CHAPTER 35
MECHANICAL FASTENERS

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35.1 INTRODUCTION

Most of the information in this chapter is not original. I am merely passing along the information I have gained from many other people and from extensive reading in this subject.

For an in-depth understanding of this inexact field of study, I would recommend two excellent books that I used extensively in the preparation of this chapter.[1,2] For a full comprehension of this topic, it is necessary to read both volumes, as they approach the topic from distinctly different points of view.

Two or more components may need to be joined in such a way that they may be taken apart during the service life of the part. In these cases, the assembly must be fastened mechanically. Other reasons for choosing mechanical fastening over welding could be:

1. Ease of part replacement, repair, or maintenance
2. Ease or lower cost to manufacture
3. Designs requiring movable joints
4. Designs requiring adjustable joints

The most common mechanical joining methods are bolts (threaded fasteners), rivets, and welding (welding will be covered in a later section).

To join two members by bolting or riveting requires holes to be drilled in the parts to accommodate the rivets and bolts. These holes reduce the load-carrying cross-sectional area of the members to be joined. Because this reduction in area as a result of the holes is at least 10–15%, the load-carrying capacity of the bolted structure, is reduced, which must be accounted for in the design. Alternatively, when one inserts bolts into the holes, only the cross section of the bolt or rivet supports the load. In this case, the reduction in the strength of the joint is reduced even further than 15%.

Even more critical are the method and care taken in drilling the holes. When one drills a hole in metal, not only is the cross-sectional area reduced, but the hole itself introduces "stress risers" and/or flaws in/on the surface of the holes that may substantially endanger the structure. First, the hole places the newly created surface in tension, and if any defects are created as a result of drilling, they must be accounted for in a quantitative way. Unfortunately, it is very difficult to obtain definitive information on the inside of a hole that would allow characterization of the introduced defect.

The only current solution is to make certain that the hole is properly prepared which means not only drilling or subpunching to the proper size, but also *reaming* the surface of the hole. To be absolutely certain that the hole is not a problem, one needs to put the surface of the hole in residual compression by expanding it slightly with an expansion tool or by pressing the bolt, which is just slightly larger than the hole. This method causes the hole to expand during insertion, creating a hole whose surface is in residual compression. While there are fasteners designed to do this, it is not clear that all of the small surface cracks of the hole have been removed to prevent flaws/stress risers from existing in the finished product.

Using bolts and rivets in an assembly can also provide an ideal location for water to exist in the crevices between the two parts joined. This trapped water, under conditions where chlorides and sodium exist, can cause "crevice corrosion," which is a serious problem if encountered.

Obviously, in making the holes as perfect as possible, you increase the cost of a bolted and/or riveted joint significantly, which makes welding or adhesive joining a more attractive option. Of course, as will be shown below, welding and joining have their own set of problems that can degrade the joint strength.

The analysis of the strength of a bolted, riveted, or welded joint involves many indeterminate factors resulting in inexact solutions. However, by making certain simplifying assumptions, we can obtain solutions that are acceptable and practical. We discuss two types of solutions: *bearing-type connections,* which use ordinary or unfinished bolts or rivets, and *friction-type connections,* which

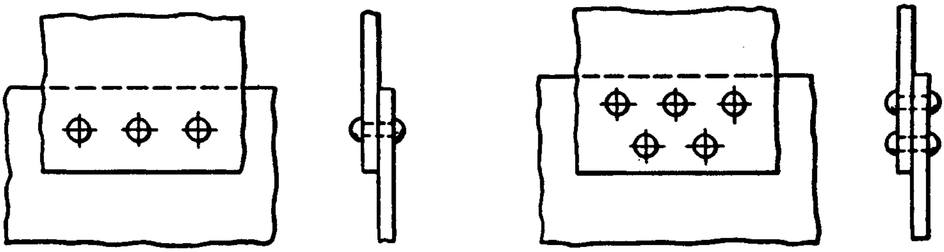

Fig. 35.1 Lap joints. Connectors are shown as rivets only for convenience.

use high-strength bolts. Today, economy and efficiency are obtained by using high-strength bolts for field connections together with welding in the shop. With the advent of lighter-weight welding power supplies, the use of field welding combined with shop welding is finding increasing favor.

While riveted joints do show residual clamping forces (even in cold-driven rivets), the clamping forces in the rivet is difficult to control, is not as great as that developed by high-strength bolts, and cannot be relied upon. Installation of hot-driven rivets involves many variables, such as the initial or driving temperature, driving time, finishing temperature, and driving method. Studies have shown that the holes are almost completely filled for short rivets. As the grip length is increased, the clearances between rivet and plate material tend to increase.

35.2 BOLTED AND RIVETED JOINT TYPES

There are two types of riveted and bolted joints: *lap joints* and *butt joints.* See Figs. 35.1 and 35.2 for lap and butt joints, respectively. Note that there can be one or more rows of connectors, as shown in Fig. 35.2*a* and *b*.

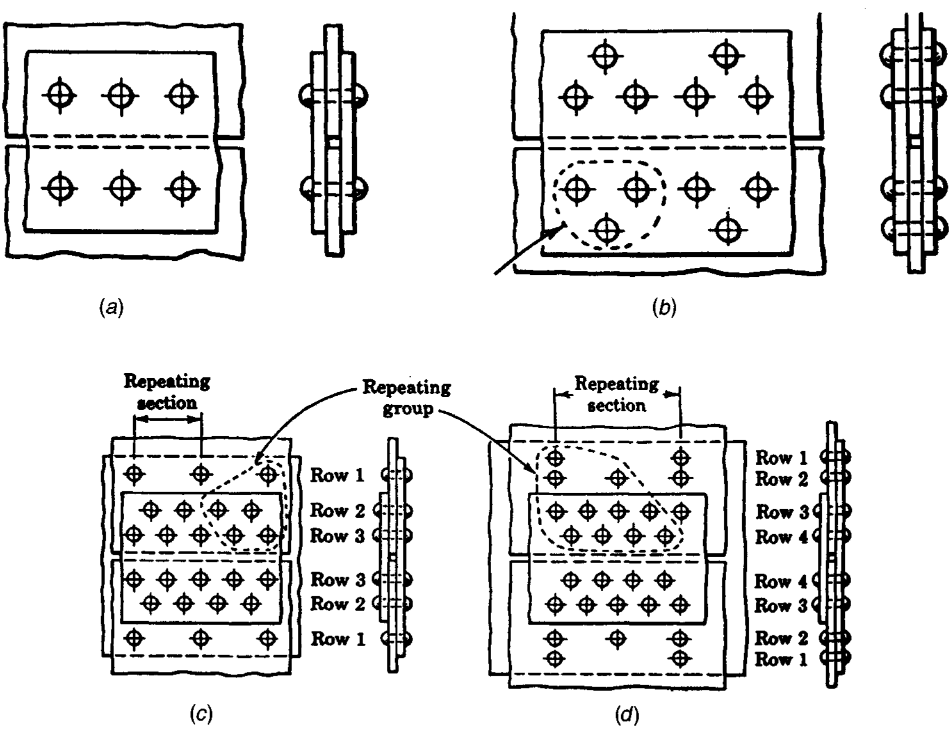

Fig. 35.2 Butt joints: (*a*) single-row; (*b*) double-row; (*c*) triple-row (pressure-type); (*d*) quadruple row (pressure-type).

