MANUAL ON FATIGUE TESTING

Prepared by
Committee E-9 on Fatigue
AMERICAN SOCIETY FOR TESTING MATERIALS
1949

Special Technical Publication No. 91

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SECTION I—INTRODUCTION[1]

Although it is nearly a century since August Wöhler started his classic fatigue tests, we see about us more fatigue testing than ever before. This is, of course, a consequence of the Machine Age in which we are living. New forms of transportation, new automatic production machinery, advances in prime movers such as the gas turbine, all demand better knowledge of materials.

In this connection, fatigue[2] of materials is of prime importance because it is a direct mechanism of failure. It has been estimated that over 80 per cent of machine failures are due to fatigue. In fact it was Wöhler's appointment to a commission for studying causes of railway wrecks which led to a study of failures of railway axles and in turn to fatigue testing.

As we see it, the most important objective of fatigue testing is to build up basic knowledge which will contribute to the design, construction and maintenance of mechanisms and structures in such a way that they are as free from failures as possible and at the same time are efficient and economical.

This Manual concerns itself with fatigue testing and not with fatigue of metals as such except for making some reference to the need for securing service data to correlate with laboratory tests. Test data and theories of failures are, therefore, outside the scope of the Manual, although a discussion of the limitations of fatigue tests is considered appropriate and important.

The purpose of the Manual is to supply information to those setting up new laboratory facilities, to aid in properly operating the equipment, and to offer advice in presentation and interpretation of the data. Some guidance is also given regarding books and references for further study. A further objective is the setting up of recommended practices which may later on be crystallized into standards.

The field covered by the Manual is largely that of so-called conventional fatigue tests of engineering materials. Service testing equipment and vibratory tables for testing completed apparatus such as radio transmitters and packaged items, came into expanded use during World War II. This type of testing, in so far as packaging is concerned, is in the scope of activity of A.S.T.M. Committee D-10 on Shipping Containers, especially Subcommittees II (Methods of Testing), IV (Performance Testing), and V (Correlation of Tests and Test Results).

In preparing the Manual, we have reviewed the following A.S.T.M. references which represent work in the direction of preferred practice in the conventional fatigue testing field:

1. "Present-Day Experimental Knowledge and Theories of Fatigue

[1] Drafted by R. E. Peterson, Manager, Mechanics Div., Westinghouse Research Labs., Westinghouse Electric Corp., East Pittsburgh, Pa.; Chairman, A.S.T.M. Committee E-9 on Fatigue. (Revised following discussion by A.S.T.M. Committee E-9.)

[2] The term fatigue, in the materials testing field, has, in at least one case, glass technology, been used for static tests of considerable duration, a type of test generally designated as stress-rupture. In this Manual, fatigue applies to failure under *repeated stress*. Although the usual concept is associated with a large number of cycles, there is no reason why the term fatigue should not be applied for a small number of cycles, if cracking and progressive failure occurs under such conditions.

Phenomena in Metals," Appendix to Report of Research Committee on Fatigue of Metals, *Proceedings*, Am. Soc. Testing Mats., Vol. 30, Part I, p. 260(1930).

2. "Note on Fatigue Tests on Rotating-Beam Testing Machines," Appendix to Report of Research Committee on Fatigue of Metals, *Proceedings*, Am. Soc. Testing Mats., Vol. 35, Part I, p. 113 (1935).
3. "Nomenclature for Various Ranges in Stress in Fatigue," Appendix to Report of Research Committee on Fatigue of Metals, *Proceedings*, Am. Soc. Testing Mats., Vol. 37, Part I, p. 159 (1937).
4. Tentative Methods of Test for Compression Fatigue of Vulcanized Rubber (D 623 - 41 T), 1949 Book of A.S.T.M. Standards, Part 6.
5. Tentative Method of Test for Repeated Flexural Stress (Fatigue) of Plastics (D 671 - 49 T), 1949 Book of A.S.T.M. Standards, Part 6.

This project was initiated at the A.S.T.M. Annual Meeting in Buffalo in 1946. While it has always been the intention that the Manual represent the combined experience of Committee E-9 on Fatigue, it was deemed expedient to assign to various individuals the responsibility for preparing drafts of the sections. This was done as follows:

I. Introduction.....	R. E. Peterson
II. Symbols and Nomenclature for Fatigue Testing .	J. M. Lessells
III. Fatigue Testing Machines.......	O. J. Horger
IV. Specimens and Their Preparation	J. B. Johnson
V. Test Procedure and Technique...	W. N. Findley
VI. Presentation of Fatigue Data....	L. R. Jackson
VII. Interpretation of Fatigue Data....	R. L. Templin
VIII. Bibliography.....	T. J. Dolan

These drafts have been circulated to and have been discussed by the committee as a whole at two annual and three spring meetings of the Society. Revisions and additions have been made to an extent that we believe the Manual represents the current practice and views of the majority of members of Committee E-9 on Fatigue. However, we still consider this to be our initial attempt and will welcome criticism and suggestions.

SECTION II—SYMBOLS AND NOMENCLATURE FOR FATIGUE TESTING[1]

PART A.—SYMBOLS USED IN FATIGUE TESTING

The American Standard Letter Symbols for Mechanics of Solid Bodies (ASA No.: Z10.3-1942)[2] are recommended. For stress, the use of S with appropriate subscripts is preferred for general purposes. The Greek symbols are generally preferred for mathematical analysis.

Symbol	Term
A	Area of cross-section
C	Cycle ratio
c	Distance from centroid to outermost fiber
D or d	Diameters
f	Frequency
I	Moment of inertia
J	Polar moment of inertia
K	Stress concentration or strength reduction factor with suitable subscript
n or N	Number of cycles
P or W	Load
q	Notch sensitivity
R	Stress ratio
S or σ	Stress, normal
S_s or τ	Stress, shear
T	Torque
t	Time
t or θ	Temperature[3]
ω	Circular frequency $= 2\pi f$

PART B.—NOMENCLATURE FOR FATIGUE TESTING[4]

Stress Cycle.—A stress cycle is the smallest section of the stress-time function which is repeated periodically and identically as shown in Fig. 1.

Nominal Stress, S.—The stress calculated on the net section by simple theory such as $S = P/A$ or $S = Mc/I$ or $S_s = Tc/J$ without taking into account the variation in stress conditions caused by geometrical discontinuities such as holes, grooves, fillets, etc.

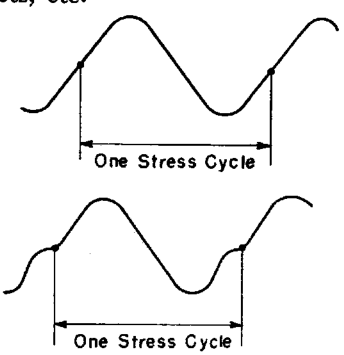


FIG. 1.—Stress Cycle.

Maximum Stress, $S_{max.}$—The highest algebraic value of the stress in the stress cycle, tensile stress being considered positive and compressive stress negative.

Minimum Stress, $S_{min.}$—The lowest algebraic value of the stress in the stress cycle, tensile stress being considered positive and compressive stress negative.

Range of Stress, S_r.—The algebraic difference between the maximum and minimum stress in one cycle, that is, $S_r = S_{max.} - S_{min.}$ For most cases of fatigue testing the

[1] Drafted by J. M. Lessells, Associate Professor of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. (Revised following discussion by A.S.T.M. Committee E-9.)

[2] Obtainable from the American Standards Association, 70 E. 45th St., New York 17, N. Y. (30 cents per copy).

[3] Use θ for temperature where time, t, is also used.

[4] The nomenclature given here refers to tensile and compressive stresses but is also applicable to shear stresses.

stress varies equally above and below zero stress but other types of variation may be experienced as shown in Fig. 2.

Alternating Stress Amplitude (or Variable Stress Component), S_a.—One half the range of stress, that is, $S_a = S_r/2$.

Mean Stress (or Steady Stress Component), S_m.—The algebraic mean of the maximum and minimum stress in one cycle, that is, $S_m = (S_{max.} + S_{min.})/2$.

Stress Ratio, R.—The algebraic ratio of the minimum stress and the maximum stress in one cycle, that is, $R = S_{min.}/S_{max.}$

Stress Cycles Endured, n.—The number of cycles which a specimen has endured at any stage of a fatigue test.

Fatigue Life, N.—The number of stress cycles which can be sustained for a given test condition.

Cycle Ratio, C.—The ratio of the stress cycles applied at a given stress level to the expected fatigue life at that stress level based on the S-N diagram, that is, $C = n/N$.

S-N Diagram.—A plot of stress against number of cycles to failure. It is usually plotted S *versus* log N, but a plot of log S *versus* log N is sometimes used.

Fatigue Limit (or Endurance Limit[5]), S_e.—The limiting value of the stress below which a material can presumably endure an infinite number of stress cycles, that is, the stress at which the S-N diagram becomes horizontal and appears to remain so. It should be noted that certain materials and environment preclude the attainment of a fatigue limit.

If the stress is not completely reversed, it is necessary to state what is meant by the fatigue limit. It may be expressed in terms of the alternating stress amplitude or the

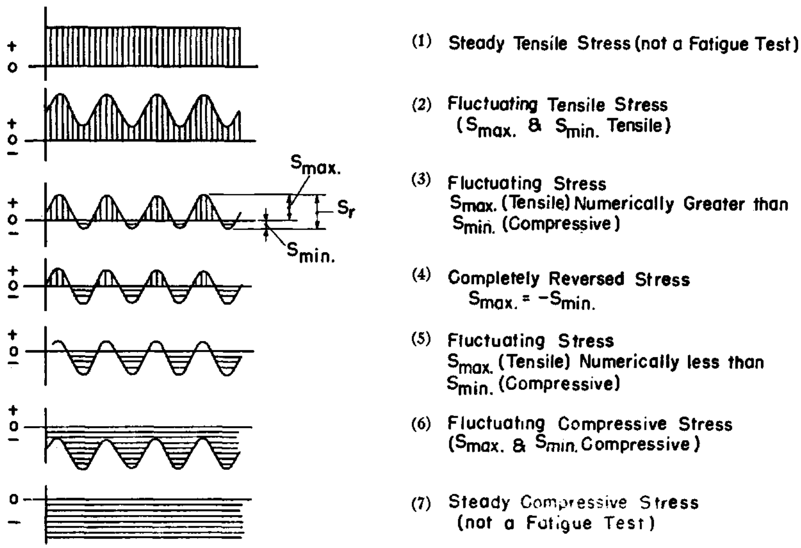


FIG. 2.—Types of Stress.

NOTE.—British Usage, which differs from the above, is as follows: a stress which does not change sign is considered as fluctuating (2 and 6); a stress which does change sign is considered as alternating (3, 4, and 5).

[5] "Fatigue limit" is considered preferable.

maximum stress; which ever method is used, it is also necessary to state the value of the mean stress, minimum stress, or stress ratio.

Fatigue Strength,[6] S_n.—The greatest stress which can be sustained for a given number of stress cycles without fracture. The number of cycles should always be given. The same considerations as given under *Fatigue Limit* apply where the mean stress is not zero.

Fatigue Ratio (or Endurance Ratio[7]**).** The ratio of the fatigue limit (or endurance limit), S_e, or fatigue strength, S_n, to the static tensile strength, S_u, that is, S_e/S_u or S_n/S_u.

Stress Concentration Factor, K_t.—The ratio of the greatest stress in the region of a notch or other stress concentrator as determined by advanced theory, photoelasticity, or direct measurement of elastic strain, to the corresponding nominal stress.

Fatigue Strength Reduction Factor,[8] K_f.—The ratio of the fatigue strength of a member or specimen with no stress concentration to the fatigue strength with stress concentration. K_f has no meaning unless the geometry, size, and material of the member or specimen and stress range are stated.

Notch Sensitivity, q.—A measure of the degree of agreement between K_f and K_t for a particular specimen or member of given size and material containing a stress concentrator of given size and shape. Thus: $q = (K_f - 1)/(K_t - 1)$. Notch sensitivity varies between zero (where $K_f = 1$) and unity (where $K_f = K_t$).

[6] "Fatigue strength" may also be considered to be a preferred general term, of which "fatigue limit" is a special case.

[7] "Fatigue ratio" is considered preferable.

[8] In some cases an author may like to use a shorter term, "fatigue notch factor."

SECTION III—FATIGUE TESTING MACHINES[1]

Fatigue testing machines (**1**)[2] may be classified as to:

1. Type of load: constant load or constant displacement.
2. Type of stress: bending, torsion, etc.
3. Design characteristics; mechanical, hydraulic, magnetic, etc.
4. Operating characteristics: resonant or non-resonant.

1. The most general classification designates machines as being either of constant load or constant displacement types. Distinguishing characteristics of these two types are:

Constant Load Type:

(*a*) Applied load or amplitude of loading is constant throughout the test.

(*b*) After the fatigue crack initiates its rate of propagation usually increases.

Constant Displacement Type:

(*c*) Applied deformation or amplitude of deformation is constant throughout the test.

(*d*) Load on specimen is reduced after fatigue crack initiates and rate of propagation of crack is usually retarded.

As a result of conditions existing in (*b*) and (*d*) above, the shape of the *S-N* curves may be different when obtained under constant loading as compared with constant displacement. Influence of type of loading can only be evaluated by introducing the time at which the fatigue crack initiates.

An important consideration in any machine is the means used to measure and maintain the forces acting on the member under test. Constant loading machines of the mechanical type may use inertia forces, dead weights or a spring system having a low spring constant, which permits convenient evaluation of the force on the specimen. In a displacement-type machine the deflection, shortening or lengthening of the specimen itself may be measured with wire-type strain gages, mirrors, or micrometer devices. Often a dynamometer is connected in series with the specimen to ascertain the force acting but careful consideration of inertia forces is required.

Displacement-type machines are sometimes equipped with automatic devices to detect and correct any change in deformation; sometimes strain conditions imposed on the specimen are adjusted and kept under more or less close observation. It is known that the specimen may develop creep or changes in elastic properties (**2**) during test; also changes in the grips or mechanical factors such as clearances in the system occur which will modify the stress in the specimen.

The over-all error in the accuracy of the machine does not usually exceed ± 3 per cent and while lower values may be attained, this is largely dependent upon the accuracy of the following factors discussed by Erlinger (**2**):

1. Static calibration.
2. Dynamic calibration as referred to actual loading on specimen.
3. Maintenance of load on specimen through the test.

2. Fatigue testing machines may be broadly divided as to type of stressing, design and operating characteristics as follows:

[1] Drafted by O. J. Horger, The Timken Roller-Bearing Co., Canton, Ohio. (Revised following discussion by A.S.T.M. Committee E-9.)

[2] The boldface numbers in parentheses refer to the list of references appended to this section, see p. 23.

Type of Stressing	*Design Characteristics*	*Operating Characteristics*
1. Rotating bending.	1. Mechanical	1. Resonant or nearly resonant.
2. Repeated bending.	2. Electromagnetic.	2. Non-resonant.
3. Axial load.	3. Hydraulic.	
4. Torsion.	4. Centrifugal force.	
5. Combined stress.	5. Pneumatic.	
6. Special machines such as those used for investigating: (*a*) Repeated impact. (*b*) Contact stresses. (*c*) Variable stress amplitude. (*d*) Structural and machine parts.		

Magnetic, centrifugal-force, and pneumatic-type machines produce a periodical disturbing force generally used to cause forced vibrations of the system containing the test specimen. This system is usually tuned to operate at or near resonance of the disturbing force in order to amplify the excitation force on the specimen. Provision for tuning adjustment is frequently obtained by connecting a spring member in series with the test specimen. When a fatigue crack develops in the specimen the natural frequency of the system is reduced and the amplitude of vibration changes. This amplitude increases if the excitation frequency approaches the natural frequency and decreases when these two frequency factors diverge from one another.

Mechanical and hydraulic machines are usually of the non-resonant type. Hydraulic and centrifugal-force machines are more often used for the testing of actual production machine parts or large structural members rather than for conventional specimens. There is an increasing trend toward the fatigue testing of full size components. In recent years, the design and performance of all types of machines have been greatly improved and their capacity increased to facilitate investigations of full size production units.

Well known fatigue-testing machines are presented here in line diagrams and discussed according to the type of stressing applied to the specimen. Any of these machines may incorporate special environment conditions such as high temperature or corrosion features.

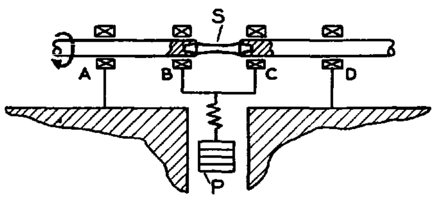
ROTATING-BENDING MACHINES

All machines in this class are of the mechanical and non-resonant type. All surface fibers at the critical section of specimens of symmetrical cross section are subjected to the maximum stresses; one complete stress reversal occurs in one revolution of the specimen. The most used machine has four point loading arrangement as shown by Fig. 3(*a*) and is known in this country as the R. R. Moore (3) machine. Here a uniform bending moment is applied over the length of the specimen. It is essential in Fig. 3(*a*) that the spacing $AB = CD$ to insure correct evaluation of the bending moment on the specimen.

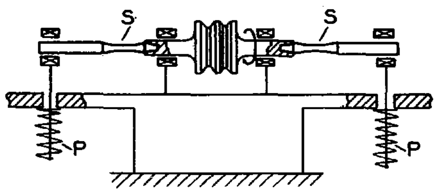
This design represents an improvement over its predecessor, the Sondericker and Farmer (4) type, because of the short specimen (5). The R. R. Moore type is commercially available in a capacity as high (6) as 10,000 in-lb. and a speed of 3600 rpm. Speeds of 10,000 rpm. are used in smaller capacity machines. It is desirable to incorporate some resiliency in the loading system, such as a spring, to minimize any inertia stresses resulting from small unavoidable vibration of the specimen. Instead of hanging weights, some machines have a scale beam loading system for convenience of stressing the specimen.

The double and single end cantilever designs in Figs. 3(*b*) and (*c*) are significant because they were first used by Wöhler (7) in his historic tests. Simplicity and low cost, particularly for testing large sections, are reasons for their extensive use. The smallest machine was built by Peterson (8) to test 0.050-in. diameter specimens. Machines as large as 8,000,000 in-lb. capacity at 600 rpm. for testing 12-in. diameter members or about half this capacity at 1550 rpm. for $9\frac{1}{2}$-in. specimens are in operation (9). Such large machines often require water-cooled heads on the ends of the specimen to stress over a considerable length of the test section.

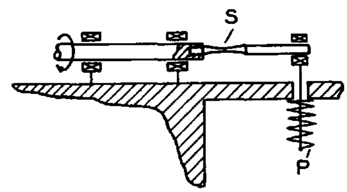
Fig. 3(*d*) represents a modification of the single end cantilever machine commercially available (11) for testing specimens, particularly those not permitting surface preparation such as wire and simi-



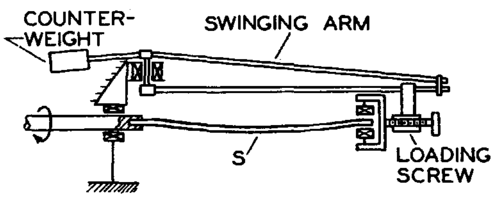
(*a*) Four Point Loading R. R. Moore Machine.



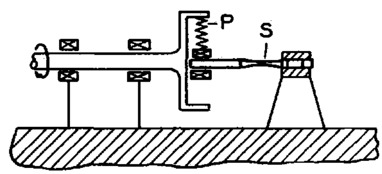
(*b*) Double-End Rotating Cantilever Machine.



(*c*) Single-End Rotating Cantilever Machin e



(*d*) Wire Tested as Euler Column. (Specimen deflection is in horizontal plane instead of vertical shown.)



(*e*) Stationery Cantilever Specimen with Rotating Load.

S—Specimen.
P—Load Applied Through Weights, Springs, Scale Beam, or Similar Means.
a and *d*—Bearing Supports.
b and *c*—Load Hangers.

FIG. 3.—Rotating-Bending Fatigue Testing Machines.

carry away the heat generated in the specimen. In these machines the bending moment varies linearly over the specimen length. This is generally no disadvantage particularly if specimens with stress concentration are being investigated; if unnotched specimens are used then the tapered design developed by McAdam (10) permits a nearly uniform

lar long length and small diameter members. The bending moment on the specimen is obtained by loading it as a pin ended column; in this way the bending stress is a maximum over the center portion of the specimen so that failure does not develop in the grip portion where the nominal stress is low. This type of machine was first used by Shelton (12) and Haigh-Robertson (13). A similar purpose rotating wire arc fatigue machine was developed by Kenyon (14).

In another machine described by H. J. Gough (1a) and H. F. Moore (1b) one end of the specimen is held rigidly in a support while the other end, which runs in a bearing, is rotated in a small circle by a revolving spring loading arrangement as in Fig. 3(*e*). A modification of this principle has been incorporated in high temperature fatigue machines (15).

A machine (16) of recent design used in France has a rotating cantilever specimen located in a vertical angular position and is motor driven at 3000 rpm. Dead weights are used to load a specimen about $\frac{1}{8}$-in. diameter which has been standardized by the Air Ministry. It is claimed that this arrangement of the specimen and loading system has a natural tendency to keep the specimen tight in the grips and also permits the use of the same cabinet for many different types of fatigue machines.

The Goodyear (17) tire cord fatigue tester is designed to simulate the condition of alternating extension and compression. The cords to be tested are laid parallel to a length of rubber hose and then covered and cured in a mold. This sample is mounted on the ends of two spindles, inflated, and rotated about its curved axis.

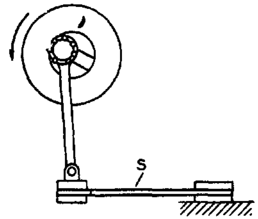
REPEATED-BENDING MACHINES

Sheet and plate materials are largely investigated by bending the specimen back and forth instead of rotating it. Many terms have been used to designate such a machine but the expression *repeated bending* is recommended. One of the advantages of this type machine is that surface preparation of the specimen is not always required although specimens are usually shaped to prevent failure in the grips; also a static preload as well as the range of applied load can be varied within wide limits.

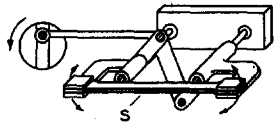
The mechanically driven machine incorporates some form of adjustable crank device, Fig. 4(*a*), and is commercially available (18) with 20,000-lb. capacity at the connecting rod and speeds up to 1750 rpm. Some machines are designed for applying torsion as well as bending loads to the specimen (18). These machines are of the constant displacement class with the bending moment increasing linearly over the specimen length. Machines have been designed to test as many as 126 cantilever specimens simultaneously (19). These machines seldom operate at speeds over 1000 rpm. due to difficulty with dynamic balancing of the crank mechanism or natural frequency of the system or both. Balancing was partly overcome in a machine for testing twelve specimens simultaneously operating at 3000 rpm. by means of a unique drive described by Johnstone (1h). Non-metallic hose (20) and wire cable (21) have been investigated by application of the old principle of flexing the specimen over a system of pulleys.

Another type of flexural machine (22), Fig. 4(*b*), applies a uniform bending moment over the length of the specimen. A buckling type of test is shown in Fig. 4(*c*) which distributes a bending moment over the test section similar to that in Fig. 3(*d*). It permits testing specimens such as spring leaves without machining or surface preparation; an appreciable volume of metal over the test section is subjected to nearly maximum stresses. The Dietz (23) machine, Fig. 4(*d*) has been used for investigating adhesives. It is seldom used for investigating metals because of the difficulty in connecting the crank rod to the specimen without introducing stress concentration leading to premature failure of the specimen.

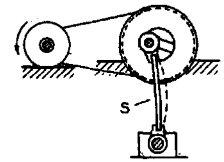
A simple oscillator form of magnetic machine (24), Fig. 4(*e*), incorporates a cantilever specimen magnetically excited



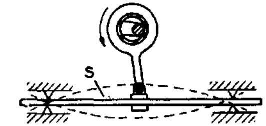
(*a*) Cantilever Specimen Actuated by a Crank or Cam.



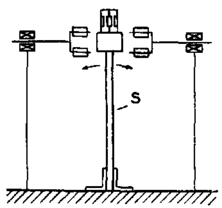
(*b*) Crank Type Machine Subjecting Specimen to Uniform Bending Moment.



