On the STRENGTH of Highly Stressed, Dynamically Loaded BOLTS and STUDS

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THE strength of most highly loaded bolts and studs is determined by the man with the wrench and not by the designer, the metallurgist, or by the manufacturing processes. In highly loaded bolts and studs we rarely find the design so bad, the fabrication practice so poor, or the material so weak as to cause failure in service provided the nut is properly tightened against reasonably rigid members.

There are, of course, advantages to be gained by careful design. Good materials and heat-treatments should be used; some manufacturing practices are better than others, but all of these are relatively unimportant in severe service in comparison to proper tightening of the nut during assembly and to maintaining that tightness during service. We should, therefore, be less concerned with design details and give more attention to the technique of tightening nuts and to the ability of the bolted assembly to maintain the required tightness.

An estimate of the relative responsibility for the strength of bolts and studs among the individuals who are concerned from design to assembly is shown in Fig. 1.

A properly tightened nut is one that applies a tension load to the bolt or stud that is equal to or greater than the external load that is to be supported in service. When this condition is fulfilled and maintained against reasonably rigid bolted assemblies, the bolt can not fail by fatigue because it can experience practically no change in stress regardless of the fluctuating nature of the operating load. It can not fail statically because, to be tightened as specified, it must be capable of supporting the greatest operating load.

■ Fatigue Strength Varies with Stress Range

Numerous fatigue tests have shown that the fatigue strength of metal decreases as the range of dynamic stress (stress change) to which the metal is subjected is increased and, conversely, that the fatigue strength is increased as the dynamic stress range is decreased. As the stress change approaches zero, the dynamic load that can be supported approaches the tensile strength of the material.

Consider the case of a bolt subjected to fluctuating load, such as in a connecting rod. If the nut is tightened just enough to make contact with the bearing cap, the load that will be applied to the bolt will vary from zero, the initial tightness, to the maximum inertia load of the piston and the connecting rod. Under this large stress change,

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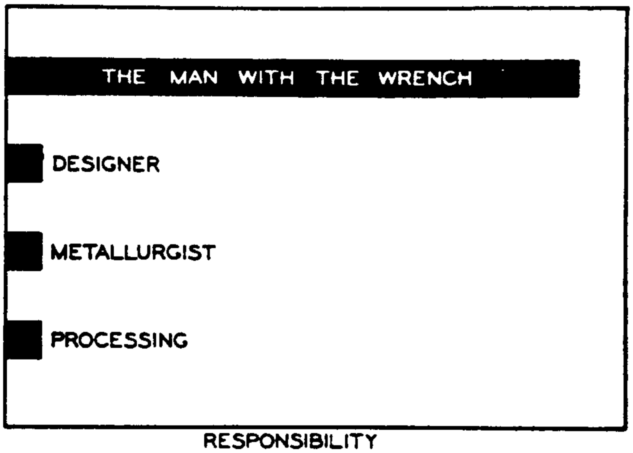

■ Fig. 1 – Relative responsibility for fatigue durability of bolts

EXTENSIVE study by Mr. Almen on the strength of highly stressed, dynamically loaded bolts and studs has led him to the following conclusions:

1. The strength of dynamically loaded, highly stressed bolts and studs is determined by the man with the wrench. Other considerations, such as design, material, and processing, are of relatively minor importance.

2. The elasticity of bolts and studs should be as great as possible.

3. The bolted members should be as rigid as possible.

4. Loss of dimension of the bolted members is particularly hazardous to the strength of short bolts and studs.

5. No adequate means are available for determining nut tightness except when the elongation of the bolt can be measured.

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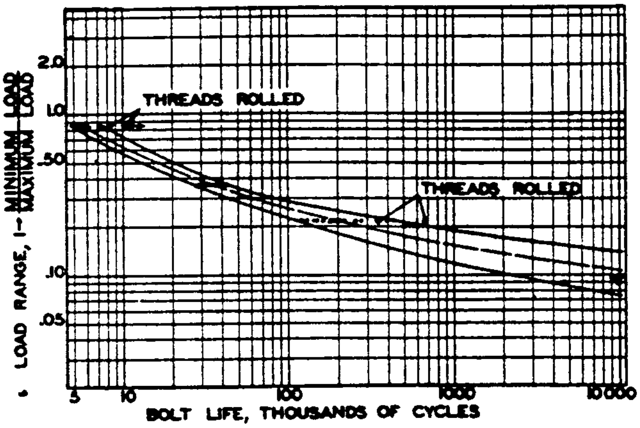


■ Fig. 4 – Bolt fatigue curve

■ Stress Fluctuation Unavoidable

When the nut is insufficiently tightened, each load application will elastically elongate the bolt and a gap will be formed between the bolted members, the magnitude of which will be reduced as the initial tightness is increased. When the nut is tightened to near the operating load, this separation of the bolted members will not occur, but there will, nevertheless, be a small elastic elongation of the bolt.

