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of the Engineering Division, N.P.L.

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Summary.—(*a*) *Purpose of Note.*—When any elastic system is subjected to a range of load, the energy stored in the system fluctuates between two values corresponding to the maximum and minimum loads of the cycle. Since the driving unit usually supplies energy at a constant rate (corresponding to losses in the machine), the energy given out by the specimen during one quarter cycle must be stored in some manner and then returned during the following quarter cycle. In existing types of fatigue-testing machine this energy is stored in a rotating mass (flywheel) and gives rise to cyclic variations in its speed. The object of this note is to indicate that the energy can be better stored in an *oscillating* mass which is, ideally, attached directly to the specimen. When this is done the full specimen load is transmitted through a direct (solid) connection and does not travel through the linkwork to the operating mechanism of the machine. As a consequence the operating mechanism can be made much lighter, the driving power required is considerably reduced and the machine can be run at a much higher speed (if desired). The dual problem of starting the machine and also ensuring stable operation can be solved by the use of a 'slipping clutch' already employed successfully in another connection.

(*b*) *Range of Note.*—The general principles governing the design of fatigue-testing machines are considered, and the operation of a machine in which the range of load is generated by resonant oscillation of a mass on the end of a spring is examined. The advantages attendant upon operation at the resonant speed are demonstrated, and a driving mechanism which ensures stable operation at the resonant frequency is described. The construction and operation of a direct-stress fatigue-testing machine embodying the principles of resonant oscillation are described and skeleton designs for a torsional fatigue-testing machine suitable for testing specimens 3 in. in diameter and for a direct-stress-fatigue testing machine to give a range of 50 ton are appended.

(*c*) *Conclusions.*—It is shown that a machine in which the range of load is generated by resonant oscillation of a mass on the end of a spring can be driven by a 'slipping clutch' device in such a manner that the machine must either give the pre-set load range corresponding to the stroke of the exciting mechanism, or fail to run at all. It is shown that under certain conditions the range of load in the specimen may be rendered effectively independent of the stiffness of the specimen. It is shown that, in existing types of testing machine the driving mechanism has to supply the maximum and minimum loads of the cycle, in the resonant spring type of machine, the driving mechanism has to supply only the forces necessary to compensate for friction damping, etc. The saving in size, weight and power output of the driving mechanism is thus considerable. As a result, the construction of large fatigue-testing machines operating at high speeds becomes feasible.

(*d*) *Further Developments.*—It is suggested that the construction of a direct-stress fatigue-testing machine giving a range of 50 ton and/or a torsional fatigue testing machine capable of testing 3-in. diameter shafts should be considered.

Introduction.—(*a*) In any fatigue-testing machine a rapidly alternating‡ force, torque, or bending moment must be applied to the specimen. Except when fracture occurs after comparatively few cycles, the net energy absorbed by the specimen per whole cycle is only a small fraction of energy stored by the specimen in the condition of maximum strain. In all existing designs of machine (Wöhlers and similar rotary types excluded) this strain energy is returned every half-cycle through the loading mechanism to the actuating member of the machine and causes a cyclic irregularity which is kept small by the provision of a suitable fly-wheel. It is clear that the mechanism of the

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† This paper was originally communicated in July, 1935.

‡ Repeated and non-alternating stress systems are of course obtained by superimposing a steady stress on to an alternating one. The imposition of such a steady stress is immaterial in the following treatment.

apparatus must be designed to transmit this large interchange of energy between specimen and fly-wheel and that this mechanism will be proportionately heavy. Further, in existing designs this transmission of energy is done inefficiently so that much more energy is dissipated in the machine than in the specimen.

The storage of energy in a specimen is proportional to the strain and therefore (for S.H.M. distortions) is a maximum when the straining head of the machine has zero velocity and is zero when the head has a maximum velocity. If, now, a suitable mass or moment of inertia is attached directly to the straining head, the energy stored in this mass, due to its movement will be supplementary to that in the specimen and the sum of the two energies—specimen and mass—will be constant. It is clear therefore that if a suitable mass be added to the straining head of the machine the operating mechanism will no longer have to transmit the energy stored in the specimen but only the losses; it can then be made very much smaller and lighter, *provided that means are provided* for giving the necessary energy to the specimen-mass system when starting the machine.

(*b*) Such a system of specimen and mass, when operating under the conditions just laid down, is of course in oscillation at its own natural frequency. The amplitude of the motion will therefore depend upon the energy supplied, the damping (energy dissipation) and upon time, if the former two are not equal. Consequently, for the arrangement to constitute a satisfactory fatigue-testing machine, an accurate and close control of amplitude is necessary. This can be done by coupling the mass through a long connecting-rod to a crank of suitable throw; for, in this way, if the amplitude tends to become too small, energy is given to the system by the crank and *vice versa*. Such an arrangement however nullifies the advantage of the proposed scheme, since, on starting up, the connecting rod has to carry the full strain energy of the specimen (and hence the maximum load) and must be proportionately strong. If, to reduce the load on the crank pin when starting, the connecting-rod be made elastic, although the machine will operate correctly at one definite speed, determined by the specimen stiffness, the amplitude will vary very rapidly with speed and the arrangement is so unstable as to be quite unusuable.