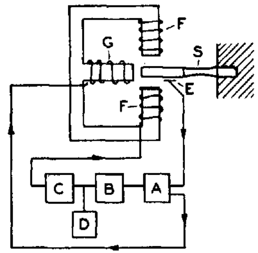
(*c*) Column Type Bending Machine.



(*d*) Simple Beam Actuated by Crank or Cam.

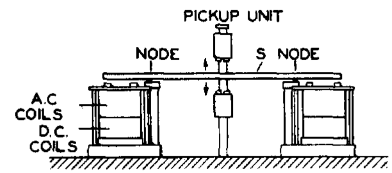


(*e*) Magnetic Type Resonant Machine.

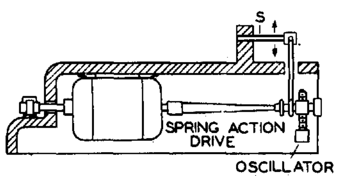


(*f*) Magnetic Type Resonant Machine.

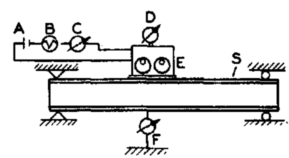
A—Preamplifier.
B—Control Circuit.
C—Power Amplifier.
D—Cycle Counter.
E—Vibration Detector.
F—a-c. Coils.
G—d-c. Coils



(*g*) Magnetic Type Resonant Machine—Rayflex.

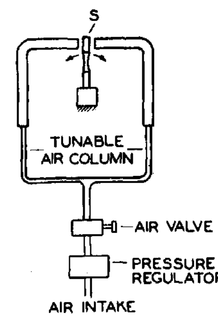


(*h*) Single-Weight Mechanical Oscillator Type Machine—Sonntag.



(*i*) Mechanical Oscillator Having Two Opposed Rotating Weights.

A—Battery or Generator.
B—Resistance.
C—Wattmeter.
D—Tachometer.
E—Oscillator.
F—Deflectometer.



(*j*) Pneumatic Type Resonant Machine.

FIG. 4.—Repeated Bending Fatigue Testing Machines.

back and forth at the free end Magnets are alternately energized by an inertia switch attached to the free end of the specimen to produce an alternating force oscillating at the natural frequency of the vibrating system. A later application (25) of this principle is shown in Fig. 4(*f*) where the vibrating force is applied to the specimen by means of a-c. and d-c. coils. The a-c. coils are powered by an electronic circuit with a current frequency equal to the natural frequency of the specimen. Amplitude of specimen vibration is kept constant by auxiliary electronic devices. A modification of Fig. 4(*f*) was also discussed by Schulz (1e). In another machine (26) for elevated temperature tests the specimen was driven at constant amplitude by a reciprocating electromagnetic motor at 7200 cycles per min.

The Rayflex machine (24), Fig. 4(*g*), supports a long specimen at the nodes for free vibration. Magnets are excited by a variable frequency alternating current and tuned to the natural frequency of the specimen. Stress is calculated from the deflection as measured by a micrometer-microscope. The magnetizing current governs the amplitude of specimen vibration.

A mechanical connection from a moving coil type loudspeaker (27) is also used to vibrate a test member. The specimen system is vibrated at its natural frequency or an electronic beat frequency oscillator supplies the required vibration. This principle has been applied to a machine (28) producing 1200 to 600,000 cycles per min. Electro-magnetically excited machines generally furnish relatively low testing forces, require small power consumption, operate at high frequency, and depend upon resonance characteristics. Such machines (29) are commercially available and may be applied as a source of power for obtaining any type of stressing.

Centrifugal force type mechanical oscillators having a single eccentric were used by early investigators. The first commercial oscillators were built by Losenhausen about 1927. A mechanical oscillator type machine by Sonntag (30), Fig. 4(*h*), incorporates a single out-of-balance rotating weight attached to the end of the cantilever specimen. One size of machine applies an alternating force up to ±20 lb, at 1800 cycles per min. A larger machine is available (31) which incorporates attachments for either bending or torsion loading of the specimen. The capacity is ±1000 lb. at 1800 cycles per min. on the end of the specimen. Still another size machine for temperature investigations through 1800 F. applies flexural loads up to ±1350 lb. at 3600 cycles per min. along with a superimposed tensile load up to 8000 lb. The Sonntag machines can be used with suitable adapters for axial tests.

Schenck (1f) also built a mechanical oscillator of about the same capacity range. Gough (1a) made leaf spring tests using an oscillator on the end of the specimen. The Schenck machine (1i) at Darmstadt suspended a number of specimens vertically on a framework so that one end of each specimen was fixed and a small mass was attached to the free end. An oscillator vibrated the framework near the natural frequency of the specimen system.

Bernhard (32) and Lazan (33) mechanical oscillators have two opposed out-of-balance weights to produce a resultant force acting in one plane. An example of the application of this type to a beam specimen is illustrated in Fig. 4(*i*). Additional use of such oscillators is later discussed under special machines because they are often adapted to testing of structural and machine elements rather than materials.

Pneumatic oscillators were used by Jenkins and Lehman (34) in 1929 and one of later design and commercially available (35) is shown in Fig. 4(*g*). The specimen is vibrated at its natural frequency by means of air pressure acting on two small pistons connected to the free end of the cantilever specimen. The pneumatic column is then tuned so that its resonant frequency coincides with that of the specimen. Johnstone (1h) and Kroon (36) also refer to the use of a similar pneumatic machine. Pneumatic vibrator equipment for general purpose testing is commercially available (37).

Axial-Load Machines

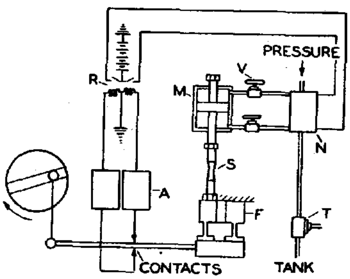
The term axial-load machines is recommended instead of the many other designations given machines of this type. They may be classified as: (*a*) non-resonant mechanical, hydraulic or magnetic systems, or (*b*) resonant type using forced vibrations excited by a magnetic or centrifugal force. Axial stressing requires large testing forces and usually rugged machine construction. The specimen is subjected to stresses more or less uniform throughout the cross section but perfect axial loading is very difficult to obtain in practice. Axial loading machines frequently give lower fatigue strength values than bending machines (38).

The earliest machine by Wöhler (7) in 1871 incorporated a direct crank drive to load the specimen by means of compressing or extending a spring connected in series with the specimen. Less than 100 cycles per min. were permissible because forced vibrations developed unless the drive had a speed considerably under the natural frequency of the spring system. Later Jasper (1b) designed a spring type machine of 4000-lb. capacity which operated at a maximum speed of about 200 rpm. In 1902 Reynolds and Smith (39) used reciprocating masses to obtain inertia forces acting axially on the test specimens. A further development of this principle was made by Stanton and Bairstow (40) in 1905 in a machine testing four specimens simultaneously and operating at 800 to 1000 cycles per min.

Moore and Krouse (1j) used a cam-operated lever working against a spring to apply axial load on a specimen at 1000 cycles per min. An elastic ring gage was used in series with the test specimen and by means of auxiliary springs any desired ratio of maximum to minimum stress was obtained. Templin (41) described a machine arranged to test four specimens simultaneously. Two variable eccentrics were used to actuate two crossheads. To each crosshead was attached one end of each of two specimens. The opposite ends were attached to link dynamometers. An eccentric crank drive and lever principle has been used by Templin (42) in machines of ±50,000-lb. capacity operating as high as 500 rpm. Crossed-plate fulcrums (43) support one end of the lever and this provision is a simple and effective means of overcoming difficulties experienced with the usual type of bearings. Wilson and Thomas (44) have previously used the crank and lever system on machines of ±50,000 and ±200,000-lb. capacity. A study was made of the inertia forces acting on the specimen for the large machine. It was found that at 180 rpm. the force on the specimen was within 2 to 3 per cent of that given by the link-shaped spring dynamometer in series with the eccentric driving rod. The bearing supports on the large machine were of unusual design.

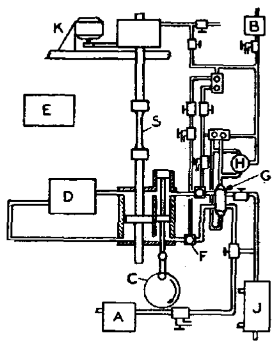
A machine by Krouse (45) is illustrated in Fig. 5(*a*). Here an elastic load lever is actuated by a variable throw crank. A flexural plate arrangement is used for pivoting the lever arm and transmitting the lever force to the specimen in a ratio of about 10 to 1. Constant maximum load is maintained on the specimen by a hydraulic unit in series with the specimen. This unit functions through an electronic circuit. This type of machine is built in 5000 to 100,000-lb. capacity range operating at 1500 rpm. for the small size and 500 rpm. for the large.

The Krouse-Purdue axial load machine in Fig. 5(*b*) derives its force from hydraulic pressure acting on a large piston directly connected to the test piece through the piston rod. Hydraulic pressure is developed from three sources depending upon the speed and specimen deformation. For direct stress tests with normal specimen deflection the pressure is derived from the small booster piston actuated by the variable throw crank, the stroke being variable while the machine is in operation. Normal operating speed using this method is 1000 cycles per min. with a maximum loading capacity of ±60,000 lb. For long-stroke testing, hydraulic pressure is used from a conventional high pressure pump, acting through a solenoid-operated four-way valve. For slow-speed testing, pressure



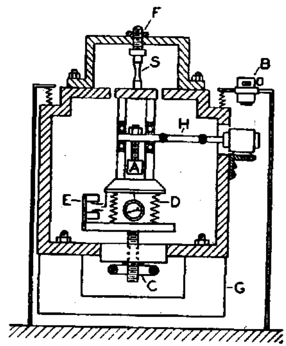
(*a*) Crank and Lever Operated Machine—Krouse.

A—Amplifiers.
V—Flow Control Valves.
F—Flexure Plates.
T—Low Pressure Throttle.
M—Hydraulic Load Maintainer.
R—Relays.



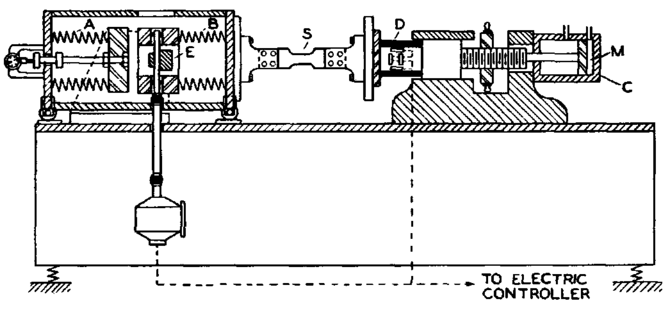
(*b*) Krouse-Purdue Direct Stress Machine.

A—Fifty five-G.P.M. Pump.
B—Five G.P.M. Pump.
C—High Speed Drive and Amplitude Control.
D—Pressure Control.
E—Electronic Unit.
F—Selector Valves.
G—Four-Way Valve.
H—Pilot.
J—Oil Cooler.
K—Hydraulic Nut.



(*c*) Single-Weight Mechanical Oscillator Type Machine—Sonntag.

A—Mechanical Oscillator.
B—Cycle Counter.
C—Preload Adjusting Screw.
D—Springs for Pre-Loading.
E—Micro-Switch Cut-Off.
F—Adjusting Screw.
H—Flexible Drive Shaft.



(*d*) Single-Weight Mechanical Oscillator Type Machine—Sonntag.

A—Preload Springs.
B—Resonance Springs.
C—Hydraulic Preload.
D—SR-4 Gage Dynamometer.
E—Eccentric.
M—Hydraulic Load Maintainer.

FIG. 5.—Axial-Load Fatigue Testing Machines.

is obtained from the low capacity load maintainer pump. Ram force is measured and controlled by the high speed differential pressure-measuring mechanism. Testing forces may be changed at the will of the operator while the machine is in operation. Attachments are available for bending, combined-stress, and torsion testing.

Findley (46) has described new apparatus which provides means for detecting and correcting strains introduced in the specimen when it is fastened to the machine. It is always difficult to obtain and determine if uniform stress is applied over the cross-section of the specimen and his apparatus incorporates features desirable on any axial load machine.

In another machine (16) used in France the gravity force from suspended weights is transferred to the specimen by means of a lever system. A member rotating on a specially shaped curved track transmits alternating tension and compression to the specimen by transferring the weight from the lever system. Testing speed is 120 rpm. and specimens about $\frac{3}{32}$ in. in diameter can be investigated.

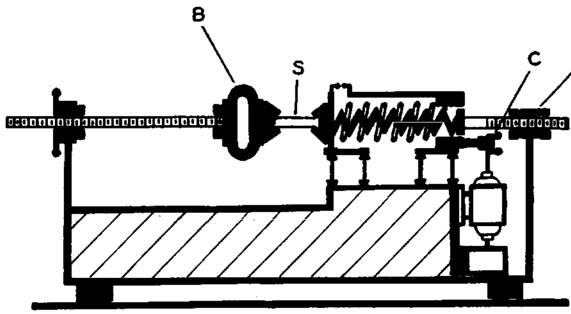
Tire cord fatigue testing machines often utilize cam-operated or eccentric-drive systems to apply intermittent stretch or snapping action to the specimen. Such machines have been built by Goodrich (47), U. S. Rubber (48), and Firestone (49).

As early as 1905 Smith (50) used unbalanced rotating masses to develop axial loading. Such mechanical oscillators are incorporated in modern machines by Sonntag as illustrated in Figs. 5(*c*) and (*d*). The former is built with a load range of 2000 lb. at 1800 load cycles per min. (31) and a larger size of 10,000 lb. at 3600 cycles per min. (51). The machine in Fig. 5(*d*) has a range of 10,000 lb. at 1200 to 1900 cycles per min. and is equipped with an automatic load maintainer (52). Aircraft structures (53) were subjected to alternating stresses through the use of mechanical oscillators of capacity as large as 50,000 at 1800 cycles per min. The vibrating system was tuned to yield a magnification factor of 5 to 50. Mounting difficulties experienced with these machines were overcome by unique methods.

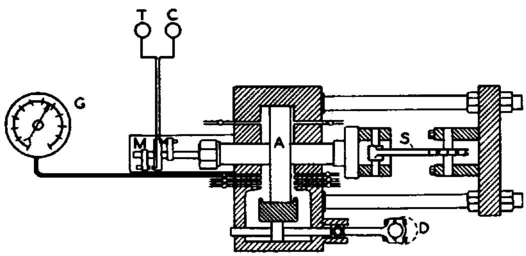
An earlier machine by Schenck-Erlinger (1d) operates on the same general principle as that of Sonntag and was built with a capacity as large as ± 30 tons at 2000 load cycles per min. Figure 6(*a*) shows how Schenck (1f) mounted a mechanical oscillator on the end of a pivoted lever from which an electronic load control also functions. Springs in series with the specimen permit tuning the system in resonance with the oscillator. Machines have been built with capacities as large as ± 10 tons at 2200 to 3000 load cycles per min. Rubber and fabric strip have also been tested in oscillator-type machines (54).

Mohr and Federhaff (1f, 55) incorporated the action of a double eccentric which was coupled to a lever system so that the difference in stroke acts on the specimen. About 720 load cycles per min. were attained.

The hydraulic-pulsator type machine shown in Fig. 6(*b*) was developed by General Motors (56). Two small diameter actuating pistons are driven by an adjustable crankshaft so that by turning one crank pin with respect to the other it is possible to vary the amount of oil displaced. This oil at high pressure is discharged to either or both sides of a large diameter main loading piston to which the specimen is connected. The travel of the large piston is controlled by leakage and bleed off. A dynamic stroke of as much as 0.170 in. can be obtained and the rate of application can vary up to 2000 cycles per min. The load capacity is 100,000 lb. in each direction. A reaction weighing fixture with electronic

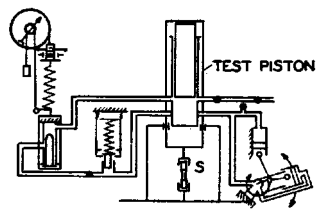


(*a*) Mechanical-Oscillator Type Resonant Machine—Schenck.
A—Adjustment for Preload. *B*—Dynamometer. *C*—Mechanical Oscillator.

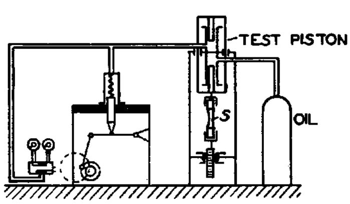


(*b*) G.M.R. Pulsator.

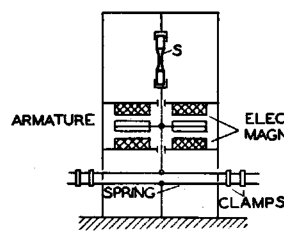
A—Loading Piston.
B—Actuating Piston.
M—Micrometer Contactors.
D—Crankshaft.
T and *C*—Neon Lamps Indicating Tension and Compression.
G—Oil Pressure Gage.



(*c*) Amsler Pulsator.



(*d*) Losenhausen Pulsator.



(*e*) Magnetic Type Machine—Haigh.

FIG. 6.—Axial-Load Fatigue Testing Machines.

control equipment is available. Adapters for torsion loading of 5000 in-lb. are provided so that any desired phase relation with simultaneous axial loading may be obtained.

Foreign-made hydraulic-pulsator machines (1f) of earlier design by Amsler (57) and by Losenhausen (58) are shown in Figs. 6(*c*) and 6(*d*), respectively. Amsler has a valveless and differential piston pump consisting of two pivoted cylinders arranged in the form of a V. By changing the angle between the two cylinders the resultant volume, fed to another cylinder in series with the specimen, can be adjusted to suit the load required on the specimen. The design by Losenhausen permits variation in load by changing the stroke of the drive. This is done by shifting the pump axis along the oscillating lever which is actuated by a crank drive. Such pulsators have been built with loading speed as high as 3000 cycles per min. and as large as ±50 ton-capacity.

Magnetically-excited alternating axial load machines were developed as early as 1911 by Hopkinson (59) and Kapp (60). Later versions are the Haigh (61) and Schenck (1i) machines. Figure 6(*e*) illustrates the former where one end of the specimen is attached to the frame of the machine and the other end to the armature. This armature operates between two magnets energized by two-phase alternating current, one phase being connected to each magnet. First the air gaps are made equal and then the clamps are moved on the spring until the machine is in resonance with the magnetic excitation without the specimen. In this way no unknown inertia forces act on the specimen. With the specimen in place the machine operates below resonance as a forced vibration machine; consequently the forces exerted on the specimen are small. The speed of loading was originally 2000 cycles per min., but the machine can be modified to obtain higher frequencies.

The Schenck machine (1e, 1i) is somewhat similar to the Haigh machine except that it operates on a resonant principle at very high frequency of as much as 30,000 load cycles per min. Means are provided for carrying away the heat developed in the specimen at such high speed of testing. An a-c. field is superimposed upon a d-c. field in the armature system. This machine, which has been used only in Germany, has also served as a means for measuring the damping characteristics of materials. A recent Amsler (62) machine of the magnetic-pulsator type operates at about 6500 cycles per min. Steel specimens of about $\frac{5}{16}$-in. diameter can be tested. This machine is also arranged for damping measurements.

TORSION MACHINES

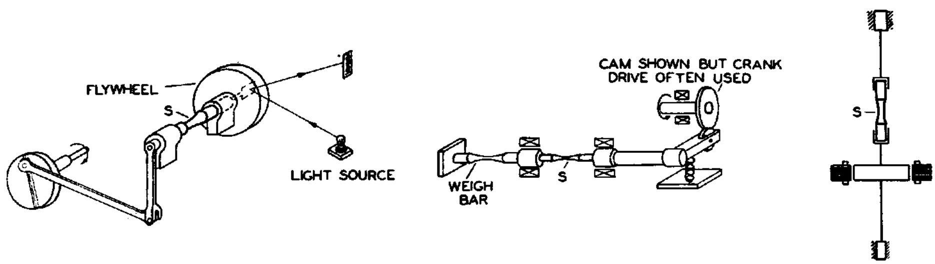
The oldest machine arrangements for developing reversed torsional stresses consisted of a crank drive connected by a leverage system to one end of the specimen with the opposite end opposed by a coil spring system for measuring the load. Such machines were used by Wöhler (7), Föppl (63), Mason (64), and Olsen-Foster (65). Owing to difficulties in measuring the inertia forces acting on the specimen these principles have not been extended to present day machines. Earlier machines by Stanton and Batson and also H. F. Moore (1b, 1j) used a dead weight on the end of a rotating cantilever beam attached to one end of the specimen.

The McAdam machine (66), Fig. 7(*a*), utilizes a crank drive on one end of the specimen with a flywheel attached to the opposite end. Torsional impulses of 1200 to 2100 per min. are applied which are near the natural frequency of the system. Stromeyer (67) had developed the same principle earlier; this machine was arranged to test two specimens simultaneously. The resonance region is very narrow for machines of this design so that small variations in motor speed from the resonant frequency of the system greatly influences the amplitude of specimen deflection. This is an important consideration in any type machine using inertia forces inasmuch as the applied stress is proportional to the square of the speed of the machine. This difficulty and corrective means, resulting in operating considerably below resonance, were discussed by H. F. Moore (68). A modification of the McAdam machine is incorporated in one built by Krouse (18) having a capacity of 50,000 in-lb. at 1500 cycles per min. Here the flywheel is excited at resonant frequency through a friction clutch and caused to oscillate by a crank drive.

Many laboratories still use torsion machines, similar to that shown in Fig. 7(*b*), with a crank drive where the specimen is in series with a weigh bar or elastic dynamometer to measure the couple produced. This idea was first projected by Rowett (69) in 1913 and present machines represent refinements in design detail over this principle. The machine used by H. F. Moore (1j) operates at 1500 rpm. and permits a range of twisting moment that can be varied from complete reversal to torsion in one direction only; attachments are also available for making reversed bending tests.

Several designs of foreign machines have incorporated a crank drive with a torsion weigh bar system. Schenck (1f) built single specimen machines (70) operating at 3000 rpm. and multiple specimen machines at 1500 rpm. An Amsler machine (1f) used a lever on the weigh bar which was connected to a spring system through adjustable wedges to facilitate the measurement of the torque on the specimen. The machine operated at 1800 rpm. Another Schenck machine (1f), operating at 3000 rpm. incorporated an elaborate optical system so as to record the hysteresis loop as a means of making damping measurements throughout the fatigue test.

The difficulties experienced with the McAdam and Stromeyer machines were overcome in another machine, similar to Fig. 7(*a*), used at Wöhler Institute (1i) The motor crank drive speed is tuned to the resonant frequency of the specimen system by means of a regulator which responds to a variation in the phase shift of 90 deg. between the excitation impulse and excited oscillation. A pendulum type brake, operating in a liquid was used to maintain the deflection constant. By means of the regulator and brake devices it is claimed that, at resonance, the angular deflection of the flywheel can be as much as 50 to 100 times as large as that of the crank lever when testing low damping capacity materials.



(*a*) Resonant Type Impulse Machine. (*b*) Weigh-Bar Type Crank Machine. (*c*) Magnetic Type Resonant Machine —Losenhausen.

FIG. 7.—Torsion Fatigue Testing Machines.

A machine (16) of recent design in France utilizes the gravity forces of suspended weights, which are alternately transferred so as to apply a torsional loading to the specimen; this is done by a special mechanical means incorporating an internal conically-geared motor drive. The machine speed is only 120 rpm. and specimens of about $\frac{5}{16}$-in. diameter can be tested. Another French machine (16) of the crank type was recently designed by Chevenard. It operates at 1500 cycles per min. and accommodates specimens about $\frac{1}{16}$ in. in diameter. Provision is incorporated for measuring damping capacity.

Mechanical oscillator type machines, similar to Fig. 5(*c*) are built by Sonntag for alternating torsion testing at 1800 load cycles per min. A small size machine (30) has a capacity of 1125 in-lb. and a larger one $\pm$15,000 in-lb. Crankshafts having $9\frac{5}{8}$-in. diameter crank pins have been investigated (71) using mechanical oscillators to obtain reversed torsional stresses. Tractor engine crankshafts (72) have been tested in reversed torsion using a two-mass resonant system excited by oscillators where the specimen serves as the spring element.

