In this article, which is part of an address given before the Detroit Chapter last winter, the thesis is argued that the machine designer should be most interested in that part of the fatigue curve (the S-N curve, so-called) which shows the cycles to failure at loads which give but limited life — above the endurance limit. Experience shows that this portion of the S-N curve becomes steeper as the test piece contains more intense stress raisers. Application of these considerations to design will be the subject of the final article next month

THE USEFUL DATA TO BE DERIVED FROM FATIGUE TESTS

By J. O. ALMEN
Research Laboratories Division, General Motors Corp., Detroit

WE STILL have far to go before we, as designers and producers of light-weight, dynamically loaded machines, can make full use of our structural materials. We must learn that we cannot tell the machine parts what they must do, but rather we must do as we are told by the machine parts. The rate at which we progress will, in large part, depend upon the accuracy of our interpretation of the complaints that are made to us by the machine elements. These complaints are not easily interpreted because they are often made in terms that we do not understand.

The conventional approach to studies of 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 has been accumulated. These data have enabled us to 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; and 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 non-ferrous metals generally do not have a fatigue limit, that rough surfaces, notches, section changes and other discontinuities lower the 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 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 in which surface irregularities, abrupt changes in section and internal stresses are almost always present.

In presenting such fatigue data, the metallurgist may be likened to the seed catalog artist who paints the beautiful flower pictures that so arouse our gardening instincts every spring. The engineer is the gardener who hopefully plants the seeds and who reaps the disappointments of his labors. The failure of the garden is usually not the fault of the seeds. The catalog pictures are intended to show what can be accomplished under favorable conditions and the artist should not be blamed for the poor garden soil, frost, drouth and the scores of other things that make the average garden look so unlike the catalog pictures. Similar disappointments are met when we attempt to apply the laboratory data on ideal specimens to the design of machine parts, more particularly for machines in which efficient use must be made of material under dynamic loading.

Economic Considerations

In the design of machines and equipment for heavy duty, 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 the 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.

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 *(a)* that stress can be determined, *(b)* that laboratory test specimens are representative of a material when that material is formed into a machine part, *(c)* that the amount and nature of the applied load is known, *(d)* that load variations occur in an orderly and predictable manner, *(e)* that representative fatigue curves can be constructed from a dozen or less specimens, and *(f)* that 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 residual stresses that have been discussed in previous articles in this paper (*Metal Progress* for February 1943, page 209, and May, page 737), 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 until we generously temper our calculated results with experience. For example, crankshafts may be stressed by the usual methods of calculation to 20,000 psi., connecting rods are stressed to 40,000, valve springs to 90,000 psi., disk clutch springs to 180,000 psi., while another form of disk 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 disk 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.

There is a growing interest in various devices employed to make direct measurement of stress such as photo-elasticity, brittle lacquers, extensometers, and similar instruments 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 of identical material, heat treatment and dimensions will show identical stress when measured by photo-elasticity or by extensometer, yet these specimens may vary widely in fatigue strength depending upon minute differences in surface finish or residual stresses. Since residual stresses are often desirable and are frequently unavoidable due to processing operations such as machining, heat treating, straightening or grinding, 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 photo-elastic tests since the specimens for this test must be free from residual stresses and must be made of another material.

Photo-elastic 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, [illegible]ally