In a butt joint, plates are butted together and joined by two cover plates connected to each of the main plates. (Rarely, only one cover plate is used to reduce the cost of the joint.) The number of rows of connectors that fasten the cover plate to each main plate identifies the joint—single row, double row, and so on. See Fig. 35.2.

Frequently the outer cover plate is narrower than the inner cover plate, as in Fig. 35.2*c* and *d*, the outer plate being wide enough to include only the row in which the connectors are most closely spaced. This is called a *pressure joint* because caulking along the edge of the outer cover plate to prevent leakage is more effective for this type of joint.

The spacing between the connectors in a given row is called the *pitch*. When the spacing varies in different rows, as in Fig. 35.2*d*, the smallest spacing is called the *short pitch*, the next smallest the *intermediate pitch*, and the greatest the *long pitch*. The spacing between consecutive rows of connectors is called the *back pitch*. When the connectors (rivets or bolts) in consecutive rows are staggered, the distance between their centers is the *diagonal pitch*.

In determining the strength of a joint, computations are usually made for the length of a joint corresponding to a repeating pattern of connectors. The length of the repeating pattern, called the *repeating section*, is equal to the long pitch.

To clarify how many connectors belong in a repeating section, see Fig. 35.2*c*, which shows that there are five connectors effective in each half of the triple row—that is, two half connectors in row 1, two whole connectors in row 2, and one whole and two half connectors in row 3. Similarly, there are 11 connectors effective in each half of the repeating section in Fig. 35.2*d*.

When rivets are used in joints, the holes are usually drilled or, punched, and reamed out to a diameter of 1/16 in. (1.5 mm) larger than the nominal rivet size. The rivet is assumed to be driven so tightly that it fills the hole completely. Therefore, in calculations the diameter of the hole is used because the rivet fills the hole. This is not true for a bolt unless it is very highly torqued. In this case, a different approach needs to be taken, as delineated later in this chapter.

35.3 EFFICIENCY

Efficiency compares the strength of a joint to that of the solid plate as follows:

$$\text{Efficiency} = \frac{\text{strength of the joint}}{\text{strength of solid plate}}$$

35.4 STRENGTH OF A SIMPLE LAP JOINT (BEARING-TYPE CONNECTION)

For bearing-type connections using rivets or ordinary bolts, we use the equation

$$P_s = A\sigma$$

For shear, this is rewritten as

$$P_s = A_s\tau = \frac{\pi d^2 \tau}{4}$$

where

P_s = the load

A = shear area of one connector

d = diameter of connector and/or hole

For the above example, friction is neglected. Figure 35.3 shows the shearing of a single connector.

Another possible type of failure is caused by tearing the main plate. Figure 35.4 demonstrates this phenomenon.

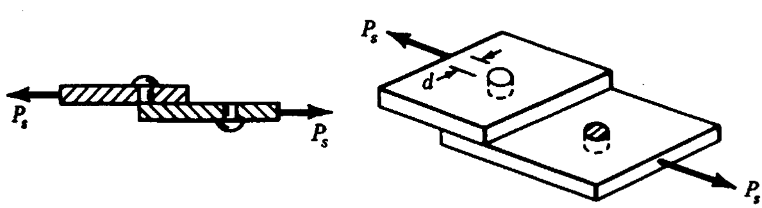

Fig. 35.3 Shear failure.

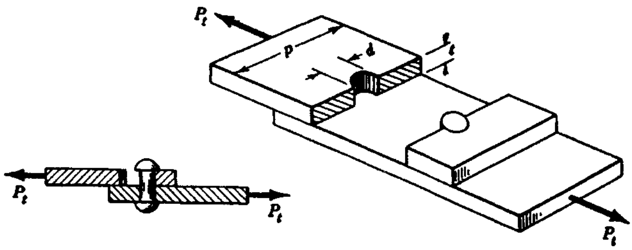

Fig. 35.4 Tear of plate at section through connector hole. $P_t = A_t\sigma_t = (p - d)t\sigma_t$.

The above failure occurs on a section through the connector hole because this region has the minimum tearing resistance. If p is the width of the plate or the length of a repeating section, the resisting area is the product of the net width of the plate $(p - d)$ times the thickness t. The failure load in tension therefore is

$$P_{\text{tension}} = A_t\sigma_t = (p - d)t(\sigma_t)$$

A third type of failure, called a *bearing failure,* is shown in Fig. 35.5. For this case, there is relative motion between the main plates or enlargement of the connector hole caused by an excessive tensile load. Actually, the stress that the connector bears against the edges of the hole varies from zero at the edges of the hole to the maximum value at the center of the bolt or rivet. However, common practice assumes the stress as uniformly distributed over the projected area of the hole. See Fig. 35.5.

The failure load in the bearing area can be expressed by

$$P_b = A_b\sigma_b = (td)\sigma_b$$

Other types of failure are possible but will not occur in a properly designed joint. These are tearing of the edge of the plate back of the connector hole (Fig. 35.6*a*) or a shear failure behind the connector hole (Fig. 35.6*b*) or a combination of both. Failures of this type occur when the distance from the edge of the plate is ~2 or less multiplied by the diameter of the connector or hole.

35.5 SAMPLE PROBLEM OF A COMPLEX BUTT JOINT (BEARING-TYPE CONNECTION)

The strength of a bearing-type connection is limited by the capacity of the rivets or ordinary bolts to transmit load between the plates or by the tearing resistance of the plates themselves, depending on which is smaller. The calculations are divided as follows:

1. Preliminary calculations to determine the load that can be transmitted by one rivet or bolt in shear or bearing *neglecting friction* between the plates
2. Calculations to determine which mode of failure is most likely

A repeating section 180 mm long of a riveted triple row butt joint of the pressure type is illustrated in Fig. 35.7. The rivet hole diameter d = 20.5 mm, the thickness of the main plate t = 14 mm, and the thickness of each cover plate t = 10 mm. The ultimate stresses in shear, bearing, and tension are respectively τ = 300 MPa, σ_b = 650 MPa, and σ_t = 400 MPa. Using a factor of safety of 5, determine

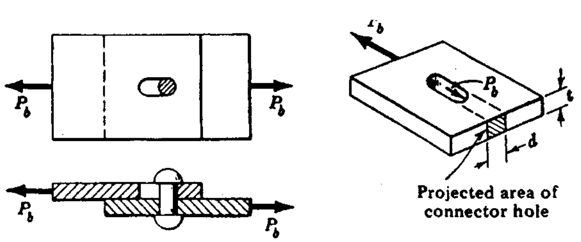

Fig. 35.5 Exaggerated bearing deformation of upper plate. $P_b = A_b\sigma_b = (td)\sigma_b$.

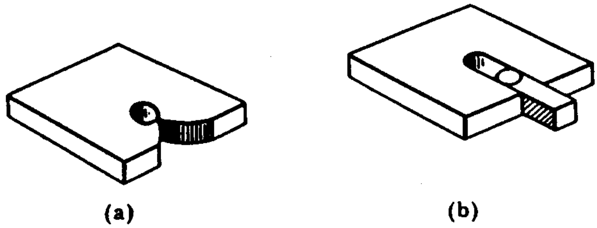

Fig. 35.6 Possible types of failure if connector hole is too close to edge of plate: (*a*) tear out; (*b*) shear behind connector.

the strength of a repeating section, the efficiency of the joint, and the maximum internal pressure that can be carried in a 1.5 m diameter boiler where this joint is the longitudinal seam.