Dorey (73) investigated full size marine shafting of the order of $9\frac{1}{2}$-in. diameter using a torsional vibration exciter. A special design of mechanical oscillator was employed. It consisted of a planetary system in which out-of-balance wheels were geared to a sun wheel and planet pinion. This exciter was connected to a two-mass torsional system where each of the masses constituted a 4-ton flywheel. The machine is run as close as necessary to the resonant frequency of the system which varied, according to specimen design, from 2400 to 2700 rpm.

Holzer (1i) also designed a magnetic type of resonant frequency machine to operate at 1200 to 1800 cycles per min. It was similar to that used by McAdam except that the crank drive was replaced by an armature connected in series with the specimen and operating in a d-c. magnetic field formed by two pairs of magnet poles. A swinging contact hammer is mounted on the flywheel which upon oscillation first switches in one pair of magnet poles, resulting in torsional impulses in one direction, and then the other pair of poles to produce impulses in the opposite direction. Damping measurements can also be made with this machine. Another German machine (1f) suspends the armature in series with the specimen as an entire vibrating unit on a torsional spring located on the inertia axis of the system. The vibrating unit is excited at its natural frequency of 4800 to 7200 cycles per min. by means of an interrupted current fed to the armature which causes it to oscillate (74). Bending tests can also be made on this machine as well as damping measurements.

A magnetic resonance type of torsion machine used by Losenhausen (1i) is shown in Fig. 7(*c*). The specimen is in series with a flywheel functioning as an armature in a magnetic field. Torsional impulses are produced by the armature when alternating current of the proper frequency and phase relationship is fed into both the rotor and stator. It is necessary that the natural frequency of the vibrating system be of the same value as that of the current source. In order to maintain the deflection of the specimen independent of line voltage variations, a brake operates on the lower portion of the armature shaft. A light beam is used to measure the deflection of the oscillating specimen. Arrangements are provided for prestressing the specimen.

The current production of Schenck machines (1d) incorporates a mechanical oscillator. These machines are of two types. One type is made in two sizes for moments up to 43 ft-lb. and 580 ft-lb. The working speed is 1800 and 3600 cycles per minute. This type machine can also be used for repeated bending as well as torsion. The second type of machine is shown in Fig. 8(*a*). It is made in three sizes for moment up to 2100, 7000, and 15,000 ft-lb. at a frequency of 700 to 3000 cycles per minute.

COMBINED-LOAD MACHINES

As early as 1916 Stanton and Batson (75) reported fatigue results using a machine for applying either reversed-torsional or reversed-bending stresses, or a combination of these two. Their machine has historical value but one of the difficulties experienced concerned vibrations at the speed used of 2000 rpm. Mounting of the specimen was such that it was not equally stiff in all angular positions of rotation and a dashpot was necessary to to minimize vibrations.

H. F. Moore (1b) used a machine which combined reversed bending with constant tension on the specimen. A bending machine of the type shown in Fig. 3(*e*) was combined with a spring load to add tensile stress. Ono (5) used a machine combining rotating bending with constant tension. The rotating bending specimen was connected to an electric absorption dynamometer. Similar machines were used by Lea and Budgen (76) and Nimhanmimie and Huitt (77) and discussed by Davis (78).

Superimposed alternating torsional stresses on pulsating tension-compression were investigated by Hohenemser and Prager (79) on a Schenck machine, Fig. 8(*a*), described by Lehr and Prager (80). A mechanical oscillator system of four rotating weights constituted the excitation source for tension-compression and a cross lever, having mechanical oscillators at each end, provided reversed-torsional loading. This machine operates at 3000 rpm. which is outside the resonance range.

The machines discussed above, with the exception of that of Stanton and Batson, are actually combined load machines in that the range of stress was changed.

A machine (1f) of the combined stress type was used by Bollenrath combining rotating bending with reversed torsion is shown in Fig. 8(*b*). A rotating bending machine, Fig. 3(*a*), was modified to include a braking arrangement consisting of magnets to provide reversed torsion loading.

The most systematic investigations were made by Gough and Pollard (81) using a machine, Fig. 8(*c*), to give a combination of reversed torsion and reversed bending in phase with each other. A vibrating arm is attached through a pivoted joint to one end of the specimen and this arm can be operated in any angular position with reference to the longitudinal axis of the specimen. In this manner any combination of bending and torsional stresses are obtained since they are proportional to the angular position of the arm. This arm is excited by a rotating out-of-balance disk, suspended from a long spring support, to which it is connected by a vertical link. The disk is operated at the resonant frequency of the system, which is 2130 rpm. Steel specimens of 0.3-in. diameter solid cross-section have been used.

By the use of attachments such as shown in Fig. 8(*d*), conventional fatigue machines for reversed bending, or reversed torsion, may be modified to give any combination of these stresses in-phase by varying the angle of the drive axis. Such modifications to standard machines have been discussed by Findley (82) and Puchner (83) and are commercially available (18). It is essential that the clamping arrangement on the specimen be designed to permit free deformation of the specimen; otherwise a secondary external moment will be imposed on the specimen. This subject as well as the determination of stress imposed on the specimen requires careful consideration as outlined by Puchner.

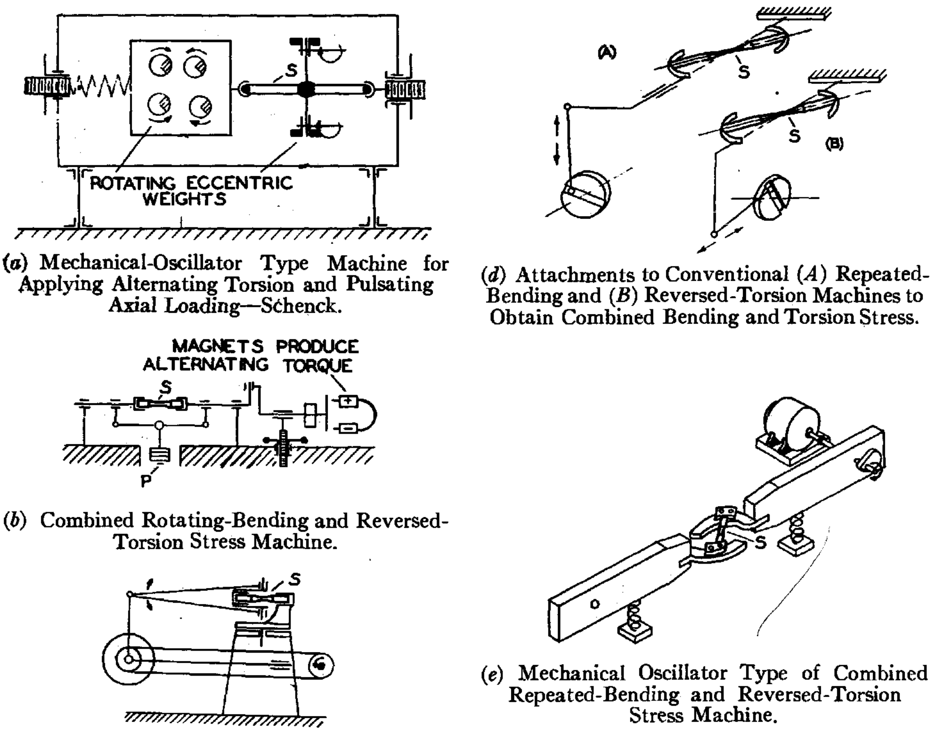
A resonant drive machine, Fig. 8(*e*), was used by Thum (84) to obtain combined reversed bending and reversed torsion loading. A mechanical oscillator system constituted the excitation source. Marin (85) also discussed some new testing machines for combined stress experiments.

REPEATED-IMPACT MACHINES

Repeated-impact tests were made before the conventional fatigue tests by Wöhler. In 1864 Fairbairn (**86**) reported tests on a wrought iron built-up girder by dropping a weight on the beam. These tests are of historical interest because a water wheel was used to lift the falling weight and an actual structural component was investigated instead of the usual small specimen. In another investigation Wöhler (**87**) used a mechanically-driven hammer to simulate an impact type of loading experienced with railway equipment. Marten (**87, 88**) tested wire rope by dropping a weight on the end of the rope at 13 times per min. Meyer (**87**) used a hammer-type impact machine operating at 50 to 60 strokes per min. to investigate mounted railroad tires.

In general, repeated-impact tests have been replaced by the usual fatigue tests. Principal reasons for loss of interest in the impact test are (1) specimen shape and rigidity of supporting means greatly influence the stresses which are difficult to maintain constant and measure, (2) speed of testing is comparatively slow, and (3) if the energy per blow is relatively high then McAdam (**89**) and Lessells (**90**), with some exceptions by Stanton (**91**), indicate that fatigue test results will arrange metals in order of merit similar to that given by the single-blow impact tests while if the energy per blow is low then the order of merit will be similar to that obtained by alternating-stress tests.

Except for the above primitive ma-



(*a*) Mechanical-Oscillator Type Machine for Applying Alternating Torsion and Pulsating Axial Loading—Schenck.

(*b*) Combined Rotating-Bending and Reversed-Torsion Stress Machine.

(*c*) Combined Repeated-Bending and Reversed-Torsion Stress Machine—Gough and Pollard.

(*d*) Attachments to Conventional (*A*) Repeated-Bending and (*B*) Reversed-Torsion Machines to Obtain Combined Bending and Torsion Stress.

(*e*) Mechanical Oscillator Type of Combined Repeated-Bending and Reversed-Torsion Stress Machine.

D—Dynamometer consisting of torsion bar and disk gages *G*.
M—Mass.
E—Mechanical oscillator.

FIG. 8.—Combined-Load Fatigue Testing Machines.

chines the earliest mechanical arrangement for making repeated-impact tests was developed for bending by Stanton (1a, 92) in 1906. A cam was used to raise a weight which, upon falling, strikes a beam specimen midway between two knife edge supports. The specimen is automatically rotated 180 deg. between each impact imposed at the rate of 100 blows per min. A modification of this principle was incorporated in another machine (8).

A tension-compression impact machine was developed by Stanton and Bairstow (1a, 93). A tup, operating in vertical guides, upon falling, strikes a special sleeve fixture arrangement holding the specimen. After each blow the sleeve is rotated 180 deg. so that during one blow the impact applies tension to the specimen and during the next blow produces compression.

A pair of swinging pendulum hammers was used by Gustafsson (94) to obtain reversed-bending-impact loading. A cantilever specimen was fixed at one end in a vertical position and struck first from one side by one hammer and then on opposite side by the other hammer. Fifty double blows were applied per min. but Moore and Kommers (95) used a similar machine at 65 blows per min.

Findley and Hintz developed a machine which used falling balls to produce impact loading. The balls were lifted by a large wheel with pockets spaced around its periphery and deposited in a runway from which they fell to strike the specimen. Lubin and Winans (1m) used a similar machine with the ball picked up by a slotted conveyor bucket.

Brown (96) made repeated impact tests on aircraft undercarriage. Cams were used to lift and drop weights on the test member. Various bolt materials and designs were also investigated using a cam-operated weight mechanism (97). Two foreign-made repeated-impact machines, operating on a principle similar to that of Stanton, were commercially available. One of these was made by Amsler (57) and operated at 600 strokes per min. Arrangements were provided for repeated-impact tension, bending or compression tests; the other was known as the Krupp machine (98).

CONTACT-STRESS TESTING MACHINES

Contact-stress problems characteristic of design members, such as gears and ball and roller bearings, have been largely investigated on machines of special design. One common machine is similar to the one made by Amsler for wear testing in that the peripheries of two rotating cylindrical disks are held together under pressure by means of a weighted lever arm. In some machines only one disk is driven (99) while in others both are driven (100) so as to obtain the same speed or a definite slippage at the contact surfaces. In still another machine both disks are driven through eccentric involute gear wheels (101). In this way sliding both toward and away from the rolling contact area occurs on the same specimen so as to reproduce all the movements that occur between two tooth faces. Conical disks (102) were used to investigate helical gear tooth action in another design of machine.

Machines have been built for the investigation of the fatigue resistance of actual gears. The automobile industry frequently uses the "four square" principle where four pairs of gears are connected together by calibrated torque bars into a closed circuit. This arrangement simplifies the means of loading the gear although some type of dynamometer (103) may be used for this purpose.

Life testing of plain bearings and ball and roller bearings was presented before a Symposium (104) on this subject. German investigations of antifriction bearings were reviewed by Jurgensmeyer (105).

VARIABLE-STRESS-AMPLITUDE MACHINES

Little descriptive information is available on fatigue machines which automatically vary the stress amplitude during the test according to some prearranged schedule. One laboratory (1k) has modified a 10,000-rpm. rotating cantilever-beam machine accommodating 0.2-in. diameter specimens. A worm-gear drive off the motor gradually slides a weight back and forth on the weigh-bar system of the machine so as to vary the stress on the specimen. An alternate arrangement provides a cam for removing or applying a supplementary load on the end of the specimen to give a two-load cycle.

Another laboratory (106) has converted an 1800-rpm. rotating cantilever beam machine testing 2-in. diameter specimens. A compression spring, having a low stiffness constant, is used to load the end of the specimen where the spring height is continuously varied by a multi-stepped flat cam. An electric time clock with rotating-drum selector operates a solenoid which in turn moves the cam to change spring height according to some predetermined time and load cycle.

The National Advisory Committee for Aeronautics Laboratory at Langley Field has converted some rotating-beam type machines, so as to vary the applied load on the specimens which are about $\frac{1}{4}$ in. in diameter. This is accomplished by using a rotating cam to actuate one end of a pivoted beam; the opposite end of the beam applies a load on the specimen through a tension spring. This same laboratory is developing an equivalent-load fatigue testing machine to simulate stress histories similar to those encountered in aircraft parts during flight. Accelerometer records of airplanes in flight are transformed into a modulated 400 cycles per sec. sound track. This sound track is fed through an electrical system which produces signals. These signals control a hydraulic servo-mechanism so as to apply a variable load on the test specimen. Loads will be produced up to 6000 lb. at 7 cycles per sec.

TESTING MACHINES FOR STRUCTURAL AND MACHINE PARTS

The present trend toward making fatigue tests on actual design members, structural components and assemblies is not a new development (107, 108). History of testing reveals that earliest investigations of fatigue phenomena such as those by Fairbairn (86), Wöhler (7, 87), and Marten (88), and Meyer (87), used full-size design assemblies. The testing of production parts often requires special testing machines or the adaptation of commercially available equipment. Resonant machines of the centrifugal type or hydraulic operated units are capable of producing large forces and are readily adaptable to universal testing requirements.

One laboratory has a general-purpose alternating-load testing machine (109) of ±150,000-lb. capacity. It is used to test structural models and assemblies rather than material. A motor-driven variable-stroke eccentric and connecting rod operate a pivoted bell crank in the form of an inverted L. The load and deflection may be varied by changing the eccentric stroke. Speed is 10 to 250 cycles per min.

The aeronautical and automotive industries have the practice of making destructive tests on full scale components. Crankshafts (71, 72), rear axles, automobile bodies, aircraft propellers (110) and airframes (53, 111) have been investigated by vibrating them near resonance. Magnetic vibrator (29, 112), hydraulic (56), or mechanical oscillator (53, 71, 72) equipment has been employed as a force excitor. A universal type of machine, operating through a crank drive, has been built for testing various automobile parts (113). The breaking strength of gears was studied by applying alternate-bending stress through a special loading arrangement (114). Coil (115) and leaf (113) springs have been tested on special machines incorporating mechanical- or hydraulic-loading means.

The Association of American Railroads has been conducting fatigue tests for over ten years of full size car axles and crank pins up through 12-in. diameter in rotating bending (9). Fissures in rails (116) and joint-bar failures (116) have been studied by rolling loaded railroad wheels over rail structures at about 55 cycles per min. Freight car truck side frames (117) and brake beams (118) have been dynamically investigated as a means of improving their fatigue resistance.

Full size drill pipe and drill collars have been tested in single-end cantilever machines (119). Lifting gear components such as chains and hooks (120), cable (121), etc. were investigated using special testing equipment. The American Welding Society is sponsoring hydraulic fatigue tests of pressure vessels.

The German practice followed in the investigation of full size components for internal combustion engines and pressure vessels has been published in much detail (1d).

A bibliography and discussion of the centrifugal force type mechanical oscillator as used for universal testing requirements is discussed in the literature (122).

Vibration or shake tables (1d, 123) are used to test instrument assemblies, wash machines, aircraft assemblies, and other production units.

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SECTION IV—SPECIMENS AND THEIR PREPARATION[1]

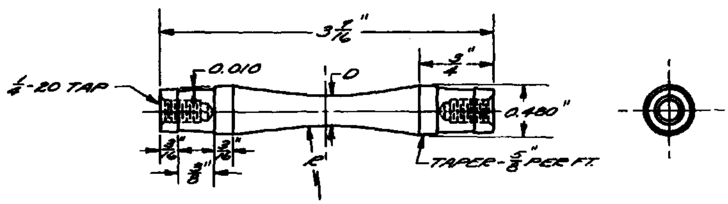
Test Specimens

Fatigue tests to determine the life of components, machines and structures are generally made on the actual parts or scaled models, or on test specimens designed to accommodate a specific type of loading. Fatigue tests to obtain *S-N* diagrams for materials are made with test specimens of relatively simple design for each product; for example, bar, sheet, tubing, and wire. Tubing and wire are usually tested with specimens of the same cross-section as the original product.

Unnotched Specimens:

The design of several test specimens for products in the form of bars and sheets are given in Figs. 9 to 15 inclusive. The simple rotating-beam fatigue specimen, Fig. 9, has been accepted and used by several laboratories, with only very minor differences in dimensions. It is used for non-metals, as well as for metals, cast and wrought. The section under test at the predetermined, calculated stress, is a narrow, cylindrical band of approximately flat contour in the middle of the beam, although, theoretically, the highest stress occurs at the section of minimum diameter.

The cantilever rotating-beam fatigue specimen, Fig. 10(*a*), has a conical section, tangent to the fillets, over which the distribution of stress is approximately uniform.



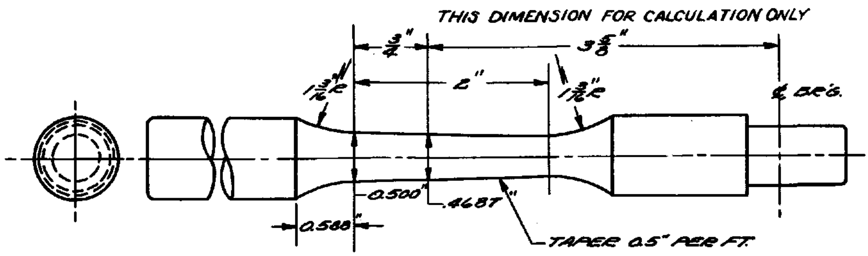
D—0.200 to 0.400 in. Selected on basis of ultimate strength of material.
R—3.5 to 10 in.

Fig. 9.—Simple Rotating-Beam Fatigue Specimen (R. R. Moore Type).

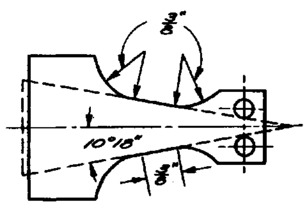
Plate and sheet bending fatigue specimens vary considerably in dimensions but usually are designed so that the load is applied at the apex of the triangle formed by extending the sides of the tapered test section, as indicated by the dash lines, Figs. 10(*b*), (*c*) and (*d*). The specimens with the larger radii at the fillets are used for soft materials. The test section is bounded by the straight, tapered sides. The beam is loaded as a cantilever.

The axially loaded specimens Fig. 11, may be gripped by external or internal threads. The test section is in the middle at the minimum diameter.

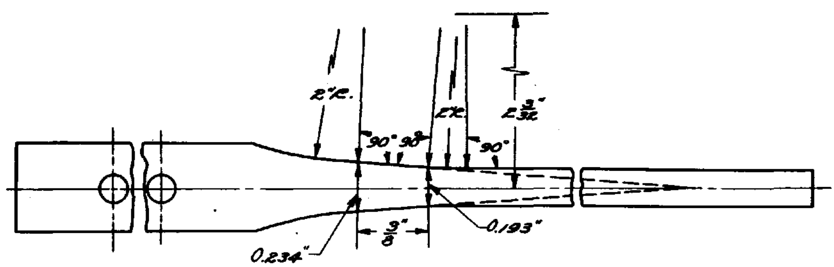
[1] Drafted by J. B. Johnson, Chief, Materials Laboratory, Engineering Div., Air Materiel Command, U. S. Department of the Air Force, Wright Field, Dayton, Ohio. (Revised following discussion by A.S.T.M. Committee E-9.)



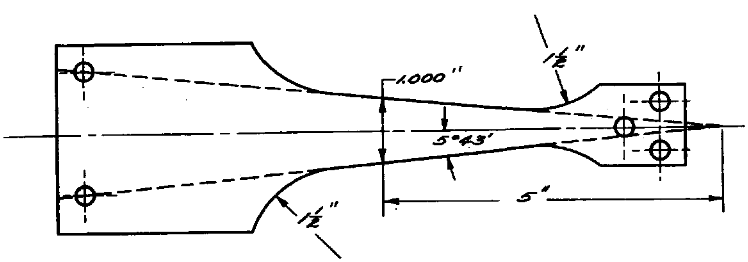
(*a*) Rotating-Beam Fatigue Specimen (McAdam Type).



(*b*) Sheet Fatigue Specimen (Bureau of Standards Type). Sheets from 0.008 to 0.031 in. gage length increased for thicker sheets.



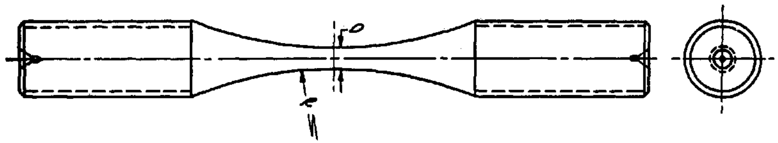
(*c*) Sheet Fatigue Specimen (Bell Telephone Co. Type).



(*d*) Plate Fatigue Specimen (Krouse Type).

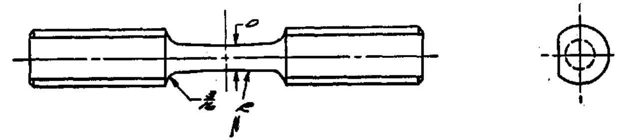
NOTE.—Specimens with longer or shorter lengths in the uniform stress section in the gage are used, depending on the thickness and ultimate strength of the material.

FIG. 10.—Cantilever Specimens.



D—Selected on basis of ultimate strength of material.
R—Three to 10 in.

FIG. 11.—Fatigue Specimen for Axial Loading (Wright Field Type).



D—Selected on basis of ultimate strength of material.
R—Five in.

FIG. 12.—Torsion Fatigue Specimen (H. F. Moore Type).

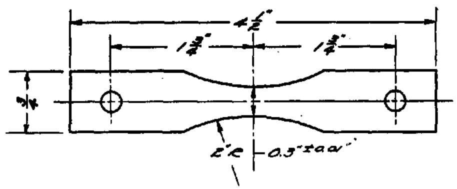


FIG. 13.—Flat Plate Cantilever Bending Fatigue Specimen for Plastic Material.

American Society for Testing Materials D 671–42 T. (University of Illinois Type.)

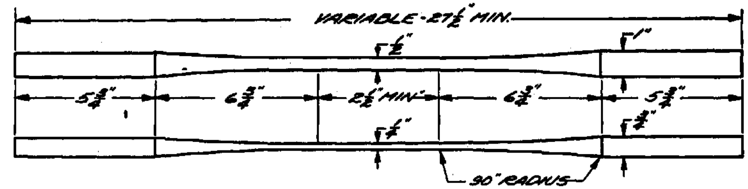


FIG. 14.—Direct Stress (Axial-Loading) Fatigue Specimens for Wood Tension Parallel to Grain. (U. S. Forest Products Laboratory.)

