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AD-A024 785

AN EMPIRICAL DESIGN PROCEDURE FOR SHAFTS WITH FATIGUE LOADINGS

DARCOM INTERN TRAINING CENTER

APRIL 1976

149211

ADA024785

Report DARCOM-ITC-02-08-76-017

An Empirical Design Procedure for Shafts with Fatigue Loadings

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April 1976

Final Report

Prepared for:

MAINTENANCE EFFECTIVENESS ENGINEERING GRADUATE PROGRAM AND
TEXAS A&M UNIVERSITY GRADUATE CENTER
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Red River Army Depot
Texarkana, Texas 75501

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER
DARCOM-ITC-02-08-75-017

2. GOVT ACCESSION NO.

3. RECIPIENT'S CATALOG NUMBER

4. TITLE (and Subtitle)
AN EMPIRICAL DESIGN PROCEDURE FOR SHAFTS WITH FATIGUE LOADINGS

5. TYPE OF REPORT & PERIOD COVERED
Final

6. PERFORMING ORG. REPORT NUMBER

7. AUTHOR(s)
James D. Grounds

8. CONTRACT OR GRANT NUMBER(s)

9. PERFORMING ORGANIZATION NAME AND ADDRESS
Dept of Maintenance Effectiveness Eng
DARCOM Intern Training Center
RRAD, Texarkana, Texas 75501

10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS

11. CONTROLLING OFFICE NAME AND ADDRESS
DARCOM Intern Training Center - USAMC
Red River Army Depot
Texarkana, Texas 75501

12. REPORT DATE
April 1976

13. NUMBER OF PAGES
40

14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)

15. SECURITY CLASS. (of this report)

15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for Public Release: DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

Approved for Public Release: DISTRIBUTION UNLIMITED

18. SUPPLEMENTARY NOTES
Research performed by James D. Grounds under the supervision of Dr. Roger J. McNichols, Industrial Engineering Department, Texas A&M University

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)
Shaft Design
Fatigue
Combined Loads
Failure Prediction

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report presents a method of designing rotating shafts subjected to combined fatigue loads. In the past, the fatigue strength, a material property necessary for design, has been determined for axial, bending, and torsional loads independently. A general approach to design with combined loads requires a fatigue strength that does not depend on the type of loading. Measured

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

20.

differences in the fatigue strengths are attributed to the size of a shaft. By developing correction factors to be used on the applied bending and torsional loads, the axial fatigue strength can be used for combined loads. The octahedral shear theory and the von Mises-Hencky failure criteria are used in describing a design procedure for solid steel shafts. This approach should be verified experimentally.

FOREWORD

The research discussed in this report was accomplished as part of the Maintainability Engineering Graduate Program conducted jointly by the DARCOM Intern Training Center and Texas A&M University. As such, the ideas, concepts and results herein presented are those of the author and do not necessarily reflect approval or acceptance by the Army.

This report has been received and is approved for release. For further information on this project, contact Dr. Ronald C. Higgins, Chief of Maintenance Effectiveness Engineering, Red River Army Depot, Texarkana, Texas.

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This report presents a method of designing rotating shafts subjected to combined fatigue loads. In the past, the fatigue strength, a material property necessary for design, has been determined for axial, bending, and torsional loads independently. A general approach to design with combined loads requires a fatigue strength that does not depend on the type of loading. Measured differences in the fatigue strengths are attributed to the size of a shaft. By developing correction factors to be used on the applied bending and torsional loads, the axial fatigue strength can be used for combined loads. The octahedral shear theory and the von Mises-Hencky failure criteria are used in describing a design procedure for solid steel shafts. This approach should be verified experimentally.

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NOMENCLATURE

A,B,C
D,E,F = coefficients for the octahedral shear theory

d = diameter

r = radius

k = conventional size factor

K = revised size factor

P,M,T = applied axial, bending, and torsional loads

σ = applied stress

S_a = uniaxial alternating stress

S_m = uniaxial mean stress

S_e = fatigue strength for alternating stresses

S_u = nominal static ultimate tensile strength

Subscripts:

tc = tension and compression (axial)

b = bending

τ = torsion

X,Y,Z = reference stress directions

a = alternating

m = mean

CHAPTER I

INTRODUCTION

Fatigue in rotating shafts is a phenomenon that has been known and studied for nearly a century. Laboratory tests and research have developed several theories of fatigue failure and analytical models. Fatigue is a complex problem that must be considered by the designer of a shaft. The tradeoff between accuracy and simplicity is perhaps the hardest decision to make.

Field data on typical shaft designs would definitely be the most accurate, but the long lifetimes expected of shafts makes this impractical. Compiling historical data on a wide range of shafts is not feasible because of the large number of parameters involved. Designers have had to resort to simple methods and large factors of safety. "Improvements in reliability can only result from the availability of additional statistical data and a clearer understanding of what can logically be deduced from them. Research in this area is as necessary as research to improve design methods and fatigue performance."(3)*

Laboratory testing is used to determine selected material properties and more recently has been used to evaluate specific shafts. Material properties are measured

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* Numbers in parentheses refer to the list of references at the end of this report.

in conjunction with a failure theory. The octahedral shear theory is the most widely accepted failure theory. The applied loads are transformed into an equivalent uniaxial load (stress). For a completely reversing (alternating) load, failure occurs at the 'fatigue strength' of the material.