When the nut is tightened, the bolt is elastically elongated and, at the same time, the bolted abutments are elastically shortened; the amount of shortening depending upon such variables as the design, the elastic characteristics of the bolted materials, and the area of the material supporting the bolt load. On application of the external operating load, the bolt is acted upon by the external load plus the load of elastic recovery of the bolted assembly. The sum of these forces will always be greater than the initial bolt tension and, therefore, the bolt will experience a change of stress at each load application regardless of the magnitude of the initial tension. The stress change will be greater as the elastic deformation of the bolted assembly is increased and it will be reduced as the rigidity of the bolted assembly is increased.

■ Stress-Strain Measurements

As a preliminary to the fatigue testing of the bolts described above, stress-strain measurements were made for each bolt. The fatigue machine used for these tests is equipped with gravity scales whereby the load applied to the specimen may be accurately weighed at any position of the load-applying device. It was, therefore, a simple matter to take stress-strain measurements on each bolt while in place in the fatigue machine by weighing the load and measuring the elongation of the bolt at frequent intervals.

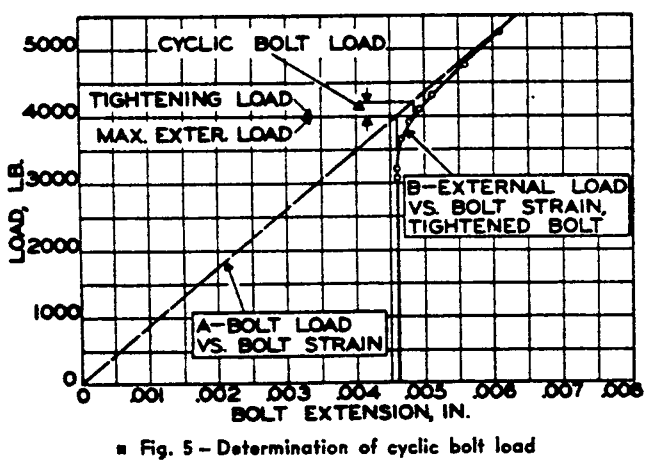
A typical plot of stress-strain data taken from one of the test bolts is shown in Fig. 5. In this plot the inclined straight line, having its origin at zero load and zero elongation, is the load extension curve of the bolt alone (curve A). The plotted points, curve B, were taken from a bolt that had been tightened against relatively rigid abutments to an initial load of 4000 lb, under which load the bolt elongated 0.0046 in. As the external load was applied to this tightened bolt, the load extension curve B, as shown by the plotted points, does not rise vertically to join the inclined load extension curve of the bolt alone (A) as would occur under ideal conditions of an elastic bolt and perfectly rigid abutments. The plotted points (B) join the vertical line at 0.0046 in. elongation and the inclined line (A) by an easy curve thus showing that the bolted members are elastic and that the load on the bolt is augmented by the elastic recovery of the bolted members.

■ Effect of Elasticity

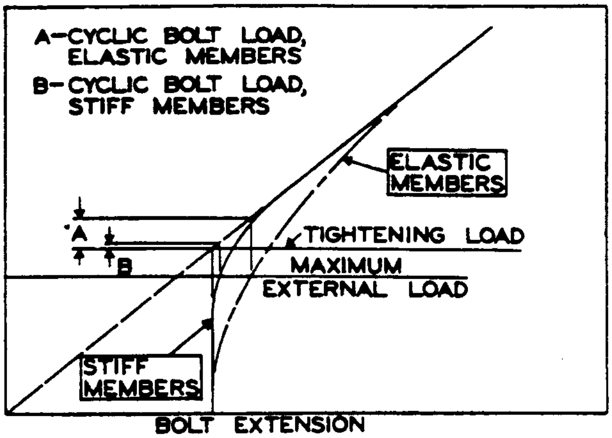
This added load increment due to the elasticity of the bolted members can be measured from the curves. For the case in which the external load is equal to the initial bolt tension, extend the bolt preload line (4000 lb) to intersect the curve of the plotted points *B*, which will be found to be at a bolt extension of 0.00485 in. Since the extension under the bolt preload was 0.0046 in., the bolt elongation has now increased 0.00025 in. Now project the new bolt extension line (0.00485 in.) vertically to intersect the load extension curve (A) of the bolt alone. This will correspond to a load of 4220 lb or a load increase of 220 lb over the initial bolt tension.