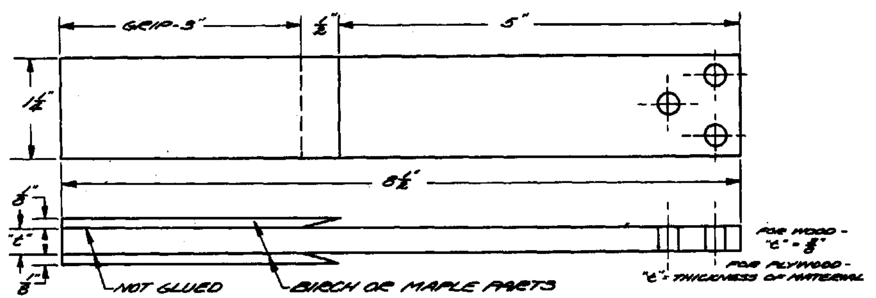
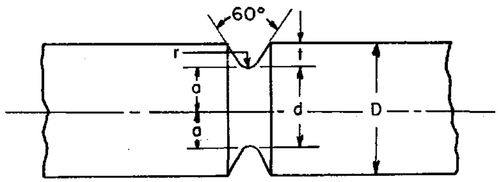


FIG. 15.—Flat Plate Cantilever Bending Fatigue Specimen for Wood or Plywood. (U. S. Forest Products Laboratory.)

The torsion fatigue specimen, Fig. 12, has a cylindrical test section tangent to the shoulder fillets.

As shown in Figs. 13, 14, and 15, the design of the test specimens for non-metallic materials may differ in some details from those for metals. However, the specimens shown in Figs. 9 to 12 inclusive are also used for non-metallics.

Radius of Curvature at Constant Depth. Angle and Width of Bar.



$D = 0.480$ in.
$d = 0.300$ in.
$d/D = 0.625$

Groove Radius, r = {0.090 in., 0.055 in., 0.020 in., 0.015 in., 0.010 in.}

r	$\frac{r}{2a}$	$\sqrt{\frac{a}{r}}$	$\sqrt{\frac{t}{r}}$	Theoretical Stress Concentration Factor	
				Axial K_t[a]	Bending K_t[a]
0.090	0.300	1.29	1.00	1.6	1.3
0.035	0.117	2.07	1.60	2.15	1.8
0.020	0.066	2.75	2.11	2.75	2.25
0.015	0.050	3.16	2.45	3.1	2.6
0.010	0.033	3.88	3.00	3.65	3.1

[a] For Poisson's Ratio $\mu = 0.3$.

$$K_t = \frac{S_{max}}{S_n}$$

$$S_n = \frac{\text{Load}}{\text{Area}} = \text{nominal stress}$$

Stress concentration factor for axial loading.

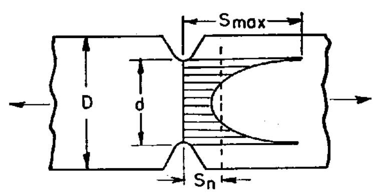


FIG. 16.—Notch Fatigue Specimens.

Notched Specimens:

Many components of machines and structures contain changes of section or contour such as fillets, grooves, holes, and threads which produce localized stress concentrations. Fatigue tests made on notched specimens, compared to those on unnotched specimens, are used as a means of evaluating the effect of these surface irregularities. Theoretical stress concentration values that encompass the range most common in engineering practice are covered by five forms of notch (Fig. 16).[2] These notches may be used in fatigue specimens subjected to any of the forms of loading previously mentioned, although most of the published data are for rotating-beam specimens. When this specimen is used with a notch it is usually machined in the form of a straight cylinder with a circumferential notch at the middle or section of maximum stress.

PREPARATION OF TEST SPECIMENS

Procedures for preparing the portion of the test specimen on which the calculated fatigue stresses are imposed require standardization to permit correlation of results between laboratories

[2] H. Neuber, "Theory of Notch Stresses, Principle, for Exact Stress Calculation," Julius Springer, Berlin, p. 181 (1937).

and to obtain comparable and reproducible values.

Cylindrical Specimens:

The surface and material directly beneath it, which may be affected by surface preparation, are controlled by the following procedures which should be followed:

1. Center ends and remove all burrs, rough machine. Center holes should be concentric to avoid eccentricity in the gage section.
2. Finish with a sharp tool and light cuts to prevent bending, overheating or cold working of the specimen. A speed of 500 rpm. with a feed of 0.0015 in. is satisfactory. For material with a hardness greater than approximately 40 Rockwell C, finish by grinding. Allow 0.003 to 0.005 in. on the diameter for polishing.

Notched Specimens:

The contour and surface of the notch requires careful control in order to obtain uniform test data, free from excessive scatter. Circumferential scratches must be avoided and the tolerance in the concentricity of the notch and the ends of the specimen must be close to control extraneous vibrational stresses. The following methods have been used for finishing a circumferential V-notch in a round specimen. A form tool having a 60-deg. included angle with a 0.010-in. radius is used in a lathe for cutting the notch in a specimen having a hardness below 40 Rockwell C. Standard abrasive thread grinding wheels are used for grinding a notch in a specimen having a hardness above 40 Rockwell C. Also standard abrasive thread grinding wheels are used for grinding a semi-circular notch in a specimen.

Specimens from Strip and Plate:

Strip and plate specimens are often tested in bending with the intent of imposing maximum fatigue stresses on surfaces of a particular nature (as-rolled, ground, sandblasted, shot peened, clad or plated, etc.). Unless study of such surfaces is intentional, however, flat specimens should also be polished. Polishing of flat specimens has not been so nearly standardized as in the case of cylindrical specimens; however, the same principles apply. It is customary to slightly round the edges (perhaps to a radius of about 0.005 in.).

Polishing:

Polishing is a cutting and not a buffing operation. The object is to remove the scratches caused by machining or grinding. The details of polishing should be considered in relation to the material concerned, the type of specimen, the kind of loading to be used, and the nature of the information sought from the test. Generally, any method which produces a smooth surface without cold-working, imposing residual stresses, overheating, or otherwise altering the material structure will constitute good polishing. All steps in the polishing procedure should be controlled with a view to producing uniformity of test conditions rather than with the thought of producing a fine finish, or highly polished, or buffed surface.

Polishing is done in successive stages with emery cloths or papers varying in fineness from No. 0 to No. 000 with a final polishing with crocus cloth, sometimes followed by rouge or suitable lapping compound. Polishing is done in a longitudinal or diagonal direction across the scratches by slowly rotating the specimen or the polisher, and simultaneously or subsequently, either one is moved in a direction parallel to the longitudinal axis of the test specimen. The surface should be free from circumferential scratches which can be seen with the unaided eye, and the depth as measured with a surface analyzer should not exceed 0.000010 in.

The detailed procedures followed by two laboratories are given.

Naval Experiment Station Procedure for Polishing Metallic Specimens Having Circular Cross-Sections:

Fatigue specimens are finished by alternate circumferential and longitudinal polishing operations. Specimens are polished while rotating in a lathe at 25 rpm. The circumferential operation is performed with a rubber disk, 4¾ in. in diameter, which is covered with canvas or broadcloth, depending on the stage of polish. The disk revolves at approximately 500 rpm. and the abrasive in suspension is applied intermittently to the periphery of the disk. The disk assembly is mounted in a framework which is attached to the lathe carriage. This frame is mounted on a vertical spindle so that it can be rotated, and is pivoted to permit raising and lowering of the disk. While operating, the disk is adjusted so that its axis makes an angle of approximately 30 deg. with the axis of the specimen. A small tension coil spring keeps the wheel in contact with the specimen.

The longitudinal polishing operation is accomplished by replacing the disk with a rubber-covered shaft ⅝ in. in diam. This spindle is covered with either abrasive paper or broadcloth, depending on the stage of polish. The spindle operates normal to the test length of the specimens at 1150 rpm. The disk and spindle speeds are obtained by the use of belt driven pulleys and a small electric motor, all mounted on the frame. The polishing disk or spindle is moved back and forth with the lathe carriage.

The polishing is accomplished in five stages of increasing fineness as in ordinary metallographic polishing. By this method the surface is made sufficiently smooth to permit examination of the structure at a magnification of 100 diameters:

Stage	Direction	Method
1......	Longitudinal	Shaft covered with No. 180 aluminum oxide cloth.
2......	Circumferential	Disk covered with canvas—polishing medium No. 600 aluminum oxide or its equivalent.
3......	Longitudinal	Shaft covered with No. 000 grit emery polishing paper washed in kerosine.
4......	Circumferential	Disk covered with broadcloth—polishing medium, levigated flour of alumina in distilled water—levigated 20 min.
5......	Longitudinal	Shaft covered with broadcloth—polishing medium, levigated flour of alumina in distilled water—levigated 40 min.

University of Illinois Polishing Procedure for Metallic Fatigue Specimens:

The polishing of round specimens is usually accomplished with the specimen rotating at a slow speed in a lathe (from 50 to 500 rpm. depending somewhat on the diameter of the specimen); the emery polishing cloth or polishing paper is wrapped around a rotating bar which is held lightly against the specimen and moved slowly with a uniform motion along the critical test portion to be polished. The rotating bar is driven at approximately 1750 rpm. by means of a flexible shaft which enables the operator to have freedom of motion in controlling the polishing. This bar, or mandrel, is held at an angle slightly less than 90 deg. to the axis of the specimen. By interchanging the position of the bar between top and bottom of the specimen (or by reversing the direction of rotation of the lathe) for each new grade of emery paper, the scratches are generated at a different angle for each new operation. It is recommended that the polishing with each new grade of paper be continued for *twice* the length of time it takes to remove visible traces of the scratches from the previous grade of paper.

In general, if a specimen has been machined with light cuts, the polishing is accomplished in three stages: (1) with No. 120X metalite cloth, (2) with No. 1 grit emery polishing paper, and (3) with No. 00 grade emery polishing paper. For large diameter specimens (those greater than about 1½ in. in diameter) in which the tool sometimes leaves a surface with somewhat rougher tool marks, an additional polishing stage is added by employing a No. 80 metalite cloth for the first operation.

By altering the angle at which the rotating bar is held with respect to the rotating specimen the scratches caused by the abrasive in the final polishing operation can be adjusted to fall along the longitudinal axis of the specimen. A liberal supply of kerosine is used on the specimen while polishing with the finer grades of paper (that is, with the No. 1 grit and the No. 00 paper). It is important that new surfaces of the polishing paper be used frequently. Change the paper or tear back to expose new areas at regular intervals in order to prevent the paper from filling up and having a burnishing effect.

The operator must take great care not to touch the polished surfaces with either his hands or with dirty rags. The final polish must immediately be protected by covering with a thin coating of vaseline (or with a corrosion inhibiting oil) applied by means of a small, clean, soft brush, reserved for that purpose.

For some of the softer metals, such as copper or aluminum, it may sometimes be found advantageous to use a final polishing operation with a

No. 0000 paper in place of or in addition to the No. 00 paper. This results in a slightly higher polish of the surface but there is danger of producing a buffing rather than polishing effect. There is no assurance that the finer scratches are any more uniform in their effects on the fatigue properties of the material than is the finish produced with the coarser No. 00 paper. The early work of Moore reported in *Bulletin 124*, Engineering Experiment Station, University of Illinois, indicated that it is somewhat doubtful whether any appreciable effect on fatigue strength is obtained by finer grades of polishing on steel than that obtained with No. 00 paper. It seems probable that the great amount of time and expense involved in polishing with finer abrasives or with rouge to obtain a metallographic finish is unnecessary for a controlled testing procedure which at best gives only relative results. The essential feature required is uniformity and reproducibility of a standard surface finish.

No specific procedures have been adopted for the polishing of rectangular specimens, but the same precautions and general sequence of operations should be used as those outlined for round specimens. In general, many of the rectangular specimens require hand operations which lead to variations from one group of specimens to the next. It is particularly important to remove any sharp fin or feather edge from the sharp corners and to minimize the tendency for final scratches to intersect the sharp corners in the critical test region.

Circumferential grooves and V-notches in cylindrical specimens may be polished in a manner similar to that described above for round specimens except that the rotating bar or mandrel is replaced by a rotating soft copper wire and the abrasive paper is replaced by a slurry of machine oil in which is suspended a fine grit abrasive powder. Two or three grades of grit are usually employed (in separate operations) finishing with a No. 600 carborundum. The copper wires are usually chosen with a radius several thousandths of an inch less than the root radius of the notch to be polished.

Electrolytic Polishing:

In some cases, final polishing of metals may be done by electrolytic removal of surface layers[3]. At present, no details of method of electropolishing can be described as typical since different metals (and sometimes different heats of a particular metal) may respond very differently to any one procedure. An important precaution is the avoidance of surface pitting or differential etching or both. An outstanding advantage is the ability to remove metal without introducing mechanical working, frictional heating, fragmentary metal particles, distorted crystal structure, or stresses. Electrolytic polishing may be adapted to a variety of specimen shapes. It affords an interesting possiblity of polishing notched specimens since even the root of a sharp notch may be so polished without cold-work.

Polishing Non-metallics:

The following procedure[4] is recommended by A.S.T.M. Committee D-20 on Plastics:

The radii are formed by milling, using a very sharp cutter and such combination of speed and feed as will give a good finish with a minimum of heating of the specimen. The test section must be polished with successively finer emery paper, finishing with No. 00 to remove all scratches and tool marks. The final polishing must be lengthwise of the specimen because even small scratches transverse to the direction of tensile stress tend to lower the fatigue strength. In order to avoid heating, all polishing must be done either by hand or with light pressure on a slowly revolving sanding drum. Care must be taken to avoid rounding the corners.

University of Illinois Procedure for Polishing Unnotched Plastic Fatigue Specimens:

In polishing unnotched plastic fatigue specimens only the critical section need be polished. Specimens with round critical sections can be polished while turning on the lathe. Specimens with square critical sections must be polished by hand, taking care not to cut the surface or excessively round the corners.

Polishing should be done with emery paper.

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[3] C. L. Faust, "Surface Preparation of Electropolishing," Pittsburgh International Conference on Surface Reactions, Corrosion Publishing Co. (1948).

[4] For details, see the Tentative Method of Test for Repeated Flexural Stress (Fatigue) of Plastics (A.S.T.M. Method D 671), 1949 Book of A.S.T.M. Standards, Part 6.

The grits used for polishing laminates should be grit Nos. 0, 00 and 000 in that order. For cellulose acetate the sequence of grit numbers should be 0, 00, 000 and 0000. The grits to be used in polish-polystyrene and their sequences hould be Nos. 0, 00, 000 and 0000. This should be followed in the case of polystyrene by rubbing with lens paper.

In polishing, all tool marks and scratches left by the preceding operation should be removed by the current size grit before proceeding to the next smaller size grit. The final strokes in the polishing operation with the finest grit should be made by hand, parallel with the specimen's axis.

PROTECTION OF TEST SPECIMENS

The surface of the test specimens may oxidize or corrode during storage or in the testing machine. A coating of mineral oil or non-corrosive grease should be applied immediately after polishing to protect the surface.

SECTION V—TEST PROCEDURE AND TECHNIQUE[1]

Upon the use of proper test procedure and exacting use of the best in technique depend the reliability of test data and conclusions drawn therefrom. This section of the Manual will discuss the appropriate test procedure and technique for fatigue tests of laboratory specimens of the type used to reveal the fatigue properties of a *material*. Of the many different fatigue machines which are in use only the rotating-beam machine will receive a complete discussion of test procedure and technique. Nevertheless, much of the discussion will apply to other types of machines and to the testing of full size parts.

Planning Tests

In planning a fatigue test program it is necessary to determine what variables are to be studied. Most variables which affect fatigue strength fall into one of the following categories: (1) material condition, (2) applied stress condition, and (3) environmental condition. Among the material conditions which may require study are: composition, structure, grain size, residual stress, and other effects of processing. Also surface treatments such as cold work, carburizing, and decarburizing may cause changes in composition, structure and residual stress whose effects may require study. The applied stress variables include: intensity of stress amplitude, intensity of mean stress, state of stress (relative values of the principal stresses), stress gradient (rate of change in stress with change in location in specimen), frequency of stress cycle, and character or shape of stress cycle. The surface finish, geometrical shape, size of specimen, or the presence of notches may cause changes in intensity of stress, state of stress, and stress gradient. Environmental conditions whose effect it may be desired to study are: temperature, relative humidity (for plastics and wood), solvents (water, oils, etc.), sunlight and other radiation (for plastics), corroding agents (water, acids, alkalies), or other chemical attack.

Of course, it may be very difficult to separate some of these variables, for example a change in size of specimen may not only change the relationship between specimen size and grain size (metals) or molecular size (plastics) or grain (wood, fabrics, etc.) but may also change the structure of the material, residual stresses, applied stress, and stress griadent. It is important, however, to be cognizant of the fact that more than one significant variable may be affected by a single change in processing the material or in the testing method.

Having decided on the variable or variables to be studied, it is then appropriate to consider the number of discrete values of each variable to be studied, the number of specimens to be tested for each change of parameter, and

[1] Drafted by W. N. Findley, University of Illinois, Urbana, Ill. (Revised following discussion by A.S.T.M. Committee E-9.)

the range of values of the variable to be covered. One of the factors which determines each of these choices is the nature of the end use for the data. When it is required to determine the effect of the variable as completely as possible the number of discrete values of the variable tested should preferably be at least five, suitable spaced throughout the range of the variable.

In instances in which an extension of the range of the variable in one direction causes conditions other than that of primary interest to be affected, it may sometimes be desirable to perform the fatigue tests in the extended range in spite of these secondary effects. For example, increasing the temperature above a certain level will cause changes in structure of the material which result in changes in fatigue strength quite apart from the effect of the temperature alone. Yet it is frequently desirable to know the fatigue strength of the material under just these conditions. It is of course desirable to obtain sufficient test data to enable separation of the effect of temperature from the effect of the change in structure, where possible.

Another example is found when the mean stress constitutes the variable under consideration. In this instance increasing the mean stress beyond a certain value may cause noticeable plastic flow to occur which may cause changes in the structure of the material, the residual stress distribution, the mean stress and the stress amplitude. In spite of these changes whose effects are difficult to evaluate, it is suggested when studying the effect of mean stress that the values of the nominal mean stress be increased at least to a stress at which noticeable plastic deformation occurs for all specimens in the *S-N* diagram at this mean stress. This suggestion is made because of the fact that it will probably be possible to analyze and interpret such data in the future.

It is suggested that at least ten specimens be tested for each condition for which an *S-N* diagram is desired. A larger number of specimens would be very desirable for establishing the *S-N* diagram accurately and indicating the variability of the material.

A complete *S-N* diagram should include as wide a distribution of tests with respect to number of cycles-to-failure[2] as possible—from 5,000 to 50,000,000 cycles may be desirable in some cases. However, many applications do not require a knowledge of fatigue behavior beyond 1,000,000 cycles, whereas others require more than 500,000,000 cycles. Some building and bridge structures, for example, experience only a few hundred thousand cycles of peak stress during their life. In still other applications fatigue data may be needed for stresses which cause failure in 500 cycles or even less. Thus the end use of the data determines the limits of the distribution of tests.

For most purposes the complete *S-N* diagram is determined by selecting stress values so as to yield a more or less uniform distribution of test points throughout the selected interval of cycles-to-failure.

When the purpose of testing is to determine the variability of a material it is suggested that at least 20 (preferably 50) specimens be prepared and tested. The specimens may all be tested at one stress level and the variability determined as the average deviation from the mean number of cycles-to-failure, or the specimens may be tested at such stresses as to yield a uniform distribution of test points along an *S-N* diagram and the

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[2] For definition of fatigue failure and determination of number of cycles-to-failure see following subsection, "Determining Number of Cycles-to-Failure," p. 56.

variability determined as the width of the scatter band formed by bounding the test points above and below with two smooth curves. It should be recognized that the deviation from the mean number of cycles-to-failure at one stress level will not necessarily be the same at another stress level.

An alternate procedure which may be useful when a large number of specimens are available is to test several specimens at each of three or four stress levels. This makes possible a statistical treatment of the data to determine the most probable value of the fatigue life at each stress level and the variability of the material as shown by the variation and standard deviation in the life of specimens tested at each stress level. In some cases the use of this procedure may necessitate running preliminary fatigue tests to determine the approximate *S-N* diagram before appropriate stress levels can be selected for the main series of tests.

For some purposes it may not be necessary to determine complete *S-N* diagrams. Instead all tests may be run with the same value of stress amplitude but distributed values of the variable being studied; and the comparison based on differences in the number of cycles-to-failure. For most studies, however, this procedure is not recommended inasmuch as the effect of a given variable on the fatigue strength is generally different for different numbers of cycles-to-failure. In fact the effect of a given variable on the fatigue strength at a large number of cycles is sometimes the reverse of the effect on the fatigue strength at a small number of cycles.

SELECTION OF FATIGUE MACHINE

Among the items which should be considered in selecting the machines for a fatigue testing program are: the type of loading must suitable for producing the desired stresses in the desired environmental conditions, the accuracy required in the data, the ease with which the machine may be operated, adjusted, and maintained, the over-all rapidity with which data may be collected, the versatility of the machine, the cost of preparing specimens, and the cost of the machine.

The most commonly used types of loading for material-type fatigue tests are bending, axial-load or torsion. The first two types of loading produce the same state of stress in smooth (unnotched) specimens; namely, one principal stress either tension or compression, and the other two principal stresses zero. This state of stress usually causes fatigue fractures along planes of maximum tensile stress for isotropic materials which have the same properties in all directions. Torsion, however, produces a state of stress in which the material is more likely to fail along planes of maximum shearing stress; namely, one principal stress tension, another an equal compression stress, and the third zero.

The versatility of fatigue machines varies considerably with type and specific design. Rotating-beam machines are inherently unsuited to tests in which the mean stress is not zero, or to tests of the original surface of materials. Unless driven from both ends, rotating beam machines have the disadvantage that the specimen must carry the torque required to overcome the friction in the outboard bearings. This condition is less important in cantilever type than in constant-bending-moment type machines, because the specimen revolves in only one bearing in cantilever machines.

Repeated-bending type machines are especially suited for tests of materials with the original surface or shot-peened or other surface conditions. They are not suitable for tests in which the mean stress is compression except when specimens notched on one side only are subjected to stresses in which the mean stress at the notch is compression, or when an unsymmetrical cross-section is used to reduce the magnitude of the tensile stress which would otherwise cause failure.

Axial-load machines are generally built heavier than bending machines for the same size specimen (owing to the relatively greater forces which the axial-load machine must supply) and require more elaborate technique in operation. They have the advantage, however, that the stress gradient over the cross-section is zero and does not change with size of specimen, which is not true of either bending or torsion tests. This has the advantage not only of eliminating stress-gradient as a variable but permits calculation of macro-stresses (stress which would exist if it were not for the minute discontinuities caused by differently oriented grains, etc.) by elementary mechanics even for strains which produce substantial plastic deformation. The rate of propagation of a fatigue crack is greater in axial-load than in bending or torsion machines. This results in a more positive determination of the end point of the tests in axial-load machines.

The construction of axial-load and torsion machines should be such that undesired stresses of unknown amount are not introduced in the test section of the specimen when the specimen is fastened to the machine. This precaution is needed since both the state of stress and the mean stress are likely to be altered by such unknown stresses. The method of gripping the specimen introduces stresses which may cause fatigue fracture to occur in the grip unless the specimen and grips are properly proportioned. Undesirable and unknown stresses may also be introduced in the test section of the specimen by the pressure of the grips if the test section is too close to the grip. Distances less than twice the diameter of the grip section from the grip should be avoided.

There are two different methods of producing the desired fluctuation of stress and strain in specimens whether they be loaded in bending, torsion or axially. The methods are constant-amplitude-of-*displacement* and constant amplitude-of-*force*. These terms apply to the deflection of, or the force developed at, some part of the actuating mechanism. The constant-amplitude-of-displacement may be produced by some form of linkage, cam, slider, or hydraulic actuator, and constant-amplitude-of-force may be produced by dead weight, spring, electro-magnet, unbalanced rotating weight, or hydraulic pulsator.

It does not always follow, however, that a constant-amplutide-of-*displacement* machine will produce a constant amplitude of strain at the significant point in a specimen throughout the life of a fatigue specimen or that a constant-amplitude-of-*force* machine will produce a constant amplitude of stress during the entire specimen life. A change in the stiffness of the specimen may be caused by the gradual development of fatigue cracks so that the *strain* in the vicinity of the cracks increases for tests in constant-amplitude-of-displacement machines and the *stress* in the vicinity of the cracks increases for tests in constant-amplitude-of-force machines as fatigue cracks progress.

