(d) *Mechanical property tests.* Deflexion method, brittle coating method, bonded wire technique, moist coating method, vibration methods involving frequency and damping or damping changes during fatigue test.
(e) *Penetrating radiation tests.*
(f) *Ultrasonic testing* Reflection or through transmission methods.
(g) *Magneto-inductive tests.*
(h) *Electrical tests.* Electrical resistance or tribo-electrical methods.

Destructive Tests

(a) Heat tinting method
(b) Chemical etching method
(c) Recrystallization method
(d) Damage line method
(e) Impact method
(f) Tensile pulling
(g) Slow-bend test
(h) Sectioning techniques

A review of experimental data on the initiation and propagation of fatigue cracks in test specimens is given in Part 2 of the reference cited.

For more detailed description of the various methods reference is made to the bibliography of the report by Demer containing about 200 references, and to the bibliography below.

References: Bennett (1956), Buchanan and Thurston (1956), Deck (1956), Demer (1955), Frost and Dugdale (1958), Frost and Phillips (1956), Hardrath and Leybold (1958), Hult (1957a, 1957b, 1958a, 1958b), McClintock (1956), Yeomans and Bellonca (1956), Weibull (1954a, 1956a, 1956b).

CHAPTER III

FATIGUE TESTING MACHINES AND EQUIPMENTS

SECTION 30. GENERAL

Fatigue testing machines may be classified from different view-points such as: purpose of the test, type of stressing, means of producing the load, operation characteristics, type of load, etc. The most appropriate sequence of these alternatives for building up a classification system depends upon who is going to use it. One system may be preferred by the manufacturer of testing machines and another by the research worker. The attitude of the latter will be taken in this chapter, which is aimed at being helpful to investigators trying to select the testing machine most suitable to their purposes. For this same reason it was decided to avoid detailed descriptions of individual machines, but to provide an ample number of references. Comprehensive reviews of the whole field are to be found in the following books: Cazaud (1949), Horger (1949), and Oschatz and Hempel (1958).

The purpose of the investigation is the most important item for the investigator, and he generally knows, when starting his investigation, what type of stressing he is going to use, whereas it may be of minor importance whether he is to use a mechanical or an electrical machine; the above-mentioned sequence will therefore be used for the classification system.

The purpose of the test will be chosen as the basis of the first-order division, the type of stressing as that of the second-order division, and the design characteristic as that of the third-order division. Each of these classes may be subdivided according to the operating characteristic, i.e. the machines may be either of the resonant type, which operate at or close to the natural frequency of the mass-spring system, or of the non-resonant type which do not.

A further basis of division is the type of load; a machine belongs either to the constant-stress amplitude type or to the constant-strain amplitude type, although some machines may easily be transformed from one type to the other, for example by inserting or removing a spring.

The first-order division consists of the following classes: (1) machines for general purposes; (2) machines for special purposes; (3) equipments for testing parts and assemblies; (4) components of fatigue testing machines; (5) calibration and checking of testing machines; and (6) accuracies of actual testing machines and equipments.

The second-order division, which will be primarily applied to the general purpose machines, but also, when possible, to other machines and equipments, consists of the following classes; (1) axial loading; (2) repeated bending; (3) rotating bending; (4) torsion; (5) combined bending and torsion; and (6) biaxial and triaxial loading.

The third-order division consists of the following classes: (1) load produced by mechanical deflexion combined, in some cases, with variable spring forces and/or reciprocating masses; (2) load produced by dead weights and/or constant spring forces; (3) load produced by centrifugal forces; (4) load produced by electro-magnetic forces; (5) load produced by hydraulic forces; (6) load produced by pneumatic forces; and (7) load produced by thermal dilatation.

The testing machines for special purposes are basically similar to general purpose machines, with some modifications and additional devices. They will be classified into: (1) high frequencies; (2) elevated or low temperatures and cyclic thermal stresses; (3) corroding environments and fretting corrosion; (4) multi-stress level tests; (5) contact stresses; (6) repeated impact; and (7) combined creep and fatigue tests.

Equipments for testing parts and assemblies have been designed for the purpose of adapting the component to conventional testing machines, but sometimes the equipment is attached directly to full-scale test pieces, such as aeroplane wings, pressurized cabins, etc. Equipment for testing of the following components will be discussed: (1) wires, tires, and ropes; (2) coil and leaf springs; (3) turbine and propeller blades; (4) large specimens, structures, beams, rails; (5) aircraft structures.

Any component of a fatigue testing machine belongs to one of the following functional parts of the machine: (1) load-producing mechanism; (2) load-transmitting members; (3) measuring device; (4) control device and shut-off apparatus; (5) counter; and (6) framework.

A careful and correct *calibration and checking of the testing machine* is an indispensable condition for obtaining reliable results; the calibration may be subdivided into: (1) static calibration and checking; and (2) dynamic calibration and checking.

Data on the *accuracies of actual testing machines and equipments* are given at the end of this chapter.

References: CAZAUD (1948), FÖPPL, BECKER and v. HEYDENKAMPF (1929), GOUGH (1926), GRAF (1929), GROVER, GORDON and JACKSON (1954), HORGER (1949), JOHNSTON (1946), LEHR (1940), MAILÄNDER (1924), MOORE and KOMMERS (1927), MOORE and KROUSE (1934), OSCHATZ (1936), OSCHATZ and HEMPEL (1958), QUINLAN (1946), RUSSENBERGER (1952), SCHULZ and BUCHHOLTZ (1931), LOCATI (1950).

SECTION 31. MACHINES FOR GENERAL PURPOSES

31.1 Axial Loading

31.11 *Load produced by mechanical deflexion and variable springs and/or masses.*—The simplest way of applying a constant-stress amplitude to a specimen consists of attaching one end of a coil spring to the specimen and imposing a reciprocating motion to the other end by means of a direct crank drive. This type of testing machine was, in fact, used by WÖHLER (1871) in his fundamental investigations. Weak springs and a heavy leverage caused a low natural frequency of the system and consequently the speed of the machine had to be limited to less than 100 c/min.

The same principle with small modifications and improvements has been used repeatedly. MOORE and JASPER (1924) introduced a variable-throw crank and a connecting-rod mechanism, which were also incorporated in a machine by MATTHAES (1935); TEMPLIN (1933) used two variable eccentrics, and MOORE and KROUSE (1934) used a cam-operated lever system. This last machine could be operated at a speed of 1000 rev/min, but in general a reduced speed of 100 to 200 rev/min was recommended to prevent vibration and to reduce undesirable inertia forces.

If the reciprocating motion is applied directly to one end of the specimen, the spring being omitted, a constant-strain amplitude machine will result, provided the testing machine, including the dynamometer, is very stiff compared to the test piece—a condition which is not always fulfilled. Machines of this type using a crank and lever system have been described by WILSON and THOMAS (1938) and LANE (1956). A study of the inertia forces acting on the specimen mounted in a large machine having a capacity of $\pm$200,000 lb showed that even at 180 rev/min the additional forces produced by the masses was some 3 per cent. A similar machine, used by ROBERTS and MCDONALD (1954), with a capacity of 100 tons and intended for testing rivet and screw joints of large sizes, had also to be limited to a speed of 180 rev/min.

For small amplitudes, difficulties may arise with the usual types of bearing, but these may be eliminated by means of flexure plate pivots as demonstrated by EASTMAN (1935), also used by ERLINGER (1941). This useful machine part is discussed in paragraph 34.2.

A double eccentric coupled to a lever system, originally introduced by MOHR (1923) and later adopted by TEMPLIN (1939) for testing structural parts, allowed an operating speed of 500 rev/min at a capacity of $\pm$50,000 lb. Other machines of this type are described by FINDLEY (1947). Reference is also made to a simplified dynamic strain equipment by WORLEY (1948).

An ingenious design, using a differential strip mechanism and eliminating the disadvantages of loose bearings, was developed by PIRKL and VON LAIZNER (1938). Another way of solving this problem by means of two counteracting, conventional bearings has been proposed by KLÖPPEL and applied in commercial testing machines.

The reciprocating motion may also be imposed on the specimen to which a mass is attached, thus producing the load in the form of inertia forces as was proposed by REYNOLDS and SMITH (1902). Further descriptions are given by SMITH (1905, 1910). A machine based on the same principle was developed by Stanton and Bairstow at the National Physical Laboratory, and by STANTON (1905). This machine had four reciprocating masses attached to two pairs of opposed cranks, thus giving complete balance in both horizontal and vertical directions. Four specimens were tested simultaneously at a speed of 1000 c/min.

A convenient machine, incorporated in the current production of the Baldwin-Lima-Hamilton Corporation consists of a shake table to which various test pieces and components can be attached. The motion of the table is obtained by the use of rotating out-of-balance weights, but the load on the specimens is actually produced by reciprocating masses.

An advantage of this type of machine is the high speed that can be achieved, but on the other hand, a very close control is necessary because an error in speed gives twice as large an error in load; complicated speed-regulating devices are therefore usually needed.

An ingenious method of producing resonant vibrations by mechanical means, called "the slipping clutch", was originated by AUGHTIE (1931) and further developed by COX and COLEMAN (1956).

References: AUGHTIE (1931), CAZAUD (1948), COX and COLEMAN (1956), ERLINGER (1941), FINDLEY (1947), HORGER (1949), LANE (1956), LEHR (1940), MATTHAES (1935), MOHR (1923), MOORE and JASPER (1924), MOORE and KOMMERS (1927, p. 91), MOORE and KROUSE (1934) OSCHATZ (1943), OSCHATZ and HEMPEL (1958), PIRKL and v. LAIZNER (1938), ROBERTS and McDONALD (1954), TEMPLIN (1933, 1939), WILSON and THOMAS (1938), WORLEY (1948), WÖHLER (1871).

31.12 ***Load produced by dead weights and/or constant spring forces.***—Springs are not always reliable, and errors in the nominal load are easily introduced by overstressing, temperature effects, and inertia. The best guarantee against such errors appears to be to use gravity forces from suspended weights. The first machine of this type was designed by Jasper as described in the book by MOORE and KOMMERS (1927, p. 91). By rotating the specimen, a stationary weight, suspended at the outer end of a lever, produces reversed axial load in the specimen.

Another design based on a similar principle was proposed by PROT and manufactured by Matra. This construction is described in a paper by OSCHATZ (1943) and in the book by OSCHATZ and HEMPEL (1958). Fluctuating axial load is transmitted from the suspended weights to the specimen by means of a member rotating on a specially shaped curved track. The speed is low, not more than 120 rev/min, and the diameter of the specimen only 2·5 mm.

References: CAZAUD (1948), HORGER (1949), MOORE and KOMMERS (1927), OSCHATZ (1943) OSCHATZ and HEMPEL (1958).

31.13 ***Load produced by centrifugal forces.***—This method of producing loads has found wide application. An early machine was designed by SMITH (1909) and later by THUM at the Material-Prüfungs-Amt, Darmstadt, as reported by THUM and BERGMANN (1937), THUM and JACOBI (1939), and THUM and LORENZ (1941). One single out-of-balance weight was rotated at a constant speed of 1500 rev/min. The centrifugal force could be changed in steps while the machine was stationary.

A few years earlier, a more complicated machine had been designed by LEHR (1930, 1931) and by LEHR and PRAGER (1939). Two pairs of weights rotating at a speed of 3000 rev/min produced a load in the horizontal direction only. The load could be changed by a phase shift while the machine was in operation.

The two preceding types did not use the principle of resonance by which the forces can be multiplied many times. As an example of such a resonant

machine may be mentioned one by ERLINGER (1936, 1938), also described by OSCHATZ (1936), in which a single rotating weight produced vibrations in a cantilever spring; in a later design (ERLINGER, 1943) this was replaced by a coil spring in order to reduce damping effects. A similar principle has been used by Sonntag.

Mechanical oscillators of this type are frequently used in modern commercial machines (Schenck, Baldwin, etc.), also as convenient means of vibrating full-scale structures and assemblies for fatigue testing purposes.

A machine of this type has also been applied to the testing of textiles as described by AMSLER (1946) and TENOT (1947).

References: ERLINGER (1936, 1938, 1943), AMSLER (1946), LEHR (1930, 1931), LEHR and PRAGER (1939), OSCHATZ (1936), OSCHATZ and HEMPEL (1958), SMITH (1909), SONNTAG (1947), TENOT (1947), THUM and BERGMAN (1937), THUM and JACOBI (1939), THUM and LORENZ (1941).

31.14 ***Load produced by electro-magnetic forces.***—Electromagnetically excited machines have the advantage of allowing very high frequencies. The first machine of this type was designed by KAPP (1911) and by HOPKINSON (1911, 1912), who attained a speed of 7000 c/min, and by HAIGH (1912, 1917). Haigh's machine, which has later been described by FOSTER (1932), has an armature placed between two magnets. One end of the specimen is attached to the framework and the other end to the armature which is connected to a double cantilever spring. The natural frequency of the system without specimen is tuned to resonance by changing the length of the cantilever. The introduction of the specimen increases the natural frequency of the system and consequently the machine operates below resonance but with compensated inertia forces.

The same principle was adopted by LEHR (1925). His machine operated with a frequency of up to 30,000 c/min and was incorporated in the production of Schenck and Co., Darmstadt. A similar design has been proposed by ESAU and VOIGT (1928).