Solution: The use of ultimate stresses will determine the ultimate load, which is then divided by the factor of safety (in this case 5) to determine the safe working load. An alternative but preferable procedure is to use allowable stresses to determine the safe working load directly, which involves smaller numbers. Thus, dividing the ultimate stressed by 5, we find that the allowable stresses in shear, bearing, and tension, respectively, are $\tau = 300/5 = 60$ MPa, $\sigma_b = 650/5 = 130$ MPa, and $\sigma_t = 400/5 = 80$ MPa. The ratio of the shear strength τ to the tensile strength σ of a rivet is about .75.

35.5.1 Preliminary Calculations

To single shear one rivet,

$$P_s = \frac{\pi d^2}{4}\tau = \frac{\pi}{4}(20.5 \times 10^{-3})^2(60 \times 10^6) = 19.8\ kN$$

To double shear one rivet,

$$P_s = 2 \times 19.8 = 39.6\ kN$$

To crush one rivet in the main plate,

$$P_B = (td)\sigma_b = (14.0 \times 10^{-3})(20.5 \times 10^{-3})(130 \times 10^6) = 37.3\ kN$$

To crush one rivet in one cover plate,

$$P_b' = (t'd)\sigma_b = (10 \times 10^{-3})(20.5 \times 10^{-3})(130 \times 10^6) = 26.7\ kN$$

Rivet capacity solution: The strength of a single rivet in row 1 in a repeating section is determined

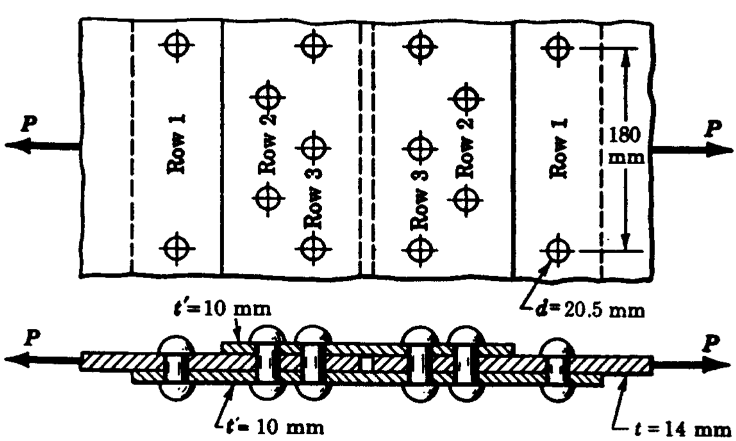

Fig. 35.7

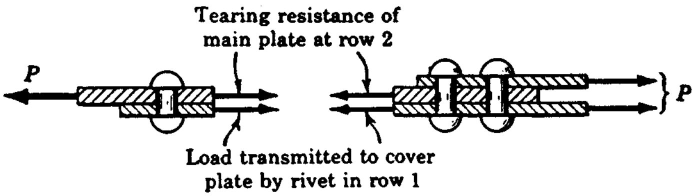

Fig. 35.8

by the lowest value of the load that will single shear the rivet, crush it in the main plate, or crush it in one of the cover plates. Based on the values in the preceding calculations, this value is 19.8 *kN* per rivet.

The strength of each of the two rivets in row 2 depends on the lowest value required to double shear the rivet, crush it the main plate, or crush it in both cover plates. From the above preliminary calculations, this value is 37.3 *kN* per rivet or $2 \times 37.3 + 74.6$ *kN* for both rivets in row 2.

Each of the two rivets in the repeating section in row 3 transmits the load between the main plate and the cover plate in the same manner as those in row 2; hence for row 3, the strength = 74.6 *kN*.

The total rivet capacity is the sum of the rivet strengths in all rows (rows 1, 2, 3), as follows:

$$P_{\text{total}} = 19.8 + 74.6 + 74.6 = 169.0\ kN$$

Tearing capacity: The external load applied to the joint acts directly to tear the main plate at row 1, and the failure would be similar to Fig. 35.4. This is calculated as follows:

$$P_{\text{tearing}} = (p - d)\,\sigma_t = [(180 \times 10^{-3}) - (20.5 \times 10^{-3})](14 \times 10^{-3})(80 \times 10^{6}) = 178.6\ kN$$

The external load applied does not act directly to tear the main plate at row 2 because part of the load is absorbed or transmitted by the rivet in row 1. Hence, if the main plate is to tear at row 2, the external load must be the sum of the tearing resistance of the main plate at row 2 plus the load transmitted by the rivet in row 1. See Figs. 35.8 and 35.9.

Thus,

$$\begin{aligned} P_{\text{tearing2}} &= (p - 2d)t\sigma_t + \text{rivet strength in row 1} \\ &= [(180 \times 10^{-3}) - 2(20.5 \times 10^{-3})](14 \times 10^{-3})(80 \times 10^{6}) \\ &\quad + 19.8 \times 10^{3} = 175.5\ kN \end{aligned}$$

Similarly, the external load required to tear the main plate at row 3 must include the rivet resistance in rows 1 and 2 or

$$\begin{aligned} P_3 &= [(180 \times 10^{-3}) - 2(20.5 \times 10^{-3})](14 \times 10^{-3})(80 \times 10^{6}) + (19.8 \times 10^{3}) + (74.6 \times 10^{3}) \\ &= 250.1\ kN \end{aligned}$$

It is obvious that this computation need not be made because the tearing resistance of the main plates at rows 2 and 3 is equal, thus giving a larger value.

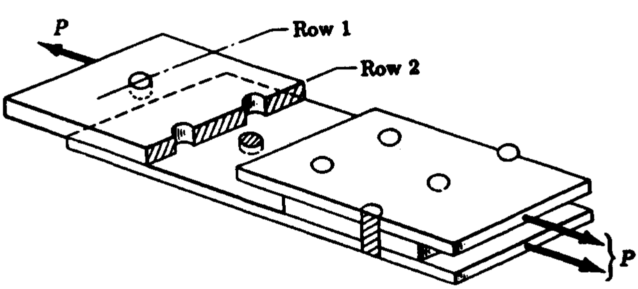

Fig. 35.9 Failure by shear of rivet in row 1 plus tear of main plate in row 2.

At row 3, the tearing resistance of the cover plates is resisted by the tensile strength of the reduced section of that row. The tensile strength of one cover plate is

$$P_c = [(180 \times 10^{-3}) - 2(20.5 \times 10^{-3})](10 \times 10^{-3})(80 \times 10^6) = 111.2\ kN$$

In an ordinary butt joint, the tensile capacity of both cover plates is twice this value. In a pressure joint, however, where one cover plate is shorter than the other, the load capacity of the shorter plate must be compared with the rivet load transmitted to it. In this example, the upper cover plate transmits the rivet load of four rivets in single shear, or $4 \times 19.8 = 79.2\ kN$, which is less than its tear capacity of $111.2\ kN$. Hence, the load capacity of both cover plates becomes

$$P_c = 79.2 + 111.2 = 190.4\ kN$$

determined by rivet shear in the upper plate and by tension at row 3 in the lower plate.

Thus, the safe load is the lowest of these several values $= 169.0\ kN$, which is the rivet strength in shear.

$$\text{Efficiency} = \frac{\text{safe load}}{\text{strength of solid plate}} = \frac{169 \times 10^3}{(180 \times 10^3)(14 \times 10^{-3})(80 \times 10^6)} = 83.8\%$$

In this discussion, we have neglected friction and assumed that the rivets or bolts only act as pins in the structure or joint—in essence like spot welds spaced in the same way as the rivets or bolts are spaced.