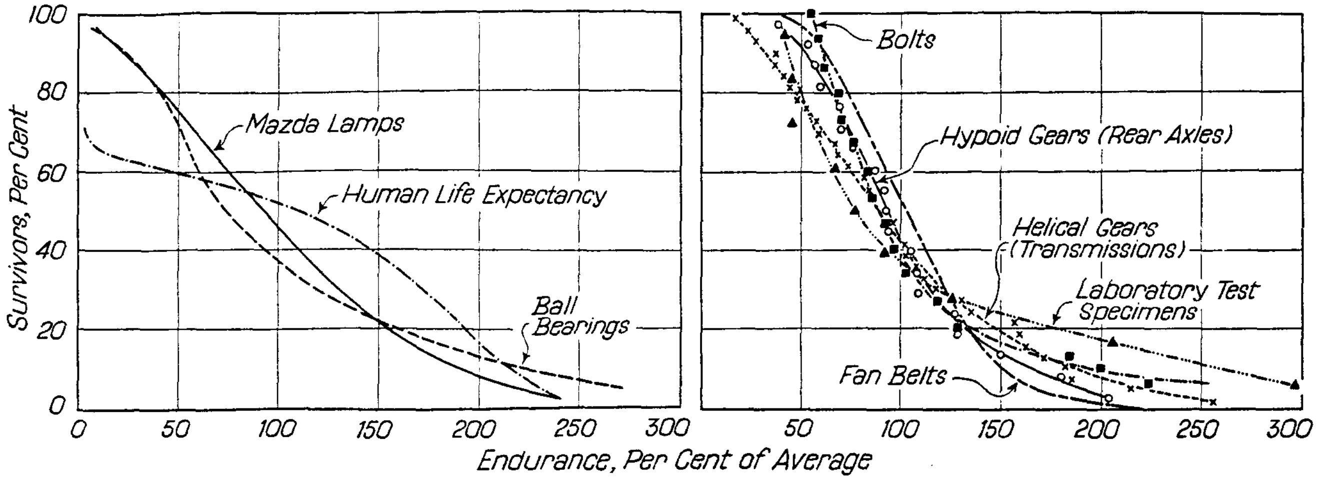


Fig. 1 and 2 — Life Expectancy Charts of Various Machine Parts Have a General Similarity, and May Be Regarded as Examples of Mortality Data

at the surface, even though it be sub-microscopic in size. Obviously, we cannot expect agreement.

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 residual stresses are often present as a result of processing or because of local heating as from bearing friction, and surfaces are subject to bruises and to corrosion of various kinds. These effects cannot be evaluated in terms of controlled stress raisers in test 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 way. Only a small percentage of the total number of stress cycles is 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?

This apparently hopeless situation is not, however, as hopeless as it seems. 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 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 which are now locked in our files, insofar as establishing a basis for evaluating material, design, or process for future use 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 persons as to expect reliable comparisons 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 Sjvall in an article on "Endurance Test Data and Their Interpretation" (*Mechanical Engineering*, Vol. 45, 1923, p. 579) 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. 1 in which the ordinate is the percentage of units surviving and the abscissa is durability in per cent of average life.

Figure 2 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, 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 in Fig. 1 and 2 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. 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 span from the best to the poorest sample was about 15 to one. These test results show that the childhood mortality is much higher in automobile transmissions than it is in automobile rear axle gears.

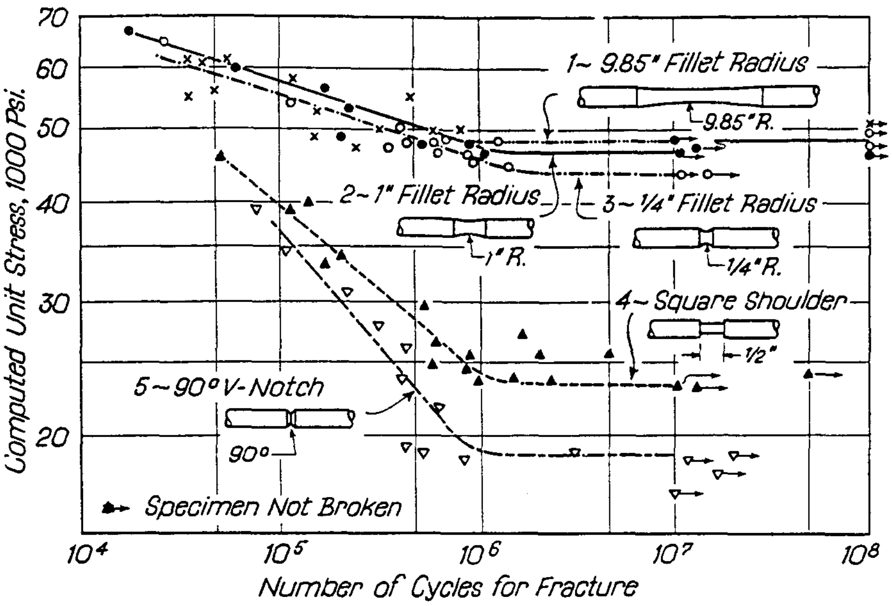


Fig. 3 — Moore & Kommers' Tests on Shape of Notch. 0.40-in. rounds of 0.49% carbon steel were water quenched and drawn at 1200° F. Diameter of reduced section is 0.275 in. in all specimens