(*c*) If the connecting-rod is made telescopic, with solid friction in the slides, the operation at once becomes quite different. As in the case of an elastic rod the crank pin load is reduced to a known small quantity (only a fraction of the full load capacity of the machine) so that starting up causes no difficulty. When operating at the correct speed, the energy required by the specimen-mass system per cycle is quite small and can easily be transmitted through the rod, which locks and operates as a rigid member. At intermediate speeds the rod can transmit, per cycle, an amount of energy equal to the frictional force in the slides (here assumed constant) multiplied by twice the motion of the end attached to the specimen. Suppose that the head of the machine be rigidly clamped and the crank is run up to the correct operating speed. Now, release the clamp: the small force due to the friction in the rod will cause a small strain of the specimen, and energy will be given to the specimen-mass system. This energy is applied as an alternating force in synchronism with the motion of the straining head of the machine, and the oscillation of the latter will therefore increase rapidly in amplitude until it attains a value equal to the throw of the crank, when the rod will lock and operate as a rigid link. An increase of amplitude beyond this value is impossible, for it would involve a reversal of the phase of the connecting-rod extension with reference to the motion, the force of friction would accordingly reverse and so would the force applied to the straining head.

At crank speeds slightly removed from the correct value, the force necessary to give forced oscillations of the specimen-mass system may be less than the limiting friction in the rod. In such conditions therefore the rod will remain rigid and the strain head will have the crank amplitude*.

It appears therefore that a system as outlined above will (i) be quite stable, and (ii) have a constant amplitude over a small range of speed, whilst (iii) requiring operating mechanism capable of transmitting loads much smaller than the capacity of the machine. Further (as a

* *Phil. Mag.*, Vol. XI, 1931, p. 517.

result of (iii)) (iv) the operating speed may be increased to 5000 to 6000 cycles/min. A preliminary experimental trial has confirmed all these conclusions (*see* Appendix).

Application of the Resonance Principle to Practicable Fatigue-Testing Machines.—The practical difficulties of the elementary machine running at resonance are all connected with the stiffness of the test piece. Firstly, this stiffness will normally be so great that, to bring the resonant frequency down to a reasonable value, a very large inertia will be required. Secondly, for the same reason, the amplitude of oscillation required to produce stresses in the neighbourhood of the safe fatigue range will be inconveniently small. Thirdly, any variation of the stiffness of the test piece during test will vary the range of stress imposed and also the 'correct' speed of the machine. The obvious method of overcoming all these difficulties is to interpose between the test piece and what has been termed the straining head a spring having a stiffness very much less than that of the test piece. If the spring be made so flexible that the stiffness of the test piece can be regarded as infinite in comparison, this modification would convert the arrangement into a constant-stress machine, and the range of stress imposed would be proportional merely to the amplitude of oscillation imposed on the free end of the spring. The stiffness of the normal test piece is not, however, so great that this requisite condition may easily be fulfilled; the effective length of spring and the amplitude of oscillation required would both be inconveniently large. A machine with a practicable value of stiffness of the stressing spring will give conditions intermediate between constant stress and constant strain. The amplitude of total strain of the stressing spring will be less than the amplitude of oscillation of the exciting mechanism by the amplitude of oscillation of the end of the test piece. In such a machine, if the effective stiffness of the test piece is not known before test or if this stiffness varies during test, the range of stress applied can only be assessed by measurement of the *strain* of the stressing spring, *i.e.*, by measurement of the amplitude both of the oscillation imposed on the free end and of the oscillation of the end of the test piece*. The complication of the dual measurement is of course inconvenient; but the more serious objection is that it is very difficult to pre-determine the range of stress imposed, and that, if the stiffness of the test piece varies during test, *both* the range of stress *and* the range of strain imposed vary in sympathy.

Compensation for Finite Specimen Stiffness.—The amplitude of oscillation of the end of the test piece reduces the strain of the stressing spring and thus causes the load applied to the test piece to be reduced. Compensation for this reduction may be provided by adding a mass or fly-wheel at the junction between the stressing spring and the test piece. Fig. 1 is a schematic representation of a torsion fatigue-testing machine.

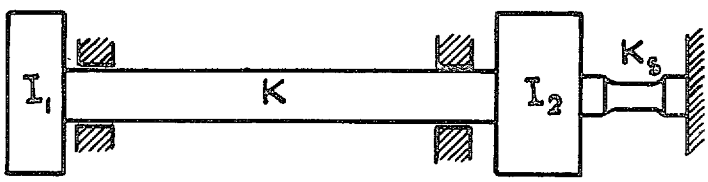

FIG. 1.

Let the angular displacement of I_1 at any time t be $\theta_0 \sin pt$ and of I_2, $\phi_0 \sin pt$.

Then $$K(\theta_0 - \phi_0) - K_s\phi_0 = -\frac{p^2 I_2}{g}\phi_0 \,, \qquad (1)$$

whence semi-range of load on test piece

$$= K_s\phi_0 = \frac{K\theta_0}{1 + (1/K_s)(K - p^2 I_2/g)} \qquad (1a)$$

* A torsion fatigue-testing machine of this type has been in use in Engineering Department of the National Physical Laboratory for some years. In this machine, however, no attempt is made to run at the resonant frequency, the exciting mechanism being direct-coupled to the stress bar and sufficiently powerful to maintain the oscillation at all speeds.