35.6 FRICTION-TYPE CONNECTIONS

In friction-type connections, high-strength bolts (generally high-strength medium carbon steel bolts plain, weathering, or galvanized finished, designated as A325 ASTM grade, or alloy steel bolts designated as A490 ASTM grade) are used and are tightened to high tensile stresses, thereby causing a large resultant normal force between the plates. Tightening of the bolts to a predetermined initial tension is usually done using a calibrated torque wrench or by turn-of-the nut methods.

If done properly (as will be discussed later), the load is now transferred by the friction between the plates and not by shear and the bearing of the bolt, as described in the previous sections. Heretofore, even though the bolts are not subject to shear, design codes, as a matter of convenience, specified an allowable shearing stress to be applied over the cross-sectional area of the bolt. Thus, friction-type joints were analyzed by the same procedures used for bearing-type joints and the frictional forces that existed, were taken as an extra factor of safety. In the ASME code, the "allowable stresses" listed in several places are not intended to limit assembly stresses in the bolts. These allowables are intended to force flange designers to overdesign the joint to use more and/or larger bolts and thicker flange members than they might otherwise be inclined to use.

Only in the non-mandatory Appendix S does section VIII of the code deal with assembly stresses, and then in relatively general terms.

The closest Appendix S comes to quantifying assembly stresses in the bolts is in Eq. (18.6) which suggests that the amount of stress you might expect to produce in the bolts at assembly is given by

$$S_A = \frac{45{,}000}{D^{1/2}}$$

where S_A = stress created in the bolts at assembly (psi)
D = the nominal diameter of the fastener (in.)

Structurally, a bolt serves one of two purposes: it can act as a pin to keep two or more members from slipping relative to each other, or it can act as a heavy spring to clamp two or more pieces together.

In the vast majority of applications, the bolt is used as a clamp and, as such, it must be tightened properly. When we tighten a bolt by turning the head or the nut, we will stretch the bolt initially in the elastic region. More tightening past the elastic limit will cause the bolt to deform plastically. In either case, the bolt elongates and the plates deform in the opposite direction (equal compressive stresses in the materials being joined). In this way, you really have a spring as shown (with substantial exaggeration) in Fig. 35.10.

The tensile stress introduced into the fastener during this initial tightening process results in a tension force within the fastener, which in turn creates the clamping force on the joint. This initial clamping force is called the *preload.* Preloading a fastener properly is a major challenge that will be discussed later.

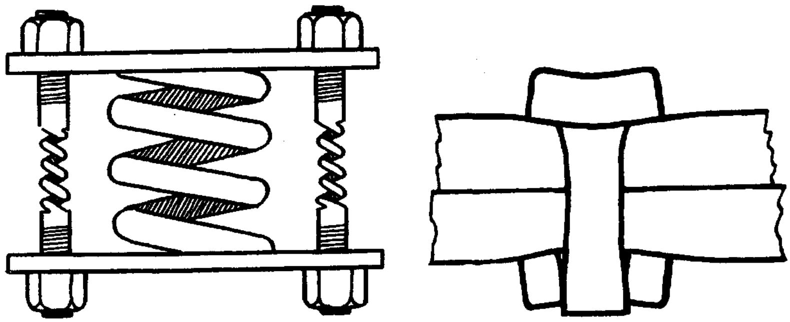

Fig. 35.10 When analyzing the behavior of a bolted joint, pretend the members are a large spring being compressed (clamped) by a group of smaller springs (bolts). When tightened, these springs distort somewhat as shown but grossly exaggerated on the right.

When a bolt is loaded in tension in a tensile testing machine, we generate a tension versus a change in length curve, as shown in Fig. 35.11. The initial straight line portion of the elastic curve is called the *elastic region*. Loading and unloading a bolt within this range of tension never results in a permanent deformation of the bolt because elastic deformation is recoverable. The upper limit of this curve ends at the *proportional limit* or *elastic limit*. Loading beyond or above this limit results in *plastic deformation* of the bolt, which is not recoverable; thus, the bolt has a permanent "set" (it is longer than it was originally even though the load is completely removed). At the *yield point*, the bolt has a specific amount of permanent plastic deformation, normally defined as 0.2 or 0.5% of the initial length. This permanent plastic deformation will increase up until the *ultimate tensile strength* (normally called the *ultimate strength* of the bolt), which is the maximum tension that can be created in the bolt. The UTS is always greater than the yield stress—sometimes as much as twice yield. The final point on the curve is the *failure* or *rupture stress*, where the bolt breaks under the applied load.

If we load the bolt well into the plastic region of its curve and then remove the load, it will behave as shown in Fig. 35.12, returning to the zero load point along a line parallel to the original elastic line but offset by the amount of plastic strain the bolt has set.

On reloading the bolt below the previous load but above the original yield point, the behavior of the bolt will follow this new offset stress strain line and the bolt will behave elastically well beyond the original load that caused plastic deformation in the first place. The difference between the original yield strength of the material and the new yield strength is a function of the "work hardening" that occurred by taking it past the original yield strength on the first cycle. By following the above procedure, we have made the bolt stronger, at least as far as static loads are concerned.

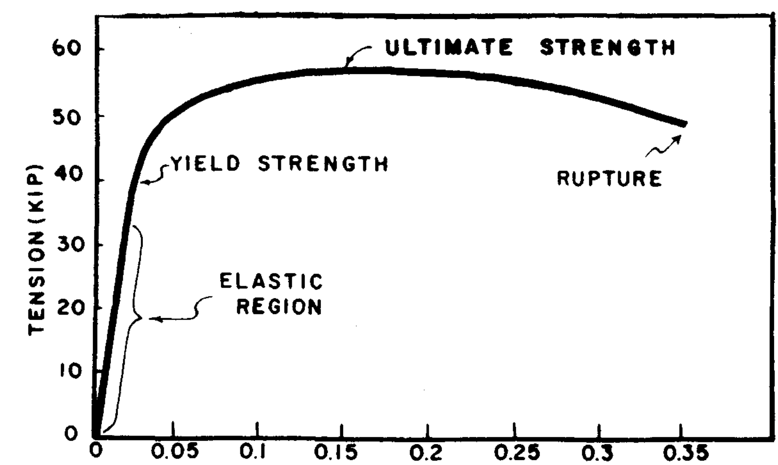

Fig. 35.11 Engineering stress-strain curve (typical).

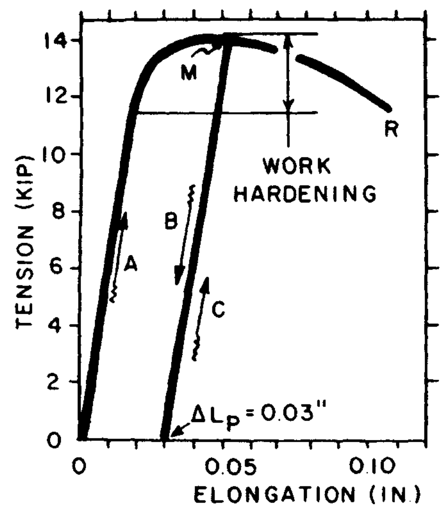

Fig. 35.12 Elastic curve for a ⅜–16 × 4 socket-head cap screw loaded (A) to point M well past the yield strength and then unloaded (B) to give permanent deformation L_p = 0.03 in. If reloaded, it will follow path (C).

This is not wise practice, however, for more brittle materials can suffer a loss of strength by such treatments. Loss of strength in ASTM A490 bolts, because of repeated cycling past the yield (under water and wind loads), has been publicly cited as a contributing factor in the 1979 collapse of the roof on the Kemper Auditorium in Kansas City.