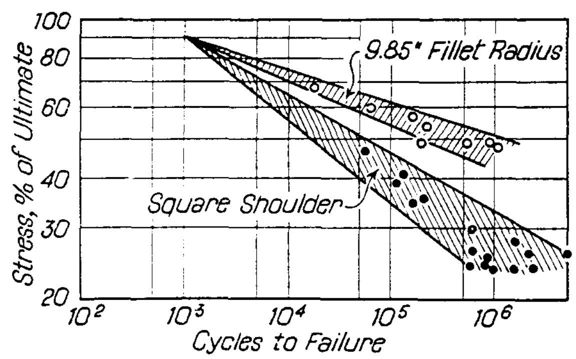


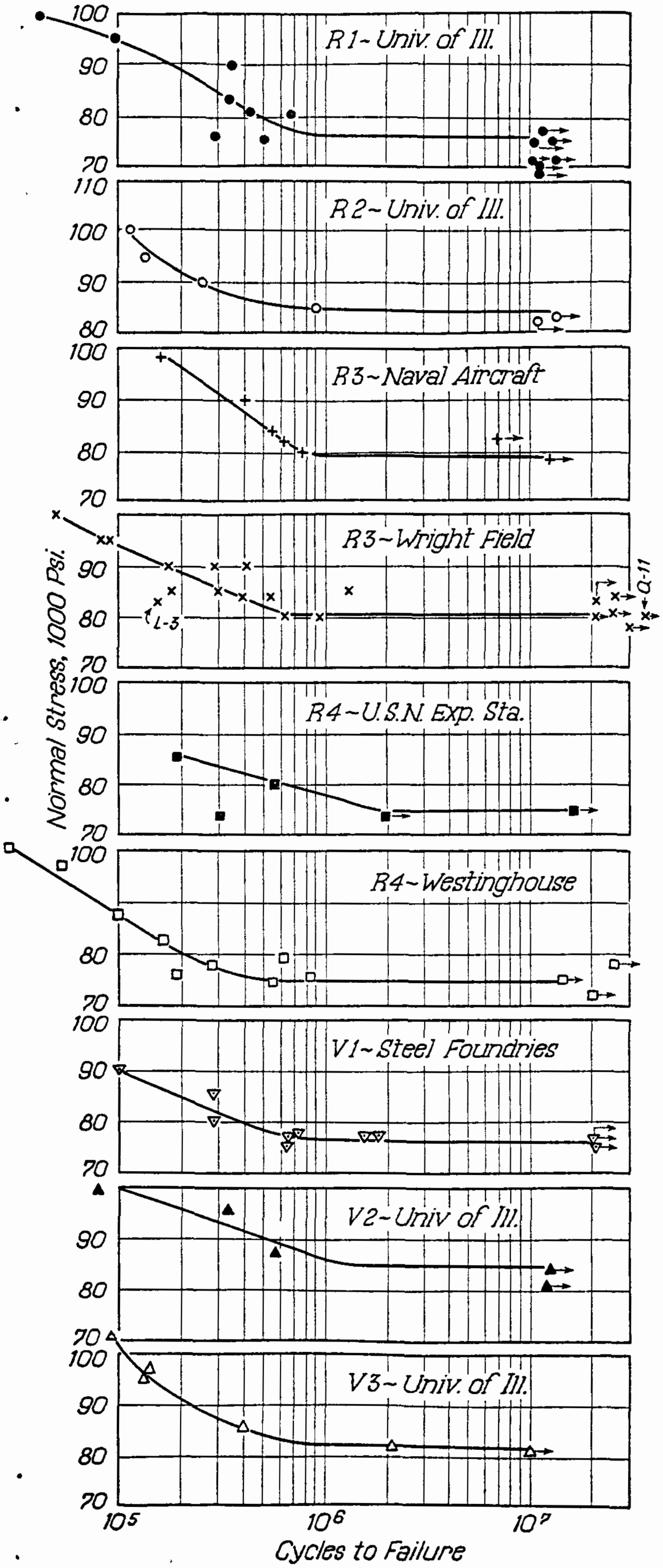
Fig. 4 — Data of Fig. 3, for Long Radius Fillet and for Square Shoulder, Plotted as Scatter Bands, Intersecting at a Point Near the Ultimate Tensile Strength of the Material (90,000 Psi.)

