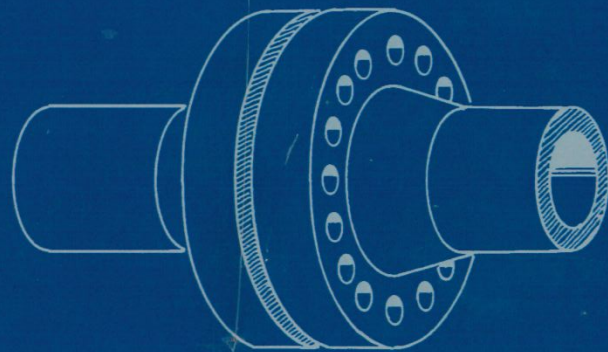


# **AN INTRODUCTION TO THE DESIGN AND BEHAVIOR OF BOLTED JOINTS**

**THIRD EDITION, REVISED AND EXPANDED**



**JOHN H. BICKFORD**

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function of the work hardening which has been done on the bolt by taking it past yield on the first cycle, as noted in Fig. 2.2. Loading it past this new yield point will create additional permanent deformation; but *unless* we take it well past the new yield point we won't damage or break the bolt by yielding it a little—in fact, we'll have made it a little stronger, at least as far as static loads are concerned.

Many bolt materials can be taken past initial, and new, yield points a number of times before they will break. The stronger, more brittle materials, however, can suffer a loss of strength by such treatment, as shown, for example, by Fisher and Struik [1]. Loss of strength in several ASTM A490 bolts, because of repeated cycling past yield (under wind and water loads), has been publicly cited as a contributing factor in the 1979 collapse of the roof of the Kemper Auditorium in Kansas City, for example [2].

### C. Stress Distribution Under Tensile Load

Let's place a bolt in a joint and load it in pure tension (this is possible if we use a hydraulic tensioner instead of a wrench to tighten the bolt). If the bolt is perfectly symmetrical, the faces of the head and nut are exactly perpendicular to the axis of the threads, joint surfaces are flat and parallel, etc. (most of which we'll never encounter in practice), loading the bolt and joint this way will produce the stress distribution shown in Fig. 2.3 [3,4].

Two points are worth noting:

Even though the bolt has been loaded in pure tension, it is well supplied with compressive stress, thanks (for example, in the shank) to Poisson's reduction and thanks to the fact that the tension built into the bolt during the tightening process must subsequently be held there by compressive forces in the nut, bolt head, and joint members.

Complex though the picture is, it's a far cry from the "truth" in most applications, where it is complicated by imperfect geometry as well as by the presence of torsional, bending, and shear stresses, as we'll discuss later. Our picture, furthermore, ignores localized, but often significant, stress concentrations in the threads.

### D. Stress Concentrations

Figure 2.3 gives us a simplified view of the directions of stress in a loaded bolt and joint. An analysis of stress magnitudes would reveal three danger points, where stress concentrations create stress levels well beyond the average. These points are the fillet, where the head joins the body; the

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### Tensile Load

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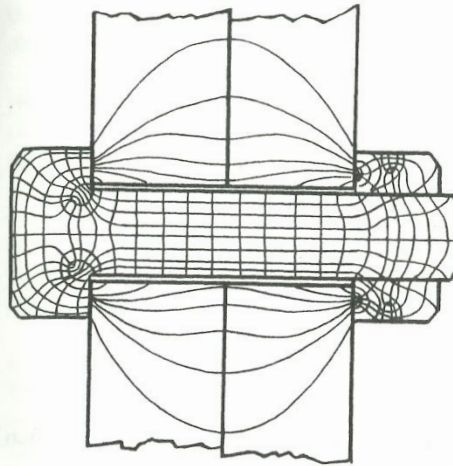


**Figure 2.3** Lines of principal tension and compression stress in a bolt loaded in pure tension (and lines of principal compression stress in the joint).

thread run-out point, where the threads meet the body; and the first thread to engage the nut. As we'll see, these are the points at which the fastener will usually fail.

### E. Magnitude of Tensile Stress

In much of what follows, we'll find it useful to know something about the magnitude of tensile stress within the fastener. For example, many of our calculations will be based on the assumption that tensile stress is zero at the free ends of the bolt and that it rises uniformly through the head to the level found in the body, as suggested by Fig. 2.4. There's a similar pattern in the threaded end, but the average stress in the threaded section is higher than the average in the body because the cross-sectional area is less in the threads.

A finite element analysis made by General Dynamics-Fort Worth [4], however, suggests a far more complex pattern. Scientists at General Dynamics say that the magnitude of tensile stress along the *axis* of the bolt does approximate that shown in Fig. 2.4, but that the magnitude of tension along other lines parallel to the axis of the bolt looks more like