SHOT BLASTING TO INCREASE FATIGUE RESISTANCE

by J. O. ALMEN
Research Laboratories Division,
General Motors Corp.

SHOT BLASTING TO INCREASE

WHILE great strides have been made in most phases of engineering and metallurgy, it is doubtful that in dynamically loaded parts we are getting more net work from our metals today than was obtainable 25 years ago. The fact that modern airplane engines weigh only about one-half as much per horsepower as the engines of World War I is primarily due to improvements in fuels and increases in engine speed. The speed and performance of airplanes have increased because of the better power-weight ratio of engines and aerodynamic improvements in propellers and airplane structures. New fabrication techniques have made possible many design improvements, better bearing materials are available, lubricants have been improved; but the basic useful strength of our structural materials remains unaltered.

Although no super-strength alloys have been discovered and no such discoveries seem to be imminent, there is much that can be done to increase materially the fatigue strength of many machine parts made from our ordinary structural materials. This fatigue strengthening does not require changes in design or in material, and in fact it does not require processes that are fundamentally new or untried. It is merely the extension of processes that, on the whole, have long and honorable histories, and the avoidance of processes and practices that are now known to reduce fatigue strength. The significance of these processes has only recently become clear through the introduction of new concepts of fatigue phenomena by which new avenues of reasoning are opened to us. These new concepts are: *Fatigue failures result only from tension stresses, never from compressive stresses and any surface, no matter how smoothly finished, is a stress-raiser.*

■ Fatigue Vulnerability

The surfaces of repeatedly stressed specimens, no matter how perfectly they are finished, are much more vulnerable to fatigue than the deeper layers. It has long been appreciated that the vulnerability to fatigue increases as the surface roughness is increased, particularly if the roughness consists of sharp notches, and more particularly if the notches are oriented at right angles to the principal stress.

The practice of carefully finishing fatigue test specimens and engine parts is, of course, a recognition of this vulnerability in so far as visible marks or scratches are concerned, even down to being sure that the final polishing marks are parallel to the direction of the applied stress. These precautions are known to be effective in increasing the fatigue strength of the specimens, and specimens finished in this manner have, therefore, come to be known as "par" bars. This name implies that fatigue specimens and machine parts approaching perfection in finish give the highest possible fatigue endurance for any particular material, and that they accurately measure the ultimate fatigue properties of that material.

[This paper was presented at the SAE War Materiel Meeting, Detroit, Mich., June 9, 1943, and the National Aeronautic Meeting of the SAE, New York City, April 8, 1943.]

IT is doubtful whether we are getting more net work from metals today in dynamically loaded parts than was obtainable 25 years ago, and no super-strength-alloy discoveries seem imminent; however, much can be done to increase the fatigue strength of many machine parts made from ordinary structural materials by merely extending processes already known to be satisfactory, and avoiding practices that reduce fatigue strength.

We have today new concepts of fatigue failure: Fatigue failures result only from tension stresses, never from compressive stresses. Any surface, no matter how smoothly finished, is a stress-raiser.

Structural materials are not rigid. Many fatigue failures can be traced to elastic deflection for which no allowance was made in design.

From experience with practical machine parts, we can only conclude that stress calculations by textbook methods are wholly inadequate unless we generously temper our calculations with experience. The accuracy of stress data from photoelasticity, brittle lacquers, extensometers, and similar methods is usually greater than by mathematical analysis, but these are far from reliable.

As a working hypothesis, it seems reasonable to assume, except possibly for very ductile metals, that:

The slope of the fatigue curve, as measured on a log-log plot, is a measure of effective stress; and fatigue curves for varying stress concentrations converge toward a point near the tensile strength of the material at some considerable number of stress cycles.

Fully 90% of all fatigue failures occurring in service or during laboratory and road tests are traceable to design and production defects, and only the remaining 10% are primarily the responsibility of the metallurgist as defects in material, material specifications or heat-treatment.

Study of fatigue of materials is the joint duty of the metallurgical, engineering, and production departments. There is no definite line between mechanical and metallurgical factors that contributes to fatigue. This overlapping of responsibility is not sufficiently understood. Until more time is devoted to searching for mechanical causes rather than metallurgical ones, we cannot make full use of our materials.

THE AUTHOR: J. O. ALMEN has long been active in General Motors research development work. He is head of the Mechanical Engineering Department No. 1 of the GM Research Laboratories.

It can be shown, however, that the so-called "par" bars are not the best specimens, but that influences akin to

FATIGUE RESISTANCE

by J. O. ALMEN

Research Laboratories Division, General Motors Corp.

notches, so far as fatigue vulnerability is concerned, are retained by the "par" specimens. It seems that the specimen surface is highly vulnerable simply because it is a surface; that there is an extra hazard in the surface layer not shared by the deeper layers. This extra surface hazard may be due to submicroscopic notch effects, or to the fact that the surface is a discontinuity, since the outer crystals are not supported on their outer faces. Whatever the reason for surface vulnerability, the evidence of its existence is strong.

Fatigue Life Increased

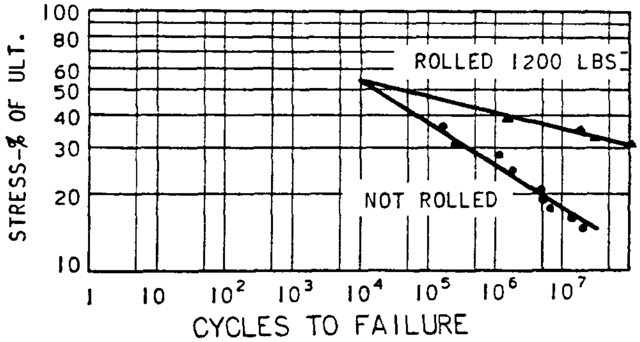
The fatigue strength of the most carefully prepared specimen will be increased if a thin layer of the specimen is pre-stressed in compression[1] by a peening operation such as peen hammering, swaging, shot blasting or tumbling, or by pressure operations by balls or rollers. This increase in fatigue strength resulting from the surface layer being stressed in compression is clearly shown by the S-N curves, Fig. 1, which compare normally finished railway axles with axles that had been subjected to a rolling operation[2]. This and many other tests show that the compressive stressed surface is effective in increasing the fatigue strength whether applied to highly finished specimens or to specimens having rough surfaces.

We are all familiar with the improvement in fatigue that may be obtained by a few cycles of overload sufficient to produce a "set" in such parts as springs. Local tension stresses from the overloads exceed the elastic limit of the material and, therefore, the tension stress at the working load is decreased. This treatment, which has long been practiced on many production items, is similar in effect to rolling or peening since, in the unloaded state, the member is stressed in compression in the areas where yield occurred during the overloading.

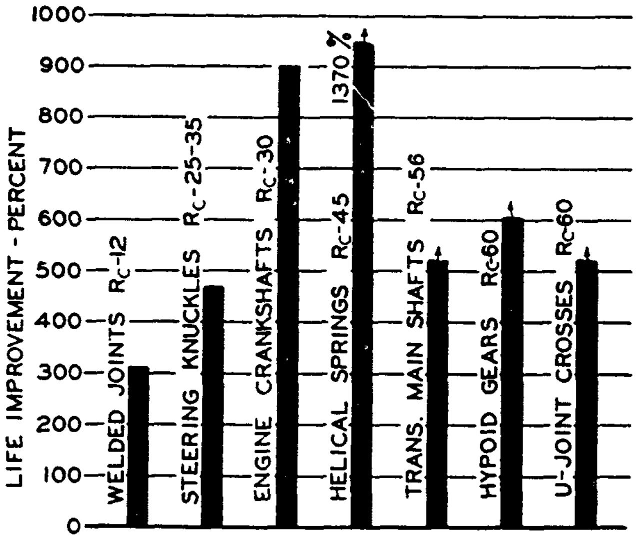
The bar chart, Fig. 2, records the increased fatigue durability resulting from shot peening of a few typical machine parts. It will be seen that the fatigue durability is increased whether the parts are hard, such as carburized gears, or soft, such as steering-gear parts, and whether the stress is completely reversed, as in crankshafts, or the stress range is small, as in preloaded springs.

Note that the fatigue durability of peened axle shafts was not increased as much as most of the other specimens. The work on these shafts was conducted a number of years ago before the technique of peening machine parts had been developed, and the relatively small increase was probably due to insufficient peening. Similar fatigue results have been obtained from a large variety of machine parts and from aluminum specimens, and there are reasons to expect that the treatment is equally effective for all metals.

The bar chart, Fig. 2, shows the fatigue durability increase as per cent gain above the durability of the same machine part before peening. Actually, durability com-



■ Fig. 1 – S-N diagram for railway axles showing change of slope due to rolling



■ Fig. 2 – Bar chart showing the benefits of surface-peening based on fatigue life

parisons cannot be made on a percentage basis alone, as is apparent when we examine the improvement in fatigue due to rolling, as shown in Fig. 1. If, in this chart, the durability comparison is made at a load equal to 55% of the ultimate strength, the percentage improvement is zero; if the durability comparison is made at a load corresponding to 20% of the ultimate strength, the percentage improvement is infinite, and, at intermediate loads, the percentage gain will, of course, be somewhere between these limits.

It is essential that this be kept in mind when interpreting the fatigue data. To illustrate, suppose that the average fatigue durability of a gear tested at high load is increased from 30,000 cycles to 70,000 cycles by suitable surface-peening, a gain of 130%. Now if this comparison is made

---

[1] See *Stahl und Eisen*, Vol. 49, April 25, 1929, pp. 575-577: "Das Drücken der Oberfläche von Bauteilen aus Stahl," by O. Föppl.

[2] See ASME Transactions, Vol. 58, September, 1936, pp. A-91-A-98: "Increasing the Fatigue Strength of Press-Fitted Axle Assemblies by Surface Rolling," by O. J. Horger and J. L. Maulbetsch.

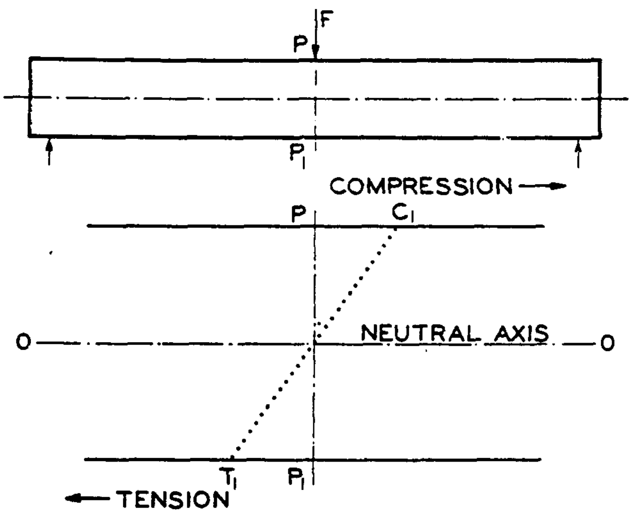
at a lower load such as to cause initial failure at 300,000 cycles, the treated gear might run 6,000,000 cycles before failure, a gain of possibly 2000%. This is because of the difference in slope of the fatigue curve representing the normal gears and the fatigue curve representing the peened gears.

Explanation of Peening Effectiveness

The most plausible explanation of the effectiveness of surface compression stress[3] is that when a load is applied to such specimens the tension stress in the surface layer becomes less by the amount of the compression pre-stress, and since fatigue failure starts from tension stress the fatigue durability of the weak surface layer is increased. However, the tension stress in the material below the pre-stressed layer is not reduced but may be actually increased, notwithstanding which, the fatigue strength of the specimen is increased. It follows, therefore, that the lower layer is inherently stronger than the surface layer.

Föppl[4] shows that the fracture in rolled specimens does not originate at the surface but in the material below the pre-stressed layer, as would be expected if the surface is sufficiently pre-stressed in compression. Similar subsurface fatigue failures, usually called fissures and attributed to faulty material, have long been known to occur in railroad rail, in which the surface is stressed in compression as a result of the cold work of heavily loaded locomotive and car wheels.

The situation can perhaps be clarified by the use of the conventional textbook stress diagram of a loaded beam, as illustrated in Fig. 3, in which a beam supported at the



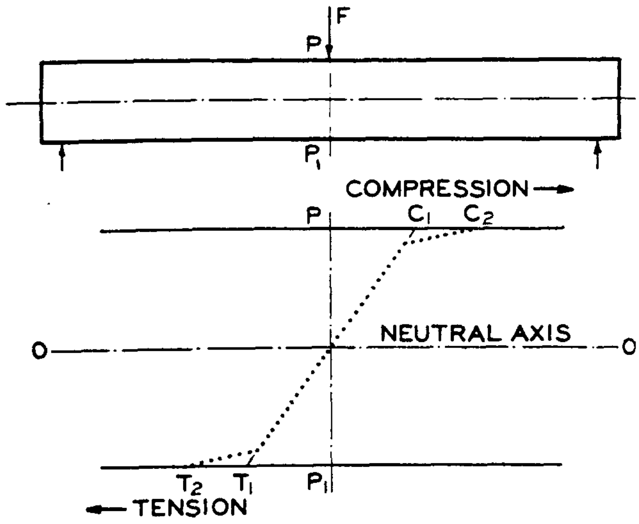
■ Fig. 3 – Conventional stress diagram of a loaded beam

ends is loaded in the central plane, P-P_1. The stress at any point in the beam is measured by the horizontal distance from the plane P, in which the load is applied, to the diagonal line T_1-C_1. The distance P-C_1 represents the compressive stress at the upper surface, the stress at the neutral axis O-O is zero and the tension at the lower surface is represented by the distance T_1-P_1.

[3] See *Stahl und Eisen*, Vol. 53, Dec. 21, 1933, pp. 1330-1332: "Die Wirkung von Eigenspannungen auf die Biegeschwingungsfestigkeit," by Hans Bühler and Herbert Buchholtz.

[4] See *Iron Age*, Vol. 126, Sept. 18, 1930, pp. 775-777, 829: "Cold Rolling Raises Fatigue or Endurance Limit," by G. S. von Heydekampf.

