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# FASTENING, JOINING, AND CONNECTING

# CHAPTER 22

# BOLTED AND RIVETED JOINTS

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## SYMBOLS AND UNITS

| | |
|---|---|
| $A$ | Cross-sectional area, $in^2$ ($mm^2$) |
| $A_B$ | Cross-sectional area of the body of a bolt, $in^2$ ($mm^2$) |
| $A_r$ | Cross-sectional area of the body of the rivet, $in^2$ ($mm^2$) |
| $A_S$ | Cross-sectional area of the tensile stress area of the threaded portion of a bolt, $in^2$ ($mm^2$) |
| $A_1, A_2, A_3$, etc. | Cross-sectional areas of individual fasteners, $in^2$ ($mm^2$) |
| $b$ | Number of shear planes which pass through the fastener; and/or the number of slip surfaces in a shear joint |
| $d$ | Nominal diameter of the bolt, in (mm) |
| $E$ | Modulus of elasticity, psi (MPa) |
| $F$ | Force, lb (kN) |
| $F_B$ | Tension in a bolt, lb (kN) |
| $F_b$ | Primary shear force on a bolt, lb (kN) |
| $F_B$(max) | Maximum anticipated tension in the bolt, lb (kN) |
| $F_{BY}$ | Tension in a bolt at yield, lb (kN) |
| $F_C$ | Clamping force on the joint, lb (kN) |
| $F_C$(min) | Minimum acceptable clamping force on a joint, lb (kN) |
| $F_J$(min) | Minimum anticipated clamping force on the joint, lb (kN) |

| | |
|---|---|
| $F_n$ | Reaction moment force seen by the *n*th bolt in an eccentrically loaded shear joint, lb (kN) |
| $F_{PA}$ | Average preload in a group of bolts, lb (kN) |
| $F_P(\text{max})$ | Maximum anticipated initial preload in a bolt, lb (kN) |
| $F_P(\text{min})$ | Minimum anticipated initial preload in a bolt, lb (kN) |
| $F_{PT}$ | Target preload, lb (kN) |
| $F_{TR}$ | Maximum external transverse load on the joint, per bolt, lb (kN) |
| $F_r$ | External shear load on the rivet, lb (kN) |
| $F_T(\text{max})$ | Maximum acceptable tension in a bolt, lb (kN) |
| $F_X$ | External tension load on a joint, lb (kN) |
| $F_1, F_2, F_3$, etc. | Secondary shear or reaction moment forces seen by individual bolts in an eccentric joint, lb (kN) |
| $H$ | Distance between the centerline of the bolt holes nearest to the edge of a joint or splice plate and that edge, in (mm) |
| $k_B$ | Stiffness of a bolt or rivet, lb/in (kN/mm) |
| $k_G$ | Stiffness of a gasket, lb/in (kN/mm) |
| $k_J$ | Stiffness of the joint members, lb/in (kN/mm) |
| $k_T$ | Stiffness of gasketed joint, lb/in (kN/mm) |
| $K$ | Nut factor |
| $l_G$ | Grip length of the fasteners, in (mm) |
| $L$ | Distance between the bolt and the nearest edge of the connected part, or to the nearest edge of the next bolt hole, measured in the direction of the force on the joint, in (mm) |
| $L_B$ | Effective length of the body of a bolt (the length of body in the grip plus one-half the thickness of the head, for example), in (mm) |
| $L_S$ | Effective length of the threaded portion of a bolt [the length of the threads within the grip plus one-half the thickness of the nut(s), for example], in (mm) |
| $m$ | Number of fasteners in the joint |
| $M$ | Moment exerted on a shear joint by an external force, lb · in (N · m) |
| $n$ | Number of threads per inch |
| $N$ | Number of cycles achieved in fatigue life test |
| $P$ | Pitch of the threads, in (mm) |
| $P_S$ | Scatter in preload anticipated from bolting tool used for assembly (expressed as a decimal) |
| $P_Z$ | Percentage loss (expressed as a decimal) in initial preload as a result of short-term relaxation and/or elastic interactions |
| $r$ | Radial distance from the centroid of a group of fasteners to a fastener, in (mm) |
| $r_n$ | Radial distance to the *n*th fastener, in (mm) |

| | |
|---|---|
| $r_s$ | Bolt slenderness ratio ($l_G/d$) |
| $r_1, r_2, r_3$, etc. | Radial distance of individual fasteners, in (mm) |
| $R_{JB}$ | Stiffness ratio ($k_J/k_B$) |
| $R_S$ | Slip resistance of a friction-type joint, lb (kN) |
| $S$ | Ratio of the ultimate shear strength of the bolt material to its ultimate tensile strength |
| $S_u$ | Minimum ultimate tensile strength, psi (MPa) |
| $S_{YB}$ | Yield strength of the bolt, psi (MPa) |
| $t$ | Thickness of a joint or a splice plate, in (mm) |
| $t_J$ | Total thickness of a joint, in (mm) |
| $T$ | Torque, lb · in (N · m) |
| $W$ | Width of a joint plate, in (mm) |
| $x$ | Coordinate distance, in (mm) |
| $\bar{x}$ | Coordinate distance to the centroid of a bolt group, in (mm) |
| $x_1, x_2, x_3$, etc. | $x$ coordinates for individual fasteners, in (mm) |
| $y$ | Coordinate distance, in (mm) |
| $\bar{y}$ | Coordinate distance to the centroid of a bolt group, in (mm) |
| $y_1, y_2, y_3$, etc. | $y$ coordinates for individual fasteners, in (mm) |
| $\Delta$ | Incremental change or variation |
| $\lambda$ | Ratio of shear stress in a bolt to the ultimate tensile strength |
| $\mu_S$ | Slip coefficient of a friction-type joint |
| $\sigma$ | Stress, psi (MPa) |
| $\sigma_B$ | Bearing stress, psi (MPa) |
| $\sigma(\max)$ | Maximum tensile stress imposed during fatigue tests, psi (MPa) |
| $\sigma_T$ | Allowable tensile stress, psi (MPa) |
| $\sigma_T(\max)$ | Maximum acceptable tensile stress in a bolt, psi (MPa) |
| $\sigma^2$ | Statistical variance (standard deviation squared) |
| $\sigma_O^2$ | Statistical variance of the tension errors created by operator variables |
| $\sigma_T^2$ | Statistical variance of the tension errors created by tool variables |
| $\tau$ | Shear stress, psi (MPa) |
| $\tau_A$ | Allowable shear stress, psi (MPa) |
| $\tau_B$ | Shear stress in a bolt, psi (MPa) |
| $\phi$ | Ratio of tensile stress in a bolt to the ultimate tensile strength |

Joints are an extremely important part of any structure. Whether held together by bolts or rivets or weldments or adhesives or something else, joints make complex structures and machines possible. Bolted joints, at least, also make disassembly and reassembly possible. And many joints are critical elements of the structure, the thing most likely to fail. Because of this, it is important for the designer to understand joints. In this chapter we will deal specifically with bolted and riveted joints, starting with a discussion of joints loaded in shear (with the applied loads at right angles to the axes of the fasteners) and continuing with tension joints in which the loads are applied more or less parallel to the axes of fasteners. As we shall see, the design procedures for shear joints and tension joints are quite different.

## 22.1 SHEAR LOADING OF JOINTS

Now let us look at joints loaded in shear. I am much indebted, for the discussion of shear joints, to Shigley, Fisher, Higdon, and their coauthors ([22.1], [22.2], [22.3]).

### 22.1.1 Types of Shear Joints

Shear joints are found almost exclusively in structural steel work. Such joints can be assembled with either rivets or bolts. Rivets used to be the only choice, but since the early 1950s, bolts have steadily gained in popularity.

Two basic types of joint are used, *lap* and *butt*, each of which is illustrated in Fig. 22.1. These are further defined as being either (1) friction-type joints, where the fasteners create a significant clamping force on the joint and the resulting friction between joint members prevents joint slip, or (2) bearing-type joints, where the fasteners, in effect, act as points to prevent slip.

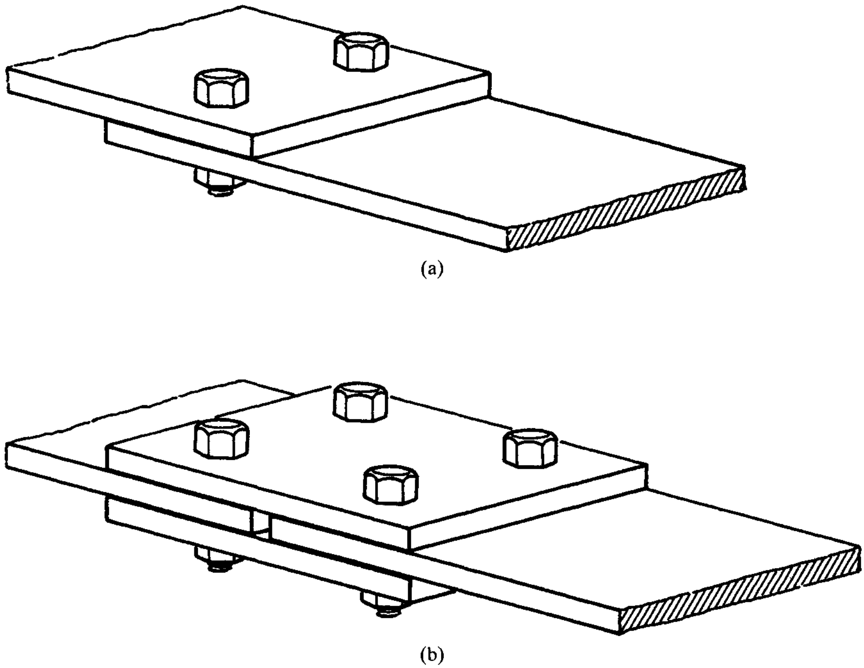



**FIGURE 22.1** Joints loaded in shear. (*a*) Lap joint; (*b*) butt joint.

Only bolts can be used in friction-type joints, because only bolts can be counted on to develop the high clamping forces required to produce the necessary frictional resistance to slip. Rivets or bolts can be used in bearing-type joints.

### 22.1.2 Allowable-Stress Design Procedure

In the *allowable-stress design procedure,* all fasteners in the joint are assumed to see an equal share of the applied loads. Empirical means have been used to determine the maximum working stresses which can be allowed in the fasteners and joint members under these assumptions. A typical allowable shear stress might be 20 percent of the ultimate shear strength of the material. A factor of safety (in this case 5:1) has been incorporated into the selection of allowable stress.

We should note in passing that the fasteners in a shear joint do not, in fact, all see equal loads, especially if the joint is a long one containing many rows of fasteners. But the equal-load assumption greatly simplifies the joint-design procedure, and if the assumption is used in conjunction with the allowable stresses (with their built-in factors of safety) derived under the same assumption, it is a perfectly safe procedure.

***Bearing-type Joints.*** To design a successful bearing-type joint, the designer must size the parts so that the fasteners will not shear, the joint plates will not fail in tension nor be deformed by bearing stresses, and the fasteners will not tear loose from the plates. None of these things will happen if the allowable stresses are not exceeded in the fasteners or in the joint plates. Table 22.1 lists typical allowable stresses specified for various rivet, bolt, and joint materials. This table is for reference only. It is always best to refer to current engineering specifications when selecting an allowable stress for a particular application.

Here is how the designer determines whether or not the stresses in the proposed joint are within these limits.