The load range experienced by the bolt under these conditions would be $1 - \frac{4000}{4220} = 0.052$, a stress range so small as to be practically zero (see Fig. 4) and the bolt load is, therefore, almost static. In like manner, the resultant bolt load may be found for any external load whether it be greater or less than the initial bolt tension.

The plot Fig. 5 was made from a fatigue test fixture in which the bolted members were relatively massive and rigid and care was used to assure parallelism and smoothness of the contacting surfaces. For these reasons the elastic compression curve of the bolted members is so steep as to be immeasureable and it, therefore, coincides with the vertical preload line at 0.0046-in. bolt extension up to a load of about 3400 lb. From this point the elastic curve (B) swings to the right and joins the bolt elastic curve (A) at about 0.0060-in. bolt extension and 5220-lb load. The elastic compression curve for this bolted assembly is such that external loads up to 3400 lb add very little to the bolt preload. The nonlinear portion of the curve is presumably due to bending of the bolted members result-



■ Fig. 5 – Determination of cyclic bolt load

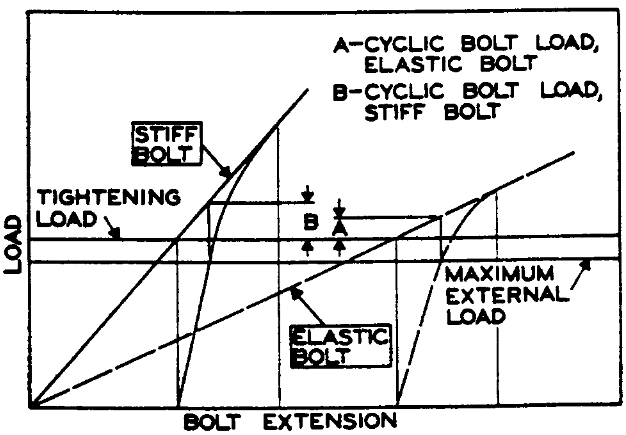


■ Fig. 6 – Effect of elasticity of bolted members

ing from imperfect surfaces and to bending of the loading beam under the external load. As the external load is increased through the nonlinear range, the cyclic increment of bolt load above the initial bolt tension increases rapidly until complete separation of the bolted parts occurs at 5220 lb and 0.0060-in. bolt extension.

The schematic chart Fig. 6 shows two elastic curves to illustrate the effect of varying the elasticity of bolted members. From the method of analysis described above, it will be seen that the cyclic load on the bolt is increased as the elasticity of the bolted members is increased. Similarly, Fig. 7 shows that the cyclic bolt load decreases as the elasticity of the bolt is increased.

The strength of a bolt may, therefore, be said to be influenced by the design of the bolted assembly and the materials used in the assembly. A properly tightened bolt will be weaker when used in combination with aluminum abutments than when used in combination with steel abutments of similar proportions because of the greater elastic deformation of aluminum. It will be weaker when used against abutments of small area than when used against abutments of large area. The number of parts making up the bolted assembly will also affect the bolt strength because mating parts will always fit imperfectly, thus adding bending deflection to elastic compression, and each part will, therefore, add to the total elastic deformation of the assembly.



■ Fig. 7 – Effect of elasticity of bolt

■ Arrangement Affects Bolt Strength

The arrangement of parts in a bolted assembly can have great effect on bolt strength. An exaggerated case of this kind is shown in Fig. 8, in which identical parts are used in two assemblies, in one of which the bolt would be weak and in the other, the bolt would be strong.

