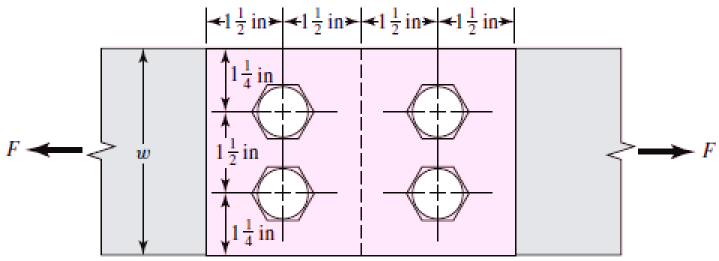
Edge shearing, or tearing, of the margin is shown in Fig. (5–16*f* and *g*), respectively. In structural practice this failure is avoided by spacing the rivets at least 1.5 diameters away from the edge. Bolted connections usually are spaced an even greater distance than this for satisfactory appearance, 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

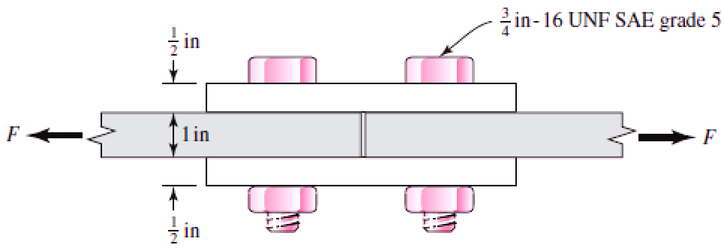
- Bearing in the bolt (all bolts participate)
- Bearing in members (all holes participate)
- Shear of bolt (all bolts participate eventually)
- Distinguishing between thread and shank shear
- Edge shearing and tearing of member (edge bolts participate)
- Tensile yielding of member across bolt holes
-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

EXAMPLE 5–2

Two 1- by 4-in 1018 cold-rolled steel bars are butt-spliced with two 0.5- by 4-in 1018 cold-rolled splice plates using four 0.75 in-16 UNF grade 5 bolts as depicted in the figure. For a design factor of n_d = 1.5 estimate the static load F that can be carried if the bolts lose preload.



(a)



(b)

Solution

From Table (3–4), minimum strengths of S_y = 54 kpsi and S_{ut} = 64 kpsi are found for the members, and from Table (5–5) minimum strengths of S_p = 85 kpsi and S_{ut} = 120 kpsi for the bolts are found.

$F/2$ 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:

$$\sigma = \frac{F}{2td} = \frac{S_p}{n_d} \qquad F = \frac{2tdS_p}{n_d} = \frac{2(1)\left(\frac{3}{4}\right)85}{1.5} = 85 \text{ kip}$$

Bearing in members, all bolts active:

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

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

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

If the bolt threads extend into a shear plane:

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

Edge shearing of member at two margin bolts: From the figure,

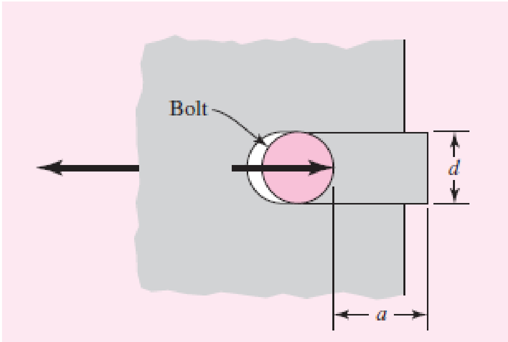
$$\tau = \frac{F}{4at} = \frac{0.577(S_y)_{\text{mem}}}{n_d}$$

$$F = \frac{4at0.577(S_y)_{\text{mem}}}{n_d} = \frac{4(1.125)(1)0.577(54)}{1.5} = 93.5 \text{ kip}$$

Tensile yielding of members across bolt holes:

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

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



Bolt
d
a

Member yield:

$$F = w\, t\, [(S_y)_{mem} / n_d] = 4\ (1)\ (54/1.5) = 144 \text{ kip}$$

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.