The selection of standard test conditions is necessary to reduce the large number of factors affecting the fatigue strength. The fatigue strength is dependent on the type of loading (axial, bending, or torsion) and that has resulted in three separate fatigue strengths being defined. The required time and expense of testing hinders the development of fatigue strengths for various loading combinations. Even simulating the operating conditions for only one particular shaft would be prohibitively expensive.

For many circumstances a less accurate but faster analytical approach using known material properties would be appropriate. The large number of assumptions and simplifications generally taken make design practices inconsistent, complicated, and controversial. This report presents an empirical method of designing shafts. By assuming a relationship between the three fatigue strengths, the octahedral shear theory can be used for combined fatigue loadings. The relationship is implemented by applying correction factors to the applied bending and torsion loads. The design procedure is described in

Chapter III and the correction factors are developed in Appendixes A and B. Appendix C shows a comparison of this method to test data. Appendix D is an example design problem. The contents of the reference material used is discussed in Chapter II.

CHAPTER II

LITERATURE SURVEY

A general method of evaluating fatigue in shafts with combined loadings was not found in the literature. The most complete and direct approach to fatigue design was found in Fatigue Design of Machine Components[9] by L. Sors. Common practices and data were shown and only a single loading could be handled. The text, Mechanical Behavior of Engineering Materials,[6] shows the classical fatigue failure theories.

An article[1] by H. A. Borchardt covers the design of three special cases of combined loads. The developed equations do not resemble an accepted failure theory and any modifications would be difficult. A report from the Defense Documentation Center[2] discusses fatigue failure theories for shafts. The former ASME Code for the design of shafts is presented, but this was withdrawn in 1955.[1] The octahedral shear theory is suggested as being the most generally applicable in evaluating the effect of combined stresses, but no design procedure is given.

Texts by R. B. Hopkins[4] and Carl C. Osgood[8] and the Metals Handbook[7] discuss in depth the factors affecting fatigue in metals. Test results are given for a small number of selected materials showing the influence of various environmental factors. Lengthy qualitative

information is also added. These references did not propose a design method for combined loads. Axial, bending, and torsional tests are made with standard .3 inch diameter smooth specimens. Additional information is usually gained by varying one parameter, such as temperature, diameter, or corrosion, at a time.

Data being generated by Dr. Kececioglu(5) describes failure by imposing combined loads upon a particular test specimen. The octahedral shear theory is used in providing distributional static and fatigue strength data. Unfortunately, these values are good for only very particular circumstances; one shaft size, one material, one combination of loadings, and one set of environmental conditions. The data would have to be estimated for a given case in order to use existing equations without testing. It will be some time, if ever, before enough data is available to do this.

H. von Philipp(9) presented a thesis in 1941 that attributes the difference between the axial and bending fatigue strengths to the size of the shaft. Since fatigue is initially a local surface failure, it is intuitive that the bending value be nearly equal to the axial value. The bending fatigue strength decreases with increasing diameter and, according to von Philipp, approaches the axial fatigue strength. The axial fatigue strength does not change appreciably with size.

With similar reasoning, von Philipp predicted that the torsional fatigue strength approaches a true shear fatigue strength as the shaft diameter increases. The octahedral shear theory assumes the ratio between the axial fatigue strength and the true shear fatigue strength to be the square root of three. This report proposes that by applying size correction factors to the applied stresses, instead of the fatigue strengths, it is possible to describe failure with the axial fatigue strength for each of the three loading conditions. This approach is supported by theories which compare actual stress values to those calculated by classical stress equations.(6)(9) For example, in bending the stresses on the shaft's surface are less than those calculated by the classical bending equation. This inflates the measured fatigue strength value.

The result is that the axial fatigue strength, which is independent of size, can be used for combined loads in the failure theory. The correction factors developed in Appendixes A and B are placed upon the applied bending and torsion stress values. Chapter III is a suggested design procedure.

CHAPTER III

DESIGN PROCEDURE

In setting a design procedure it is first necessary to outline the circumstances in which the procedure can be used. An objective of this report is to make the procedure as easy to use as possible. The limitations to be imposed are necessary for the use of the given equations and graphs. By deviating very little from the accepted theory it is hoped that this approach can be used for other classes of problems.

Only solid circular steel and steel alloy shafts are considered. In steel shafts with relatively long lifetimes the fatigue strength approaches a value known as the endurance limit, as shown in Figure 1. If a value other than the endurance limit is used, or a material other than steel is used, the graphs may need to be revised. The largest expected value of each mean and alternating load must necessarily be used. For cases where this is too conservative, an approach such as Miner's linear accumulation theory(2)(6)(9) or Harris and Lipson's cumulative damage relation(2) may possibly be incorporated.

The procedure is broken into eight basic steps. This is only a suggested design procedure for fatigue in shafts and has not been evaluated experimentally. Attention should be given to the conclusions of this report.