*Stresses within the Fasteners.* The shear stress within a rivet is

$$\tau = \frac{F}{bmA_r} \tag{22.1}$$

The shear stress within each bolt in the joint will be

$$\tau = \frac{F}{A_T} \tag{22.2}$$

A bolt can have different cross-sectional areas. If the plane passes through the unthreaded body of the bolt, the area is simply

$$A_B = \frac{\pi d^2}{4} \tag{22.3}$$

If the shear plane passes through the threaded portion of the bolt, the cross-sectional area is considered to be the tensile-stress area of the threads and can be found for Unified [22.4] or metric [22.5] threads from

**TABLE 22.1** Allowable Stresses

| Material | Source | Comments | Allowable stress | | |
|---|---|---|---|---|---|
| | | | Tension kpsi (MPa) | Shear kpsi (MPa) | Bearing† kpsi (MPa) |
| ASTM A325 bolts | 1 | Used in bearing-type joints with slotted or standard holes, and | | | † |
| | | some threads in shear planes | .... | 21.0 (145) | |
| | | no threads in shear planes | .... | 30.0 (207) | |
| ASTM A325 bolts | 1 | Used in friction-type joints with standard holes and surfaces of | | | † |
| | | clean mill scale | .... | 17.5 (52) | |
| | | blast-cleaned carbon or low-alloy steel | .... | 27.5 (190) | |
| | | blast-cleaned inorganic zinc rich paint | | 29.5 (203) | |
| ASTM A490 bolts | 1 | Bearing-type joints with slotted or standard holes, and | | | † |
| | | some threads in shear planes | .... | 28.0 (193) | |
| | | no threads in shear planes | .... | 40.0 (276) | |
| ASTM A490 bolts | 1 | Friction-type joints with standard holes and surfaces of | | | † |
| | | clean mill scale | .... | 22.0 (152) | |
| | | blast-cleaned carbon or alloy steel | .... | 34.5 (238) | |
| | | blast-cleaned inorganic zinc-rich paint | | 37.0 (255) | |
| ASTM SA193 Grade B7 at an operating temperature of | 2 | Used for bolts‡ | | | |
| −20°F | ... | ........................ | 18.8–25 (130–172) | | |
| +650°F | ... | ........................ | 18.8–25.0 (130–172) | | |
| +850°F | ... | ........................ | 16.3–17.0 (112–117) | | |
| +1000°F | ... | ........................ | 4.5 (31) | | |

**TABLE 22.1** Allowable Stresses (*Continued*)

| Material | Source | Comments | Allowable stress: Tension kpsi (MPa) | Shear kpsi (MPa) | Bearing† kpsi (MPa) |
|---|---|---|---|---|---|
| ASTM SA31 rivets | 3 | Used in SA515 plate | .... | 9 (62) | 18 (124) |
| ASTM A502-1 rivets | 3 | Used in A36 plate | | 13 (93) | 401 (276) |
| ASTM A36 joint material | 4 | | 22 (152) | 14.5 (100) | 48.6 (335) |
| 58-kpsi ultimate tensile steel: joint material | 5 | Joint length 25 in (with A325 bolts)<br>Joint length 80 in (with A325 bolts) | ....<br>.... | 23.2 (160)<br>29 (200) | |
| 100-kpsi ultimate tensile strength steel: joint material | 6 | Joint length 20 in (with A490 bolts)<br>Joint length 90 in (with A490 bolts) | ....<br>.... | 50 (345)<br>40 (276) | |
| ASTM A440 joint material | 7 | Based on a safety factor of 2.48:1 ($S_u/\sigma_T$) | 25.4–28.2 (175–194) | | |
| ASTM A514 joint material | 7 | Based on a safety factor of 2:00:1 ($S_u/\sigma_T$) | 50–65 (345–448) | | |
| ASTM A515 joint material | 3 | Stress in net section | 14 (95) | | |

†The allowable bearing stress for either A325 or A490 bolts is either $LS_u/2d$ or $1.5S_u$, whichever is least.

‡The stress allowed depends on the diameter of the bolts. The material cannot be through-hardened, so larger sizes will support less stress.

SOURCES:

1. "Structural Joints Using ASTM A325 or A490 Bolts." AISC specification, April 14, 1980, pp. 4–5.
2. "ASME Boiler and Pressure Vessel Code," Sec. VIII, Div. I, American Society of Mechanical Engineers, New York, 1977. Table UCS-23, pp. 208–209.
3. Archie Higdon, Edward H. Ohlsen, William B. Stiles, John A. Weese, and William F. Riley, *Mechanics of Materials*, 3d ed., John Wiley and Sons, New York, 1978, p. 632.
4. John W. Fisher, "Design Examples for High Strength Bolting," *High Strength Bolting for Structural Joints*, Bethlehem Steel Co., Bethlehem, Pennsylvania, 1970, p. 52.
5. John W. Fisher and John H. A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints*, John Wiley and Sons, New York, 1974, p. 124.
6. Ibid., p. 127.
7. Ibid., p. 123.

Unified: $$A_S = \frac{\pi}{4}\left(d - \frac{0.9743}{n}\right)^2$$

(22.4)

Metric: $$A_S = \frac{\pi}{4}(d - 0.9382P)^2$$

Here is an example based on Fig. 22.2. The bolts are ASTM A325 steel, $m = 5$ bolts, $F = 38\,250$ lb (170.1 kN), $d = ¾$ in (19.1 mm), $b = 2$ (one through the body of each bolt, one through the threads), and $n = 12$ threads per inch (2.12 mm per thread).

The total cross-sectional area through the bodies of all five bolts and then through the threads is

$$5A_B = \frac{5\pi(0.75)^2}{4} = 2.209 \text{ in}^2 \text{ (1425 mm}^2\text{)}$$

$$5A_S = \frac{5\pi}{4}\left(0.75 - \frac{0.9743}{12}\right)^2 = 1.757 \text{ in}^2 \text{ (1133 mm}^2\text{)}$$

The shear stress in each bolt will be

$$\tau = \frac{F}{A_T} = \frac{38\,250}{2.209 + 1.757} = 9646 \text{ psi (66.5 MPa)}$$

which is well within the shear stress allowed for A325 steel bolts (Table 22.1).

*Tensile Stress in the Plate.* To compute the tensile stress in the plates (we will assume that these are made of A36 steel), we first compute the cross-sectional area of a row containing the most bolts. With reference to Figs. 22.2 and 22.3, that area will be

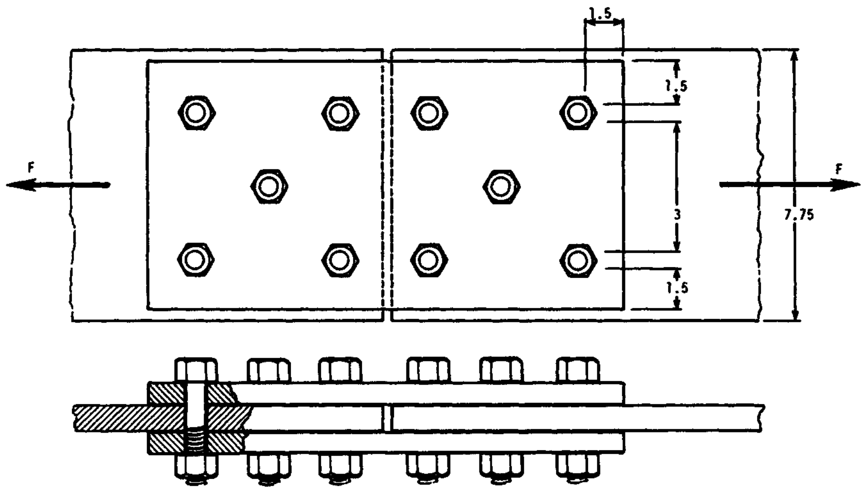



**FIGURE 22.2** Shear joint example. The joint and splice plates here are each ¾ in (19.1 mm) thick. Dimensions given are in inches. To convert to millimeters, multiply by 25.4.

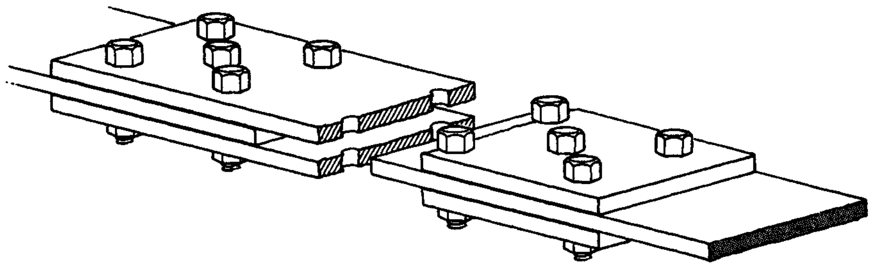


**FIGURE 22.3** Tensile failure of the splice plates. Tensile failure in the plates occurs in the cross section intersecting the most bolt holes.

$$A = 0.75(1.5) + 0.75(3) + 0.75(1.5) = 4.5 \text{ in}^2 \text{ (2903 mm}^2)$$

The stress in two such cross sections (there are two splice plates) will be

$$\sigma = \frac{F}{A} = \frac{38\,250}{(4.5)2} = 4250 \text{ psi (29.3 MPa)}$$

These plates will not fail; the stress level in them is well within the allowable tensile-stress value of 21.6 kpsi for A36 steel. In some joints we would want to check other sections as well, perhaps a section in the splice plate.

*Bearing Stresses on the Plates.* If the fasteners exert too great a load on the plates, the latter can be deformed; bolt holes will elongate, for example. To check this possibility, the designer computes the following (see Fig. 22.4):

$$\sigma_B = \frac{F}{mdl_G}$$

For our example, $l_G = 2.25$ in (57.2 mm), $m = 5$, and $d = 0.75$ in (19.1 mm). Then

$$\sigma_B = \frac{38\,250}{5(0.75)(2.25)} = 4533 \text{ psi (31.3 MPa)}$$

Note that the allowable bearing stresses listed in Table 22.1 are greater than the allowable shear stresses for the same plate material.

*Tearout Stress.* Finally, the designer should determine whether or not the fasteners will tear out of a joint plate, as in the lap joint shown in Fig. 22.5. In the example shown there are six shear areas. The shear stress in the tearout sections will be

$$\tau = \frac{100\,000}{6(0.75)(2)} = 11\,111 \text{ psi (76.6 MPa)}$$

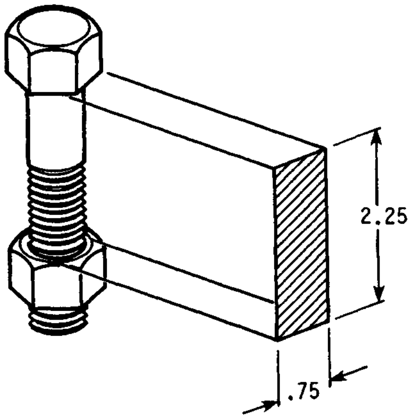



**FIGURE 22.4** The bearing area of a bolt. The dimensions given are those used in the example in the text for the joint shown in Fig. 22.2. Dimensions are in inches. Multiply by 25.4 to convert to millimeters.

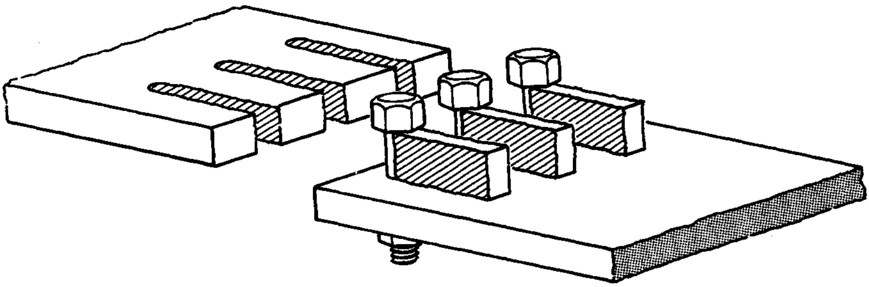


**FIGURE 22.5** Tearout. The pieces torn from the margin of the plate can be wedge-shaped as well as rectilinear, as shown here.

where $F$ = 100 kip (445 kN)
$H$ = 2 in (50.8 mm)
$t$ = ¾ in (19.1 mm)

***Friction-type Joints.*** Now let us design a friction-type joint using the same dimensions, materials, and bolt pattern as in Fig. 22.1, but this time preloading the bolts high enough so that the friction forces between joint members (between the so-called faying surfaces) become high enough to prevent slip under the design load.

*Computing Slip Resistance.* To compute the slip resistance of the joint under a shear load, we use the following expression (from Ref. [22.6], p. 72):

$$R_S = \mu_S F_{PA} bm \tag{22.5}$$

Typical slip coefficients are tabulated in Table 22.2. Note that engineering specifications published by the AISC and others carefully define and limit the joint surface conditions that are permitted for structural steel work involving friction-type joints. The designer cannot arbitrarily paint such surfaces, for example; if they are painted, they must be painted with an approved material. In most cases they are not painted. Nor can such surfaces be polished or lubricated, since these treatments would alter the slip coefficient. A few of the surface conditions permitted under current specifications are listed in Table 22.2. Further conditions and coating materials are under investigation.

To continue our example, let us assume that the joint surfaces will be grit blasted before use, resulting in an anticipated slip coefficient of 0.493. Now we must estimate the average preload in the bolts. Let us assume that we have created an average preload of 17 kip in each of the five bolts in our joint. We can now compute the slip resistance as

$$R_S = \mu_S F_{PA} bm = 0.493\ (17\,000)(2)(5)$$

$$= 83\,810 \text{ lb } (373 \text{ kN})$$

*Comparing Slip Resistance to Strength in Bearing.* The ultimate strength of a friction-type joint is considered to be the lower of its slip resistance or bearing strength. To compute the bearing strength, we use the same equations we used earlier. This time, however, we enter the allowable shear stress for each material and

**TABLE 22.2** Slip Coefficients

| Surfaces | Source | Typical slip coefficient $\mu_S$ |
|---|---|---|
| Free of paint or other applied finish, oil, dirt, loose rust or scale, burrs, or defects. Tight mill scale permitted | 1 | 0.45 |
| Clean mill scale | 2 | 0.35 |
| Hot dip galvanized | 2 | 0.16 |
| Hot dip galvanized, wire brushed | 2 | 0.3–0.4 |
| Grit blasted | 3 | 0.331–0.527 |
| Sand blasted | 3 | 0.47 |
| Metallized zinc sprayed (hot) onto grit blasted surface | 4 | 0.422 |

SOURCES:
1. Specification BS 4604: Part 1: 1970, British Standards Institution, London, 1970.
2. *High Strength Bolting for Structural Joints,* Bethlehem Steel Co., Bethlehem, Pennsylvania, 1970, p. 14.
3. John W. Fisher and John H. A. Struik, *Guide to Design Criteria for Bolted and Riveted Joints,* John Wiley and Sons, New York, 1974, p. 78.
4. Ibid., p. 200.

then compute the force which would produce that stress. These forces are computed separately for the fasteners, the net section of the plates, the fasteners bearing against the plates, and tearout. The least of these forces is then compared to the slip resistance to determine the ultimate design strength of the joint. If you do this for our example, you will find that the shear strength of the bolts determines the ultimate strength of this joint.

