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Electrical Resistance Wire Strain Gauges, with Particular Reference to Possible Errors in their Use for Static and Dynamic Measurements.

By F. AUGHTIE, Ph.D., M.Sc., A.M.I.Mech.E., A.M.I.E.E.

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Synopsis.

After a brief description of the general principles of operation of the electrical resistance wire strain gauge and of the different forms of construction now available, and a mention of glues for use with the gauges, bridge circuits are shown and described for the measurement of static strain. These are developed from single station null methods where the bridge is balanced for each reading, to deflection methods, where strain is determined from the galvanometer reading, and to multiple station arrangements. This section concludes with arrangements suitable for use when slip rings are required to permit the gauges to be used upon rotating shafts.

Attention is next given to the measurement of non-static strains and since this problem always necessitates the use of valve amplifiers, attention is drawn to possible sources of distortion and in particular of phase distortion, a matter which is not infrequently overlooked. It is shown how the use of a "carrier" system effectively eliminates this distortion, which would otherwise be very serious when recording either "slow transients" or very slow cyclic phenomena. The paper concludes with a reference to dynamic use ("quick transients" and rapid cyclic phenomena) and describes methods of calibrating the equipment.

Throughout the paper stress is laid on possible sources of error and the precautions (usually quite simple) whereby these errors can be eliminated, because this aspect is either omitted or only touched upon in many existing publications. It is hoped that this will help prospective users to obtain accurate results and so avoid disappointment rather than discourage them from using the gauges.

Introduction. Types of Gauge.

Although the fact that the electrical resistance of a conductor varies with mechanical strain has been known for many years, and the change with bulk strain, caused by hydraulic pressure, has been used as a pressure gauge, the electrical resistance wire strain gauge as now used, is a comparatively recent development. It consists of a length of wire which is firmly secured to an insulating layer, usually of paper. For use, the complete gauge is glued to the surface of the test object. The gauge responds to linear strain in the direction of the length of the wire. The force necessary to strain the wire itself must obviously be transmitted through the glue (and paper) and to the metal of the wire through its surface. It is therefore desirable that the circumference/area ratio should be as high as possible. This implies a wire of small diameter. The size used commonly lies between 0·0015 and 0·0005in. Thin strip would be preferable and has been tried, but it is not available commercially so that, at the present time, circular wire is universally used. The wire used is too fine for direct connection to the leads, and intermediate connections, integral with the complete gauge, are therefore provided.

In a few cases a single length of wire can be used, but usually the total length of wire necessary to meet the needs of accuracy and sensitivity is greater than the desired gauge length, and the wire is then formed into a multiple zig-zag. According to the method of construction, the zig-zag is formed in one of three ways. In the first the wire is cemented on the surface of a piece of paper in a series of parallel lengths joined by semi-circular arcs; in the second the wire is wound around a piece of paper so that alternate limbs are parallel, and at a slight angle to the intermediate limbs; in the third the wire is woven as the weft of a composite fabric. These different forms of construction each have particular advantages and limitations. For instance, the second, or wound form permits the construction of gauges of very short length (down to 0·06in.) and is well adapted to the use of a thermo-setting plastic as the bonding material. On the other hand, it requires more paper than the first, or flat grid, form; this increases the mechanical stiffness of the gauge so that greater strength is required in the glue. At the same time the resin impregnation renders the paper much less permeable to the glue solvent so that, at air temperatures, a much longer drying time is required. The flat grid form, being usually manufactured with a cement similar to the glue used for fixing the gauge usually requires a drying time of one day or less. The woven type, being even more permeable than the flat grid type gives a much shorter drying time, but does not seem to be quite so uniform in characteristics.

In the great majority of cases, a gauge, once glued down, cannot be removed and re-used, so that individual calibration of gauges is not practicable. Even if it were, the particular glue film used to fix the gauge to the test object is effectively part of the gauge. The performance of the gauge is thus determined essentially, though not necessarily entirely, by the glue film. The relation is, however, so abruptly quantitative as to be virtually qualitative: that is, a glue is either satisfactory, or not, and an intermediate stage where the effective sensitivity is appreciably reduced is so unpredictable that it is not satisfactory.

Several different glues have been suggested and tried out fairly extensively, but there seems to be almost universal agreement that, within its limitations, what may be described as a good-quality celluloid cement is the most suitable. Its main limitation is that it softens at relatively low temperatures, and is not reliable above say 50°-60° C. For higher temperatures, fish glue is satisfactory provided the humidity is low, but it may become very brittle and fail completely after a period of some weeks. Some glues made from thermo-setting resins are also suitable for higher temperatures (200° C.) but, on steel surfaces at least, sometimes fail to adhere. Gluing to aluminium and its alloys is, in general, very much easier.

The glues just considered are solvent glues and set by the evaporation of a volatile solvent. The time of setting is determined by the rate of loss of solvent which depends upon temperature and the concentration of solvent vapour in the surrounding atmosphere. Drying is thus accelerated by raising the temperature and, to a lesser degree, by the use of fans. If heat is used the temperature should not be raised too rapidly, otherwise bubbles may form owing to boiling of the solvent. Numerous experiments have shown that it is necessary *i.e.* if the auxiliary slide wire is set such that its presence has no effect upon the initial balance. In practice, therefore, imperfect temperature compensation has to be accepted as the price of working from a fixed zero on the main slider. The error introduced depends partly upon the value of α and of range of t, and partly upon the departure of the ratio of M_o/C_o from the value for balance at zero of the main slide wire (generally unity). If M_o/C_o varies by more than one or two per cent. from this value errors in strain of a few parts per hundred thousand may arise from temperature changes of the order of 10° C.

For cases where the errors arising from this source become serious the arrangement of Fig. 5 is suggested. This is derived from Fig. 1b in which each M-C pair has its own P-Q pair of arms, all the P's being fixed, and of the same value, whilst the Q's consist of

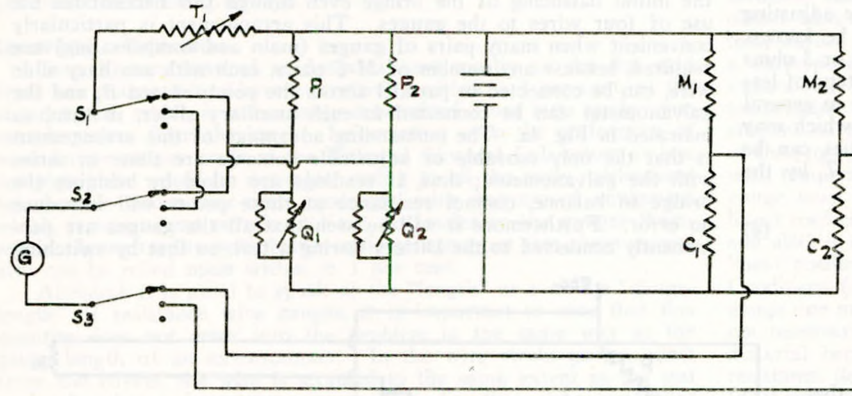

FIG. 5.

fixed resistances shunted by variable resistances to secure initial balance. The same conductance box T_1 is shunted across P_1, P_2, etc. in turn to determine the strain at each station, the galvanometer being simultaneously switched from bridge to bridge by the other two poles S_2 and S_3 of a three-pole multi-way switch. It will be noted that the contacts of S_1 and S_2 are in parallel, and it is probable that G could be directly connected to T_1, thus reducing the three-pole to a two-pole switch without error.

Static Measurements. Deflection Methods.

The preceding treatment has assumed that the strain is determined by re-balancing the bridge, but any of the arrangements so far described can be used as a deflection method, instead of a null method, simply by reading the galvanometer deflection instead of re-balancing the bridge. It can be shown fairly easily that the galvanometer current, in an unbalanced bridge, is determined by the proportionate change of resistance in one arm. Although the relation is not exactly linear, over the range commonly encountered in strain gauge work, the error thus introduced is negligible. A possible source of error in a deflection method, often overlooked, is the change of galvanometer resistance with temperature. Maximum sensitivity is obtained when the resistance of the galvanometer is equal to that between the terminals of the bridge to which it is connected. Thus, if the galvanometer coil has a temperature coefficient of 0·4 per cent. per degree C., the effective coefficient for the galvanometer and circuit is 0·2 per cent. A change of temperature of 5° C. will therefore produce a change in sensitivity of 1 per cent.*

When a null method is adapted for deflection working, there is usually no difficulty in calibrating the galvanometer in terms of strain. It is only necessary to change the ratio arms from the "zero" or initial balance point by the amount corresponding to the required strain (usually 1/1000) and observe the deflection. It is frequently convenient to include in the galvanometer circuit an adjustable series resistance (or a shunt) so that the scale reading can be converted to strain by a convenient factor. When a common galvanometer is switched from circuit to circuit, to permit the reading of strain at a number of stations, one point needs attention; the scale factor may not be exactly the same for all bridge circuits. This effect is unlikely to be serious except for the circuit of Fig. 4b. If necessary, extra resistances can be inserted, for each station, between the slider of each initial balancing resistance and its corresponding contact stud of the selector switch. These must, of course, be adjusted individually, *i.e.* by switching the galvanometer from gauge to gauge and setting the appropriate resistance so that the desired deflection is given when the bridge balance is altered by the required amount.

An experienced observer will obtain a null balance on a bridge almost as quickly as the galvanometer deflection can be read, so that the deflection method saves very little time in return for the (generally) reduced accuracy. The method is, however, particularly convenient if it is desired to take more or less automatic records by photographic means, since it is sufficient to take a photograph of the galvanometer with an ordinary camera. It becomes of particular value when it is desired to take simultaneous readings of strain at numerous positions, because each position can have an associated bridge and galvanometer, and the bank of galvanometers can be photographed upon a common film. This arrangement represents a practicable solution of certain cases where the loading rate, whilst more rapid than for "static" tests, is still sufficiently slow to permit ordinary galvanometers to follow without undue error.