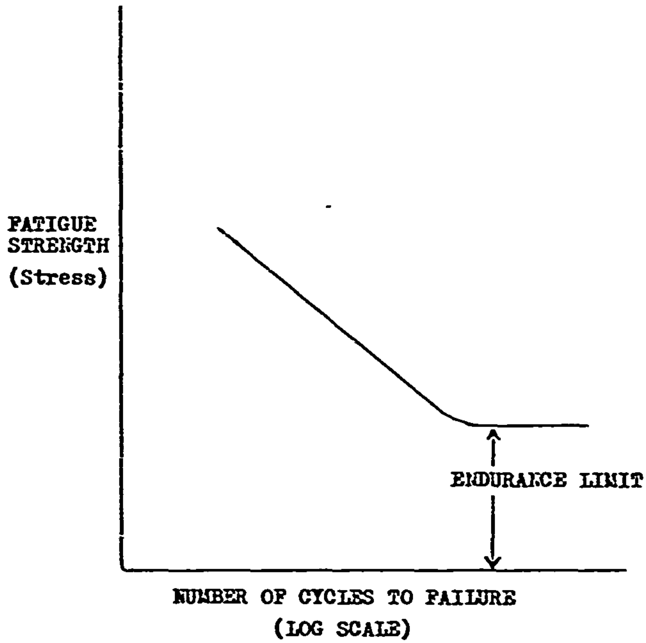


FIGURE 1. FATIGUE STRENGTH vs CYCLES TO FAILURE

STEP 1. A specific shaft design must be proposed. If necessary, this procedure can be iterated until an acceptable shaft is found. The critical sections of the shaft must be analyzed separately. The diameter at a section, the standard axial fatigue strength (Se_{tc}), and the nominal ultimate tensile strength (S_u) of the material must be specified.

STEP 2. The applied loads on the shaft must be known. These are in the form of axial (P), bending (M), and torsion (T) loads at the section to be analyzed. Figure 2 illustrates the mean and alternating values of these loads. Figure 3 shows the relationship of the loads to the shaft.

STEP 3. The appropriate stresses must be calculated. These stresses are depicted in Figure 3.

Axial	Bending	Torsion
$\sigma_{tc_m} = \dfrac{4P_m}{\pi d^2}$	$\sigma_{b_m} = \dfrac{32M_m}{\pi d^3}$	$\sigma_{XY_m} = \dfrac{16T_m}{\pi d^3}$
$\sigma_{tc_a} = \dfrac{4P_a}{\pi d^2}$	$\sigma_{b_a} = \dfrac{32M_a}{\pi d^3}$	$\sigma_{XY_a} = \dfrac{16T_a}{\pi d^3}$

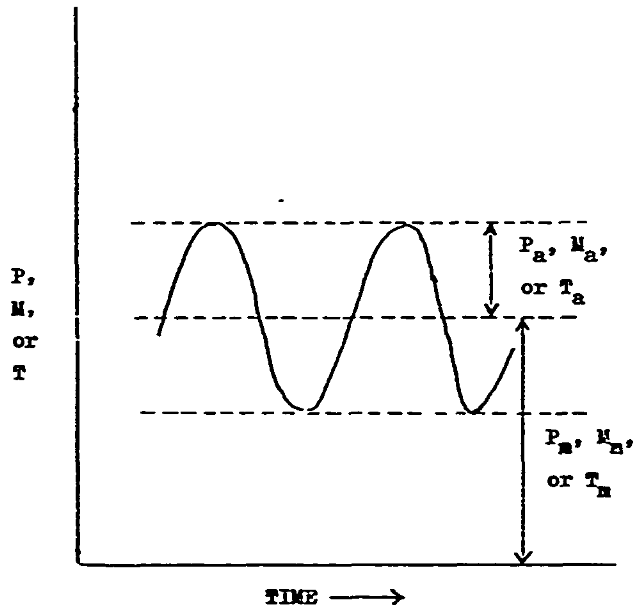
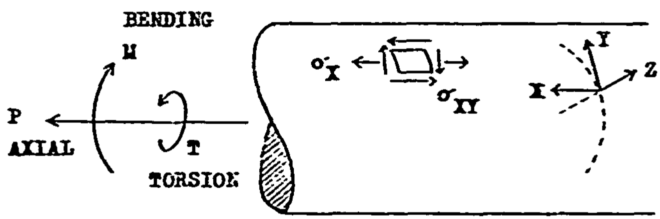


FIGURE 2. MEAN AND ALTERNATING LOADS



$$\sigma_X = \sigma_{tc} + \sigma_b$$

FIGURE 3. LOADS AND STRESSES ON A SHAFT

STEP 4. An adequate factor of safety must be determined by the designer. The use of the shaft, the accuracy of the input information, and the complexity of the loading must be considered. Since a subjective decision must be made, the designer should consult as many references as possible.

STEP 5. Standard material values are used for design and corrections must be made for stress concentrations, temperature, and surface condition. This is done by forming a coefficient for each applied stress.

$A\sigma_{tc_a}$ $B\sigma_{b_a}$ $C\sigma_{XY_a}$ $D\sigma_{tc_m}$ $E\sigma_{b_m}$ $F\sigma_{XY_m}$

Factors which are normally used on the applied stress, such as stress concentration, are numerators in the coefficients. Factors which are conventionally used on the material properties, such as temperature and surface condition, are denominators in the coefficients. For example, assume that for a particular shaft the stress concentration factor for alternating bending is 1.5. The elevated operating temperature is known to increase the bending fatigue strength of a standard specimen by 10%. A machined surface with no corrosion reduces the bending fatigue strength by 30%. $B = 1.5 / (1.10 \times .70) = 1.95$

STEP 6. The size correction factors for bending, K_b, and torsion, K_τ, must be determined from Figures 4 and 5. These factors are fairly conservative and can be revised if enough material information is available, see Appendixes A and B.