While this is a satisfactory enough stress diagram for static loads, it does not agree with the behavior of fatigue specimens. However, if we modify the diagram Fig. 3 as is shown in Fig. 4, in which T_1-T_2 represent an added

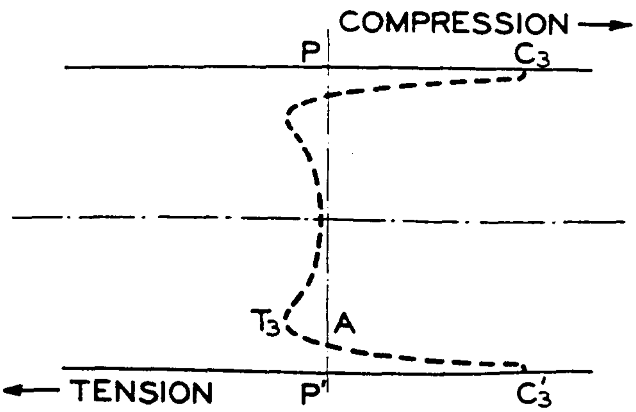


■ Fig. 4 – Modified stress diagram of a loaded beam showing surface vulnerability

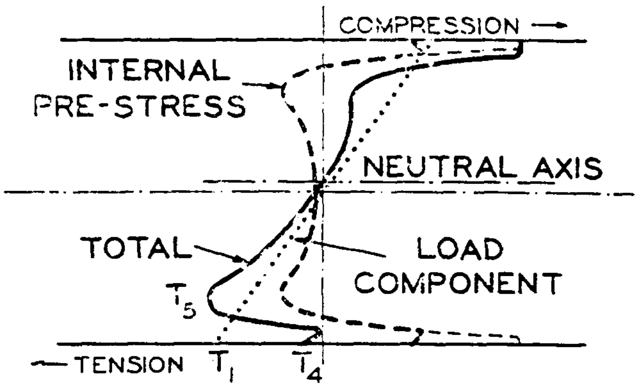
increment of tension stress in the surface, we have a reasonable representation of the surface fatigue vulnerability. For a sharply notched surface, the additional stress increment T_1-T_2 is relatively great. As the surface roughness is decreased, the increment T_1-T_2 decreases, but no matter how well polished the specimen may be, there still remains an additional surface stress as measured by fatigue tests.

Stress Patterns

Fig. 5 represents the residual stress pattern in an unloaded beam that has been rolled or peened, as has been described, in which C_3-P and C'_3-P' represent the magnitude of compressive pre-stresses, and T_3-A represents the magnitude of the tension pre-stress to balance the compressed stresses in the surfaces. After this beam has been loaded from either side through one stress cycle, as in a reversed fatigue test, the compression pre-stress will be reduced if the applied load raises the total compression



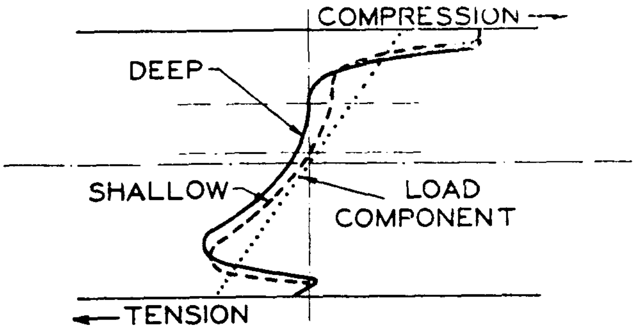
■ Fig. 5 – Probable stress diagram of an unloaded beam with pre-stressed surfaces stress above the yield point. The stress diagram for such a pre-stressed beam supporting an external load is shown in Fig. 6, in which the effective tension stress, T_4, at the surface may be less than the stress T_5, below the surface,



■ Fig. 6 – Stress diagram for a loaded, pre-stressed beam

in which case failure would start below the surface, as noted by Föppl. Note also that the neutral axis is displaced from the geometric center of the beam and that the tension stress, T_5, below the surface is greater than in the beam that had not been pre-stressed, as is shown by the dotted lines.

The magnitude of the subsurface tension stress in a loaded beam having pre-stressed surfaces will vary with the amount of compression pre-stress and with the depth of the pre-stressed layer. Fig. 7 shows that the subsurface



■ Fig. 7 – Stress diagram for a loaded, pre-stressed beam showing the influence of the depth of the pre-stressed layer

tension stress may be greater for a deeply pre-stressed layer than for a layer of lesser depth.

It seems evident that the improvement in fatigue strength by compressive pre-stress is due to the reduction in tension stress when loaded in the vulnerable surface layer and that the increased compressive stress in a specimen stressed from zero to a maximum in either direction does no harm, probably because of adjustment of compression stress in the pre-stressed layer through yield.

Further evidence of the extra vulnerability of the surface layer is found in the behavior of specimens having increased strength in a thin surface layer, as in thinly carburized or cyanided specimens or in thinly nitrided specimens. Fatigue failures in such specimens also start below the surface and show greater fatigue strength than the same material in the unclad state. A nitrided specimen is probably superior to the other forms of hard cladding because, in addition to the higher physical properties of the surface layer, this layer is in a state of compression, and it is, therefore, less notch sensitive.

■ Residual Thermal Stresses

While on the subject of beneficial internal stresses, mention should be made of the surface compressive stress obtainable by heat-treatment. By a rapid quench, it is possible, through thermal contraction alone, to trap compressive stress in the surface and corresponding tension stress in the core, but this method, although showing some benefit in fatigue, is not as effective as the other methods that have been discussed. This subject will be discussed later in this paper.

Perhaps the most spectacular use of surface compression stress by heat-treatment through thermal contraction alone is tempered glass which, because of its great strength, is used in some parts of modern automobiles. This glass is prepared from normal glass by rapidly cooling the surfaces by means of air jets. The cooled surfaces contract causing the relatively plastic center to yield in compression. As the center of the glass cools and contracts it becomes stressed in tension, with consequent compressive stress in the surfaces.

■ First Use of Surface Compression

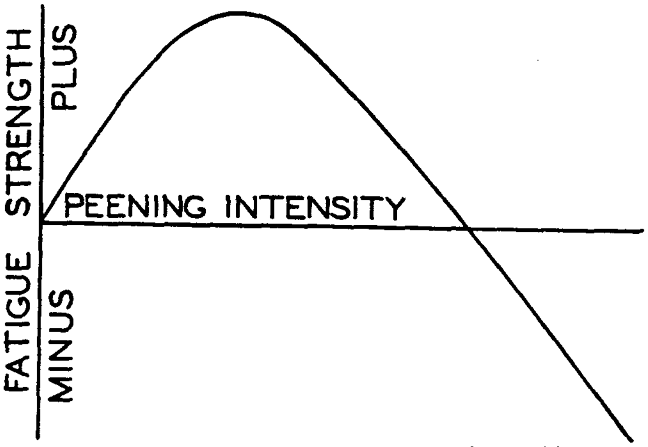
The idea of surface compression to improve the strength of steel is probably as old as steel itself. It has probably been discovered, forgotten and rediscovered many times. Certainly every village blacksmith knew and practiced the art in making wagon and buggy springs, axles, and other heavily loaded parts. After these parts were forged into shape they were severely hammered to improve their strength and, no doubt, the same procedure was followed by the ancient sword makers. Likewise, mill and ship shafts were cold-worked by the application of small rollers at high pressure after machining because of the greater strength that was known to result.

Cold-working of metals increases the hardness of most metals, including steel, at least in the range of low hardness, it usually results in internal stresses of varying degrees and patterns, it alters the physical properties and sometimes fractures the material. With the known sensitivity of materials to fatigue, it is obvious that we must learn how to control cold-work just as we have had to learn how to control heat-treatment in order that we may benefit by the good effects and overcome the evil effects. We would not think of specifying a heat-treatment without stating whether the temperature should be raised or lowered and in which order and to what extent; yet that is the way we now think of cold-work. Cold-working can be good or bad depending upon how it is done and for what purpose.

■ Pre-Stressing May Be Overdone

Papers have been published showing that cold-working of the surface so as to produce a layer stressed in compression increases the fatigue strength of the parts to which it is applied, but we are not told the amount of the pre-stress or the depth of the pre-stressed layer. Both of these values are presumably important in obtaining optimum results for any particular specimen, but it is probable that the values should not be the same for all sizes of specimens, for all materials, or for hard and for soft specimens.

Several instances are known in which the strength of machine parts and specimens has been decreased by too intense surface peening. Fig. 8 is presented as showing the probable effect of peening or rolling, particularly on thin sections. The fatigue strength is increased as the intensity of peening or rolling is increased until a maximum improvement is obtained. With more intense peening or rolling, the fatigue strength rapidly decreases below the original strength and the part will be damaged due to excessive internal tension stresses.



■ Fig. 8 – Effect of peening intensity on fatigue life

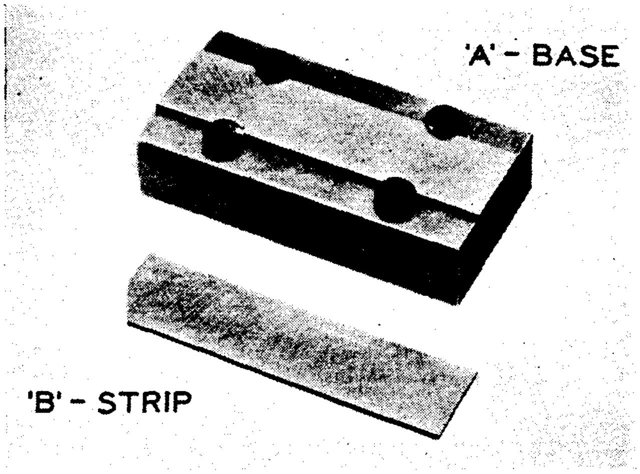
It would, therefore, seem important to control the compression stressed layer as to stress magnitude and depth with considerable accuracy by proper selection of the curvature of the rolling or peening instruments and by the pressure that is applied. The precise amount of surface compressive stress that is required for optimum fatigue strength is known for a few specimens only. It will vary with the shape and section thickness of the machine part, with the hardness, and with the kind of metal being treated. For the present, we must frequently rely upon the not too accurate sense of proportion that is developed by experience to indicate the treatment that should be applied to any given machine part.

When the layer is stressed in compression (by applying sufficient pressure on the work by rollers or by peening) to a degree exceeding the yield strength of the metal in compression, the amount of residual stress is presumably at least equal to this yield strength.

The depth of the stressed layer is probably roughly proportional to the instantaneous area over which the pressure is applied, and to the pressure intensity. The depth of the compression stressed layer in a railroad rail[5] should be greater than the depth of the compression stressed layer in the same material if small rollers at the same pressure intensity were used instead of large car wheels. Under these circumstances, the initial point of fracture should appear at corresponding depths. Such evidence as is available indicates this to be true.

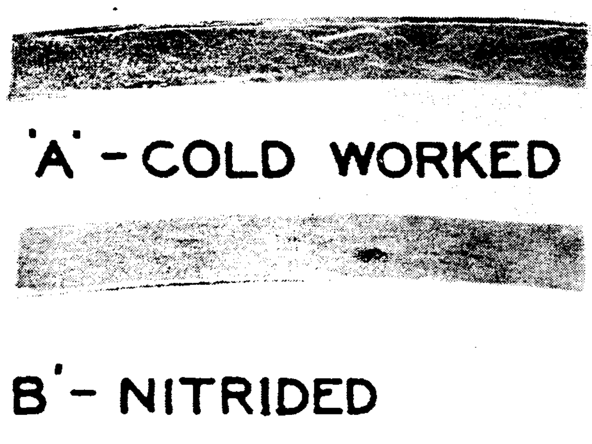
■ Instrument Measures Peening

A simple and practical method for indicating the compression in the stressed layer consists of a thin flat strip, Fig. 9B, that is attached to a heavy base as shown in Fig. 9A This strip is rolled or peened with the same intensity that is given to the machine part, and when it

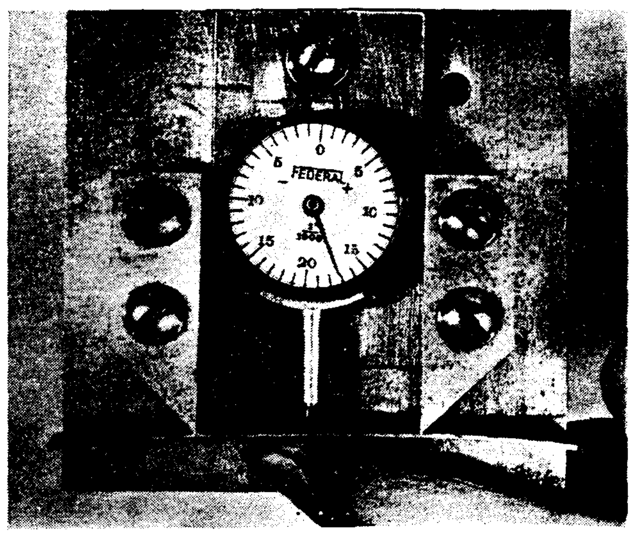


■ Fig. 9 – Apparatus for indicating the compression in the stressed layer

is removed from the base it will be found to be curved, as shown in Fig. 10A, with the convex surface on the cold-worked side. The curvature of the strip may be measured by an indicator, as shown in Fig. 11, which can then be interpreted in terms of the depth of the stressed layer.



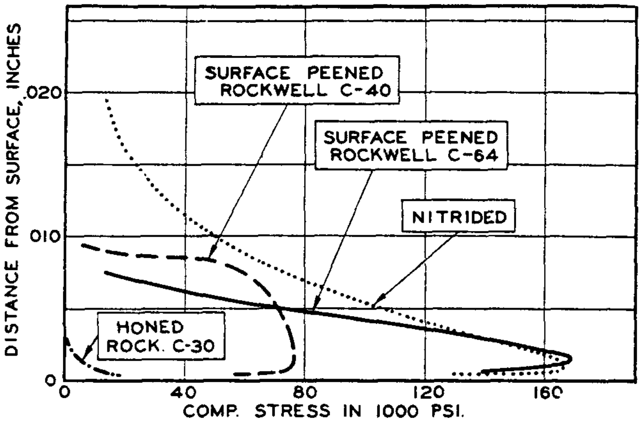
■ Fig. 10 – Measuring strips



■ Fig. 11 – Curvature indicator

[5]See *Journal of the Iron and Steel Institute*, Vol. 66, No. II, 1927, pp. 265-282: "The Work-Hardening of Steel by Abrasion," by E. G. Herbert.