Thus the range of load on the test piece can be made independent of its stiffness if

$$(K - p^2 I_2/g) = 0. \qquad \text{(2)}$$

If, in order to allow mean stress to be applied, a second stressing spring K' is added (Fig. 2), relation (1) becomes

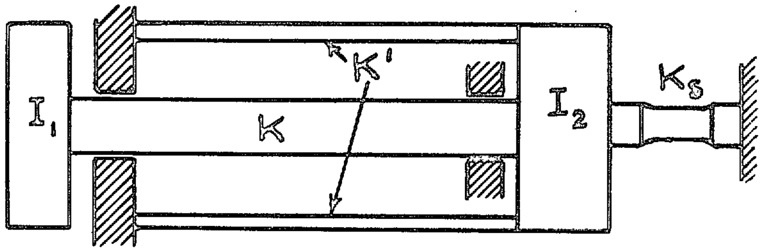

FIG. 2.

$$K(\theta_0 - \phi_0) - K_s\phi_0 - K'\phi_0 = \frac{p^2 I_2}{g}\phi_0 \qquad \text{(3)}$$

or

$$K_s\phi_0 = \frac{K\theta_0}{1 + (1/K_s)(K + K' - p^2 I_2/g)} \qquad \text{(3a)}$$

So that the relation (2) becomes $(K + K' - p^2 I_2/g) = 0$. (4)

Neglecting friction and damping, the couple required to maintain the oscillation of the disc I_1 is

$$\left\{K(\theta_0 - \phi_0) - \frac{p^2 I_1}{g}\theta_0\right\} \sin pt \qquad \text{(5)}$$

$$= \left\{1 - \frac{p^2 I_1}{Kg} - \frac{K}{K_s + K + K' - p^2 I_2/g}\right\} K\theta_0 \sin pt. \qquad \text{(5a)}$$

The condition that this couple should be zero is

$$1 - \frac{p^2 I_1}{Kg} - \frac{K}{K_s + K + K' - p^2 I_2/g} = 0. \qquad \text{(6)}$$

If equation (4) is satisfied, the semi-range of load applied to the test piece is $K\theta_0$, i.e., the load on the test piece is dependent only on the angle of oscillation of the disc I_1. If equation (6) is satisfied, the exciting torque required is only that necessary to overcome frictional forces*. Unless provision be made for varying I_1 or I_2 in sympathy with K_s†, p is the only variable and equations (4) and (6) can be simultaneously satisfied only by K_s infinite and one finite value of K_s.

There are thus three possibilities:—

(i) To run the machine at the frequency p_0 defined by the equation $K + K' = p_0^2 I_2/g$ and to provide sufficient friction in the connecting rod to permit operation of the machine at this speed over a range of values of the true resonant frequency p defined by equation (6) by insertion of the range of values of K_s for which provision must be made.

(ii) To operate as in (i) at the frequency p_0 and to satisfy equation (6) by varying I_1. (The alternative of varying I_2 is obviously less convenient since it involves varying p_0 also.)

(iii) To operate always at the true resonant frequency p (dependent upon K_s) and to choose the values of K, K', I_1 and I_2 in such a manner as to make the variation of the ratio $K_s\phi_0/K\theta_0$ over

* Including internal friction in the specimen.
† It is assumed that K and K' must be regarded as fixed.

the normal range of K_s the least possible. These possibilities are examined in detail below.

(*a*) *Constant Frequency*

Equation (6) defining the resonant frequency at the input end, may be written

$$1 - \frac{p^2}{p_1{}^2} - \frac{K}{K_s + (K + K')(1 - p^2/p_0{}^2)} = 0 \qquad \ldots \quad \ldots \quad \ldots \qquad (6a)$$

where $p_1{}^2 = Kg/I_1$ and $p_0{}^2 = \dfrac{(K + K')g}{I_2}$.

p_1 is at our disposal (by adjusting I_1) and may be chosen to satisfy equation (6a) with $p = p_0$ for any one value of $K_s(= K_s')$

i.e.,
$$\frac{p_0{}^2}{p_1{}^2} = 1 - \frac{K}{K_s'}.$$

Equation (6a) then becomes

$$1 - \left(1 - \frac{K}{K_s'}\right)\frac{p^2}{p_0{}^2} - \frac{1}{\dfrac{K_s}{K} + \left(1 + \dfrac{K'}{K}\right)\left(1 - \dfrac{p^2}{p_0{}^2}\right)} = 0. \qquad \ldots \quad \ldots \qquad (6b)$$

This equation may be used to determine the variation of the resonant frequency p with variation of the ratio K_s/K for various values of the ratios K_s'/K and K'/K. For this purpose, it is convenient to write the equation in the form

$$K_s/K = \frac{1}{1 - \left(1 - \dfrac{K}{K_s'}\right)\dfrac{p^2}{p_0{}^2}} - \left(1 + \frac{K'}{K}\right)\left(1 - \frac{p^2}{p_0{}^2}\right). \qquad \ldots \quad \ldots \quad \ldots \quad \ldots \qquad (6c)$$

For values of p in the neighbourhood of p_0, the second term on the right-hand side is small in comparison with the first. Variations of the value of K' therefore make very little difference and we shall consider only the case $K' = K$.

In Fig. 3 are shown a number of curves of p/p_0 against K_s/K for values of K_s'/K, from 2·8 to 10. These curves show that, the lower the value of K_s'/K, the greater is the variation of p/p_0 with K_s/K, so that it is advantageous to keep K small in comparison with the normal values of K_s.