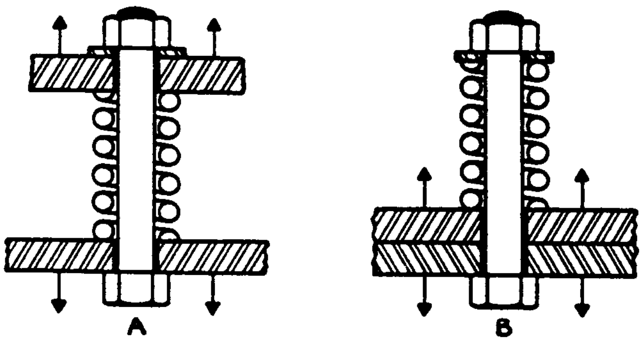
In Fig. 8A, a bolt is tightened against two stiff plates that are spaced apart by a low-rate spring. The external load is applied against the bolt by the two stiff plates. In this case the coil spring is a part of the bolted assembly and since it is very elastic as compared to the elasticity of the bolt we may, for purposes of discussion, consider its rate to be zero. When an external load equal to the initial bolt tension is applied to the plates, the total bolt load will be twice the initial tension, because the spring load will not be appreciably reduced by the additional extension of the bolt and its load will be added to the external load. The stress range experienced by this bolt will, therefore, be $1 - \frac{1}{2} = 0.5$. For the stress magnitudes of the bolts shown in Fig. 4, it will be seen that at a stress range of 0.5 early failure may be expected.

In Fig. 8B, the external load is also applied to the plates, but the low-rate spring now is a part of the bolt instead of a part of the bolted assembly as in Fig. 8A. We, therefore, have a very elastic bolt tightened against relatively very rigid members. When an external load equal to the initial bolt tension is applied to the plates, the load between the plates will be reduced to zero but the bolt will experience no appreciable change in load. The stress range will be $1 - 1 = 0$; that is, the load will be static and fatigue failure can not occur (see Fig. 4).

■ Practical Elastic Considerations

While the arrangements shown in Fig. 8 are rather fanciful, we have assemblies in actual practice that are somewhat similar. Fig. 9 shows a condition frequently met in practice in which a gasket is clamped between two mating plates. The gasket is compressed locally opposite the bolt, thus bending the plates until they serve as the spring shown in Fig. 8A. When the external load is applied, the bolt will feel the elastic load of the bent plates plus the external load, a condition that invites trouble.

Also in such assemblies the bolt or stud is usually short



■ Fig. 8 – *A*: spring is part of abutments – bolt is weak; *B*: spring is part of bolt – bolt is strong and relatively inelastic and we, therefore, have all the conditions necessary for large stress range and resulting fatigue failure. When gaskets are necessary, the gasket area should be large in the vicinity of the bolt and relatively smaller in proportion to the distance from the bolt, in order that the unit pressure on the gasket may be maintained with minimum bending of the bolted surfaces. The gasket should be as inelastic as possible and still serve its purpose.

A practical assembly having the characteristics of the assembly in Fig. 8B is shown in Fig. 10, in which a spring washer is clamped between the bolt head or nut and the bolted assembly. When such a spring washer is dimensioned to support the required external load or the initial bolt tension, whichever is the greater, within the elastic range of the washer, the effective bolt elasticity is increased and, therefore, the bolt fatigue strength is increased.

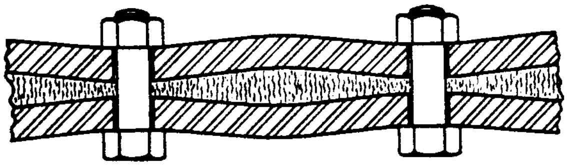
Likewise any other expedient that will increase the rigidity of the bolted assembly or that will increase the elasticity of the bolt will increase the fatigue strength of the bolt.

■ Tightness Must Be Maintained

Although properly tightened bolts are stronger as the rigidity of the bolted assembly is increased and as the elasticity of the bolt or stud is increased, a bolt or stud that is incapable of maintaining an initial tension equal to the external tension load is quite likely to fail in severe service. Thus, we find relatively greater mortality among short bolts and studs than among long ones because of their lesser elastic yield.

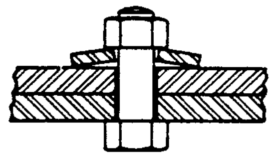
Consider a short stud such as is often used to attach engine cylinders to the crankcase. The thickness of the cylinder flange plus the thickness of the washer may be less than ½ in. When the nut is tightened, the stud is elastically elongated through the length from near the bottom of the nut to near the last thread engaging the crankcase. Assuming that this distance is ½ in. and that the nut, at proper tightness, stresses the stud to 120,000 psi, the stud will then be elastically elongated 0.002 in. If during operation, the bolted assembly is reduced in thickness by 0.001 in. through wear, corrosion, embedding, or by displacement of material such as soft plating, the stud will lose one-half of the required tension, and fatigue failure will follow.