The chart shown in Fig. 12 records the stress magnitude and the depth of the stressed layer at constant cold-work intensity of two such test strips. The cold-worked surfaces of these strips, the Rockwell C hardness being respectively 64 and 40, were honed away in small increments and the curvature was measured with the removal of each thin layer. The changing curvature as metal was removed provided data from which the compressive stress in each laver could be calculated, with the results shown in the chart. As would be expected, because of the higher yield point, the harder specimen was found to be more highly stressed than the softer specimen.



■ Fig. 12 – Magnitude and depth of stress imposed by various surface treatments

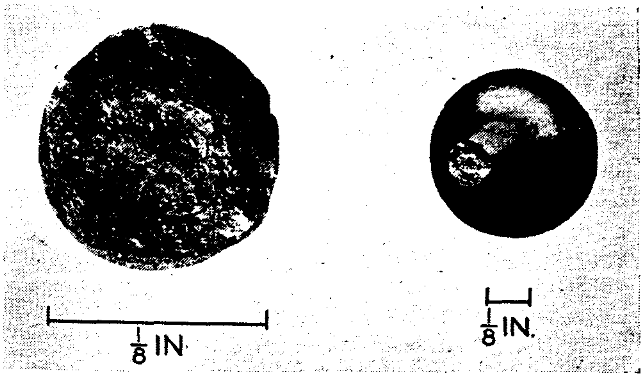
Also shown in this chart is the surface compressive stress in a nitrided specimen as a result of the nitriding. The procedure for this experiment was the same as for measuring the stress due to peening except that the face of the specimen that was in contact with the heavy base was plated to limit the nitriding to the outer face of the strip. On removal from the base after nitriding, the strip was curved convex on the nitrided side, as is shown in Fig. 10B. It seems, therefore, that the well-known resistance of nitrided specimens to fatigue is primarily due to the compressively stressed surface layer.

■ Overdose of Nitriding

Although the usual experience with nitriding is that it greatly improves fatigue strength, it is possible to overdo nitriding just as it is possible to overdo surface stressing by peening and rolling. The high compressive surface stress that results from nitriding must, of course, be balanced by internal tension stress of equal total value. When deep nitriding is applied to light sections, the unit internal tension stress may reach dangerous proportions.

Fig. 13 shows a part that was greatly reduced in strength as a result of nitriding, its fatigue durability being only 1 or 2% as great as the fatigue durability of the same part not nitrided. The diameter of the part at the point of failure was approximately ⅛ in. The depth of the nitrided layer was about 0.020 in., the area of which is equal to about 60% of the area of the section, as is shown by the circle in the enlarged view. From the nitriding, compressive-stress diagram shown in Fig. 12, it is evident that the internal tension stress must have been very great.

It is also known that internally nitrided cylinder barrels are more prone to fail by cracking than cylinder barrels that are not nitrided, the reason being that the stress due



■ Fig. 13 – The effect of deep nitriding – failure from severe tension stress

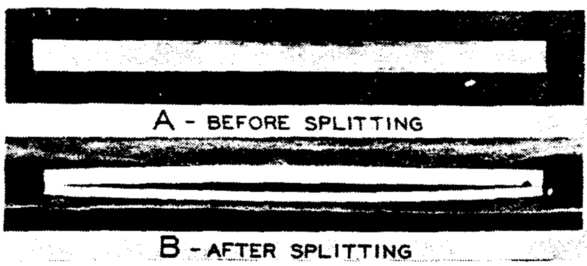
to nitriding is added to the stress from gas pressure. Care must, therefore, be used in nitriding thin sections to gage the depth of the nitrided layer in proportion to the thickness of the section being nitrided.

■ Residual Stress from Honing

While the peened specimens used for the experiment shown in Fig. 12 were being honed as has been described, it was found that the strips did not fully recover their original flat form. To determine if this residual curvature was due to a "set" in the material or was the result of honing, other flat strips that had not been peened were honed. These strips developed the same curvature as the residual curvature in the peened specimens, demonstrating that honing produces a compressively stressed layer. The approximate magnitude of this honing stress is also shown in the chart given in Fig. 12. This raises a question as to the state of surface stress in the carefully prepared fatigue specimens favored for laboratory fatigue tests, since additional tests have shown that lapping also introduces surface compressive stress.

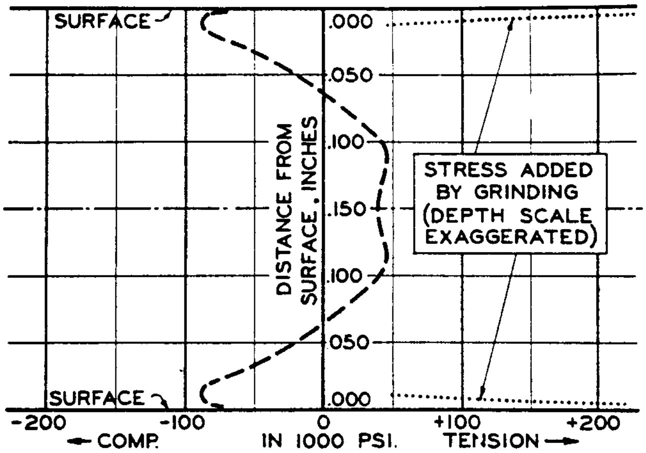
■ Residual Stress from Carburizing

The carburized layer in a carburized part is stressed in compression, as is graphically shown in Fig. 14. Two



■ Fig. 14 – Residual compressive stress resulting from carburizing and hardening the upper and lower faces of a specimen

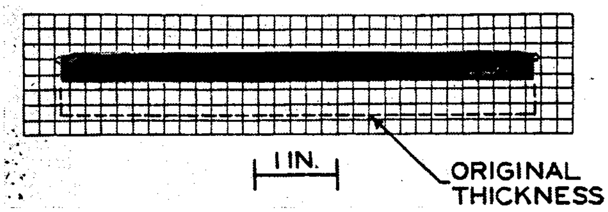
opposite faces of this ½-in. square specimen were carburized, while the other two faces were protected by copper plating. The specimen was quenched and tempered in the usual manner, after which it was split with a saw as shown in Fig. 14B. Note that the parts are curved convex on the outer faces, indicating compressive stress in these faces. Analysis of the internal stresses in another carburized member by a method similar to that described for peened and nitrided strips indicated the internal stress pattern shown in Fig. 15. Of interest here is the magnitude of the compressive stress in the carburized layer and the reduced compressive stress, possibly even tension stress, in a thin surface layer. When carburized parts, such as bearing races, wristpins, and gear teeth, are ground we may expect the surface to be stressed in tension, as is indicated by the dotted lines shown at the right in Fig. 15.



■ Fig. 15 – Magnitude and depth of residual stress due to carburizing and hardening

The compressive stress in the carburized layer may be a hazard for members stressed in tension, as was shown for the nitrided part, Fig. 13, because the tension stress in the core is equal to the working load plus the tension load due to the compressive pre-load of the case. For members stressed in bending and in torsion, however, the internal compressive stress in the carburized case of ordinary depth improves the fatigue strength of the part except for the thin surface layer which, especially after grinding, is severely stressed in tension. It is, however, a simple matter to convert this thin tension stressed layer into stress in compression by suitable peening or rolling operations, as was indicated in Fig. 12, with resultant large gains in bending and torsion fatigue strength.

The residual stress in crankshafts and other parts hardened by induction heating and probably also in flame-hardened parts resembles the residual stress in carburized and hardened parts, as shown in Fig. 16. The hardened



■ Fig. 16 – The effect of induction hardening – the convex curve of the upper surface indicates compressive stress

upper surface of this specimen was straight at the original thickness. Note that after removal of the material indicated by the dotted line, the upper surface is curved convex, indicating compressive stress. More complete analysis indicated that a thin surface (upper) layer was possibly stressed in tension. In these treatments as in carburizing, the hardened layer is stressed in compression because in undergoing the phase change to the hard state, the density of the steel is reduced and therefore the hardened layer seeks to occupy more space. A thin surface layer however may be stressed in tension.

With internal stresses of the magnitude shown in Fig. 15, we can readily understand why carburized parts are prone to warp during heat-treatment, especially if the design is not symmetrical with respect to the internal stresses.

■ Low-Temperature Quenching Stresses

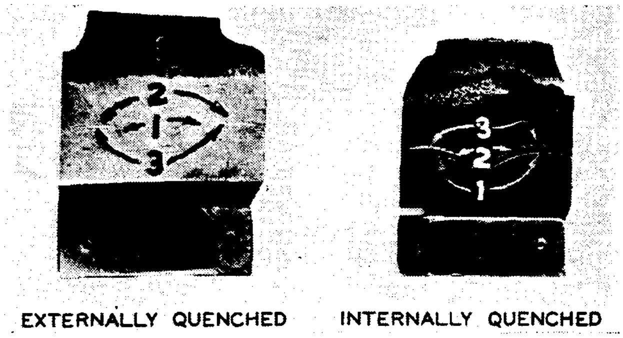
Residual stresses due to quenching from relatively low temperatures may reach considerable magnitudes and may be harmful or helpful to fatigue durability depending upon whether the trapped stresses augment or diminish the tension stresses from the applied loads. An interesting case of this kind occurred in a water-cooled aluminum cylinder head, as shown in Fig. 17, that failed by fatigue

■ Fig. 17 – Method of indicating quenching stresses in a water-cooled aluminum cylinder head – fatigue failure occurred on the water side of the combustion-chamber wall

on the water side of the combustion-chamber wall. Measurements of residual stress disclosed that the water side of the combustion-chamber wall was stressed in tension and the combustion-chamber side of the wall was stressed in compression. This internal stress pattern was one of the same kind as the stress from the gas pressure against the combustion-chamber wall, and the resultant stress was, therefore, the sum total of the residual stress and the gas pressure stress.

The residual stresses in this case were caused by quenching the cylinder heads from 980 F by immersion in cold water. The outer surfaces of the casting were cooled while the inner water-jacket surfaces, especially at the thick section, were still hot. Thermal contraction of the outer surfaces imposed compressive stresses of such magnitude as to cause yield in the still hot and therefore weaker water-jacket surface. As cooling progressed, the metal that had been stressed beyond the yield point contracted thermally, leaving tension stress on the water-jacket side and the corresponding compression stress on the combustion-chamber side. The retention of the residual stress in the thick combustion-chamber wall was aided by the thinner outer wall of the water jacket, which was stressed in compression.

A visual indication of the residual stress pattern is shown by the scribed arcs at the left in Fig. 18. These arcs were drawn from centers in a steel bar bolted to the opposite side of the casting section, as is shown in Fig. 17. The arcs indicated by the numeral 1 were drawn when the cylinder head was intact. The casting was then sectioned as shown in Fig. 17 except that the outer side of the water jacket had not been cut and the arcs indicated by the numeral 2, of the same radius as before, were drawn. Finally the outer side of the water jacket was removed and the arcs indicated by the numeral 3, still of the same radius, were drawn. Note the direction of movement of



■ Fig. 18 – Variation in internal stresses from different quenching methods

the metal with each operation and the order of the stress that is indicated. As a correction of the undesirable residual stress shown by this test, another cylinder head was given the same heat-treatment except that it was quenched internally by forcing cold water through the water jacket. The order of internal stresses was measured in the same manner as has been described, with the results shown at the right in Fig. 18. Note that when internally quenched, the stress pattern in the combustion-chamber wall is reversed, leaving the inner side in compression and the outer side in tension. Since these trapped stresses are of opposite sign to the operating stress, the resultant stress is the difference between the residual and operating stresses instead of their sum as when the casting was quenched externally.

■ Fatigue Life Increased

Fatigue tests were conducted on cylinder heads quenched by both methods, the results of which were 2,000,000 to 3,000,000 stress cycles to failure for the externally quenched heads, and 5,000,000 to 6,000,000 stress cycles for the internally quenched heads.

Additional fatigue tests were made on internally quenched heads in which the aging treatment at 350 F was omitted in order to avoid reduction of the favorable stress pattern. These heads endured more than 14,000,000 stress cycles at the same test load without failure.

Similar residual stresses are known to occur in many other heat-treated and quenched aluminum parts. It is also known that many aluminum parts show better fatigue resistance when they are drawn to a higher temperature than that which gives the greatest tensile strength, presumably because unfavorable residual stresses resulting from quenching are thereby reduced. It is probable that similar stresses can be trapped in steel by quenching from tempering temperatures[6]. Such residual stresses may be favorable or unfavorable depending on the shape of the part, the temperature gradient, and the direction of heat flow.

■ Corrosion Promotes Fatigue

Fatigue failures in many machine parts are traceable to corrosion of several kinds or to other forms of surface damage that occur in service. In normal machine parts, even slight corrosion or bruising is very potent in encouraging fatigue fractures because each pit interrupts the continuity of the surface and increases the local stress. The damaging effect of corrosion or bruising is prevented on the surfaces that are adequately protected by compressive pre-stress because the local tension stress cannot reach dangerous values until the pits or bruises have progressed sufficiently to penetrate the compressively stressed layer. This was forcefully demonstrated in fatigue tests of a machine part that failed alternately in a badly formed fillet or in the region of a clamp remote from the fillet where fretting corrosion occurred. The durability of the part could not be increased by improving the fillet because this would merely transfer all failures to the fretted area at about the same durability. After peening, however, the fatigue durability was found to have increased several hundred per cent and large additional gains were then possible by improving the form of the fillet without failure in the corroded area. The peening did not prevent corrosion but it did prevent the ill effect of corrosion in promoting fatigue.

Similar protection against the effects of corrosion and of surface bruises is afforded by nitriding[7], carburizing, and other treatments that produce compressively stressed surfaces. The working face of a gear tooth may be severely pitted, creating a fatigue hazard, but the bending fatigue strength may not be impaired because the carburized layer is compressively stressed and the surface is compressively stressed by the cold-work of mating teeth.

■ Surface Finishes

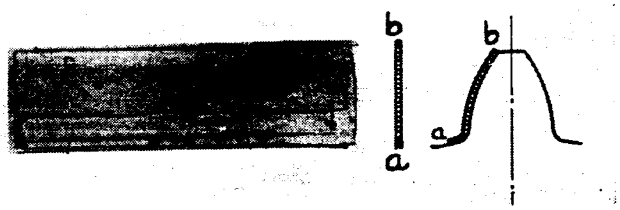
Efforts to improve products by improving surface finish may sometimes have the opposite effect. Highly finished surfaces and fillets may lead to a false sense of security if, as the result of machining or straightening operations, the parts have high internal stresses of the wrong kind.