The life span ratios given above should not 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, as normally cut, they have less tolerance for warpage and deflection errors than rear axle gears and they are, therefore, subjected to greater variations of stress concentration.

The percentage variation in life of machine parts will also change as the test load is changed. When tests are conducted at high load 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. The reason for this variable will become clear when we examine the form of the scatter band of adequate fatigue 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, 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

Fig. 5 — Selection of Data From an A.S.T.M. Cooperative Research, Wherein Several Laboratory Workers Determined the S-N Diagram of a Given Steel



no interest in data at any stress except the stress at which the specimen will endure indefinitely.

In practical testing of machine parts, it should be obvious that comparisons of material, design, or processing 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 S-N curve.

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-logarithmic 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-log charts the points tend to lie on a straight line instead of on a curved line, as is usual when they are plotted on semi-log charts.

Figure 3 shows a series of fatigue specimens used by Moore and Kommers (and described in their book on "Fatigue of Metals", published by McGraw-Hill Book Co.) to determine the effect on fatigue of varying degrees of stress concentration. The resulting fatigue curves, plotted on logarithmic coordinates, are also shown. The authors compare these specimens on the basis of calculated stress at the fatigue endurance limit; that is, the stress

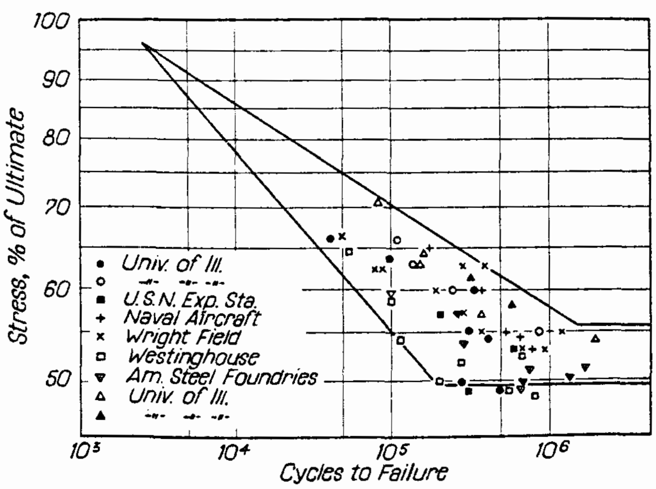


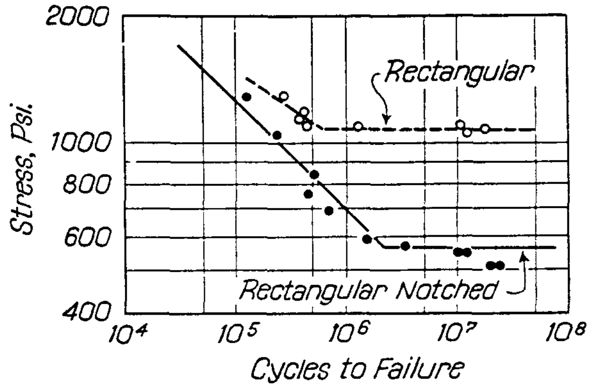
Fig. 6 — Re-Plot of Points in Graphs at Left to Show Scatter of Results at Loads Above Endurance Limits

at the knee where the curve becomes horizontal. However, as stated 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.

Fatigue curves of machine parts, no matter how well they are 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% to be obtainable.

The difference in slope of 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, and automobile transmission and rear axle gears.