Although a common pair of ratio arms can be used for the separate strain gauge pairs, this course may lead to errors from mutual interference unless the ratio arms are of inconveniently low resistance. It is therefore preferable to have a separate bridge circuit for each strain gauge pair. When this is done it is practicable and preferable to remove the arrangement for initial balance of the bridge from between the main and compensating gauges to the ratio arms, so that the circuit reverts to that of Fig. 2. The question of calibration remains, however, because it is uneconomical to use an accurately calibrated slidewire, or the like, merely for the purpose of securing initial balance.

The most satisfactory solution is the circuit of Fig. 6 where the slide wire is shunted by the series combination of two resistances R_1 and R_2 and the calibrating resistance R_3 is connected by a switch S between the junction of R_1 and R_2, and the top of P. If $P = Q$ (as is supposed) and $R_1 = R_2$, then the four resistances P, Q, R_1 and R_2 form a balanced bridge with the slide wire as the conjugate of the other (external) resistance R_3. It follows, therefore, that no component of the current through R_3 flows through the slide wire, *i.e.* the potential of all points on the slide wire is changed equally by closing the switch S, and it can be shown that this approximates most

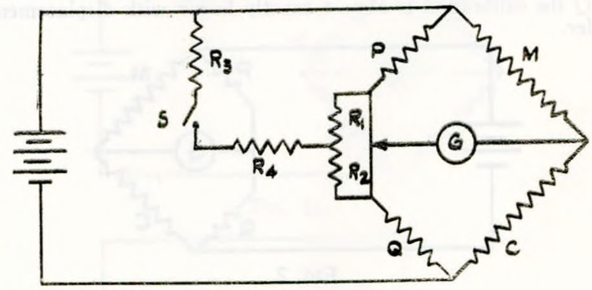

FIG. 6.

closely to what is required. It is obvious that a number of separate but similar bridges, as Fig. 6, can be used and supplied from a common battery. Then if the values of P, etc. are the same for all bridges a common resistance R_3, with a multi-way switch, can be used for successive calibration of the bridges. This is particularly convenient because the required value of R_3 is usually fairly high. When the bridges are slightly dissimilar a common resistance R_3 can be used for the bulk of the calibrating resistance, together with auxiliary separate resistances as R_4. If the bridges are all the same then the several resistances R_4 (one for each bridge) can, of course, be omitted.

Use of Slip Rings.

The preceding treatment has assumed throughout that connections to the gauges are made by continuous conductors, and it may be mentioned here that it is preferable, where possible, to use a single rather than a stranded conductor, because the latter may show small variations of resistance unless good contact is made to every strand at both ends. Emphasis has also been laid upon the desirability of eliminating switch contacts from positions directly in series with the

* The choice of galvanometer resistance so given for maximum sensitivity will usually cause the galvanometer to be overdamped so that observation is much protracted. The use of a galvanometer of lower resistance, to reduce the damping, will of course reduce the error due to change of resistance with temperature.

main or compensating gauges. In certain cases, however, it is necessary to make measurements upon a rotating member, and this involves the use of slip rings. To reduce errors from changes of contact resistance at the slip rings, two courses are possible. One is to keep the variations below the permissible error. This requires the selection of the best materials for brush and slip ring and the use of a gauge of high resistance; in many cases a number of gauges can be connected in series. The author has no direct experience of the use of slip rings, but understands that a satisfactory combination is silver-graphite brushes on stainless-steel slip rings. Practice in the U.S.A. also apparently favours a similar brush but working upon a silver slip ring.

The second, and preferable course when possible, is to remove the brush-slip ring resistance from the bridge circuit and place it in series with the battery or galvanometer circuit where any variation causes a scale, or sensitivity error, and not an error of zero. The simplest way of doing this is to mount the whole bridge upon the moving part and bring out the four corners via slip rings. This places one pair of rings in series with the supply leads (and battery) and the second in series with the galvanometer. If the galvanometer circuit is of, say, 1,000 ohms resistance, then a variation of brush-slip ring resistance of as much as 1 ohm will alter the sensitivity by only 0·1 per cent. The expression "galvanometer circuit resistance" is not that of the galvanometer alone, but includes also the resistance of the bridge circuit itself, measured between the two corners to which the galvanometer is connected. This can be shown from a detailed analysis of the circuit.

In a similar way the second pair of rings are in the battery circuit, and since this obviously includes the bridge itself, quite appreciable variations in contact resistance give rise to only small errors of sensitivity.

An additional advantage of placing the whole bridge upon the moving part is that it can then frequently be arranged that all the bridge members are active gauges, or at least that half are active and the remainder are compensating gauges. This gives an increase of sensitivity over the case where two members are fixed ratio arms*. On the other hand, the placing of the whole bridge on the moving part generally makes it impossible to provide a simple adjustment for initial balance, and this balance must usually be obtained by alteration to a short length of resistance wire included in one arm.

When, for some reason, it is not possible to place a complete bridge circuit upon the moving part, or when the number of slip rings must be restricted, then in certain cases an alternative is to provide each slip ring with two separate brushes. One brush on each ring is then used to supply an almost constant current to the gauge, from an external circuit consisting of a battery and a resistance many times that of the gauge; the second pair is then used to measure the potential difference across the gauge, due to this current. It should be noted that, with this arrangement, the full potential across the gauge appears across the "potential" brushes, so that changes, arising from strain, cannot be directly observed as in the case of a bridge.

In the preceding discussion, for simplicity "brush" has been used to indicate *one* connection from a fixed external circuit to a slip ring. In practice however, it is always desirable to use several brushes in parallel to constitute the "brush". When ordinary slip rings are used it is best to provide three separate brushes, equally spaced around the ring so that, if the ring is slightly eccentric, one brush is always being pushed out by the ring.

If there is appreciable vibration perpendicular to the axis of the shaft, but less or none along the axis, it is preferable to use "face" or "disc" rings. As before, multiple brushes should be used, and it is advantageous whenever possible, to provide a pair of brushes facing one another, on opposite sides of the disc. These should be connected together by a relatively stiff spring so that if the disc runs out of truth in its plane each brush tends to keep the other in contact.

Dynamic Measurements. Amplifier Distortion.

All the preceding treatment has dealt essentially with "static" measurements, *i.e.* steady strains which are either under control (as in a dead-weight loading test) or at least vary so slowly that there is no appreciable change in the few seconds required either to balance the bridge or to read the galvanometer. Resistance wire gauges are, however, even more useful when it is required to record varying strains, but the equipment required is different.

It can be shown fairly easily that with a deflection method, as already described, the maximum power delivered to the galvanometer is of the order of a micro-Watt. This requires a moderately sensitive galvanometer with a period of the order of a second or so. It is quite impossible to operate any existing recording unit of short period directly from the gauge or bridge circuit and it therefore becomes necessary to use an amplifier. Although d.c. amplifiers* have been designed, they are much more elaborate than ordinary resistance capacity coupled amplifiers; they require elaborately stabilized power supplies and also, when the input voltage is small, they require special means and adjustment to minimize zero drift. At the present stage of development they cannot be regarded as purchasable pieces of equipment which can be used by even semi-skilled operators: in general they require not so much skilled as sympathetic handling.

Ordinary resistance capacity amplifiers, on the other hand, have now reached a high degree of reliability and, by known and well-established methods of design can be given very stable characteristics. They are therefore well suited for use with strain gauges, but it is essential that their limitations be realized, otherwise serious errors may result.

Although the problem can be considered in other ways, the performance of an amplifier is usually expressed in terms of the input-output relation for a sinusoidal input, as a function of the frequency of the input. On this basis an amplifier can distort the input signal† in four ways:—

(*1*) The amplification may not be independent of the amplitude of the input signal (for a fixed frequency). This is *amplitude*, or *non-linear* distortion. Since the output, being not purely sinusoidal, can be analysed into a fundamental (of the same frequency as the input) and a number of harmonics, this type of distortion is sometimes called harmonic distortion.

(*2*) An amplifier free from amplitude distortion may (and in fact always will) be found to give an amplification which varies with the frequency of the applied sinusoidal input. This is *frequency* distortion.

(*3*) If the input and output are compared, it may be found that they are not exactly in phase‡, *i.e.* the two peak values do not occur simultaneously but are separated by a time interval. This is *phase* or *delay* distortion.

(*4*) The amplifier may add to the signal other, and false, components which may be either periodic, arising from the power supply, *i.e. hum*, or of a random nature, *i.e. noise*.

In practice, non-linear distortion can be made negligibly small by correct design. When it occurs it can always be reduced by reducing the amplitude of the signal through the amplifier, provided, of course, that it has not arisen from ageing of one of the valves or failure of one of the components.

Every amplifier causes both frequency and phase distortion, because no amplifier will handle the complete range of zero to infinite frequency. Although there is, in fact, a relation between the two types of distortion, so that an improvement of one accompanies, or is accompanied by, an improvement in the other§, it is generally convenient to consider them separately.

Most people are familiar with frequency distortion. It can be experienced by adjustment of the "tone control" of a wireless set or gramophone. Corresponding data or curves of performance of oscillograph amplifiers are commonly given in the instruction books. Unfortunately, these curves can be very misleading, because they omit entirely the accompanying phase distortion; this latter distortion (within very wide limits) passes quite undetected by the ear but for most engineering purposes the eye, not the ear, is the relevant organ and phase distortion of a complex (*i.e.* non-sinusoidal) waveform can alter its whole shape and its peak values.