STEP 7. The octahedral shear theory uses the combined loadings to create equivalent uniaxial stresses, S_a and S_m. The required input comes from steps 3, 5, and 6.

$$(S_a)^2 = (A\sigma_{tc_a} + BK_b\sigma_{b_a})^2 + 3(CK_\tau\sigma_{XY_a})^2$$

$$(S_m)^2 = (D\sigma_{tc_m} + EK_b\sigma_{b_m})^2 + 3(FK_\tau\sigma_{XY_m})^2$$

STEP 8. Failure is defined by the von Mises-Hencky ellipse. The ellipse can be plotted for a visual representation of the stresses. If the following equation is satisfied, the shaft is adequately designed to prevent fatigue failure for the specified operating conditions.

$$\left(\frac{\text{factor of safety} \times S_a}{S_{e_{tc}}}\right)^2 + \left(\frac{\text{factor of safety} \times S_m}{S_u}\right)^2 < 1$$

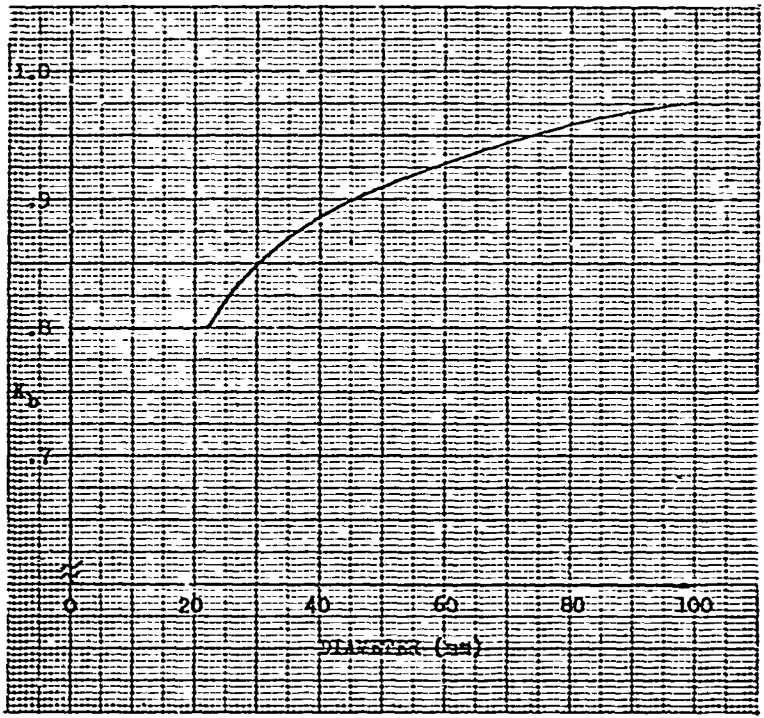


FIGURE 4. CONSERVATIVE BENDING STRESS CORRECTION FACTOR

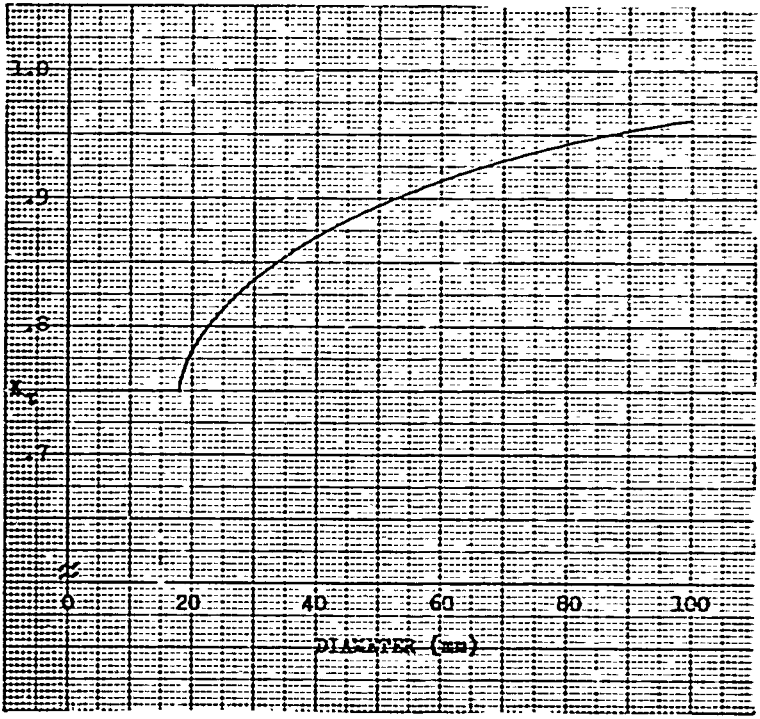


FIGURE 5. CONSERVATIVE TORSIONAL STRESS CORRECTION FACTOR

CHAPTER IV

CONCLUSIONS AND RECOMMENDATIONS

An empirical design procedure for shafts with fatigue loadings is presented in this report. A shaft with combined fatigue loadings can be evaluated with this procedure. Many design considerations are made in the selection or evaluation of a shaft. In some circumstances only a few standard shaft sizes are available to be picked from. Other times a requirement for small deflections, protection from a single fail-safe load, or other conditions are critical in selecting shaft size. If a shaft design shows inadequate fatigue characteristics there are things that can be done besides increasing the shaft size. Fatigue problems could be solved by various surface treatments, reducing stress concentrations, or changing the type of steel.