Practicable limitations on the size of the testing machine and the desirability of using a fairly long test piece render it necessary to provide for values of the ratio K_s/K at least as low as 5 and preferably lower still. Therefore, if I_1 is to be definitely fixed, K_s'/K cannot be made much more than 7, for which value of K_s'/K values of K_s/K from 3 to infinity can be accommodated with a variation of resonant frequency from $0{\cdot}90p_0$ to $1{\cdot}08p_0$.

The main disadvantage of variation in the resonant frequency lies in the reaction on the actuating mechanism. If the inertia I_1 is forced to oscillate at a frequency different from the natural frequency, the actuating mechanism has to supply a torque in phase with the angle of oscillation. The forces necessary to produce this torque react on the actuating mechanism and tend to vary its speed. Thus if the oscillation is produced by a crank and connecting-rod mechanism, the forces in the connecting-rod tend to slow the crank down in one quarter of a revolution and to accelerate it in the next. A fly-wheel attached to the crank will help to diminish the effect; but the motor speed cannot be maintained truly constant and the oscillation of I_1 will not be simple harmonic. The introduction of harmonics will cause the balance for oscillations of I_2 to be inaccurate and the equation $K_s\phi_0/K\theta_0 = 1$ will be rendered inexact.

The torque (other than that due to friction, etc.) that the actuating mechanism has to supply is from equations (5a) and (6b), putting $p/p_0 = 1$, $(K/K_s' - K/K_s)K\theta \sin pt$; but from equation (6c)

$K/K_s \simeq 1 - (1 - K/K_s')p^2/p_0^2$. Therefore the torque required is approximately

$$(1 - K/K_s')(p^2/p_0^2 - 1)K\theta_0 \sin pt$$
$$= \frac{p_0^2}{p_1^2}\left(\frac{p^2}{p_0^2} - 1\right) K\theta_0 \sin pt$$
$$= \left(\frac{p^2}{p_0^2} - 1\right)\frac{p_0^2 I_1}{g} \sin pt. \qquad \text{(8)}$$

Therefore in designing the machine, it is necessary to provide for a definite range of values of p/p_0; but this range should be as small as possible.

(*b*) *Constant Frequency: Tuning by varying I_1 in steps*

Referring again to Fig. 3, it will be seen that, if it is required to keep the variation of p/p_0 within any particular limits, provision for a range of values of K_s/K greater than that which can be covered by one of the curves of this diagram may be made by using two or more curves. Thus if p/p_0 is to be allowed to vary from $0{\cdot}97$ to $1{\cdot}03$, we can use either of the schemes:

$I_1 = 0{\cdot}32 I_2$	$K_s/K = 2{\cdot}4$ to $3{\cdot}3$
$I_1 = 0{\cdot}375 I_2$	$K_s/K = 3{\cdot}3$ to $5{\cdot}0$
$I_1 = 0{\cdot}43 I_2$	$K_s/K = 5{\cdot}0$ to $11{\cdot}2$

or

$I_1 = 0{\cdot}35 I_2$	$K_s/K = 2{\cdot}8$ to $4{\cdot}0$
$I_1 = 0{\cdot}40 I_2$	$K_s/K = 4{\cdot}0$ to $6{\cdot}7$
$I_1 = 0{\cdot}45 I_2$	$K_s/K = 6{\cdot}4$ to $22{\cdot}2$

Similar schemes of other limits of p/p_0 and K_s/K can easily be devised. It will be noticed that the necessary variations I_1 are not large, so that the adjustment would be easy to make in practice. It would of course be necessary to estimate the value of K_s/K before test; but the estimate need not be very exact, and, if the first value proved to be very much in error, the machine could be stopped and the value of I_1 re-adjusted.

(*c*) *Input Tuned: Frequency Variable*

Using the same notation as in (1), equation (6c) may be taken to define the resonant frequency p in terms of K_s/K for given values of K'/K and K_s'/K. The 'load factor' $K_s\phi_0/K\theta_0$ is given by equation (3a), *i.e.*,

$$\frac{K_s\phi_0}{K\theta_0} = \frac{K_s/K}{(K_s/K) + (1 + K'/K)(1 - p^2/p_0^2)} = \frac{K_s}{K}\left(1 - \frac{p^2}{p_1^2}\right). \qquad \text{(10)}$$

Curves of $K_s\phi_0/K\theta_0$ against K_s/K for various values of K_s'/K are shown in Fig. 4.

The disadvantages of this type of machine are that the correct speed of operation has to be determined afresh for each specimen and that the range of load applied to the specimen depends to some extent upon its stiffness. If the stiffness of the test piece were always at least 7 times the stiffness of the stressing spring, the variation of the range of load applied could be made negligible; but, if this course were not practicable, this type of machine would probably prove less convenient than a machine of the type operating at constant frequency.

Variation of Stress with Frequency.—In each of the three possible types of machine, the maintenance of a constant amplitude of the free end of the stressing spring ensures that the load range applied to the test piece is also constant only provided the speed of the machine has the value p_0 in cases (*a*) and (*b*) or the value p (defined by equation (6c)) in case (*c*). Small variations of speed about the correct value will however have only a small effect on the load range.

Thus $F = \dfrac{K_s\phi_0}{K\theta_0} = \dfrac{K_s/K}{(K_s/K) + (1 + K'/K)(1 - p^2/p_0^2)}$

Therefore $\dfrac{\text{percentage variation of load range}}{\text{percentage variation of speed}} = (p/F)(\partial F/\partial p)$

$$= \frac{2p^2/p_0^2}{[K_s/(K + K')] + 1 - p^2/p_0^2}$$

In cases (1) and (2), the correct value of p is p_0, so that

$$(p/F)(\partial F/\partial p) = 2[(K + K')/K_s] = 4K/K_s \text{ if } K = K'.$$

The percentage variation of load range is therefore less than the percentage variation of speed of $K_s/K > 4$.