Under ideal conditions these two types of machines should give identical results. There are two conditions, however, which tend to promote differences in results even for completely reversed stress cycles: (1) If the rate of crack growth is small for the loading conditions and material considered, then the inherently greater rate of crack development with constant-amplitude-of-force machines will result in a fewer number of cycles to complete fracture of the specimen. (2) If internal friction in the material causes a temperature rise in the specimen sufficient to change the modulus of elasticity of the specimen (true of many plastics) the effect is similar to that described above when the stiffness is changed by crack growth. The decrease in stiffness produces a small increase in amplitude of strain in the specimen (for constant-amplitude-of-displacement machines), the amount of which depends on the flexibility of the machine and on the portion of the specimen in which the temperature did not increase. The amplitude of stress decreases an amount equal to the difference between the effect of the increased strain and the decrease in modulus of elasticity.

On the other hand for a constant-amplitude-of-force machine the decrease in stiffness of the specimen resulting from a temperature rise produces an increase in amplitude of strain proportional to the change in stiffness. The amplitude of stress remains unchanged provided: the change in amplitude of deflection of the machine does not change the amplitude of force or the relation between stress and applied force, provided the stress is below the new proportional limit, and provided fatigue cracks have not decreased the cross-section of the specimen and introduced stress concentrations. See also Section III, Fatigue Testing Machines.

Differences between the usual operating speeds of fatigue machines are not usually important in ordinary fatigue testing of metals. However, under very slow operating speeds or stress cycles in which the mean stress is not zero creep or relaxation will affect the fatigue results when the temperature is high enough for noticeable creep or relaxation to occur. The lowest temperature at which creep or relaxation is important may be anywhere from below room temperature to 1200 F. or higher depending on the material. If the mean stress is produced by a constant force, creep will occur, but if it is produced by a constant deflection, relaxation will occur. The results of fatigue tests will therefore be different for the two different methods of producing the mean stress.

There are two conditions under which the speed of testing may be expected to have a pronounced effect on test results. As mentioned above, when the mean stress is not zero and the temperature is high enough for creep or relaxation to occur, either the mean stress or mean strain (depending on the type of machine) will change with time. Since the change in mean stress or strain occurs with time rather than number of cycles the amount of change occurring during a given number of cycles in a test having a given initial mean stress and alternating stress amplitude will be different for different speeds of testing.

Similarly in corrosion fatigue tests the effect of corrosion is a function of time rather than simply the number of cyles so that the test results will be influenced by the speed of testing.

The speed of testing of some machines can be varied over wide limits. However, many others are inherently constant-speed machines. This is particularly true of any type of machine when designed to operate at resonance. In selecting a fatigue machine for a given application it should be borne in mind that higher testing frequencies promote higher specimen temperatures and that increasing the testing frequency beyond a certain point may not increase the rapidity of data collection appreciably since the time required to set up the specimens may be a substantial portion of the time required to obtain a complete *S-N* diagram.

The accuracy of fatigue machines

depends not only on the original construction but also upon maintenance of the machine and upon the technique used.

SELECTION OF SAMPLE AND PREPARATION OF SPECIMENS

The selection of the sample from which fatigue specimens are to be prepared is a step in the procedure whose importance cannot be over-emphasized. The significance of the test results depends on the care used in selecting the sample.

Before the sample is selected it is necessary to know the purpose of the tests. If the object of the test is to determine the variability of a material then one or more specimens should be taken from a representative location in each of several different bars, plates, moldings, castings, or other forms whose variability is to be studied. The plates, bars, etc., from which specimens are prepared should be taken from different billets, heats, etc., at random.

If the object of the tests is to determine the effect of some parameter such as mean stress, test temperature, etc., then all other variables should be excluded as nearly as possible and the sample selected should be as uniform and free from defects as possible. With this objective it is desirable that all specimens be obtained from one heat of steel or one sheet of plastic, etc.

Some of the things to look for in appraising a sample are: homogeneity of composition, micro-structure, and grain size; isotropy of mechanical properties; freedom from segregation, micro-cracks, residual strains, inclusions, delamination (plastics), knots, unfavorable orientation of grain (wood), etc.

The machining and polishing of specimens should be done in accordance with the discussion in Section IV, Test Specimens. Three things are to be avoided if possible: (1) overheating the specimens during machining, (2) cold-working the material at the surface of the specimen, and (3) repeatedly stressing the specimen during machining by excessive vibration. The order of machining operations is sometimes important. When specimens are prepared on a lathe, it is suggested that the test section of the specimen be machined last and that the feed be toward the head stock in order to minimize the chance of repeatedly stressing the material in the test section of small specimens. If the specimens are stress relieved or otherwise heat treated after machining, care should be exercised to avoid such defects as decarburization and oxidation, unless these are the variables under test.

Test specimens for rotating-beam machines must be prepared very carefully to insure that the grip ends are accurate and the axis of one end coincides with the axis of the other so that the specimen will run true in the machine. Also the test sections should be round—not elliptical in cross-section. Non-circular test sections make stress determination for rotating-beam and torsion-fatigue tests particularly uncertain. Care must be exercised in the preparation of specimens for axial-load fatigue tests to insure that the test section is concentric with the grip ends of the specimen. Otherwise appreciable bending stresses may be introduced due to eccentric loading.

It is especially difficult to machine small diameter specimens concentric owing to their flexibility. When turning such specimens the chief source of trouble is the center holes in the specimens. A satisfactory procedure is to drill the center holes in the bar stock, mount between centers on a lathe, machine the bar to a cylindrical shape, grip the machined blank in a collet and *machine* the center holes, then mount between accurately ground and aligned centers to finish-machining the specimen.

Of course concentricity of the specimen parts does not insure freedom from unwanted stresses from other sources such as misalignment or wear of the grips.

MEASURING SPECIMENS

The precise measurement of specimens and determination of the necessary accuracy is not as simple a procedure as may appear at first glance. The object of measuring the specimen is to provide data from which to calculate the load, bending moment or twisting moment needed to produce the desired stresses.

If we set as our goal the calculation of the desired load, moment or torque to an accuracy of let us say $\pm$ 1 per cent (permissible relative error) then the precision of the measurements required can be determined. The equations for load, moment, or torque for a specimen in the elastic condition under axial loading, bending or torsion respectively may be written as follows:

$$P = AS = kL^2 S \ldots\ldots (1)$$

$$M = \frac{I}{c}S = kL^3 S \ldots\ldots (2)$$

$$T = \frac{J}{c}S = kL^3 S \ldots\ldots (3)$$

where:

P = load,
A = area of cross-section,
S = desired stress,
k = dimensionless constant depending on the shape of the cross-section of the specimen and the type of loading,
L = a dimension of the cross-section of the specimen,
M = bending moment,
I = moment of inertia,
c = distance from neutral axis, to extreme fiber,
T = torque, and
J = polar moment of inertia.

The relative error in multiplication is approximately equal to the sum of the relative errors of the numbers for small errors (1).[3] Thus for Eq. 1, 2, and 3 above if S and k are considered to be exact, then the relative error of measurement of each dimension (L) must not be greater, than the permissible relative error of the product (P, M, or T) divided by the number of dimensions (L's). For instance in the axial-load tests, Eq. 1, each dimension of the cross-section of the specimen must be measured so accurately that its relative error is less than $\frac{\pm 1 \text{ (per cent)}}{2 \text{ (L's)}} = \pm \frac{1}{2}$ per cent if the load desired is to be known within $\pm$ 1 per cent. Also for bending and torsion each dimension of the cross-section must have a relative error less than $\pm \frac{1}{3}$ per cent if the desired moment is to be known within $\pm$ 1 per cent.

Thus for bending fatigue tests of small specimens such as thin sheets the measuring instruments must have considerable precision.

However, when one dimension can be determined within a smaller relative error than the other, then a larger relative error can be tolerated in the remaining dimension than in that of the first. It is only necessary that the sum of the relative errors in all numbers being multiplied be equal to or less than the allowable error in the product. This is particularly helpful in tests of thin sheet (especially under axial-loading) since the width of the sheet can be measured within a smaller relative error than the thickness, with instruments having the same absolute error.

When round specimens are used for axial-load or torsion or rotating-beam fatigue tests it is suggested that two mutually perpendicular diameters of the specimen be measured and that the average of these measurements be used

[3] The boldface numbers in parentheses refer to the list of references appended to this section, see p. 63.

in calculating the load or moment. But when round specimens are used in repeated-bending fatigue machines the diameter d_1 in the plane of bending is of greater importance than the diameter d_2 in the neutral plane. Differences between these two diameters may be approximately accounted for by using the following equation in computing the bending moment:

$$M = S\frac{I}{c} = \frac{S\pi}{32}\, d_2\, d_1^2 \ldots\ldots (4)$$

However, for most purposes it is sufficiently accurate to simply use the diameter in the plane of bending with the ordinary equation. Except for axial-load fatigue tests, it is suggested that round specimens not be used if they deviate more than 1 per cent from true round since the ordinary equations for stress are then of doubtful accuracy.

The accuracy with which the forces can be applied to a specimen should be considered together with the accuracy with which the load can be calculated, that is, the accuracy of measurement of the specimen. For example if the permissible relative error in the calculation of the stress is $\pm\ 1\frac{1}{2}$ per cent, then, considering the worst possible combination of errors, the sum (without regard for sign) of the relative error in calculating the desired load as described above and the relative error involved in applying the load, must not be greater than the permissible value $\pm\ 1\frac{1}{2}$ per cent. If the load can be applied within say $\pm\ \frac{1}{2}$ per cent, then the relative error in the load calculation must not be more than, $\pm\ 1$ per cent, but if the relative error in load application may be as large as say $\pm\ 1$ per cent then the relative error in the load calculation must not be more than $\pm\ \frac{1}{2}$ per cent.

The concentricity of the test section compared to the grip ends should be determined for axial-load fatigue specimens. This is most conveniently done by mounting the specimen freely on centers and rotating it in the field of an optical comparator. The amount of permissible eccentricity, e, in round specimens may be calculated approximately by comparing the bending stress due to the eccentricity with the uniformly distributed stress due to the axial load. This comparison yields the following relation:

$$e = \frac{d}{800}\,\Delta S \ldots\ldots\ldots (5)$$

where:
d = diameter of the specimen and
ΔS = allowable error in stress expressed in per cent.

Care should be exercised when measuring specimen dimensions with micrometers to avoid the burnishing action which such measurement may have on the specimens and to avoid the possibility of scratching the specimen. A scratch often will act as a stress-raiser thus altering the test results. Instead of micrometers, traveling microscopes or optical comparators may be used to good advantage, especially for measurement and examination of notched specimens. When micrometers are used they should be equipped with ball contact points with radii less than the radius of the curved surface to be measured.

In measuring materials having a low elastic modulus, such as plastics, it is important to note that errors may be introduced with a micrometer by squeezing the material if more pressure is applied than just enough to insure contact.

When fatigue tests are conducted at extreme temperatures it may sometimes be desirable for research purposes correct the dimensions of the specimen for thermal expansion or contraction when the specimens are measured at room temperature. For example, any dimension of a steel specimen at 1600 F. is about 1 per cent larger than the same dimension at room temperature. Thus a bending moment calculated from the dimensions at room temperature would be about 3 per cent too low for a test of steel at 1600 F.; similarly an axial load would be about 2 per cent too low. For ordinary engineering purposes, however, it is not customary to make this correction.

STRESS OR STRAIN DETERMINATION

Quantities to be Determined From the Stress Cycle to Measure the Severity of the Stress Condition:

To define the severity of the stress condition, it is necessary to determine: (1) the mode of variation of the cycle of stress or strain, (2) the intensity of stress, strain, or internal potential energy values, at particular locations in the stress cycle, and (3) the state of stress existing during the stress cycle.

The variation of stress with time in a machine or structure may be very irregular. In laboratory fatigue testing, however, it is usual to apply a stress cycle which approaches a sinusoidal variation with time. When the mode of variation of the stress cycle is sinusoidal it is only necessary to determine any two of the following four quantities, together with the testing frequency, in order to completely define the stress cycle (or strain cycle): alternating stress (or strain), mean stress (or strain), maximum stress (or strain), or minimum stress (or strain).

The state of stress is determined by calculating the magnitude and sense of the three principal stresses at the particular point in the specimen and the particular portion of the stress cycle under consideration. While it is not always necessary to explicitly state the three principal stresses, sufficient information should always be given from which the three principal stresses can be determined. In unnotched fatigue specimens the state of stress can usually be determined if the type of loading to which the specimen is subjected is stated, as for example, axial load, torsion, bending, or a stated combination of bending and torsion.

All of the above stress conditions must be stated in reporting fatigue tests since it is known that each of them affects the results obtained.

Stresses Determined by Calculation from Forces:

Stresses cannot be measured directly and are usually determined from applied forces by calculation. The equations used for calculating stresses from applied forces in simple members such as used in laboratory fatigue tests are given in numerous texts and handbooks for unnotched members under axial, bending, and torsion loads which produce stresses within the proportional limit.

Stresses in notched members, that is, members with holes, fillets, grooves, and so forth, are usually determined by first computing the nominal stress from the elementary formula. In making these computations, the dimensions of the cross section of the part are considered to be those of the net section. Because of the change in stress distribution resulting from the presence of notches the stress values calculated by the simple formulas are *nominal stresses*, and in general, give stress values which are lower than those actually existing at the highest stressed point in the cross section.

The stress at the highest stressed point in members having notches of various types may be calculated by following the procedure outlined in the Appendix at the end of this section.

Strain Determined by Direct Measurements of Local Deformation:

While stresses cannot be measured directly, strain may be so measured with reasonable precision.

Instruments used for measuring strain are called strain gages and may be classified in various ways. For the present purposes we may first classify them in two groups according to whether they are suitable for strain measurement under static conditions or suitable for strain measurement under dynamic conditions. In an irregularly shaped member the strain may vary appreciably from point to point. Since a strain gage either measures the change in length between two particular points a finite distance apart or averages the strain over a finite distance, the utility of a strain gage for such application depends not only on the ease with which it may be operated and the precision of the measurement, but depends also upon having as short gage length as possible Unfortunately, as gage lengths are decreased, both the precision of the measurements and the ease with which the instruments may be manipulated decrease.

There are four general methods of strain measurement (2) in common use today: mechanical, optical and electrical strain measuring instruments, and brittle coatings. Each of these methods can be used either for static strain measurements or for dynamic strain measurements if the proper instruments are selected. The mechanical gages include instruments having systems of multiplying levers (the Huggenberger) (3), and instruments having multiplying levers or mechanical dials (the Berry and Whittemore) (4).

Chief disadvantages of these gages are relatively long gage length ($\frac{1}{2}$, 2, and 10 in., respectively), friction, and various difficulties involved in the means of contact between the gage and the specimen. Strain gages operating on optical principles include Tuckerman (5) strain gages, interferometer (6) types, and the General Motors (7) or Gadd gage. The chief disadvantage of the Tuckerman gage lies in its 1-in. gage length. The disadvantages of the interferometer types are the extreme sensitivity which make them especially vulnerable to vibration and the fact that they must be continuously observed during straining of the specimen. The chief disadvantages of the General Motors strain gage, which has a gage length as short as $\frac{1}{16}$ in., are the manual skills required to manipulate the instrument and slippage and flow of the material at the point of contact between the gage and the specimen.

The currently important electrical strain gages are of two types: the magnetic strain gage and the resistance-strip strain gage. The strain sensitive element in the latter may be either a carbon strip (8) or a very fine wire (8, 9). The magnetic strain gages (8, 10, 11) are limited to applications in which they can be securely screwed to a relatively stiff specimen. The resistance-strip strain gages are probably the most versatile and most widely used strain gages at the present time. Disadvantages of the resistance-strip strain gages are: the small magnitude of the change in resistance encountered which calls for resistance measuring equipment of extreme precision; sensitivity of the strain gages to strain crosswise to the axis of the gage; possible relaxation of the adhesive used in cementing the gage to the specimen or lack of adherence of the gage to the specimen; and the impossibility of direct calibration of an individual gage in the case of the wire grid type. The carbon-strip gages have a greater change in resistance for a given change in strain but also have a greater sensitivity to strains crosswise to the axis of the gage, which is a disadvantage.

Brittle coatings (12) are useful in surveying members to locate points of highest stress, but have the disadvantage of being sensitive to changes in temperature and humidity and of yielding results of a low order of precision. They probably have the shortest gage length of any strain measuring device, practically zero.

Strain measurements may be made under dynamic conditions (13) as in a fatigue test by all of the methods of strain measurement enumerated for static tests. However, the conditions imposed by dynamic conditions render many of the instruments enumerated above unsuitable for dynamic strain determination. A mechanical instrument suitable for dynamic strain measurement is the DeForest scratch extensometer (14). Disadvantages of this instrument are that it must be screwed to the work, it is not direct indicating, indicates maximum and minimum strain only, and has a gage length of 2 in. The Tuckerman optical strain gage is suitable for dynamic measurements when the frequency of the strain cycle is not too high and the mass of the instrument not too important.

The electrical gages of both magnetic and resistance types are suitable for dynamic strain measurements. Of these the resistance-strip gages are the most versatile and probably the most widely used. The use of resistance-strip strain gages in measurements of strain on vibrating or reciprocating members involves no difficulty other than those of accurately measuring small changes in resistance and indicating or recording these changes faithfully. However, when strains in a rotating member are to be determined slip rings and brushes must be used to connect the strain gage or gages to the resistance measuring equipment. Such slip rings must be of good quality since variations in contact resistance between the slip ring and brushes would distort the indicated strain reading.

NOTE.—A suitable material for slip rings is sterling silver and a suitable material for brushes is National Carbon Co. grade 619 carbon.

Brittle coatings may be used for strain measurements under dynamic conditions (15). As in the case of static strain measurements, brittle coatings are useful for determining the location of regions of highest stress or strain, but the quantitative measurement of strain is not very accurate by this method.

Stresses Calculated from Strains and Vice Versa:

Often it is desired to calculate stresses from measured strains or to calculate strains from computed stress values. Such calculations are readily accomplished when the stresses and strains are within the proportional limit. Formulas for calculating stresses from strain measurements and vice versa are to be found in numerous publications (16, 17, 18, 19), together with special formulas, (19, 20, 21, 22) slide rules, nomographs, (23, 24) graphical procedures (25) and machines (26, 27) for computing stresses from three or more intersecting gage lines.

Stresses Determined by the Use of Models or Analogies:

Stresses in members or specimens of irregular shape may be determined in many cases by the use of models or analogies. Among the most important of these are the photoelastic method, the membrane analogy, and the electrical analogy. The photoelastic method is described in detail in books by Coker and Filon, (28) and by Frocht (29). The photoelastic method has been widely used particularly for determining stress-concentration factors. It also offers a means of determining stresses at the interior of three-dimensional models.

The membrane analogy has been used to determine stresses in shafts in torsion and beams in bending and is discussed for these cases by Timoshenko (16). Measurement of electrical potential (16) has been used to determine stress distributions in a few instances.

Stresses Beyond the Proportional Limit:

Stresses beyond the proportional limit may be calculated from applied loads or moments for certain cases such as axial-load, bending, and twisting. In unnotched cylindrical members subjected to axial loads, stresses beyond the proportional limit may be calculated by dividing the applied load by the cross-sectional area existing at that load. Stresses produced during the first loading beyond the porportional limit may be calculated by a more involved procedure reviewed by Nadai (30) for pure bending, and for torsion of a circular cross-section member, when the stress-strain relationship is known.

The methods reviewed by Nadai are based on the assumptions: that the relationship between stress and strain in a member in torsion or bending is the same as the relationship observed for homogeneous tension or homogeneous shear; that the strain in a plastically bent beam is proportional to the distance from the neutral surface; and that the strain in a shaft in torsion is proportional to the distance from the center of the shaft. Experiments have shown these assumptions to be reasonably accurate for some of the simple shapes.

Residual stresses resulting from bending or twisting beyond the proportional limit can be calculated from principles reviewed by Nadai (30) and discussed in detail by Sidebottom (31) for bending. These calculations are based upon the same assumptions outlined above for stresses beyond the proportional limit plus the additional assumption, also verified by experiment, that upon unloading a bent or twisted member the decrease in stress is a linear function of the decrease in strain with the same modulus of elasticity as that exhibited below the proportional limit during the first application of load.

Unknown residual stresses may also be calculated approximately from the distortion of the crystal lattice observed by X-ray defraction patterns (32). For certain types of members, residual stresses may be calculated from observations of strains and deflections of the member when it is mutilated (33) by peeling off the outside surface (34), boring out the interior (35), or sawing it into strips (31), etc. Relatively large residual stresses may also be measured approximately by coating the surface of the parts with a brittle lacquer and then drilling a hole into the surface (36). The relief of stresses caused by drilling the hole produces cracks in the brittle lacquer characteristic of the type of residual stress present before drilling.

CALIBRATION OF EQUIPMENT

Fatigue-testing machines should be calibrated. Those types of machines which involve springs or electrical controls should be calibrated frequently. In some machines the mechanism for applying the load, torque, or bending moment requires calibration while in other machines the strain or the deflection-sensing mechanism requires calibration.

In all fatigue-testing practice it is most desirable to calibrate the machine under operating conditions, that is, under dynamic conditions. However, this procedure is always difficult, and in some machines, under certain conditions, it may be better to use a static calibration procedure than a dynamic calibration of questionable reliability.

Static Calibration:

Rotating-beam fatigue machines may usually be calibrated statically. In almost all machines of this type the loads are applied by means of dead weight, either by direct application of the weight to the end of the specimen in a cantilever-beam machine, or to the center of an equalizer link in a constant-bending-moment type machine.

In other machines of the rotating-beam type, the loads maybe applied through a lever and poise mechanism. Calibration of such machines should include accurate weighing of all levers, and other parts of the loading system, together with experimental determination of the center of gravity of these parts. This information together with the dimensions of the lever system can be employed to calculate a calibration constant or to construct a calibration chart relating the applied weight or position of the poise to the load, bending moment, or torque applied to the specimen. One should make sure that the calculations compensate for the tare weight of the overhanging portion of the specimen and its extension in cantilever-beam fatigue machines. In the case of four-point loading, that is, constant-bending-moment machines, one should make sure that the effective weight or bending moment produced by each of the bearing housings with shaft assembly, one-half specimen, etc., are equal, or that their effect has been entirely removed by counter-balancing each of these assemblies separately.

With some lever-and-poise-loaded machines calibration may be accomplished by placing a weighing scale in such a position as to receive the direct thrust from the lever system to the specimen and comparing the indicated load at different settings of the poise with the load measured on the scale. The tare introduced by the weight of the unsupported portion of the specimen and other parts of the machine must be taken into account.

In large size rotating-beam machines, axial-load machines, and other types of fatigue machines, spring dynamometers are frequently employed to indicate the magnitude of load applied to the specimen. Such dynamometers may be calibrated by direct application of weight or by suitably mounting the dynamometer in a universal testing machine. Care should be exercised in applying loads to these dynamometers to insure: that they are loaded in the same manner as they are loaded in the fatigue testing machine; that the method of applying the calibration load does not change the effective stiffness of the dynamometer; and that unknown side thrusts are not applied to the dynamometer during calibration. It is always necessary to take into account the tare weight of parts of the fatigue-testing machine lying between the dynamometer and the specimen (including that portion of the specimen lying between the dynamometer and the test section of the specimen). The effect of this tare weight may frequently be compensated by shifting the zero of the dynamometer dial indicator. In case the dynamometer is placed in the fatigue machine in such a way that other springs such as flexible guide links are in parallel with the dynamometer, the effect of the stiffness of these springs must be taken into account or shown to be negligible.

Dynamic Calibration:

Two methods may suitably be employed for dynamic calibration: (1)

direct measurement of deflections of parts of the machine and (2) measurement of strain by resistance-strip strain gages.

To calibrate a fatigue machine by direct measurement of deflection the machine may first be represented as a system of springs, rigid members and rigid connecting links. The accuracy of the calibration will depend on the accuracy with which the machine may be represented by this schematic approximation. In general the large, stiff parts of the machine may be considered rigid, relative to the specimen, flexible connecting pivots, spring dynamometer, ball bearings, etc. which act as springs.

It is then necessary to devise a means of measuring the relative movement between the various parts of the machine and its frame in such a way that the deflection of the specimen (that is, the angular or linear displacement of one end with respect to the other end) is determined. This may usually be accomplished by mounting a micrometer, micrometer microscope or dial gage to the frame of the machine by a stiff bracket so that the deflection of the parts of the machine may be determined under both static and dynamic loads. It is essential that the mounting be proportioned and damped so that its vibration is negligible at the frequency for which the machine is to be calibrated.