A modern machine of this type has been developed by RUSSENBERGER (1945), also described by RUSSENBERGER and FÖLDES (1955), and is now incorporated in the current production of the Amsler Co. The system, consisting of two masses connected through the specimen and the dynamometer in series, vibrates at its natural frequency which, by changing one of the masses, can be tuned to a frequency from 3000 to 18,000 c/min with a capacity of ± 1 ton and ± 5 tons.

Very high frequencies (30,000 c/min) have been attained by VOIGT and CHRISTENSEN (1932) and KÖRBER and HEMPEL (1933) and up to 60,000 c/min in a machine by THOMPSON, WADSWORTH and LOUAT (1956).

References: ERLINGER (1936, 1938), ESAU and VOIGT (1928), FOSTER (1932), HAIGH (1912, 1917), HOPKINSON (1911, 1912), KAPP (1911, 1912, 1917), KÖRBER and HEMPEL (1933), LEHR (1925), RUSSENBERGER (1945), RUSSENBERGER and FÖLDES (1955), SCHULZ and BUCHHOLTZ (1931), THOMPSON, WADSWORTH and LOUAT (1956), VOIGT and CHRISTENSEN (1932).

31.15 ***Load produced by hydraulic forces.***—Very high loads (up to ± 100 tons or more) and large dynamic amplitudes are obtainable by means of hydraulic machines, and various types of commercial machine are now available. The first machines consisted of a pulsator attached to the conventional tensile testing equipments. Later on, designs for the specific purpose of fatigue testing have been evolved.

The problem of changing the load while the machine is in operation has been solved in two different ways. In one, the pump consists of two identical pivoted cylinders, and by changing the angle between them the resultant volume fed to another cylinder in series with the specimen is adjusted to give the required load (Amsler); instead of pivoting one of the cylinders, both cylinders may have a fixed position, and the phase is then changed by means of a differential gear (MAN). Alternatively, the stroke of the pump piston of a single cylinder may be changed (Losenhausen).

A description of an Amsler machine, using the first method, is given by SCHICK (1934) and of the Losenhausen machines, using the second method, by RATHKE (1931) and POMP and HEMPEL (1933, 1936). A hydraulic pulsator, developed by General Motors Corporation, is described by UNDERWOOD and GRIFFIN (1946). The design is somewhat different from the preceding types in that oil at high pressure is discharged to either or both sides of a large-diameter piston connected to the specimen. The travel of the piston is controlled by leakage and bleed-off.

A French pulsator which is combined with a Trayvou universal testing machine is mentioned in the book by CAZAUD (1948, p. 91).

The characteristic feature of hydraulic fatigue testing machines is that the speed is rather limited. For large machines a speed of 500 to 1000 c/min is possible. For smaller machines, as for example the Losenhausen machines with a capacity of ± 3 tons and ± 10 tons, speeds from 500 to 3000 c/min may be used. The load capacity of the above-mentioned GMC pulsator is $\pm 100{,}000$ lb at a maximum speed of 2000 c/min and a stroke of 0·17 in.

References: CAZAUD (1948), DIEPSCHLAG, MATTING and OLDENBURG (1935), POMP and HEMPEL (1933, 1936), RATHKE (1931), SCHICK (1934), UNDERWOOD and GRIFFIN (1946).

31.16 *Load produced by pneumatic forces.*—The only machine of this type has been proposed by Lehr, and a description will be found in the book by OSCHATZ and HEMPEL (1958, p. 183). The main data are: load ± 100 tons, stroke ± 5 mm and speed 1200 c/min. The load is regulated with the machine in operation by changing a volume between the pump, which works at a constant stroke, and the cylinder attached to the specimen.

Reference: OSCHATZ and HEMPEL (1958, p. 183)

31.17 *Load produced by thermal dilatation.*—An original idea for producing cyclic strains was introduced by COFFIN and HEAD (1956). The device was based on the principle of heating and cooling columns in parallel with the test specimen. The thermal expansion and contraction were controlled by thermocouples spot-welded to each column. The cycling speed is of necessity very low. Two full cycles of strain were imposed per minute. This device was used for a study of the fatigue behaviour of cold-worked metal.

The joint effect of temperature and stress cycling was investigated in an apparatus developed by COFFIN and WESLEY (1953). A thin-walled tubular specimen was constrained at each end and alternately heated and cooled. The inner diameter of the specimen was 0·5 in. and the thickness of the wall 0·02 in. which allowed a cycling rate of 4 c/min.

A more complicated stress distribution in the specimen is applied by means of thermal dilatation in a method used at the Westinghouse Research Laboratories for the purpose of screening or grading materials according to their resistance to cyclic temperature conditions. This method which was mentioned by Kemeny in a discussion of a paper by COFFIN (1954b) consists of thermal cycling by induction heating of small disks. The desired temperature is reached after 3 or 4 sec, and is limited to a thin layer around the periphery. After the heating cycle, the test piece is allowed to cool in air until all surfaces are below 800°F, when the specimen is quenched in water. In this way, cracks can be produced within 50 to 100 temperature cycles.

References: COFFIN (1954b), COFFIN and READ (1956), COFFIN and WESLEY (1953)

31.2 Repeated Bending

31.21 *Load produced by mechanical deflexion.*—All machines belonging to this type work on the constant-strain amplitude principle, although a constant moment would be easily maintained in many of the machines by an adjustment while the machine is in operation.

The simple principle of this type of machine consists of bending back and forth in the same plane of the specimen. The forced motion of one or of two points of the specimen is usually produced by an adjustable crank. Various mechanisms are described by HORGER (1949, p. 10). In some of the machines, the stroke can be changed while the machine is in operation, as mentioned by JACQUESSON and LAURENT (1950).

The bending moment may either vary or be constant over the length of the specimen. The former usually results in a simpler design, but the second alternative is preferable from the testing view-point because a larger volume is tested, and irregularities in the material are consequently easier to detect.

The earlier machines were of the first type. In the machines by UPTON and LEWIS (1912), modified by LAUDENDALE, DOWDELL and CASSELMAN (1939), and in those by MOORE (1930), the free end of the specimen is given a back and forth motion by means of a crank. The bending moment consequently increases linearly over the length of the specimen. Sometimes the width of the specimen is made to decrease linearly, so that a constant stress is produced over the larger part of the specimen.

A convenient method of eliminating failure in the grips is to load the specimen as a buckling column; as the moment is proportional to the deviation from the straight line through the ends of the specimen, the moment is a maximum in the middle portion of the specimen and is zero at the grips.

A uniform bending moment over the length of the specimen is realized in many different ways. In a DVL machine developed by MATTHAES (1933), the specimen is attached to two levers, one having a fixed end and the other given a reciprocating motion by means of an adjustable crank. Another design by Erlinger (1938) solves the problem by having the midpoint of the specimen fixed and applying movements along circular arcs to the ends of the specimen. A third machine, designed and constructed at the National Physical Laboratory, is described by Low (1956). The ends of the flat test piece are given appropriate angular movements and all tensile loads are eliminated by leverages; the curvature at the test section and thus the maximum strain is measured by a spherometer. The speed of this machine was controllable between limits of 300 and 600 c/min. By means of a hand rig a speed of about 3 c/min could be attained. In order to localize the strain in the test section as far as possible, steel plates were clamped to the ends of test pieces from sheet material, while in test pieces from bars the test section was reduced in thickness.

The bending machines are easily adapted for testing a large number of specimens simultaneously. An early design at the Bell Laboratory is described by Townsend and Greenall (1929) and by Greenall and Gohn (1937), allowing the simultaneous testing of 126 specimens. In a modified construction by Gohn and Morton (1949) and by Gohn (1952), the number of specimens was reduced to 24. Both mean static strain and alternating strain are adjustable. The speed of this machine (3000 c/min) is exceptionally high for this type of machine. Another fast machine intended for 12 specimens is described by Johnstone (1946).

These machines seldom exceed 1000 c/min because of the low natural frequency of the system, but a machine for 18,000 c/min is mentioned by Jacquesson and Laurent (1950).

A quite different principle for producing bending moments was introduced by Ottizky (1936, 1938). A cantilever specimen is rotated and its free end is loaded through a ball bearing by the constant force of a coil spring. A steady bending moment can be superimposed by means of a beam spring rotating together with the specimen.

References: Dietz (1944), Erlinger (1938), Gohn (1952), Gohn and Morton (1949), Greenall and Gohn (1937), Horger (1949), Jacquesson and Laurent (1950), Johnstone (1946), Laudendale, Dowdell and Casselman (1939), Low (1956), Matthaes (1933), Moore (1930), Oschatz and Hempel (1958), Townsend and Greenall (1929), Upton and Lewis Casselman (1939), Low (1956), Matthaes (1933), Moore (1930), Oschatz and Hempel (1958), Townsend and Greenall (1929), Upton and Lewis (1912), Ottizky (1936, 1938).

31.22 ***Load produced by dead weights.***—This way of producing bending moments rotating in relation to the specimen has been used frequently, but does not appear to have been used for producing fluctuating bending moments in a fixed plane of the specimen.

31.23 ***Load produced by centrifugal forces.***—A very convenient and frequently used method of producing repeated bending stresses in specimens consists of mechanical oscillators attached to the test piece. An early investigation by Gough (1926) used a single out-of-balance weight attached to the free end of leaf springs.

The mechanical exciter may also be mounted in the centre of the span between the nodes. Tests of this type have been carried out on various test pieces by many investigators, including those named in the references below.

Mechanical oscillators having two opposed out-of-balance weights in order to produce a resultant centrifugal force acting in one plane only were designated by Bernhard (1937) and by Lazan (1942). Commercial oscillators of this type were being built in 1927 by Losenhausen and are now available also from Schenck, Baldwin, and other manufacturers. They are almost always run with a speed close to the natural frequency of the specimen system. Further references are given in Section 33.

References: Banks (1950), Benda and Gallant (1954), Bernhard (1937), Gallant and Benda (1954), Gough (1926), Lazan (1942), Mailänder (1939), Newman and Coates (1956), Percival and Weck (1947), Unksov (1956).

31.24 ***Load produced by electro-magnetic forces.***—Most of the machines of this type are based on the same principle as the preceding type, in which a cantilever specimen or a beam is excited to vibrate in resonance. If the specimen is supported at the nodes and vibrates in its fundamental free-free bending mode, failure in the grip portion of the specimen is definitely eliminated.

One of the earlier machines was designed by Jenkins (1925), a wire-bending machine which operated at a frequency of 1000 c/s. A similar machine developed by Ruttman (1933) used a cantilever specimen which was excited by magnets alternately energized by an inertia switch attached to the free end of the specimen. Other machines of this type was described by von Heydenkampf (1929), Müller (1937), Wilkinson (1939), Roberts and Gregory (1951), and Dolan (1951). In the Dolan machine, the difficulty of controlling the amplitude of the resonant vibration within narrow limits was solved by employing a new and simple circuit actuated from a micrometer screw used to pre-set the amplitude desired.

A modification was introduced by Lessells and Brodrick (1956) which made it possible to apply the Prot method of determining the fatigue limit. This method requires a continuously increasing amplitude of vibration at any desired rate. Failure of the specimen at an early stage was detected by the reduction in the natural frequency, which was used also to control the automatic shut-off.

In some cases, e.g. when testing the specimen at elevated temperatures, it may be more convenient to have the vibrating system as a separate unit and to produce the forces on the test member by a mechanical connexion. This method was used by Bleakney (1938) and by Brueggeman, Krupen and Roop (1944) for testing aeroplane wing-beam specimens, and by Welch and Wilson (1941) for testing material at high temperatures.

An interesting torsional vibrator producing bending moments was developed by Wade and Grootenhuis (1954, 1956) by which a wide range of frequencies was attained (from 24 to 3835 c/s). The specimen had a rectangular cross-section and was made to vibrate in the free-free mode by purely torsional oscillations at one of the nodal points.

References: BLEAKNEY (1938), BRUEGGEMAN, KRUPEN and ROOP (1944), DOLAN (1951), VON HEYDEKAMPF (1929), JENKIN (1925), LESSELLS and BRODRICK (1956), MÜLLER (1937, 1939) ROBERTS and GREGORY (1951), RUTTMAN (1933), WADE and GROOTENHUIS (1954, 1956), WELCH and WILSON (1941), WILKINSON (1939)

31.25 ***Load produced by hydraulic forces.***—No reference to machines of this type have been found in the literature, but by means of suitable attachments axial-load or torsional machines may be used for this purpose.

31.26 ***Load produced by pneumatic forces.***—Extremely high frequencies may be attained by this type of machine. The first machine of this type was designed by JENKIN and LEHMAN (1929). Small beam specimens were made to resonate in the free-free mode by an air stream. The frequency was 18,000 c/s. A similar machine, designed and constructed at the National Bureau of Standards, is described by VON ZEERLEDER (1930). The frequency was 12,000 to 20,000 c/s. A method of testing turbine blades with pneumatic oscillators is discussed by KROON (1940).

A machine by QUINLAN (1946, 1947) consists of two small pistons connected to the free end of a cantilever specimen which is vibrated at its natural frequency by air pressure. A pneumatic column is tuned so that its resonance frequency coincides with that of the specimen. It is of considerable interest that fatigue cracks too small to be detected by X-ray or Zyglo tests have a measurable influence on the frequency, which gradually decreases with the growth of the crack. This method also allows internal cracks to be detected before they appear at the surface. The same principle of generating vibrations has been used by MEREDITH and PHELAN (1948) and also by LOMAS, WARD, RAIT and COLBECK (1956) who studied the speed effect on several different materials.