Greater safety is assured when longer studs are used. For example, when under the same conditions as described above, a stud of 2-in. effective length is used, the loss of tension will be only one-eighth of the initial tension; that is, a loss of 0.001 in. from a total elastic elongation of



■ Fig. 9 – Bolts subject to fatigue – elastically deformed plates and gasket increase maximum and cyclic bolt loads

■ Fig. 10 – Spring augments bolt elasticity – bolt life is increased



0.008 in. This would be serious only when other factors, such as initial tension, are near the low limit.

It will, therefore, be seen that among the major bolt and stud hazards, particularly when they are short, must be included loss of thickness of the bolted assembly as a result of surfaces that can embed, soft-plated coatings that can be displaced, gaskets that can be plastically compressed, and materials that can yield at elevated operating temperatures.

■ Yield Point May Be Exceeded

In tightening nuts, it is far better to stress the bolt above the yield point of the material than to risk undertightness. Since for adequate size bolts, the load is practically static, there can be no harm in yield provided that the plastic deformation is not so great as to reduce the static strength of the bolt. The amount of yield that can be tolerated will depend on the length of the bolt, the design of the bolt, and the characteristics of the material. When the body of the bolt or stud is equal in diameter to the outside diameter of the threads, yield will be concentrated at the roots of the threads whether the bolt be long or short. In such cases, little yield, as measured by bolt elongation, can be tolerated. For bolts or studs in which the body diameter is equal to or less than the thread root diameter, greater yield, as measured by bolt extension, can be tolerated since yield will occur over the entire body length.

It is not intended to recommend that all bolts should be tightened above the yield point but only to show that, where necessary, the practice may be followed provided proper caution is used. The necessity for tightening above the yield point would occur in such cases as when the bolt yield strength is only slightly greater than the maximum external tension load or when cotters are used in combination with castle nuts. In the latter case, it is often necessary to advance the nut beyond the required tightness, in order to admit the cotter since the nut should not be slackened to meet the cotter pin hole. For short bolts, two cotter holes should be provided at such angles as to register with the notches in the nut at 30-deg intervals, thus reducing the overtightness in critical cases.

■ Bending Loads

In addition to tension loads, a large percentage of bolts and studs are subjected to bending loads of various magnitudes. Stress changes from repeated bending loads are generally of smaller magnitude than stress changes from tension loads but since these occur simultaneously, the stress change in the bolt will usually be their sum. As in the case of repeated tension loads, the repeated bending loads, in normal designs, will be reduced as the nut tightness is increased and for the same reason. Also as in the case of tension loads, there will always be a small change in bending load regardless of nut tightness because of stressing roller, but this effect was probably slight as compared to the compressive prestress.

Although the rolling operation increased the fatigue resistance of these bolts in the ratio of two to one, it will be seen from Fig. 4 that the rolling was only the equivalent of a slight change in stress range as would be obtained by greater initial bolt tension. Specifically, the unrolled bolts tested at stress range 0.846 would have had the same average durability as the rolled bolts had the nuts been turned 15 deg tighter. The unrolled bolts tested at stress range 0.215, which more nearly represents normal practice, would have had the same average durability as the rolled bolts had the nuts been turned 4 deg tighter. This comparison is not made to discourage improved fabrication practice, but only to emphasize the importance of proper tightening.

Commercial rolled threads can and often do take advantage of compressive prestressing intentionally or otherwise. If rolled threaded bolts are heat-treated after rolling, the prestress is lost. To retain the prestress, rolled threads should not be heated sufficiently to relieve the stress set up by the rolling or the bolts should be given a final rolling after heat-treating.

Stressing the thread roots beyond the yield point probably has an effect similar to overstressing springs, which is known to be beneficial in increasing their fatigue durability. Overloading should have other effects because it should result in better distribution of the load among the threads engaging the nut as well as reducing local stress concentration in the most highly loaded thread.

The details of design and the finish of the shank of bolts is perhaps of the same order of importance as the design and finish of the threads. The diameter of the shank should be made as small as possible consistent with the required strength in order to increase the elasticity of the bolt both in elongation and in bending. Generous fillets should be provided at all section changes and the reduced diameter should be carried up to the threaded section. Additional gain may be had by compressively prestressing the shank surface as by rolling or by shot peening.