Most shafts have unique features in their design. Suggesting a very tightly constrained approach would not fill the objectives of this report. It is the designer's responsibility to select the input to the design. This method makes use of the work previously done on fatigue.

Tests should be designed to evaluate this procedure. Improvements in fatigue design for combined loads is necessary. Guidelines for testing and listing test results need to be established.

APPENDIX A

DEVELOPMENT OF THE BENDING CORRECTION FACTOR, K_b

For practical reasons the standard test specimen has a small diameter, .3 inch. Tests have shown that the axial fatigue strength does not change appreciably with size.[4] The bending fatigue strength decreases for larger shafts and has been shown to approach the axial fatigue strength.[9] Figure 6 shows this theory, but few large shafts have been tested.

The actual stress on the surface of the specimen in bending is believed to be less than that calculated by the bending equation, refer to Figure 7. As the diameter is increased, the true stress value approaches the calculated value. Conventionally, a size correction factor is applied to the bending fatigue strength. By assuming the bending fatigue strength approaches the axial fatigue strength, a size correction factor can be used on the applied bending stress. The axial fatigue strength could then be used for axial, bending, and combined loads.

$\sigma_b = k_b S_{e_b}$ k_b is the conventional factor

$K_b \sigma_b = S_{e_{tc}}$ K_b is the proposed factor

Solving for K_b: $$K_b = \frac{S_{e_{tc}}}{k_b S_{e_b}}$$

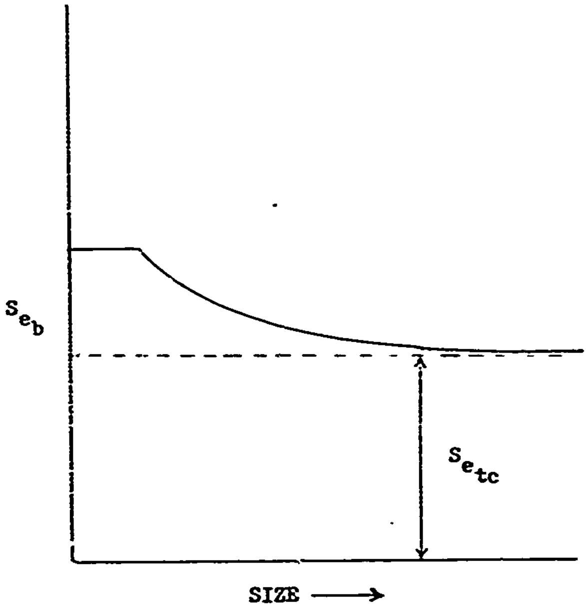


FIGURE 6. BENDING FATIGUE STRENGTH vs DIAMETER

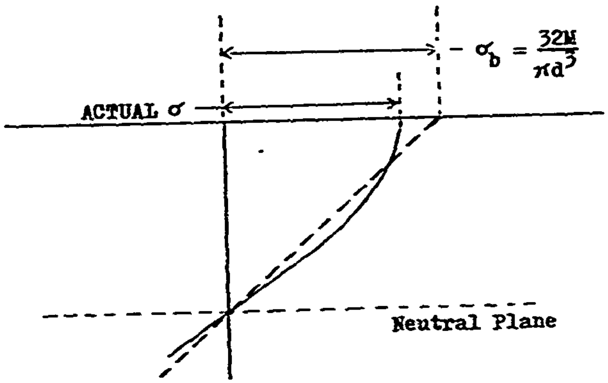


FIGURE 7. BENDING STRESS

To be conservative, K_b should be larger than its correct value. For large shafts, k_b approaches Se_{tc}/Se_b and K_b should approach one. For standard test specimens, k_b equals one and the ratio, Se_{tc}/Se_b, is between .65 and .78 for most steels.[9] von Philipp[9] predicted the ratio to be .59. Because of the inverse nature of the two correction factors, Figure 8, the von Philipp line is not conservative for K_b. What is conservative in reducing the applied bending stress is not conservative in reducing the bending fatigue strength. If the correct ratio, Se_{tc}/Se_b, is known for a particular material, a line proportional to the sequel of the von Philipp line could be constructed. For use in this report, Figure 4, a conservative ratio, .8, is used.