## 22.2 ECCENTRIC LOADS ON SHEAR JOINTS

### 22.2.1 Definition of an Eccentric Load

If the resultant of the external load on a joint passes through the centroid of the bolt pattern, such a joint is called an *axial shear joint.* Under these conditions, all the fasteners in the joint can be assumed to see an equal shear load.

If the resultant of the applied load passes through some point other than the centroid of the bolt group, as in Fig. 22.6, there will be a net moment on the bolt pattern. Each of the bolts will help the joint resist this moment. A joint loaded this way is said to be under an *eccentric shear load.*

### 22.2.2 Determine the Centroid of the Bolt Group

To locate the centroid of the bolt group, we arbitrarily position $xy$ reference axes near the joint, as shown in Fig. 22.7. We then use the following equations to locate the centroid within the group (Ref. [22.1], p. 360):

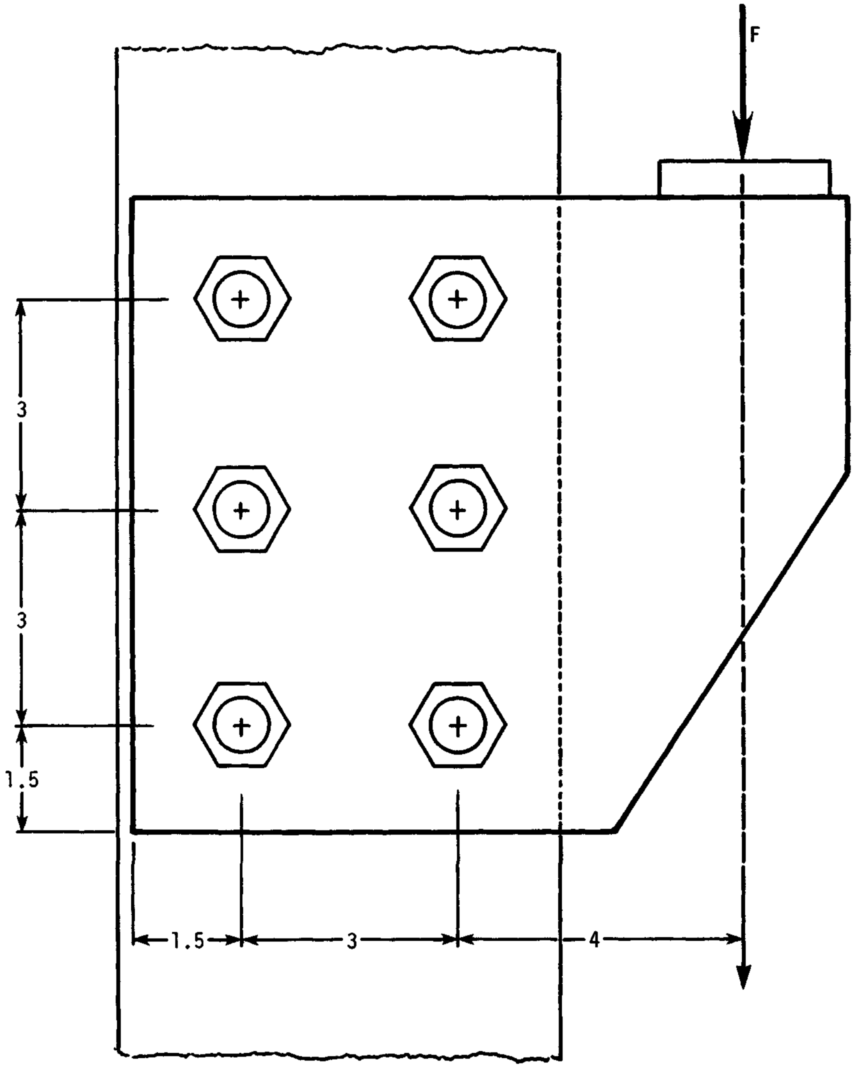



**FIGURE 22.6** Eccentrically loaded shear joint. For the example used in the text, it is assumed that the bolts are ¾—12 × 3, ASTM A325; the plates are made of A36 steel; the eccentric applied load $F$ is 38.25 kip (170 kN).

$$\bar{x} = \frac{A_1x_1 + A_2x_2 + \cdots + A_6x_6}{A_1 + A_2 + \cdots + A_6}$$

$$\bar{y} = \frac{A_1y_1 + A_2y_2 + \cdots + A_6y_6}{A_1 + A_2 + \cdots + A_6} \qquad (22.6)$$

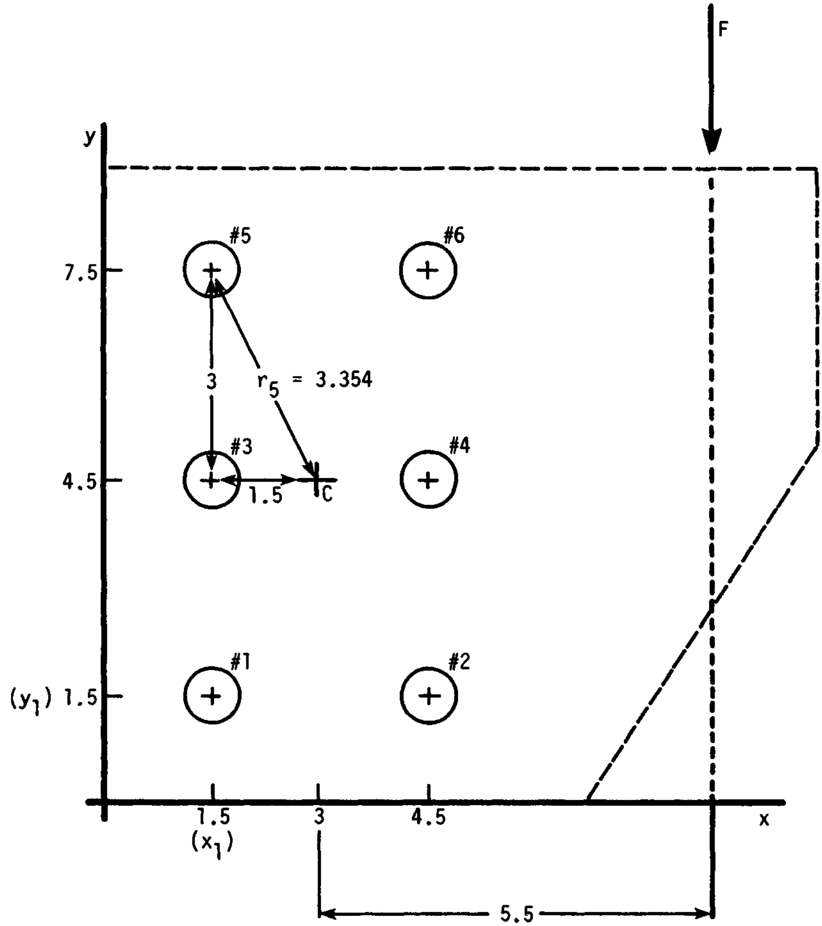



**FIGURE 22.7** The centroid of a bolt pattern. To determine the centroid of a bolt pattern, one arbitrarily positions coordinate axes near the pattern and then uses the procedure given in the text. I have used the edges of the splice plate for the $x$ and $y$ axes in this case. Multiply the dimensions shown (which are in inches) by 25.4 to convert them to millimeters.

For the joint shown in Fig. 22.6 we see, assuming that $A_1 = A_2 =$ etc. $= 0.442$ in$^2$ (285 mm$^2$),

$$\bar{x} = \frac{0.442(1.5 + 4.5 + 1.5 + 4.5 + 1.5 + 4.5)}{6(0.442)} = 3 \text{ in } (76.2 \text{ mm})$$

Similarly, we find that $\bar{y} = 4.5$ in (114.3 mm).

### 22.2.3 Determining the Stresses in the Bolts

***Primary Shear Force.*** We compute the primary shear forces on the fasteners as simply (see Fig. 22.8)

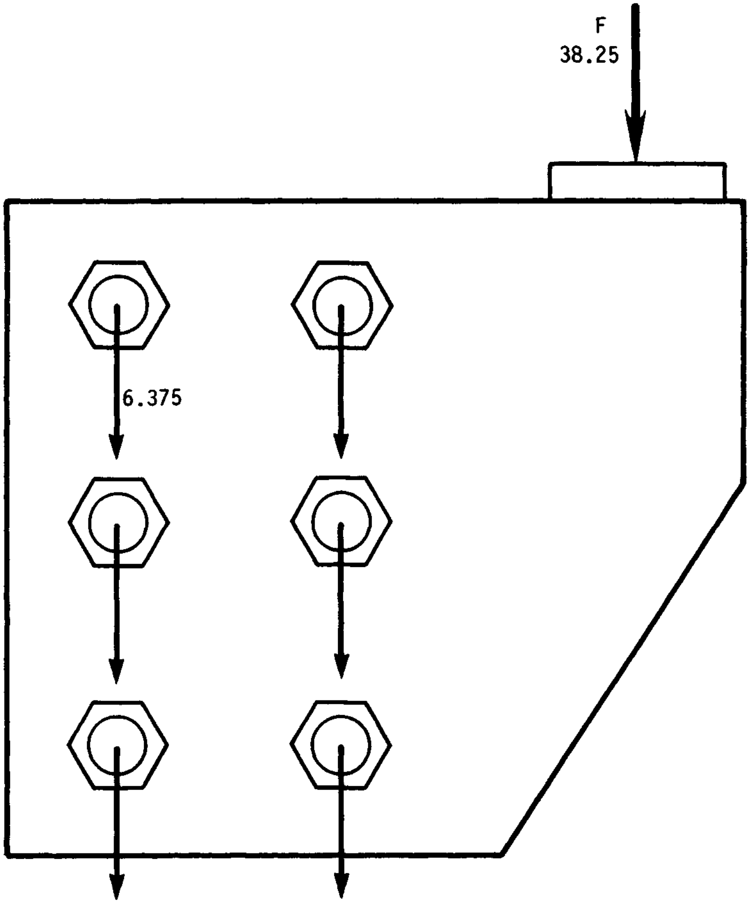



**FIGURE 22.8** Primary shear forces on the bolts. The primary forces on the bolts are equal and are parallel. Forces shown are in kilopounds; multiply by 4.448 to convert to kilonewtons.

$$F_b = \frac{F}{m} = \frac{38\,250}{6} = 6375 \text{ lb (28.4 kN)}$$

***Secondary Shear Forces.*** We next determine the reaction moment forces in each fastener using the two equations (Ref. [22.1], p. 362):

$$M = F_1 r_1 + F_2 r_2 + \cdots + F_6 r_6 \tag{22.7}$$

$$\frac{F_1}{r_1} = \frac{F_2}{r_2} = \frac{F_3}{r_3} = \cdots = \frac{F_6}{r_6} \tag{22.8}$$

Combining these equations, we determine that the reaction force seen on a given bolt is

$$F_n = \frac{Mr_n}{r_1^2 + r_2^2 + \cdots + r_6^2} \tag{22.9}$$

Let us continue our example. As we can see from Fig. 22.7, we have an external force of 38 250 lb (170.1 kN) acting at a distance from the centroid of 5.5 in (140 mm). The input moment, then, is 210 kip · in (23.8 N · m). The radial distance from the centroid to bolt 5 (one of the four bolts which are most distant from the centroid) is 3.354 in (85.2 mm). The reaction force seen by each of these bolts is (see Fig. 22.9)

$$F_5 = \frac{210\,375(3.354)}{4(3.354)^2 + 2(1.5)^2} = 14\,255 \text{ lb } (63.4 \text{ kN})$$

***Combining Primary and Secondary Shear Forces.*** The primary and secondary shear forces on bolt 5 are shown in Fig. 22.9. Combining these two forces by vectorial means, we see that the total force $F_{R5}$ on this bolt is 12 750 lb (56.7 kN).

Let us assume that there are two slip planes here—that one of them passes through the body of the bolt and the other passes through the threads as in the ear-

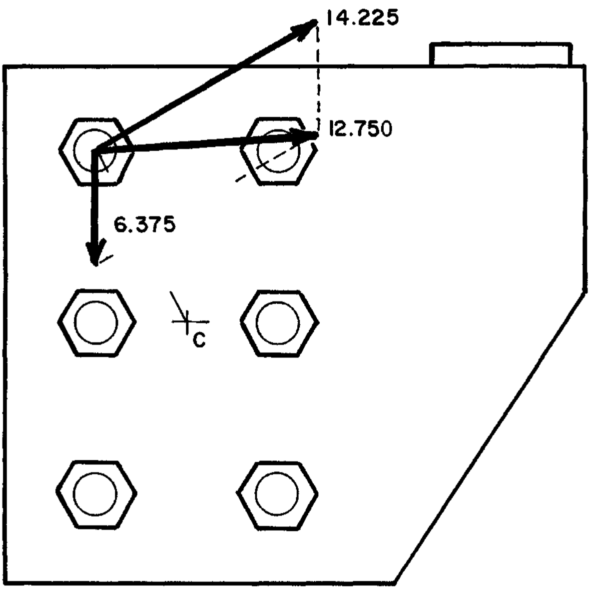



**FIGURE 22.9** Combining primary and shear forces. I have selected one of the four *most distant* bolts to calculate the secondary shear force, 14.255 kip (63.4 kN), which has a line of action at right angles to the radial line connecting the bolt to the centroid. The resultant of primary and secondary forces on this bolt is 12.750 kip (56.7 kN).

lier example illustrated in Fig. 22.2. The shear area of bolt 5, therefore, is (see Sec. 22.1.2 for the equations) 0.793 in$^2$ (511 mm$^2$).