An adequate discussion of phase distortion would be far too lengthy for inclusion here, and the problem of specifying the permissible degree of phase distortion (as a function of frequency) is, perhaps, one of the most difficult design problems. It is therefore only possible to make a few general comments. In practice, the effect of phase distortion is always to "erode" the wave shape, *i.e.* corners are rounded; peaks are rounded and reduced. It is also the primary cause of straight portions of a waveform being reproduced as slightly curved lines. As an illustration of the relative importance of phase and frequency distortion, one can compare the distortion of a sine wave and a square wave, of the same fundamental frequency, when passed through a normal amplifier. It has been shown that when the frequency distortion of the sine wave is only 0·015 per cent., some parts of the square wave have an error of 5½ per cent. This shows

* But not the double sensitivity, frequently claimed, except for the alternative special conditions (*a*) galvanometer of very high (infinite) resistance, (*b*) ratio arms limited to the same resistance as the gauges. This last restriction is usually artificial.

* *i.e.* amplifiers which respond accurately to very slow changes of input voltage.

† "Signal" is a convenient term for the output from the strain gauge, *i.e.* the input to the amplifier, or this same wave form as it is amplified through the amplifier.

‡ This is additional to the phase reversal which occurs in a single stage.

§ By special methods, phase distortion can be reduced without any accompanying change of frequency distortion.

very clearly that it is unwise to rely only on curves of frequency distortion when assessing the performance of recording equipment; it shows also that a more sensitive and practicable criterion is the fidelity with which the equipment reproduces a square wave. This method of testing is beginning to come into use and will probably replace frequency response tests except perhaps purely for acoustic work, where, as already suggested, phase distortion is frequently of negligible importance*.

"Hum" in the output of an amplifier usually arises from the H.T. supply and can be reduced by improved smoothing. There is an increasing tendency in laboratories to use special electronically-regulated power units which reduce both hum and "noise" coming from the supply mains. Except in very high-gain amplifiers, when inherent noise of the first stage may become comparable with the signal, noise is almost invariably a sign of a faulty component, *e.g.* valve resistance or condenser.

All four forms of distortion except noise and hum originating in the grid-cathode circuit of the first stage can be considerably reduced—by a factor of 100 or more—at the expense of loss of amplification by the use of negative feed-back. Unfortunately, no commercial units are available which incorporate negative feed-back and also give sufficient amplification for strain gauge work.

In practice the most serious form of distortion is almost always phase distortion, owing to phase advance of the lower frequency components of the signal caused by the coupling condensers which form an essential part of the resistance-capacity amplifier. Unfortunately, these condensers cannot be made indefinitely large, partly for reasons of bulk and partly because trouble would then be experienced from the same causes as arise in the case of d.c. amplifiers. This distortion is most serious for transient phenomena (as contrasted with cyclic phenomena).

"Slow Transients" Carrier Methods.

The most satisfactory way of eliminating distortion of low frequency, or slow transient phenomena, is to supply the gauge with alternating current of a suitably high frequency—1,000 cycles/sec. is commonly satisfactory. The out-of-balance signal from the bridge then has the form of a 1,000 cycles/sec. a.c. modulated by the phenomenon it is desired to record, *i.e.* the wave-form of the phenomenon is given by the envelope of the output. The simplest arrangement is to record the modulated output wave directly (after amplification), but in some cases it is preferable to add a rectifier or "detector" exactly as in a wireless set so that the modulating wave form is obtained directly and the modulated wave or "carrier" (*i.e.* the 1,000 cycles/sec. current) is eliminated from the record.

It is interesting to note how the use of an a.c. "carrier", as just described, eliminates or renders negligible the errors due to frequency and phase distortion. If the phenomenon or signal is a simple sinusoidal variation of frequency f_s and the a.c. or carrier supply to the bridge has a frequency of f_c then the output from the bridge can be expressed as:—

$$(a+b \operatorname{Cos} 2\pi f_s) \operatorname{Cos} 2\pi f_c$$

where a is a constant which represents the output when the gauge is unstrained, *i.e.* it depends upon the initial balance of the bridge, and b corresponds to the strain of the gauge.

$$\begin{aligned} \text{Now} \quad & (a+b \operatorname{Cos} 2\pi f_s) \operatorname{Cos} 2\pi f_c \\ & = a \operatorname{Cos} 2\pi f_c + b \operatorname{Cos} 2\pi f_c \operatorname{Cos} 2\pi f_s \\ & = a \operatorname{Cos} 2\pi f_c + \frac{b}{2} \operatorname{Cos} 2\pi(f_c+f_s) + \frac{b}{2} \operatorname{Cos} 2\pi(f_c-f_s) \end{aligned}$$

The original signal frequency f_s has thus been replaced by two new frequencies (f_c+f_s) and (f_c-f_s). By suitable choice of f_c, both of these can be made to have such a value that a normal amplifier will handle them with negligible distortion.

If the signal is not sinusoidal, it can be analysed into its component frequencies and it will be seen that if f_s is the highest frequency present, the amplifier must handle a frequency band $2f_s$ wide with its centre at f_c. It should be particularly noted that as f_s becomes lower, the width of the band becomes smaller. There is therefore no difficulty in the case of signals of very low frequency.

Attention should be drawn to one point. If the bridge is exactly balanced initially (so that $a=0$) the output is $b \operatorname{Cos} 2\pi f_c \operatorname{Cos} 2\pi f_s$, and the sign of $\operatorname{Cos} 2\pi f_s$, *i.e.* whether the strain is initially tensile or compressive cannot be determined because $\operatorname{Cos} 2\pi f_c$ is varying cyclically. Or, in other words, one cannot tell whether the upper or lower envelope is the correct one. To avoid this ambiguity it is usual to arrange so that the bridge is not exactly balanced initially, so that a has a value preferably slightly greater than b. It is then easy to determine the sign of the strain from a knowledge of the direction of the initial unbalance.

Although the use of alternating current, as shown, solves most of the problems of the amplifier, it introduces another complication and risk of error. This arises because a bridge circuit, when balanced for d.c. is not necessarily balanced for a.c. because of the presence of capacities, and sometimes inductances also, which, of course, are without effect upon the d.c. balance. The general problem of a.c. bridges is much too extensive for treatment here, but a few brief remarks may be of value. A.c. bridges may be divided into two general classes, (*a*) those in which the balance condition is ***dependent***

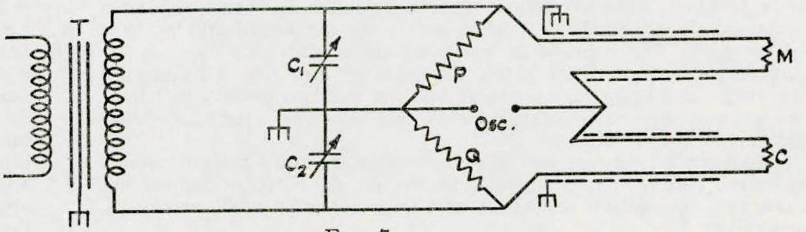

FIG. 7.

on the supply frequency, and (*b*) those in which the balance condition is ***independent*** of the supply frequency. Now it has already been shown that, in addition to the supply, or carrier, frequency f_c, there are frequencies (f_c+f_s) and (f_c-f_s) present; hence it is important, for use with strain gauges, to use a bridge circuit whose balance is independent of frequency, otherwise the reproduced signal may suffer distortion. In practice this is best achieved by using a symmetrical arrangement as shown in Fig. 7. The two ratio arms P and Q (one of which must be variable to obtain initial "balance") are each shunted by variable condensers C_1 and C_2. One of these can be omitted if the remaining one is connected across the appropriate arm. Alternatively, the two condensers can be combined as a single differential condenser. The capacity required will in practice depend mainly upon the capacity unbalance of the output winding of the transformer T, and will usually be less than $0{\cdot}003\mu F$ (across one arm). An ordinary radio three-gang condenser with the three units in parallel will supply a "swing" or variation of about 0·00125, and one or two fixed condensers can be connected in parallel if a larger capacity is required. The leads to M and C should be of approximately equal length and should be shielded in an earthed screen of wire braid as shown. Ordinary lead-covered cable also provides excellent screening. To calibrate the complete layout one of the ratio arms may be changed by an amount corresponding to a known strain by any of the methods already described. It is important to note that the arms P and Q must be wound non-inductively.

Cameras and Oscillographs.

For the recording of phenomena which last for a period of some seconds, it is almost always essential to use a moving film camera. This can either be somewhat similar to a cinema camera, except that the film moves ***uniformly*** instead of intermittently, or for shorter records a drum camera can be used. The width of the film must be chosen with regard to the accuracy required and, for multiple recording units, according to the number of separate records to be included. Except for the very simplest form of record, it is not practicable to allow neighbouring records to overlap. The type of oscillograph used is best chosen also with reference to the number of records required on a single film. When only one record is required, a cathode-ray oscillograph is probably of greatest utility, as synchronizing marks can be added to the record in a number of ways, *e.g.* by momentarily extinguishing the spot or by abruptly displacing it sideways for a brief interval. A time scale is automatically provided by the a.c. supply itself, provided that this is of known frequency and that the film speed is fast enough to resolve the separate cycles. Alternatively, if synchronizing marks are not required, of if there is no risk of confusion, the spot can be regularly extinguished or displaced sideways to provide a time scale.

With a cathode-ray oscillograph, one other point needs attention. Oscillograph tubes can usually be obtained with two or sometimes

* One peculiarity of phase distortion deserves comment. The difference between the input and the output (for a single sinusoidal frequency) can be expressed either as a fraction of the complete cycle (in degrees say) or directly as a time interval. Now, if for a complex wave the time interval, or delay, is the same for all components, this results in the complete wave being reproduced unchanged in shape, but delayed by this time interval. Thus if only the wave shape (and magnitude) is of interest, irrespective of its time of occurrence, this delay of the whole wave form is unimportant, and the recorded wave form is free from distortion.

three types of screen characterized essentially by the length of "afterglow", *viz.* a long afterglow, lasting for some seconds, a type with virtually no afterglow, and an intermediate type. (The second type, no afterglow, usually has a blue fluorescence; the others are commonly green but reference should be made to the makers). For photographic work on moving film, the tube should have no afterglow, otherwise the record will appear "smudged".