Measurements of amplitude of motion with a dial gage or micrometer during operation of the fatigue machine are made by gradually approaching the vibrating member with the plunger of the dial or micrometer until contact is made. Contact may be established by a combination of feel, sight, and sound; or electric contact indicating devices may be used. Due to the very short duration of contact allowed for accuracy when machines operate at relatively high speeds, both techniques may be difficult to apply satisfactorily.

When corresponding values of static and dynamic deflection of appropriate parts of the machine have been determined for several different amplitudes of load or deflection of the machine, the values should be related to the strain (or deflection) of the specimen by means of the observed geometry of the deflected parts. The ratio of the strain (or deflection) of the specimen under dynamic conditions to that under static conditions may then be determined for each amplitude of the machine and a calibration curve plotted. This ratio will not always be independent of the amplitude of the machine owing to non-linear deflection characteristics of ball bearings and some bolted connections.

The method of dynamic calibration outlined above has the disadvantages that: strain in the specimen is not measured directly; the small changes in amplitude of deflection are difficult to measure with precision; the flexibility of the nominally rigid parts may not be negligible; and the distribution of the dynamic deflection along the specimen may differ from the static distribution.

Resistance-strip strain gages, such as the wire gage, provide a means of calibration by direct measurement of strain in or near the critical section of the test specimen. Unfortunately presently available gages may fail by fatigue in a few thousand cycles at strain amplitudes as low as 0.002 in. per in. However, calibration at higher amplitudes of strain may be accomplished by mounting the gage on a portion of the specimen where the cross-section is larger than at the critical section.

The output of resistance strain gages is measured by means of a Wheatstone bridge (8) which may be either an a-c. or d-c. bridge. However, a d-c. bridge is satisfactory when used with ordinary a-c. amplifiers only within a frequency range of about 50 to 50,000 cycles. The gain of ordinary amplifiers is usually not constant for frequencies below this range. Since much of the currently available fatigue equipment operates at 30 cycles per sec. (1800 rpm.) ordinary a-c. amplifiers are usually not satisfactory unless used with an a-c. Wheatstone bridge. Procedures for calibrating fatigue machines using an a-c. bridge circuit have been described (37, 38). Such circuits employ an oscillator supplying a high frequency carrier voltage to opposite corners of a Wheatstone bridge.

One side of the bridge is the active strain gage cemented to the specimen in the testing machine. In an adjacent side of the bridge an inactive gage of the same type is placed. This gage should be cemented to a piece of the same type of material and same size as the specimen and subjected to the same ambient temperature as the specimen in order to provide compensation for ambient temperature and temperature due to electrical resistance of the gages. This procedure does not correct for the effect of temperature created in the test specimen by hysteresis.

To calibrate a fatigue machine dynamically, the maximum and minimum strain in the specimen for static load or deflection settings of the fatigue machine are measured. Then with the fatigue machine in operation the magnitude of the maximum strain and the minimum strain are obtained. Considerable experience and good judgment are required to make these measurements with adequate precision.

The ratio of the range (maximum minus minimum) strain under dynamic conditions to that under static conditions may then be computed for each amplitude setting of the fatigue machine and a calibration curve plotted as in the previous method of calibration.

It is essential that the electrical circuit itself be calibrated against known strains to make sure that the circuit and observational technique are not frequency sensitive. Specifically it should be determined that the same strain amplitude is indicated by the electrical circuit for the straining frequency used in the fatigue machines as for the identical strain applied under static load. A dynamic calibrator which may be used for this purpose has been described by Ball (39).

MANIPULATION OF MACHINES

A procedure which may be followed and precautions to be observed in using rotating-beam fatigue machines in testing round specimens is described together with general precautions for other type fatigue machines.

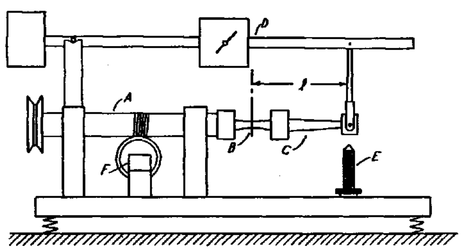


FIG. 17.—Cantilever Type Rotating-Beam Fatigue Machine.

A rotating cantilever-beam fatigue machine is shown diagramatically in Fig. 17. It consists of a motor-driven spindle, *A*, to which the specimen, *B*, is attached co-axially by means of a split collet, draw-in bolt or other suitable means. A shaft extension, *C*, is fastened to the outer end of the specimen by means of another split collet and the specimen is bent downward by a load applied to the end of the shaft extension, *C*. The load is produced by a beam and poise mechanism, *D*.

A pure-bending type rotating-beam fatigue machine is shown diagramatically in Fig. 18. This machine consists of two spindles contained in housings, A and B. The spindles are connected co-axially to the ends of specimen, C, by split collets, draw-in bolts or other suitable means. This assembly is supported flexibly on the posts, D and E, and weight, F, is applied through links, G and H, in such a way that equal bending moments are applied to both ends of the specimen. A motor, I, rotates the spindles, specimen and revolution counter, J.

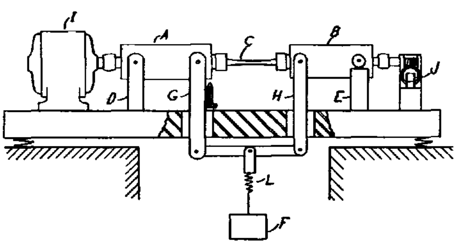


FIG. 18.—Pure-Bending Type Rotating-Beam Fatigue Machine.

For a general discussion of fatigue machines see Section III of this Manual, p. 6.

Procedure:

1. Measure the diameter, d, of the specimen using the procedure previously described in this Section (see p. 44). See Section IV, p. 30 for a general discussion of specimens and their preparation.

2. On cantilever type machines measure the bending moment arm, Fig. 17, that is, the distance from the point on the end of the shaft extension at which the load is applied to the center of the test section of the specimen. It is most convenient if this distance is kept constant by machining the specimens so that the ends are all the same distance from the center of the specimen, and the end of the specimens are seated against a shoulder in the collet each time.

3. Determine the amplitude of stress S_a to be used. It is usually best practice to start with a relatively high stress (well above the fatigue strength at 10^8 cycles) and reduce the amplitude of stress by small increments in succeeding specimens. It should be borne in mind that specimens tested at stresses substantially below the fatique strength at 10^8 cycles contribute nothing to the determination of fatigue characteristics, so that the time required for such tests is wasted. If nothing is known about the probable fatigue strength of the material it will usually be safe to use a value of $\frac{2}{3}$ the ultimate strength of the material in tension as the stress in the first fatigue test.

4. Compute the value of $\frac{I}{c} = \frac{\pi d^3}{32}$ for the round specimen.

5. Compute the load, $P = \frac{S_a}{l} \cdot \frac{I}{c}$ to be applied to the specimen.

6. Clamp the specimen to the shaft extension, C, and spindle, A, Fig. 17, or to the spindles, A, and B, Fig. 18, by means of the split collets or other fastenings provided. Rotate the spindle slowly to determine whether the specimen and extension shaft run true. If the specimen and extension shaft do not run true, loosen the collets, rotate the specimen relative to the collets a quarter turn or so, retighten the collets and try again. The specimen (on pure-bending type machines) or the end of the shaft extension, C (on cantilever type machines), should run true within about 0.005 in. or less, otherwise excessive vibration may result if the machine speed is at or below the critical speed of the shaft and the mean stress will not be zero if the machine operates at a speed above the critical speed for the shaft.

7. Set the poise or add weights to produce the desired stress in accordance with following subsection, "Precautions for Rotating-Beam Machines," p. 55.

8. Record the initial reading of the revolution counter, *F*, Fig. 17; *J*, Fig. 18.

9. Start the machine. If the specimen being tested is quite flexible the end of the shaft extension, *C*, Fig. 17, or links, *G* and *H*, Fig. 18, should be clamped or held with the fingers while the machine is gaining speed. If this is not done, the specimen may be damaged by severe vibration if the machine passes through a critical speed during starting.

10. Set the electric cutoff, *E*, Fig. 17, *K*, Fig. 18, so that it will stop the machine when the specimen has fractured. For specimens of large diameter or for some fiber-filled plastics it is desirable to adjust the electric cutoff so that the machine will stop when the deflection of the specimen has increased by a certain percentage (say 25 per cent) of the initial deflection caused by the applied load. This procedure is desirable because of the wide differences in rate at which the fatigue cracks progress in different specimens after the cracks have been initiated. These differences in rate of crack propagation cause differences in the number of cycles-to-failure which are not directly related to the number of cycles required to initiate the fatigue crack. For a general discussion of this problem, see following subsection, "Determining Number of Cycles-to-Failure," p. 56.

11. When the specimen has failed, record the final counter reading and compute the number of cycles sustained. Make sure that the date, material, specimen number and name of operator have been recorded.

12. Test another specimen using a somewhat lower value of stress but the same procedure as outlined above, and repeat until enough data have been collected to define the *S-N* diagram as described in preceding subsection, "Planning Tests," p. 38.

13. When the *S–N* diagram is not clearly or continuously defined by results of preceding tests, other stress values higher than the last one used may be inserted between the initially selected stress values to aid in completing the diagram.

General Precautions:

While mounting and adjusting the specimen, care should be exercised to avoid scratching or other marking of the specimen with wrenches or other objects since such marks may alter the fatigue strength of the specimen. Touching the test section of specimens with the fingers should also be avoided as this is very likely to cause corrosion with consequent alteration in fatigue strength.

All fatigue-testing machines should be mounted on suitable springs, rubber mounts, or cork, in order to isolate them from the vibration of surrounding machines. Extraneous vibration or vibration "beats" are an indication that the isolation is inadequate.

All bearings in the machine should be maintained in good order: they should turn freely and without evidence of roughness; the races of anti-friction bearings should fit tight enough so that they do not turn in the housing or on the shaft; and the looseness of the bearings should be very slight.

Check the machine for excessive vibration. Excessive vibration may be reduced by changing the relation between the operating speed and the natural frequency of the machine by means of some of the following expedients: increase or decrease the speed of the machine (all tests in a given series should, however, be run at or near the same speed), change the stiffness of the mechanical system as by using a larger specimen diameter, or change the moment of inertia of mass of the extension shaft or lever and poise, or other parts of the vibrating system.

Machines should be arranged so that they will not restart automatically after a power failure if they tend to vibrate severely when starting up, as for example when the machine passes through a critical speed before reaching the operating speed, or if special attention is required during starting for any other reason.

Precautions for Rotating-Beam Machines:

Rotating-beam machines are frequently equipped with a flexible spring, *L*, Fig. 18, between the load and the specimen to reduce the stress resulting from vibration caused by misalignment of the specimen in the grips. This procedure will not necessarily improve the situation, since the chosen spring together with the applied weight may change the vibration characteristics of the machine so as to place the operating speed at or near a resonance frequency. The probability of such an occurrence will be minimized by selecting springs as flexible as practical for this purpose.

Specimens in rotating beam machines equipped with split collets for straight shank specimens will sometimes gradually work out of the collet. This action is due to flexing of the specimen within the collet, and can usually be cured by making the diameter of the grip end of the specimen larger relative to the test section. If this is not possible, then the test section of the specimen may be made smaller, or some means of positively locking the specimen in the grip may be used—such as a draw-in bolt.

"Fretting corrosion" may occur in the collet due to the rubbing resulting from the flexing mentioned in the preceding paragraph. When it is not possible to use the remedy given above, the corrosion may be materially reduced by plating the grip ends of metal specimens with copper, or hard chrome or in some cases by cold-rolling the grip ends.

Precautions and Notes on the Operation of Other Type Machines:

In repeated-bending machines, the lever arm and one-half specimen, considered as a unit rotating about an axis through the center of the test section of the specimen, should be proportioned so that its center of percussion is located at the wrist pin. This should be done in order to avoid exciting the specimen and lever-arm to vibrate in modes other than the fundamental. The lever arm may tend to vibrate sideways if the natural frequency for such vibration is near that of the motor speed.

Adjustable throw cranks, where used, should be accurately constructed so that the connecting rod always moves in a plane. There should be little or no sideways movement of the connecting rod at any amplitude.

Fretting corrosion may occur at the joints at the ends of the specimen in all type machines. With some materials this condition may result in fracture of the specimen at the joint. The fretting corrosion and tendency for fracture at the joint may be reduced by inserting a thin spacer of vulcanized fiber or a rubber gasket between the specimen and the part to which it is joined. Such joints may loosen as a result of relaxation of the spacer if the material is not properly selected and applied.

It is important that all locking screws, and bolted joints in the machine be tight at all times. Variation in the degree of tightness may affect the dynamic calibration of many machines.

Under some conditions substantial inelastic action may occur in the specimen during the first few cycles. This action will be manifested by a permanent set or by changes in the maximum and minimum dynamometer readings (where a dynamometer is used) in successive cycles at the same adjustment settings of the machine. When this occurs with a machine equipped with a dynamometer, the adjustments may be altered each cycle (while turning the crank over by hand) until further changes nearly cease to occur. Then the machine may be started.

It should be pointed out that the results obtained in this way will not be identical with results obtained by operating the machine at the initial setting of the adjustments, in instances in which substantial ineleastic action occurs.

When a permanent set is observed the mean stress in a cycle will not be accurately given by the ordinary equations but may be computed approximately by more involved procedures referred to in preceeding subsection, "Stress or Strain Determination," p. 46. Since on unloading the stress-strain relation is essentially linear, the range of stress (and hence the amplitude of stress) will be the value given by the ordinary equations.

Machines should be checked periodically to determine whether slippage has occurred in the grips or elsewhere. Corrections should be made if the test has been monitored frequently enough so that the number of cycles affected by the slippage and the error in stress is insignificant.

Ordinarily machines which produce constant-amplitude-of-force should be operated at a constant-amplitude-of-force regardless of changes in strain which may occur in the specimen; and machines which produce constant-amplitude-of-displacement should be operated at a constant-amplitude-of displacement in spite of changes in stress which may occur in the specimen during the test.

However, there have been numerous cases in which for expediency constant-amplitude-of-force machines have been intermittently or continuously adjusted either by hand or by automatic apparatus to make them apply a nearly constant-amplitude-of-displacement. The reverse has also been used as an expedient; constant amplitude-of-displacement machines have been operated at nearly constant-amplitude-of-force by making adjustments as needed.

It is wise to examine the shut-off mechanism periodically to insure that it is operating correctly and has been properly adjusted. This mechanism should ordinarily be so adjusted that a slight pressure of the hand on an appropriate part of the machine or specimen will actuate the shut-off mechanism.

DETERMINING NUMBER OF CYCLES-TO-FAILURE

The determination of the number of cycles-to-failure must be based on a definition of fatigue failure. Fatigue failure may be said to occur when a specimen has completely fracured into two parts. However, such a definition has certain objections in some instances. In large members or test specimens a fatigue crack, once formed, may require a great many cycles to propagate itself through the specimen. This is especially true of specimens subjected to bending or torsion and even more true if the machine is of the constant-amplitude-of-displacement type.

Wide differences in the rate at which fatigue cracks progress after the cracks have been initiated are found for some materials in identical specimens tested at the same conditions. These differences in rate of crack propagation are the result of differences in the geometry of crack formation and cause differences in the number of cycles-to-failure which are not directly related to the number of cycles required to initiate the fatigue crack.

For laboratory fatigue specimens of

small size, the use of complete fracture is usually satisfactory for axial-load machines, and constant-amplitude-of-force machines for any type of loading. In such cases it is only necessary to provide a switch which will stop the machine when the specimen fractures and a counter to indicate the number of cycles of operation. The difference between the initial reading and final reading of the counter is then the number of cycles to failure.

Some other definition of failure should be used in most tests of large size specimens, and tests with constant-amplitude-of-displacement machines (especially in torsion tests of some metals and bending or torsion tests of fibrous materials).

In torsion tests of metals the geometry of crack formation will occasionally be such that the two parts of a completely broken specimen will hang together so that the usual switch for shutting off a machine will not function. Thus the number of cycles-to-failure of the specimen will be indeterminant.

A similar difficulty is encountered in testing fibrous materials such as laminated plastics and wood. Such materials when tested in constant-amplitude-of-deflection fatigue machines develop fatigue cracks whose propagation may be arrested by the fibrous nature of the material and the fact that the crack is being propagated into a field of low stress so that the specimen does not completely fracture for many cycles (if at all) even though fatigue cracks of considerable severity are present.

For the conditions described above fatigue failure may be arbitrarily said to have occurred when a visible crack appears in the specimen or part. This definition of fatigue failure is used in inspection of parts in service and sometimes in laboratory tests, but has the disadvantage of depending to a large extent on the intensity of lighting used during the inspection and the quality of eyes and patience of the operator. The method also requires almost continuous monitoring of the test.

In another arbitrary definition of fatigue failure used by Findley (40) fatigue failure is said to have occurred when the stiffness of the specimen has been reduced by a specified amount as a result of a fatigue crack. This change in stiffness may be detected by a change in deflection of the specimen in constant-amplitude-of-force fatigue machines, or by a change in the load, torque, or bending moment applied to the specimen in constant-amplitude-of-displacement fatigue machines. In the latter instance a suitable dynamometer may be used in series with the specimen to measure the change in load, torque, or bending moment.

The change-in-stiffness method has the disadvantage, in some types of machines, of requiring more elaborate apparatus and testing technique than sometimes used. Also when testing materials whose stiffness is sensitive to small changes in temperature the decrease in stiffness resulting from a rise in temperature due to internal friction may be difficult to separate from the decrease in stiffness resulting from a fatigue crack.

Usually the load-deflection curve of a cracked fatigue specimen will not be a straight line owing to the opening and closing of the fatigue crack, and the curve may not even have polar symmetry for completely reversed bending. For these reasons the decrease in maximum load is not a satisfactory measure of the decrease in stiffness. A load-deflection curve for a cracked and an uncracked specimen in bending is shown in Fig. 19. The decrease in slope of the upper portion of the curve for the cracked specimen compared to the slope of the curve before the specimen was cracked is equal to the change in stiffness of the member. For a constant amplitude-of-deflection fatigue machine the change in slope is conveniently determined by measuring the loads at two different deflections, Δ and Δ_{max}, Fig. 19, before the fatigue test has started and after a crack has developed. The deflection Δ_{max} is conveniently taken as the maximum deflection and the deflection Δ should be a reproducible deflection large enough to be above the knee in the curve for the cracked specimen caused by the opening of the crack.

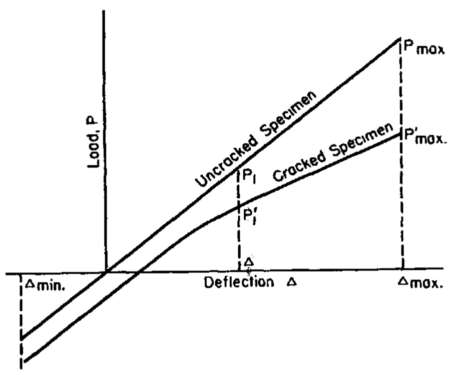


FIG. 19.—Load-Deflection Diagram for an Uncracked and a Cracked Specimen in Bending or Torsion.

The ratio, R_E, of the stiffness of the cracked specimen to the stiffness of the uncracked specimen is then given by the following:

$$R_E = \frac{P'_{\max} - P'_1}{P_{\max} - P_1}$$

A suitable value of this ratio to use in defining fatigue failure in small specimens of metals is 7 to 8. For plastics a value of 3 to 4 is perhaps more practical owing to the change in stiffness of plastics resulting from the increase in temperature during a fatigue test.

TEMPERATURE EFFECTS

In laboratory fatigue testing one object is to determine the S–N diagram of the material at a *given temperature.* But, this requirement is especially difficult to meet under some conditions. All materials dissipate mechanical energy under repeated stressing as a result of several types of internal movements and rearrangements of the constituents of the material. This energy appears as heat which collects in the material and causes a rise in temperature. The maximum temperature developed in the specimen depends not only on the total amount of heat generated but on the distribution of heat sources within the specimen, and the heat transfer coefficient of the material of the specimen, surrounding atmosphere, surrounding parts, contact surfaces, etc.

Thus the temperature rise will be greatest: for materials having the largest energy dissipation (hysteresis damping), for specimens having repeated loads which produce uniform stress over the cross-section of the specimen, for large-size specimens, and for specimens of low thermal conductivity tested in still air in machines having a relatively small volume of metal in the grips to effect the removal of heat from the specimen.

An undesirably high rise in temperature of the specimen may be reduced by some of the following expedients:

1. Reduce the size of the specimen.
2. Use a bending type fatigue test instead of an axial load test.
3. Use a specimen of round cross-section instead of a rectangular specimen (in bending).
4. Reduce the speed of operation of the testing machine.
5. Improve the rate of heat removed by operating the specimen in a stream of liquid, such as a non-corrosive oil or an air blast. It is important to make sure that the coolant does not in itself alter the material through such processes as corrosion, or other chemical action, absorption of the coolant by the material, solvent action of

the coolant, or by increasing or decreasing the rate of loss of volatile constituents in the material.

The temperature developed in a fatigue specimen during a test is difficult to measure accurately, since it is not possible to insert a thermocouple in the material without creating a stress concentration which would cause a greater amount of heat to be generated at the location of the thermocouple.

One method of approximating the temperature of the specimen in the most highly stressed region (which is usually at the surface) is to fasten a small thermocouple bead to the specimen at the most highly stressed point with a small piece of tape and determine the temperature indicated by the thermocouple in the usual manner. This method is of course subject to errors of heat transfer from the specimen to the couple, from the couple to the air and lead wires, etc.

The effect of a rise in the temperature of the specimen due to hysteresis damping during a fatigue test is felt in several ways. Each specimen tested in an $S–N$ diagram under these conditions is subjected to a different temperature because of the higher temperatures produced by larger amplitudes of stress. Also the temperature is not constant during the tests because many stress cycles usually elapse before temperature equilibrium is reached and the amount of energy dissipated per cycle changes with the number of cycles sustained and extent of fatigue damage. Thus when the fatigue strength of the material is markedly affected by small changes in temperature, it is difficult to coordinate and interpret the test data of an $S–N$ diagram to tell what temperature should be related to the observed fatigue strength.

A change in temperature may be accompanied by an appreciable change in stiffness of the material. When the change in temperature occurs during the fatigue test different effects are produced depending on the type of testing machine employed. If a constant-amplitude-of-displacement type fatigue machine is employed the amplitude of the alternating stress will decrease as the temperature increases. If a constant-amplitude-of-force fatigue machine is employed the amplitude of the strain cycle imposed on the specimen is increased as the temperature increases.

Another temperature effect which should be examined is caused by differential thermal expansion. An increase in the temperature of the specimen due to hysteresis damping, or an increase in temperature of parts of the fatigue machine caused by radiant heat or heat from bearings, may in some types of fatigue machines cause differential expansions which will alter the mean stress of the stress cycle. Often such changes will not produce much change in the fatigue strength (defined in terms of the amplitude of the alternating stress) because of the fact that the effect of small changes in the mean stress on the fatigue strength is slight.

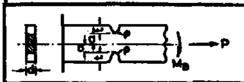
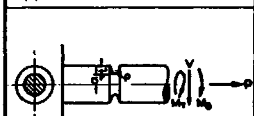
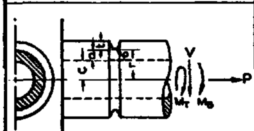
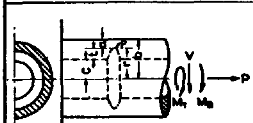
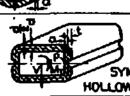
CHECKING

In fatigue testing, constant checking and comparing with previous data are required to avoid the many errors which are easily introduced and which may cause erroneous conclusions to be drawn from the data.

Revolution counters which cannot be set to zero are an advantage since it is not possible to err by failing to reset the counter and if the initial reading is omitted, the data for the previous test will supply the missing information. It is often impossible to detect an error due to failure to reset a resetable counter.

The procedure used should be recorded

TABLE I.—GRAPHIC PRESENTATION OF NOTCH PROPLEMS WHICH CAN BE TREATED BY THE STRESS CONCENTRATION FACTOR NOMOGRAPHS, GIVEN BY FIGS. 20 AND 21.