References: JENKIN and LEHMAN (1929), KROON (1940), LOMAS, WARD, RAIT and COLBECK (1956), MEREDITH and PHELAN (1948), QUINLAN (1946, 1947), ROBERTS and NORTHCLIFFE (1947), VON ZEERLEDERER (1930).

31.3 Rotating Bending

31.31 ***Load produced by mechanical deflexion.***—If a bent wire is rotated about its curved axis, a simple and efficient method of producing constant strain amplitudes is obtained. Machines of this type were designed by KENYON (1935) and are also described and used by VOTTA (1948). If the wire arc is circular, a constant bending moment over the length of the specimen results. This is of advantage, if the specimen can be given such a shape that failure does not occur in the grip portion of the specimen, but otherwise it is desirable that the end moments be small. This problem was solved by Haigh and Robertson who introduced the principle of loading the test piece as a buckling column. This idea was adopted by SHELTON (1931, 1933, 1935) and by GILL and GOODACRE (1934).

Instead of using an axial load, CORTEN and SINCLAIR (1955) attained the same result by having the drive end of the wire rigidly fixed, the other end of the specimen being free to rotate in the plane of bending and following a curved path of such a form that the fixed end of the specimen is subjected

to zero moment. The movable end of the specimen fits into a miniature bearing and housing which are free to rotate and assume the configuration imposed by the specimen. Rapid changes of deflexion are possible due to the small masses, and this machine is therefore suitable for programme testing.

References: CORTEN and SINCLAIR (1955), GILL and GOODACRE (1934), KENYON (1935), SHELTON (1931, 1933, 1935), VOTTA (1948)

31.32 ***Load produced by dead weights and/or constant spring forces.***—This type of machine employs either a rotating specimen or a rotating load. The first design constitutes the classical high-speed fatigue machine, introduced by WÖHLER (1871). The merit of this principle lies in the fact that all inertia forces are easily eliminated.

In its simplest form, the rotating-beam specimen is provided at the free end with a ball bearing which is loaded by a dead weight or a constant spring force calibrated by a dead weight. An early design developed by KROUSE (1934) and also described by MOORE and KROUSE (1934) was capable of speeds up to 30,000 rev/min. Such machines are extensively used and have a wide application. Specimens of diameter from 0·05 in. (PETERSON, 1930) up to 12 in. have been tested, the latter specimens requiring bending moments of 8,000,000 lb in.

In these machines, the bending moment varies linearly over the length of the specimen. This may be quite acceptable if the specimen is notched, but in an unnotched specimen a uniform stress over the length is preferable. For this purpose MCADAM (1921) introduced a tapered specimen which satisfies this condition.

Another method of producing uniform stresses over the length is to apply a constant bending moment over the length of the specimen. Four-point loading provides a good solution of this problem, the specimen being supported by two ball bearings while two other bearings are loaded by weights. If a large battery of machines is used simultaneously, it is convenient to replace the weights by coil springs and to set their elongations by a common calibrated weight. Machines of this type were introduced by LEHR (1925) and also by R. R. Moore as described by OBERG and JOHNSON (1937). A spring-loaded machine has recently been described by CORON (1953).

A rotating-beam machine with superimposed fluctuating axial loading was developed by ROMUALDI, CHANG and PECK (1954).

There are methods other than the four-point loading method for producing constant moments. LEHR (1940) attached a cross-lever to the free end of the cantilever specimen and loaded it by two springs acting in opposite directions. This construction was simplified by using one spring only. In this way the axial load is not completely eliminated, but it can be made negligible by using a lever of sufficient length. This modification was introduced by THUM and BERGMANN (1937) and is discussed also by THUM (1942) and by SAUL (1942).

Machines of this type, allowing the simultaneous testing of a large number of specimens, have been designed by PROT (1937) capable of testing thirty specimens and also by KELTON (1946).

The constant-moment machines make it necessary to give the specimen a suitable shape to avoid failure in the grips. In some cases, where this measure is undesirable—as for example when testing wires—it is better to apply a non-uniform bending moment to the specimen.

The second method of producing a bending stress rotating in relation to the specimen, is to keep the specimen stationary and to rotate the bending moment. This principle was used by GOUGH (1926) and by MOORE and KOMMERS (1927). A similar machine was designed by DORGELOH (1929). The specimen is held rigidly in a support, while the other end is rotated in a small circle by a revolving load arrangement. An advantage of the non-rotating specimen is that it can be examined and cracks can be detected while the machine is in operation. Also at elevated temperatures, where the measurement of the surface temperature is needed, the rotation introduces complications.

References: CORON (1953), DORGERLOH (1929), ERLINGER (1941), GOUGH (1926), GUTFREUND (1951), HOWELL and HOWARTH (1937), JATZKEWITSCH (1949), KELTON (1946), KROUSE (1934), LEHR (1925, 1940), MCADAMS (1921), MCKEOWN and BLACK (1948), MOORE and ALLEMAN (1931), MOORE and KOMMERS (1927), MOORE and KROUSE (1934), OBERG and JOHNSON (1937), OSCHATZ and HEMPEL (1958), PROT (1937), ROMUALDI, CHANG and PECK (1954), SAUL (1942), THUM (1942), THUM and BERGMANN (1937), TIEDEMANN, PARDUE and VIGNESS (1955), WÖHLER (1871).

31.4 Torsion

31.41 ***Load produced by mechanical deflexion and inertia forces.***—Machines of this type were developed by WÖHLER (1871), FÖPPL (1909) and ROWETT (1913) using a crank drive acting directly on a specimen in series with a coil spring or a torsion weight bar or even an optical system recording the hysteresis loop and thus allowing a study of the damping at different stages of the damage process (LEHR, 1930). Other contributions to the development of this type of machine has been described by MASON (1917, 1921), MOORE and KOMMERS (1921) and others. Commercial machines of different capacity are now available.

A different principle was introduced by STROMEYER (1914) who used a crank drive connected to one end of the specimen while a flywheel producing the load was attached to the opposite end of the specimen. Two specimens could be tested simultaneously. This machine did not operate at resonance.

A resonant machine was designed by MCADAM (1920) and by BUSEMANN (1925). As a torque bar and a flywheel constitute a system with very small damping, the amplitude is very dependent on the speed of the machine, and artificial damping is sometimes needed.

Torsional oscillations may also be maintained by means of a slipping clutch. This principle was used by Krouse and later by Aughtie and by Cox and Coleman as explained in paragraph 34.1.

References: BUSEMANN (1925), FÖPPL (1909), GUTFREUND (1950), HANSTOCK and MURRAY (1946), LEHR (1930), MAILÄNDER (1939), MAILÄNDER and BAUERSFELD (1934), MASON (1917, 1921), MCADAM (1920), MOORE and KOMMERS (1921), OSCHATZ (1934), PAUL and BRISTOW (1952), ROWETT (1913), SPÄTH (1938), STROMEYER (1914), WÖHLER (1871).

31.42 ***Load produced by dead weights.***—This principle has not been used very much because of the limited speed due to inertia forces. A machine of this type, however, developed by H. F. Moore and by Stanton and Batson as reported in the book by MOORE and KOMMERS (1927, p. 102). The torsional moment was produced by a rotating cantilever beam provided with a dead weight which was attached to one end of the specimen. A speed of 1000 rev/min could be used.

A more complicated design of a similar kind has been produced by PROT (1937). The torsional loading is produced by suspended weights by means of an internal conical gear.

References: MOORE and KOMMERS (1927, p. 102), PROT (1937)

31.43 ***Load produced by centrifugal forces.***—Mechanical oscillators mentioned in paragraphs 31.13 and 31.23 may quite well be used for producing reversed torsional vibrations. Crankshafts of diesel engines have been tested in this way by LEHR and RUEF (1943) and full-size marine shaftings of $9\frac{1}{2}$ in. diameter by DOREY (1948) using a planetary system in which out-of-balance wheels were geared to a sun wheel and planet pinions. An electronic method of speed control was claimed to be capable of regulating the nominal stresses in the specimen within 1 per cent.

THUM and BERGMANN (1937) tested specimens in reversed torsion using a two-mass resonant system excited by a rotating out-of-balance weight. The same method was applied to the testing of tractor engine crankshafts by ROSEN and KING (1946) and by PAUL and BRISTOW (1953) for testing large crankshafts. The torsional moment was 28,000 kg cm at a speed of 300 rev/min.

Mechanical oscillators are now incorporated in the current production of several manufacturers.

References: DOREY (1948), HORGER (1949), LEHR and RUEF (1943), PAUL and BRISTOW (1953), ROSEN and KING (1946), THUM and BERGMANN (1937).

31.44 ***Load produced by electro-magnetic forces.***—Several machines of this type have been designed, all being of the resonant type. Most of them consist of an armature acting as a flywheel connected in series with the specimen and excited either by feeding into the stator an electric current of a frequency close to the natural frequency of the system, as in the Losenhausen machine described by VON BOHUSZEWICZ and SPÄTH (1928), or by some automatic device as for example a swinging contact hammer mounted on the flywheel as done by Holzer and described by FÖPPL, BECKER and VON HEYDENKAMPF (1929). In some cases two flywheels are introduced in the swinging system for the purpose of eliminating bending vibrations (FÖPPL and PERTZ, 1928). A similar design was further developed by ESAU and KORTUM (1930), KORTUM (1930), HOLTSCHMIDT (1935).

The preceding machines apply reversed torsion to the specimen and it is difficult to introduce a steady torsional moment. An improvement in this respect is incorporated in the production of Amsler and also by HENTSCHEL and SCHWEIZERHOF (1954).

References: VON BOHUSZEWICZ and SPÄTH (1928), ESAU and KORTUM (1930), FÖPPL, BECKER and VON HEYDEKAMPF (1929), HENTSCHEL and SCHWEIZERHOF (1954), HOLTSCHMIDT (1935), HUBRIG (1936), KORTUM (1930), PERTZ (1928).

31.45 ***Load produced by pneumatic forces.***—By a slight modification, QUINLAN (1946) had adapted his machine for exciting high-frequency bending vibrations into a torsional machine. This machine is particularly fitted for tests at elevated temperatures.

Reference: QUINLAN (1946).

31.5 Combined Bending and Torsion

31.51 ***Load produced by mechanical deflexion.***—Conventional testing machines may be used to apply combined bending and torsional loads to the specimen by means of suitable attachments. Such devices are described by BRUDER (1943) and by NISHINARA and KAVAMOTO (1943). Designs and features of such attachments for converting Krouse plate-bending fatigue machines and Sonntag vibratory fatigue machines have been developed by FINDLEY (1945) and FINDLEY and MITCHELL (1953) and also by PUCHNER (1946) and have been incorporated in the current production of Krouse.

References: BRUDER (1943), FINDLEY (1945), FINDLEY and MITCHELL (1953), FRITH (1948) NISHINARA and KAVAMOTO (1943), PUCHNER (1946).

31.52. ***Load produced by centrifugal forces.***—Machines for the specific purpose of combining bending and torsional loads are generally based on centrifugal forces. A machine designed by LEHR and PRAGER (1933) consisted of a mechanical oscillator with four rotating out-of-balance weights which produced axial loading, while a cross-lever having mechanical oscillators at each end provided reversed torsional loading. This machine is a non-resonant machine. Further details are given by HOHENEMSER and PRAGER (1933).

Another machine for similar purposes was designed by GOUGH and POLLARD (1935, 1936, 1937). Any combination of bending and torsional stresses was possible by means of a vibrating arm attached through a pivoted joint to one end of the specimen. This arm, which could be operated in any angular position with reference to the longitudinal axis of the specimen, was excited by a rotating out-of-balance disk, operated at the resonant frequency. Additional steady loads could be produced by the cantilever spring supporting the mechanical oscillator as described by GOUGH (1949, 1950).

The use of mechanical oscillators for producing reversed bending and torsion is also reported by THUM and BERGMAN (1937) and by THUM and KIRMSER (1943).

References: GOUGH (1949, 1950), GOUGH and COX (1932), GOUGH and POLLARD (1935, 1936, 1937), HOHENEMSER and PRAGER (1933), LEHR and PRAGER (1933), STANTON and BATSON (1916), THUM and BERGMAN (1937), THUM and KIRMSER (1943).

31.53 ***Load produced by electro-magnetic forces.***—Starting from conventional rotating-beam machines, additional steady or fluctuating

torsional moments were applied to the specimen by ONO (1921), by LEA and BUDGEN (1926) and by Bollenrath as reported by OSCHATZ (1936).

References: LEA and BUDGEN (1926), ONO (1921), OSCHATZ (1936), OSCHATZ and HEMPEL (1958).

31.6 Biaxial and Triaxial Loading

The majority of the combined fatigue stress tests reported have been made by subjecting a specimen of circular cross-section to combined bending and torsion as described above. The range of biaxial principal stress ratios is restricted by this method to from 0 to −1·0, i.e. to biaxial stresses of opposite signs.

A wider range of possible stress combinations can be obtained by subjecting tubular specimens to internal fluctuating pressure and static or fluctuating axial stress, or by combining torsional fatigue and external static pressure.

In the case of tubular specimens subjected to internal pressure, the thickness of the wall decides whether biaxial or triaxial stress result. Biaxial stress is obtained by means of thin-walled tubular specimens, whereas a thick cylinder, subjected to internal pressure and supporting its own end load, can be considered to be subjected to a uniform triaxial tensile stress acting throughout the wall thickness with a superimposed shear stress varying from a minimum at the outside to a maximum at the bore. The ratio of the triaxial tension to the shear stress changes with the ratio of the external to the internal diameters of the cylinder.