We can now compute the shear stress within this bolt:

$$\tau = \frac{F_{R5}}{A_5} = \frac{12\,750}{0.793} = 16\,078 \text{ psi}$$

This is less than the maximum shear stress allowed for A325 steel bolts (see Table 22.1), and so the design is acceptable.

It is informative to compare these results with those obtained in Sec. 22.1.3, where we analyzed a joint having similar dimensions, the same input load, and one less bolt. The axial load in the earlier case created a shear stress of only 9646 psi (66.5 MPa) in each bolt. When the same load is applied eccentrically, passing 5.5 in from the centroid, it creates 16 078/9646 times as much stress in the most distant bolts, even though there are more bolts this time to take the load. Be warned!

## 22.3 TENSION-LOADED JOINTS: PRELOADING OF BOLTS

In the joints discussed so far, the bolts or rivets were loaded in shear. Such joints are usually encountered in structural steel work. Most other bolted joints in this world are loaded primarily in tension—with the applied loads more or less parallel to the axis of the bolts.

The analysis of tension joints usually centers on an analysis of the tension in the fasteners: first with the initial or *pre*load in the fasteners when they are initially tightened, and then with the working loads that exist in the fasteners and in the joint members when external forces are applied to the joint as the product or structure is put into use. These working loads consist of the preload plus or minus some portion of the external load seen by the joint in use.

Because clamping force is essential when a joint has to resist tension loads, rivets are rarely used. The following discussion, therefore, will focus on bolted joints. The analytical procedure described, however, could be used with riveted joints if the designer is able to estimate the initial preload in the rivets.

### 22.3.1 Preliminary Design and Calculations

***Estimate External Loads.*** The first step in the design procedure is to estimate the external loads which will be seen by each bolted joint. Such loads can be static, dynamic, or impact in nature. They can be created by weights such as snow, water, or other parts of the structure. They can be created by inertial forces, by shock or vibration, by changes in temperature, by fluid pressure, or by prime movers.

***Fastener Stiffness.*** The next step is to compute the stiffness or spring rate of the fasteners using the following equation:

$$k_B = \frac{A_S A_B E}{L_S A_B + L_B A_S} \tag{22.10}$$

***Example.*** With reference to Fig. 22.10, $A_S = 0.232$ in$^2$ (150 mm$^2$), $L_B = 2.711$ in (68.9 mm), $A_B = 0.307$ in$^2$ (198 mm$^2$), $E = 30\ (10)^6$ psi (207 GPa), and $L_S = 1.024$ in (26 mm). Thus

$$k_B = \frac{0.232(0.307)(30 \times 10^6)}{1.024(0.307) + 2.711(0.232)} = 2.265 \times 10^6 \text{ lb/in } (0.396 \text{ N/mm})$$

***Stiffness of a Nongasketed Joint.*** The only accurate way to determine joint stiffness at present is by experiment. Apply an external tension load to a fastener in an actual joint. Using strain gauges or ultrasonics, determine the effect which this external load has on the tension in the bolt. Knowing the stiffness of the bolt (which must be determined first), use joint-diagram techniques (which will be discussed soon) to estimate the stiffness of the joint.

Although it is not possible for me to give you theoretical equations, I can suggest a way in which you can make a rough estimate of joint stiffness. This procedure is based on experimental results published by Motosh [22.7], Junker [22.8], and Osgood [22.9], and can be used only if the joint members and bolts are made of steel with a modulus of approximately $30 \times 10^6$ psi (207 GPa).

First compute the slenderness ratio for the bolt ($l_G/d$). If this ratio is greater than 1/1, you next compute a stiffness ratio $R_{JB}$ using the empirical equation

$$R_{JB} = 1 + \frac{3(l_G)}{7d} \tag{22.11}$$

The final step is to compute that portion of the stiffness of the joint which is loaded by a single bolt from

$$k_J = R_{JB} k_B$$

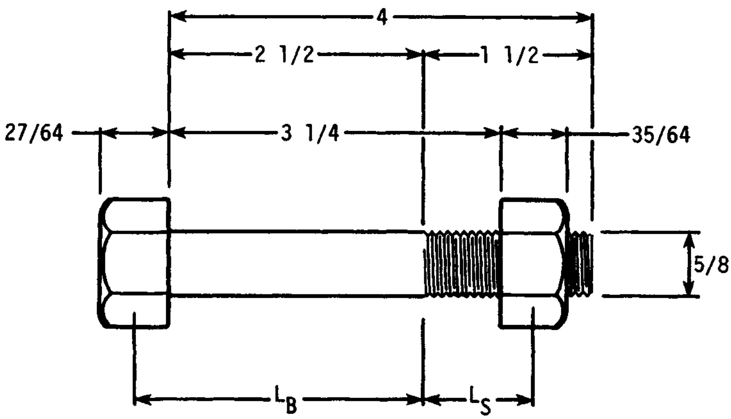



**FIGURE 22.10** Computing the stiffness of a bolt. The dimensions given are those used in the example in the text. This is a ⅝—12 × 4, SAE J429 Grade 8 hexagon-head bolt with a 3.25-in (82.6-mm) grip. Other dimensions shown are in inches. Multiply them by 25.4 to convert to millimeters.

If the slenderness ratio $l_G/d$ falls between 0.4 and 1.0, it is reasonable to assume a stiffness ratio $R_{JB}$ of 1.0. When the slenderness ratio $l_G/d$ falls below 0.4, the stiffness of the joint increases dramatically. At a slenderness ratio of 0.2, for example, $R_{JB}$ is 4.0 and climbing rapidly (Ref. [22.6], pp. 199–206).

***Example.*** For the bolt shown in Fig. 22.10 used in a 3.25-in (82.6-mm) thick joint,

$$R_{JB} = \frac{3(3.25)}{7(0.625)} = 3.23$$

Since we computed the bolt stiffness earlier as $2.265 \times 10^6$ lb/in (396 kN/mm), the joint stiffness will be

$$k_J = 3.23(2.265 \times 10^6) = 7.316 \times 10^6 \text{ lb/in (1280 kN/mm).}$$

***Stiffness of Gasketed Joints.*** The procedure just defined allows you to determine the approximate stiffness of a nongasketed joint. If a gasket is involved, you should use the relationship

$$\frac{1}{k_T} = \frac{1}{k_J} + \frac{1}{k_G} \tag{22.12}$$

You may have to determine the compressive stiffness of the gasket by making an experiment or by contacting the gasket manufacturer, since very little information has been published on this subject (but see Chap. 25). A few typical values for pressure-vessel gasket materials are given in Table 22.3, but these values should be used for other gaskets with caution.

Note that the stiffness of a gasket, like the stiffness of everything else, depends on its cross-sectional area. The values given in Table 22.3 are in terms of pressure or stress on the gasket versus deflection, not in terms of force versus deflection. Before you can combine gasket stiffness with joint stiffness, therefore, you must determine how large an area of the gasket is loaded by a single bolt (total gasket area divided by the number of bolts). This per-bolt area is multiplied by stress to determine the stiffness in terms of force per unit deflection. For example, the compressed asbestos gasket listed in Table 22.3 has a total surface area of 11.2 in$^2$ (7219 mm$^2$). If it is clamped by eight bolts, the per-bolt area is 1.4 in$^2$ (903 mm$^2$). The stiffness is listed in Table 22.3 as $6.67 \times 10^2$ ksi/in (181 MPa/mm). In force terms, per bolt, this becomes $6.67 \times 10^5$ (1.4) = $9.338 \times 10^5$ lb/in ($1.634 \times 10^2$ kN/mm).

The stiffness values given in Table 22.3 are for gaskets in use, after initial preloading. Gaskets exhibit a lot of hysteresis. Their stiffness during initial compression is a lot less (generally) than their stiffness as they are unloaded and reloaded. As long as the usage cycles do not take the stress on the gasket above the original assembly stress, their behavior will be repetitive and elastic, with only a little hysteresis, as suggested by Fig. 22.11. And when analyzing the effect of loads on joint behavior, we are interested only in how the gaskets act as they are used, not in their behavior during assembly.

### 22.3.2 Selecting the Target Preload

Our joint will perform as intended only if it is properly clamped together by the fasteners. We must, therefore, select the preload values very carefully.

**TABLE 22.3** Gasket Stiffness

| Source | Gasket | Dimensions, in (mm) | | | | Stiffness kpsi/in (MPa/mm) |
|---|---|---|---|---|---|---|
| | | ID | OD | w | t | |
| 1 | Spiral-wound, asbestos-filled (300-lb class) | 5 (127) | 5.75 (146) | 0.375 (9.52) | 0.175 (4.45) | $4.71 \times 10^2$ (127.6) |
| 1 | Spiral-wound asbestos-filled (600-lb class) | 4.75 (121) | 5.75 (146) | 0.5 (12.7) | 0.175 (4.45) | $6.95 \times 10^2$ (188.3) |
| 1 | Compressed asbestos | 4 (102) | 5.5 (140) | 0.75 (19) | 0.062 (1.59) | $6.67 \times 10^2$ (180.7) |
| 2 | Flat stainless steel double-jacketed asbestos-filled | 6.5 (191) | 7.5 (216) | 0.5 (12.7) | 0.125 (3) | $43.3 \times 10^2$ (1176) |
| 2 | Solid oval ring-style 950 soft iron | 5.438 (138) | 6.314 (160) | 0.469 (9.7) | 0.688 (14.3) | $27.5 \times 10^2$ (747) |

SOURCE:
1. H. D. Raut, André Bazergui, and Luc Marchand, "Gasket Leakage Behavior Trends: Results of 1977–79 PVRC Exploratory Gasket Test Program," *Welding Research Council Bulletin* no. 271, WRC, New York, October 1981, Figs. 16 and 18.
2. André Bazergui and Luc Marchand, "PVRC Milestone Gasket Tests—First Results," report submitted to the Special Commission on Bolted Flanged Connections of the Pressure Vessel Research Committee of the Welding Research Council, September 1982, Figs. 12 and 13.

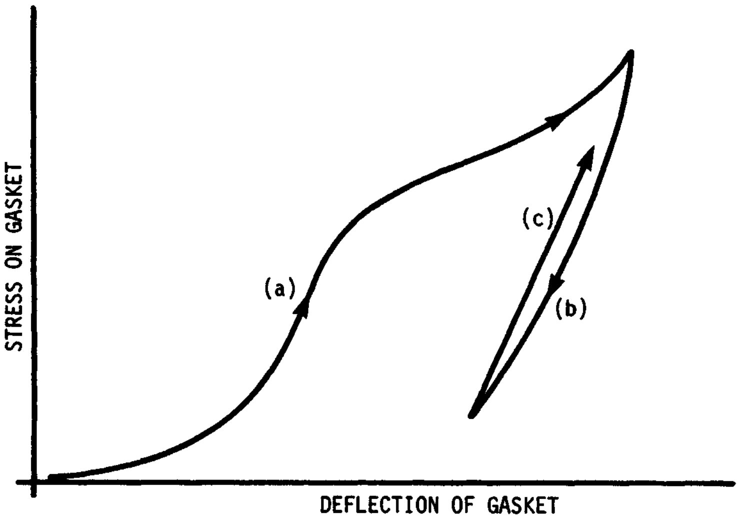



**FIGURE 22.11** Typical stress versus deflection characteristics of a spiral-wound, asbestos-filled gasket during (*a*) initial loading, (*b*) unloading, and (*c*) reloading.