In case (c), the useful parts of the curves of Fig. 4 lying above the line $F = 1$, correspond to values of $p > p_0$, therefore the value of $(p/F)(\partial F/\partial p)$ is greater in case (c) than in cases (a) or (b). This is an additional reason why the constant-speed type of machine is preferable.

Reaction on Foundation of Machine.—It will be seen that whatever the type of machine, the load applied to the specimen is transmitted directly to the foundation of the machine. It will be necessary therefore either to have a fairly heavy foundation or alternatively to balance the machine by adding a second mass or fly-wheel oscillating in opposite phase. The balancing arrangement would almost certainly require no driving mechanism but would operate by resonance, as a 'vibration damper'. If such a device were fitted it would be advantageous to mount the whole machine on a spring suspension so that the energy dissipation would be reduced to a minimum. This would produce a corresponding reduction in the power required by the machine.

APPENDIX

Design of Actual Testing Machines.—In the discussion of the principles of the new type of fatigue-testing machine, a machine giving alternating torsion stresses has been considered; but the same principles can be applied to direct-stress or flexural fatigue-testing machines. Some precautions must of course be taken to prevent the excitation of oscillations in modes other than that required; but this requirement will not ordinarily cause any difficulty. In the case of torsion, the provision of the necessary restriction by bearings may easily be visualized; it is for this reason that the torsion case has been used for purpose of illustration.

A direct-stress fatigue-testing machine embodying the major principle discussed in the first section of this paper has been constructed in the Engineering Department of the N.P.L. In this machine, which is shown diagrammatically in Fig. 5, the required alternating force was obtained by vibrating a loaded cantilever; provision had therefore to be made for separating the alternating force from the alternating bending moment produced at the fixed end of the cantilever and applying only the former to the test piece. For this purpose the cantilever was anchored to a bracket supported on two parallel double cantilever springs bolted to the bed-plate of the machine. The bracket was thus constrained to move in a line parallel to the direction of oscillation of the main spring; whilst its motion in this direction was resisted by tensile or compressive forces set up in the test piece by which it was also connected to the bedplate of the machine. The main spring was tuned to resonate at the resonant frequency of the straining head on the supporting springs, so that the force applied to the test piece was merely the shearing force at the root of the main spring due to its vibration*.

The main spring was excited by a crank and connecting-rod which actuated a plunger working in a slide attached to the free end of the cantilever. The friction of the plunger in the slide could be varied within certain limits.

This machine was successfully operated at 5000 to 6000 cycles per minute and gave a load range of about ± 600 lb sufficient to break mild steel specimens $\frac{1}{8}$-in. in diameter. (The range of bending moment produced proved sufficient to fracture in bending the two $\frac{3}{4}$-in. bolts by which the cantilever spring was clamped to the straining head.) The main spring was about 10-in. long by 3-in. wide by $\frac{3}{8}$-in. thick and the amplitude of vibration at the free end was about $\pm \frac{1}{8}$ in. The power (including fairly high losses) required to drive the machine was about 1/20 h.p.

It will be appreciated from these figures that the new type of machine has a much higher mechanical efficiency than other existing types, and that the construction of really large fatigue-testing machines of this type is quite feasible. As an example, a schematic design excluding the driving mechanism for a torsion fatigue-testing machine capable of stressing 3-in. diameter shafts to ± 20 ton/sq in. is shown in Fig. 6. As a further more detailed example a schematic design for a 50-ton direct-stress fatigue-testing machine is shown in Fig. 7 and is described below.

Outline Design for a Direct-Stress Machine of 50 *ton Capacity Suitable for a Maximum Specimen Strain of* $0 \cdot 2$ *in.*—For a specimen strain of $0 \cdot 2$-in. total, a displacement of the mass of 3-in. total gives a convenient ratio of K_s/K and at the same time suitable sizes of mass and springs, at a working speed of 3000 cycles per minute.

The suggested layout is given in Fig. 7. The oscillating mass consists of two steel (or C.I.) plates about 23-in. diameter $\times$ $\frac{5}{8}$-in. thick held about 14 in. apart by bolts and distance pieces. The straining head of the machine is located between these two plates by batteries each of 15 springs (arranged in two concentric circles of 5 and 10 springs respectively). The straining head itself is a plate about 33-in. diameter $\times$ $1\frac{1}{4}$-in. thick. The outer edge of this plate is held between batteries each of 15 springs compressed between fixed rings attached to the frame of the machine. The purpose of these outer springs is to supply the mean load.

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* No compensation for finite specimen stiffness was attempted.

The whole machine is balanced by a second mass, made up of two discs with 30 similar springs and excited in opposite phase to the main mass. This can be located either at the bottom of the machine (in a pit) or immediately below the main mass.

All springs will be similar, about 6 coils 4-in. diameter, $\frac{3}{4}$-in. wire. By the use of a large number the construction is simplified and asymmetry due to non-uniformity is reduced.

Adjustment of load will be provided by a variable-throw crank built up as an eccentric sleeve on a fixed eccentric. Rotation of the former upon the latter will give a variation of stroke from 0 to 3 in. Approximate sizes of bearings are indicated upon the sketch.