Biaxial fluctuating stresses are obviously easier to produce, because a thin-walled tube requires a comparatively small pressure, which may be produced by means of oil from a pump. Such apparatuses, test specimens and methods of testing are described by MARIN (1947, 1948, 1949a, 1949b), MARIN and SHELSON (1949), MARIN and HUGHES (1958), BUNDY and MARIN (1954) and also by MAJORS, MILLS and MCGREGOR (1949). Similar arrangements will be found in publications by MAIER (1934), MORIKAWA and GRIFFY (1945), LATIN (1950), and ROŠ and EICHINGER (1950).

Fatigue under triaxial stress has been studied by MORRISON, CROSSLAND and PARRY (1956). This paper gives a detailed description of the machine used and a discussion of special features required in a machine for this purpose, such as glands, core bar, and pressure measurements. The high-pressure system consists of a ram reciprocating in a closed cylinder filled with oil driven by a variable-stroke mechanism (PARRY, 1956). A remarkable observation is reported. It was found that the fatigue limit for unprotected cylinders subjected to repeated internal pressures was astonishingly low. The fatigue strength of the cylinder could, however, be raised considerably either by honing the bore after heat-treatment or by protecting the bore from the fluid by a thin film of rubber. If the cylinder after honing was heat-treated at 600°C [the material used was Vibrac V 30 (En 25T)] *in vacuo*, then the strengthening effect was removed.

Another way of producing triaxial stresses is to subject a specimen to torsional fatigue with superimposed high static fluid pressure as described by CROSSLAND (1956). The most difficult part of the machine is obviously the seal surrounding the oscillating shaft which is heavily loaded by the high fluid pressure. A successful construction, called the Morrison seal, is described in detail by CROSSLAND (1954).

Finally, reference is made to a method of producing triaxial loads by means of cube-shaped test pieces, developed by WELTER (1948).

References: BUNDY and MARIN (1954), CROSSLAND (1954, 1956), LATIN (1950), MAIER (1934), MAJORS, MILLS and MCGREGOR (1949), MARIN (1947, 1948, 1949), MARIN and HUGHES (1942), MARIN and SHELSON (1949), MORIKAWA and GRIFFY (1945), MORRISON, CROSSLAND and PARRY (1956), PARRY (1956) ROŠ and EICHINGER (1950), WELTER (1948).

SECTION 32. MACHINES FOR SPECIAL PURPOSES

32.1 High Frequencies

The classification into high and low frequencies is rather arbitrary. Since there are now commercially available fatigue testing machines which are capable of speeds up to 12,000 and even 18,000 c/min, it seems reasonable to put the lower limit of high frequency at 30,000 c/min.

The only workable way of obtaining these high frequencies by mechanical means is by using rotating-beam machines. This method was developed by KROUSE (1934) who attained a speed up to 30,000 rev/min using an air-turbine driven rotating-beam machine.

Other high-frequency machines are of the resonant type, the vibrations being excited either electromagnetically or pneumatically.

Of the first type may be mentioned some axial-load machines. SCHULZ and BUCHHOLTZ (1931), VOIGT and CHRISTENSEN (1932), and KÖRBER and HEMPEL (1933) achieved frequencies of 30,000 c/min, whereas THOMPSON, WADSWORTH and LOUAT (1956) reached 60,000 c/min and VIDAL, GIRARD and LANUSSE (1956) even 480,000 c/min. Electromagnetic wire-bending machines by JENKIN (1925) and by WADE and GROOTENHUIS (1954) were capable of 30,000 and 230,000 c/min respectively. A resonant torsion machine by HANSTOCK and MURRAY (1946) operates at a frequency of 90,000 c/min.

The highest frequencies so far produced were obtained by means of pneumatic bending machines. A machine simulating the vibrations of turbine blades attained speeds of 150,000 c/min, while another machine using small beam specimens which were made to resonate in the free-free mode by an air excitation method developed by JENKIN and LEHMAN (1929) reached the highest frequency so far recorded for testing purpose, namely 1,080,000 c/min.

References: HANSTOCK and MURRAY (1946), JENKIN (1925), JENKIN and LEHMAN (1929), KROUSE (1934), KÖRBER and HEMPEL (1933), LOMAS, WARD, RAIT and COLBECK (1956), SCHULZ and BUCHHOLTZ (1931), THOMPSON, WADSWORTH and LOUAT (1956), VIDAL, GIRARD and LANUSSE (1956), VOIGT and CHRISTENSEN (1932), WADE and GROOTENHUIS (1954, 1956).

32.2 Elevated or Low Temperatures and Cyclic Thermal Stresses

Some modifications of conventional fatigue testing machines are generally needed to enable them to be used for testing at elevated temperatures. The rotating-beam machine are not easily arranged for this purpose but a note on such a machine for tests at 200°C is given by PHILLIPS and THURSTON (1951). For this reason machines have been designed where the specimen is stationary and the load rotates. Such machines were developed by DORGERLOH (1929), JATZKEWITSCH (1949), and MCKEOWN and BACK (1948).

Machines specifically intended for testing at elevated temperatures have been designed by HOWELL and HOWARTH (1937), BERNSTEIN (1949), and MARKOWITZ, SMIJAN and MICHAJEW (1949) and special machines for ceramic materials by DICK and WILLIAMS (1952). Test equipments and technique are described by SMITH (1944), REGGIORI and ERRA (1953). VIDAL (1955) combined the temperature with a corroding atmosphere of combustion gases.

Pneumatic, bending, or torsional machines are easily adapted for tests at elevated temperatures by placing a cylindrical, resistance-wound furnace around the specimen as demonstrated by QUINLAN (1946).

Fatigue tests of welds at elevated temperatures were conducted by AMATULLY and HENRY (1938).

An apparatus for testing at low temperatures is described by RUSSELL and WELCKER (1931) and fatigue machines for low temperatures and for miniature specimens have been developed by FINDLEY, JONES, MITCHELL and SUTHERLAND (1952).

A comparatively new field of research is the resistance of materials to cyclic thermal stresses. Thermal fatigue is due either to the anisotropic thermal expansion of the crystals or to temperature gradients as explained by ALLEN and FORREST (1956) who postulate that resistance to thermal fatigue can be determined only from dynamic experiments. A simple method consists of subjecting the specimen, rigidly clamped at its ends, to thermal cycles.

Alternatively, fatigue tests for purposes of comparison can be made at a number of constant temperatures under conditions of constant alternating strain. It is important that the frequency of the stress cycle should be comparable with that occurring in service, since at high temperatures dynamic ductility depends on the frequency.

An apparatus for carrying out either of the two types of tests indicated above has been developed by COFFIN and WESLEY (1953) and is also described by COFFIN (1954a). A detailed investigation of the behaviour of an austenitic steel was carried out by COFFIN (1954b). The apparatus was quite simple; a thin-walled tubular specimen (inner diam. 0·5 in., wall thickness 0·02 in.) was constrained at each end and alternately heated and cooled. The cycling rate was 4 c/min.

Another test, used at the Westinghouse Research Laboratories, and described by Kemeny in the discussion of the paper by COFFIN (1954b) uses disks 1 to 1·65 in. in diameter and 0·11 to 0·25 in. thick which are thermally cycled by induction heating. A thin layer around the periphery is brought to the desired temperature in 3 to 4 sec. After the heating cycle the pieces are allowed to cool in air down to 800°F, when the specimens are quenched in water. The temperature distribution may be controlled by using the indicating paint "Tempilaq". Cracks can be produced within 50 to 100 temperature cycles.

References: AMATULLY and HENRY (1938), BERNSTEIN (1949), COFFIN (1954a,b), COFFIN and WESLEY (1953), DICK and WILLIAMS (1952), DORGERLOH (1929), FINDLEY, JONES, MITCHELL and SUTHERLAND (1952), HOWELL and HOWARTH (1937), JATZKEWITSCH (1949), MARKOWITZ, SMIJAN and MICHAJEW (1949), MCKEOWN and BACK (1948), PHILLIPS and THURSTON (1951), QUINLAN (1946), REGGIORI and ERRA (1953), RUSSELL and WELCKER (1931), SMITH (1944).

32.3 Corroding Environments and Fretting Corrosion

Conventional fatigue testing machines of any type may be used for corrosion fatigue tests with the addition of a means for applying the corrosive solution or atmosphere to the specimens under test. It is essential that the action of corrosion and stressing be simultaneous, and that the temperature be kept constant. It was found by GOULD (1936) that tests in a constant temperature room give points which plot with less scatter than tests conducted in the open laboratory.

Although any type of stressing may be used, different types produce very different results. In the region of normal working stresses, axial stresses give about five times as long a fatigue life as do rotating bending stresses of the same value, as demonstrated by GOULD (1949). This result is explained by the fact that the electric currents flowing under axial loading are of lower intensity than those flowing under the rotating-bending action, thus producing slower fatigue damage.

Some remarkable comments made by Gould have a bearing on corrosion tests and may be presented here. He states that when deciding the way in which the corrosive is to be applied, it may be realistic to consider the fact that with industrial metals it is highly probable that pure mechanical fatigue is a phenomenon which is perhaps non-existent in actual service and even in the laboratory is realized only by invoking the ultimate of refinement in technique. It has been found that the air fatigue limit of the metal was raised appreciably by running the test in a hard vacuum and by excluding oxygen (GOUGH and SOPWITH, 1932) or in a concentrated, pure solution of corrosion inhibitor (GOULD, 1933).

The first systematic investigations into corrosion fatigue were made by HAIGH (1917) and by MCADAM (1926, 1927a,b,c) who used a stream of the corrosive guided along the test piece. Other ways of serving the corrosive are as a spray (GOUGH and SOPWITH, 1933) or as a drip on to a tape which carries a meniscus of fluid over a selected portion of the specimen (GOULD and EVANS, 1939) or by pouring sea-water on to torsional and rotating-beam specimens from an overhead tank (HARA, 1956). Technique and apparatus for testing in an atmosphere of combustion gases are described by VIDAL

(1955). Recently, a comprehensive review relating to corrosion fatigue has been made by GOULD (1956).

References: GOUGH and SOPWITH (1933), GOULD (1936, 1949, 1956), GOULD and EVANS (1939), HAIGH (1917), MCADAM (1926, 1927a,b,c), VIDAL (1955).

Fretting corrosion in connexion with contact friction and its detrimental influence on the fatigue limit was first observed as premature failure of fatigue test specimens in the grip portion. It has been found that the degree of damage of this type is greatest under perfectly dry conditions.

Conventional rotating-beam tests on shafts with pressed-on collars have been carried out by PETERSON and WAHL (1935). Extensive investigations using a similar method are reported by HORGER (1953, 1956).

Another way of producing the necessary pressure is to use clamps in which known high lateral pressures can be applied to specimens under test in a fatigue testing machine, either in plane bending (CORTEN, 1955) or in axial loading (CORNELIUS, 1944). The latter method has been extensively used by FENNER, WRIGHT and MANN (1956).

A third method has been applied by ODING and IVANOVA (1956). A specimen was vibrated in reversed bending and was provided with two bent plates attached to the specimen, thus producing the desired contact friction at the critical part of the specimen. It was found that the fatigue limit was equal to zero or, at any rate, was very small.

References: CORNELIUS (1944), CORTEN (1955), FENNER, WRIGHT and MANN (1956), HORGER (1953, 1956), ODING and IVANOVA (1956), PETERSON and WAHL (1935).

32.4 Multi-stress Level Tests

In order to simulate service loads, the stress levels must be changed during the lifetime of each individual specimen. This can be done either by means of programme testing or by spectrum testing. In the first method, a limited number of stress amplitudes are selected and to each of them is attributed a certain number of stress reversals, chosen on the basis of extensive records of statistical frequencies. Each stress cycle of a given amplitude is repeated a certain number of times, large amplitudes a smaller number than small amplitudes. The programme is composed of these stress levels following after each other either according to a fixed pattern or at random. The second method, spectrum testing, is defined by the condition that two conse cutive stress cycles always differ in amplitude. In this case too the sequence of stress amplitudes either follows a fixed pattern or is completely random.

A non-random programme testing may, of course, be performed by hand in any conventional testing machine, but an improvement is obtained by using automatically controlled machines.

The simplest programme consists of two stress levels only. Axial fatigue testing machines for applying a sequence of loads of two amplitudes have been developed by SMITH, HOWARD, SMITH and HARWELL (1951) and by MCPHERSON (1952). A dual-amplitude rotating-bending machine was designed by CORTEN and SINCLAIR (1955).

Several commercial machines capable of subjecting the specimen to a preassigned programme until fatigue failure occurs are now available. Descriptions and details of such machines are given by BECKER (1949, 1950), HALL and SINNAMON (1952), DRYDEN, RHODE and KUHN (1952), ZÜNKLER (1956), and DEUTLER (1956). One of the most complete programme machines constructed by Schenck and based on proposals by Gassner and Federn is provided with twin drives. Small, fast stress reversals are produced by a crankshaft with constant stroke acting on a spring system in resonance, while high, slow stress cycles are produced by hydraulic means. The programme can be changed within a wide range.

