Table 8-4

<table>
<thead>
<tr>
<th>Screw Material</th>
<th>Nut Material</th>
<th>Safe ( p_{br} ), psi</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>Bronze</td>
<td>2500-3500</td>
<td>low speed</td>
</tr>
<tr>
<td>Steel</td>
<td>Bronze</td>
<td>1600-2500</td>
<td>10 fps</td>
</tr>
<tr>
<td>Steel</td>
<td>Cast iron</td>
<td>1800-2500</td>
<td>8 fps</td>
</tr>
<tr>
<td>Steel</td>
<td>Bronze</td>
<td>800-1400</td>
<td>20-40 fps</td>
</tr>
<tr>
<td>Steel</td>
<td>Cast iron</td>
<td>600-1000</td>
<td>20-40 fps</td>
</tr>
<tr>
<td>Steel</td>
<td>Bronze</td>
<td>150-240</td>
<td>50 fps</td>
</tr>
</tbody>
</table>

Table 8-5

<table>
<thead>
<tr>
<th>Screw Material</th>
<th>Steel</th>
<th>Nut Material</th>
<th>Brass</th>
<th>Brass</th>
<th>Cast Iron</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel, dry</td>
<td>0.15-0.25</td>
<td>0.15-0.23</td>
<td>0.15-0.19</td>
<td>0.15-0.25</td>
<td></td>
</tr>
<tr>
<td>Steel, machine oil</td>
<td>0.11-0.17</td>
<td>0.10-0.16</td>
<td>0.10-0.15</td>
<td>0.11-0.17</td>
<td></td>
</tr>
<tr>
<td>Bronze</td>
<td>0.08-0.12</td>
<td>0.04-0.06</td>
<td>—</td>
<td>0.06-0.09</td>
<td></td>
</tr>
</tbody>
</table>

Table 8-6

<table>
<thead>
<tr>
<th>Combination</th>
<th>Running</th>
<th>Starting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Soft steel on cast iron</td>
<td>0.12</td>
<td>0.17</td>
</tr>
<tr>
<td>Hard steel on cast iron</td>
<td>0.09</td>
<td>0.15</td>
</tr>
<tr>
<td>Soft steel on bronze</td>
<td>0.08</td>
<td>0.10</td>
</tr>
<tr>
<td>Hard steel on bronze</td>
<td>0.06</td>
<td>0.08</td>
</tr>
</tbody>
</table>

Ham and Ryan\(^1\) 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 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 presence of a blend of stochastic and deterministic viewpoints. In most cases the threat is from over-proof loading of fasteners, and this is best addressed using 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 E-26 for dimensions. The diameter of the washer face is the same as the width across the flats of the hexagon. The thread length

---

\(^1\) Ham and Ryan, An Experimental Investigation of the Friction of Screw-threads, Bulletin 247, University of Illinois Experiment Station, Champaign-Urbana, IL.
of inch-series bolts, where \( D \) is the nominal diameter, is

\[
L_T = \begin{cases} 
2D + \frac{1}{4} \text{ in} & L \leq 6 \text{ in} \\
2D + \frac{1}{2} \text{ in} & L > 6 \text{ in}
\end{cases}
\]  

(8–13)

and for metric bolts is

\[
L_T = \begin{cases} 
2D + 6 & L \leq 125 \text{ } D \leq 48 \\
2D + 12 & 125 < L \leq 200 \\
2D + 25 & L > 200
\end{cases}
\]  

(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 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 E–27. 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 E–28. 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.

### 8–4 Joints—Fastener Stiffness

When a connection is desired which can be disassembled without destructive methods and which is strong enough to resist external tensile loads, moment loads, and shear loads,
Figure 8-10

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

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or a combination of these, then the simple bolted joint using hardened-steel washers is a good solution. Such a joint can also be dangerous unless it is properly designed and assembled by a trained mechanic.

A 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 pre-tension 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 which 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 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 grip is the thickness of the top member plus that of the washer.

The stiffness of the portion of a bolt or screw within the clamped zone will generally consist of two parts, that of the unthreaded shank portion and that of the threaded portion.
Figure 8–11
Types of heads used on machine screws.

(a) Round head
(b) Flat head
(c) Fillister head
(d) Oval head
(e) Truss head
(f) Binding head
(g) Hex head (trimmed)
(h) Hex head (upset)

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.

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

$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2} \quad \text{or} \quad k = \frac{k_1k_2}{k_1 + k_2}$$

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,
Screws, Fasteners, and the Design of Nonpermanent Joints

8-13
A connection loaded in by the forces $P$. Note the two washers. A simplified method is used to represent the screw. Note how the threads go into the body of the connection. This is usual and is

![Diagram of a loaded connection with washers and screw threads](image)

8-14
Hexagon-head cap screws are used to fasten the head to the body. Note an O-ring seal.

![Diagram of a hexagon-head cap screw](image)

$$k_T = \frac{A_t E}{l_t} \quad k_d = \frac{A_d E}{l_d} \tag{8-16}$$

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

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

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 $k_b$; and in the case of 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 stiffness 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