The friction clutch between the cross-head and oscillating mass will be operated by oil pressure so that the machine can be started very easily. The same oil pump will supply forced lubrication to all bearings.

The machine will run from a synchronous motor and will probably require about 3 h.p.

It is proposed to compensate for angularity of the connecting-rod by providing for the crank to rotate at a non-uniform angular velocity, corresponding to sinusoidal motion of the cross-head. This can be done by mounting an assembly of three fly-wheels upon the main shaft, their locations and moments of inertia being so adjusted that the mechanical impedance of the crank to angular oscillations is zero at 50 and 150 cycles/sec. It can be shown that if this is done no force of 100 and 200 cycles can be applied to the cross-head, and the operation will be quite stable*. At the same time the mechanical impedance of the crank can be made infinite at 100 cycles and this will prevent cyclic irregularities from the ordinary inertia terms in the motion of the connecting-rod.

The form of the fixed head has not been sketched as this can be of any convenient form.

Tuning of the main mass to correspond to the specimen stiffness can be done either (*a*) by the addition of small masses bolted on to the main one, (*b*) preferably by displacing mercury (through a pipe coiled as a helical spring) into a cylinder attached to the mass and fitted with a spring-loaded plunger. In this way smooth tuning can be obtained.

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* It is desirable to test this theoretical deduction by a small scale model.

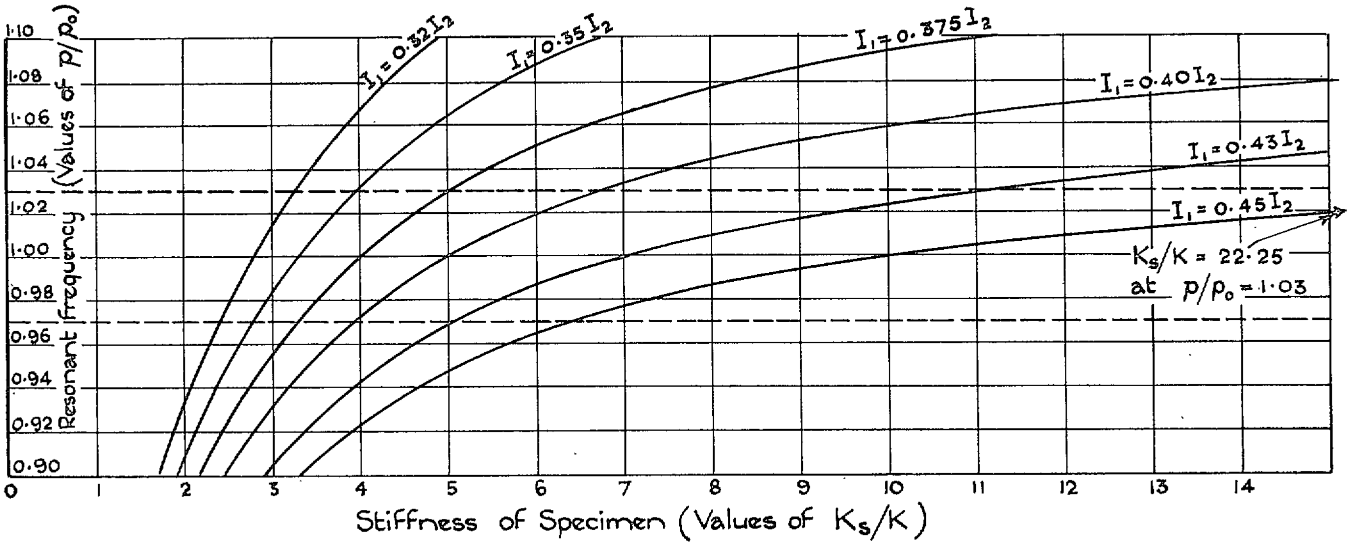

FIG. 3. Variation of resonant frequency with stiffness of specimen.