A more realistic simulation of service load is obtained by means of random programme testing. A machine for this purpose was designed and constructed by FREUDENTHAL (1953, 1956). This machine operates on the principle of a conventional vertical rotating-beam machine with the added feature that the load can be arbitrarily varied between zero and a maximum so as to form a prescribed sequence of sufficient length to eliminate any effect of the periodicity. The sequence is recorded on a tape which is run through a reading device consisting of a group of contact springs and a group of relays closing a combination of circuits delivering the prescribed current pulses through the loading coil. Random load fatigue tests on simple specimens have also been carried out by FINNEY and JOHNSTONE (1955). BRUTON, COHEN and HIND (1956) developed a random load controller for fatigue testing of full-scale structures.

The simplest pattern of spectrum testing is that proposed by PROT (1937) for the specific purpose of determining the fatigue limit rapidly. The specimen is subjected to a linearly increasing stress amplitude until failure occurs. This type of spectrum is usually produced by a rotating-beam machine in which the loading weight, consisting of water, increases continuously.

Another solution of the problem is given by BRODRICK, KHEIRALLA and BABCOCK (1956) and LESSELLS and BRODRICK (1956). In this machine the specimen is magnetically excited in a free-free bending mode. A system of electronic controls is provided, so as to produce a continuously increasing amplitude at any desired rate.

For service simulating purposes, HARDRATH and UTLEY (1952) used a rotating-beam machine with a mechanism by which the stress amplitude is varied according to a predetermined pattern. A cam is rotated at 1 rev/min and the specimen at 10,000 rev/min. Two different cams were used, one producing a stress amplitude which varied sinusoidally with time while the other produced stress amplitudes varying according to an exponential function for most of its travel.

A similar machine was designed by LOCATI (1952), and another by SERENSEN (1956).

An interesting method of producing random spectrum loading is introduced by HEAD and HOOKE (1956). The "random noise generator" consists essentially of a thyraton valve giving a large random output voltage which is amplified and excites a moving-coil vibrator which produces an equivalent

bending moment in the fatigue specimen. The average output voltage of the amplifier is maintained at a constant value by a stabilizer.

Finally, a machine of quite a different type may be mentioned. STARKEY and MARCO (1954) have designed a machine which produces a multi-harmonic, uniaxial stress by superposition of fundamental and second-harmonic sinusoidal stress-time waves. The load is produced by cam-operated plungers on a common volume of hydraulic fluid. A machine for similar purposes has been designed and constructed by SERENSEN (1956). Torsional load is produced by two pairs of out-of-balance weights, rotating at different speeds and resulting in stress cycles of polyharmonic form. Different combinations of the first and second harmonics were investigated.

References: BECKER (1949, 1950), BENDA and GALLANT (1954), BRODRICK, KHEIRALLA and BABCOCK (1956), BRUTON, COHEN and HIND (1956), CORTEN and SINCLAIR (1955), DEUTLER (1956), DRYDEN, RHODE and KUHN (1952), FINNEY and JOHNSTONE (1955), FREUDENTHAL (1953, 1956), HALL and SINNAMON (1952), HARDRATH and UTLEY (1952), HEAD and HOOKE (1956), LESSELLS and BRODRICK (1956), LEBER (1954), LOCATI (1952), MCPHERSON (1952), NISHIHARA and YAMADA (1950), PROT (1937), SERENSEN (1956), SMITH, HOWARD, SMITH and HARWELL (1951), STARKEY and MARCO (1954), TAPLIN and FINDLEY (1952), ZÜNKLER (1956).

32.5 Contact Stresses

A very direct method of testing specimens subjected to pulsating contact stresses was devised by KENNEDY (1956). He used two steel balls which were pressed against each other by means of a rig consisting of a rotating shaft which, through two cranks, caused an oscillatory motion in a second shaft, arranged to impart its motion to the loading device. The mating sizes of the pairs of balls were respectively 2 in. and ½ in. diameter. The larger ball was considered to be the test specimen. An important feature of this testing device is an ultrasonic flaw detector.

A different method of producing contact stresses is described by MACKS (1953) and by BUTLER, BEAR and CARTER (1957). The rig consists of two balls driven at high speed on the inner surface of a cylinder race by an air jet from three nozzles. Ball loading results from centrifugal forces. Speed control and automatic failure shut-down systems are provided.

The most common method of producing contact stresses is by rotating a pair of cylindrical disks which are pressed against each other. In some machines one of the disks is driven (WAY, 1935 and BUCKINGHAM, 1944), whereas in other designs both are driven thus allowing a definite amount of slip at the contact surface.

Conical disks are also used to simulate helical gear tooth action in a laboratory machine by WALKER (1947).

A convenient method of testing complete gears is to connect two pairs of wheels in a closed circuit and to apply the load by means of torque bars or coil springs. In this way the effect circulates within the assembly and only the losses due to friction have to be produced by the motor. As an example of this arrangement an investigation by KNOWLTON and SNYDER (1940) is mentioned.

Extremely high contact stresses are developed in ball and roller bearings, and machines for testing complete bearings are used by all manufacturers. Loads are produced by weight or spring-loaded leverages. Some methods of life testing of both plain bearings and ball and roller bearings are described in ASTM STP No. 70 (1946): *Symposium on testing of bearings* and also in a book by JURGENSMEYER (1937).

References: BUCKINGHAM (1944), GUYOT and SCHIMKAT (1950), HORGER (1949), JURGENSMEYER (1937), KNOWLTON and SNYDER (1940), KENNEDY (1956), MACKS (1958), MELDAHL (1939), NISHIHARA and YAMADA (1950), VIDAL, GIRARD and LANUSSE (1956), WALKER (1947), WAY (1935).

32.6 Repeated Impact

The effect of an impact depends entirely upon the shape of the test piece and the rigidity of the framework, and in consequence reproducible results are difficult to obtain. This makes the impact method of testing less reliable than conventional fatigue testing and it is now not very much used. As early as 1864 Fairbairn carried out "experiments to determine the effect of impact vibratory action and long continued changes of load on wrought iron girders".

A typical way of producing repeated impacts is the one used by STANTON (1906). A cam raises a weight which strikes a beam specimen midway between two knife edge supports at a rate of 100 blows per minute. The specimen is rotated 180° between impacts. The speed is restricted, due to the condition that the vibrations have to vanish between the blows. This delay depends, of course, on the shape of the specimen and on the material.

A machine of similar design was developed by Amsler and Co. as described by SCHICK (1934). The test piece permitted an operation speed of 600 strokes per min. Arrangements were provided for tension, bending, and compression impact tests.

A somewhat different principle was used by Roos (1912). His machine consisted of a pair of swinging pendulum hammers acting on a cantilever specimen which was fixed at one end and struck alternately from two sides by the hammers. Fifty double blows were applied per minute. A similar machine was developed by MOORE and KOMMERS (1927). The rate was slightly higher, being 65 blows per minute. Reference is also made to a paper by SEAGER and TAIT (1938).

For the purpose of testing the resistance of plastics to impact, FINDLEY and HINTZ (1943) employed a novel method. Balls were lifted by a large wheel with pockets and deposited in a runway from which they dropped on the specimen. It is of interest to note that calculations of stress produced by impact permitted correlation with fatigue test. A somewhat modified method of transporting the balls was used by LUBIN and WINANS (1944).

References: FINDLEY and HINTZ (1943), LUBIN and WINANS (1944), MOORE and KOMMERS (1927), ROOS (1912), SCHICK (1934), SEAGER and TAIT (1938), STANTON (1906).

32.7 Combined Creep and Fatigue Tests

The development of high-temperature machines such as gas turbines, operating under complex stressing conditions, depends to a large extent on the production and use of special metals. Simple creep and fatigue tests alone are inadequate in determining the behaviour of metals under intermittent working conditions. From a practical point of view it may be advantageous to simulate the essential features of the stress and temperature conditions imposed on a particular machine component during its working life.

From this viewpoint KENNEDY (1956) and KENNEDY and SLADE (1956) have designed and developed an apparatus to examine the more fundamental aspects of these problems, permitting a complex stress programme to be applied and the deformation recorded. In addition, facilities are incorporated for examining the stress relaxation at constant strain or the strain relaxation at constant stress.

The new feature of the machine is the electromechanical stressing system, and particularly its application to creep testing. The stress is imposed by means of a mechanical spring. The extension of the spring is automatically regulated so that the test is conducted at either constant stress or constant load. The test piece is mounted in a temperature-controlled enclosure, its upper end connected through a set of parallel-motion springs to an electromagnetic vibrator. A pick-up in parallel with the vibrator enables the motion of the upper end of the test piece to be measured. The variable force is measured by a barium titanate crystal. The creep stress is applied by driving a reversible motor to pull down the lower end of a spring. Any desired intermittent stressing sequence can be applied by a programmer which switches the motor on or off.

Another machine of this type, combining the rupture test and fatigue test, is described by MANJOINE (1949).

References: KENNEDY (1956), KENNEDY and SLADE (1956), MANJOINE (1949).

SECTION 33. EQUIPMENTS FOR TESTING PARTS AND ASSEMBLIES

33.0 General

The difficulty of correlating the fatigue properties of standard test specimens with those of actual machine parts and components is explained by differences in material properties, shape and fabrication. As an illustration we may take a gas-turbine blade having a crescent-like cross-section. In cast blades, the metal usually has a finer grain structure at the points of the crescent than in the heavier mid-section. As grain structure has a pronounced effect on the fatigue properties, conclusions based on tests on standard specimens from the same material may be quite misleading. In the same way, differences in stress distribution due to the shape and differences in surface condition due to the fabrication make tests on actual components a necessity.

The testing of actual design members and assemblies is, in fact, older than the testing of standardized specimens. Fairbairn in 1864 carried out experiments on full-size wrought iron girders subjected to impact vibratory loads and Wöhler (1858–1870) started his famous investigations by applying rotating-bending tests to full-size railway axles.

The shape and the size of many components prevent the use of standard fatigue machines and accordingly several testing machines and equipments have been designed for specific and limited purposes.

The following items will be discussed in the present section: (1) wires, tyres, and ropes; (2) coil and leaf springs; (3) turbine and propeller blades; (4) large specimens, structures, beams, rails; (5) aircraft structures.

33.1 Wires, Tyres and Ropes

The purpose of testing wires in fatigue is to examine the properties of the material and the effect of heat treatment, cold working, and metallographic and mechanical surface condition. A particular difficulty arises from the fact that the test piece cannot be given a suitable shape to prevent failure in the grips and does not permit surface preparation for this purpose. This condition restricts the choice of testing machine. The most common type of stressing is bending and torsion for the reason that they are more sensitive to changes in the surface properties.

There are also, however, a few wire testing machines in which axial loading is applied to the specimen. POMP and DUCKWITZ (1931) and POMP and HEMPEL (1938), for example, used an electro-magnetic machine based on a d.c. motor vibrated in resonance by an a.c. current. Another type of machine by KENYON (1940) employs the inertia forces of three masses connected to three specimens inserted between two wobble plates which produce reciprocating motions with a phase difference of 120°. A third machine by AMSLER (1946) is based on resonant vibrations produced by centrifugal forces acting on two specimens in series, one on each side of a vibrating lever.

Fatigue testing of wires by means of conventional rotating-beam machines with constant moment over the length of the specimen was used by WAMPLER and ALLEMAN (1939), while KENYON (1935) and similarly VOTTA (1948) developed rotating wire-arc machines with the specimen submerged in an oil bath to prevent transverse vibrations. The constant moment over the length of the specimen makes it difficult to eliminate failure at the grip. For this purpose SHELTON (1931, 1933, 1935) and GILL and GOODACRE (1934) used the buckling column principle introduced by Haigh and Robertson by means of which the moment in the grip portion is practically zero. Instead of loading the wires as a pin-ended column, CORTEN and SINCLAIR (1955) attained the same effect by automatically keeping the distance between the ends of the curved wire at the correct distance for a moment-free end load. ROSSETTI (1953) developed a new machine for testing wire ropes, the load being a combination of bending and tension.

The preceding machines may be used for purposes other than testing wires. A more realistic simulation of service loads on ropes and cables is obtained

in a machine by FOREST and HOPKINS (1932) who used the old principle of flexing the cable over rotating pulleys of different diameters, and simultaneously rotating the cable which was subjected to a constant tension. The friction was reduced by a slow rotation of the pulley. A similar idea was used by WOERNLE (1930) with the modification that both ends of the cable were driven in order to reduce torsional moments on the cable.

A special machine for testing car tyre cords was invented by MALLORY (U.S. Patent No. 2,412,524) and further developed and used by KENYON (1945), BRADSHAW (1945), BUDD and LARRICK (1945) and LARRICK (1945). An axial oscillator-type machine for testing textiles and rubber is described by TENOT (1947).

References: AMSLER (1946), BRADSHAW (1945), BUDD and LARRICK (1945), CORTEN and SINCLAIR (1955), DEFOREST and HOPKINS (1932), GILL and GOODACRE (1934), KENYON (1935, 1940, 1945), LARRICK (1945), MALLORY (U.S. patent No. 2,412,524), POMP and DUCKWITZ (1931), POMP and HEMPEL (1938), ROSSETTI (1953), SHELTON (1931, 1933, 1935), TATNALL (1937), TENOT (1947), VOTTA (1948), WAMPLER and ALLEMAN (1939), WOERNLE (1930).

33.2 Coil and Leaf Springs

Earlier machines for testing of coil springs were of the direct displacement type in which some mechanism, such as a cam, applied a known compressive distortion to the spring. Some were designed for a single specimen, whereas others allowed the simultaneous testing of a large number of specimens. An example of the first type is a machine by ZIMMERLI (1940), and of the second type machines described by TATNALL (1937) and by OSCHATZ (1940).