When machine polishing is done by the use of abrasive paper, cloth wheels, or abrasive-covered felt wheels, sufficient heat is often generated to induce serious surface-tension stresses and thus promote instead of prevent fatigue failures.

In ground surfaces such as shafts, wristpins, and gear teeth, the grinding operations may introduce high surface tension stresses that, from the standpoint of fatigue strength, often do more harm than good. The surface-tension stresses from grinding are often so great as to produce visible or magnaflux surface cracks, but whether detectable or not, surface tension is frequently very serious.

Fig. 19 is a magnaflux transfer print on transparent cellulose tape showing surface fractures in a ground gear tooth. This tooth failed by spalling originating in these surface fractures. Since fatigue cracks start on the side of the gear tooth that is loaded in tension, the effective stress is the grinding pre-stress plus the working stress. Fre-

[6] See *Engineering*, Vol. 154, Aug. 14, 1942, pp. 134-135: "Quenching of Steel after Tempering and the Impact Test," by L. E. Benson.
[7] See *Metals and Alloys*, Vol. 5, June, 1934, pp. 129-130: "Effect of Notches on Nitrided Steel," by J. B. Johnson and T. T. Oberg.

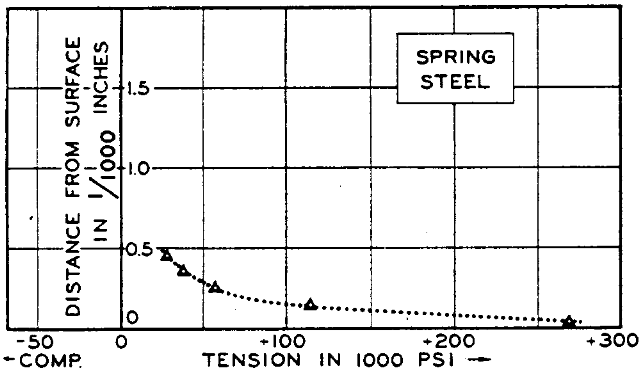


■ Fig. 19 – Surface fractures in a gear tooth – the result of grinding

quently, we find that a hardened part will show a file-soft skin after grinding, which not only promotes fatigue but is also susceptible to seizure and galling.

Internal stresses of the wrong kind are perhaps the most insidious of all fatigue hazards because we can seldom know their magnitude or the pattern in which they are distributed within the material or whether they are alike for all commercially identical machine parts. Internal stresses may be the result of operating conditions such as occur in brake drums, clutch plates, or other friction surfaces where the instantaneous temperature in a thin layer is so great that, under thermal expansion, the surface layer is stressed beyond the yield point in compression. When the source of heat is removed, the heated surface layer is quenched by the adjacent cool metal and, under thermal contraction, it is so severely stressed in tension that fractures often occur. This is, of course, the same thing that happens in machine polishing and in grinding unless great care is used.

The magnitude of surface tension stress in a specimen that was ground in accordance with normal commercial practice is shown in Fig. 20. A specimen of annealed



■ Fig. 20 – Magnitude and depth of the residual stress caused by grinding a spring-steel specimen

spring stock, 1/16 in. in thickness, 1 in. wide, and 7 in. long was ground to a depth of 0.002 in. After grinding, the previously straight specimen was found to be curved concave on the ground side, indicating tension stress. Very thin layers were then removed from the ground surface by hand honing until the specimen regained its initial straightness. Measurements of the change in curvature with each thin layer removed permitted calculation of the stress distribution as is shown in the chart. Surface stresses of this magnitude are not unusual in ground production parts, but we are seldom aware of their presence unless actual failure has occurred.

Obviously, a stress of 270,000 psi, a stress just below the fracture point of full-hard steel, could not be supported by the steel in the annealed state, from which it follows that the stress layer was hardened by the heat cycle of the grinding operation to not less than Rockwell C 55. The extreme thinness of the hardened layer presents an interesting problem in hardness measurements, as is shown in Table 1.

TABLE I

	Unground	Ground
Rockwell B	88	89
Rockwell C	5	5
Vickers Brinell	193	199

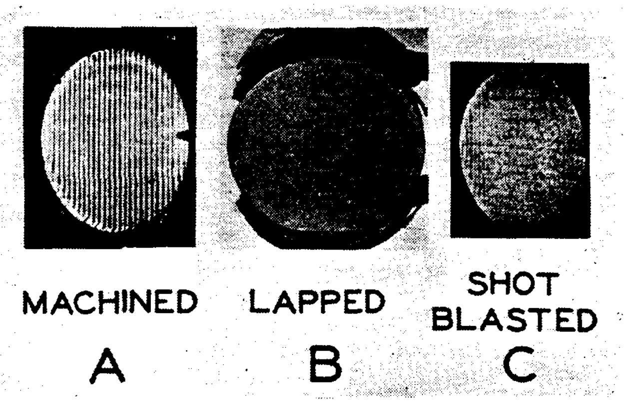
This table demonstrated the futility of our normal hardness-measuring technique for measuring the hardness of the most significant portion of our machine parts, the surface layer.

Internal stresses often result from the cooling of castings and forgings or from the vigorous heat transfer of heat-treating. Many parts, such as crankshafts, axle shafts, and camshafts require straightening during processing. Since the straightening operation is usually done at room temperature and since the part is rarely stress relieved after straightening, the result is severe internal stresses.

■ Machining Damage

In turning, milling, and other machine operations, it is necessary that metal be removed at a minimum cost, and therefore the cutting tools must often take deep cuts at high feed rates. Since metal cutting is more accurately described as a metal-tearing operation so far as stresses are concerned, we need not be surprised to find serious internal stresses to considerable depths after machining. When metal cutting has been unusually severe or after operations such as punching and shearing, we often find that the surfaces are actually fractured. Finish machining or grinding rarely goes deep enough to remove the internally stressed metal from previous rough machining and, of course, these finishing operations add stresses of their own. Whenever it is economically practicable, internal stresses that produce tension in any surface layer subjected to cyclic tension stress should be reduced or removed or, better still, converted to compressive stress by suitable treatment because all fatigue failures are due to tension stresses.

In connection with machining damage, an interesting and perhaps important observation has recently been made which indicates that the layer "injured" by machining is deeper than is generally believed. It also shows that the "injured" material does not recover by heating for long periods at high temperatures. Fig. 21A shows a bar of 4615 steel as it appeared after rough machining on a shaper. This piece was then carburized for 8 hr at 1700 F, cooled in the box, reheated to 1500 F, quenched in oil, and drawn at 300 F for 1 hr. The machined surface was then ground in a direction at right angles to the shaper marks to a depth of 0.0055 in. below the last visible tool mark, after which it was polished as shown in Fig. 21B. Finally, the polished surface was shot blasted, whereupon the machining marks (vertical lines) and the grinder marks (horizontal lines) reappeared as shown in Fig. 21C, showing that the material is not uniform in resisting the shot blasting, notwithstanding the long period at elevated temperature. There is no evidence at present that the effect brought out by this experiment is significant in fatigue.



■ Fig. 21 – A steel part, rough machined as shown in *A*, after being heated at a high temperature for a long period, was ground and polished smooth, as shown in *B* – shot blasting the polished surface again brought out the machining marks, as shown in *C*

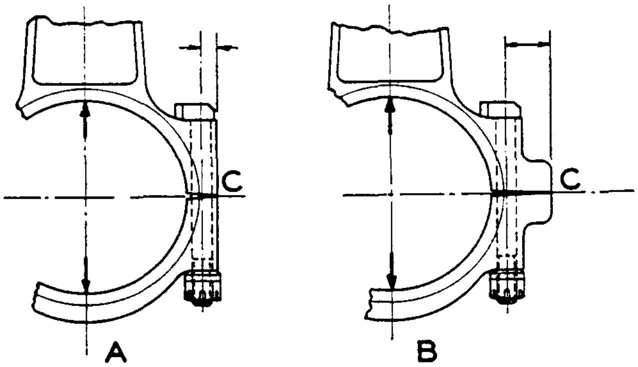
It is presented here merely to emphasize that there is much that is not known about our materials and processes.

■ Bolt Failures

The fatigue vulnerability of bolts and studs has been discussed in many papers[8], 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 the fatigue vulnerability due to insufficient bolt tightness. A bolt or stud should be tightened to a load equal to or 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 and the bolt load is, therefore, 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 bolt.

An exaggerated case of this kind is shown in Fig. 22A,



■ Fig. 22 – Connecting-rod design – *A* causes excessive tension stress on the bolt, *B* is an improved design

in which the bolts are excessively stressed in tension and in bending because the distance 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. Fig. 22B illustrates an improved design in which the fulcrum point *C* is farther 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 member, as is frequently encountered in practice.

If the bolt in Fig. 22B should fail by fatigue, the failure could still not be charged to insufficient bolt strength because, as stated, if the initial bolt tension is less than the applied 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 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 illustrates a case in which a bolt fatigue failure is not the fault of either bolt strength or of design but is chargeable to bad assembly practice.

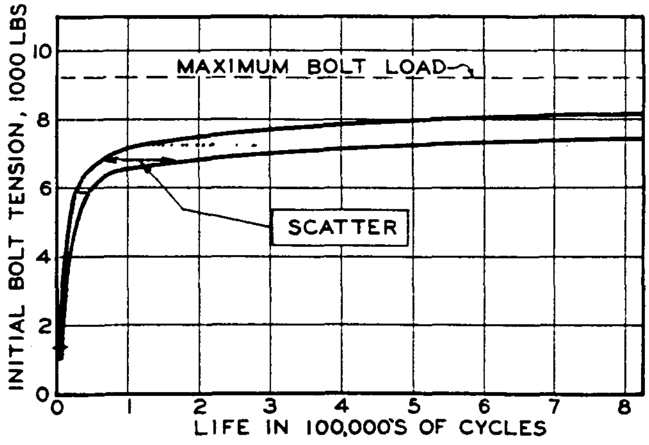
The vulnerability to fatigue as a function of bolt tightness is shown in Fig. 23. In these tests, all bolts were subjected to a cyclic tension load of 9210 lb, but were tightened to initial tensions of 1420, 5920, 7220, and 8420 lb. Fifteen bolts were tested in each of the three lower groups as shown in the graph in order to establish partially 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 was not failed after 10,000,000 stress cycles. These are not shown on the graph because they would compress the scale to undesirable proportions. The bolts used in these tests were ⅜ in. in diameter accurately dimensioned and finished. The threads were uniform, 24 threads per in. and ground to close limits.

■ Stress Range in Bolts

The stress range to which these bolts were subjected is the difference between the initial load and the maximum operating load, and 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 from these tests as shown 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 of the threads in compression. In these rolled threaded bolts, the fatigue durability of the threads was increased sufficiently to cause failure in the bolt 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 evident that the fatigue strength of bolts and studs stressed in tension is dependent upon the initial tension applied by the nut and upon 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 wherever possible. Short studs or bolts are more vulnerable to

[8] See "Prevention of the Failure of Metals under Repeated Stress," by Battelle Memorial Institute, John Wiley & Sons, Inc., 1941.

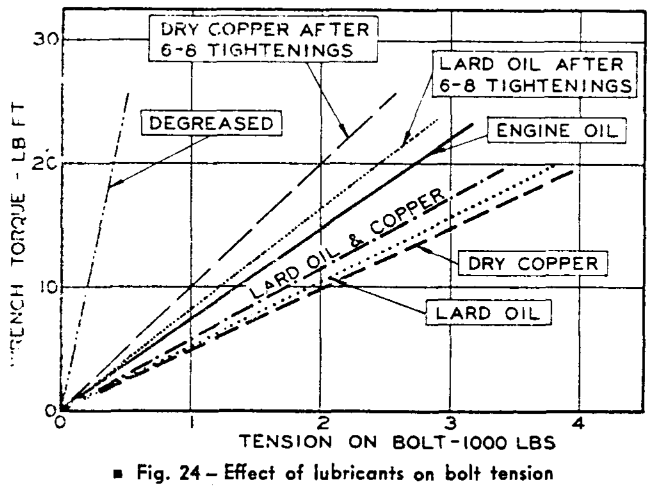


■ Fig. 23 – Effect of initial tightness on bolt life

fatigue than long ones. Their elastic extension is small, and therefore a slight loss of dimension by corrosion, plastic deformation, or wear will lose preload and they will fail by fatigue.

The practice of plating screw threads with soft metal to avoid corrosion is a definite fatigue hazard and should be avoided. The soft plate is so weak that it squeezes out from between the loaded thread surfaces and reduces the stud or bolt tension, thus promoting fatigue failure. A slackening of 0.001 in. on a stud holding a ½-in. flange may result in a loss of 50,000 psi preload. If protective plating must be used, it should be of a hard metal and of minimum thickness.

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 to the wrench is very unreliable because of the variability of the friction. Fig. 24 records tension measurements plotted



■ Fig. 24 – Effect of lubricants on bolt tension

against wrench torque in lb-ft for 5/16-in. diameter capscrews having 24 threads per in. It will be seen that the bolt tension varied as much as 10 to 1 for constant wrench torque depending upon the lubricant that was used. The mechanical efficiency of this bolt varied from 1 to 10%, as may be calculated from the chart.

There is little need for metallurgical examination of failed bolts or studs, or for considering design changes until it has been shown that the failure was not the fault of the man on the wrench.

Preloading 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 bushing tightly into the spring eye.

■ Materials Are Elastic

A common cause of fatigue vulnerability is the belief apparently held by many designers and engineers that our structural materials are rigid. Many fatigue failures can be traced to elastic deflection for which no allowance was made in the design. Elastic deformation of mating parts may be such as to concentrate the load in a small region, as occurred under the conditions described for the bolt in Fig. 22A.