Except with a double-beam tube, which permits the recording of two phenomena quite easily, the provision of multiple records upon a common film involves a fairly complex mechanical and optical layout. For many engineering purposes, therefore, where the supremely rapid response of the cathode ray oscillograph is not essential, "mechanical" oscillographs are useful since they greatly facilitate the recording of numerous separate phenomena. These instruments are commonly moving coil galvanometers designed and constructed to have a very high natural frequency, but there has recently been developed a piezo-electric oscillograph which, it is understood, has a somewhat higher natural frequency than the majority of moving coil instruments. Nearly all these oscillographs employ photographic recording, but there are a few moving-coil instruments which give a pen record. These are obviously much more convenient to use for many purposes, but are only suitable for lower-speed operation.

Dynamic Measurements.

The remaining field of application of strain gauges is to short transient phenomena, such as impact testing and to cyclic phenomena at and above medium speeds. In this work, usually termed dynamic measurement, an alternating-current supply to the bridge may not be permissible, because if the frequency is raised high enough to give the necessary detail trouble due to stray capacitance of the leads, etc. becomes excessive. Fortunately these phenomena, because of their relatively high speed, include no, or negligible, very low-frequency components which suffer serious phase distortion in resistance-capacity amplifiers, so that the gauge can be supplied with d.c. and the resulting "signal" can be amplified directly.

Any of the d.c. bridge circuits can be used, with the galvanometer replaced by an amplifier and oscillograph, but in practice a simpler arrangement has additional advantages. One function of the bridge in *e.g.* Fig. 4, is to arrange that the initial steady voltage across the gauge does not deflect the galvanometer. In dynamic work, only the rapid changes of resistance, and the resulting small voltage changes, are of interest. These can very easily be "filtered" from the steady voltage across the gauge by means of a series condenser of suitable size. The circuit therefore becomes that of Fig. 8. The gauge M is

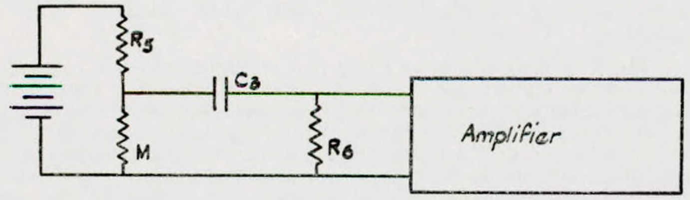

FIG. 8.

connected in series with a resistance R_5 across a battery. The amplifier, with an input grid resistance R_6 is connected to the gauge through a condenser C_3; the time constant of this latter combination must, of course, be long compared with the duration of the phenomenon to avoid phase distortion.

If E is the battery voltage and e the steady voltage across the gauge

$$\frac{e}{E}=\frac{M}{R_5+M}$$

If e becomes $e+\Delta e$ when M becomes $M+\Delta M$

$$\frac{\Delta e}{E}=\frac{e+\Delta e}{E}-\frac{e}{E}=\frac{M+\Delta M}{R_5+M+\Delta M}-\frac{M}{R_5+M}$$

$$=\frac{M.R_5+M^2+R_5\Delta M+M.\Delta M-M.R_5-M^2-M\Delta M}{(R_5+M+\Delta M)(R_5+M)}$$

$$=\frac{R_5\Delta M}{(R_5+M)(R_5+M+\Delta M)}=\frac{R_5\Delta M}{(R_5+M)^2}\text{ approx}$$

$$\text{Now }\frac{\Delta e}{e}=\frac{\Delta e}{E}\cdot\frac{E}{e}=\frac{(R_5\Delta M)}{(R_5+M)^2}\cdot\frac{(R_5+M)}{M}$$

$$=\frac{\Delta M}{M}\cdot\frac{R_5}{R_5+M}$$

$\Delta M/M$ is of course (strain)$\times$(strain sensitivity) the other factor $R_5/(R_5+M)$ approaches unity as R_5 becomes very large compared with M. It is very convenient to use a value of R_5 which makes this factor 0·995 say, so that it can be taken as unity with negligible error, but this requires that $R_5=200\ M$, and therefore that $E=201\ e$. Now it is convenient to work with e of the order of 20 volts at least (so that Δe for 0·001 strain is about $0{\cdot}001\times2{\cdot}2\times20=0{\cdot}044$ volt) and this would require E to be of the order of 4,000 volts. This is inconveniently and even dangerously high. Fortunately, it is possible to replace R_5 by a pentode valve so connected that while its effective resistance to changes of current is of the order of tens of megohms, the steady voltage drop across it, for a reasonable working current, is only one or two hundred volts. Where such an arrangement cannot be used, R_5 must be restricted to a value of the order 10 M or less and the appropriate value of $R_5/(R_5+M)$ must be used.

It will be noted that Fig. 8 differs from all previous circuits in that no compensating gauge is shown. If desired a compensating gauge can be used in place of R_5, at the expense of loss of sensitivity, but for the class of work now under consideration a compensating gauge is very rarely necessary, because temperature changes during the working interval are too small to cause appreciable error. Even if a compensating gauge be used, it will only be effective in favourable circumstances because of the difficulty of ensuring that both gauges are at the same temperature.

Calibration of Dynamic Equipment.

Calibration of equipment for dynamic tests can be done in several ways. The most convincing is to include in series with M a small fixed resistance which can be put in or out of circuit, thus giving a step record on the oscillograph corresponding to a known change of resistance. Alternatively, and preferably, a high resistance can be shunted across M. Both these methods are direct, *i.e.* they relate a known resistance change with oscillograph deflection. They therefore make it unnecessary to know E (or e), $R_5/(R_5+M)$ and the oscillograph sensitivity (including amplifier) separately. However it is rarely possible to observe the height of the step visually with the desired accuracy, so that a photographic record must be taken, and in many cases this requires some synchronization of exposure with the switching operation. Another method is to measure E, or the current through the gauge, *i.e.* $E/(R_5+M)$ and to determine the oscillograph and amplifier sensitivity separately by applying a small known a.c. voltage to the input and observing the record so obtained. This method is fairly popular, but is open to one objection which is often overlooked. The only measurement which can easily be made upon the cathode-ray tube itself is that of peak-to-peak deflection, whereas the input voltage is usually measured with a pointer instrument which generally reads either the average or the R.M.S. value. If the waveform is *accurately* sinusoidal these quantities are related in a known way, but if there are harmonics present an uncertainty of a few per cent. may arise.

The method which the writer prefers is first to determine the deflectional sensitivity of the oscillograph tube by applying known d.c. deflecting potentials to the plates, and then to determine the "gain" of the amplifier by balancing it against the "loss" or potential dividing ratio of an accurate resistance network. The cathode-ray tube is used for this comparison in the following way. Each Y plate is joined to an X plate, so that when an auxiliary a.c. supply is connected to the X plates a trace is obtained on the screen which is inclined to both X and Y axes. This trace is called the "nominal 45° line". It is only at a geometrical 45° to both X and Y axes if the two sets of plates have equal deflexional sensitivities. A line is inked on the screen at the same slope as this trace, and provides a reference for equality of voltages on X and Y plates. The Y plates are next disconnected from the X plates and reconnected to the amplifier output. The amplifier input is connected to a potentiometer, of accurately known ratio, and the amplifier gain control is adjusted until the trace lies on the marked "nominal 45°" line. It is easily seen that when this is done the voltage amplification of the amplifier is equal to the dividing ratio of the potentiometer.

This method is easily sensitive to $\pm1\%$, and the sensitivity can be doubled if arrangements are made to replace the marked nominal 45° line by an unsilvered glass plate perpendicular both to this line and to the screen. Adjustment is now made until half of the trace appears to coincide with the reflection of the other half. It should be noted that comparison is made, in effect, over the whole cycle, instead of only at the peaks, thus the waveform is unimportant (within limits mentioned later) and it is frequently convenient to use the 50 cycles/sec. supply mains.

An added advantage of the method is that, if a variable frequency oscillator is available, it can be used to determine the distortion of the amplifier. Non-linear distortion is at once shown up by the trace departing from a straight line. (Trapezium distortion* in the C.R. tube also gives a curved trace, but this appears also in the initial stage when the amplifier is not used; thus trapezium and non-linear distortion can be distinguished). It is obvious that frequency distortion (in the absence of phase distortion) would be shown by different values of gain at different frequencies. Finally, phase distortion is shown by the line opening out into an ellipse.

Conclusion.

The writer has attempted a general survey of the application of strain gauges to the three general classes of problems, static work, dynamic work and the intermediate class of "slow transients". If emphasis has been laid on possible troubles and errors, it is because these are not infrequently omitted from earlier publications, and because they can so easily creep in unsuspected unless the worker is aware of possible errors. A further point is that there is rarely any indication of incorrect operation of electrical recording equipment.

Acknowledgment.

The work described above has been carried out as part of the research programme of the National Physical Laboratory, and this paper is published by permission of the Director of the Laboratory.

Discussion.

Mr. E. J. Heeley (Visitor), opening the discussion, said that one of the difficulties in connection with that work was to make sure that the strain gauge behaved in the same way as the piece of metal to which it was attached. They had experienced difficulties, about which the author had warned them, in getting the gauge to adhere correctly to the material, and it had been found that a lot of that trouble was due to having to use a "glue" that could not be heated, because it was impracticable to heat the object to which the gauge was attached. By carefully limiting the amount of solvent, however, they had managed to make those gauges follow the behaviour of the metal, and they had had some excellent calibrations.

Mr. S. Archer (Visitor) said that, as the author pointed out, the snares which beset the path of the unwary user of strain gauges could be many. They were, therefore, all the more grateful to him for indicating so clearly the various precautions which were necessary if correct results were to be obtained from that most valuable tool, so comparatively recently presented to the engineer.

