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Fatigue Failures in Common Machine Parts

IN AN ARTICLE in the February issue of this magazine the general problem of fatigue failure was discussed, and it was pointed out that in the great majority of cases failure starts at a surface where high tension stresses are localized. It was demonstrated that if such a surface were prestressed in compression the liability to such fatigue failure would be correspondingly lessened, a fact which is utilized in practice by peening the surface. Methods of doing this, and of estimating the resulting compressive stresses, were also outlined.

Bolt Failures

The fatigue vulnerability of bolts and studs has been discussed in many papers (listed in the excellent book by the Battelle staff on "Prevention of the Failure of Metals Under Repeated Stress") and the improvements resulting from reducing the diameter of the bolt body and from pressure rolling of the threads have been adequately recorded. Insufficient attention, however, has been given to fatigue vulnerability due to insufficient bolt tightness.

A bolt or stud should be tightened to a load exceeding the maximum working load. When properly tightened against rigid members, a bolt or stud cannot fatigue because there can be no change in stress; the bolt load is static even though the load applied to the bolted member oscillates at high frequency from zero to a maximum. This rule must, however, be applied with caution because all bolted members are elastic in some degree and the design of the bolted members may be such that the applied load is greater than can possibly be supported by the fastenings. An exaggerated case of this kind is shown in Fig. 1 *(a)* in which the connecting rod bolts are excessively stressed in tension and in bending because the distance L from the bolt to the point C is small and, since the bolted parts tend to bend about the point C as a fulcrum, the tension and bending loads in the bolts are great. An improved design is shown in Fig. 1 *(b)*, in which the fulcrum point C' is further removed from the bolt and, therefore, the tension and bending loads are reduced. This is a case in which fatigue failure of one member is due to faulty design of another. We must remember that, like an aching tooth, the one that seems to hurt is not necessarily the one that should be pulled.

If the bolt in Fig. 1 *(b)* should fail by fatigue it could still not be charged to insufficient strength because, as stated above, if the initial bolt tension is less than the maximum external load the stress range under repeated loads is increased. Let us suppose that the bolt is tightened just enough to bring the surfaces into contact without appreciable tension. Under alternating stress the stress range would be from

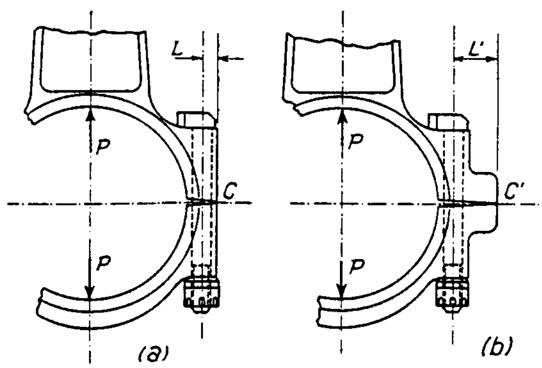


Fig. 1—Fatigue Failure of Bolt Due to Elastic Changes in Bolted Member Cured by Increasing the Effective Length L', to the Right

zero to maximum and fatigue failure could only be avoided by greatly increasing the bolt strength. As the initial bolt tension is increased the stress range is decreased until it approaches zero when the initial bolt tension is equal to or greater than the maximum working load. This would illustrate a case in which a failure is not the fault of either bolt strength or design but is chargeable to bad assembly.

The vulnerability to fatigue as a function of bolt tightness is shown in Fig. 2. In the tests plotted all bolts were subjected to a cyclic tension load of 9210 lb. but were tightened to initial tensions of 1420, 5920, 7220 and 8420 lb. Fifteen bolts were tested at each of the three lower loads in order to establish the scatter band for this kind of specimen. Only two bolts were tested in which the initial tension was 8420 lb., one of which failed after 4,650,000 stress cycles and the second had not failed after 10,000,000. Bolts used in these tests were $\frac{3}{8}$ in. diameter, accurately dimensioned and finished; threads were U. S. form, 24 threads per in. and ground to close limits.