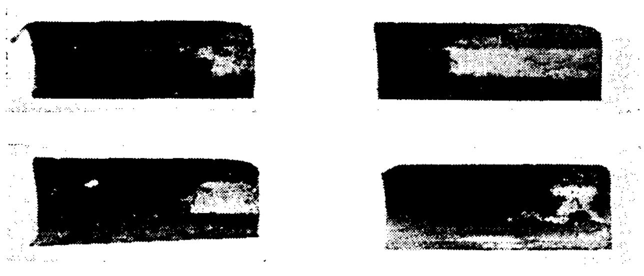
Under operating conditions, a crankshaft may be so elastically deformed in twisting and in bending that the bearings are only partially effective in supporting the load. The bearings are frequently found to be plastically deformed or worn "bell mouthed" to accommodate the elastic gyrations of the crankshaft.

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 over which the load is assumed to be uniformly distributed, 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 by:

1. The elastic characteristics of the housing in which they are contained.
2. The elastic characteristics of the bearings by which they are supported.
3. The elastic characteristics of the shafts upon which they are mounted.
4. The elastic characteristics of the gears themselves.
5. The accumulated dimensional errors in all the supporting parts as well as the errors in the cutting of the gears.
6. The necessary and accidental clearances in the supporting parts.
7. Misalignment of supporting parts through thermal expansion.
8. 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 for the parallelism between mating teeth, as visioned by the designer, to 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 deflection of some of the supporting parts is not linear with respect to the load. As ordinarily designed, the load on gear teeth is never uniformly distributed over the length of the teeth but is always concentrated toward one end of the teeth. This localization of the load is shown in Fig. 25, which is a record of the contact impressions of gear teeth under load in a commercial gear box. Load localization cannot often be seen by examination of a gear that has been in service because, usually, each tooth of each gear makes



■ Fig. 25 – Contact impressions of gear teeth under load in a commercial gear box

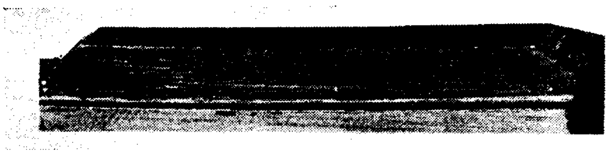
contact with all of the teeth in the mating gear and, therefore, the summation of all contacts under all load conditions will be seen by the examiner.

■ Localized Gear-Tooth Load

The illustration shown in Fig. 26A is of a gear that failed in service. This gear was "rescued" while on its way to the metallurgical department to find out what was wrong with the material to cause the fatigue failure. Note that the failed tooth is broken at one end which, incidentally, is typical of almost all failed gear teeth. An adjoining unbroken tooth shown in Fig. 26B tells us that failure occurred because only a small part of the tooth was actually supporting the load in spite of the generous tooth length that was provided by the designer. This gear would have been just as durable had it been designed to one-fifth the tooth width that was actually provided. Clearly, this is a mechanical and not a metallurgical problem. The real trouble was inadequate support of the gears and other mechanical errors as enumerated.

■ Fig. 26 – Localized tooth load of a gear that failed in service

A – Gear tooth that failed in service



B – Nature of contact, as shown by adjoining unbroken tooth – load was concentrated on left end

It may fairly be argued that this is an unusually severe case and that it is not typical of gear fatigue. Actually the most unusual thing about it was that it could be diagnosed before it was cut into sections and the evidence etched away, as so often happens in metallurgical examinations.

In the 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.[9]

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[10], hypoid, and in some spur and helical gears. In "rigidly" mounted gears, this is accomplished by curving the teeth barrel shape or the equivalent in such a 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.

Load concentrations at the ends of gear teeth can sometimes be avoided by increasing the elastic deflection, as is done in simple spur reduction gears by providing the gear with a thin diaphragm web located central of the gear width, or by more complex construction in other forms of gears, but it cannot be done by accurate dimensions alone.

■ Gear-Tooth Pitting

The pitting of gear teeth is a form of fatigue that is induced by tensile stress from compression loads on the contacting tooth surfaces. The magnitude of the compression stress varies with relative curvature of the contacting teeth in accordance with the Hertz formula, it 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 tooth barrel shaping or equivalent, also reduce the contact compressive stress. The relative curvature and, therefore, the contact 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 the 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 the automobile transmission. 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. This form of lubrication will wash away the heat of friction while it is still on the surfaces of the teeth and will prevent excess oil from reaching the contacting teeth, provided, of course, the sump is dry.

■ Generalized Fatigue "Laws"

The conventional approach to studies of the fatigue of metals is through laboratory tests on several arbitrary forms of fatigue specimens. During the many years that such tests have been made, a vast amount of fatigue data have been accumulated. These data have enabled us to

[9] See *Automotive Industries*, Vol. 77, Sept. 25, 1937, pp. 426-432, Oct. 9, 1937, pp. 488-493: "Factors Influencing the Durability of Automobile Transmission Gears," by J. O. Almen and J. C. Straub.
[10] See ASTM Proceedings, Vol. 35, Part II, 1935, pp. 99-135: "Rear Axle Gears; Factors which Influence their Life," by J. O. Almen and A. L. Boegehold.

formulate somewhat generalized "laws" on the behavior of various specimens subjected to repetitive stresses of several kinds.

We have found that steel, under most laboratory conditions of repetitive stress has a fairly well-defined limit of stress, known as the fatigue endurance limit, below which it will endure for an infinite number of stress cycles; that the fatigue endurance limit of steel is roughly proportional to the ultimate strength of the material but that the proportionality varies with the range of the applied stress. We also know that, under certain other test conditions, steel does not have a fatigue endurance limit, that nonferrous metals generally do not have a fatigue limit, that rough surfaces, notches, section changes, and other discontinuities are detrimental to fatigue strength. These and many other "laws" have been established through laboratory tests under controlled conditions.

The preferred laboratory fatigue test specimen is very carefully prepared to avoid all surface imperfections, abrupt section changes, internal stresses, and other stress-raisers. This is considered necessary because the investigator is usually interested in the inherent properties of the material undergoing test, and he naturally seeks to eliminate all factors that would tend to obscure these inherent properties. There can be no objection to this procedure as it refers to the test specimens, but the data thus obtained have little bearing on the fatigue characteristics of machine parts made from the same material and given the same heat-treatment, because in machine parts surface irregularities, abrupt changes in section, and internal stresses are almost always present.

■ Economic Requirements

In the design of machines and equipment for heavy duty, where weight is not important and where the number of units produced is small, the present practice of designing to large factors of safety is justified because the expense involved in preparing designs to approach exact requirements would far exceed the savings in weight and material.

The same economic considerations that justify overdesign in low-production-volume equipment demand designs of low weight and high stress in many machine parts where weight is all-important, as in airplanes or in large-production-volume machines, such as automobiles, where both weight and cost must be considered. Obviously, the dynamically loaded parts of such machines should be designed with accurate knowledge of their fatigue strength.

■ Laboratory Fatigue Data

When we try to apply quantitatively the accumulated laboratory fatigue data to such design problems, we find that they are almost useless. Published data on fatigue assume that:

1. The operating stress can be determined.
2. Laboratory test specimens are representative of a material when that material is formed into a machine part.
3. The amount and nature of the applied load is known.
4. Load variations occur in an orderly and predictable manner.
5. Representative fatigue curves can be constructed from a dozen or less specimens.
6. Machine parts must be stressed below the fatigue limit to be successful.

These assumptions are not justified in practical design.

■ Stress Cannot Be Calculated

From the data on internal stresses that have been discussed, we may reasonably have some misgivings about the reliability of our stress calculations. From experience with practical machine parts we can only conclude that stress calculations by textbook methods are wholly inadequate unless we generously temper our calculated results with experience. For example, by the usual methods of calculation, crankshafts may be stressed to 20,000 psi, connecting rods may be stressed to 40,000 psi, valve springs 90,000 psi, disc clutch springs to 180,000 psi, while another form of disc spring supports, by calculation, 600,000 psi. Obviously, some of these stress values are ridiculous, but the formulas used in each case conform to the "laws" of mechanics. The actual stress in crankshafts is probably several times 20,000 psi, while the 600,000 psi in the disc spring is not reached because of yielding in local highly stressed regions.

The unreliability of stress calculations has almost been forgotten by seasoned designers because they no longer take the numerical values of their stress calculations literally. Instead, they have learned by experience that, by the usual methods of calculation, the numerical values have different meanings for different machine parts; that is, somewhat rough empirical correction factors are applied.

■ Extensometer Readings of Doubtful Value

There is a growing interest in various devices employed to make direct measurements of stress, such as by photoelasticity, brittle lacquers, extensometers, and similar instrumentation, in the belief that these devices will provide accurate stress data. The accuracy of stress data from such measurements is usually greater than can be obtained from the most involved mathematical analysis, but that they are far from reliable can easily be shown by fatigue tests. Two specimens may vary widely in fatigue strength depending upon minute differences in surface finish or internal stresses. Since internal stresses are often desirable and are frequently unavoidable due to processing operations, such as machining, heat-treating, straightening, or grinding, as has been discussed, and since surface finishes vary all the way from rough forgings to lapped or honed surfaces, there is little reason to expect accuracy from extensometer readings, and even less for photoelastic tests, since photoelastic specimens must be free from internal stress and must be made of another material.

Photoelastic and extensometer readings are measures of elasticity in which the changes in dimensions are the statistical average of all of the material involved in the measurement. Fatigue tests provide a strength measure of the weakest portion of the material involved, usually at the surface, even though it be submicroscopic in size. Obviously, we cannot expect agreement between fatigue measures of stress and the stress readings obtained from elastic measurements alone.

Even if stress could be determined, the fatigue data from laboratory specimens could not be used because machine parts cannot be finished with the care and exactness that is given laboratory specimens. Abrupt section changes cannot be avoided, high internal stresses are often present as a result of processing or because of local heating as from bearing friction, surfaces are subject to bruises and to corrosion of various kinds. These effects cannot be evaluated in terms of arbitrary stress-raisers in controlled laboratory specimens.

■ Operating Loads Rarely Known

In the kind of machines under discussion, the dynamic loads are rarely constant for any appreciable time but vary up and down the load scale in an unpredictable manner. Only a small percentage of the total number of stress cycles are at maximum load, and this percentage will not be the same in the hands of any two operators.

This brings up the question of damage by overstress and recovery by understress as has been observed by several investigators in tests of laboratory fatigue specimens. No doubt such effects occur also in dynamically loaded machine parts, but how are such laboratory data to be applied to machine parts when the schedule of overload and underload is beyond control?

The development of engineering materials, designs, and processes requires that we conduct laboratory tests by which these factors may be evaluated, but to devise a reliable laboratory test is far from simple. The common belief that we can reproduce the conditions of service in a laboratory test is wholly erroneous. By the time the laboratory investigator on any particular part has provided for all of the conditions that occur in service he will have a complete machine in actual service.

The useful strength of materials in dynamically loaded parts is the fatigue strength of such parts under actual service conditions. No other measure will suffice. The strength of a part cannot be determined by tests of the kind that are commonly made in laboratories, many of which are used because they are easy to perform and not because they give useful information.

■ Compromise Treatments

Many materials and processes have been graded and are still being graded by laboratory tests which are now known to have been very costly to industry. For example, the fiction that a carburized part should have a hard case to resist wear, and a tough core to resist breakage, arose from laboratory impact tests. In these tests, the strength of the part was judged by the number or intensity of hammer blows it would withstand before fracture. Since gear teeth resisted impact fracture in accordance with the physical properties of the core, it seemed logical to specify heat-treatments to bring out the best compromise between the imagined requirements of the case and the core. Being compromises, these heat-treatments were not the best for either region.

If, instead of counting the number of impacts or measuring the intensity of hammer blows to produce fracture, the gear tooth had been examined after the first impact, the tooth would have been found bent, and therefore ruined, and it would make no difference how many more blows were required to fracture the tooth.

This compromise heat-treatment resulted in reducing the quality of many millions of gears before it was realized that gear teeth fail by fatigue and that fatigue failure, for the usual depth of carburizetion, always originates at the surface of the case. From this evidence, it became clear the heat-treatment should consider the requirements of the carburized case only, and that the properties of the core were relatively unimportant, because, in bending and in torsion, the core serves mainly as a stuffing for the case.

■ Physical Tests

Several kinds of impact tests are still being used and impact specifications appear in many drawings, but no man can explain and substantiate the significance of the test in terms of the service strength of machine parts.

Elongation and reduction of area are carefully measured and are prominent in our specifications, but we do not know their meaning in terms of serviceability of machine parts. We are told that "brittleness" must be avoided, but no matter how brittleness is defined it does not explain why this property is necessarily more harmful than ductility. Most machine parts that are plastically deformed are just as surely failed as if they were broken. We are asked to believe that machine parts generally must possess relatively high ductility and they must therefore be heat-treated to develop this property. However, when we really get down to applying severe dynamic loads, we forget about ductility and specify high hardness that certainly is well within the range of "brittleness" in the usual meaning of the word. Strong fatigue-resistant gear teeth are file hard. Wristpins, ball bearings, roller bearings, shafts, and cams are hard, and they are strong and fatigue-resistant because they are hard.

A gear tooth is just as surely a spring as the coil that actuates a valve. Why, then, must the one be hard and "brittle" and the other be relatively soft and "ductile"? Why can we not avail ourselves of stronger hardened materials? The answer may lie in our concept of brittleness. We do not fear "brittleness" from hardness when hardness is obtained by nitriding. Nitrided surfaces are not notch-sensitive because they are stressed in compression.

Notch sensitivity is probably the inability of a non-ductile material to yield locally and thus reduce tension stresses in local highly stressed regions, such as notches and scratches. The amount of ductility that is required to overcome "brittleness" depends upon the amount of yield that is necessary to reduce local tension stresses. If the surface is sufficiently pre-stressed in compression, local yielding is not required and therefore non-ductile materials will not be "brittle." As we improve our understanding of "brittleness" we may expect to use steels at higher hardness in many parts for which we now specify ductility. We will then gain from the greater inherent strength as well as from the increased strength obtained by compressively stressed surfaces.

The most significant of our easily performed laboratory tests is hardness. Since the static strength of most materials is roughly proportional to hardness, we will know the approximate static strength of a part if the hardness is accurately measured. However, the popular hardness testers such as Brinell or Rockwell are incapable of the accuracy that is required because they penetrate too deeply into the material being tested, and therefore they do not measure the characteristics of that most important part of the material, the surface layer.