***Acceptable Upper Limit for the Tension in the Bolts.*** In general, we always want the greatest preload in the bolts which the parts (bolts, joint members, and gasket) can stand. To determine the maximum acceptable tension in the bolt, therefore, we start by determining the yield load of each part involved in terms of bolt tension. The force that will cause the bolt material to yield is

$$F_{BY} = S_{YB} A_S \tag{22.13}$$

Let us begin an example using the joint shown in Fig. 22.12. We will use the bolt illustrated in Fig. 22.10. Let us make the joint members of ASTM A441 steel. The yield strength of our J429 Grade 8 bolts is 81 kpsi (558 MPa), worst case. For the bolts, with $A_S = 0.232$ in$^2$ (150 mm$^2$),

$$F_{BY} = 81 \times 10^3\ (0.232) = 18.8 \times 10^3 \text{ lb } (83.6 \text{ kN})$$

For the joint, we determine the yield load of that portion of the joint which lies under the head of the bolt or under the washer (using the distance across flats of the head or nut to compute the bearing area). If our joint material is ASTM A441 steel

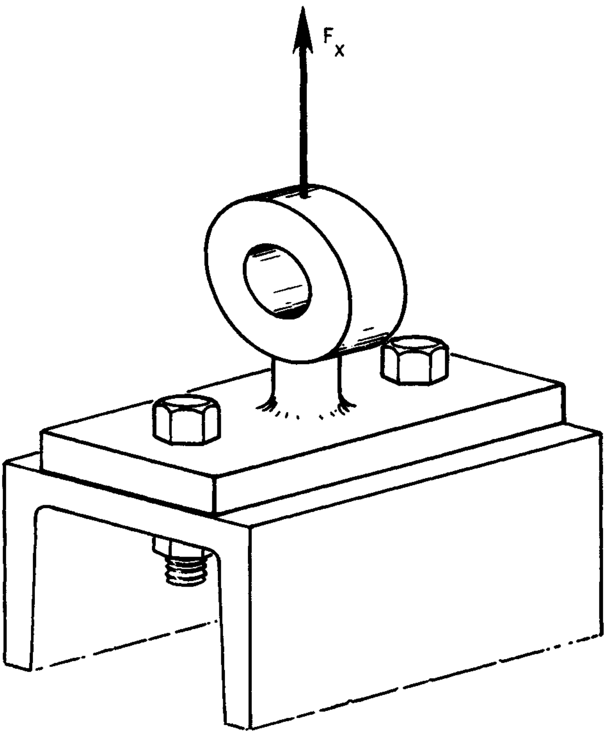



**FIGURE 22.12** Joint loaded in tension. This is the joint analyzed in the text. The bolts used here are those shown in Fig. 22.10.

with a yield strength of 40 kpsi (276 MPa), we would find that a *bolt* force of 27.6 kip (122 kN) would yield the joint. We take the lesser of the joint or bolt yield loads, $18.8 \times 10^3$ lb (83.6 kN) as the yield load of the system.

Next, since we are planning to tighten these fasteners by applying torque to the nuts, we subtract 10 percent from the yield strength of the fastener to allow for the torsional stresses which will be developed in the fastener as it is tightened. If we were planning to use a high-pressure lubricant on the threads, we might subtract only 2 to 4 percent for torsion. Since no lube will be used in our example, we subtract 10 percent, making the upper limit $16.9 \times 10^3$ lb (75.2 kN).

This will remain our upper limit unless the fasteners will also be subjected to shear stress; or unless code limits, stress corrosion problems, or the desire for a safety factor forces us to reduce it further. We will assume, in our example, that they do not.

Before continuing, however, let us see what we would have to do if the bolts *did* see a combined tension and shear load. This might happen, for example, in a bearing-type joint in which we planned to preload (tension) the bolts to a significant percentage of yield. There are other types of joint, of course, which are subjected to both tensile and shear loads in use. Any shear load on the bolt will "use up" part of the strength of the bolt, leaving less capacity for tensile loads (Ref. [22.6], p. 226).

Under these conditions, the maximum acceptable tensile stress in a bolt can be determined using any of the static failure theories of Chap. 28. Here, we select the equation

$$\left(\frac{\lambda}{S}\right)^2 + \phi^2 = 1 \qquad (22.14)$$

where $S$ = the ratio of the shear strength to the tensile strength (typically 0.6) for bolt steels. Equation (22.14) is a form of the maximum-shear-stress theory.

***Acceptable Lower Limit for the Clamping Force on the Joint.*** When we are computing the maximum acceptable forces, we focus on the joint, because its behavior can be seriously affected if the interface forces become too small. The joint, for example, might leak, it might vibrate loose, or it might have a short fatigue life. To determine the lower acceptable limit, we must consider each potential failure mode separately, estimate the minimum preload required to control that particular problem, and then select the *highest* of these several minimum requirements to establish the minimum for the system.

This is one of the more difficult steps of our procedure. In fact, we may be able to determine the acceptable minimum preload only by making fatigue or vibration tests or the like. (We will consider fatigue problems at length in Sec. 22.5.) There are some rules of thumb, however, which we can apply.

If our joint is a friction-type shear joint, or if it will be subjected to transverse vibration, we want a minimum preload which will prevent joint slip under the maximum anticipated shear load. This load is

$$F_C(\min) \geq \frac{F_X}{S} \qquad (22.15)$$

If we are dealing with a foundation bolt or something where it is only necessary to avoid separation of the joint members, the minimum acceptable preload can be zero. If we are dealing with a gasketed joint, we will have to worry about the minimum acceptable gasket pressure required to keep that gasket from leaking.

***Example.*** Let us assume for our ongoing example that we are concerned only about separation of the joint members. Minimum acceptable clamping force, therefore, is zero.

***Select an Initial Preload Target.*** We now, rather arbitrarily, select an initial preload target that is somewhere between the acceptable minimum and acceptable maximum bolt tensions which we computed earlier. Let us try 60 percent of the acceptable maximum of 16.9 kip (75.2 kN) or, in our example, $10.1 \times 10^3$ lb (45.1 kN).

### 22.3.3 Estimating Actual Upper Limit on Bolt Tension

We must now determine whether or not the tension we will actually develop in any bolt will exceed the maximum acceptable tension, given our preliminary target preload and a consideration of the tools, lubricants, and procedures we are planning to use during assembly. We must also consider the effects of the external tension loads which will be placed on this joint after assembly.

***Tool Errors.*** We can select many different types of assembly tools. Each choice carries with it certain accuracy implications; some tools can produce preload in the fasteners with far greater precision than can other tools. Table 22.4 lists some of the many possibilities. We will assume for our example that we are going to use a manual torque wrench and must face a potential scatter in preload for a given torque of ±30 percent.

***Operator Problems.*** Even if we used perfect tools, we would see some scatter in the resulting preload because of operator problems. Are the operators skilled, properly trained, or tired? Do they care about their work? Are the bolts readily accessible?

Let us assume for purposes of our example that the operators will contribute an estimated ±10 percent additional scatter in preload. We do not just add this 10 percent to the 30 percent we assigned to the torque wrench when we assess the combined impact of tools and operators. We use the statistician's method for combining the variances of two variables, as follows:

$$\sigma^2 = \sigma_T^2 + \sigma_O^2 \tag{22.16}$$

In our example this suggests that the combined variance will be

$$\sigma^2 = 30^2 + 10^2 = 1000$$

giving us a ±3 sigma "deviation" (square root of the variance) of ±31.6 percent.

We have selected a target preload of $10.1 \times 10^3$ lb. Consideration of tool and operator scatter gives us

$$F_P(\text{max}) = F_{PT} + 0.316 F_{PT} = 13.3 \times 10^3 \text{ lb (59.2 kN)}$$

$$F_P(\text{max}) = F_{PT} - 0.316 F_{PT} = 6.91 \times 10^3 \text{ lb (30.7 kN)}$$

***Effects of External Tension Load.*** Now let us see what happens when an external tension load is placed on the preloaded joint. Although it is difficult to do this in

**TABLE 22.4** Tool Accuracy

| Control parameter and type of tool | Source | Reported scatter in preload, % |
|---|---|---|
| *Torque* control with | | |
| 1. Power wrench | 1 | ±23 to 28 |
| 2. Hand wrench | 2 | ±21 to 81 |
| 3. Hand wrench | 3 | −20. 4 to 99 |
| 4. Hand wrench plus torque multiplier | 4 | ±70 to150 |
| 5. Dial or click wrench | 4 | ±60 to 80 |
| 6. Wrench with electronic readout | 4 | ±40 to60 |
| 7. Hand wrench | 2 | ±12 |
| 8. Air-powered tool with torque feedback | 5 | ±20 |
| 9. Hand wrench in laboratory conditions | 6 | ±30 |
| 10. Air tool with one shot clutch | 7 | ±30 |
| 11. Stall torque air tool | 7 | ±35 |
| 12. Hand wrench | 7 | ±30 |
| *Torque-turn* control with | | |
| 13. Yield control system (computer-controlled air tool) | 8 | ±8 |
| 14. Turn-of-the-nut procedure used in structural steel work | 9 | ±15 |
| 15. Logarithmic rate method (LRM†) controlled air tool | 11 | ±2.2 to 2.6 |
| 16. Turn-of-the nut | 6 | ±15 |
| *Miscellaneous* methods | | |
| 17. Strain-gauged load washers | 10 | ±15 |
| 18. Strain-gauged bolts | 10 | ±1 |
| 19. Swaged lockbolts | 10 | ±5 |
| 20. Bolt heaters | 12 | ±15 |
| 21. Air-powered impact wrench | 5 | ±50 |
| 22. Manual slug wrench | 3 | −48 to + 50 |
| 23. Air-powered impact wrench | 4 | −300 to +150 |
| 24. Hydraulic tensioners controlled by "large vernier gauge readout" | 4 | ±20 |
| 25. Torque-time control on air tool | 5 | ±11 |
| 26. Operator feel | 6 | ±35 |
| 27. Fastener elongation | 6 | ±3 to 5 |
| 28. Ultrasonic control of preload | | ±1 to10 |

†Trademark of Rockwell International

SOURCES:

1. Robert J. Finkelston and P. W. Wallace, "Advances in High Performance Mechanical Fastening," SAE Paper No. 800451, 1980, p. 6.
2. Results of tests conducted privately.
3. Roly Laird, "The Nuts and Bolts of Preventing Failure Where Pressure Is Hot," *Engineer* (GB), vol. 245, no. 6330, July 21, 1977, p. 37.
4. Steven F. Aaronson, "Analyzing Critical Joints," *Machine Design*, January 21, 1982, p. 95.
5. *Investigation of Threaded Fastener Structural Integrity*, Southwest Research Institute, San Antonio, Texas, October 1977, p. 3.
6. *Fastener Standards*, 5th ed., Industrial Fastener Institute, Cleveland, Ohio, 1970, p. N-12.
7. Edwin Rodkey, "Making Fastened Joints Reliable—Way to Keep 'em Tight," *Assembly Engineering*, March 1977, pp. 24–27.
8. Robert J. Finkeltson, "Optimized Bolt Tightening Takes to the Field," SPS Laboratories, Jenkintown, PA., March 1981.
9. E. Donald, "Fatigue-Indicating Fasteners; A New Dimension in Quality Control," *Fastener Technology*, March 1979.
10. Larry D. Mercer, "How Swaged Lockbolts Optimize Fastener Preload," National Design Engineering Conference, Chicago, Ill., 1982.
11. S. Eshghy, "Tension by Ultrasonic Stretch," privately published, June 1982.
12. Carl Osgood, "How Elasticitiy Influences Bolted-Joint Design," *Machine Design*, March 1972, p. 106.

practice, we usually assume that a tension load is applied between the head of the fastener and the nut or tapped hole at the other end. Such a load would have a *worst-case effect* on the tension in the fasteners and on the clamping force on joint members, so this assumption is a safe and conservative one.

Any such tension load, no matter how small, will add to the tension in the bolts, increasing the length of the bolts slightly and thereby reducing the clamping force on the joint interface.

Not all the external load applied to the bolt is seen by the bolt, however. Part of the external load merely replaces part of the outward force which the joint initially exerted on the bolts that were clamping it. This can be illustrated by what engineers call a *joint diagram,* such as that shown in Fig. 22.13. Note that the external load applied to the bolt is equal to the sum of the changes which occur in the bolt and joint. It is equal, in other words, to the increase in tension in the fasteners plus the decrease in compressive load in the joint. We say that one part of the external load has been "absorbed" by the bolts; the rest has been absorbed by the joint (Ref. [22.6], pp. 199*ff*).

The relative stiffness or spring rate of bolt and joint determines how much of the load each will absorb. In Fig. 22.13 the joint stiffness is 3.23 times that of the bolt, as determined by our previous calculation of the stiffness ratio $R_{JB}$ for the bolt in Fig. 22.10. The joint, therefore, will absorb approximately seven-tenths of any applied external tension load.

We should note in passing that the effects of an external *compressive* load can also be illustrated by a joint diagram, such as that shown in Fig. 22.14. This time the tension in the bolt is reduced and the compression in the joint is increased simultaneously by the single external load. The portions absorbed by fasteners and joint are again proportional to their relative stiffness.

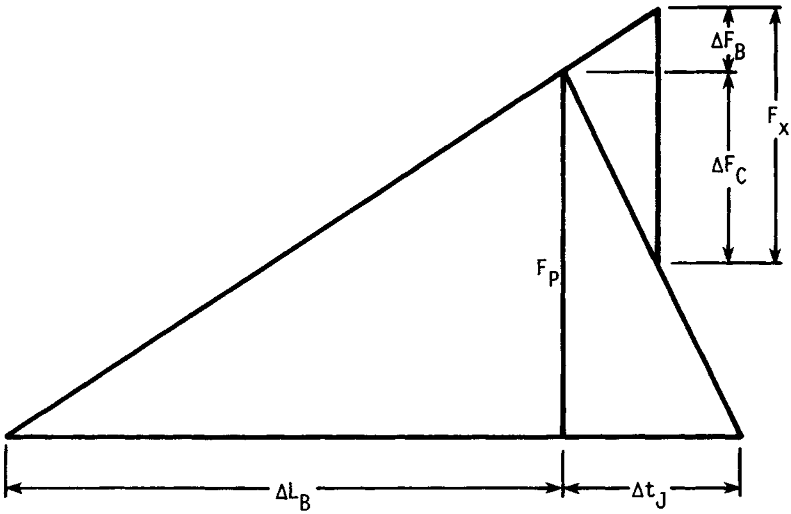



**FIGURE 22.13** Joint diagram for a joint loaded in tension. A joint diagram consists of two force-elongation diagrams, one for the bolt and one for the joint material loaded by that bolt put front-to-front (Ref. [22.6], pp. 199–206). It illustrates the combined elastic behavior of bolt and joint.