The answer to the question "how much preload" we should place on the joint is currently impossible to answer other than in generalities, ranging from "We always want the maximum clamping force the parts can stand" to "The more the better, up to, but *probably not exceeding the yield stress.*"

35.7 UPPER LIMITS ON CLAMPING FORCE

When determining the amount of clamping force required to combat self-loosening or slip or a leak, we are establishing the essential minimum of force. In each of these situations, additional clamping force is usually desirable from an added safety point of view or is at least acceptable. Commonly used criteria for setting the upper limit follow.

35.7.1 Yield Strength of the Bolt

There is currently a lot of debate about this in the bolting world, as most feel it is unwise to tighten bolts beyond the yield in most applications, although torquing beyond yield is growing in popularity for automotive and similar applications. In general, however, *we usually don't want to tighten bolts beyond their yield point.*

35.7.2 Thread Stripping Strength

We would never want to tighten fasteners past the point at which their threads will strip.

35.7.3 Design-Allowable Bolt Stress and Assembly Stress Limits

We need to follow the limits placed on bolt stresses by codes, company policies, and standard practices. Both structural steel and pressure vessel codes define maximum design allowable stresses for bolts. To distinguish between maximum design stress and the maximum stress that may be allowed in the fastener during assembly, we need to look at the design safety factor. These two will differ—that is, maximum design allowables will differ if a factor of safety is involved. For structural steels, bolts are frequently tightened well past the yield strength even though the design allowables are only 35–58% of yield. Pressure vessel bolts are commonly tightened to twice the design allowable. Aerospace, auto, and other industries may impose stringent limits on design stresses rather than on actual stresses to force the designer to use more or larger bolts.

35.7.4 Torsional Stress Factor

If the bolts are to be tightened by turning the nut or the head, they will experience a torsion stress as well as a tensile stress during assembly. If tightened to the yield stress, they will yield under this combination. If we plan to tighten to or near the yield stress, we must reduce the maximum tensile stresses allowed by a "torquing factor." If using as received steel on steel bolts, then a reduction in the allowable tensile stress of 10% is reasonable. If the fasteners are to be lubricated, use 5%.

35.7.5 Shear Stress Allowance

Since we are designing a clamp, we are primarily interested in the clamping force that the bolts are going to exert on this joint, and therefore the tensile stress. If the bolts will also be exposed to a shear stress, we must take this into account in defining the maximum clamping force, since the shear stress will reduce the amount of strength capacity available for the tensile stress. The following equation can be used to determine how much shear stress the bolt can stand if subject to a given tensile stress or vice versa:

$$\frac{S_T^2}{G^2} + T_T^2 = 1.0$$

where

S_T = the ratio between the shear stress in the shear plane of the bolt and the UTS of the bolt
T_T = the ratio between the tensile stress in the bolt and the UTS of the bolt
G = the ratio between the shear strength and the tensile strength of the bolt (0.5—0.62 typically) if computed on the thread stress area. It is best to compute both S_T and T_T using the equivalent thread stress area rather than the shank area

35.7.6 Flange Rotation

Excessive bolt load can rotate raised face flanges so much that the ID of the gasket is unloaded, opening a leak path. The threat of rotation, therefore, can place an upper limit on planned or specified clamping forces.

35.7.7 Gasket Crush

Excessive preload can so compress a gasket that it will not be able to recover when the internal pressure or a thermal cycle partially unloads it. Contact the gasket manufacturer for upper limits. Note that these will be a function of the service temperature.

35.7.8 Stress Cracking

Stress cracking is encouraged by excessive tension in the bolts, particularly if service loads exceed 50% of the yield stress at least for low alloy quenched and tempered steels.

35.7.9 Combined Loads

These loads include weight, inertial affects, thermal effects, pressure, shock, earthquake loading, and so on. Both static and dynamic loads must be estimated. Load intensifiers such as prying and eccentricity should be acknowledged if present. Joint diagrams can be used (see later section) to add external loads and preloading. The parts designed must be able to withstand *worst case combinations of these pre- and service loads.*

Figure 35.13 shows the many factors considered above: each as a function of the UTS of the bolt. The chart describes a hypothetical situation in which a code limit, reduced by a safety factor, was used to pick a desired upper limit on bolt stress of 62% of the UTS of the bolt. Also shown is the minimum clamping required to prevent leaking, vibration, or fatigue problems, again modified by a factor of safety. This suggests a desired lower limit on clamp force corresponding to a bolt tension of 48% of the UTS of the bolt. Remember that these loads *are in-service bolt loads.* Know also that when we preload a joint, we can do so only approximately because (1) when tightening one joint at a time loading one joint influences the load on those joints already preloaded, and (2) after preloading, stress relaxation occurs, reducing the load from the initial torqued value to something less. Both of these factors have a large influence on the total preload achieved in the joint.

In general, fasteners relax rapidly following initial tightening, then relax at a slower rate, as shown in Fig. 35.13.

Figure 35.14 shows the residual stresses in a group of 90 2¼-12 × 29 4330 studs that were tightened by stretching them 79% of their yield stress with a hydraulic tensioner. The studs and nuts were not new but had been tightened several times before these data were taken. Relaxation varied from 5% to 43% of the initial tension applied in these apparently identical studs. In many cases, similar scatter in relaxation also occurs after torquing.

Charts of this sort can be constructed on the basis of individual bolts or multibolt joints. Limits can be defined in terms of force, stress, yield instead of UTS, or even assembly torque.

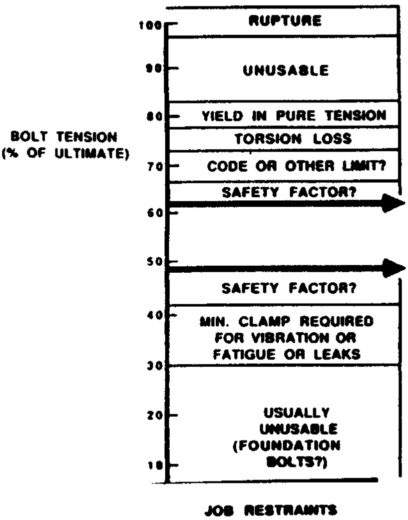

Fig. 35.13 Chart summarizing the design decisions made for a hypothetical joint.

35.8 THEORETICAL BEHAVIOR OF THE JOINT UNDER TENSILE LOADS

In this section, we will examine the way a joint responds when exposed to the external loads it has been designed to support. This will be done by examining the elastic behavior of the joint. When we tighten a bolt on a flange, the bolt is placed in tension; it gets longer. The joint compresses in the vicinity of the bolt.

We need to plot separate elastic curves for the bolt and joint members by plotting the absolute value of the force in each of the two vertical axes and the deformation of each (elongation in the bolt and compression in the joint) on the horizontal axes. See Fig. 35.16.

Three things should be noted.

1. Typically the slope (K_B) of the bolts elastic curve is only ⅓ to ⅕ of the slope (K_J) of the joints elastic curve; i.e., the stiffness of the bolt is only ⅓ to ⅕ that of the joint.
2. The clamping force exerted by the bolt on the joint is opposed by an equal and opposite force exerted by the joint members on the bolt. (The bolt wants to shrink back to its original length and the joint wants to expand to its original thickness.)
3. If we continue to tighten the bolt, it or the joint will ultimately start to yield plastically, as suggested by the dotted lines. In future diagrams, we will operate only in the elastic region of each curve.