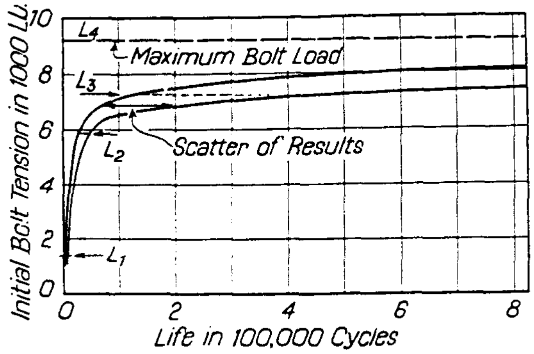


Fig. 2—Effect of Tightness (Setting-Up Tension) on Life of Bolt in Alternating Tension

The stress range to which these bolts were subjected is the difference between the initial tension and the maximum operating load; since it is known that the fatigue durability is increased as the stress range is decreased, we would expect results of the order that were obtained in the chart. All failures occurred in the threads except in a few cases in which the threads were rolled in a manner to pre-stress the roots in compression. In these rolled-threaded bolts the fatigue durability of the threads was increased sufficiently to cause failure in the shanks. When the surfaces of the bolt shanks were also compression pre-stressed by peening, the failures were again transferred to the threads but, of course, at prolonged durability.

These tests, therefore, also show that the fatigue durability of cut and ground screw threads can be increased by rolling and indicate that compression pre-stressing of the surface of pure tension members is effective in increasing their fatigue strength.

It is therefore evident that the fatigue strength of bolts and studs stressed in tension is dependent upon the *initial tension* applied by the nut, *plus* the elasticity of the bolted members. Therefore, washers, lock washers, gaskets and other units that add to the elasticity of the bolted assembly are definite fatigue hazards and should be avoided whenever possible.

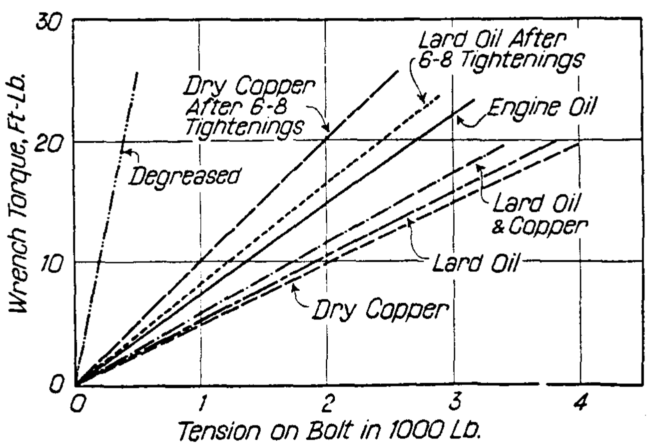
The initial tension applied by the nut is difficult to determine unless the elongation of the bolt or stud can be measured. Measurement of the torque applied by the wrench gives very unreliable information because of the variability of friction. Thus, Fig. 3 records tension measurements plotted against wrench torque in ft-lb. for $\frac{5}{16}$-in. bolts having 24 threads per in. It will be seen that the bolt tension varied as much as ten to one (1000%) for constant wrench torque depending upon the lubricant used; that is to say, a 10 ft-lb. torque puts a 200-lb. tension on a degreased bolt, and a 2000-lb. tension on the same bolt covered with lard oil. The mechanical efficiency of this bolt varied from 1 to 10%, as is shown in the chart.

Pre-loading of cyclically stressed members to reduce the stress range and thus to increase their fatigue durability is not restricted to bolts, but may be applied to many machine parts. For example, the stress range in leaf spring eyes can be reduced by pressing a tight bushing into them.

Elastic Movements in Crankshafts

A common cause of fatigue vulnerability is the belief, apparently held by many designers and engineers, that our structural materials are rigid. As a matter of fact, many fatigue failures can be traced to elastic deflection for which no

Fig. 3—Effect of Lubrication on Bolt Tension When Set-Up by a Given Torque on Wrench



allowance was made in the design. It has already been shown in Fig. 1 that elastic deformation of mating parts may be such as to concentrate the load in a small region.

Under operating conditions a crankshaft may be so elastically deformed in twisting and in bending that the bearings are only partially effective in supporting the load. The bearings are frequently found to be plastically deformed or worn "bell mouthed" to accommodate the elastic gyrations of the crankshaft! Proper attention to this detail is likely to increase the size of the shaft to a degree where its unit working stresses are surprisingly low.