TYPE OF NOTCH	CASE NO	TYPE OF LOADING	FORMULA FOR THE NOMINAL STRESS	$\sqrt{\frac{t}{\rho}}$ STRESS-CONCENTRATION FACTOR (SHALLOW NOTCHES) COEFFICIENT SCALE:	$\sqrt{\frac{a}{\rho}}$ STRESS-CONCENTRATION FACTOR (DEEP NOTCHED) COEFFICIENT CURVE:	$\sqrt{\frac{r}{\rho}}$ AUXILIARY FORMULA COEFFICIENT CURVE:
	1	TENSION	$\frac{P}{2da}$	b	1	—
	2	BENDING	$\frac{3M_B}{2da^2}$	b	2	—
	3	TENSION	$\frac{P}{da}$	b	3	—
	4	BENDING	$\frac{6M_B}{da^2}$	b	4	—
	5	TENSION	$\frac{P}{2da}$	b	5	—
	6	BENDING	$\frac{3M_B t}{2d(b^3-t^3)}$	a	5	—
	7	TENSION	$\frac{P}{\pi a^2}$	b	6	—
	8	BENDING	$\frac{4M_B}{\pi a^3}$	b	7	—
	9	SHEAR	$\frac{1.23V}{\pi a^2}$	a	8	—
$\mu = 0.3$	10	TORSION	$\frac{2M_T}{\pi a^3}$	a	9	—
	11	TENSION	$\frac{P}{\pi(r^2-c^2)}$	b	5	1
	12	BENDING	$\frac{4M_B r}{\pi(r^4-c^4)}$	b	5	2
	13	SHEAR	$\frac{(1.23r^2+2.77c^2)V}{\pi(r^4-c^4)}$	a	10	3
$\mu = 0.3$	14	TORSION	$\frac{2M_T r}{\pi(r^4-c^4)}$	a	10	4
	15	TENSION	$\frac{P}{\pi(b^2-r^2)}$	b	5	5
	16	BENDING	$\frac{4M_B r}{\pi(b^4-r^4)}$	b	5	6
	17	SHEAR	$\frac{(2.77b^2+1.23r^2)V}{\pi(b^4-r^4)}$	a	10	7
$\mu = 0.3$	18	TORSION	$\frac{2M_T r}{\pi(b^4-r^4)}$	a	10	8
	19	SHEAR	$\frac{V}{\pi a r}$	a	10	—
	20	TORSION	$\frac{M_T}{2\pi a r^2}$	a	10	—
	21	SHEAR	$\frac{VQ}{2aJ}$ 1)	a	10	—
GIVEN SYMMETRICAL HOLLOW SECTION	22	TORSION	$\frac{M_T}{2aF}$ 2)	a	10	—

1 Q = STATIC MOMENT OF THE UPPER HALF SECTION WITH RESPECT TO THE ZERO LINE.
J = MOMENT OF INERTIA OF THE ENTIRE SECTION WITH RESPECT TO THE ZERO LINE.

2 F = AREA ENCLOSE BY THE CENTERLINE OF THE BOUNDARY WALL (-----)

μ = POISSON'S RATIO

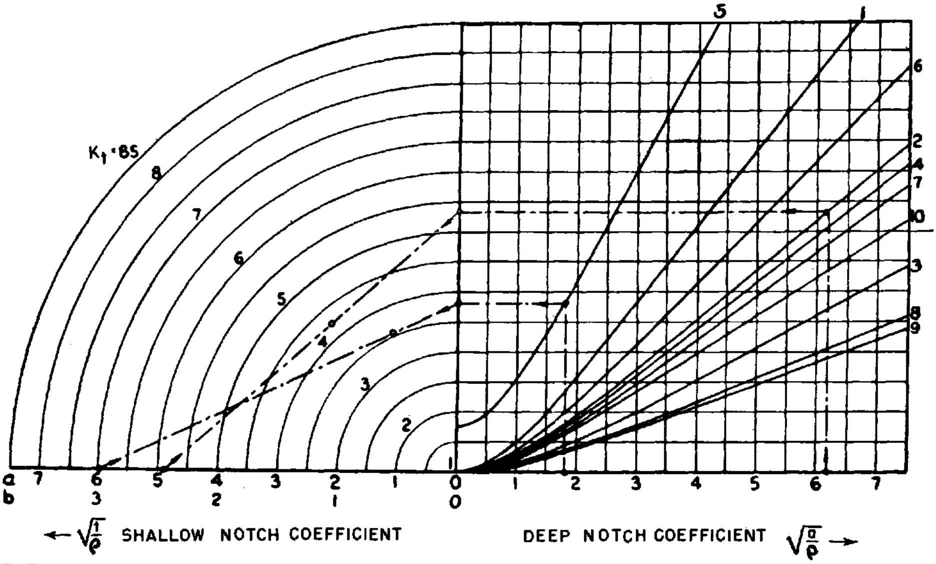


FIG. 20.—Nomograph for the Determination of the Ideal Stress-Concentration Factor.

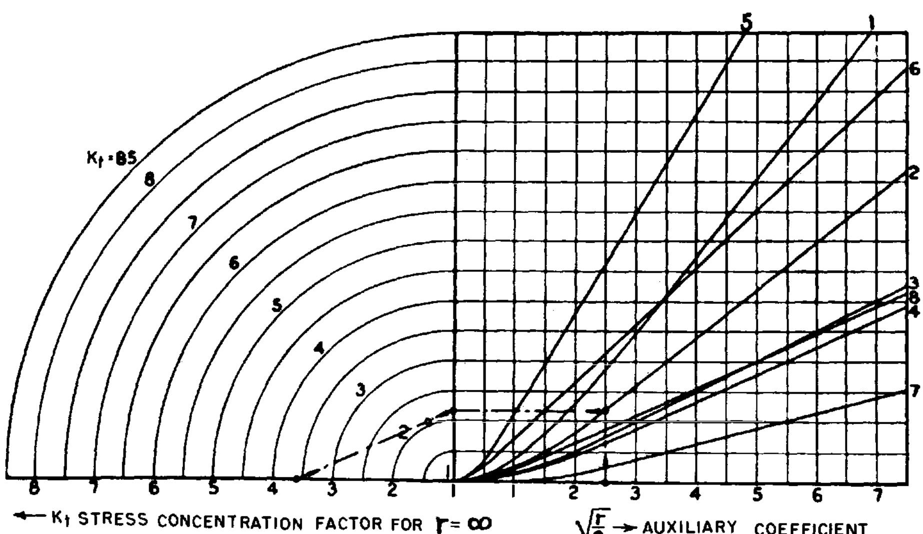


FIG. 21.—Supplementary Nomograph for Circumferential Notches, Provided with an Axial Hole

and frequent reference made to it to insure that changes in procedure or manipulation of the machine are not being introduced which may alter the test results.

It is also important to check the equipment and calibrate the machine at intervals. Some of the things to check are listed under precautions in the preceding subsection, "Manipulation of Machines," p. 52. Calibrating procedures are described in the preceding subsection, "Calibration of Equipment," p. 49.

After the data for a series of fatigue tests have been obtained and plotted as described in Section VI, Presentation of Data, all the test points which are inconsistent with the general trend of the data should be given an especially rigorous examination for errors in material preparation, machining or polishing procedure, accidental scratches, errors in computation or observation, errors in manipulating the machine, machine calibration, or errors in plotting the data.

Finally the fatigue strength and general trend of the data should be compared with existing data on similar materials in order to expose any gross errors which may thus far have escaped notice.

APPENDIX

THEORETICAL DETERMINATION OF STRESS CONCENTRATION FACTORS

By application of the assumptions in the theory of elasticity the local (macro) stresses existing in the region of notches have been calculated for several different types of members by various investigators. Much of this work has been coordinated by Neuber (41) and presented in the form of ratios of the local stress calculated by theory of elasticity in the region of a notch to the nominal stress calculated from elementary formulas. These ratios define the effect of the notch in altering the largest stress developed in the members and are called *stress concentration factors*. The ratios referred to as stress concentration factors theoretically represent ratios of the largest of the principal stresses developed in the region of a notch to the largest of the principal stresses in the corresponding unnotched member.

For most members, the ratios are concerned with values of principal stresses which lie in the same direction for both notched and unnotched members. Whereas, in some instances, the greatest principal stress in a notched member will have a different direction from the greatest principal stress in the corresponding unnotched specimen.

The theory of stresses developed in notched members was summarized by Neuber in the form of the nomographs and table shown in Figs. 20 and 21 and Table I.

The use of these nomographs in calculating stress concentration factors and actual stresses at the root of a notch is illustrated in the following two examples:

Example 1.—An external notch on both top and bottom of a rectangular beam in bending. The condition of this example corresponds to line 2 of Table I. If the following dimensions are given, $\rho = 0.0984$ in., $t = 0.5906$ in., $a = 3.740$ in., $d = 1$ in., and $M_B = 1{,}000$ in-lb., then the shallow notch coefficient $\sqrt{\dfrac{t}{\rho}} = 2.45$, and the deep notch coefficient $\sqrt{\dfrac{a}{\rho}} = 6.16$, and the nominal stress $S\ nominal = \dfrac{3M_B}{2da^2} = 107.5$ psi. Table I indicates that scale b should be used with the shallow notch coefficient and curve 2 should be used with the deep notch coefficient in determining the stress concentration factor from the nomograph in Fig. 20. In Fig. 20, proceed from a point on the horizontal axis corresponding to a deep notch coefficient of 6.16 perpendicularly upward to an intersection with curve 2 then horizontally to the left to an intersection with the vertical axis of the diagram. From this point draw a connecting line as shown to the point on the horizontal axis corresponding to a shallow notch coefficient of 2.45 using scale *b*. This line is tangent to a circle in the left hand quadrant of the figure having a radius of 4.28 which is therefore the value of the stress concentration factor K_t. That is, $K_t = 4.28$. From the definition of stress concentration factor the stress S at the root of the notch may be calculated as follows:

$$S = K_t \cdot S_{nominal} = 460 \text{ psi.}$$

Example 2.—A circular beam in bending containing a circumferential external notch and an axial hole. The conditions for this problem are listed under line 12 of Table I. If the following dimensions are given, $\rho = 0.1575$ in., $a = 0.5118$ in., $t = 1.417$ in., $r = 0.9843$ in., and $M_B = 1000$ in-lb., then the shallow notch coefficient $\sqrt{\frac{t}{\rho}} = 3$, the deep notch coefficient $\sqrt{\frac{a}{\rho}} = 1.80$, the auxiliary coefficient $\sqrt{\frac{r}{\rho}} = 2.50$, and the nominal stress $S_{nominal} = \frac{4M_B\, r}{\pi(r^4 - c^4)} = 1410$ psi. From Fig. 20 a value of stress concentration factor, $K_t = 3.60$ is found by using the shallow notch coefficient with scale *b*, and the deep notch coefficient with curve 5 in the manner described in the preceding example. But this stress concentration factor is the value which would obtain for a large hole, that is for r equal infinity. A correction may now be made by using Fig. 21 and the auxiliary coefficient to determine the stress concentration factor for the actual hole. In Fig. 21 proceed from the auxiliary coefficient $\sqrt{\frac{r}{\rho}} = 2.50$ upward to an intersection with curve 2, then proceed to the left to an intersection with the vertical axis. From this point a straight connecting line is constructed to a point on the horizontal axis corresponding to a stress concentration factor of 3.60 for a member with r equal infinity. This line is tangent to a circle in the left quadrant of Fig. 21 having a radius of 2.08. This value is the desired stress concentration factor. That is, K_t is equal to 2.08. The stress at the root of the notch may now be calculated as in the previous example by multiplying the stress concentration factor by the nominal stress. A value of 2930 psi. is found.

In addition to the method presented above for calculating the stresses in specimens of simple shapes solutions for the stresses in members of many other shapes have been obtained and reported in the technical literature and in books such as Seely (42), Timoshenko (16), Love (17), Roark (18), and Neugebauer (43). In addition, many problems of stress calculation which cannot be handled readily by ordinary mathematical procedures may be solved by numerical methods as described in the book by Southwell (44) and in various papers in in the literature.

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SECTION VI—PRESENTATION OF FATIGUE DATA[1]

This section is mainly devoted to the presentation of data obtained from conventional types of fatigue machines. Nonstandard tests of various types are discussed briefly in order to bring out analogies and differences between the so-called "standard tests" and tests designed to fit particular situations.

Description of Test Specimens

Since in most cases fatigue fractures start at the surface, the method of surface preparation is of primary importance. As discussed in Section IV on Specimens and Preparation, it is desirable to finish the test pieces in a well-defined manner. This usually means nothing; however, the purpose of this data may be to study other types of surfaces, such as "as-rolled" surfaces. Whatever the method of surface preparation used, it is desirable to list characteristics which may affect fatigue strength:

1. The procedure used in polishing, if any,
2. The presence of decarburization, carburization, nitriding, or other surface treatments if the material is steel,
3. Whether or not the surface was shot-peened, sandblasted, or treated in any other way that might affect the surface strength or the internal stress distribution in the test piece,
4. The presence and characteristics of coatings such as electroplating, cladding, etc. and,
5. Any other known surface characteristics (roughness, etc.).

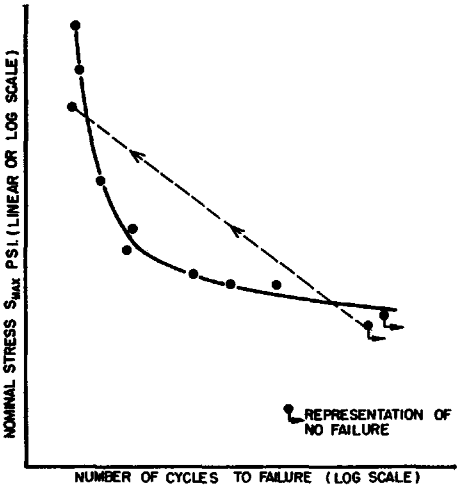

FIG. 22.—Representation of S-N Curve for Completely Reversed Stress.

General Considerations in the Presentation of Fatigue Data

In investigations of fatigue strength of materials there are, in general, two types of data to be presented:

1. The variation in fatigue lifetime with intensities of stress cycles. This type of data is used for making the so-called S-N curves.
2. Variations in stress conditions for failure at a constant number of cycles.

[1] Drafted by L. R. Jackson, Research Supervisor, Battelle Memorial Institute, Columbus 1, Ohio. (Revised following discussion by A.S.T.M. Committee E-9.)

The Presentation of S-N Curves:

A common test is carried out by applying completely reversed stresses. One method of presenting fatigue data from this type of test is in the form of *S-N* curves of the type illustrated in Fig. 22. In diagrams such as Fig. 22, the nominal stress in a repeated stress cycle is plotted against the number of cycles at which failure occurs.

The figure also represents some other conventions which have been widely used. Points representing failure are, of course, clear-cut. In many cases, however, failure has not occurred when the test is stopped, in which case the number of cycles at the time of stopping the test are represented by a point with an arrow. Frequently, it is desired to find out whether a test specimen, from a test stopped before failure, has been damaged by the stress history it has received. In such cases, the stress is raised and the test piece broken. To indicate this sequence, the point representing failure (see Fig. 22) is joined to the corresponding "arrowed" point by a dotted line.

Data from rotating-beam tests are always represented as shown in Fig. 22; data from other types of tests may also be represented in the same manner if the mean stress (S_m) in the cycle is zero.

In some cases, it may be desirable to learn something regarding the rate at which damage from repeated stress is occurring at a given stress level. A procedure which has been used for this consists of the following steps:

1. A conventional fatigue curve of the type represented in Fig. 22 is obtained.

2. A test specimen is run at some stress cycle higher than the endurance limit for some fraction of its expected life, and

3. The stress is then lowered to the endurance limit and the test is continued. If the test piece fails it is said to have been damaged.

By running a number of test specimens at each of a series of stress levels, it is possible to determine the position of a so-called "zero damage" line. For details as to how this method is used, the reader is referred to descriptions such as the one in the reference cited[2].

The choice as to whether a linear or log scale should be used for representation of stress lies with the experimenter. A log scale facilitates the comparison of data by virtue of the fact that curves will have the same shape regardless of the units in which the stress is expressed;

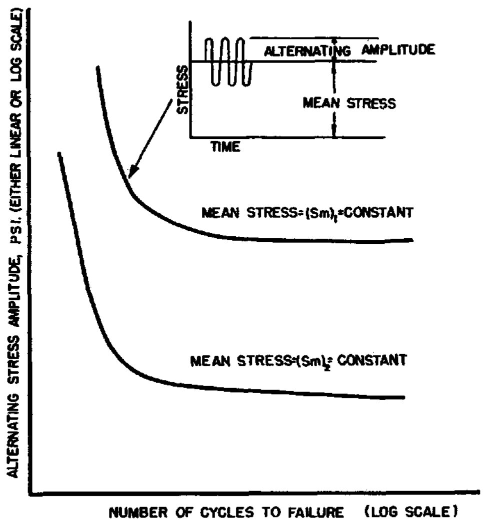

FIG. 23.—Representation of *S-N* Curves Taken at Constant Mean Stress.

furthermore, the log plot has the virtue that it gives per cent of error the same value at all locations in stress. The idea has been expressed that the data appear to be more nearly linear on a log plot; however, the linearity sometimes observed may have no fundamental significance.

When the mean stress in the loading cycle is other than zero, other methods must be used for conveying this informa-

[2] Battelle Memorial Institute, "Prevention of the Failure of Metals Under Repeated Stress," John Wiley and Sons, Inc., p. 92 (1941).

tion. Figures 23, 24, and 25 illustrate methods that have been used. Figures 23 and 25 show, respectively, the representation of data taken at constant mean stress and at constant minimum stress. The conventions for individual points on the curves are the same as those illustrated in Fig. 22.

Figure 24 illustrates the use of "constant R" curves. Points which determine one of these curves are obtained by applying a series of stress cycles with decreasing maximum stress and adjusting the minimum stress in each case so that it is a constant percentage of the maximum stress. When the minimum stress is compression, the ratios become negative; for example, for completely reversed stress, the ratio is -1. The conventions for representing "no failure", etc., are the same as shown in Fig. 22.

The choice of these various methods of taking and representing the data is in many cases immaterial. If enough data are being taken to plot a family of curves, then the family obtained by any one method can be recomputed and plotted for the other methods.

In cases where it is undesirable to obtain enough data to plot an entire family of curves, the choice of method will depend on the service application under consideration. For example, in studying materials for use in combustion-engine cylinder walls, it might be desirable to obtain data at constant minimum stress, while for certain cases in leaf-spring materials, the data might best be obtained at constant mean stress.

The Presentation of Data for Failure at a Constant Number of Cycles:

Figures 26 and 27 illustrate two methods of plotting data of this type. Figure 26 shows a diagram illustrated by Peterson[3] which has gained wide use.

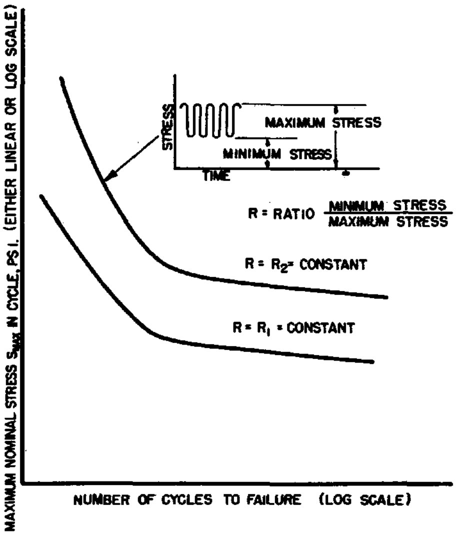

FIG. 24.—Representation of *S-N* Curves Taken at Constant Stress Ratio.

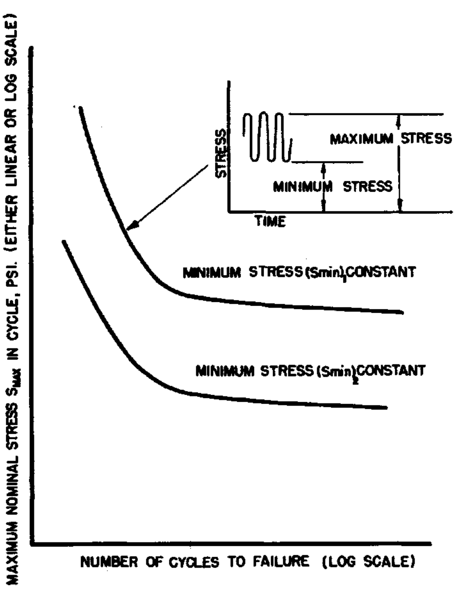

FIG. 25.—Representation of *S-N* Curves Taken at Constant Minimum Stress.

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[3] R. E. Peterson discussion on "Nomenclature on Range in Stress in Fatigue," *Proceedings*, Am. Soc., Testing Mats., Vol. 37, Part I, pp. 162–163 (1937).

It will be noted from Fig. 26 that the mean stress and maximum or minimum stress are plotted to the same scale. By handled by Fig. 26. This method is known as the Haigh-Soderberg method of presentation.

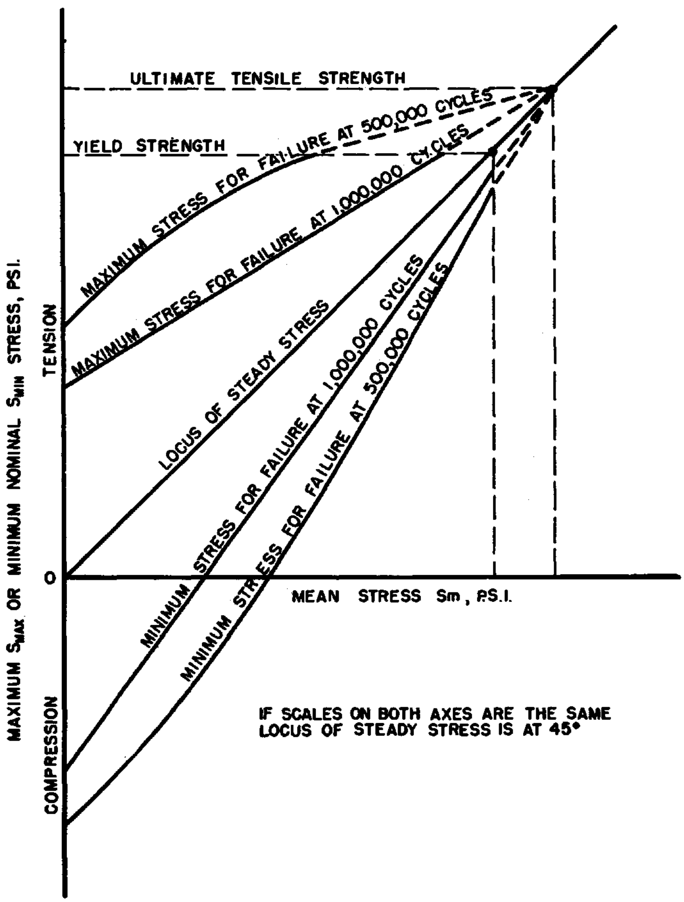

FIG. 26.—Diagram for Representation of Failure, Under Axial Loading, at Constant Number of Cycles.

this means, the locus of steady stress is at 45 deg. to the mean-stress axis. It should also be pointed out that, in the figure, the curves are shown "dotted" above the yield strength. This practice calls attention to the fact that fatigue phenomena for stresses above the yield are complicated by plastic flow.

Figure 27 illustrates another method of presenting the same type of data as is handled by Fig. 26. This method is known as the Haigh-Soderberg method of presentation.

Methods of Handling Scatter in Fatigue Data:

Experience has shown that considerable scatter may be expected in fatigue data. In general, variation in the number of cycles-to-failure of between 50,000 and 100,000 is not considered significant particularly near the endurance limit; however, this cannot be cited as a general rule, since many cases arise in which the background of experience for a given material may be sufficiently detailed that a difference in the number of cycles-to-failure of 50,000 may be highly significant.