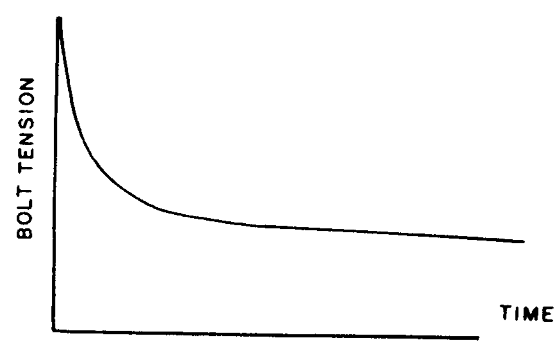

Fig. 35.14 Most short-term relaxation occurs in the first few seconds or minutes following initial tightening, but continues at a lesser rate for a long period of time.

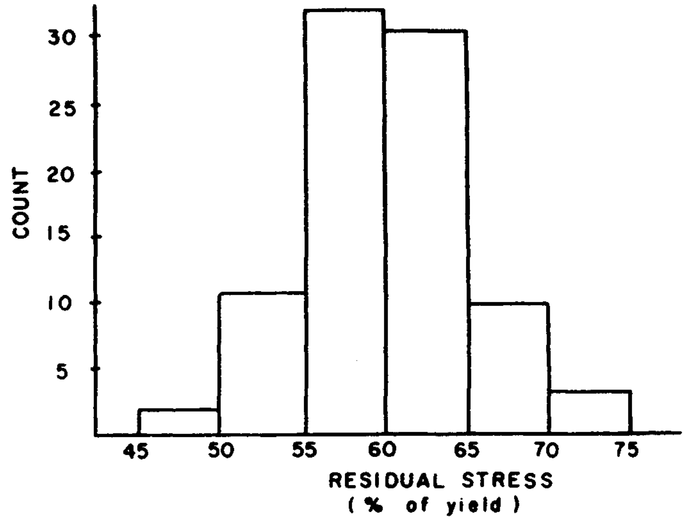

Fig. 35.15 Residual stress as a percentage of yield strength, following removal of tension. Studs were all tensioned to 79% of yield. Torqued to 500 lb-ft.

Rotscher first demonstrated what is called a *joint diagram* (Fig. 35.17).

In Fig. 35.17, the tensile force in the bolt is called the *preload* in the bolt and is equal and opposite to the compressive force in the joint.

If we apply an additional tension force to the bolt, this added load partially relieves the load on the joint, allowing it (if enough load is applied) to return to its original thickness while the bolt gets longer. Note that the increase in the length of the bolt is equal to the increase in thickness (reduction in compression) in the joint. In other words, the *joint expands to follow the nut as the bolt lengthens.*

Because the stiffness of the bolt is only ⅓ to ⅕ that of the joint, for an equal change in the strain, the change in load in the bolt must be only ⅓ to ⅕ of the change in the load in the joint. This is shown in Fig. 35.18.

The external tension load (L_X) required to produce this change of force and strain in the bolt and joint members is equal to the increase in the force on the bolt (ΔF_B) plus the reduction of force in the joint (ΔF_J):

$$L_X = \Delta F_B + \Delta F_J$$

The above relationship is demonstrated in Fig. 35.19.

Any external tension load, no matter how small, will be partially absorbed in replacing the force in the bolt (ΔF_B), and partially absorbed in replacing the reduction of force that the joint originally

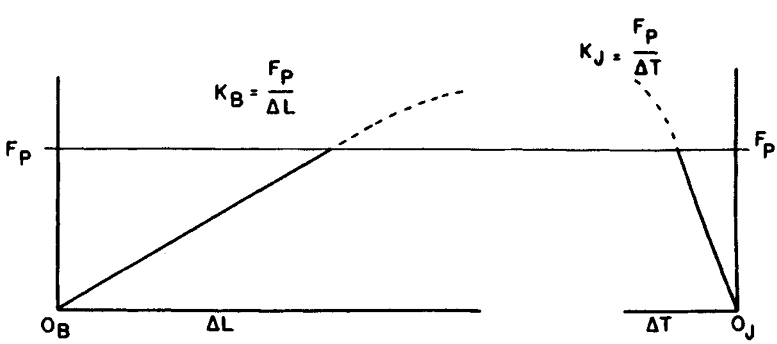

Fig. 35.16 Elastic curves for bolt and joint members.

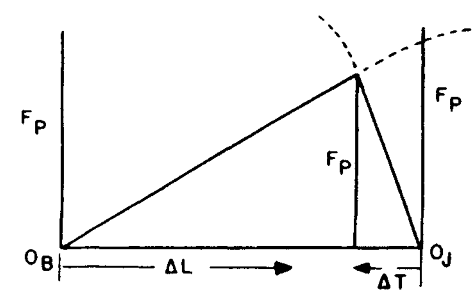

Fig. 35.17 The elastic curves for bolt and joint can be combined to construct a joint diagram. O_B is the reference point for bolt length at zero stress. O_J is the reference point for joint thickness at zero stress.

exerted on the bolt (ΔF_J). The force of the joint on the bolt plus the external load equal the new total tension force in the bolt—which is greater than the previous total—but the change in bolt force is less than the external load applied to the bolt. See Fig. 35.20, which recaps all of this.

We can change the joint stiffness between the bolt and the joint by making the bolt much stiffer (i.e., a bolt with a larger diameter). The new joint diagram resulting from this change is shown in Fig. 35.21.

Note that the bolt now absorbs a larger percentage of the same external load. For another example, see Fig. 35.22 for a softer bolt (less stiff). The joint will see a smaller percentage of a given load.

That the bolt sees only a part of the external load, and that the amount it sees is dependent on the stiffness ratio, between the bolt and the joint, have many implications for joint design, joint failure, measurement of residual preloads, and so on.

35.8.1 Critical External Load

If we keep adding external load to the original joint, we reach a point where the joint members are fully unloaded, as in Fig. 35.23. This is the critical external load, which is not equal to the original preload in the bolt but is often equal to the preload for several reasons.

1. In many joints, the bolt has a low spring rate compared to the joint members. Under these conditions, there is a very small difference between the preload in the bolt and the critical external load that frees the joint.
2. Joints almost always relax after first tightening with relaxations of 10–20% of the initial preload being not uncommon. If a bolt has ⅕ the stiffness of the joint, then the critical external load to free the joint members is 20% greater than the residual preload in the bolt when the load is applied. Therefore, the difference between the critical external load and the present preload is just about equal and opposite to the loss in preload caused by bolt relax-

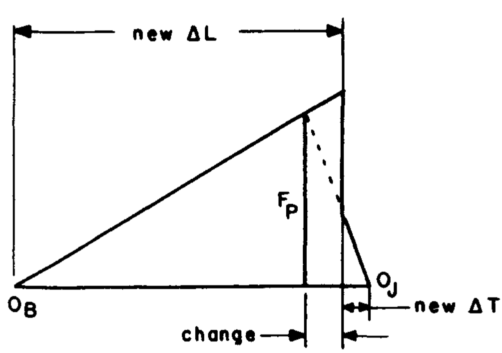

Fig. 35.18 When an external tension load is applied, the bolt gets longer and joint compression is reduced. The change in deformation in the bolt equals the change in deformation in the joint.