The force necessary to apply direct compressive deformation to a heavy spring of this type is considerable and requires a large input of power. Due to the weight of the moving parts these machines are slow.

A considerable increase in speed can be obtained by using a machine working on the resonance principle, in which the spring-mass system is oscillated at its natural frequency (LEA and HEYWOOD, 1927). Another machine of this type is described by COATES and POPE (1956), in which the oscillating system consists of two masses arranged between four springs in series. A periodic force produced by a pair of out-of-balance weights is applied to the lower mass. This particular system has two degrees of freedom: the first, in which the two masses move in phase, thus causing no fluctuating stresses in the two central springs; and the second, in which the two masses move in opposite directions, with a central collar acting as a node. The forces are determined by the amplitudes of oscillation which are measured to an accuracy of $\pm 0{\cdot}005$ inch by means of vibrographs fastened on the two masses and the collar. When fracture of any spring occurs, the balance of the system is destroyed and the central collar starts moving, thereby acting on a cut-off system which breaks the power supply to the main driving motor.

Resonant machines for the simultaneous testing of a large number of springs have also been constructed. As an example may be mentioned one of Bauart Reicherter, described in the book by OSCHATZ and HEMPEL (1958, p. 231). At the same time, 100 to 180 springs may be subjected to a displacement of 40 mm at a speed of 1800 to 2400 c/min.

The testing of leaf springs is also based either on constant displacement amplitude or on resonance of the spring-mass system. Several commercial machines of the first type are available. An early design of the second type was developed by BATSON and BRADLEY (1931) in which the excitation force was produced by a crank, leverage and coil spring system. A similar machine was designed by LEHR (1932).

References: BATSON and BRADLEY (1931), COATES and POPE (1956), LEA and HEYWOOD (1927), LEHR (1932), OSCHATZ (1940), OSCHATZ and HEMPEL (1958), TATNALL (1937), ZIMMERLI (1940).

33.3 Turbine and Propeller Blades

Special fatigue-testing machines in which the blade of a propeller is excited to its natural mode of vibration by means of mechanical oscillators are discussed by GARDINER (1949).

Turbine blades may be excited electro-magnetically. Such devices operate at high frequencies and require very small power consumption. Electrical methods of inducing and detecting such vibrations are described by SNOWBALL (1949). An electrostatic method for the same purpose has been developed by STRAND–HAGEN and SOMMER (1956).

References: GARDINER (1949), SNOWBALL (1949), STRANDHAGEN and SOMMER (1956).

33.4 Large Specimens, Structures, Beams, Rails

Fatigue testing of large specimens and full-size members requires special testing machines or equipment attached direct to the test piece. A frequently used device is the mechanical oscillator consisting of a single rotating eccentric, as developed by Losenhausen Werke already in 1927, or of two opposed out-of-balance weights as described by OSCHATZ (1934), THUM and BERGMAN (1937), LAZAN (1942) and others. Vibration-testing techniques for large specimens are reviewed by SCHREYER and YOST (1956).

Electromagnetic excitation is also used. A convenient method of vibrating a test member is to use a mechanical connexion from a moving-coil type loudspeaker as described by BLEAKNEY (1938).

Similar methods may be used for testing structures. Fatigue machines for testing structural units are discussed by TEMPLIN (1939). The fatigue testing of structures by the resonance method is discussed by HEYWOOD (1953) and by MEYER (1954). Notes on the automatic control of testing equipment are given by HEWSON (1954) and a new resonance vibration excitor and controller was developed by LAZAN, *et al.* (1952).

In some cases it may be convenient to attach hydraulic equipment to a structure to produce the load. Arrangements of levers, jacks, loading frames, and special supports are reviewed by OWEN (1943).

A control equipment for the fatigue testing by means of hydraulic jacks of a large variety of components such as highly loaded undercarriage, wing

or tailplane attachments, and pressurized components such as radoms and parts of the air, fuel and hydraulic system has been developed at the English Electric Company Ltd. by MOORE (1956). This equipment is capable of subjecting the component to a programme loading consisting of six loading steps with a number of load cycles arbitrarily selected between one and 999 cycles.

The main part of the equipment consists of a load setting and counting unit which controls the number of cycles of each stress level. This unit controls the supply of hydraulic fluid to the loading rig by means of a solenoid-operated valve. The load applied to the test component is measured by resistance strain gauges included in a bridge network. The bridge can be unbalanced by a number of potentiometers, one for each value of maximum and minimum load. On starting the cycle, the load is gradually increased until the maximum value is reached. At this point, the bridge network will be balanced and the output signal from the bridge will pass through zero and change in phase. This causes the bi-stable switch to reverse the flow, and the load returns to the minimum load value, from which the cycling is started again.

A typical method of testing large welded beams by means of mechanical excitors mounted in the centre of the span between the nodes is described by PERCIVAL and WECK (1947). A similar test on thin-gauge box-section beams was carried out by NEWMAN and COATES (1956) with the modification that the excitor was mounted at one end of the beam with a balancing mass at the other. To keep the test frequencies within 3500 to 5000 c/min the beams were made 6 ft in length.

Similar tests on rails have been conducted by BANKS (1950). A length of 15 ft gave a natural frequency of 1920 c/min.

In tests by ROESLI, LOEWER and ENEY (1954) the passage of trucks over bridge members was simulated.

References: BANKS (1950), BLEAKNEY (1938), HEWSON (1954), HEYWOOD (1953), LAZAN (1942), LAZAN, BROWN, GANNSETT, KIRMSER and KLUMPP (1952), MEYER (1954), MOORE (1956), NEWMAN and COATES (1956), OSCHATZ (1934), OWEN (1943), PERCIVAL and WECK (1947), ROESLI, LOEWER and ENEY (1954), SCHREYER and YOST (1956), TEMPLIN (1939), THUM and BERGMAN (1937).

33.5 Aircraft Structures

Mechanical oscillators have frequently been used for subjecting aircraft structures to alternating stresses, as described by FOSTER and SELIGER (1944) and by MOLYNEUX and BROADBENT (1946). A comparison of the endurance of various aircraft structures under fluctuating load was made by FISHER (1949). Aeroplane wing-beams were tested by BLEAKNEY (1938) and by BRUEGGEMAN, KRUPEN and ROOP (1944) by using a mechanical connexion from a moving-coil type loudspeaker. Tests for studying crack propagation in fuselages and small- and full-scale cylinders were made by HARPUR (1958) and impact tests on aircraft undercarriage by BROWN (1947). Cams were used to lift and drop weights on the test member.

Developments in methods of strength testing pressurized fuselages are reported by HOTSON (1949). Observed failures of pressurized fuselages varied in character from minor rupture to catastrophic explosion. In order to answer the question of whether the character of the failure can be controlled, DOW and PETERS (1955) subjected stiffened cylinders of 2024 aluminium alloy to internal pressure and cyclic torsion, thus simulating stress conditions of cutouts in the side of a pressurized cabin in flight.

Full-scale aeroplane wing structures have been tested to destruction by several investigators. FEARNOW (1951) subjected two C-46D wings to resonant vibrations of constant amplitude by means of a testing rig which consisted of prime mover, reduction gear box, line shafting, adjustable eccentric, and an excited spring. Concentrated masses were attached to the wing to reproduce flight stresses corresponding to load factor values of $1 \pm 0{\cdot}625$ g over approximately 45 per cent of the span. Each wing was instrumented with fatigue-detector wires at points where previous tests by brittle-lacquer techniques had indicated high local stress concentrations. It is remarkable that the decrease in natural frequency was small and could not be used to detect incipient cracks, a method which has been used with great advantage when testing specimens of simple shape (cf. QUINLAN, 1946). In this case, the frequency did not decrease more than 2 c/min out of 106 c/min when as much as 55 per cent of the tension material had failed.

A detailed description of a similar fatigue rig and the preparation of the wing specimen is given by MCGUIGAN (1953). Tests on C-46 "Commando" aeroplane wings with this machine are reported by MCGUIGAN, BRYAN and WHALEY (1954). The tests were conducted at a resonant frequency of 108 c/min at four different stress levels (each wing subjected to one stress level only). The wings were instrumented with a number of wire resistance strain gauges and crack-detecting copper wires in the vicinity of expected stress raisers. Fatigue tests on typical two-spar light alloy structures (Meteor 4 tailplanes) were conducted by RAITHBY (1951).

The fatigue strength of CA-12 "Boomerang" wings was determined in a similar manner by JOHNSTONE, PATCHING and PAYNE (1950). The vibrations were excited by a stroking machine driving through a spring. The load was controlled by a deflexion indicator. The wing was excited on one side only. The bending restraint at the supporting points must therefore be small to obtain sympathetic vibration of the other side.

The vibration method is not suitable for applying high load ranges of low frequencies. For this purpose hydraulic loading rigs are preferable. Such a rig is described by PATCHING (1951) in an interim note on fatigue testing of P51D "Mustang" wings. The frequency is very low, being only 10 c/min. An extension of this investigation including not less than 72 Mustang mainplanes and using a combination of the previously mentioned vibration and hydraulic methods is reported by KEPERT and PAYNE (1956) and by PAYNE (1956). The hydraulic loading rig was used for stress levels leading to fatigue failure in less than 50 kc in combination with dead weights and screw jacks, whereas tests at low load ranges were made in the vibration

loading rig. Each wing was subjected to one stress level only until failure occurred, with the exception that some of the wing specimens were subjected to pre-loads of different magnitude, some as high as 95 per cent of the ultimate failing load.

The preceding investigations were all of the constant-amplitude type, i.e. each specimen was subjected to one stress level only. A fatigue equipment for simulating the flight stresses on aircraft by a programme loading is described by JOHNSTONE and MOODY (1953). Fatigue machines of this type have been used extensively for testing aircraft structures and various components in Germany since 1939 (GASSNER) and are now commercially available (Schenck and others).

References: BLEAKNEY (1938), BROWN (1947), BRUEGGEMAN, KRUPEN and ROOP (1944), DOW and PETERS (1955), FEARNOW (1951), FISHER (1949), FOSTER and SELIGER (1944), HARPUR (1958), HOTSON (1949), JOHNSTONE and MOODY (1953), JOHNSTONE, PATCHING and PAYNE (1950), KEPERT and PAYNE (1956), MCGUIGAN (1953), MCGUIGAN, BRYAN and WHALEY (1954), MOLYNEUX and BROADBENT (1946), PATCHING (1951), PAYNE (1956), RAITHBY (1951).

SECTION 34. COMPONENTS OF FATIGUE TESTING MACHINES

34.0 General

Any fatigue testing machine is composed of the following structural components: (1) a load-producing mechanism which generates the alternating load (or displacement) to which in some cases is added a steady load; (2) load-transmitting members such as grips, guide fixtures, flexure joints etc., by which the load produced is transmitted in such a way as to produce the desired stress distribution within the specimen; (3) measuring devices which permit the setting of the nominal upper and lower load limits; (4) a control device for maintaining the load throughout the test and sometimes automatically correcting changes in force or deformation arising during the test; (5) counter and shut-off apparatus which counts the number of stress reversals imposed on the specimen and stops the testing machine after a given number of cycles, at complete fracture of the specimen, or at some preassigned change in deformation or frequency; (6) a framework, supporting the various parts of the machine and, if necessary, arranged to reduce the vibratory energy transmitted to the foundations.

34.1 Load-producing Mechanisms

The loads may be produced by various methods: mechanical, electromagnetic, etc., as mentioned in Section 30.

The simplest way is to attach one end of a coil spring to the specimen and to give the other end of the spring a reciprocating movement by means of a crank. The use of single tandem springs will often produce torsional vibrations which may be eliminated by the addition of a detuning inertia. Parallel-motion springs are described by JONES (1951). If the speed of the crank shaft is well below the natural frequency of the spring-mass system, the forces on the crank are approximately equal to the forces acting on the specimen. A considerable reduction of the load on the crank is possible, however, by running the crankshaft at a speed close to the natural frequency of the system.

Another convenient method of producing the exciting forces by mechanical means is to use oscillators consisting of one, two, or even four rotating out-of-balance weights. With four weights it is possible to adjust the force while the machine is in operation by shifting the relative phase between the two pairs of weights.

A third method of exciting the spring-mass system, known as the "slipping clutch", was originated by AUGHTIE (1931) and further developed and used by Cox and COLEMAN (1950). An application of this device is also described by O'CONNOR and MORRISON (1956). The clutch is moved back and forth by a variable throw crank, rubbing against a surface of the mass and exciting the system to vibrate at its natural frequency. The force transmitted to the specimen is constant within a reasonable range of speeds of the driving motor; its magnitude depending on the spring system and the throw of the driving crank, which in some applications by Cox could be varied while the machine is running.

The fact that very small forces are required to maintain a spring-mass system in vibration at its natural frequency makes electromagnetic excitation very suitable for testing purpose. Various machines based on this principle have been designed. As an illustration, a device developed by DOLAN (1951) will be described because of its simplicity. The vibrating system consists of two heavy masses attached to the ends of the specimen which acts as a bending spring. This assembly is suspended by links and soft springs from a frame that allows the assembly to vibrate as a tuning fork. One of the masses is excited by means of a short drive rod actuated by the electromagnetic excitor. Attached to the second mass is a velocity-sensitive pickup which generates an electric signal that is amplified and fed back to the driving coil of the excitor. The main difficulty arising when operating a system in resonant vibrations is the control of the amplitude within narrow limits. This problem has been solved in a very simple and successful way which will be described in paragraph 34.4.