At Lloyd's Register of Shipping they had found the electrical resistance strain gauge a most versatile and reliable servant, both for static and dynamic work. Dr. Dorey, in his paper before the Institution of Naval Architects in 1944, described some static work carried out by the Society on boiler pressure parts including a water-tube boiler steam drum. Static work since carried out by the Research Department at Lloyd's Register included: (*a*) The measurement of internal strains in the hydraulic cylinder of a 3,000-ton extrusion press. That particular job presented some rather formidable difficulties, since the gauges had had to be subjected not only to oil, but also to a pressure of up to 6,000lb. per sq. in. It was suspected that the high fluid pressure normal to the gauge surface might have affected the strain sensitivity, but careful calibration showed no such effect. Insulation difficulties were overcome by coating the gauges and "Pyrotenax" leads with insulating varnish and using a high-grade transformer oil for the pressure medium. Standard N.P.L. gauges were used, applied with American "Duco" glue and accelerated drying. (*b*) A more recent application had been the gauging of the shell and bands of a banded pressure vessel at pressures of up to 1,000lb. per sq. in. In that case the gauges were standard N.P.L. applied with "Durofix" and accelerated drying. (*c*) Another interesting test had been the measurement of the progressive increases in load coming upon launching triggers during the launching of an oil tanker. The trigger was of the multi-lever type and the strain gauges were applied to measure the bending strains in the primary lever. Standard N.P.L. type gauges were used, applied with a hot-setting wax in that case. (*d*) A novel application now under way was the checking of screw shaft alignment by the measurement of reversed bending strains in the line shafting. The alignment would first be checked statically by means of the turning gear during one revolution and later at sea dynamically using slip rings on the shaft.

Among the dynamic uses of the strain gauge of which the Society had had experience were included (*i*) the checking of the torque balance between the two intermediate quill shafts of a locked train double reduction gear box; (*ii*) the measurement of reversed torsional fatigue strains in a 10-in. dia. mild steel shaft up to 16,000lb. per sq. in. (strain of about $\pm$ 0·0005) at a frequency of 2,400 vibrations per minute, and (*iii*) the recording of torsional impact strains in the driving shaft of a rolling mill as the billet entered and left the rolls. In that case the output from the strain gauges on the shaft was passed to the amplifiers and oscillographs of the Society's Sperry M.I.T. equipment.

Generally the Society's experience had shown the following procedure to be reliable:—For static work with standard N.P.L. type mechanically strong gauges a hot-setting wax should be used where possible. That would give a strain sensitivity of about 2·2. In cases where heating was inadmissible, a good cellulose glue such as "Durofix" or "Duco" in conjunction with accelerated drying should be used. It had been found that in the case of "Durofix" a reduced strain sensitivity of about 1·95 had to be accepted with standard N.P.L. gauges. Seccotine had been tried out, but had given disappointingly erratic results. The null method should be used on the Wheatstone bridge with the cathode ray oscillograph as balance indicator.

Strain gauges should be applied to a calibrating bar at the same time and using the same technique as for the test gauges to be used on the job. The insulation resistance of gauges and equipment to earth should be tested before and after the tests.

For dynamic work the technique for adhesives was as for static work, but care should be taken that possible temperature rises due to running heat of shafts in the vicinity of bearings, pinions, etc., were safely below the softening temperature of the adhesive.

The practice mentioned by the author of placing the entire Wheatstone bridge on the shaft, thus very largely reducing the effect of variations in slip ring-brush resistance, was very desirable, particularly where the strains being measured were not large.

The use of the carrier wave method of recording low frequency strains to avoid phase distortion in amplifiers represented a very considerable advance on the electronic side, and should prove of value in the study of hull and engine vibration problems.

It seemed that the scope of the electrical resistance strain gauge was practically unlimited; in fact, all that now remained was for the medical profession to adopt that method for assessing the degree of strain-hardening in the human being under modern austerity conditions!

Mr. R. J. Keig (Associate) said that resistance wire strain gauges had been in regular use at the Admiralty Engineering Laboratory for a number of years and several possible sources of error had become apparent. Most of the applications of those gauges at A.E.L. had been concerned with the measurement of static strains with measuring circuits of the type shown in Fig. 4A and using celluloid cement as an adhesive. It soon became apparent that at least two serious sources of error could occur. These could be referred to as "drift" and "creep".

The term "drift" was used to describe the relatively slow but erratic drifting of the galvanometer reading, which could occur before any load was applied to the specimen. That trouble had been found to be almost invariably due to the presence of moisture in the paper of the gauge and in the cement. The leakage path due to that moisture provided a high resistance shunt across the gauge, thus lowering its effective resistance. The presence of moisture in the gauge could readily be detected by using a megger to determine the insulation resistance of the gauge to earth. It had been found that an insulation resistance of at least 5 megohms was essential if the accuracy of the results was to be maintained, and 50 megohms or over, which could generally be attained under reasonably favourable conditions, was desirable.

On large structures situated on board ship or in an open workshop, it was often difficult to prevent the absorption of moisture from the atmosphere by the gauges unless special care was taken in drying out the gauges thoroughly and in applying a protective coating to the gauges and leads. Various protective coatings had been tried and a solution of polythene in xylene, now commercially available, had been found most satisfactory.

The term "creep" was used to describe the trouble which could occur when the solvent in the cement had not evaporated completely. That left the cement in a slightly plastic condition and caused a slow creep of the strain reading towards zero if the load on the specimen

* Trapezium distortion occurs when the potentials of a pair of deflecting plates are not symmetrical with respect to earth, unless the tube is designed to avoid this error. It consists of the dependence of the Y deflection sensitivity upon the potential of one X plate. It is so called because, when it occurs, the envelope of a high order Lissajous figure is trapezoidal instead of rectangular.

was maintained for a short time. As it was often necessary to maintain the load on the specimen for a considerable period, particularly when readings were being taken at a large number of gauge stations, that phenomenon could be very troublesome. The remedy would appear to be primarily in the use of a cement made up from constituents of the highest possible purity. A longer drying period at a higher temperature was often helpful, although not necessarily a complete cure and was not often practicable on large structures.

The trouble would appear to arise when the solvent in the cement contained a small percentage of impurities of high boiling point. It was felt that it would be of great assistance to workers in that field if a cement were readily available which was made up to a rigid specification and which could be relied upon to give consistently uniform results.

Small changes of temperature between the active and compensating gauges had also been troublesome. That might be due to draughts or to radiation from adjacent objects or direct sunlight. The thermal insulation of the active and compensating gauges by means of felt or cotton-wool pads had been found to be an effective cure for that trouble.

In conclusion, he would give an example of the accuracy of measurement attainable with resistance wire strain gauges. Those gauges had been in use for some time at A.E.L. for measuring small torques of the order of 20lb.-ft in a shaft running at about 1,500 r.p.m. Four gauges were used on the shaft to form a complete bridge circuit, the four corners of the bridge having been connected to four silver slip rings. Three silver-loaded carbons brushes were fitted to each slip ring. The bridge was balanced to zero by means of an external high-resistance shunt, and the torque reading was made by measuring the deflection of a small cantilever fitted with four gauges forming an identical bridge. The two bridges were backed off against one another and a galvanometer was used to indicate balance. With that equipment it had been found possible, under reasonable conditions, to measure mean torque consistently to an accuracy of 1 per cent.

Mr. A. G. Mapp (Visitor) said that the aircraft industry had used strain gauges since 1938, using at first gauges of 20,000 ohms resistance made of moulded carbon. The ordinary radio moulded carbon resistance was more strain sensitive than the wire-wound strain gauge, but had a strain sensitivity which might vary from a figure of 8-20, and so each gauge had to be calibrated when stuck in place. The temperature co-efficient was also so high that only alternating stresses might be measured, the measurement of steady stress being initially impossible.

However, attention was directed to the wire-wound gauge as soon as information of the preliminary developments in the U.S.A. became known in this country, but owing to the initial start with gauges of 20,000 ohms resistance, the tendency had been to use gauges having rather higher resistance values than those used by the N.P.L.

For alternating stress work, gauges of 5,000 and 10,000 ohms resistance were commonly used with a gauge of 2,000 ohms resistance for steady stress measurements. In the application to alternating stress measurements on propeller blades, where connection between the gauge and the amplifier had to be taken through slip rings running under severe conditions of vibration, the high resistance gauge allowed more latitude on slip ring performance and gave a higher signal voltage to the amplifier. The best brush-slip ring combinations had been found to be silver graphite brushes on pure silver for small diameter rings, with copper graphite brushes on B.6 bronze rings for the larger diameters. Brush pressures of about 2½ times the maker's figure were always used, with rubbing velocities up to 10,000ft. per min.

A source of error not mentioned by the author was the tendency of the actual cement to allow a slight slipping of the gauge. If a gauge was loaded to high values of strain (say 0·005) and this strain left on for some long period, the galvanometer spot indicating the strain showed a slow creep in a direction given by a release of strain; in some cases leading to an error of, say, 2 to 3 per cent. The cellulose cements were the worst offenders in this respect, while the synthetic resin bonded gauge that was stuck on to the specimen with this cement and the whole polymerised *in situ* showed the least error.

Another problem was that of corrosion of the gauge in salt-laden or corrosive atmospheres. The best solution to this problem, so far, had been to cover the gauge with a layer of Everetts wax, or with "Pliofilm" held on with this wax.