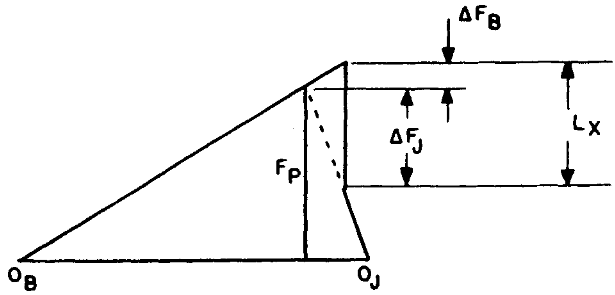

Fig. 35.19 Because the bolt and joint have different stiffness, equal changes in deformation mean an unequal change in force. ΔF_B is the increase in bolt force; ΔF_J is the decrease in clamping force in the joint. L_X is the external load.

ation. In other words, the critical external load equals the original preload before bolt relaxation.

35.8.2 Very Large External Loads

Any additional external load we add beyond the critical point will all be absorbed by the bolt. Although it is usually ignored in joint calculations, there is another curve we should be aware of. The compressive spring rate of many joint members is not a constant. A more accurate joint diagram would show this. See Fig. 35.24.

For joint diagrams, as shown in Fig. 35.19, we can make the following calculations where

F_P = initial preload (lb, N)
L_X = external tension load (lb, N)
ΔF_B = change in load in bolt (lb, N)
ΔF_J = change in load in joint (lb, N)
ΔL, $\Delta L'$ = elongation of the bolt before and after application of the external load (in., mm)
ΔT, $\Delta T'$ = compression of joint members before and after application of the external load (in., mm)
$L_{\text{xcritical}}$ = external load required to completely unload the joint (lb, N) (not shown in diagram)

The stiffness (spring constants) of the bolt and joint are defined as follows:

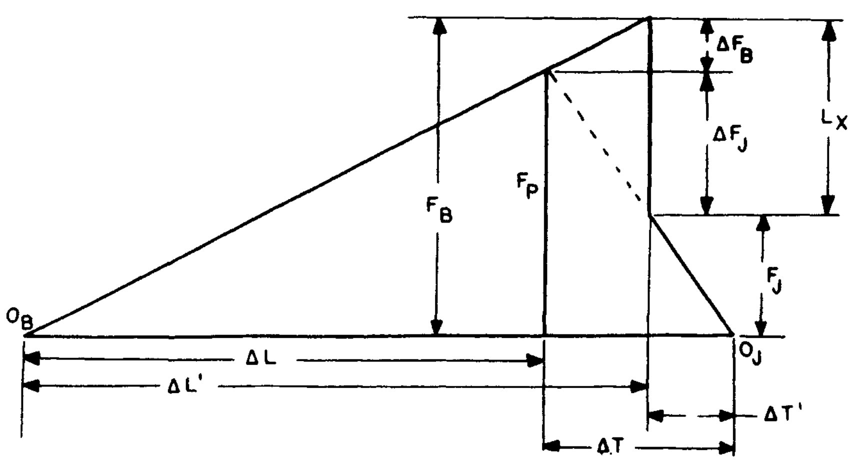

Fig. 35.20 Summary diagram. F_P = initial preload; F_B = present bolt load; F_J = present joint load; L_X = external tension load applied to the bolt.

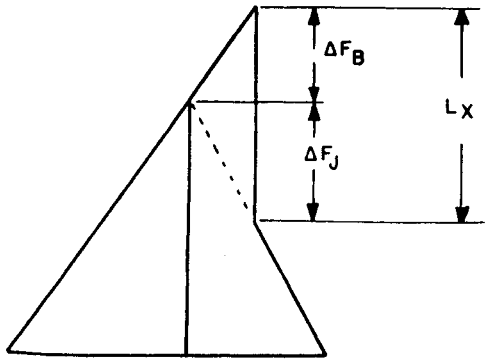

Fig. 35.21 Joint diagram when stiffness of the bolt nearly equals that of joint.

$$\text{for the bolt } K_B = \frac{F_P}{\Delta L} \text{ for the joint } K_J = \frac{F_P}{\Delta T}$$

$$\text{by manipulation } \Delta F_B = \frac{(K_B)}{(K_B + K_J)} \times L_X \text{ until joint separation, after which}$$

$$\Delta F_B = \Delta L_x \text{ and } L_{x\text{ critical}} = F_P\left\{1 + \frac{K_B}{K_J}\right\}$$

Different diagrams must be used if the external load is applied at different planes; but the bolt loads and critical load computed for these situations are only a little different than those calculated from the above diagrams and are often used in joint design and analysis even though they are based on external load applications points that will probably never be encountered in practice. (Remember that we have assumed that the external loads are applied to both ends of the bolt.)

An accurate description of loads created by pressure, weight, shock, inertia, and so on are transferred to the joint by the connected members. This description requires a detailed stress analysis (finite element). A far simpler way to place the load is to define "loading planes" parallel to the joint interface and located somewhere between the outer and contact surfaces of each joint member. Designers then assume that the tensile load on the joint is applied to those loading planes. Joint material between the loading planes will then be unloaded by a tensile load. Joint material outboard of the planes will be trapped between plane of application of the load and the head or nut of the

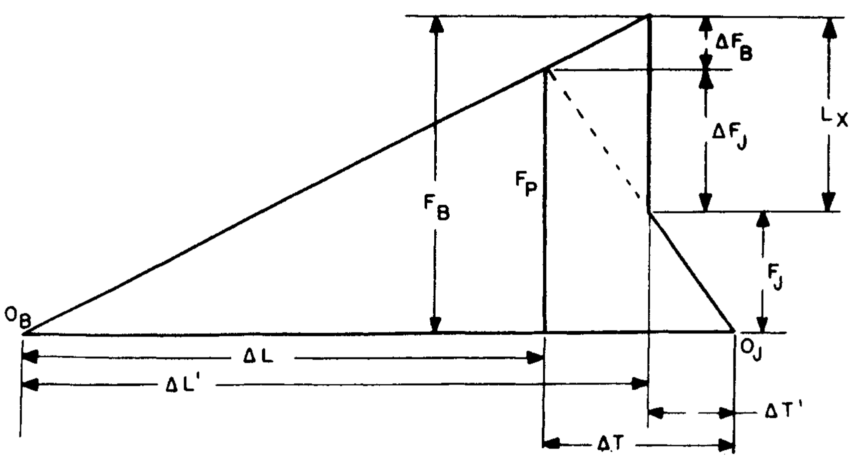

Fig. 35.22 Joint diagram with softer bolt and stiff joint.

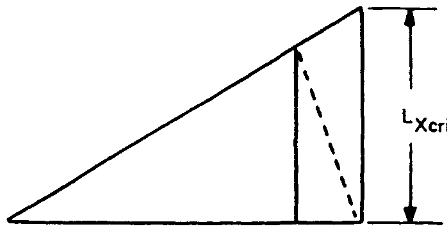

Fig. 35.23 A critical external load (L_{xcrit}) fully unloads the joint (but not the bolt).

fastener. Therefore, loading planes coincide with the interface between upper and lower joint members. To illustrate this, see Fig. 35.25, where (*a*) represents the original assumption and (*b*) the new assumption.

If the same tensile force, however, were applied *to the interface* between the upper and lower joint members, both the bolt and the joint (in compression) would be loaded by the external load. In this case, we are relieving these flange-on-flange forces rather than adding to them. In the joint, this means that the external load reduces the flange-on-flange force without increasing the total force in either of the flange members or the bolt. The joint diagram for the above situation is shown in Fig. 35.26.