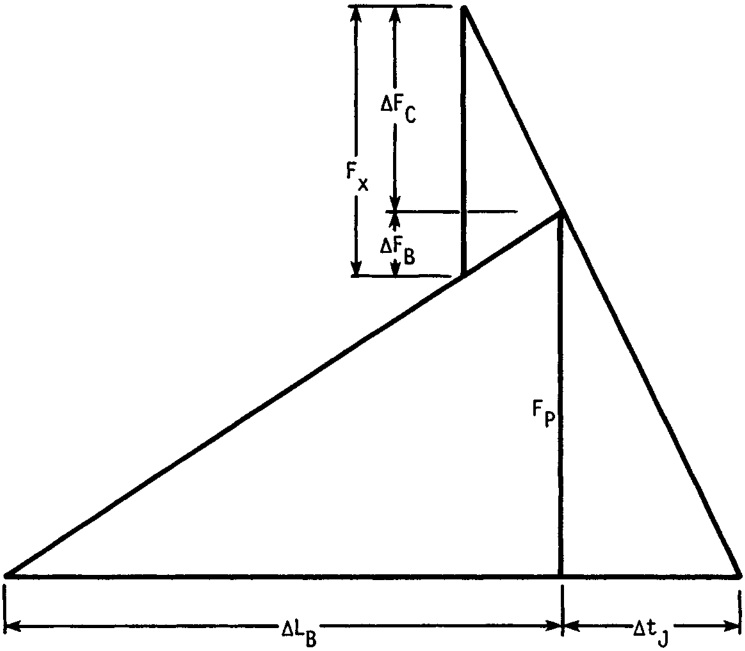



**FIGURE 22.14** Joint diagram for a joint loaded in compression.

Note that *any* external compression or tension load will alter both the tension in the fasteners and the compression in the joint. This is contrary to the widely held belief that there will be no such change in either member until and unless the external tensile load exceeds the preload in magnitude.

If the external tensile load exceeds the initial preload, then all clamping force will have been removed from the joint and the bolts will see, in its entirety, any additional tension load placed on that joint.

One nice feature of the joint diagram is that it allows us to derive expressions which define joint behavior, such as the following:

$$k_B = \frac{F_P}{\Delta L_B} \qquad k_J = \frac{F_P}{\Delta t_j} \tag{22.17}$$

$$\Delta F_B = F_x\left(\frac{k_B}{k_J + k_B}\right) \tag{22.18}$$

$$\Delta F_C = F_x\left(1 - \frac{k_B}{k_J + k_B}\right) \tag{22.19}$$

Let us continue the example we started in Fig. 22.12 by assuming an external tensile load of 10 kip (44.5 kN) per bolt has been placed on the system and computing the estimated effect of the external load on the bolt tension:

$$F_B = 5000\left[\frac{2.265 \times 10^6}{(7.316 + 2.265) \times 10^6}\right] = 1182 \text{ lb } (5.26 \text{ kN})$$

***Estimated Maximum Tension in the Bolts.*** We can now combine all the effects which we have studied to determine the maximum anticipated tension in the bolts under the worst-case situation:

$$F_B(\max) = F_P(\max) + \Delta F_B$$

$$F_B(\max) = 13.3 \times 10^3 + 1.182 \times 10^3 = 14.5 \times 10^3 \text{ lb } (64.5 \text{ kN})$$

So our anticipated maximum is less than the acceptable maximum preload of $16.9 \times 10^3$ lb (75.2 kN). We can therefore continue with our analysis. Note that if the anticipated maximum had exceeded the acceptable maximum, we would have had to lower our target preload somewhat and try again.

### 22.3.4 Estimating Actual Lower Limit on the Clamping Force

To determine the lower limit on clamping force, we follow a procedure similar to that used for determining the maximum tension to be expected in the fasteners, this time subtracting, not adding, from target preload the tool and operator scatter and that portion of the external load which reduces the clamping force on the joint members ($\Delta F_C$ in Fig. 22.13). When considering the lower limit, we must also consider one other effect: the short-term relaxation of the joint following or during initial tightening.

***Relaxation Effects.***
*Embedment Relaxation.* When joint and fastener are first assembled, especially if we are dealing with new parts, they contact each other only on the microscopically rough high spots of their contact surfaces. These high spots will be loaded past the yield point and will creep and flow when first placed under load until enough additional surface area has been brought into play to stabilize the process. Other plastic flow can occur in thread roots, in the head-to-body fillet, and even in some whole threads, causing further relaxation. All these effects are lumped together under the term *embedment relaxation.* Typically they will lead to a loss of 2 to 10 percent of initial preload in the first few seconds or minutes after initial tightening. Let us assume a 5 percent loss of our ongoing example.

*Elastic Interactions in the Joint.* Achieving perfect initial preload is not our only problem when we tighten a multibolt joint. When we tighten the first fasteners, we stretch them and partially compress the joint. When we subsequently tighten other fasteners in the same joint, we further compress the joint. Because this act allows previously tightened fasteners to contract a little, it reduces the tension in those fasteners even if we achieved perfect *initial* preload in each when we first tightened it.

The effect a fastener has on a previously tightened fastener is illustrated in Fig. 22.14. The new fastener applies a compressive load to the joint which had been previously preloaded by the first bolt.

***Estimating Minimum Clamping Force.*** Elastic interactions can be a special problem in large-diameter and/or gasketed joints. Since our example is neither, and involves only two bolts, we will assume that these interactions cost us, worst case, only 25 percent of initial preload in that bolt tightened first. Worst-case minimum clamping force in the vicinity of that bolt, therefore, becomes

$$F_C(\min) = F_P(\min) - F_X\left(1 - \frac{k_B}{k_J + k_B}\right) - F_P(\min)P_Z$$

$$F_C(\min) = 6.91 \times 10^3 - 5000\left[1 - \frac{2.265 \times 10^6}{(7.316 + 2.265) \times 10^6}\right] \tag{22.20}$$

$$- 6.91 \times 10^3(0.05 + 0.25) = 1019 \text{ lb } (4.53 \text{ kN})$$

So our computed anticipated minimum is greater than the zero minimum acceptable clamping force determined earlier. If it had not been acceptable, we would, of course, have had to readjust our target preload and/or select more accurate tools and/or revise our design.

### 22.3.5 Achieving a Desired Preload

In most cases we will try to achieve our target preload by using a wrench of some sort to apply torque to the nut or to the head of the bolt, accepting the resulting scatter in preload which this implies. We will discuss the torque-preload relationship at length in the next section. First, however, let us take a brief look at some of our other options for tightening bolts.

***Hydraulic Tensioners and Bolt Heaters.*** On large bolts we do not have to use a wrench. We can use a hydraulic tensioner which exerts a pure tension on the bolt, grabbing a few threads which stick out past the nut. Once the tool has stretched the bolt by the desired amount, the nut is run down to retain the tension. Then the tool lets go. At first glance this sounds like a perfect answer to some of the torquing and friction uncertainties we shall consider in Sec. 22.4, but there are other problems. The amount of preload retained by the fastener's nut is never the same as the preload introduced by the tool, because the nut must embed itself in the joint to pick up the loads originally supported by the much larger feet of the tensioner. This *elastic recovery loss* can equal 10 to 80 percent of the initial tension, depending on whether the fastener is relatively long (smaller loss) or short, and depending on how much torque was applied to the nut when it was run down (more torque, less loss).

Hydraulic tensioners, however, are superb tools when it comes to preloading large fasteners. They can be gang-driven from a single hydraulic pump and so can tighten several fasteners simultaneously with the same initial tension in each. This can be a very important feature when you are tightening large joints, especially if they are gasketed.

Tensioners also eliminate the galling problems often encountered when we attempt to torque large fasteners (3 in in diameter or more). The male and female threads are not turned relative to each other under heavy contact pressure with the tensioner. So tensioners have a place, but they do not provide perfect control of bolt preload.

Another nonwrench sometimes used to preload large fasteners is a bolt heater. This is inserted in an axial hole running down the center of the bolt. The bolt gets longer as it gets hot. When it is hot, the nut is run down against the joint to retain the increase in length produced by the heat. Since this is a crude way to preload bolts, the process must be controlled by other means. Dial gauges or micrometers are usually used to measure the net change in length of the bolts after they have cooled. If they have been stretched too much or too little, the bolts must be reheated and the nuts run down again. The process takes skill but is widely used on large fasteners (again 3 in or so and larger).

***Microprocessor-Controlled Torque-Turn Tools.*** Hydraulic tensioners and bolt heaters make it possible to tighten fasteners without suffering the uncertainties of the torque-preload relationship, but they can be used only on large-diameter fasteners. For smaller ones, we need something else. One relatively new approach—microprocessor-controlled tools—measures both applied torque and the turn of the nut to monitor and/or control fastener preload. Most of the presently available systems are designed for automatic or semiautomatic assembly in mass-production operations (automotive, primarily), but there are manual versions of some of them. They can control preload to ±2 to 5 percent if the joints are relatively soft (preload builds up smoothly as the bolts are tightened) and reasonable control is maintained over fastener dimensions and lubricity.

Some of these systems are designed to tighten every fastener to the yield point. This provides good preload accuracy (control is based on the act of yield rather than the torque-friction-preload relationship). But not all joints can be tightened safely to the yield point of the fastener. Although there is considerable debate on this point, many designers feel that yield-point tightening can lead to fatigue failure or rupture unless future external loads can be predicted and controlled.

***Turn-of-the-Nut Control.*** There is one place where tightening to or past the yield point is the norm; structural steel joints have been tightened this way for half a century using a carefully designed process called *turn-of-the-nut.* The fastener is first tightened to 60 to 80 percent of yield by the application of torque (usually with an air-powered impact wrench). The location of one corner of the nut is then noted, and a wrench is used to give the nut a specified half turn or so (depending on the size of the fastener and whether or not it is being used on a flat or tapered joint member). This amount of turn always takes the bolt beyond yield. Since external loads can be predicted, however, and are generally static rather than dynamic, the process is a safe and effective way to control preload.

***Ultrasonic Control of Preload.*** Ultrasonic instruments are sometimes used instead of torque and/or turn-of-the-nut to control preload (Ref. [22.6], p. 157). This technology has been used in a few aerospace applications for nearly a decade and is just starting to emerge in the commercial marketplace.

Presently available instruments send bursts of sound through the fastener and measure the time it takes for these wavefronts to travel through the fastener, echo off the far end, and return to the transducer. As the fastener is tightened, the time required for this round trip increases because the fastener gets longer, and so the path length is increased. Also, the velocity of the sound waves decreases as the stress level increases.

Microprocessors in the instruments sort out the change-in-length effect from the velocity effect and display either the change in length of the fastener or the average stress level in the tensile-stress area of the threads. Either of these quantities can be used to estimate fastener preload with better accuracy than is possible with torque or torque-turn controls.

One advantage of ultrasonics is that it can also be used in some cases to measure residual or working loads in the fasteners, as well as initial loads. You can use it to detect the effects of elastic interactions, for example, and therefore to compensate for such interactions. You can measure residual loads days or even years after initial tightening, which is never possible with torque and/or turn means.

Ultrasonics can be used with any sort of wrench, as well as with tensioners or heaters, to tighten fasteners.

## *22.4 BOLT TORQUE REQUIREMENTS*

### 22.4.1 The Problem

Although torque is the most common way to tighten a fastener, it is not a very good way, usually, to control the preload developed within the fastener. As we saw in Table 23.4, we must expect to see a scatter of ±30 percent or worse in the preload we achieve if we are using torque tools to tighten the fasteners. This scatter is acceptable in most applications, however. We compensate for it by overdesign, using larger bolts than might otherwise be necessary, for example.

Many factors affect this scatter in preload. These include such things as the finish on nuts, bolts, and joint members; the age, temperature, quantity, condition, and type of the lubricants used, if any; the speed with which the fasteners are tightened; the fit between male and female threads; the size of the holes and their perpendicularity with respect to joint surfaces; and the hardness of all parts. There is no way in which we can control or predict all the variables in a given situation, and so we must always expect and accept a considerable scatter in preload results when we use torque to control the tightening operation.

### 22.4.2 Selecting the Correct Torque

Having said all this, we must still select an appropriate torque to produce, or attempt to produce, the target preload we have established for our design. Our best bet is to use the so-called short-form torque equation to make an estimate. This equation is

$$T = KdF_{PT} \tag{22.21}$$

The nut factor $K$ is an experimental constant, a *bugger factor,* if you will, which defines the relationship which exists between applied torque and achieved preload in a given situation. The only way to determine what $K$ should be in your application is to make some actual experiments in which you measure both torque *and* preload and compute the mean $K$ and the scatter in $K$. If accuracy is not a big concern or you are merely trying to select the proper size of wrench or determine the approximate preloads you will achieve, then it is safe to use a nut factor listed in Table 22.5.