Fig. 8 — Findley Shows That Endurance Test Specimens of Cellulose Acetate Exhibit Same Tendencies as Steel



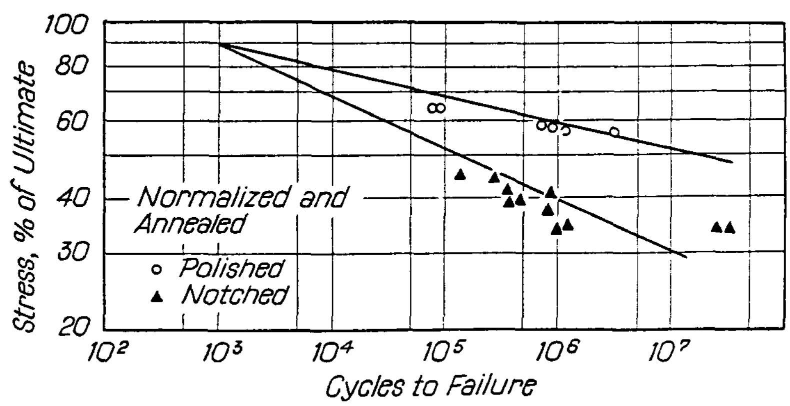


Fig. 7 — Oberg and Johnson's Tests (on S.A.E. 4134 Aircraft Steel, Normalized and Annealed) Designed to Show Influence of Notch

The lines plotted in Fig. 3 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.

Scatter of Test Points

The scatter of test points is due to unavoidable differences in test specimens no matter how carefully they are made, and 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 of the specimens. This is for the same reason that the average slopes for the rotating beam specimens shown in Fig. 3 increase with the severity of the stress concentration due to shape of notch.

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. The distribution of test points within this scatter band at any constant load will be of the order shown in Fig. 2.

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. 3 would all tend to converge toward a point in the vicinity of 1000 cycles and 90,000 psi., somewhat as indicated in Fig. 4. In this chart the fatigue slopes of specimens 1 and 4 of Fig. 3 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 withstand approximately the same stress 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 through yield, thus reducing the influence of local highly stressed points. For very brittle material, the intersection point of the fatigue c[illegible] for t[illegible] type of specimens shown in Fig. 3 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.

Apparent Disagreement of Test Data

The research committee on Fatigue of Metals of the American Society for Testing Materials recently sponsored a cooperative test program in which several laboratories conducted independent fatigue tests on identical material (heat treated S.A.E. 4340) under similar test conditions. The results were reported in an A.S.T.M. research

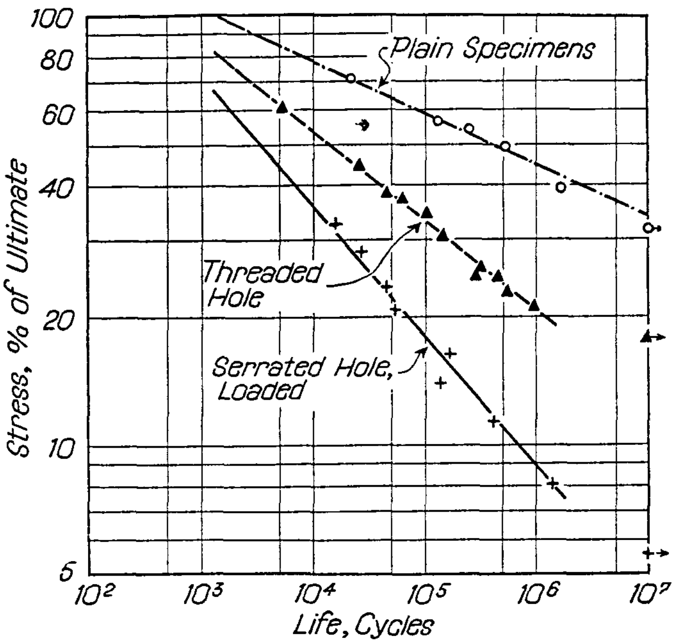


Fig. 9 — Fatigue Tests on Duralumin, Re-Plotted on Log-Log Scale

report presented at the June 1941 annual meeting, and from this the group of plots shown in Fig. 5 is 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. 6, we begin to see a semblance of order. All of the points lie within a scatter band of the same converging form that is shown in Fig. 4.