Statistical methods[4] have been developed for handling scatter in data; however, it is rare, if ever, that a case arises in which enough fatigue data are available to allow the intelligent application of statistical methods. While no hard and fast rules for handling scatter can be laid down, the following summary of expedients that have been used by various experimenters may be helpful in deciding how a particular set of data should be handled:

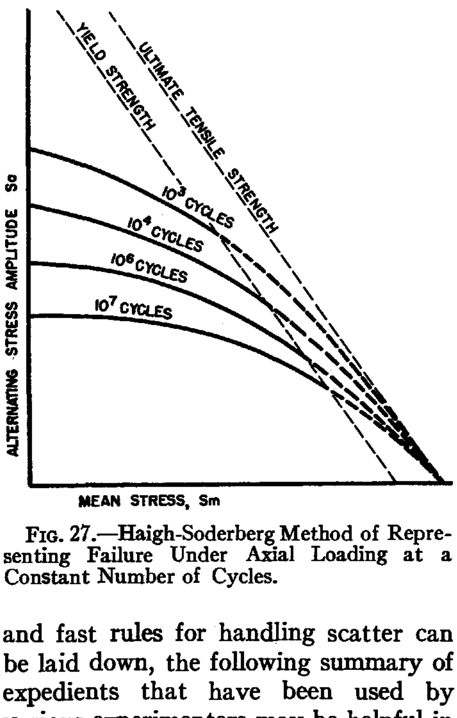

FIG. 27.—Haigh-Soderberg Method of Representing Failure Under Axial Loading at a Constant Number of Cycles.

1. Care should be taken to choose the scales for plotting data so that they are consistent with the precision of measurements. Frequently, data are plotted on too open a scale and the scatter thereby revealed is more characteristic of the precision of measurement than it is of the material. In particular, scatter near the endurance limit is more fairly appraised when the scale of stress is correctly chosen.

2. When a set of data shows excessive scatter, it is always desirable to find out whether instructions for the preparation of test pieces have been consistently followed, before attempting to interpret the scatter as a characteristic of the material.

3. If only a few points lie far off a curve established by a larger number of points, the individual test pieces responsible for the excessive scatter should be carefully examined for accidental flaws before appraising the value of these points.

4. When all of the factors within the control of the experimenter have been appraised, the fitting of a curve to the experimental points may be accomplished by such methods as the following:

(*a*) The most common method is to judge by eye the best location for a smooth curve threading its way through the points, draw it in free hand, and later smooth it out with a French curve. While the rigor of this method cannot be defended, it is seldom that a curve drawn by any other rule will deviate in a significant manner.

(*b*) Some experimenters assume that the fatigue curve is a straight line above the endurance limit when plotted on log-log coordinates. They then fit the best straight line to the points by the method of least squares. This method has the virtue that it provides a systematic method for specifying how data are to be handled.

(*c*) For reasonably conservative design purposes, it is sometimes desirable to bracket the data with a band, and

[4] R. E. Peterson, "Approximate Statistical Method for Fatigue Data," ASTM BULLETIN, No. 156, January, 1949, p. 50 (TP 12).

then use the lower limit of this band for design purposes.

(*d*) When, after eliminating as much scatter as can be explained by controllable variables, the data still show so much scatter that it is not possible to draw a reasonable curve through the points, no conclusions regarding fatigue strength should be drawn.

(*e*) When enough data are available, it is sometimes possible to use statistical methods of analysis.[4, 5]

COMPUTATIONS OF STRESS VALUES IN PRESENTING DATA ON FATIGUE STRENGTH

It has been almost uniformly conventional to report stresses in fatigue test pieces as nominal stresses. That is, for beam-type test specimens the stress is reported as the stress at the minimum section as computed from conventional beam formulae. In axial-stress tests, the stress is reported as the load divided by the minimum area in the test section. While this convention provides a uniform method of computing a relation between test load and geometry of test specimens, there are two general cases in which the stress so computed is fictitious.

Actual Stresses for Nonlinear Elastic Conditions:

The conventional formula for computing the stress in the outer fiber of a beam from a knowledge of its geometry and the distribution and magnitude of external loads depends on assuming a linear relation between stress and strain, and on identical compression and tension modulii of elasticity of the material. For many materials these two assumptions are not justified, and in order to compute the maximum stresses with some precision, it is necessary to make a detailed analysis based on a complete stress-strain curve for the material extending into the compression region. Plastic yielding also makes it difficult to determine accurately the state of stress existing in a flexural fatigue specimen.

Stress Computations in the Presence of Stress Concentrators:

Changes in cross sections such as notches, holes, fillets, etc., always introduce stress concentrations in fatigue test specimens whether they are of the bending or axial-stress type. While it is common practice to report nominal stresses on the basis of the minimum section and ignore stress concentrations, the effect of stress concentrations on fatigue strength is fully appreciated and many devices have been adopted by various experimenters in order to take these factors into account.

The problem of stress concentrations has been approached from two viewpoints which reach toward different objectives:

1. The objective of determining the relation between the magnitude of stress concentration and the geometry of the part under test without regard for the material, and

2. The objective of determining the reaction of various materials to the same geometrical stress raiser.

Relation Between Geometry and Stress Concentration:

In reporting data from fatigue tests on notched[6] specimens, it is desirable to give the value of the theoretical stress-concentration factor for the particular type of notch and loading used. This factor, K_t, is the ratio of maximum stress to nominal stress for an ideally homogeneous and elastic material. Values of K_t are commonly obtained from mathe-

[5] A.S.T.M. Manual on Presentation of Data, Am. Soc. Testing Mats. (1943).

[6] For convenience, the term "notched" specimen is here used to denote any specimen with a geometrical irregularity (notch, groove, fillet, hole, etc.).

matical calculation or from photoelastic analysis. For many cases, values of theoretical stress-concentration factor have been tabulated in available references.[7] A chart from one of these references is given in the Appendix to Section V.

Reporting the value of K_t serves to define the notch used in the tests and allows comparison of theoretical stress

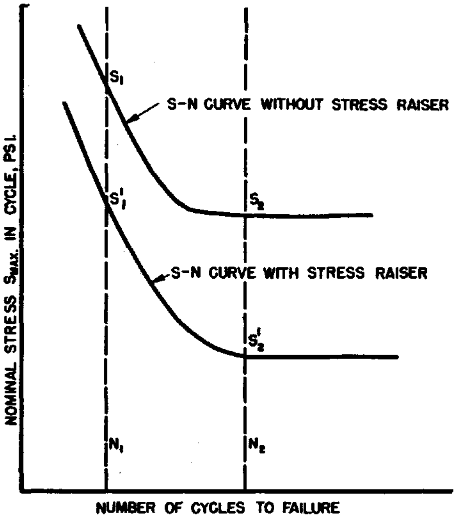

FIG. 28.—*S-N* Curves for Completely Reversed Stress Used to Estimate Strength Reduction Factor.

concentration and measured fatigue-strength reduction in the test pieces used. It is not common practice to use K_t to "correct" stress values in the notched fatigue tests. Observation of notch-fatigue values are still reported in terms of nominal stress as illustrated in Figs. 28 and 29.

[7] Useful references are: H. Neuber, "Kerbspannungslehre," Julius Springer, Berlin. This book has been translated by the David Taylor Model Basin where it is listed as *Translation No. 74*. The German text has been reprinted by Edwards Brothers, Inc., Ann Arbor, Mich. (1944).

G. H. Neugebauer, "Stress Concentration Factors and Their Effect on Design," *Product Engineering*, Vol. 14, pp. 82–87 and 168–172 (1943).

R. J. Roark, "Formulas for Stress and Strain," McGraw-Hill Book Co., Am. Soc. Mechanical Engrs., Design Data, Part I (1938).

From the nominal stress values observed in fatigue tests on unnotched and notched specimens, values of "fatigue strength-reduction factor",[8] K_f, are computed. Thus, in the case illustrated in Fig. 28

$$K_f = \frac{S_1}{S_{1'}} \text{ at lifetime } N_1 \text{ (cycles)}$$

$$K_f = \frac{S_2}{S_{2'}} \text{ at } N_2 \text{ (cycles)}$$

etc.

This measured "fatigue strength-reduction factor" is to be considered as

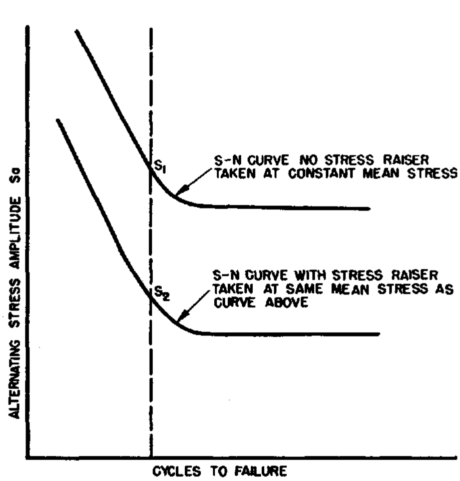

FIG. 29.—Constant Mean Curves Used to Estimate Strength Reduction Factor.

a measure of reduction in fatigue strength of the particular specimen or member tested rather than a basic reduction in fatigue strength of the material.

Even under completely reversed stress tests, it is found that the fatigue strength-reduction factor varies with:

1. The material and its condition (heat treatment, aging, etc.).
2. The condition of the surface. It is desirable to take all possible precautions that the root of a notch

[8] Sometimes the term "fatigue notch-factor" is used.

be polished in the same manner (and with the same minimum of cold work) as the surface of the comparison unnotched test piece.
3. The type and severity of notch.
4. The kind of loading (bending, axial, torsion, etc.).
5. The lifetime at which it is evaluated. (K_f is usually small at short lifetimes and often approaches K_t at long lifetimes).
6. Almost any other testing conditions (temperature, corrosion, etc.).

In view of the importance of so many factors, it is particularly necessary that all details be listed in reporting notch-fatigue tests.

As long as the mean stress is zero, the above convention is reasonably satisfactory with the exception that it does not take into account complex stresses. Most stress raisers produce stresses in more than one dimension. This means that, while the stress system used to obtain the S-N curve without a stress raiser is uniaxial, the stress system in the presence of a stress raiser is usually biaxial or triaxial; thus, factors other than the stress concentration are present, and the two curves are not on a strictly comparable basis.

Some attempts have been made to generalize the stresses used as a basis of comparison so that, for example, both are the maximum shear stresses, or both represent shear-strain energy, etc.; however, not enough work of this type has been done to demonstrate the correct method of comparing results from different types of stress systems or even to show that there is any single correct method. A summary of such methods is given by Peterson[9]. Methods of computing the effect of stress concentrators are also discussed in Section V.

[9] R. E. Peterson, "Application of Stress Concentration Factors in Design," *Proceedings*, Soc. Experimental Stress Analysis, Vol. 1, No. 1, pp. 118–128 (1943).

Strength Reduction Factors When the Mean Stress is Not Zero:

The methods of presenting data to be used in computing the strength reduction factor for the case where the applied load has not completely reversed each cycle are at present in a very unsatisfactory state. The reason for this is that, as mentioned above, the material in the region of highest stress at a stress concentrator yields slightly in many cases and thereby transfers part of its load to less heavily loaded regions. This occurs when the load is completely reversed as well as when there is a mean load; but when the load has completely reversed each cycle, it is reasonable to assume that the stress in the region of highest concentration also reverses each cycle. When there is a mean load, however, the stress cycle in the region of highest concentration where the fatigue cracks start is not predictable. Two conventions have been used to handle this situation.

The first is illustrated in Fig. 29. As in Fig. 28, the strength reduction factor is computed as the ratio of S_1 to S_2. Families of curves such as those illustrated in Fig. 27 are also useful for this method of computation. The principal difficulty with this method is that, while loads can be applied to test pieces containing stress raisers which will theoretically provide the same mean stress as was used on tests without a stress raiser, the redistribution of stress by yielding in the region of highest stress concentration complicates the situation so that there is no assurance that the comparison is actually being made at the same mean stress. Fortunately, in many materials the alternating stress component in a fatigue test is not especially sensitive to the mean stress over quite wide ranges in stress. This fact makes it possible to use the method outlined above without encountering too many contradictions. When using this method, however, it should be kept in mind that a value for strength reduction factor obtained by the method has only a limited meaning.

A second method of computing strength reduction factors is illustrated in Fig. 30. In this figure, it will be noted that, instead of using constant-mean-load curves, constant-ratio curves are used. Otherwise, the method of calculation is the same, that is, the strength reduction factor is the ratio of S_1 to S_2. (Fig. 30).

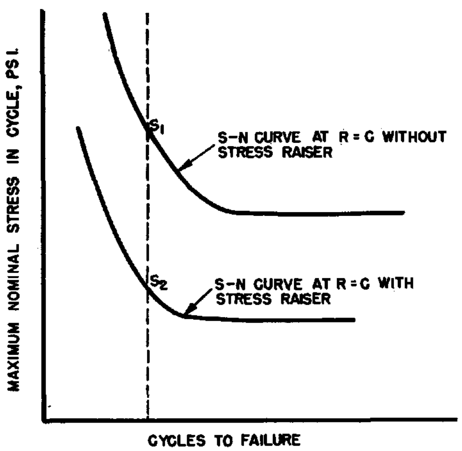

FIG. 30.—Use of Constant R Curves to Estimate Strength Reduction Factor.

Both methods may be criticized, the latter results in a definitely variable K_f factor while the former results in a fairly uniform K_f and is considered by some to be preferable for this reason. The former method corresponds to the common design practice of applying a K factor to the variable component only.

PRESENTATION OF FATIGUE DATA UNDER SPECIAL TESTING CONDITIONS

Corrosion Fatigue:

While all of the remarks made previously apply also to the presentation of fatigue data obtained under corrosive conditions, there are some other factors which require special attention. For most materials and test conditions, the speed of test is of minor importance; however, in corrosion-fatigue tests, the speed is of primary importance. For example, at the high-stress end of the fatigue curve, high-speed tests would produce failure by fatigue before corrosion effects have time to produce any damage. Similarly, at low stresses, such phenomena as stress corrosion may produce failure of a type which resembles a fatigue failure, but which would have occurred from the action of static stress alone in about the same time.

The exact conditions under which corrosion occurs should be described as carefully as possible. In view of the fact that both fatigue and corrosion phenomena require time for operation, considerable variation in results can be obtained with minor changes in testing conditions. For this reason in the design of experiments in corrosion fatigue, it is advisable to attempt to simulate the service conditions to which the data are to be applied as closely as possible. In most cases, it will not be possible to simulate these conditions exactly because time involved would be too long; however, when either stress or corrosion conditions are accentuated in order to accelerate the test, results should be used with caution.

High-Temperature Fatigue Tests:

Here again speed effects may be important, because creep phenomena may overshadow fatigue effects.

In running fatigue tests over a range in temperature, it will be found that, under fluctuating tensile loading, there will be a temperature above which the brittle failures usually associated with fatigue phenomena no longer occur; however, a failure somewhat akin to fatigue will occur in that, after a certain number of stress cycles, creep or plastic flow will accelerate rapidly.

The critical temperature for this change in type of phenomena under repeated stress is not a constant for a given material but depends on the speed of test, the method of loading (presence or absence of mean load), and whether the loading is in bending or is axial.

As in corrosion fatigue, tests in high-temperature fatigue should be designed to simulate intended service as much as possible if data are to be used directly in design.

SPECIAL FATIGUE TESTS ON LARGE STRUCTURES

Frequently the design and fabrication of a structure are more important in determining its performance under repeated stress than is the material from which it is made, and it is desirable to determine the performance of a complete structure or an element of a structure under repeated stress.

In most cases of this type, not only is the detailed stress analysis of the structure complicated and uncertain, but also the exact location and magnitude of service loads are not known. Furthermore, the time and expense involved in these tests make it imperative that the number of tests to be performed be kept to a minimum.

There are several techniques that have been used which aid in obtaining the maximum amount of information from the fewest number of tests.

In the first place, one of the most informative guides to the design of experiments is the study of service failures in the structure under consideration. When the external load cycles are adjusted by trial and error so that the location and appearance of the failure resembles that observed in service, the test is very useful in comparing alternate designs and materials.

It is always desirable to measure the stress at the location of failure, so that results can be compared with known fatigue properties of the material in order to find out whether the material is performing in a normal manner and whether or not there is a "size effect". That is, whether fatigue strength of the structure is compatible with data on small test pieces.

The stress change at the location of failure can, in many cases, be measured with the use of wire strain gages of the SR-4 type provided that the location is known; however, considerable help can be secured by the use of brittle lacquers which can be used to locate regions of high stress when loads are applied as they are in service.

In large structural elements, machined, or otherwise shaped from a single piece, it will be found that there will be a "size effect" which will make fatigue-strength results lower than would be expected from conventional fatigue tests on small pieces of the same material. Whenever possible, it is desirable to measure stresses as carefully as possible so that "size effects" can be separated from stress-concentration effects. Furthermore, it will sometimes be found that failures in large parts originate in relatively large defects. When this occurs, the fact should be noted again in order to avoid confusing possible "size effects" with the homogeneity of the structure.

When large structures are fabricated from sheets or other thin sections bolted, riveted, welded, or otherwise fastened together, "size effects" such as are noted in large monoblock parts are not so apparent; however, another effect may may enter. It has frequently been noted that in newly made fabricated parts not all of the fastenings are carrying their proper share of the load. In service, this condition is usually corrected by slight yielding at the overloaded fastenings; proof loading will also, in most cases, effect the necessary readjustments. A low-stress fatigue test may not apply enough stress to relieve the overloaded fastenings but may be high enough to produce fatigue failures at the locations of overload. Such results give an erroneously low value for the fatigue strength of the structure and precautions should be taken to see that the structure is in a condition to allow a fair appraisal of its strength.

SECTION VII—INTERPRETATION OF FATIGUE DATA[1]

Fatigue data are normally obtained from three general types of tests, which may be designated as the material type, the structural type, and the actual service type. The material type of fatigue test is the conventionalized or in some instances standardized one, using rather small specimens of various sizes and shapes, depending on the mode of stressing to be imposed and the testing machine involved. The structural type of fatigue test for this classification should be broadly interpreted to include machine parts and assemblies. The actual service type of fatigue test is, as its name implies, a test under actual service conditions.

The results obtained from the material type fatigue test are generally considered to be indicative of the inherent fatigue strengths of the materials tested, when subjected to specific types of repetitive loading and environment. In making such tests major attention is accorded the material, but a number of other factors are involved whose effects upon the results obtained must be considered for proper interpretation of the test data. The more important of these factors include:

1. Specimen geometry (size and shape),
2. Specimen preparation (machining, polishing),
3. Testing machine (design, mechanical condition, operation),
4. Testing technique,
5. Speed of testing,
6. Type of stressing, and
7. Environment.

Consideration of these factors is important because frequently minor differences in them can cause appreciable differences in fatigue results. Tolerable discrepancies in these factors are generally smaller than those considered satisfactory for static tests of the materials.

Data from the material type of fatigue test may be advantageously used for the following purposes:

1. Comparing the behavior of different materials subjected to repeated stresses,
2. Comparing the effects of various manufacturing practices or processes upon the fatigue properties of materials,
3. As a guide in developing new materials, or manufacturing processes,
4. Comparing the behavior of materials in various environments, while subjected to repeated stresses,
5. Comparing effects of various simple geometrical factors such as different sizes and shapes of notches and different surface finishes,
6. Establishing correlations with other mechanical properties, different types of stressing, chemical composition, etc.,
7. Checking the quality of different lots of a given material, and
8. Evaluating the effects of surface treatments such as case hardening, decarburization, nitriding, shot peening,

[1] Drafted by R. L. Templin, Assistant Director of Research and Chief Engineer of Tests, Aluminum Company of America, New Kensington, Pa. (Revised following discussion by A.S.T.M. Committee E-9.)

and plating upon the fatigue properties of materials.

In general, data from fatigue tests of this type are not directly applicable for engineering design purposes. This type of test gives quantitative data, but ordinarily these data can be used only qualitatively by the designer and operator of structures and machines. Some of the factors which preclude satisfactory correlation of material fatigue test data and service life expectancy are differences between actual and assumed loading, environmental conditions, and differences between actual and nominal stresses.

The structural type of fatigue test, in addition to material and the factors mentioned previously in connection with the material fatigue test, includes two other factors of primary interest to designers and users of structures. These are design and fabrication. The specimens used in making the tests range from simple joints, bearings, beams, columns, frames, and hydrostatic pressure units, to parts of actual structures and machines and occasionally full-size complete structures or machines. These specimens, however, are tested by being subjected to simulated assumed service loads, the application of which is usually accelerated for the purpose of reaching the end points of the tests quickly. The effects of the time and environment factors are often omitted from such tests and this fact should be considered when interpreting the data obtained.

The structural type of fatigue tests appears to have particular merit when used for the following purposes:

1. Revealing stress concentrations and design or fabrication faults,
2. Comparing specimens of a given structure made of different materials,
3. Comparing different designs for a specific structure,
4. Comparing structures made by different fabrication procedures,
5. Developing better designs or fabrication procedures for structures,
6. Establishing design criteria (satisfactory nominal stresses) for given conditions of repetitive loading conditions,
7. Correlation with actual service data toward obtaining an acceptable, relatively quick, means of predicting service behavior and life, and
8. Correlation of data from structural units with behavior of complete structures so as to be able to predict the performance of the actual complete structures from the tests of its components.

The efficacy of the structural type fatigue test for the last three purposes indicated, so far has not been satisfactorily demonstrated and may never be fully realized, but much effort is being expended toward reaching the goals mentioned. This type of fatigue test is generally considered to give results that are more nearly indicative of the behavior of actual structures, than does the material type of test.

Service tests of actual structures often furnish the most satisfactory fatigue data, particularly if the service conditions are determined together with the corresponding effects upon the structure. The procedure of designing, building, and putting into service a pilot or sample structure has been frequently followed. Observation of the performance of such pilot structures furnishes data useful in correcting faults or making improvements in subsequent similar structures. In order that corrections and improvements can be made sooner and more intelligently, much effort is being made to measure loads and resulting stresses under actual service conditions. These data, when correlated with the structural type fatigue results, should aid considerably in establishing the merits or shortcomings of the accelerated structural type fatigue tests.

When interpreting fatigue data, it is generally more satisfactory to make comparisons on the basis of stress or load at a given number of cycles than on the number of cycles or life at a given stress, since a relatively small change in stress usually corresponds to a very marked change in the number of cycles. The number of cycles is of particular interest when considering fatigue strengths but may be of less significance in the case of endurance limits. A longer life at a given stress is often not considered significant unless it is on the order of two or more times the reference value.

The end points for fatigue tests of the material type are in most cases quite evident, since it is customary to take the number of cycles corresponding to the occurrence of the first visible crack in the specimen. For many of the different kinds of specimens used in structural type fatigue tests the end-point criteria just mentioned can also be used. As the structural type specimens become more complex, however, more consideration must be given to the selection of acceptable end points. The occurrence of the first crack in such specimens may be observed long before the specimen ceases to withstand the repeated loading conditions imposed. Under actual service conditions it is, of course, quite customary to make minor repairs or changes when evidence of potential fatigue failures occur but the observance and correction of these incipient failures are not usually interpreted as the end points in the service life of any given structure.

Another type of repetitive loading test frequently used is in reality a vibration type of test. In such tests it is customary merely to shake or vibrate specimens with little if any consideration being given to the actual stresses or loads or amplitudes involved, yet some effort may be made to simulate service conditions. Results from these tests are nearly always reported in number of cycles withstood under the imposed conditions but are nevertheless difficult of satisfactory interpretation because the actual stresses and amplitudes vary appreciably with slight variations in dimensions, weight, design, or fabrication of the specimens. In general, vibration *per se* does not connote eventual fatigue failure but the magnitude of the actual stresses and the total number of times they are repeated under conditions of vibration are the factors of primary significance.

SECTION VIII—BIBLIOGRAPHY[1]

The following brief bibliography has been compiled to list a few selected articles for those who may desire to make a more comprehensive study of some of the research reports giving other techniques of testing and factual data. No attempt has been made to present a complete bibliography but rather to guide the reader to a few publications which cover a variety of topics. These papers are each fairly comprehensive and also list a number of additional sources for further study. For instance, the 1946 printing of "Prevention of Failure of Metals Under Repeated Stress," reference No. 5, contains a very complete bibliography listing 914 articles, and "Fatigue Properties of Aircraft Materials and Structures," reference No. 6, contains a listing of 1028 articles, mainly on fatigue studies. In general, many papers dealing with the various phases of the strength of materials and members subjected to diverse test conditions involving repeated stressing are presented each year in the publications of all the engineering technical societies.

[1] Drafted by T. J. Dolan, Research Professor of Theoretical and Applied Mechanics, University of Illinois, Urbana, Ill. (Revised following discussion by A.S.T.M. Committee E-9.)

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