## *22.5 FATIGUE LOADING OF BOLTED AND RIVETED JOINTS*

When a bolt or joint member suddenly and unexpectedly breaks, it has probably failed because of fatigue. This is certainly one of the most common modes of failure for bolted joints. The designer, therefore, should learn how to cope with it.

**TABLE 22.5** Nut Factors

| Lubricant or coating on the fastener | Source | Nut factor: Reported mean | Nut factor: Reported range |
|---|---|---|---|
| 1. Cadmium plate | 1 | 0.194–0.246 | 0.153–0.328 |
| 2. Zinc plate | 5 | 0.332 | 0.262–0.398 |
| 3. Black oxide | 1 | 0.163–0.194 | 0.109–0.279 |
| 4. Baked on PTFE | 1 | 0.092–0.112 | 0.064–0.142 |
| 5. Molydisulfide paste | 2 | 0.155 | 0.14–0.17 |
| 6. Machine oil | 2 | 0.21 | 0.20–0.225 |
| 7. Carnaba wax (5% emulsion) | 2 | 0.148 | 0.12–0.165 |
| 8. 60 Spindle oil | 2 | 0.22 | 0.21–0.23 |
| 9. As received steel fasteners | 3 | 0.20 | 0.158–0.267 |
| 10. Molydisulfide grease | 3 | 0.137 | 0.10–0.16 |
| 11. Phosphate and oil | 3 | 0.19 | 0.15–0.23 |
| 12. Copper-based anti seize compound | 3 | 0.132 | 0.08–0.23 |
| 13. As received steel fasteners | 4 | 0.20 | 0.161–0.267 |
| 14. Plated fasteners | 4 | 0.15 | |
| 15. Grease, oil, or wax | 4 | 0.12 | |

SOURCES:
1. Values given are typical results from a very large and unpublished set of experiments on ASTM A193 B7 studs treated with various coatings. The tests were made in 1979–1980. Mean values for *K* varied with the diameter of the studs tested and the torques applied in various test series.
2. Kazuo Maruyama, Makoto Masuda, and Nobutoshi Ohashi, "Study of Tightening Control Methods for High Strength Bolts," *Bulletin of the Research Laboratory Precision Machine Selection,* Tokyo Institute of Technology, N46, September 1980, pp. 27–32.
3. John H. Bickford, *An Introduction to the Design and Behavior of Bolted Joints,* Marcel Dekker, Inc., New York, 1981, p. 429.
4. *Fastener Standards,* 5th ed., Industrial Fastener Institute, Cleveland, Ohio, 1970, p. N-16.
5. Edwin Rodkey, "Making Fastened Joints Reliable—Ways to Keep 'em Tight," *Assembly Engineering,* March 1977, p. 24.

### 22.5.1 Spotting a Fatigue Problem

It is usually easy to diagnose a fatigue failure. Here are the clues:

1. *Cyclic Loads* Fatigue failures always occur under cyclic tension loads.
2. *No Advance Warning* Fatigue failure is always sudden and almost always unexpected. The parts do not neck-down or wear out before they fail.
3. *Appearance of the Break Surface* If you examine the surface of a part which has failed in fatigue, you will usually find that a section of the surface is smooth, sometimes almost polished. Another portion of the surface, surrounding the first, may be a little rougher but is still basically smooth. The remainder of the surface will be very rough indeed.
4. *Typical Failure Points* The parts tend to fail at points of high stress concentration. Figures 22.15 and 22.16 show the most common failure points.

### 22.5.2 Estimating Fatigue Life

Many factors affect the fatigue life of any machine part, including fasteners. Such things as shape, heat treatment, surface finish, the mean load stress, the magnitude of load excursions, and the material all play a role. If you know the basic strength of the part, however, you can use the methods of Chap. 29 to estimate fatigue strength or endurance limit. Table 22.6 gives you the strength information you will need to do this for fasteners. You will find other information pertinent to the fatigue of joint members in Chaps. 32 and 29.

The term *proof strength* in Table 22.6 deserves explanation. It is common to test the strength of fasteners by applying tension loads to them. The *proof load* of a given fastener is the highest tensile force which can be applied to it without causing a permanent set to the fastener. The *proof strength* can then be determined by dividing the proof load by the tensile-stress area of the threads [Eq. (22.4)].

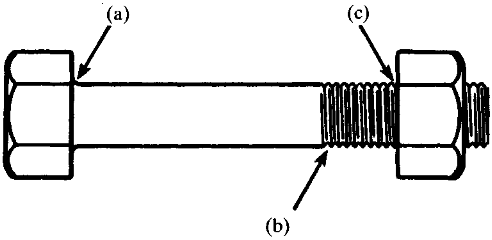



**FIGURE 22.15** Typical failure points of a bolt. (*a*) Failure at head fillet; (*b*) failure at thread runout; (*c*) failure at first thread to engage the nut.

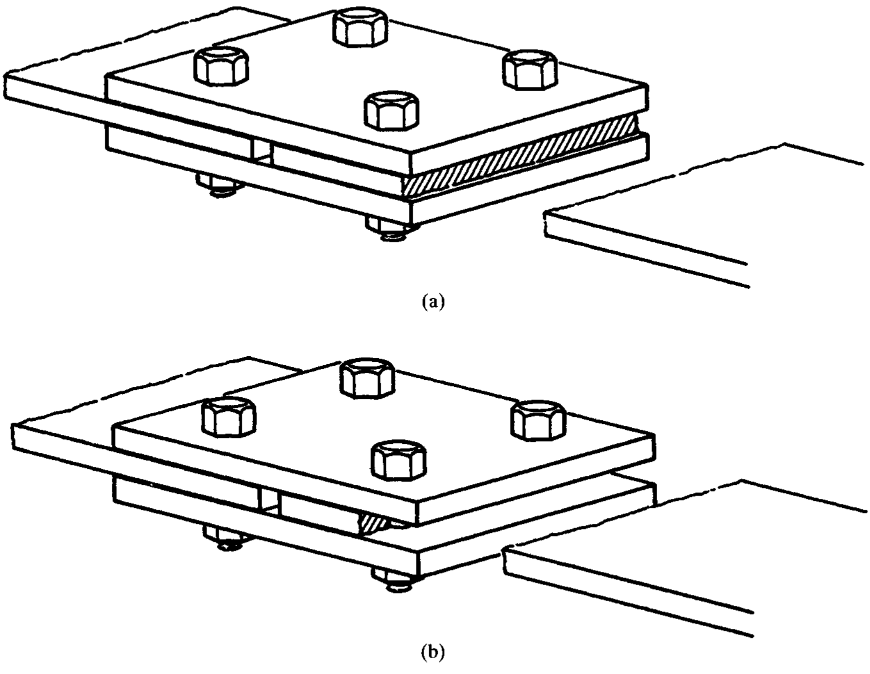



**FIGURE 22.16** Typical fatigue failures in joints loaded in shear. (*a*) Failure occurs in the gross cross section, near the place where the splice plates and joint plates meet, in a friction-type joint; (*b*) failure occurs in a net cross section, through a line of bolts, in a bearing-type joint.

**TABLE 22.6** Specifications and Identification Markings for Bolts, Screws, Studs, Sems,[a] and U Bolts[b]

| SAE grade | ASTM grade | Metric grade[c] | Nominal diameter in | Proof strength,[d] kpsi | Tensile strength,[d] kpsi | Yield strength,[d,e] kpsi | Core hardness Rockwell min/max | Grade identification marking | Products[f] | Material |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | A307 | 4.6 | 1/4 thru 1 1/2 | 33 | 60 | 36 | B70/B100 | None | B, Sc, St | Low- or medium-carbon steel |
| 2 | . . . | 5.8 | 1/4 thru 3/4 | 55 | 74 | 57 | B80/B100 | None | B, Sc, St | Low- or medium-carbon steel |
| | | 4.6 | Over 3/4 thru 1 1/2 | 33 | 60 | 36 | B70/B100 | None | B, Sc, St | Low- or medium-carbon steel |
| 4 | . . . | 8.9 | 1/4 thru 1 1/2 | 65[g] | 115 | 100 | C22/C32 | None | St | Medium-carbon, cold-drawn steel |
| 5 | A449 or A325 Type 1 | 8.8 | 1/4 thru 1 | 85 | 120 | 92 | C25/C34 |  | B, Sc, St | Medium-carbon steel, Q&T |
| | | 7.8 | Over 1 thru 1 1/2 | 74 | 105 | 81 | C19/C30 |  | B, Sc, St | Medium-carbon steel, Q&T |
| | | 8.6 | Over 1 1/2 to 3 | 55 | 90 | 58 | . . . |  | B, Sc, St | Medium-carbon steel, Q&T |
| 5.1 | . . . | 8.8 | No. 6 thru 5/8 | 85 | 120 | . . . | C25/C40 |  | Se | Low- or medium-carbon, Q&T |
| | | 8.8 | No. 6 thru 1/2 | 85 | 120 | . . . | C25/C40 |  | B, Sc, St | Low- or medium-carbon, Q&T |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 5.2 | A325 Type 2 | 8.8 | $\frac{1}{4}$ thru 1 | 85 | 120 | 92 | C26/C36 | | B, Sc | Low-carbon martensite steel, fully killed, fine-grained, Q&T |
| 7[h] | . . . | 10.9 | $\frac{1}{4}$ thru $1\frac{1}{2}$ | 105 | 133 | 115 | C28/C34 | | B, Sc | Medium-carbon alloy steel, Q&T |
| 8 | A354 Grade BD | 10.9 | $\frac{1}{4}$ thru $1\frac{1}{2}$ | 120 | 150 | 130 | C33/C39 | | B, Sc, St | Medium-carbon alloy steel, Q&T |
| 8.1 | . . . | 10.9 | $\frac{1}{4}$ thru $1\frac{1}{2}$ | 120 | 150 | 130 | C31/C38 | None | St | Elevated temperature drawn steel-medium carbon alloy or G15410 |
| 8.2 | . . . | 10.9 | $\frac{1}{4}$ thru 1 | 120 | 150 | 130 | C35/C42 | | B, Sc | Low-carbon martensite steel, fully killed, fine-grained, Q&T |
| . . . | A574 | 12.9 | 0 thru $\frac{1}{2}$ | 140 | 180 | 160 | C39/C45 | 12.9 | SHCS | Alloy steel, Q&T |
| | | 12.9 | $\frac{5}{8}$ thru $1\frac{1}{2}$ | 135 | 170 | 160 | C37/C45 | 12.9 | SHCS | Alloy steel, Q&T |

[a]Sems = screw and washer assemblies.
[b]Compiled from ANSI/SAE J429j; ANSI B18.3.1-1978; and ASTM A307, A325, A354, A449, and A574.
[c]Metric grade is *xx.x*, where *xx* is approximately $0.01S_{ut}$ in MPa and *.x* is the ratio of the minimum $S_y$ to $S_{ut}$.
[d]Multiply the strengths in kilopounds per square inch by 6.89 to get strength in megapascals.
[e]Yield strength is stress at which a permanent set of 0.2% of gauge length occurs.
[f]B = bolt, Sc = screws, St = studs, Sc = Sems, and SHCS = socket head cap screws.
[g]Entry appears to be in error but conforms to the standard ANSI/SAE J429j.
[h]Grade 7 bolts and screws are roll threaded after heat treatment.

SOURCE: From Joseph E. Shigley and Charles R. Mischke, *Mechanical Engineering Design*, 5th ed., McGraw-Hill, 1989; reproduced by permission of the authors and the publisher.

### 22.5.3 Reducing Fatigue Problems

There are a lot of things you can do to minimize fatigue problems.

***Material and Part Selection and Care.*** Materials with higher tensile strengths tend to have better fatigue lives than those with lower tensile strengths, at least up to an ultimate tensile strength of 200 kpsi (1379 MPa) or so. It also helps to select a material having low-notch sensitivity.

Avoid decarburization of the parts. Decarburization can weaken part surfaces and make it much easier for initial cracks to form.

Make sure that nut faces and the undersurface of the bolt head are perpendicular to the axis of the bolt threads and that the holes are perpendicular to the surfaces of the joints [22.10]. Two degrees of angularity can reduce fatigue life to only 25 percent of normal.

Lubricate the threads [22.10]. If nothing else, this can reduce corrosion problems, and corrosion is a main source of initial cracks.

If using fasteners with a tensile strength above 150 kpsi (1034 MPa), do *not* use lubricants containing sulfides, since these can contribute to stress-corrosion cracking, which will accelerate fatigue failure [22.11].

Grit blast the surfaces of joints loaded in shear before assembling because anything which increases the slip resistance improves fatigue life ([22.2], p. 120).

***Prevent Crack Initiation.*** Polish, but do not hard coat, bolt surfaces, or shot peen the surfaces, or roll bolt threads after heat treatment. Do anything and everything possible to avoid corrosion of bolts or joint members (see Chap. 35).

***Reduce Load Excursions.*** Even if the magnitude of external loads imposed on a joint are beyond the designer's control, there are many things which he or she can do to reduce the variations in load seen by a given joint. And these variations, or *load excursions,* are a key issue. We always want to keep the ratio between minimum load and maximum load seen by the parts as close to unity as possible.