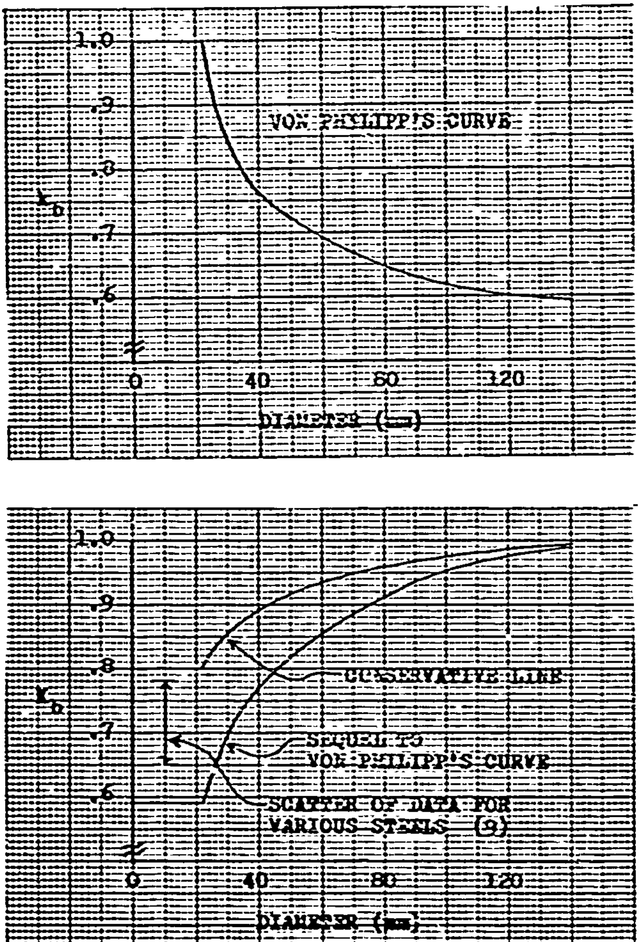


FIGURE 8. SIZE CORRECTION FACTORS FOR BENDING

APPENDIX B

DEVELOPMENT OF THE TORSION CORRECTION FACTOR, K_τ

Fatigue occurs at a lower stress for shear stresses than for axial stresses. According to the octahedral shear theory, the ratio between the normal and shear fatigue strengths is equal to $\sqrt{3}$. It has also been shown that the torsional fatigue strength decreases with increasing size, as did the bending fatigue strength. Using the same reasoning as in Appendix A, it is proposed that $Se_\tau\sqrt{3}$ approaches Se_{tc} as the shaft size increases.

Von Philipp determined that the true shear fatigue strength should be three quarters of the value obtained in the standard test,[9] see Figure 9. The conventional size factor, k_τ, is applied to the fatigue strength measured in the standard test.

$$\sigma_{XY} = k_\tau S_{e_\tau}$$

To use the octahedral shear theory for combined stresses, it is necessary to relate torsion to the axial fatigue strength.

$$\sqrt{3}(K_\tau \sigma_{XY}) = S_{e_{tc}}$$

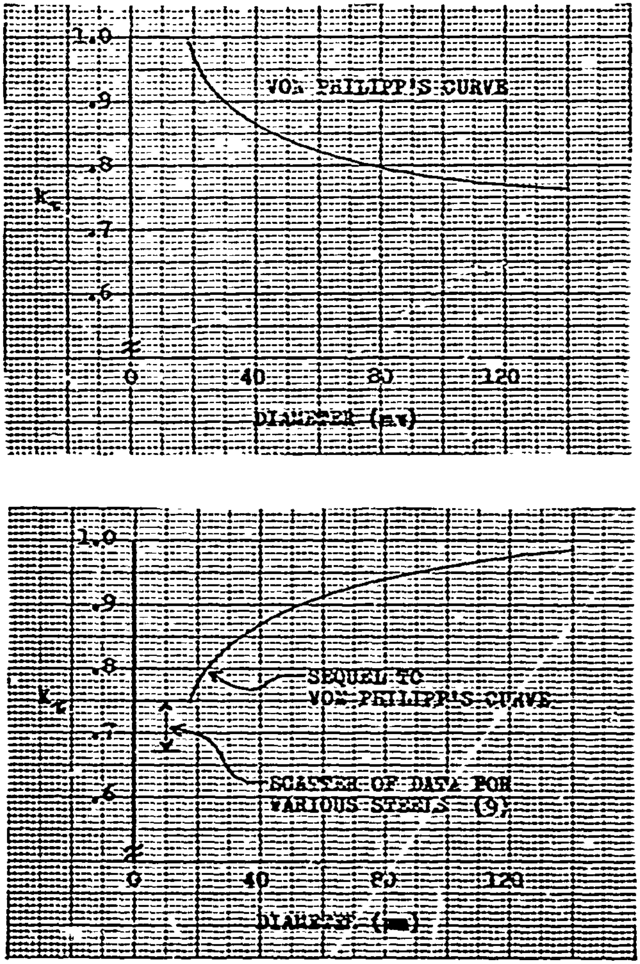


FIGURE 9. SIZE CORRECTION FACTORS
FOR TORSION

The proposed correction factor, K_τ, can be found by equating the previous two equations.

$$K_\tau = \frac{S_{e_{tc}}}{k_\tau S_{e_\tau}\sqrt{3}}$$

For large shafts, K_τ should approach one. For the standard shaft, k_τ is equal to one and the ratio, Se_{tc}/Se_τ, is between 1.16 and 1.30 for most steels.(9) Substituting this into the last equation results in K_τ values from .67 to .75. K_τ must be larger than its actual value to be conservative. Therefore, the sequel to von Philipp's line, Figure 9, is reasonably conservative. What is conservative for K_τ should not be conservative for k_τ. Experiments have shown that the von Philipp curve for the conventional torsion factor is not conservative.(9) If the correct ratio, Se_{tc}/Se_τ, is known for the standard specimen of a particular material, a line proportional to the sequel of the von Philipp line can be plotted.