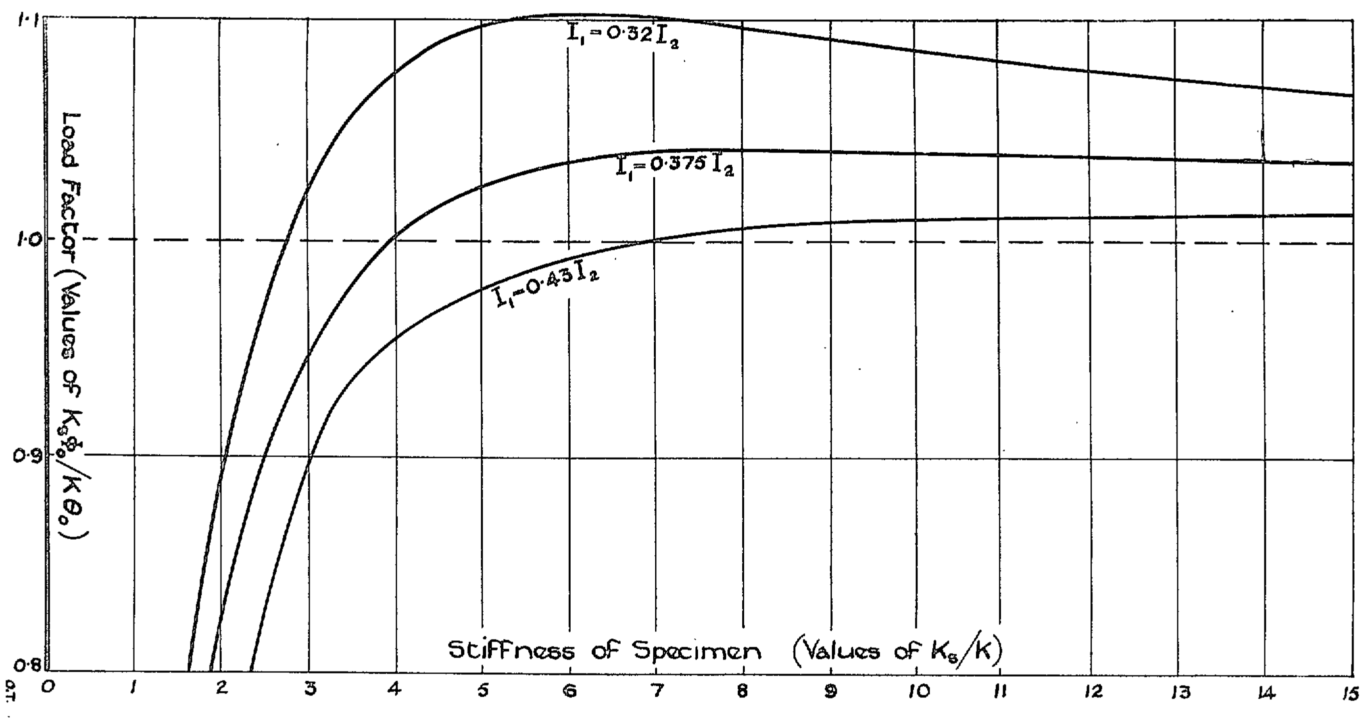

FIG. 4. Variation of load factor (at resonant speed) with stiffness of specimen.

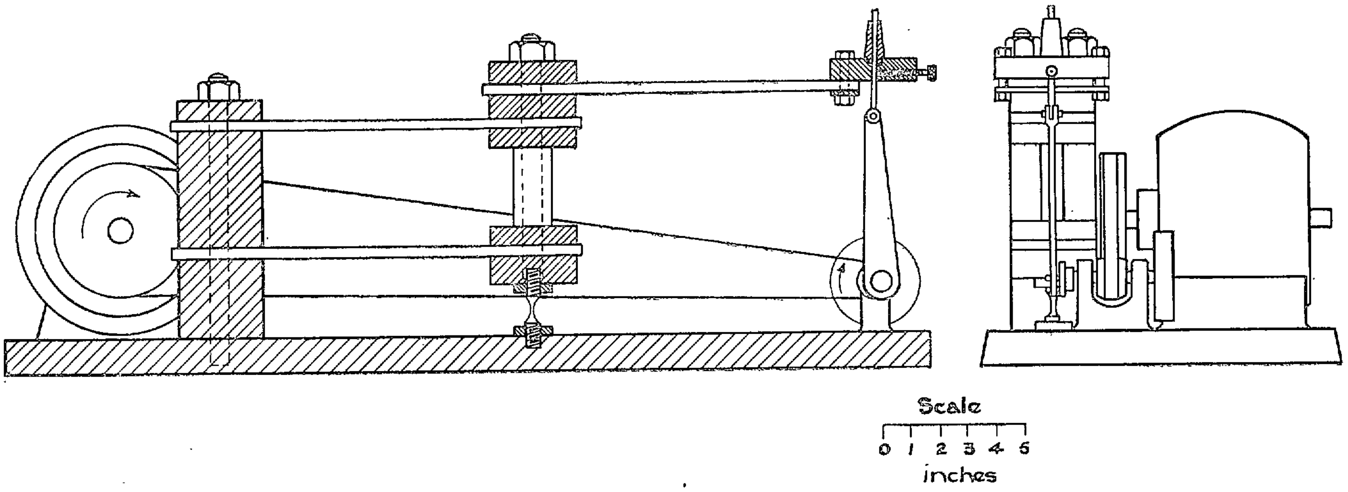

FIG. 5. Experimental direct-stress fatigue-testing machine.