As to methods of measurement, they were using somewhat similar methods to those described in the paper but, owing to the higher resistance values used, more advantage was taken of methods involving substitution of the measuring resistance in the bridge arms, together with the use of radio type "wafer" switches for the necessary switching operations of multiple gauge circuits. However, the use of galanometer deflection as the measure of strain was the almost universal method for most steady stress work. Alternating current carrier methods had been tried, but were not used to any large extent because of the added complexity of the bridge balancing methods and the necessity for using screened wire in any job involving a large number for gauges in a small space. The aircraft industry rather tended to combine the methods of alternating and steady stress measurement by the use of a mechanical interrupter switch or "chopper". The Wheatstone bridge circuit was used in the usual way with the output terminals of the bridge connected directly to the input of a convential a.c. amplifier. Mechanically-driven switch contacts were placed across the amplifier input so as periodically to short the amplifier, say at a rate of 200 cycles per second. A square wave was thus produced on the resulting record, such that the amplitude was proportional to the d.c. unbalance voltage appearing across the bridge output terminals. Thus, all that was required to change from a steady to an alternating stress measurement was to close a switch supplying power to the interrupter drive motor.

To give some idea of the outside limits actually met in practice, two figures could be quoted, one a recorded alternating stress having a frequency of 6,500 cycles per second and containing at least a third, if not higher, harmonic, and the second, a measurement involving a strain gauge stuck on the periphery of an impeller where the centrifugal field was approximately 75,000 x gravity.

Mr. C. J. Laing (Visitor) said that he was concerned in a small way with the use of electrical resistance wire gauges. He had found that in static tests on bars loaded centrally, if the gauge was stressed in tension and the resistance/load curve plotted, on removing the load, the resistance was less than when starting. An investigation showed that this was partly due to galvanometer hysteresis. However, this defect did not account for the entire error and the fault was attributed to the adhesive.

He asked if a particular type of proofing could be recommended which would prevent oil from entering the gauges during working. He was concerned with tests on Diesel engines and this was an important factor in obtaining an accurate reading.

Mr. J. L. Thompson (Visitor) said he spoke as an electronic engineer rather than as a user, but there were one or two points that he desired to raise.

The author, very wisely, had drawn attention to the enormous importance of phase distortion in amplifiers, and very misleading results could be obtained unless phase distortion was low.

Phase distortion in amplifiers could, however, be made extremely low. It was admittedly a difficult problem, but they had been working on some strain gauge equipment and had succeeded in designing an amplifier with a phase angle of 3½° at half a cycle, which meant that from five cycles upwards the phase distortion was negligible. Quite apart from the phase distortion due to the coupling networks in the amplifier, the output transformer introduced very considerable phase distortion at low frequencies, but by using a cathode follower output stage and feeding the galvanometer direct, this source of phase distortion could be completely eliminated.

The author had pointed out that some of the ancillary equipment for use with strain gauge recording could be almost as complex as the strain recorder proper. For example, they were faced with the problem of producing 50 c.p.s. markers superimposed on the strain record film in situations where the normal 50 c.p.s. controlled mains supply was not available. This had meant producing a valve maintained tuning fork and multi-vibrator system in order that the necessary degree of frequency stability could be maintained, whilst several stages of differentiating amplifiers were necessary in order to reduce the width of the marker line to less than 0·5mm.

They had found also that in certain cases it was extremely desirable to be able automatically to select all the gauges that were to be recorded and not to have to go round the equipment operating switches in order to bring in the various banks of gauges during the actual test run. They had accordingly developed an equipment that had facilities for accepting a total of 48 strain gauges, any six of which could, in turn, be switched into a six-channel amplifier and on to a six-channel recorder by means of an automatic telephone type dial. This telephone dial, together with the camera "start" control, trace number indicator, film footage indicator, etc., was housed in a small control box which could be used to control remotely the complete strain recording equipment. By this means it was possible. for example, for the pilot of a single-seater aircraft to carry out both the required manœuvres and the control of the strain recorder.

These points were mentioned because there was a tendency to think that really good strain gauge equipment had to come from the United States. In England that were manufacturers who were producing equipment that was every bit as good as the American and, in some cases, possessed certain advantages and facilities which the American equipment did not.

Mr. A. F. C. Timpson, M.B.E. (Member of Council), proposed a hearty vote of thanks to the author for his excellent treatment of a difficult subject. This was seconded by Mr. J. R. Henderson (Member) and carried with acclamation.

BY CORRESPONDENCE:

Mr. A. C. Rankin: The question of strain sensitivity was rather interesting. The author had remarked upon the deviation of the value found for most alloys, $S=2{\cdot}2$, from the value evaluated from Poisson's ratio, $S=1{\cdot}6$. Since there was no variation in the strain sensitivity of specific resistivity with the stresses involved, at least within 1 per cent., and this in both compression and tension, this factor did not account for the discrepancy.

In consequence, it was not unreasonable to assume an abnormal value of Poisson's ratio in fine wires, but, since the maximum possible value for this constant was 0·5, corresponding to a perfect fluid, the maximum value of S became only 2·0. It was highly unlikely that this maximum value should be reached, so the discrepancy which still existed was greater than 0·2 in 2·2, *i.e.* 9 per cent.

It appeared, therefore, that both of these possibilities must be discarded and the variation looked for elsewhere. It seemed more likely that an abnormal specific resistivity existed in small cross-sections, and this could readily be investigated. It would be of interest also to compare strain sensitivity values using d.c. and a.c. bridges.

With regard to the question of recording, particularly where a number of strain gauge stations was involved, it might be of interest to note that commercial development in this country had produced a mechanical multi-pen recorder, allowing up to eight stations per instrument. The frequency response in this case was 0-100 cycles per second flat, the pen having a natural frequency of some 70 cycles per second with a system just less than critical damping, thus allowing for pen deflection being independent of frequency at 100 c.p.s. Recording was carried out by a dry electrical recording paper, and gave a fine, clear, black trace. The paper drive could operate up to 15 cm./sec. driven by a mechanically-governed constant-speed motor, in which case auxiliary pens dotted the paper, controlled by the a.c. mains, tuning fork, or oscillator, depending upon the accuracy desired. Alternatively, a fully synchronous motor could be used, with a preprinted time trace on the paper, driven from mains or a tuning-fork control.

Finally, it should be mentioned that this recorder had been developed in conjunction with strain gauge work on both transient and cyclic phenomenon. Four channels were now employed, incorporating 2 kc/s oscillators with automatic gain control, resistance and capacity bridges (with a phase discriminating circuit and bridge balancing allowing for balancing of in-phase and quadrature components, and a miniature C.R.T. for visual verification), push pull amplifiers with cathode followers (distortion being kept very low), demodulators and recorder.

The main applications of this equipment were in aircraft vibration studies, but the use of these mechanical recorders had, in general, been fairly wide, wherever transient and cyclic phenomena were studied.

Mr. J. Edwards, B.Sc., Wh.Sc., A.M.I.Mech.E.: The paper was a valuable contribution to the literature available to the engineer interested in strain measurements, since it set down in concrete form the essential information required to apply the resistance wire gauge to static and dynamic studies and also enumerated some of the possible errors arising in their use.

It was felt, however, that more emphasis could have been given to two very common sources of error known as "drift" and "creep", and it is hoped the comments below on these and less noticeable effects would amplify the author's remarks, and enable users to obtain that accuracy so often claimed for resistance wire strain gauges.

Resistance wire gauges had been in use at the Admiralty Engineering Laboratory for some years, and from the experience gained in the static and dynamic testing of structures it was evident that drift, due to lack of stability of the gauges, was the source of error most commonly encountered. The prime cause of this lack of stability, which could be detected as a loss of bridge balance without corresponding strain on the structure was a decrease of insulation resistance between the gauge wires and earth and was generally due to the gradual absorption of moisture into the adhesive between the gauge and the structure. In several cases the effect of this drift could be minimized by controlling the load cycles to be of short duration and returning quickly to the zero load state to check the amount of drift occurring, but in many more cases the loading conditions could not be repeated and the gauges were required to remain stable over extended periods, in some cases months, usually under the difficult conditions of extremes in weather. In such cases drift must be prevented at all costs, and even in those cases of short loading cycles it was as well to avoid the necessity of applying at best a rather indeterminate correction factor. The order of the drift encountered could be shown by the fact that a change of insulation resistance from 6 megohms to 5 megohms on one gauge of the bridge would produce a drift equivalent to about 180lb. per sq. in. stress in steel. Moisture could have a more serious effect by causing electrolysis of the resistance wires when a current was passing through the gauge, with a subsequent change in the gauge sensitivity. The degree and rapidity of moisture absorption was dependent upon the adhesive used and the porosity and the thickness of the paper on which the gauge was formed, and again upon the degree of protection given to the gauges. In general, it could be said that it was unsatisfactory to work over a long period with gauges of less than 70 megohms insulation resistance, and over a short period (2 hrs.) with gauges of less than 15 megohms. These figures were of course approximate and the ideal of infinite insulation resistance should always be aimed at. Tests had shown that gauges attached by celluloid cements were less stable than those attached by bakelite resin, and more information regarding the relative stability characteristics of these and other cements in use would be welcome. With the present celluloid cements, optimum stability should result from thorough initial drying, with subsequent protection of the gauge and adhesive from absorption of water by a water-resistant coating, the main requirements of which might be laid down as follows:—

(i) It must be an insulator, and completely waterproof.

(ii) It must not react on the cement or any other constituent of the gauge, nor change the sensitivity in any way.

(iii) It must form a watertight bond with steel, duralumin, and the more common of other structural materials.

A suitable protective coating commercially available had been found to be a solution of polythene in xylene. This had been used at the A.E.L. under severe conditions, and providing a good bond was also made with the covering of the wires to the gauge leads, had proved satisfactory. It was easily applied and quick drying and also afforded a slight degree of mechanical protection.

Random drifts could occur due to inadequate temperature compensation, sudden temperature changes (draughts, etc.), appreciable differences between temperature of the atmosphere and the structure, etc.; consequently the best possible thermal contact between the compensating gauge plates and the metal of the structure should exist, and thermal insulation and shelter should be provided over both active and compensating gauges. When extreme accuracy was required and appreciable temperature changes were expected, a lesser effect due to the coefficient of resistance of the wiring might become important, and in such a case the active and compensating gauge leads should be made equal in length and of the same wire.