This development has assumed that the ratio, $\sqrt{3}$, from the octahedral shear theory is correct. The existing data seems to fit this theory well, but much more data is needed on larger shafts and shafts with combined loads.

APPENDIX C

COMPARISON TO TEST DATA

Tests conducted at the University of Arizona Reliability Research Laboratory provide information about the loads that caused a particular shaft design to break.(5) These loads can be used to see how closely this design procedure predicts failure. Grooved AISI 4340 steel shafts were subjected to alternating bending and steady torsion loads. Ungrooved specimens were tested to obtain the ultimate tensile strength of the material. The shaft dimensions are shown in Figure 10.

The standard bending fatigue strength is fifty percent of the ultimate tensile strength.(4)(7) For most steels this value is between fourty and sixty percent. The axial fatigue strength is estimated to be seventy percent of the bending fatigue strength. Since only one alternating stress is present and the shaft size is small, any errors due to this estimate are eliminated by using .70 for K_b, see Appendix A. A more accurate value for Se_{tc} would be needed for multiple alternating loads.

$$S_u = 165 \text{ Kpsi}$$

$$Se_{tc} = 165 \times .50 \times .70 = 58 \text{ Kpsi}$$

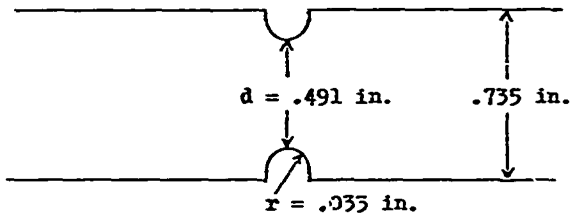


FIGURE 10. TEST SHAFT

Since only two stresses are present, alternating bending and steady torsion, only two stress coefficients need to be determined, B and F. The bending and torsion stress concentration factors are computed to be 2.04 and 1.58 respectively.(9) Figure 11 shows typical surface condition effects for alternating bending.(2) It is assumed that the groove had a ground surface, the fatigue strength being reduced to 88% of the standard test value. The surface condition is, in this case, assumed to have a negligible effect on the static tensile strength and the mean stresses do not require correction.

$$B = 2.04 \; / \; .88 = 2.32$$

$$F = 1.58$$

The torsion size correction factor, K_τ, is .75, from Figure 5. Five different loading conditions for failure were determined. The equivalent mean and alternating stress is calculated according to the octahedral shear theory.

$$S_a = BK_b\sigma_{b_a} = 2.32 \times .70 \times \sigma_{b_a}$$

$$S_m = \sqrt{3}FK_\tau\sigma_{XY_m} = 1.73 \times 1.58 \times .75 \times \sigma_{XY_m}$$

	σ_{b_a}	σ_{XY_m}	S_a	S_m
1	33.7	0	54.7	0
2	32.0	17.5	52.0	35.8
3	30.4	44.0	49.4	90.0
4	24.1	55.7	39.2	114.2
5	22.9	88.2	37.2	181.0

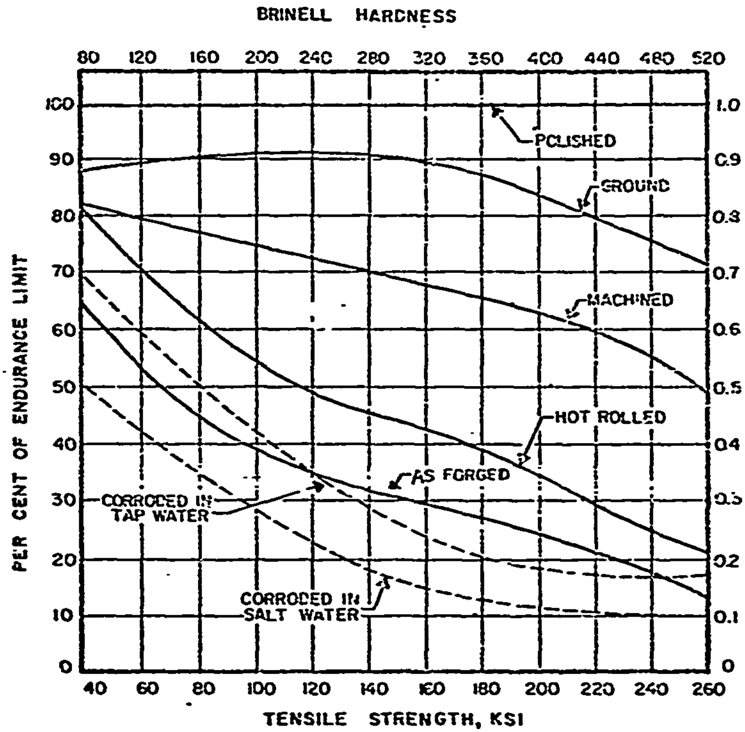


FIGURE 11. THE EFFECT OF SURFACE FINISH ON FATIGUE STRENGTH

Failure is predicted by the von Mises-Hencky ellipse.

$$\left(\frac{S_a}{S_{e_{tc}}}\right)^2 + \left(\frac{S_m}{S_u}\right)^2 = 1$$

The correlation to measured values, Figure 12, is good except for the last data point. This could be attributed to plastic stresses that violate the classical equations used for elastic stresses. By these results, this design approach closely approximates fatigue failure and should be conservative when large mean stresses are encountered. The ultimate tensile strength for a notched specimen was not used, even though it had been measured, because this would not generally be available to the designer without testing. These results must not be taken as defining this procedure's accuracy because only two types of loads were applied to a small shaft. The concept of using the axial fatigue strength and ultimate tensile strength from a standard test is emphasized.