In [illegible]ing, it is interesting to note that in Fig. 5 we find 9 test points at 85,000 psi. load which are the points plotted in the life expectancy curve, Fig. 2, to show 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 unavoidable variations

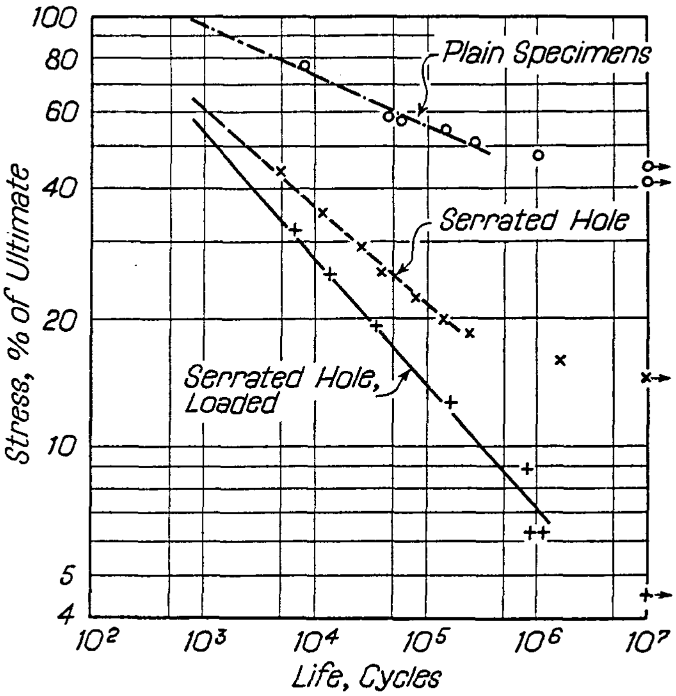


Fig. 10 — Similar Results From Appropriate Tests on "Electron", a Strong Magnesium Alloy

between "identical" specimens. As previously stated, this tendency toward convergence is seldom apparent in the published curves because the investigators have plotted their data on linear ordinates and logarithmic abscissas and always there are insufficient test points.

A few other diagrams will now be given, replotted from published data, to the same scale as Fig. 4, 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. (Note that this is the reciprocal of the slope as ordinarily used in engineering but it is a more convenient form for our purpose.)

T. T. Oberg and J. B. Johnson report in the *Proceedings* of the American Society for Testing Materials, Vol. 37, Part 2, 1937, page 199, a comparison between polished and notched specimens.

This is shown in Fig. 7 with results similar to the experiments by Moore and Kommers shown in Fig. 3.

Surface treatment of the test specimens other than the degree of smoothness has a marked effect on fatigue strength. In a paper on "Increasing the Fatigue Strength of Press-Fitted Axle Assemblies by Surface Rolling (*Transactions*, American Society of Mechanical Engineering, Vol. 58, 1936, p. A-91), O. J. Horger and J. L. Maulbetsch compared normal well-finished specimens with specimens that had been subjected to a rolling operation which introduced compressive stresses in the surface layer. The rolled specimens showed considerably better endurance, as shown in Fig. 3 of the first part of this paper (February *Metal Progress*, page 212).

Since the rolled specimens were pre-stressed in compression the subsequent tension stresses during the test were reduced as is shown in Fig. 6 of the February article, page 213; 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.

Tendency Persists in Other Materials

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.

W. M. Findley, in *Modern Plastics*, Vol. 19, 1941, page 57, described fatigue tests on cellulose acetate specimens with the replotted results shown in Fig. 8. As is usual there are not as many test points as are needed to define the region of convergence but the trend is definite.

Fatigue tests on duralumin, Fig. 9, and magnesium, Fig. 10, in plain and various notched specimens were reported by H. Burnheim in *Luftfahrtforschung*, Vol. 19, No. 2-3, March 1941. These re-plotted charts are more satisfactory than most published data in that the region of convergence is more clearly defined.

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 that the 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. In an article in next month's *Metal Progress* it will be shown how these ideas can be utilized in practice.