The other common trouble met with was known as "creep" and was usually due to an unsatisfactory adhesive or sticking technique; this became apparent as a change in the strain indication under constant load and was probably due to either slip or plastic flow occurring in the cement layer between the gauge and structure. The direction of the change was usually opposite to that of the applied strain, indicating that the gauge was trying to return to its initial condition, although some cases had been recorded in which the change was in the same direction as the applied strain: the latter might be due to using an exceptionally thick adhesive layer. It was considered that if a good quality celluloid cement was used, creep could be avoided by ensuring complete evaporation of the volatile solvent of the cement. It might also be of use to apply one or two load cycles to the gauges before commencing the test in order to relieve any internal stress in the adhesive due to setting.

Another source of error encountered at A.E.L. had been the overlap of the gauge resistance wire on the nichrome connecting leads. If the weld was injudiciously placed during manufacture of the gauge, then "working" of the leads when the gauge was subjected to dynamic strain might cause resistance changes of indefinite and varying amounts at that point.

It should be emphasized that gauges should be calibrated under approximately the same strain ratios to which they were to be subjected in service. All resistance wire strain gauges consisting of more than a single strand of wire were in some degree sensitive to strains transverse to the axis of the gauge, and if multi-strand gauges were calibrated for uni-axial stress only, they might give rise to an error of up to 3 per cent. if used in an equal bi-axial stress field. The error in the measurement of strain in the axial direction was considerably increased as the ratio of transverse stress to axial stress was increased. It followed that if reliable strain data were to be obtained for a structure in which the ratios and directions of the principal strains were unknown, the gauges should have as little as possible of the wire laid in a direction perpendicular to the axis of the gauge. The transverse sensitivity of the gauges might be as much as +3 per cent. of the axial sensitivity, and might in some cases be negative (i.e. the effective axial sensitivity was reduced) dependent on the method of winding the gauges.

An effect not generally realized by investigators employing resistance wire strain gauges was that the strain sensitivity factors for most wires in the plastic region was 2 or very slightly greater, whereas in the elastic range this factor was constant at 2·63 for nichrome, 6·1 for platinum, etc., dependent on the wire used. Thus any gauge used in both elastic and plastic ranges should have a calibration factor of 2 in the former range to ensure a continuous linear response.

Thermo-electric effects between gauges and leads should be avoided, and might be eliminated by making sure that all the junctions of the bridge are at the same temperature, or by employing an a.c. bridge method of measurement.

Finally, a question arose regarding wire strain gauges mounted on a high-speed shaft or rotor, as to whether or not spurious voltages could be generated in the gauge windings due to passing rapidly through local magnetic fields.

The above sources of error, together with those enumerated by the author should not discourage prospective users, since a pre-knowledge of these and the methods of combating them would enable the resistance wire strain gauge to be used to its best advantage in the field of stress analysis.

Dr. S. Livingstone Smith (Director of Research, The British Shipbuilding Research Association): The paper was a valuable addition to the published data on wire resistance strain gauges, and the inclusion of detailed information on possible sources of error, particularly in electrical measurement, when using these gauges would be of great benefit to those who used wire resistance strain gauges for strain measurement.

The writer thought it could be accepted that wire resistance strain gauges could, for static and dynamic loading, give an accurate measure of the strain in a wide variety of members, provided the external conditions to which the member was subjected were favourable for the use of this principle of strain measurement, and also provided that experienced personnel with highly-skilled technique were available to carry out the tests.

It was apparent from the discussion, however, that experienced users were still occasionally meeting with difficulties other than those of electrical measurement, even in controlled laboratory tests. In this connection, it would be very helpful to prospective users of electrical resistance strain gauges to have more specific guidance on such points as the general technique of mounting the gauges in different circumstances, protection from moisture, and heat, etc. This would be of great benefit to those who used wire resistance strain full-scale structures such as ships at sea and on running machinery where conditions could not be controlled as in a laboratory.

If the author could supply information on this, or if the practical experience of various investigators could be collated and made generally available, it would be of inestimable service to those who were desirous of using electrical resistance strain gauges.

The Author's Reply to the Discussion.

Much of the discussion touched upon the problem of gluing. He realized that that was a very difficult problem, but as the field of strain gauges was very large he had had to limit the paper.

One or two speakers had mentioned the difficulty with regard to glue and also the question of creep. N.P.L. practice, whenever possible, was to bake the gauges, which got rid of most moisture and kept the glue hard. Under those conditions creep and hysteresis were unknown. Sometimes a strain gauge might be held responsible for hysteresis when it was not the gauge at all but in the object under test.

He endorsed the remarks of a speaker on the question of insulation. It was of vital importance. That could be seen by the fact that if one used a 2,500-ohm gauge, the change of resistance which was equivalent to a strain of one thousandth, was also roughly the same change in resistance as would be given by shunting that gauge with a 1-megohm resistance. Although they had been using 2,500-ohm gauges (and not 200 which was applicable to the American type gauges), he personally thought that the best value of resistance for static uses was rather less, and he was inclined to put it between 500 and 1,000 ohms.

He believed that it was possible to use comparatively light switches even for switching lower resistance gauges. In fact, the Americans appeared to use comparatively small switches for switching gauges directly. The circuits shown in the paper aimed at avoiding switching in gauge circuits and placing those switches in a circuit where their resistance was relatively unimportant. Although it was possible to have a switch of light construction which would give results, sooner or later it would lead to trouble, and there was nothing more annoying than having switch contact trouble in the middle of a test.

One speaker mentioned the question of the use of carrier methods, and they were working on possible ways of improving matters in that connection. The main difficulty with the carrier method was the problem of quadrature balance, that was, the bridge might be thrown out of balance due to change of capacity.

It could be shown quite easily that, if a bridge was initially balanced and was thrown out of balance due first to a small resistance change and then to a capacitance (or reactance) change, the two component voltages so produced were 90° out of phase. To determine the amount of resistive unbalance in the presence of capacity unbalance it was therefore necessary to use an instrument sensitive to the one component but unaffected by the other. An ordinary dynamometer wattmeter would do this, but it was insufficiently sensitive. Other phase sensitive detectors had been developed, but they used rectifiers which might not have a sufficiently constant performance. The laboratory was therefore trying to develop a method free from these troubles by using a cathode-ray tube but, instead of having the beam operating continuously, to arrange to switch it on for a brief interval, once per cycle, at the instant when the unwanted component of out-of-balance (due to change of capacity) was passing through zero. In this way the deflection due to resistive unbalance could be observed, in the presence of a small capacity unbalance, without the use of non-linear devices.

Reverting to the question of glues, he thought a lot more still had to be discovered in that connection. He did not think anyone knew why a glue adhered, much less why it failed to adhere.

He desired to supplement the last speaker's remarks by one further addition. With regard to phase distortion, he had said that at half a cycle/sec. it had been brought down to $3\frac{1}{2}$°. In the body of the paper, work was referred to which actually had been taken from figures the author had taken from an American publication, where frequency distortion of a sine wave was 0·015 per cent., but a square wave was subject to an error of $5\frac{1}{2}$ per cent. Actually that corresponded to phase distortion of only 1°.

One speaker asked for a method of oil-proofing gauges and the author could not give him any information on that point. He had not used them under oil.

The main criticism he had to answer was that he had concentrated on overcoming electrical errors and had said very little about gluing troubles. This was a deliberate choice and he felt it was justified by the fact that so many workers had already discovered for themselves the gluing trouble, and in many cases, how to overcome it, whereas the electrical errors he had tried to warn users against were generally more subtile and could easily pass unnoticed. The contributions of many speakers on the question of gluing were extremely valuable and they partially met the enquiries of others. Discrepancies sometimes occurred between the practice or experience of different workers; these were probably due mostly to environment, particularly temperature and humidity.

On the question of insulation resistance (leakage) he should add one point. If gauges had to be used for any length of time in a damp atmosphere, where perceptible leakage might occur, then it was important to earth the *positive* terminal of the battery. In this way electrolytic corrosion of the fine wire of the gauge would be avoided.

The contribution from Mr. Archer was very welcome as indicating the growing field of use of the gauges. The author was, however, surprised to learn that no effect had been observed due to fluid pressure on the gauges, as in one test made at the N.P.L. there was evidence that fluid pressure might have an effect. He noted that the Research Department of Lloyd's Register of Shipping obtained and used a strain sensitivity of 1·95 when using Durofix.

This corresponded to a partially set glue. Doubtless they were satisfied that they could reproduce this condition, but he must warn other users against adopting this figure without sufficient preliminary experiments to satisfy themselves that they could reproduce this figure with their own personal technique. Both Mr. Keig's and Mr. Mapp's remarks were valuable contributions and neither seemed to call for reply on any specific points. The remarks of Mr. Laing and Mr. Thompson had already been answered.

The author did not quite understand the first part of Mr. Rankin's communication. He thought that it was generally accepted that the specific resistance of materials was affected by strain. Mr. Rankin's suggestion in his third paragraph was no solution to the problem he posed, because a hypothesis of an abnormal specific resistance of material in small sections would not explain an abnormal *variation* of resistance with known strain of the sample.

With magnetic materials and certainly with magnetostrictive materials a different law of resistance change against strain would be expected using alternating current, but it would probably be necessary to use radio frequencies. A similar effect might possibly occur if non-magnetic gauges were glued to a magnetostrictive specimen.

The author was a little disappointed that, after he had stressed the predominating importance of phase distortion over frequency distortion, Mr. Rankin described some equipment purely in terms of its frequency response. As, however, the latter described the frequency response of a system, having a natural frequency of 70 cycles/sec. as "flat" to 100 cycles/sec. (by suitable choice of damping), it seemed that his standards were lower than those of most users, and the associated frequency distortion (which admittedly was very much less than that caused by a resistance capacity amplifier at low frequencies) could probably be accepted as not serious.