Some say the minimum bolt tension should always be more than half the maximum bolt tension. Others recommend a preload that is at least two to three times the magnitude of the worst-case external load to be applied to the joint. Because of the large number of variables involved, such rules will apply only to certain applications. Nevertheless, they give you an idea of the importance of minimizing load excursions.

It helps to increase the ratio between the stiffness of the joint and the stiffness of the bolt ($k_J/k_B$) so that the joint will absorb a larger percentage of the applied load excursions. There are many ways to do this. For examples, see Fig. 22.17*a* and *b*. Reducing the body of the bolt to nine-tenths of the nominal diameter is sometimes recommended.

It helps to compensate for initial preload loss and relaxation effects by retightening the bolts after they have relaxed. By the same token, try to avoid vibration loss of preload by providing damping and/or by periodic retightening of the nuts and/or by using special vibration-resistant fasteners.

***Reduce Stresses in Parts.*** Make sure there are at least three threads above and below the faces of the nut (Fig. 22.17*c*). Do not let the thread run-out point coincide with the shear plane of the joint (Fig. 22.17*d*). Roll the threads instead of cutting

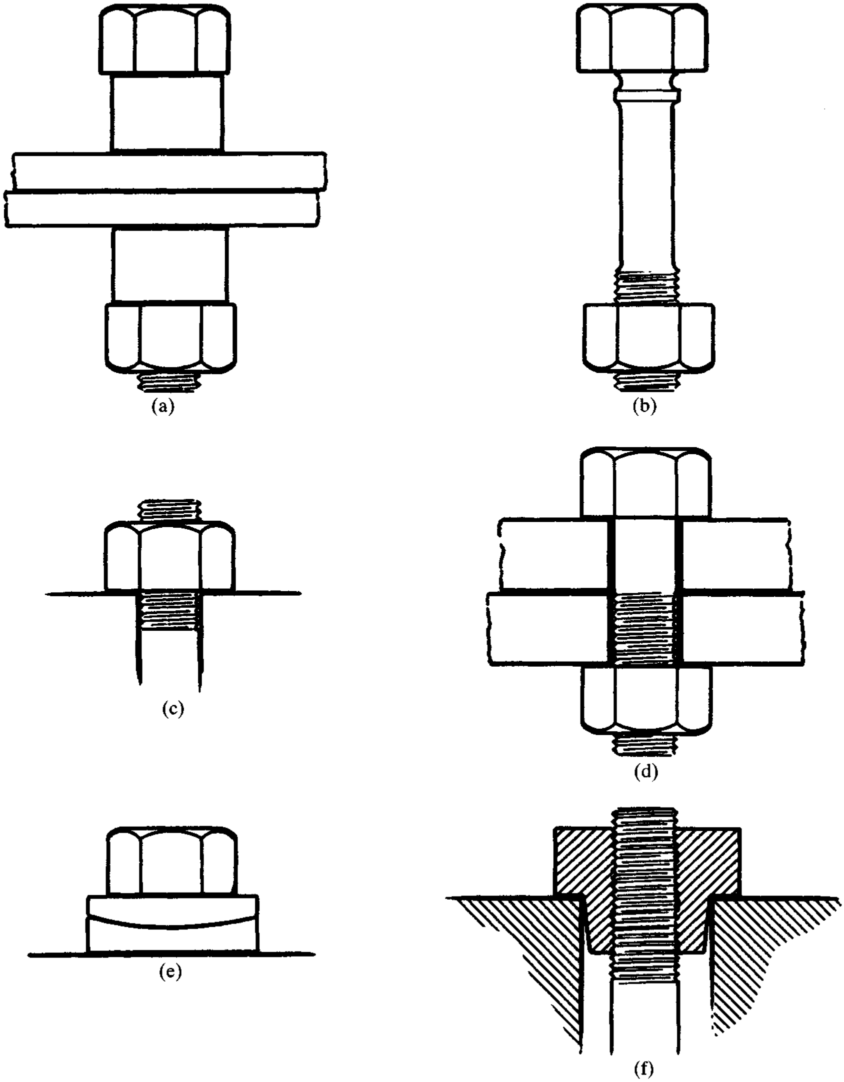



**FIGURE 22.17** Ways to improve the fatigue life of bolts. (*a*) Use collars to increase the length-to-diameter ratio of the bolts; (*b*) turn down the body of a bolt to reduce its stiffness; (*c*) make sure that there are at least three threads above and below a nut to reduce thread stress concentrations; (*d*) it also helps to avoid the situation, shown here, where thread run-out coincides with a shear plane in the joint, or (*e*) to use spherical washers to help a bolt adjust to bending loads, or (*f*) to use tension nuts to reduce thread stress levels. All figures shown are improvements except (*d*).

them, and if possible, roll them after heat treatment instead of before [22.12]. Use a large root radius in the threads.

Use a large head-to-body fillet, and use elliptical fillets instead of round fillets [22.13]. Use spherical washers to minimize bending effects (Fig. 22.17*e*). Use Class 2 threads instead of Class 3. Use tension nuts for a smoother stress transition in the bolts (Fig. 22.17*f*). Use nuts that are longer than normal. Make sure the thread-to-body run-out is smooth and gradual.

## 22.6 PROGRAMMING SUGGESTIONS FOR JOINTS LOADED IN TENSION

Figure 22.18 shows the flowchart of a computer program which might be used to design bolted joints loaded in tension. We start by entering dimensions, strengths, external loads, and the like. Next, we compute the cross-sectional areas of the bolt and the stiffness of bolt and joint members. The program assumes that the joint is not gasketed.

Next, we compute the maximum acceptable tension in the bolt, basing this either on a code or specification limit or on the yield strength of the bolt. If the bolts are to see a combination of tension and shear loads, the acceptable upper limit of tension must, of course, be reduced.

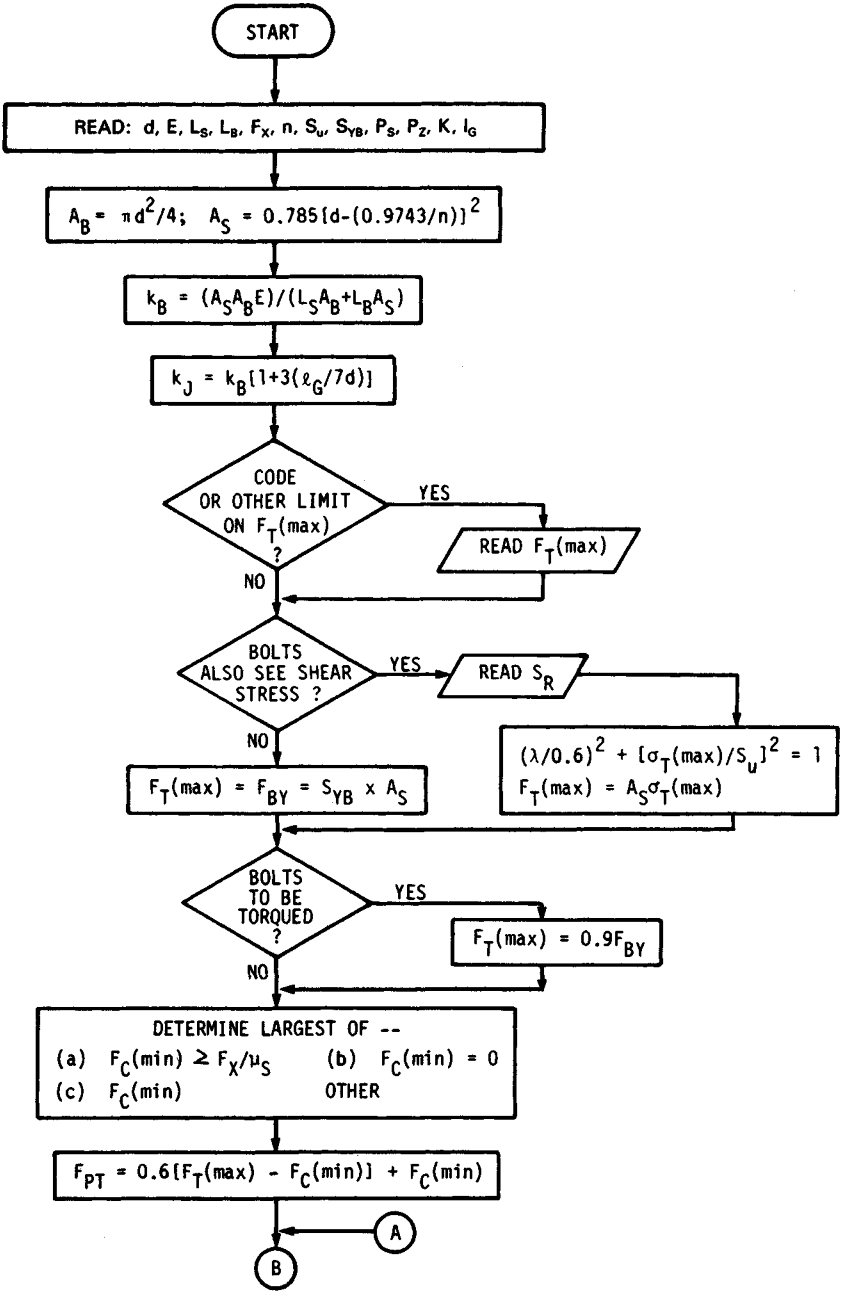
The next step is to determine the acceptable lower limit on the clamping force in the joint. This will usually be a more complicated procedure than suggested by the flowchart. If gaskets are involved, for example, it would be necessary to calculate minimum clamping force using the procedures and equations of the ASME Boiler and Pressure Vessel Code, or the like. If all we are concerned about is transverse slip or total separation, we could use the equations shown in the flowchart.

We complete the definition of our design specifications by computing a target preload and then printing out the upper and lower acceptable limits, the force required to yield the bolt, and the target preload. It is useful to know these things if we need to revise the target preload at a later point in the program.

Having determined the acceptable upper and lower limits, we now take a series of steps to estimate the actual limits we will achieve in practice based on our target preload and estimates of such things as tool scatter and joint relaxation. During this part of the procedure we also introduce the estimated effects of the external tension loads on the joint, assuming linear joint behavior. The equations used here are derived from the joint diagram in Fig. 22.13.

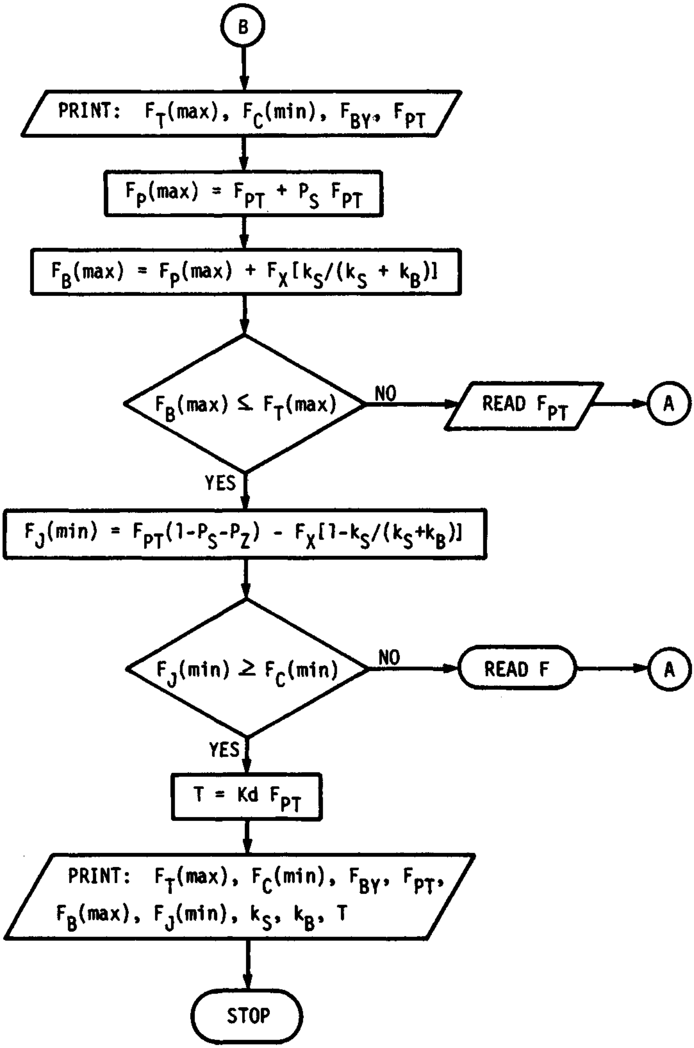
We compute the anticipated upper limit on bolt tension first, and then we compute the anticipated lower limit on clamping force. We compare them, one at a time, to the acceptable limits. We recycle, choosing a new target preload, if the anticipated limits fall outside of the acceptable limits. In some cases we will not be able to satisfy our specifications merely by modifying the target preload; we may have to choose new joint dimensions to enlarge the range between upper and lower limits or choose more accurate tools to reduce the range between the upper and lower limits anticipated in practice.

When our conditions are satisfied, we complete the program by computing the torque required to aim for the target preload. Then we print out the final values of the parameters computed.




(a)

**FIGURE 22.18** Flowchart describing a computer program which could be used to design non-gasketed joints loaded in tension. Chart continues on next page.



(b)

**FIGURE 22.18** *(Continued)*

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