Chemical analysis can only indicate the responsiveness to heat-treatment and can measure the potential strength of a steel only by its probable hardenability. Since strength is proportional to hardness, all properly heat-treated steels of equal hardness are equally strong regardless of their compositions.

Laboratory hardenability tests are now coming into general use. This test has much merit provided that we understand its meaning and that we do not debase it, as we are so prone to do by applying arbitrary hardenability specifications without considering the requirements of each particular part. Through-hardenability (approximately uniform hardness through the section) can be very important for parts that are stressed in tension, but it is difficult to see why through-hardness is necessary in parts that are loaded in bending or in torsion, because in such members the stress decreases somewhat linearly with depth, reaching zero at the neutral axis. For this kind of loading, it would seem to be more important to develop heat-treatments that give the type of internal stress shown in Fig. 15 because, being pre-stressed negatively to the applied tension load, the dynamic load-carrying capacity is greatly increased.

The standard laboratory tensile test is, of course, incapable of indicating the useful bending or torsion strength of pre-stressed specimens, particularly when the pre-stressing is deep, as shown in Fig. 15. For such specimens, the tensile test cannot even distinguish between harmful and beneficial pre-stressing. Both would probably show decreased tensile strength, whereas under dynamic bending or torsion loads one would show greatly decreased fatigue strength and the other greatly increased fatigue strength.

However, we have done a reasonably satisfactory job in the past without worrying overmuch about the shortcomings of the methods used. We may be certain that we will do better in the future as more experience is gained and it is in the accumulation and organization of this experience that we can best serve the needs of the future. It is probable that fatigue studies will play increasingly important parts in future designs; but these studies will be based on fatigue tests of actual, full-scale machine parts instead of on laboratory specimens.

■ Fatigue Tests on Machine Parts

Fatigue tests of full-scale machine parts have been made by many laboratories for a long time, but since these tests have usually been made for the purpose of comparing one material, design, or process with another material, design, or process, the tests have been run at arbitrary constant loads without thought to the fatigue-curve characteristics and often without adequate correlation with service requirements. Because of this procedure, we have made little use of the vast quantities of such fatigue data as are now locked in our files in so far as establishing a basis for evaluating material, design, or process for the future is concerned.

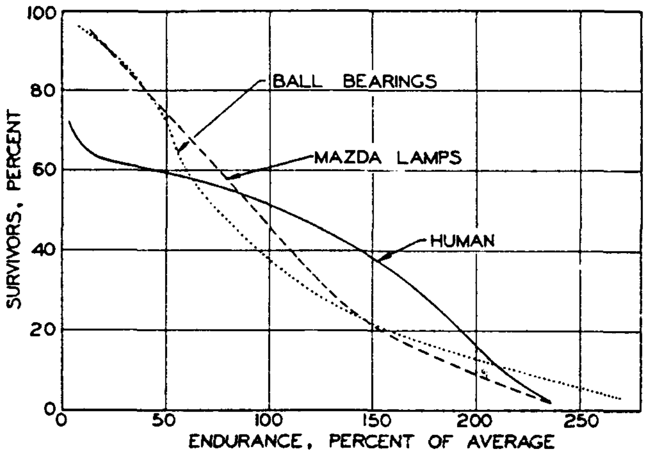
In the few cases where fatigue data on machine parts have been properly organized, we find that they reveal astonishing amounts of fundamental information about the many variables that are present in machine elements, many of which are not even qualitatively revealed by ideal laboratory fatigue specimens.

■ Fatigue Data Are Mortality Data

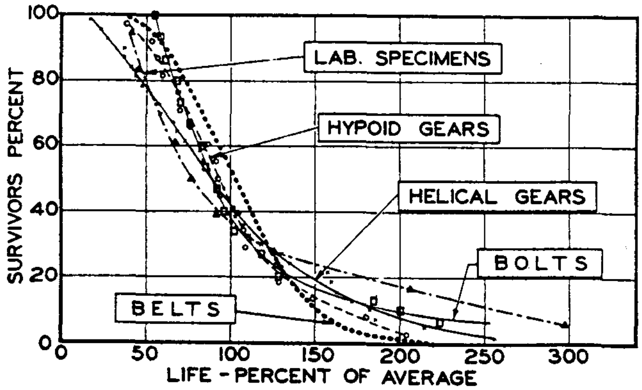
Fatigue data are mortality data and it is just as absurd to expect that reliable actuarial tables can be constructed from mortality data on a half-dozen individuals as to expect that reliable comparisons can be made from fatigue tests on a half-dozen machine parts. When a sufficient number of machine parts are fatigue tested at constant load and plotted in the manner of the well-known mortality curve for human life expectancy, we find remarkable similarity to human mortality experience. Heindlhofer and Sjovall[11] have shown life expectancy curves for commercially identical ball bearings, for commercially identical mazda lamps, and for human beings. These curves are shown in Fig. 27, in which the ordinate is the percentage of units surviving and the abscissa is durability in per cent of average life.

Fig. 28 is a life expectancy chart at constant load for commercially identical transmission gears in complete automobile transmissions, for commercially identical rear-axle gears in complete automobile rear axles, for commercially identical automobile fan belts[12], for commercially identical bolts, and for a group of ideal laboratory fatigue specimens. Similar life expectancy curves will result whether applied to mountain ranges or to the hairs on our heads.

Although the general form of all life expectancy curves is the same, they differ in detail. Note that the expectancy curves for machine parts (Figs. 27 and 28) do not extend to zero life as is the case in the human expectancy curve. Infant mortality is avoided in machine parts because the parts having a low potential life are rejected by factory inspection, a practice that is not followed for humans.



■ Fig. 27 – Comparative endurance – life expectancy curves for ball bearings and lamps compared with the life expectancy curve for human beings



■ Fig. 28 – Similarity of various life expectancy curves – as shown by machine parts, belts, and laboratory specimens

■ Variation in Durability

Another important difference is the relative life span for various machine parts. Note that for automobile rear-axle gears, the life span of the most durable unit was about four times the life span of the poorest unit, but for automobile transmission gears the life ratio from the best to the poorest was about 15 to 1; that is, childhood mortality is higher in automobile transmissions than in automobile rear axles. The life span ratios given should not

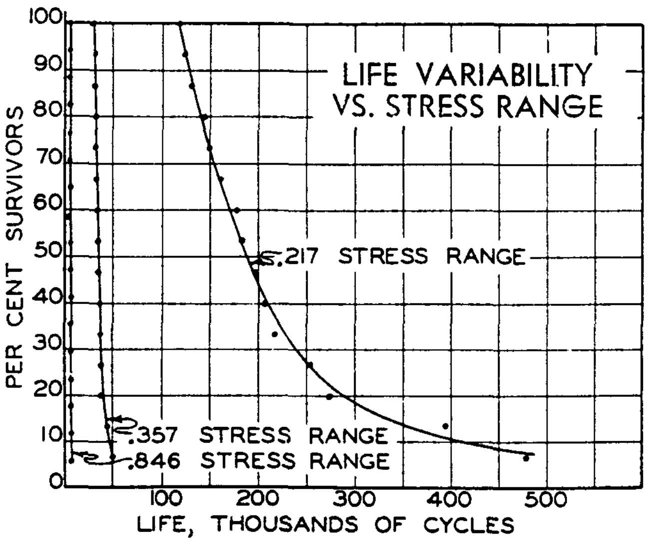
[11] See *Mechanical Engineering*, Vol. 45, October, 1923, pp. 579-581: "Endurance-Test Data and their Interpretation," by K. Heindlhofer and H. Sjovall.

[12] Private communication, R. S. Carter, Goodyear Tire & Rubber Co.

be taken literally because there are not enough test points in either curve to define their limits. As the number of test points is increased, the life ratio of the best to the poorest will increase but the "scatter" will be greater for transmission gears because the variability of stress resulting from end contact is greater.

The percentage variation in life of machine parts will also change as the test load or load range is changed. When tests are conducted at high load or high load range to produce fatigue failure after relatively few stress cycles, the percentage variation from the best to the poorest will be less than if the test is conducted at a lower load to produce fatigue failure after a relatively large number of stress cycles. This is shown in Fig. 29, in which the life variation of the bolts used to determine Fig. 23 is recorded. Note short life and the small variation in life for the bolts that were given low initial tightness (large stress range), and the greater life and relatively great variation in life for the bolts that were given high initial tightness (low stress range). The reason for this variable will become clear when we examine the form of scatter band of fatigue data from a sufficiently large number of fatigue tests.

In the class of light machines where weight must be conserved, it will probably never be possible to design mechanisms to withstand all the abuses that are encountered in service. If an airplane engine, for example, should be so sturdily designed that the shortest lived of each of its numerous parts was failure-proof under all the abusive conditions that may be experienced in service, the engine would be so heavy as to be impractical. As we learn how to increase the durability of each machine element, we will reduce but not eliminate failure hazards. Instead, progress will demand that we take advantage of such improvements by reducing the weight or by increasing the power output.



■ Fig. 29 – Life variability of bolts for different stress ranges

■ Insufficient Test Data

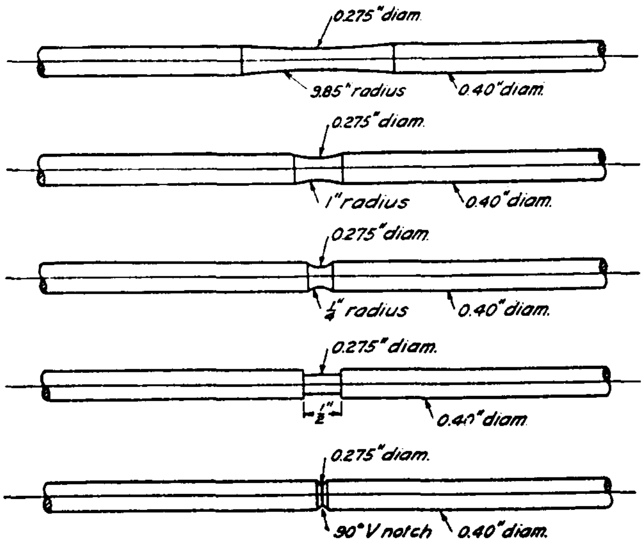
Reliable life comparison of machine parts demands a large number of tests unless the life difference is very great. It is obvious from the mortality charts that have been shown that, on the basis of a few tests, the poorer design, material, or process may rate higher than the better design, material, or process. Yet nowhere in the literature do we find fatigue data approaching even the minimum requirements of reliability. The reason is largely that most of the investigators in this field, particularly in work on steel, assume that we have no interest in data at any stress except the stress at which the specimen will endure indefinitely.

In practical fatigue testing of machine parts, it should be obvious that comparisons of material, design, or processes cannot be made unless the tests are run to failure and the comparisons are made on the number of stress cycles each will endure. This is true whether or not the part being tested is required to withstand, in service, a very large number of stress reversals at maximum load such as a crankshaft or a relatively small number of stress reversals at maximum load, such as chassis springs. Since all representative tests are made at loads that result in failure by fatigue, our interest lies not in the fatigue endurance limit where for steel, under most test conditions, life is infinite, but in that portion of the fatigue curve to the left of the "knee" where life is finite, that is, the sloping part of the curve.

■ Fatigue Curve Slope

The characteristics of the sloping part of the fatigue curve have been obscured in most of the published S-N plots (a) by never having enough test points and (b) by the popular custom of plotting fatigue data on semi-log charts. In the very few cases where published data contain a considerable number of test points, we find that when plots are constructed on log charts, the points tend to lie on a straight line instead of on a curved line as when they are plotted on semi-log charts.

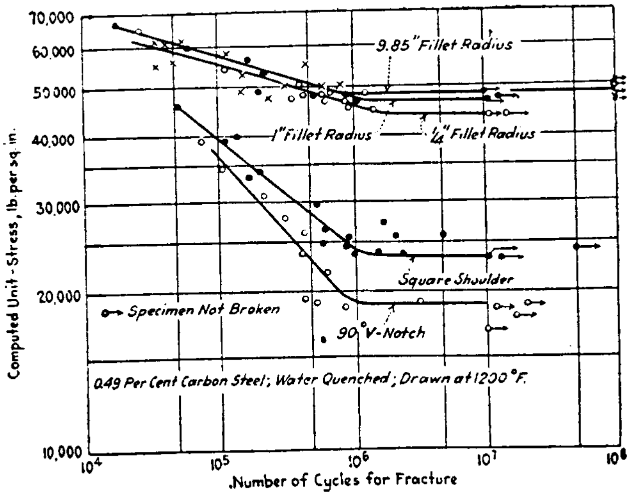
Fig. 30 shows a series of fatigue specimens used by Moore and Kommers[13] to determine the effect on fatigue of varying degrees of stress concentration.



■ Fig. 30 – Specimens used to study the effect of shape on endurance limit

The resulting fatigue curves, plotted on logarithmic coordinates, are shown in Fig. 31. The authors compare these specimens on the basis of calculated stress at the fatigue endurance limit; that is, the stress at the "knee" where the curve becomes horizontal. However, as stated

[13] See "Fatigue of Metals," by Moore and Kommers, McGraw-Hill Book Co.



■ Fig. 31 – S-N curves for the specimens shown in Fig. 30

above, our interest is in the finite life region of the diagram; that is, in the characteristics of the curve lying to the left of the "knee." Observe that as the "notch" severity of the specimen section is increased, the slope of the curve increases, and that the curves, if extended leftward, tend to cross one another.

■ Machine Parts Fatigue Curves

Fatigue curves of machine parts, no matter how well finished or how carefully rejected for detectable flaws, almost invariably show steeper slopes than are shown by well-finished fatigue specimens and, therefore, presumably the fatigue strength of a material as determined by ideal test specimens is not obtainable when that material is formed into a machine part. Permissible stress at the fatigue limit of a machine part may be less than 10% of the ultimate strength of the material, whereas laboratory test specimens may indicate 50% or more as obtainable.

The difference in slope of the fatigue curves suggests that this characteristic promises a way whereby we may eventually greatly improve our accuracy in determining the strength of machine parts. This is now being done in rating the load capacity of ball bearings, roller bearings, automobile transmissions, and rear-axle gears.

The lines plotted in Fig. 31 are intended to represent the averages for the specimens tested. Note the wide scatter of the test points and the increasing scatter of the points as the slope increases. Note also that, generally, the scatter decreases toward the left of the diagram. The significance of this scatter is not apparent in the diagram due to the limited number of test points, there being an average of only 12 failed tests for each type of specimen.