In the above case, both elastic curves (bolt and joint) are drawn on the same side of the common vertical axis that represents the original preload or F_P. This is done because both springs are loaded by the external force.

When the external load equals the original preload in the bolt, it will have replaced all of the force that each joint member was exerting on the other. Note that neither bolt deformation nor joint deformation has changed to this point.

Increasing the external load beyond this point now adds to the original deformation of both the bolt and the joint members. That is, the bolt gets longer and the joint compresses more. The joint diagram merely gets larger. See Fig. 35.27 (dashed lines represent the original joint diagram; the solid lines represent the new joint diagram). Note that at all times, both the bolt and the joint see the same total load.

The mathematics of a tension load at the interface are very simple and can be determined by inspection of the joint diagram above. The change in the bolt force is

$$\Delta F_B = 0$$

until the external load exceeds the preload (F_P), after which

$$\Delta F_B = L_X - F_P$$

$$\text{or further } \Delta F_B = \text{further } \Delta L_X$$

The critical external load required to cause joint separation is

$$L_{x\ \text{critical}} = F_P$$

and this is true regardless of the spring constants, or spring constant ratios of the bolt and joint members.

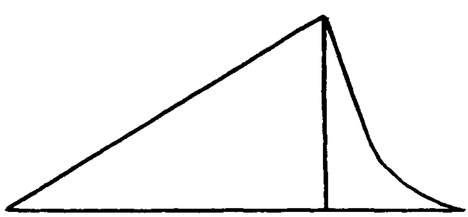

Fig. 35.24 The spring rate of the joint is frequently nonlinear for small deflections.

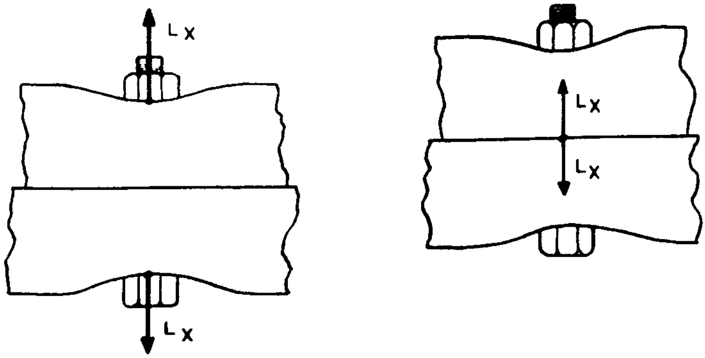

Fig. 35.25 The external tension load is applied at the joint interface.

For our first joint diagrams, the keys were that the change in elongation of the bolt under an external load equaled the change in compression of the joint, but the changes in force were unequal. In the present case, where the tension load is applied at the interface, the deflections are not equal, but the force in the bolt is always equal to the forces in the flange. This is the inverse of the first joint diagrams.

The mathematics above tells us that there will be no change in the force seen by the bolt when tension loads are applied at the interface *until the external load exceeds the original preload in the bolt.* On the other hand, interface loading gives us a critical external load (i.e., joint separation) that is equal to the preload. This is less than the load required for separation when the tension loads are applied at joint surface. The load capacity of the interface joint is *less* than the load capacity of the original joint.

Maximum bolt load, working change in bolt load, and the critical external load are important design factors. They are different for different loading planes; hence the importance of loading planes in our calculations. Loading planes that are in between the two situations that we have looked at will result in "in-between" values for all factors of interest. Designers can assume that the true values lie somewhere between the two conditions we have shown (depending on the position of the actual loading plane)—which, by the way, represent the limiting conditions; that is, head-to-nut and joint interface. Assuming the "worst case" scenario, which is given in Fig. 35.23 (or even a loading plane halfway to the interface) can be so conservative as to seriously affect the assumptions about the amount of change in bolt load actually created by an external load. Experimental data on "compact" and other pressure vessel and piping flanges report that the changes in bolt load in such rigid joints are often only about one-tenth of the change that would be predicted by Fig. 35.18 and the following equation:

$$\Delta F_B = \frac{(K_B)}{(K_B + K_J)} \times L_X$$

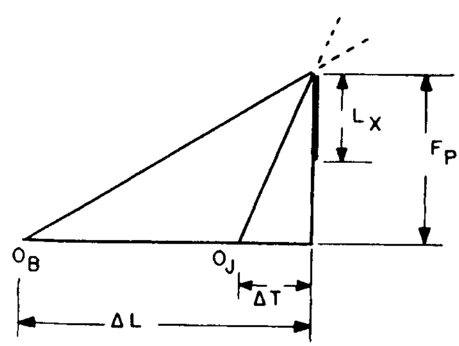

Fig. 35.26 Joint diagram when external tension load is applied at joint interface. ΔL elongation of bolt; ΔT compression of joint; F_P = original preload.

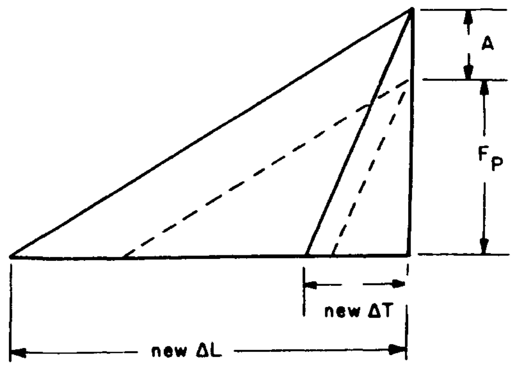

Fig. 35.27 The external load applied to the joint interface has exceeded the critical load by an amount = A.

This suggests that a joint designed to the above equation might have larger and/or more numerous bolts than necessary to support pressure loads the bolts will never see. The ASME Boiler and Pressure Vessel Code takes an even more conservative point of view than that described by the above equation to introduce a factor of safety. This code assumes that the bolts see 100% of external load L_X, not an amount reduced by the stiffness ratio.

35.9 EVALUATION OF SLIP CHARACTERISTICS

A slip-resistant joint is one that has a low probability of slip at any time during the life of the structure. In this type of joint, the external applied load usually acts in a plane perpendicular to the bolt axis. The load is completely transmitted by frictional forces acting on the contact area of the plates fastened by the bolts. This frictional resistance is dependent on (1) the bolt preload and (2) the slip resistance of the fraying surfaces.

Slip-resistant joints are often used in connections subjected to stress reversals, severe stress fluctuations, or in any situation wherein slippage of the structure into a "bearing mode" would produce intolerable geometric changes. A slip load of a simple tension splice is given by

$$P_{\text{slip}} = k_s m \sum_{l=1}^{n} T_i$$

where

$$k_s = \text{slip coefficient}$$
$$m = \text{number of slip planes}$$
$$\sum_{l=1}^{n} T_i = \text{the sum of the bolt tensions}$$

If the bolt tension is equal in all bolts, then

$$p_{\text{slip}} = k_s\, m\, n\, T_i$$

where

$$n = \text{the number of bolts in the joint}$$

The slip coefficient K_s varies from joint to joint, depending on the type of steel, different surface treatments, and different surface conditions, and along with the clamping force T_i shows considerable variation from its mean value. The slip coefficient K_s can only be determined experimentally, but some values are now available, as shown in Table 35.1.

35.10 INSTALLATION OF HIGH-STRENGTH BOLTS

Prior to 1985, North American practice had been to require that all high-strength bolts be installed and provide a high level of preload, regardless whether or not it was needed. The advantages in such an arrangement were that a standard bolt installation procedure was provided for all types of con-