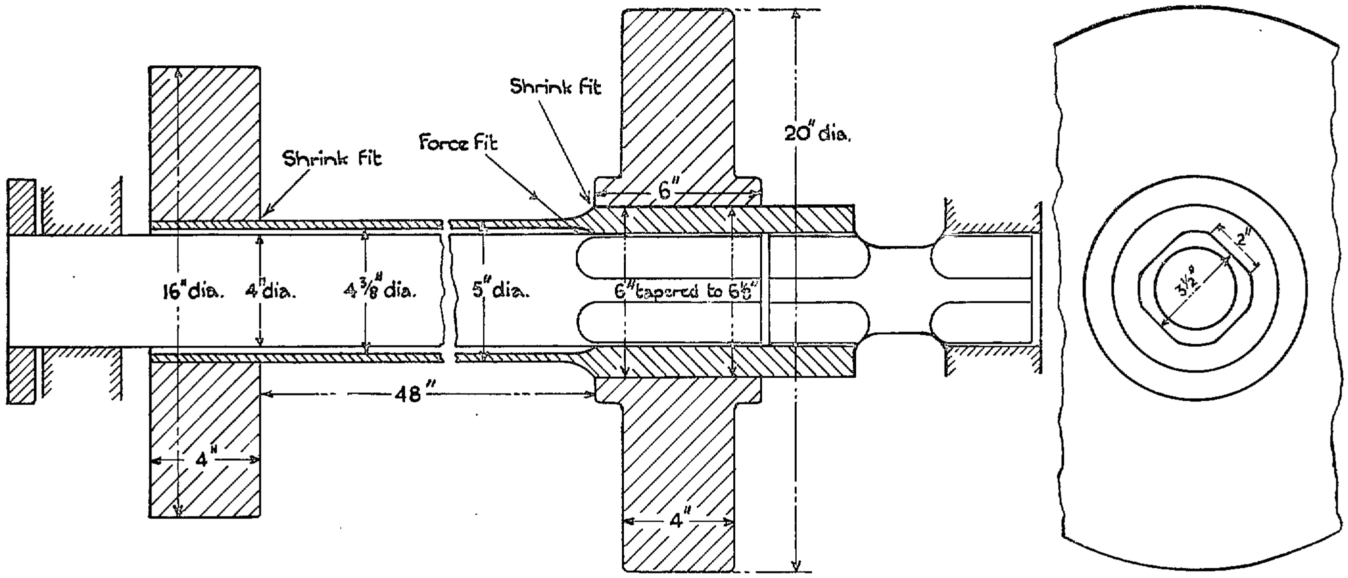

FIG. 6. Schematic design of torsion fatigue-testing machine for 3-in. diameter specimens.

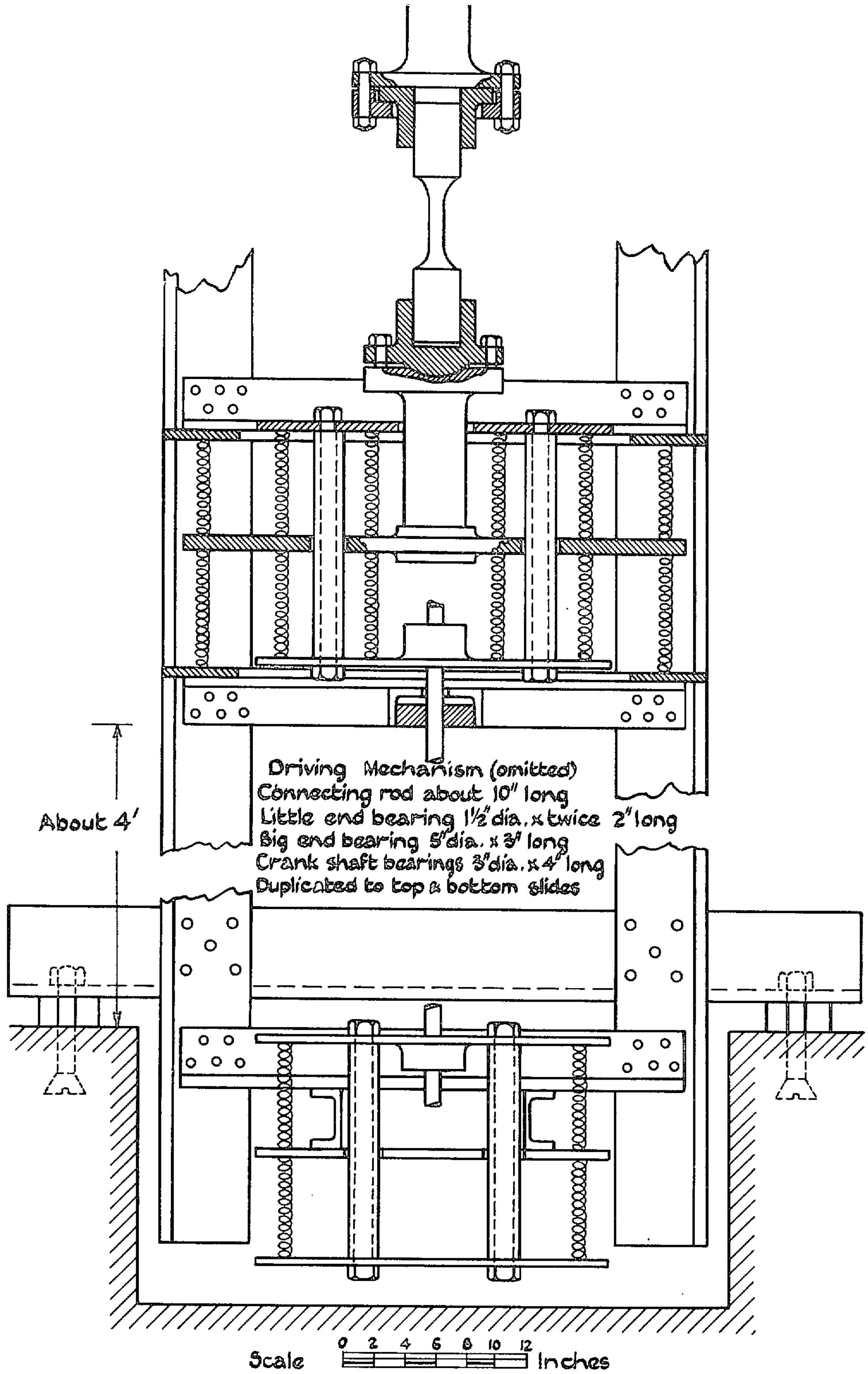

FIG. 7. Schematic diagram of direct-stress fatigue-testing machine giving load range of 50 ton.

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