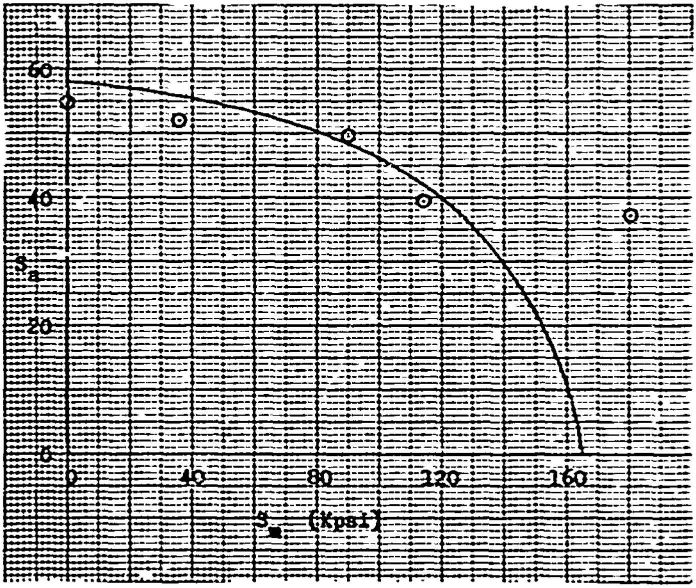


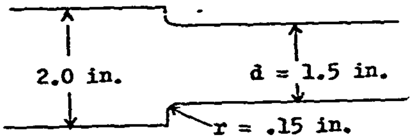
FIGURE 12. COMBINED BENDING AND TORSION
TEST RESULTS

APPENDIX D

DESIGN EXAMPLE

STEP 1. A small airplane propeller shaft, crankshaft, may be subjected to axial, bending, and torsional loads. The section to be analyzed is a step increase in diameter about six inches behind the propeller.

$S_{e_{tc}} = 54$ Kpsi $\qquad S_u = 170$ Kpsi



STEP 2. The axial load is caused by the thrust of the propeller. The bending loads are caused by the weight of the propeller and an allowable amount of difference between the two blades. The torsion loads are caused by the pulsations of the small motor.

$P_a = 0 \qquad P_m = 106$ lb

$M_a = 125$ in-lb $\qquad M_m = 143$ in-lb

$T_a = 1200$ in-lb $\qquad T_m = 1260$ in-lb

STEP 3. The stresses are calculated from the loads.

$$\sigma_{tc_a} = 0 \qquad \sigma_{tc_m} = 60 \text{ psi}$$

$$\sigma_{b_a} = 377 \text{ psi} \qquad \sigma_{b_m} = 430 \text{ psi}$$

$$\sigma_{XY_a} = 1800 \text{ psi} \qquad \sigma_{XY_m} = 1900 \text{ psi}$$

STEP 4. The factor of safety is chosen to be 1.9.

STEP 5. The stress coefficients must be estimated.

	Axial	Bending	Torsion
Stress Concentrations[9]	1.82	1.66	1.41

Possible water corrosion, from Figure 11, gives a factor of .22 for the alternating stresses. A factor of .90 is placed on all stresses for a 200°F temperature condition.[7]

$$B = 1.66 \,/\, (.22 \times .90) = 8.4$$

$$C = 1.41 \,/\, (.22 \times .90) = 7.1$$

$$D = 1.82 \,/\, .90 = 2.02$$

$$E = 1.66 \,/\, .90 = 1.85$$

$$F = 1.41 \,/\, .90 = 1.57$$

STEP 6. The size correction factors are taken from Figures 4 and 5.

$$K_b = .88 \qquad K_\tau = .87$$

STEP 7. The equivalent uniaxial mean and alternating stresses, S_m and S_a, are calculated.

$$(S_a)^2 = (8.4x.88x377)^2 + 3(7.1x.87x1800)^2$$

$$S_a = 19.4 \text{ Kpsi}$$

$$(S_m)^2 = (2.02x60 + 1.85x.88x430)^2 + 3(1.57x.87x1900)^2$$

$$S_m = 4.6 \text{ Kpsi}$$

STEP 8. The shaft can now be evaluated.

$$\left(\frac{1.9 \times 19.4}{54}\right)^2 + \left(\frac{1.9 \times 4.6}{170}\right)^2 = .47 < 1$$

The shaft is adequate for the specified conditions. By using conservative values in this procedure, a shaft can be easily checked for possible fatigue failure. A better representation of the stress condition can be obtained by plotting the ellipse. as in Figure 12. If the stresses are near the ellipse, more accurate material values and stress coefficients are recommended if they are available.

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