Hydraulic machines are convenient to use when a large capacity is required and low frequencies can be tolerated. The loading rigs may easily be adapted to a wide range of applications, in particular by controlling the supply of hydraulic fluid to the loading rig by means of a solenoid-operated valve. Such an equipment, developed by MOORE (1956), has been described in paragraph 33.4.

Although hydraulic methods do not permit high frequencies, the contrary is true of pneumatic methods. As already mentioned (32.1), LOMAS, WARD, RAIT and COLBECK (1956) could easily attain speeds up to 150,000 c/min with an extremely simple design of fatigue machine. A tuning cavity adjustable by a piston, a tube, and the specimen is all that is needed. Another advantage is that the resonant peak in a pneumatic circuit is quite broad and

flat, unlike resonance in an electrical circuit. Automatic amplitude control is, therefore, is most cases unnecessary.

References: AUGHTIE (1931), COX and COLEMAN (1950), DOLAN (1951), JONES (1951), LEHR (1930), LOMAS, WARD, RAIT and COLBECK (1956), MOORE (1956), O'CONNOR and MORRISON (1956).

34.2 Load-transmitting Members

An essential feature of a fatigue testing machine, which is of extreme importance if the machine is to function efficiently and give reliable results, is the way in which the load is transmitted from the machine to the specimen, i.e. the design of the grips. The grips must fulfil two conditions; first, they must not introduce extraneous stresses leading to failure in the grip portion of the specimen, and second, they must not distort the prescribed stress distribution within the specimen.

The first condition is particularly difficult to satisfy when no surface preparation of the specimen is permitted and the specimen cannot be shaped to prevent such failure. Under these conditions only certain types of machine are acceptable, e.g. repeated bending machines or machines of the pin-ended column type.

Even if the specimen is given a suitable shape to satisfy the first condition, the second difficulty remains, i.e. the prevention of unexpected stresses introduced into the specimen when it is mounted in the machine. In particular, precautions must be taken in connexion with axial-loading machines.

It is easy to show that in order to limit the error to not more than 1 per cent of the stress applied to a rectangular specimen of cross-section $b \times h$, the line of load must not deviate from the geometrical axis of the specimen by more than 0·002 h. This problem has been discussed by MORRISON (1940). It is obvious that no specimen which depends upon screwed ends for load application is satisfactory in this respect, as pointed out by O'CONNOR and MORRISON (1956), who also suggest that appalling inaccuracies which may easily range from ten to twenty or more per cent have been incurred by those who have taken insufficient care with this one requirement. This statement appears to be in good agreement with their own results. The specimens of a push-pull fatigue testing machine were finished not only on the gauge length with meticulous care, but also had their ends ground to extremely close limits ($\pm$0·0001 in.). Each end was thus secured, by means of split collets and a nut, axially in a loading bar. These bars were themselves supported in a massive stress-free cast-iron frame in which were four holes, bored and lapped with extreme care to guide the bars coaxially. In spite of these precautions—certainly surpassing current practice—careful dynamic calibration, using triple resistance strain-gauge extensometers attached to the specimen, and statically checked by triple optical extensometers, indicated that the maximum stress exceeded the mean stress by about $1\frac{1}{2}$ per cent.

These results are confirmed by an investigation by FINDLEY (1947). By means of an apparatus which will be described in the following paragraph (3), he found that the specimen was distorted when mounted in the testing machine by an amount corresponding to bending stresses seldom less than 10 or 20 per cent of the mean stress. These distortions could be considerably reduced by means of a six-component correction system incorporated in the testing machine.

Self-alining grips are described by RUSSELL, JACKSON, GROVER and BEAVER (1944) and by GROVER, BISHOP and JACKSON (1951). Measurements with bonded wire strain gauges have shown that, with careful loading, the grips gave uniformity of stresses in a sheet specimen to about ± 500 lb/in^2, which was 5 per cent of the maximum stress.

Five different systems of grips for fatigue testing wires are described by SOETE and VANCROMBRUGGE (1949).

A torsion grip to ensure that only pure torque is applied to the specimen is described by CHODOROWSKI (1956). The square end of the specimen is keyed to an inner member, whose inclined faces are positioned in the machine by four hardened steel balls held in an outer member. The balls are backed by grub screws and bear against hardened steel rollers.

An interesting method of mounting specimens in a torsional vibrator in such a way that reversed plain bending in the free-free mode resulted was developed by WADE and GROOTENHUIS (1956). After having tried several ways, the ultimate form described in the paper consisted of transmitting the torque through a hardened knife-edge integral with a silver-steel shaft. The specimen, supported only at one node, was located on the opposite face by a central probe.

Another detail of practical use for supported levers in testing machines, which is a simple and effective means of overcoming difficulties experienced with the usual type of bearing and knife edges, is the cross-spring pivot which consists of one or two pairs of crossed flat springs. It is of practical use in cases where only a limited angle of rotation is required. The points where the springs intersect functions primarily as the pivot point. Since a cross-spring pivot has no sliding parts there is no need of lubrication. The deflexion is, however, accompanied by reaction forces, though they are usually very small. This constructional element has been examined by EASTMAN (1935). It has been studied experimentally by YOUNG (1944) and theoretically by HARINGX (1949).

The application of the load to structural members, aircraft wings, and the like, is not always easy. A device which in many cases has rendered good service is the tension pad which is glued to the test piece. It has been examined by OAKS and HOWELL (1956).

Thin sheet specimens in compression are inclined to buckle. A means of preventing buckling is to clamp the specimen between guide plates. A description of such guides is given by BRUEGGEMAN and MAYER (1944, 1948) and also by RONDELL and DUYN (1950). In papers by GROVER, BISHOP and JACKSON (1951) and by GROVER, HYLER, KUHN, LANDERS and HOWELL (1953) some features of this device and the influence on the accuracy are presented.

It was found that if the guide plates are too tight and specimens are not perfectly flat, an appreciable fraction of the applied load goes in friction,

but if the guide plates are too loose, the specimen buckles on the compression part of the cycle and bending stresses may become large. On the basis of previous experience, the guide plates were made to allow a clearance of 0·0025 in. between either surface of the specimen and an oiled paper. With clearance increased by a 0·005 in. shim separating the guide plates, however, there was evidence of significant buckling.

A difficult problem is to transmit an axial or torsional load to a specimen inside a thick cylinder to which high internal pressure is applied. A solution of this problem is reported by MORRISON, CROSSLAND and PARRY (1956), the main feature being a gland which provides simultaneously little friction and little leakage. The gland described is of the "unsupported area" principle, in that a heavy block called the gland body is forced by the liquid pressure on to a rubber packing-ring whose extrusion is prevented by chamfer rings. The inner surface of the rubber presses on a thin extension of the gland body. It is reported that since the technique of producing a really good finish on the ram and in the gland with the correct clearance had been mastered, and the optimum packing thickness ascertained, the leakage was extremely small—of the order of a drop per minute—yet the ram could be easily pushed to and fro and rotated by hand when the pressure held was 20 tons/in^2.

References: BRUEGGEMAN and MAYER (1944, 1948), CHODOROWSKI (1956), CROSSLAND (1954), EASTMAN (1935), GROVER, HYLER, KUHN, LANDERS and HOWELL (1953), HARINGX (1949), MORRISON (1940), MORRISON, CROSSLAND and PARRY (1956), OAKS and HOWELL (1956), O'CONNOR and MORRISON (1956), RONDELL and DUYN (1950), SOETE and VANCROMBRUGGE (1949), WADE and GROOTENHUIS (1956), YOUNG (1944).

34.3 Measuring Devices

In the machines producing a constant amplitude of deflexion and those where the load is produced by dead weights or constant spring forces no measuring device is required. This is also the case when a variable load is measured by the extension or compression of a calibrated spring with the reservation that the modulus of elasticity for steel, measured statically, does not remain unchanged at frequencies above 5000 c/min as stated by ERLINGER (1935). Otherwise some means of measuring either a deflexion or a force is needed.

A very simple device for measuring large amplitudes is the vibrograph, which consists essentially of two *diagrams* with sloping *lines*, one of them fastened to the vibrating mass, the other stationary. Vertical oscillations of such a *diagram* causes an apparent horizontal movement of the point of intersection of the *lines*. Readings to an accuracy of 0·002 in. are obtainable.

Various other optical methods such as microscopes or mirrors are frequently used.

A convenient electrical method, easily adapted for control purposes, is the resistor transducer. As an example, the stressing unit by KENNEDY and SLADE (1956) may be mentioned, where the transducers were able to detect a movement of 0·001 in. over a total range of 0·5 in.

An interesting method of measuring the applied load in an axial-loading machine based on centrifugal forces was developed by LEHR (1930). The frame-work of the machine was free to move in horizontal direction and its amplitude, measured by a microscope, was used as a measure of the applied load.

Torsional moments are easily and precisely measured optically by observing the twist of a torque-bar calibrated against a dead weight. This type of spring is less affected by inertia forces than coil and leaf springs and is therefore applicable at high frequencies. An application is reported by CROSSLAND (1956) and by CHODOROWSKI (1956). By using interchangeable bars, a sensitivity of measurement always better than $\pm\frac{1}{4}$ per cent could be expected.

By making use of an idea suggested by Parry, it was possible to measure the torque at any point in the cycle while the machine was running at full speed. The method consists of fitting a contact breaker on the driving shaft, so arranged that it may be manually placed in an arbitrary angular position of the stress cycle. This contact breaker triggers a stroboscope which illuminates the scale, used in conjunction with a telescope and mirrors on the torque bar.

The measurement of forces may be done by mechanical dynamometers of various designs. A versatile electrical method is obtained by the use of bonded wire strain gauges. As the change in resistance is very small, a precise measurement of high-frequency variable forces requires a specialized technique. A null method for this purpose has been developed by ROBERTS (1952), another by GUSTAVSSON and OLSSON (1956).

The strain gauge may be bonded to a calibrated bar, thus serving as a high-frequency dynamometer. It may also be bonded directly to the test piece without adding any perceptible inertia, and is then an excellent means of controlling the desired stress distribution within the specimen.

In hydraulic testing machines, the problem arises of measuring the fluid pressure. Up to pressures of 30 tons/in^2 the approximate pressure may be measured by a Bourdon tube gauge, but for more accurate measurements, or for higher pressures, a dead-weight piston gauge may be used. The weight carrier of the dead-weight gauge is continuously rotated as described by PEARCE (1952). For fluctuating pressures, the pressure effect on the resistance of a manganin coil has been extensively used by Bridgman. Another straightforward method is to measure, optically or by strain gauges, the diametral expansion of a thick-walled cylinder. The latter method has been used, for example, by MORRISON, CROSSLAND and PARRY (1956).

References: CHODORWSKI (1956), ERLINGER (1935), GUSTAFSSON and OLSSON (1956), KENNEDY and SLADE (1956), LEHR (1930), MORRISON, CROSSLAND and PARRY (1956), PEARCE (1952), ROBERTS (1952).

34.4 Control Devices and Shut-off Apparatuses

In most cases the selected stress level remains the same throughout the greater part of the test, but many machines require a certain time before a stationary state is reached, and in the later stage of the damage process when

localized yielding or cracking of the test piece occurs, a substantial change may occur. Possible changes are revealed by the measuring devices indicated in the preceding paragraph, and appropriate adjustment may then follow either by hand or automatically. The first alternative is, in general, used during the starting period, but in modern machines and equipments an automatic control throughout the test is frequently required, particularly in equipments of the resonant type. This procedure is, of course, easier when the load is produced by electrical forces but it is quite feasible in connexion with other machines.

As an example of such control methods in connexion with mechanical oscillators, reference is made to a paper by PERCIVAL and WECK (1947). A similar device is used in spring fatigue testing machines developed by COATES and POPE (1956). A spring and dash-pot lever mechanism is actuated by one of the vibrating masses and controls a servo-motor which varies a resistance in the electrical circuit of the main driving motor.

A simple control system which has been found to operate in a stable and satisfactory manner for long periods of continuous operation was developed by DOLAN (1951). A micrometer screw is attached to one of the two vibrating masses. At a given amplitude which can be pre-set to give a desired magnitude the contact made by this screw develops a small pulse from a battery. This pulse is smoothed and spread out over approximately a half period of the vibration and then subtracted from the generating current. The control circuit can be adjusted during operation so that the micrometer makes contact, say, every second or third cycle of vibration. The shut-off apparatus consisted of a piece of piano wire on which was slipped a small brass weight free to slide up and down the wire. The length of wire was adjusted to give a natural frequency slightly higher than the resonant frequency of the spring-mass system. Any drop in amplitude or frequency of the exciter caused the small weight to slide down the wire and actuated the shut-off.

In the torsional vibrator by WADE and GROOTENHUIS (1956), mentioned in paragraph (32.1), resonance was maintained by using an electrical feedback system, the initial signal being derived from the motion of the specimen by means of an induction coil pick-up. An elaborate, completely electronic amplitude and control system is described in the publication.

A load-control system in connexion with a hydraulic, non-resonant fatigue testing machine by MOORE (1956) has already been described in paragraph (33.4).

A load-control system of quite a different character is that developed by KENNEDY (1952, 1953) in a unit for combined creep and fatigue testing. It is required that the applied load be decreased in relation to the changing cross-section in order to maintain a constant stress. Assuming that there is no change in the density of the metal, this condition is equivalent to the condition that the product of the load applied and the specimen length shall be constant. This condition is simply achieved by arranging that the resistances in the opposite arms of a Wheatstone bridge vary according to the load and the length. For this purpose, two resistor transducers were connected across a load spring and a creep spring, respectively.