■ Nut Tightening Practices

The secondary specifications of bolts and studs, such as chemical composition, heat-treatment, thread dimensions, body dimensions, and finish, are carefully controlled, but the really important consideration, nut tightness, is very loosely specified or entirely ignored. Among the reasons for this lack of control is the failure of engineers and metallurgists to appreciate the vital importance of nut tightness and the inadequacy of available means of measurement.

The most popular means of specifying nut tightness is through the use of torque wrenches, but it is no exaggeration to say that a good mechanic who has developed wrench "feel" is more reliable than the most elaborate torque wrench.

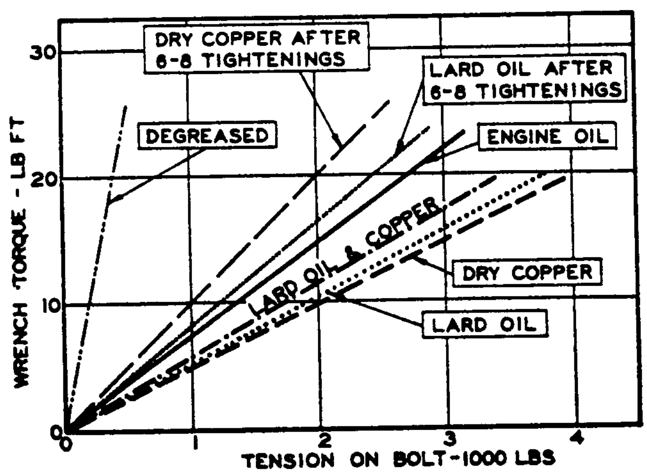
By specifying nut torque, all nuts of the same size are presumed to give the same bolt tension. However, the friction between the nut and the abutment, and the friction between the threads of the bolt and the nut are so variable that the bolt tension will vary over wide limits. Fig. 12 shows that bolt tension may vary as much as ten to one under extreme conditions of lubrication as from degreased steel to copper plate as a lubricant. The variation in practice, however, will rarely be more than three to one or less than two to one. This is, of course, still so great as to condemn nut torque measurement as a specification for so critical an operation as preloading severely stressed bolts. It would mean much to industry if means can be found for reducing the variability of nut friction to tolerable limits and, thus admit the use of the torque wrench as an accurate tool.

A recent innovation in nut tightening technique is the use of the nut as a rough micrometer. The nut is first tightened sufficiently to seat firmly all contacting surfaces, after which it is fully released and retightened "finger tight" which is taken as zero setting. From this setting, the nut is turned through a specified angle to produce the desired bolt or stud tension. The results from this procedure are more accurate than from nut torque specifications, but it is very cumbersome. The angle through which the nut should be turned must be varied with bolt size, bolt length, number of threads per inch, kind of bolted material, area of bolted material, and other variables. The technique is probably justified in spite of the complications in original assembly and the greater complications in servicing shops because of the greater assurance of tightness that is obtainable. It serves, also, to emphasize the desperate need for nut-tightening accuracy.

In the cases where both ends of the bolt are accessible, good accuracy may be assured by measuring bolt elongation by micrometer or by indicator. This practice is followed by most aircraft-engine builders for accessible bolts with satisfactory results. At least one manufacturer tightens connecting-rod bolts to a micrometer extension so great as to exceed the yield point for the bolts lying near the low limit of hardness. No trouble has been experienced since this practice was adopted.

■ Utilizing Spring Members

It is probable that in many instances spring members such as are shown in Fig. 10 can be profitably used. This would add to the bolt elasticity, thus providing (1) greater safety against loss of tension by embedding, corrosion, yielding, and so on, and (2) greater safety by reducing the stress range experienced by the bolt. The deflection of the spring member could probably be used as a measure of the bolt tension with reasonable accuracy since the deflection



■ Fig. 12 – Effect of lubricants on bolt tension could be relatively great and, therefore, easily measured. It would be necessary to design these spring members to support the desired bolt tension within the elastic range of the spring and to avoid clamping them solidly against the abutments. They could be made in the form of disc springs or as beams depending upon the available space and only one size would be required for each bolt size.

Conclusions

1. The strength of dynamically loaded, highly stressed bolts and studs is determined by the man with the wrench. Other considerations, such as design, material, and processing, are of relatively minor importance.

2. The elasticity of bolts and studs should be as great as possible.

3. The bolted members should be as rigid as possible.

4. Loss of dimension of the bolted members is particularly hazardous to the strength of short bolts and studs.

5. No adequate means are available for determining nut tightness except when the elongation of the bolt can be measured.