Elasticity in Gear Trains

Perhaps the most generally misunderstood of all machine elements are the several classifications of gears. As ordinarily designed there is only one thing certain about gears and that is that they will not function as intended by the designer! When laying out a set of gears on the drafting board, the mating gear teeth are represented by parallel straight lines but no matter how carefully the gears are cut and heat treated the mating teeth will never again be parallel except by accident — and then only through a small load range.

The nature of the contact between two mating gear teeth is influenced *(a)* by the elastic characteristics of the housing in which they are contained, *(b)* by the elastic characteristics of bearings by which they are supported, *(c)* by the elastic characteristics of the shafts upon which they are mounted, *(d)* by the elastic characteristics of the gears themselves, *(e)* by the accumulated dimensional errors in all the supporting parts as well as the errors in the cutting of the gears, *(f)* by the necessary and accidental clearances in the supporting parts, and *(g)* by the amount and nature of the warpage in heat treating — to give the metallurgist some of the responsibility.

The result of all this is that it is virtually impossible that the parallelism between mating teeth as visioned by the designer can exist in practice. If it should chance that two mating gear teeth are parallel at some load, they cannot be parallel at any other load because the elastic deflections of some of the supporting parts are not linear with respect to the load. As usually designed, the load on gear teeth is never uniformly distributed over the length of the teeth but is always concentrated toward one end. This localization of the load is shown in Fig. 4, which is a record of the contact impressions of gear teeth under load in a commercial gear box. Load localization cannot often be seen on examining a gear that has been in service because each tooth of each gear usually makes contact with all of the teeth in the mating gear, and the summation of all contacts under all load conditions will therefore be seen by the examiner.

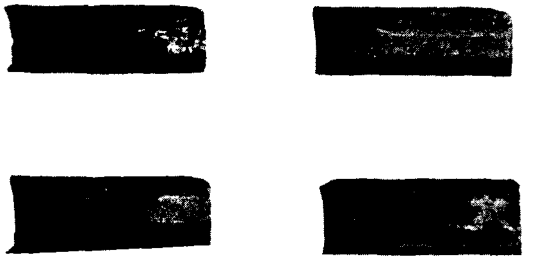
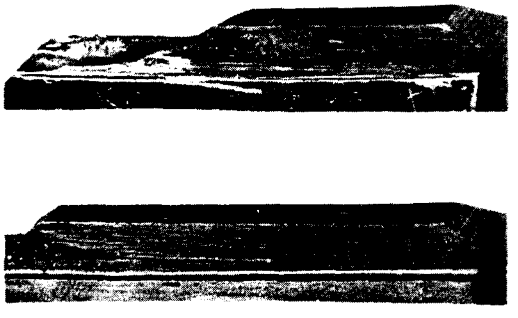


Fig. 4 — Four Teeth Taken From a Spur Gear After Operation in a Commercial Gear Box; the Polished Areas Show That Contact Was Concentrated at One End

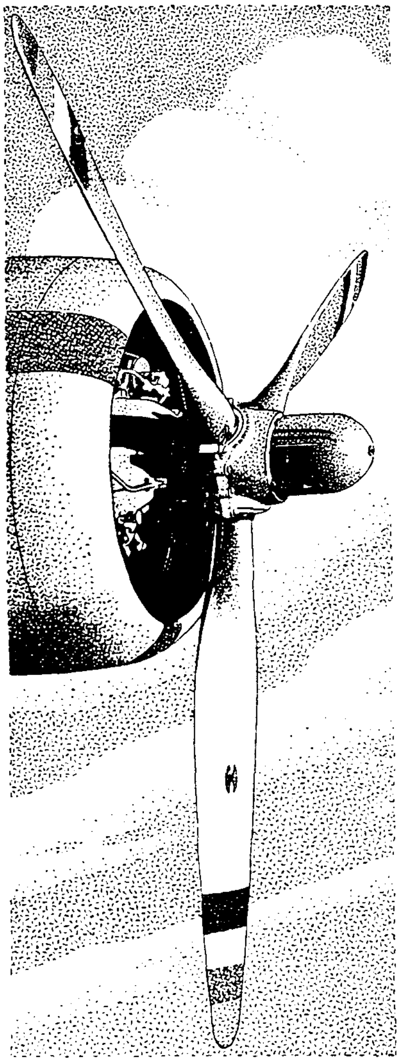
The illustration Fig. 5, however, is from a gear that failed in service. This gear was "rescued" while on its way to the metallurgical department to find what was wrong with the *material!* Note that the failed tooth is broken at one end — which, incidentally, is typical of almost all failed gear teeth. The adjoining unbroken tooth tells us that failure occurred because only a small part of the tooth was actually supporting the load, in spite of the generous length provided by the designer. This gear would have been just as durable had it been designed with one-fifth the tooth width. Clearly this was a mechanical and not a metallurgical problem; the real trouble was inadequate support of the gears and other mechanical errors.