References: COATES and POPE (1956), DOLAN (1961), KENNEDY (1952, 1953), MOORE (1956), PERCIVAL and WECK (1947), WADE and GROOTENHUIS (1956).

34.5 Counters

The fatigue life is defined by the number of cycles imposed on the specimen until failure or some other specified event occurs. When the testing machine is driven by a rotating motor, the life is simply measured by a counter giving the number of revolutions from the start until the motor is stopped. This method is not feasible, however, in electrically excited resonant-type machines. In this case, it may be possible to measure the fatigue life directly in time (minutes) on condition that the frequency does not change by a measurable amount, and then to obtain the number of cycles by direct multiplication of the time and the frequency. This method was used by DOLAN (1951).

It is, of course, better to measure the exact number of cycles by means of an electric clock as done, for example, by WADE and GROOTENHUIS (1956).

In programme testing machines it is required that the load be changed automatically after a prescribed number of cycles. This may be realized by means of mechanical or electrical counters. A review of various counters of these types used in Germany is given by BECKER (1950). High-frequency pulses and electrical signals are conveniently counted by means of dekatron counters as done, for example, by KENNEDY and SLADE (1956). Another application in programme testing of this type of counter is described by MOORE (1956).

References: BECKER (1950), DOLAN (1951), KENNEDY and SLADE (1956), MOORE (1956), WADE and GROOTENHUIS (1956).

34.6 Frameworks

The different components of a testing machine such as guiding bars, loading frames, leverage, pivots and bearings are assembled and supported by a framework.

In order to reduce to a minimum the energy transmitted to the foundation and to isolate the testing machine and its parts from vibrations of surrounding machines, the framework sometimes includes a suspension rig, or is mounted on suitable springs, or placed on rubber mounts or cork. Undamped dynamic vibration absorbers have also been used (O'CONNOR and MORRISON, 1956).

As already mentioned, LEHR (1930) provided the framework with means to allow free horizontal vibrations, which were used as a measure of the load.

References: LEHR (1930), O'CONNOR and MORRISON (1956).

SECTION 35. CALIBRATION AND CHECKING OF TESTING MACHINES

35.0 General

The purpose of a fatigue-testing machine is to apply to the test piece an alternating load producing a well-defined stress distribution. This distribution should be reproducible within narrow limits, a requirement which

includes two aspects: the load should be reproduced with sufficient accuracy, and it should be transmitted to the test piece without undue scatter. For this purpose, the measuring devices of the machine should be calibrated and the proper function of its components should be checked at intervals in order to detect and eliminate the many errors which are so easily introduced. The demand of reduced scatter is—or should be—particularly severe in relation to fatigue machines. The reason why errors in these machines are so detrimental is explained by the fact that the end product of a conventional fatigue test is an observed fatigue life and this life is greatly affected by errors in load. This statement is easily demonstrated by an elementary calculation.

Suppose that the relation between load S and life N is represented by the expression (cf. Section 85)

$$S = (S_u - S_e)(N/B + 1)^{-a} + S_e$$

It then follows that

$$\frac{\mathrm{d}S}{S} : \frac{\mathrm{d}N}{N} = -a \frac{(S - S_e)}{S} \frac{N}{(N+B)}$$

or, for large values of N

$$-a(S - S_e)/S$$

Taking as a common average value $a = 0{\cdot}5$ and supposing that $(S - S_e)/S = 0{\cdot}1$, which corresponds to a stress level approximately 10 per cent above the fatigue limit, then it follows that

$$\mathrm{d}N/N = 20\ \mathrm{d}S/S$$

which implies that an error in the load of −3 per cent (which is frequently exceeded) corresponds to an increase in fatigue life of no less than 60 per cent. A stress level closer to the fatigue limit, say 5 per cent above it, implies that an error of 3 per cent in load corresponds to an error in life of 120 per cent.

The behaviour of a stationary fatigue machine is quite different from the machine under operating conditions with regard to the sources of errors, and, therefore, a static calibration is not sufficient but must be completed by the more complicated dynamic calibration.

35.1 Static Calibration and Checking

From the preceding it follows that the examination of the testing machine should include not only the load-producing mechanism but also the transmitting elements.

The calibration depends, of course, upon the method by which the load is produced. In machines where the load is generated by dead weights and constant spring forces or by lever and poise mechanism and transmitted to the specimen through a lever system, weights and forces have to be carefully controlled including weighing of all levers and other parts of the loading system, together with an experimental determination of the centres of gravity of these parts. This information, together with the geometry of the lever system, can be employed to calculate a calibration constant or to construct a calibration chart relating the applied dead weight or position of the poise or the reading of the spring deflexion to the load, bending moment, or torque actually applied to the specimen.

In machines where the load is produced by reciprocating masses or by centrifugal forces, it is of paramount importance to know exactly the speed of the machine, as an error in speed corresponds to a doubled error in load.

In hydraulic and pneumatic machines, pressure gauges should be carefully calibrated at intervals.

The greatest source of scatter in a fatigue testing machine appears, however, to be the grips and the guiding fixtures. For this reason, extraneous bending moments and twisting, introduced by the grips in axial-loading machines in particular, and friction caused by guide fixtures should be measured and, if possible, eliminated. There is reason to believe that appalling inaccuracies, easily ranging from ten to twenty or more per cent, may be incurred, if sufficient and meticulous care is neglected, as stated by O'Connor and Morrison (1956).

A valuable contribution to the solving of this problem has been presented by Findley (1947) by an apparatus which was designed with provision for detecting and correcting strains introduced into the specimen when it is fixed in the testing machine. The device consists of means for measuring six components of distortion in the upper end of the specimen with respect to the lower end. Axial load is detected by readings of a dynamometer dial. Displacement in either of two horizontal directions, bending in either of two planes, and twisting are indicated by means of five small dials indicating relative movement of two aluminium plates which are clamped by split collets to the upper and lower straight sections of the specimen. This detector is balanced statically about the centre-line of the specimen. Before a specimen is placed in the testing machine, the detector is clamped to the specimen and the five dials are set to zero. The specimen is then fastened to the testing machine by means of special collet-type chucks, which are so designed that the specimen and detector can be inserted in the machine without moving the heads of the machine from their normal position. Results obtained by means of this detector are indicated below.

The joint effect of all factors mentioned above (except the grips) may be determined by placing a weighing scale or a dynamometer in the machine and measuring repeatedly the result for different readings on the load scale.

A reliable and simple dynamometer is the "Morehouse proving ring" which is an elastic steel ring, designed primarily for determining static loads by micrometer measurements of the deflexion of the ring. Rings of less than 100,000 lb capacity can be carefully calibrated in precision dead-weight machines. A calibration factor, varying with deflexion, may thus be obtained for tension and compression, together with a temperature correction factor. The calibration factor remains constant over several years with normal care. This device has been thoroughly examined by Wilson, Tata and Borkowski (1946). In an adaption of a 25,000 lb ring by Wilson and Johnson (1937), where the micrometer and reed were replaced by an adjustable screw and an electrical contact with a neon glow lamp as

indicator, the sensitivity of the device when controlled by dead-weight loading was found to be less than 2 lb.

References: Findley (1947), Wilson and Johnson (1937), Wilson, Tata and Borkowski (1946).

35.2 Dynamic Calibration and Checking

When the testing machine is in operation, a new source of error, sometimes of considerable magnitude and non-existent in a stationary machine, appears due to unintentional inertia forces. In combination with springs or other flexible members, resonant vibrations will be generated which result in an appreciable increase in the errors in load.

An inertia effect which actually exists in every fatigue testing machine is introduced by the deflexion of the test piece in the load direction, thus imposing vibratory movements to the grips. In many cases this effect may be negligible but in others, as for example in hydraulic machines, this effect may result in errors in the load amounting to some 30 per cent of the maximum load. These inertia forces, acting on the specimen, may be eliminated by applying a spring force to the grip of such a magnitude that this spring-mass system has a natural frequency equal to that of the testing machine. This idea was introduced by Haigh (1912) in his electromagnetic testing machine, but is not easily applied to many types of machine. It is, however, possible to calculate a correction factor which may be applied to the nominal loads. As the additional forces on the grips are proportional to the masses and to the square of the speed, which are known, and to the deflexion of the specimen, which can be measured, a chart relating the correction factor with the deflexion of the specimen per unit load and the speed consists of a family of straight lines, each one corresponding to a certain frequency. This problem has been discussed by Hempel (1939), von Philipp (1942) and Pischel (1953) and has also been applied by Hempel and Fink (1953).

The influence of other inertia forces on the actual load is generally much more difficult to detect and eliminate. The flexibility of parts supposed to be rigid and the complicated distribution of masses in many machines make it hard to anticipate the natural frequencies of many possible spring-mass system that may develop within the machine. Such resonance regions are usually—because of small damping—very narrow. These difficulties and constructive means for dealing with them are discussed in papers by Mason (1917, 1921), Moore (1921), McAdam (1924), and others.

Many machines are rather weak in transverse directions, and this may result in large transverse vibrations at one or more specific speeds of the machine.

The preceding considerations emphasize the necessity of dynamic calibration. Some of these methods do not simulate closely enough the properties of the specimen. Methods of this type are those using a dynamometer or a proving ring as, for example, demonstrated by Thurston (1948). Valuable and even indispensable as they may be, the effect of the grip is not revealed, as only the total load and not its eccentricity is measured.

For this purpose, the most reliable method of measuring the load distribution within the specimen is, for the present, the application of electrical resistance strain gauges directly to the test piece. The technique of this valuable tool of measurement is now very well developed, and some of its merits will be indicated in Chapter IV. Reference will here be given to two comprehensive reviews, viz. one by ROBERTS (1946) and another by HUGGENBERGER and SCHWAIGERER (1958).

Suffice it to mention here, that an accuracy better than one per cent will require an advanced technique, and that strain gauges are not very resistant to repeated strains. Fatigue failure will be expected to occur after a few thousand cycles at strain amplitudes of the magnitude 0·2 per cent.

References: HAIGH (1912), HEMPEL (1939), HUGGENBERGER and SCHWAIGERER (1958), MASON (1917, 1921), MCADAM (1924), MOORE (1921), v. PHILLIPP (1942), PISCHEL (1953) ROBERTS (1946), THURSTON (1948), WILSON and JOHNSON (1937)

SECTION 36. ACCURACIES OF ACTUAL TESTING MACHINES AND EQUIPMENTS

In this section, data will be given of accuracies actually attained and measured in fatigue testing machines. Up to the present, however, such data are scarce and not easily given in general form, as the accuracy depends upon the individual care of calibration, static and dynamic, and on the proper maintenance of the machine and its function. This problem has been discussed by ERLINGER (1936).

According to a comprehensive survey of various conventional machines, FINK and HEMPEL (1951) and HEMPEL and FINK (1953) found that the accuracy depends upon three different factors: (1) the design of the machine; (2) the use of the machine and resulting wear in the bearings; (3) the proper manipulation of the machine according to established instructions.

In the above-mentioned investigations which were carried out by means of electrical strain gauges, deviations of the actual load from the nominal load of more than 30 to 40 per cent were observed in some cases. When caused by uncorrected inertia forces, the accuracy could be substantially improved by applying correction factors as mentioned above, but in some machines the errors resulted from the design of the machine or neglected maintenance of its proper function. The latter objection applied particularly to hydraulic machines.

Even if errors of this magnitude, though actually existing perhaps more frequently than anticipated, may be removed without excessive difficulty, it appears, on the other hand, that accuracies better than 3 per cent will require considerable skill. As previously mentioned, this statement is confirmed by the results obtained by O'CONNOR and MORRISON (1956), who after many precautions could attain an accuracy in the load of about $1\frac{1}{2}$ per cent.

In the fatigue testing of coil springs by COATES and POPE (1956), the stresses produced under dynamic straining conditions were determined by electrical strain gauges. Examination of the results showed that an accuracy

of $\pm 2\frac{1}{2}$ per cent was attainable in the middle ranges of load amplitude, but that in the lower ranges the percentage accuracy was not so high.

GROVER, HYLER, KUHN, LANDERS and HOWELL (1953) found that the accuracy of load-measuring apparatus is approximately 1 per cent. Frequent monitoring revealed, however, that the loads sometimes but rarely changed as much as 3 per cent during any given test. If guide plates were used, (in compression tests) the accuracy of the load was estimated to be about ± 5 per cent.

KEPERT and PAYNE (1956) examined the fatigue characteristics of a typical metal wing using a vibration rig. The accuracy of loading was checked by numerous electric strain gauge readings and also by deflexion measurements during the test. They concluded that the applied load is accurately known within ± 5 per cent.

As a general conclusion, it may be stated that an accuracy of ± 3 per cent seems to be generally accepted as satisfactory, that in some cases the error may be considerably much larger, and that an accuracy of 1 per cent is comparatively seldom attained.

These results may perhaps appear to be too pessimistic when rotating bending machines are concerned, but, in fact, in the author's experience errors in the load of ten or more per cent may easily occur, if vibrations due to eccentric mounting of the specimen are not effectively eliminated by inserting sufficiently weak suspension springs between the dead weight and the specimen.

References: COATE and POPE (1956), ERLINGER (1936), FINK and HEMPEL (1951), GROVER, HYLER, KUHN, LANDERS and HOWELL (1953), HEMPEL and FINK (1953), KEPERT and PAYNE (1956), O'CONNOR and MORRISON (1956).