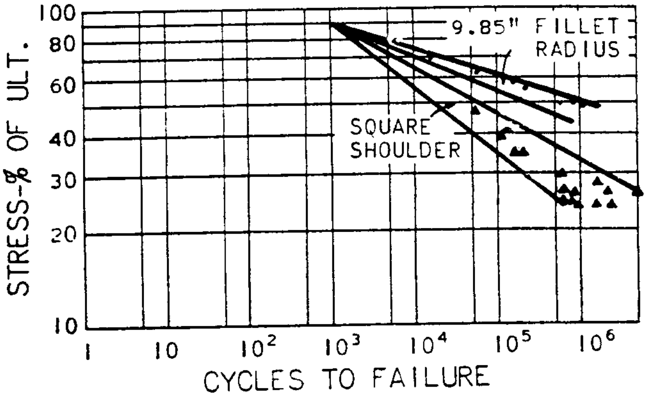
■ Slope Indicates Stress Concentration

The scatter of test points is due to unavoidable differences in test specimens no matter how carefully they are made. Since these differences constitute varying degrees of stress concentration, the fatigue line representing the poorest of a group of specimens should lie on a steeper slope than the fatigue line representing the best specimens. This is for the same reason that the average slopes of the specimens shown in Fig. 30 increase with the severity of the stress concentration as shown in Fig. 31.

[14] See Report of the Research Committee on the Fatigue of Metals, ASTM Meeting, June, 1941.

The test points for any group of specimens would, therefore, be expected to lie within a scatter band diverging from the region of high stress, and will be of the order shown in Fig. 32.

If a sufficient number of specimens had been tested, and if the stress scale proportionality were the same for all specimens, it is probable that the sloped lines in Fig. 31 would all tend to converge toward a point in the vicinity of 1000 cycles and 90,000 psi, somewhat as indicated in Fig. 32, in which the fatigue slopes of specimens 1 and 4



■ Fig. 32 – Probable form of scatter band of specimens 1 and 4, Fig. 30

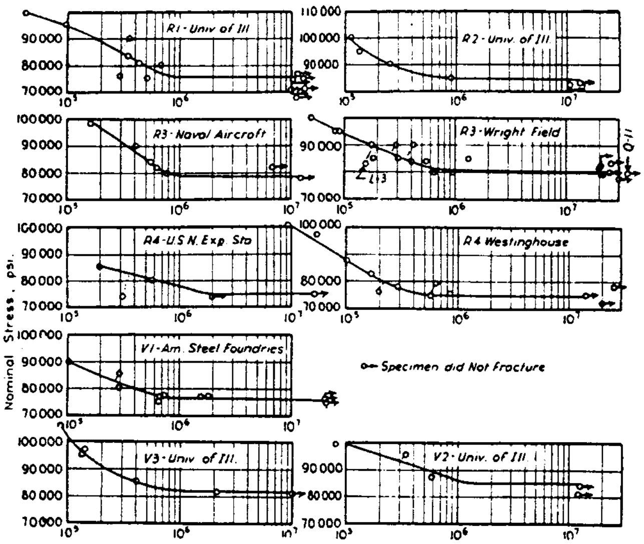
of Fig. 30 are shown as converging bands rather than lines. This region of intersection is suggested because the ultimate strength of the material tested by Moore and Kommers was approximately 95,000 psi and, obviously, if the stress scale is correct for each type of specimen, they would all have approximately the same strength at one stress cycle. The point of intersection would probably be at a considerable number of stress cycles because the ductility of the material permits adjustment of stress yield, thus reducing the influence of local highly stressed points.

For very brittle material, the intersection point of the fatigue curves for the type of specimens shown in Fig. 30 would probably be near the ultimate strength and nearer one cycle of stress.

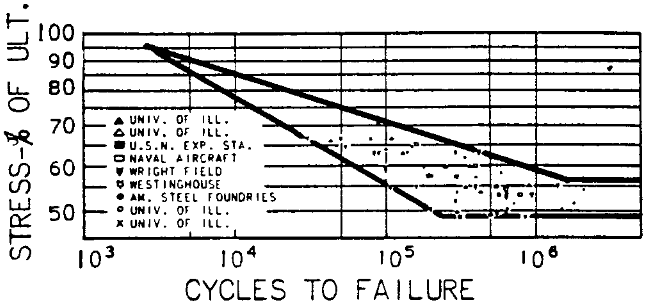
There are not now available sufficient data on any specimens to complete a group of fatigue diagrams to the region of intersection. Knowledge of the characteristics of fatigue curves at high stress would be valuable in industry since it would greatly facilitate interpretation of fatigue tests on machine parts. Such tests could be evaluated in terms of the slope of the fatigue curve, which would also give a clue to the actual stress, if desired, in the part being tested.

The research committee of the ASTM recently sponsored a cooperative test program in which several laboratories conducted independent fatigue tests on identical material (heat-treated SAE 4340) under similar test conditions. The results were reported in an ASTM research report[14] from which the group of plots shown in Fig. 33 were taken. Note the wide disagreement between the curves from the several laboratories in the fatigue limit as well as in the sloping part of the curves. When all of the 59 individual failed points are plotted on a log-log chart, as is shown in Fig. 34, we begin to see a semblance of order, in that all of the points lie within a scatter band of the same converging form as is shown in Fig. 32.

In passing, it is interesting to note that in Fig. 33 we find nine test points at 85,000 psi load, which are the points plotted in the life expectancy curve, Fig. 28, to show



■ Fig. 33 – Semi-log plot of data reported by the ASTM Research Committee on Fatigue of Metals



■ Fig. 34 – Log-log replot of data plotted in Fig. 33

that the life variation of laboratory specimens is of the same order as the life variation of gears and other machine parts.

Published data on fatigue of metals contain numerous tests showing the same general trend of increasing slope with increasing stress concentration whether due to differences in specimen shape, specimen size, mechanical working of specimen surface, surface coatings, fillet radii, surface finish, or to variations between "identical" specimens. As previously stated, this tendency toward convergence is often not apparent in the published curves because the investigators have plotted their data on linear ordinates and logarithmic abscissa, and always there are insufficient test points. The following diagrams copied from published papers have, when necessary, been replotted for the sake of uniformity on logarithmic coordinates to the same scale as used by Moore and Kommers in which the stress scale is four times the scale of stress repetitions. The slopes of the curves are calculated as the measured horizontal distance multiplied by the scale ratio divided by the measured vertical distance:

$$\frac{\text{(Abscissa} \times \text{Scale Ratio)}}{\text{Ordinate}}$$

Note that this is the reciprocal of the slope as ordinarily used in engineering, but it is a more convenient form.

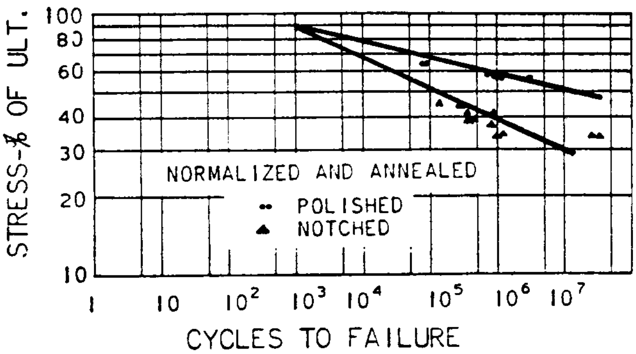
[15] See ASTM Proceedings, Vol. 37, Part II, 1937, p. 199: "Fatigue Properties of Metals Used in Aircraft Construction at 3450 and 10,600 Cycles," by T. T. Oberg and J. B. Johnson.

[16] See *Modern Plastics*, Vol. 19, September, 1941, pp. 57-62, 78: "Mechanical Tests of Cellulose Acetate," by W. M. Findley.

[17] See *Luftfahrt-Forschung*, Vol. 18, March 29, 1941, pp. 102-106: "Uber den Einfluss von Bohrungen mit Gewinden und Kerbverzahnungen auf die Zeit- und Dauerfestigkeit von Leichtmetall-Flachstüben," by H. Bürnheim.

Oberg and Johnson[15] report a comparison between polished and notched specimens, Fig. 35, with results similar to the exeperiments by Moore and Kommers, Fig. 31.

Surface treatment of the test specimens, other than the

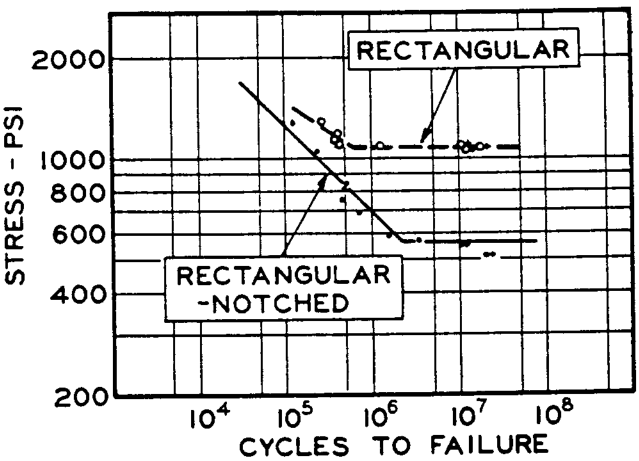


■ Fig. 35 – S-N diagram for a metal used in aircraft construction – 4134 steel

degree of smoothness, has a marked effect on fatigue strength. Horger and Maulbetsch[2] compared normal well-finished specimens with specimens that had been subjected to a rolling operation which introduced compressive stresses in the surface layer with the results shown in Fig. 1. Since the rolled specimens were pre-stressed in compression, the subsequent tension stresses during the test were reduced, as is shown in Fig. 6; hence the difference in the slope of the curves for the two types of specimens. Since this treatment would be ineffective in a tensile test, the lines should converge in the manner shown.

When replotted on log-log charts, published fatigue curves on other materials than steel exhibit the same tendency to converge toward the left and to increase their slope as notch effects are increased.

Findley[16] conducted fatigue tests on cellulose acetate specimens with the replotted results shown in Fig. 36. As

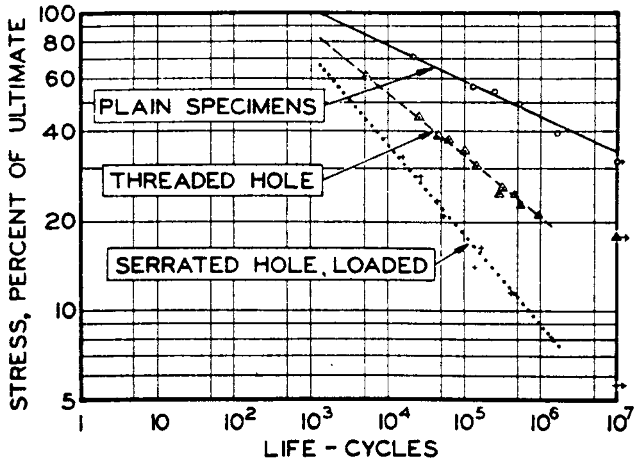


■ Fig. 36 – S-N diagram for cellulose acetate

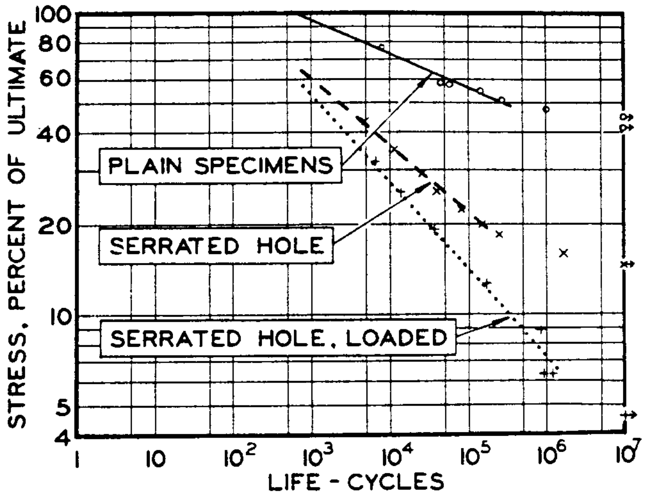
is usual, there are not as many test points as are needed to define the region of convergence but the trend is definite.

Bürnheim[17] reported fatigue tests on duralumin (Fig. 37), and magnesium (Fig. 38), in plain and various notched specimens. These replotted charts are more satisfactory than most published data in that the region of convergence is more clearly defined.

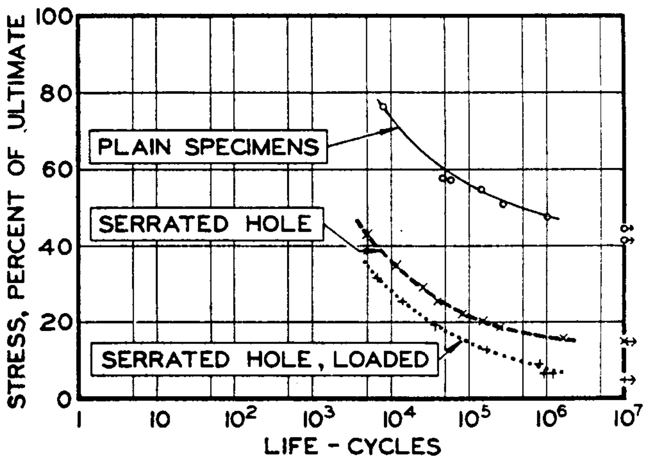
Fig. 39 is the original semi-log plot of the fatigue data on magnesium that is shown in the replot of Fig. 38. Note



■ Fig. 37 – S-N diagram for duralumin showing the effect of stress-raisers



■ Fig. 38 – S-N diagram for magnesium alloy (Electron) showing the effect of stress-raisers – log-log plot



■ Fig. 39 – S-N diagram for magnesium alloy (Electron) showing the effect of stress-raisers – semi-log plot

that when plotted in this manner, the data lose much of their significance and may not safely be extrapolated.