Fig. 5 — Broken Tooth and Unbroken Partner That Prove the Trouble Was Due to Concentration of Load at Left End



It may fairly be argued that this is an unusually severe case and that it is not typical of gear fatigue. But actually the most unusual thing about it was that it could be diagnosed before it was cut into sections and the evidence etched away.

In case of fatigue failure of mating helical gear teeth of equal strength, fatigue will always occur in the tooth that is loaded on its acute angled end, because the section is weaker at this end. Mating helical gears should be offset so that contact cannot occur on the acute angled end by any mode of deflection. This is possible only where the torque is constant in direction, as pointed out by the present author in a paper in *Automotive Industries*, Sept. 25 and Oct. 9, 1937, entitled "Factors Influencing the Durability of Automobile Transmission Gears".



All gear teeth should be designed to afford a degree of tolerance for deflections, machining errors, and warpage — as has long been standard practice in spiral bevel, hypoid and in some spur and helical gears. This is accomplished by curving the elements of the tooth contour in such manner as to concentrate the load near the centerline of the gear width and thus avoid load concentration at the weaker extreme ends of the teeth.

Warpage — The statement made in the first part of this paper (page 210 of *Metal Progress* for February) that there is no practical difference, from the standpoint of fatigue, between the various alloy steels must be amended when these steels are formed into gears, because warpage during heat treatment after cutting is one of the many processing errors that results in high fatigue vulnerability of gear teeth. However, the fatigue vulnerability due to non-uniform warpage can also be reduced by design, as has been described immediately above.

Pitting of Gear Teeth

Pitting of gear teeth is a form of fatigue that is induced by compression loads on the contacting tooth surfaces. The magnitude of the compression stresses varies with relative curvature of the contacting teeth in accordance with the Hertz formula; it also varies with the degree of load concentration at the ends of the teeth and with the applied load. The load that may be carried varies with the hardness and therefore with the strength of the material, with the temperature, and with the manner in which the lubricant is applied.

The design factors that are effective in reducing the load concentration at the ends of the teeth also reduce the compressive stress. The relative curvature (and therefore the compressive stress) can be varied by the choice of pressure angle. In general there is little to be gained by designing wide face gears except the doubtful satisfaction of dealing with smaller stress numbers.

In high speed gears, pitting may occur when gears are transmitting no load. This is sometimes seen in the reverse idler gear of automobile transmissions. Although this form of transmission trouble is rare and occurs only when other conditions, such as hardness, are unfavorable, it serves to emphasize the part played by the lubricant in promoting fatigue. A reverse idler running submerged in oil will trap the oil between the gear teeth and if the clearances are small will induce extremely high surface pressures. We are all familiar with the high temperatures that are generated in gear boxes when too generously supplied with oil, but we do not always interpret this as a fatigue hazard.

High speed gears should be lubricated by jets of low viscosity oil directed at the teeth as they are *coming out* of mesh, not on the incoming side. Proper lubrication will wash away the heat of friction while it is still at the surface and will prevent excess oil from reaching the contacting teeth, providing the sump is dry.

There is evidence indicating that oil further contributes to pitting fatigue by entering surface fissures where, under hydrostatic pressure, the fissures are extended until pieces are lifted out of the surfaces of the teeth. This is described by Stewart Way in an article on "Gear Tooth Pitting" in *Electric Journal*, 1936, p. 175.