From the foregoing it seems reasonable, as a working hypothesis, to assume that, except possibly for very ductile metals, *the slope of the fatigue curve, as measured on a log-log plot, may be considered a measure of effective stress; and fatigue curves for varying stress concentrations converge toward a point near the tensile strength of the material and at some considerable number of stress cycles.*

It should be remembered that the tensile strength in a test in which the load is slowly increased is lower than in a tensile test in which the load is maintained for a very short time[18] as in a fatigue test, and also that there is a considerable variation in the tensile strength of any material as measured by a number of tensile test specimens. Therefore, the tensile strength on a fatigue chart would actually plot as a band and not as a line and would lie above the normal tensile value. Likewise, the lines of a fatigue plot would converge to a region above the normal tensile strength and would probably not meet at a point. However, the inclusion of these variables would considerably complicate the above hypothesis and since they occur in a region of the fatigue plot that has little or no practical value, they may, for the present be ignored.

The application of this hypothesis to the fatigue strength of machine parts has some important implications. A large variety of machine elements are constantly being tested for relative durability in the laboratories of industries engaged in the manufacture of light-weight, high-output machines. In most cases these fatigue tests are intended to compare one design, material, or process with another.

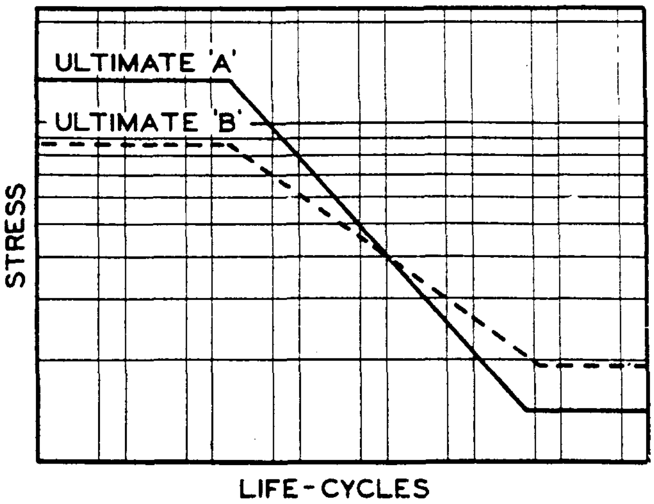
It is axiomatic that nothing can be learned in regard to limiting loads except through tests to destruction and, therefore, the fatigue tests for practically all parts are run to failure and the comparison is made on the number of stress cycles at constant load that each part will withstand. As stated above, this procedure is followed regardless of whether, in practice, the part in question is stressed below the fatigue limit or whether it is a part requiring relatively short life at maximum stress.

This method of evaluating test results is subject to serious error for several reasons. If it is true that fatigue curves radiate from a point in the high-stress region, it is obvious that comparisons of specimens cannot be made on a percentage basis only, since the percentage difference will vary all the way from zero to infinity depending upon the load that is applied during the test. Furthermore, since the scatter band for each test part should also radiate from the same point, as was shown in Fig. 32, the width of the band in terms of life may be several hundred per cent, as shown in Fig. 34; therefore, unless a considerable number of tests are run for each part, there is no assurance that whatever life difference is found is not just the chance location of these particular test points within the scatter band. It is easily possible that the better design, material, or process will apparently rate lower than the poorer design, material, or process if insufficient tests are made.

It is possible that the average fatigue curves for two materials having different tensile strengths and yield points will cross at some point in the finite life region due to differences in sensitivity to stress-raisers. In such cases, life comparisons may be positive for one material at one test load and negative for the same material at another test load. The diagram shown in Fig. 40 illustrates such a situation.

It is evident, therefore, that true comparisons can only be obtained through fatigue tests on a sufficient number of parts at varying loads to outline the slopes of the scatter-band limits. While this may appear to be an impractical requirement, it is not so difficult as it seems. It is only necessary that the results of the present routine tests be

18 See Institution of Mechanical Engineers Proceedings, Vol. 141, April, 1939, pp. 175-185: "Deformation and Fracture of Mild Steel under Cyclic Stresses in Relation to Crystalline Structure," by H. J. Gough and W. A. Wood.

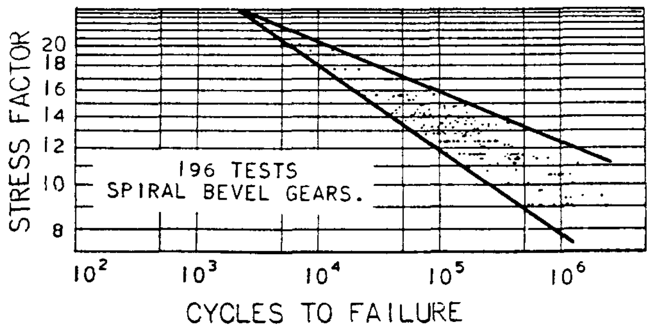


■ **Fig. 40 – Fatigue curves for two materials having different strengths**

accumulated on a fatigue diagram, and in a relatively short time fatigue curves and their scatter bands will be available for a large variety of machine parts.

Only occasionally are fatigue tests on machine parts run at various loads. In the very few cases where data from a reasonable number of such tests, run at sufficiently large load differences, are available on commercially identical parts, a reasonable number being 100 or 200, we find that the scatter of the test points when plotted on logarithmic coordinates falls within a well-defined pattern. This pattern tends to radiate from a point at high stress and a low number of stress cycles and to diverge to a broad band at low stress and high number of stress cycles just as was suggested in Fig. 32.

This is clearly shown in Fig. 41, which is a fatigue diagram of about 200 complete automobile and truck rear axles of various makes and sizes.[10] The stress scale shown in this diagram is not actual stress but is believed to be



■ **Fig. 41 – Fatigue diagram showing scatter band for fatigue tests of 196 spiral bevel rear-axle gears**

proportional to actual stress. The axles were tested at loads to produce failure of one or more pinion teeth through the range of from 7000 cycles to 1,000,000 cycles.

The scatter of the test points is due to variations in one or more of the many variables that are always present in commercially similar parts, such as internal stresses, fillet radii, cutter scratches, bearing, shaft and housing deflections, warpage in heat-treatment, and so on.

The slope of the average durability line, calculated as the horizontal distance divided by the vertical distance measured on logarithmic coordinates is approximately seven, while the slopes of the upper and lower limits of the scatter band are respectively nine and five.

The intersection point at the left of the diagram should lie near the ultimate strength of the material (approximately 300,000 psi) which, if proved, would supply us with a measure of actual stress for the entire diagram.

The diagram shown in Fig. 41 is not ideal as a proof of the scatter band or the intersection point, since it includes a variety of axles made from various alloy steels variously heat-treated, for which the stresses were calculated by an empirical formula.

Satisfactory determination of the characteristics of the scatter band would require a large number of fatigue tests on one form and size of specimen made of one type of material similarly heat-treated and tested to produce failure over a range of stress repetitions from as near a single cycle of stress as possible to the fatigue limit.

Data approaching these requirements[19] have been accumulated by the various ball- and roller-bearing manufacturers, but the published data are not yet extensive enough to define the form of the scatter band. Particularly, more data are needed in the very low stress range and in the very high stress range.

However, fatigue data on ball and roller bearings need not in all particulars agree with fatigue data on other forms of machine parts, since failure of rolling bearings usually originates below the surface of the material. Surface influences, which play so important a part in fatigue of ordinary machine parts are, therefore, absent in rolling bearings. This would be expected to influence the permissible stress, and possibly the form of the scatter band. The scatter band as reported by Macauley[20] and by Styri[21] is parallel to the average life curve throughout the life range shown.

Ball and roller bearings are also peculiar in that their curves do not show a fatigue endurance limit as is usually found in fatigue specimens. According to the catalog ratings, the sloped lines continue to more than a billion inner race revolutions, and since there are several stress cycles per revolution, we do not find a "knee" in these curves up to more than five billion stress cycles.[22]

We seek to determine actual stress only as a step in predicting the adequacy or inadequacy of our designs. Any other means that will enable us to predict the performance of our designs will do as well. Ball- and roller-bearing manufacturers do not consider stress at all in their catalog ratings but rely entirely upon tabulated load capacities as determined by service experience that has been correlated with laboratory test data on complete bearings. In practice, we are not only unable to calculate or to measure but we do not even know the manner of load applications in service on the majority of machine parts.

Laboratory fatigue testing of light-weight, high-output machine parts, as well as other laboratory tests such as on fuels, oils, tire wear, and so on must be definitely correlated with service data on the part in question before the results can be accepted. This requires that, for fatigue, tests must be devised that will agree with failures that occur in normal service as to the location of points of fracture and the character of the fractures. It is not important that the test procedures agree with the preconceived notions of service loading.

---

[19] See *The Ball Bearing Journal*, No. 3, 1927, SKF Industries, Inc.
[20] See *The Automobile Engineer*, Vol. 13, July, 1923, pp. 213-223: "The Endurance of Ball Bearings," by A. W. Macauley.
[21] See *Mechanical Engineering*, Vol. 47, June, 1925, pp. 490-492: "General Properties of Ball Bearings," by Haakon Styri.
[22] See *The Ball Bearing Journal*, No. 3, 1937, SKF Industries, Inc.

The slope of the finite life portions of the S-N diagram has been discussed from the standpoint of fatigue tests at constant stress range. Most of the test data presented hereto have been taken from the specimens in which the stress was completely reversed. However, many machine parts are otherwise stressed, as for example, the case of properly tightened bolts in which the stress range approaches zero. Valve springs are stressed through a relatively narrow range in one direction only, being preloaded to approximately 25,000 psi stress, which is increased to approximately 90,000 psi when the valve is fully open. Gear teeth usually are loaded from zero to a maximum stress in one direction only. Crankshaft stresses may be somewhat more complex, being completely reversed in bending during each revolution, while transmitting torque in one direction only.

Many experiments have been conducted to determine the effect of varying the stress range, but again interest lay in the stress at the fatigue limit, and few data are available on the change of slope of the curve with stress range. However, since the stress at the fatigue limit increases as the stress range decreases, as has been amply demonstrated, it follows that the slope must decrease (become flatter) as the stress range decreases.

Moore and Kommers[13] present a modified Goodman diagram from which the fatigue slope for ideal specimens

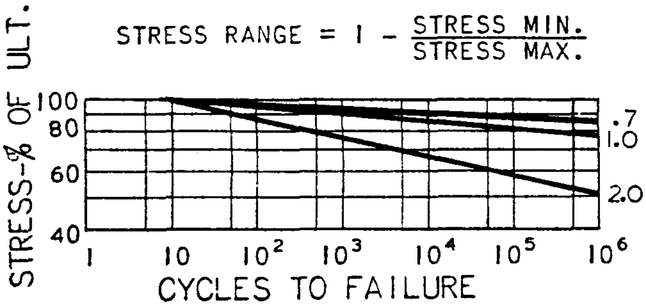


Fig. 42 – Effect of stress range on fatigue life – the slope of the fatigue curves varies with the stress range as well as with the stress concentration

may be constructed for any stress range. Fig. 42 shows a replot of this modified Goodman diagram for three stress ranges. The upper curve represents a small stress range similar to that of automobile valve springs; the second curve represents a stress range of from zero to maximum as for gears; and the lowest curve represents complete reversal of stress as may occur in a crankshaft.

The slopes of these curves are respectively 80, 48, and 17. We thus see that the slope of the fatigue curve varies with the stress range as well as with the stress concentration and, therefore, the hypothesis that the slope of the fatigue curve is a function of the usual conception of stress is no longer tenable. If, however, we state that the slope of the fatigue curve is a function of effective stress, the hypothesis will apply for any stress range.

The methods now used for coordinating laboratory tests with service experience are too haphazard to be completely reliable. Service failures must, obviously, be infrequent, and when true fatigue failures do occur, it is usually the result of harder-than-normal service combined with a specimen lying on the lower fringe of the fatigue scatter band.

Since failures must be infrequent, it is highly important that failed parts be examined by competent observers in order that the true cause of the trouble may be determined. Clear evidence of fatigue failures does not prove that the failed part was primarily responsible. A bolt may fatigue because it was not properly tightened during assembly; a gear may fatigue due to improper support or to a failed bearing; a crankshaft may fatigue due to inadequate or maladjusted vibration damper; and so on without end. It sometimes happens, therefore, that immediate corrections are made to the wrong part and recognition of the true trouble is sometimes greatly delayed.

Laboratory fatigue tests on machine parts must not only duplicate service failure as to location of fracture, but they must, in some cases, produce failure in approximately the same number of stress cycles if accurate life comparisons are to be made. This requires that we distinguish between normal operating stresses and the relatively infrequent overloads that cause the failure.

Fully 90% of all fatigue failures occurring in service or during laboratory and road tests are traceable to design and production defects, and only the remaining 10% are primarily the responsibility of the metallurgist as defects in material, material specification, or heat-treatment. While this ratio is not a measure of the quality of workmanship contributed by each department, there can be no doubt that the metallurgist has a better appreciation of his responsibility for fatigue failures than has the designer, the engineer, or the man in the production department.

You are familiar with the routine that is followed when a failed part is received by the laboratory. The fracture is examined and is found to be due to fatigue, the material is analyzed for composition, sections are studied for all the many things that are metallurgically important and a report is written describing the things that are and are not up to par. But no matter how many possible metallurgical causes of trouble are found, such examination is far from sufficient unless the failure is also examined for design faults and possibly bad lubrication and assembly practice. Most of the failed parts should not be sent to the metallurgist at all, but unfortunately very few engineers or production men are adequately trained in diagnosing fatigue trouble and, therefore, failures are seldom examined for contributing mechanical causes. Most of our engineers pass all fatigue problems on to the metallurgical department with the implication that something must be wrong with the material or with the heat-treatment.

The study of fatigue of materials is properly the joint duty of the metallurgical, engineering, and production departments. Unless all of these departments have an understanding of fatigue phenomena and the factors that promote fatigue, they cannot recognize their individual responsibilities for the product they manufacture. There is no definite line of demarcation between mechanical and metallurgical factors that contribute to fatigue, and there must, therefore, be very close cooperation between the metallurgist and the engineering fatigue specialist, if such there is, or the metallurgist must possess the qualifications of the metallurgist, designer, and machinist. This overlapping of responsibility is not sufficiently understood in industry and hence the engineers are constantly demanding new metallurgical miracles, instead of correcting their own faults. It would be very helpful if the metallurgists would be less willing to look for metallurgical causes of fatigue and insist that equally competent examination for mechanical causes be made. Until this is done, we cannot hope to make full use of our engineering materials.