The author fully agreed with Mr. Edwards that insulation resistance required particular attention and, as the error due to a change of leakage was proportional to the square of the gauge resistance, he thought that in practice the actual gauge resistance used should be chosen primarily by reference to the conditions of use. The value of 2,500 ohms to which N.P.L. gauges were originally made was suitable for dynamic use, where leakage rarely caused appreciable error, but for long term static measurements a much lower value was preferable. On the other hand, on large structures appreciable errors of apparent strain sensitivity and of zero might arise from lead resistance and its change with temperature, respectively, when the gauge resistance was as low as 100 ohms. However, it was possible to eliminate lead resistance (as by the use of a Kelvin bridge) so that, on balance, the low resistance gauge had the advantage under very difficult conditions, but so far as the author was aware only one small group of workers were following this line.

Dr. S. L. Smith had, in effect, asked for the gluing problem to be dealt with as fully as the electrical side. The author, however, had had to limit the length of the paper and, since gluing troubles were nearly always much more obvious than electrical ones, he had thought it more valuable to give details about the latter. He thought that there was both scope and need for a separate paper on gluing, though this would be a difficult subject since experience showed that one worker could obtain consistently good results with a technique which another would condemn as bound to fail.

In conclusion the author wished to thank all contributors for their kind reception of his paper and to express his appreciation of the honour the Institute had conferred by their invitation to give the paper.

ADDITIONS TO THE LIBRARY.

Presented by the Publishers.

B.S. 61: Part 2: 1946. Screw Threads for Copper Tubes. 14p., illus., price 2s. net, post free.

B.S. 1304: 1946. "Ready-to-Fit" Thermal Insulating Materials for Hot and Cold Water Supply and Central Heating Installations for Small Dwellings. 14pp., price 2s. net, post free.

B.S. 41: 1946. Cast Iron Spigot and Socket Flue or Smoke Pipes. 17pp., illus., price 2s. net, post free.

—British Standards Institution.

Radio Acoustic Ranging (R.A.R.). By Commander K. T. Adams, United States Coast and Geodetic Survey. (With one plate). Publication 3782.

The David W. Taylor Model Basin. By Rear Admiral Herbert S. Howard, U.S.N., Director, David W. Taylor Model Basin. (With four plates). Publication 3783.

Research for Aeronautics—Its Planning and Application. By W. S. Farren, Director, Royal Aircraft Establishment. Publication 3784. —Smithsonian Institution, Washington, D.C.

Register of Ships, 1946. The British Corporation Register of Shipping and Aircraft, 14, Blythswood Square, Glasgow, C.2.

The Journal of the Institution of Electrical Engineers.
Vol. 93. Part I. No. 63. March, 1946.—General.
Vol. 93. Part III. No. 22. March, 1946.—Radio and Communication Engineering.

The Nickel Bulletin. Bound Volume 17, 1944. The Mond Nickel Company, Limited, Grosvenor House, Park Lane, London, W.1.

Ocean Waves and Swell. By G. E. R. Deacon, D.Sc., F.R.S., 1946, 11pp., 4 illus., 1s. The Challenger Society, c/o British Museum (Natural History), Cromwell Road, S.W.7.

Patents and Income Tax—Explanatory Notes. No. 490.

Income Tax—Notes on Allowances for Machinery or Plant. No. 430.

Income Tax—Notes on Allowances for Industrial Buildings. No. 410. —Board of Inland Revenue, Somerset House, W.C.2.

Scientific Research and Income Tax—Explanatory Notes. No. 470. Issued by the Board of Inland Revenue.

The Testing of Internal Combustion Engines. By S. J. Young and R. W. J. Pryer. The English University Press Ltd., London, 1945, 200pp., 87 illus., 8s. 6d.

The authors in the preparation of this small book have drawn upon their experience and material accumulated in the course of their work in the Internal Combustion Engines Laboratory at Loughborough College.

The book covers the general principles of testing procedure and deals with engines of a type likely to be found in well-equipped laboratories.

In the introductory chapter the authors divide engine testing into four categories; routine and acceptance tests; comparative tests; research testing and educational tests; and although it is the last category which the authors have kept in mind as being the type of test with which a heat engines class is principally concerned, the instruction given should enable the student to acquire the knowledge necessary for carrying out the more comprehensive type of commercial test, which must form the basis of "research" testing.

Chapters are devoted to general test procedure, engine timing, brake horse-power tests, fuel consumption tests, engine indicating heat losses, measurement of air consumption, and miscellaneous experiments.

In two appendices attention is drawn to the need for consideration of dynamometer size and characteristics in relation to the speed r.p.m. characteristics of the engine to be tested, and also the importance of taking dynamometer readings only when the dynamometer arm is in the correct position for which it was calibrated.

The book is clearly printed and well illustrated, and it fulfils the need for a small book dealing with the testing of internal combustion engines from the point of view of the average technical college; in addition it should prove valuable to the professional engine tester, as it gives that guidance necessary to enable him to appreciate the significance of the data obtained on engine tests.

When Ships Go Down. By David Masters. Eyre & Spottiswoode Ltd., London, sixth impression, 1945, 356pp., illus., 7s. 6d. net.

Wrecked surface ships may be salved for further service, for their cargo, or the material of which they are composed, but the prime concern in attempts to salve submarines is almost always the lives of the men shut up in them, who too often face a slow and horrid death. The most poignant stories in this book are therefore those of the salvage of submarines such as, for example, H.M.S. "Poseidon" and the American "S4". In the case of the "S4" the weather prevented the heroic efforts of the divers from raising the submarine in time to save the crew, but they carried on, and by raising the ship, gained new and valuable experience in submarine salvage technique.

The author discusses the fitting of safety devices on submarines such as eyes for lifting them, escape locks, emergency mark buoys, and air valves fitted to the hull, but apparently safety devices are not popular among submarine officers, and he quotes one case where a valve fitted to take an outside air pipe connection was persistently blocked. Nevertheless in this book he advocates lifting eyes as an integral part of a submarine's equipment "just in case they may one day prove the deciding factor that means life to the men manning one of these craft". One wonders what submarine officers would say as a result of the more recent experiences of the last war, for this book was first published in 1932.

There are other tales of salvage such as the recovery of the "Egypt's" gold, the raising of ships of the German Fleet in Scapa Flow, and the mystery of the Dutch liner "Tubantia", torpedoed in the North Sea in March, 1916. Certain passages in the book which obviously "date" it might well have been revised. Although many new stories of salvage remain to be told, the old ones lose none of their interest, and the volume loses little, if any, of its original appeal.

Manual of Firemanship—Part 7—Fireboats and Ship Fires. (Issued under the authority of the Home Office Fire Service Department). Published by His Majesty's Stationery Office, London, 1944, 149pp., profusely illus., price 2s. 6d. net.

Lloyd's Register of Yachts 1946. (Supplement to the 1939 edition).

The Furnaceman's Manual. Ministry of Fuel and Power Fuel Efficiency Bulletin, No. 43, May, 1946, 29pp., 22 figs., can be obtained free on application to the Ministry of Fuel and Power, 2, Little Smith Street, Westminster, or from the Ministry's Regional Offices.

The Welding of Cast Iron. (2nd edition). By L. Tibbenham, M.I.Mech.E. Sir Isaac Pitman & Sons, Ltd., London, 1945, 120pp., 74 figs., price 7s. 6d. net.

This little book is a useful addition to technical literature on the welding of cast iron. It deals only with the oxyacetylene welding process, which is the process most commonly used for cast iron; the low pressure and high pressure acetylene systems, together with the gas generating equipment, are also fully described.

Much of the information contained in the book is of a practical nature, but due emphasis has been given to metallurgical theory, so that the book should be of wider interest than to welders alone. Engineers and others who may be responsible for deciding the nature of weld repairs to iron castings will find appropriate guidance in these pages.

No doubt the author had to limit the scope of his treatment of the subject, but from the engineer's point of view some reference to physical properties of welds in cast iron would not have been out of place.

The Englishman and the Sea. An anthology. Edited by Christopher Lloyd. George Allen & Unwin, Ltd., 1946, 160pp., 7s. 6d. net.

Some twenty years have elapsed since the late William Blane read his paper at the Institute on "Poetry and the Engineer", under the presidency of the late Sir Fortescue Flannery. Members who were present will recall the inspiration they derived both from the lecture and from Sir Fortescue Flannery's introduction of the Author and his subject. If the same capacity for literary appreciation exists among engineers to-day, there are many to whom this present anthology can be commended. The reviewer can endorse the Publishers' statement that "the noblest expressions of what we as a nation owe to the sea, and the sacrifice involved, are here represented. It is a maritime, not merely a naval, anthology. It is a book for all those who love (or hate) the sea".

Additive Engine Oils. By G. A. Zamboni. (Compiled and edited by the Editorial Staff of the Petroleum Educational Institute, 704, South Spring Street, Los Angeles 14, California). The Mitre Press, Mitre Street, London, E.C.3, 1945, 143pp., profusely illus., 17s. 6d. net.

There is one very good reason why "Additive Engine Oils" by Zamboni (The Mitre Press) should be read by all motor engineers. This is because it clearly indicates the great and ever-growing complexity of modern lubricating problems and the consequent necessity for lubricants to be intelligently planned for their particular applications. The days when any old oil would do passed with the development of the modern high speed engine and its stringent requirements.

When purchasing oil, the importance of the choice of a reputable brand, backed by an adequate research and development organisation can clearly be inferred.

The unattractive reproduction and the irritating manner in which the text is broken up with illustrations must be tolerated by the reader if he is to obtain benefit from this work.

It may be felt that the frequent use of rather far-fetched analogies is taken to limits which tend to be tiresome to the reader. Some analogies used are questionable and in particular those provided to describe the nature of an additive may be mentioned. These occur on pages nine and ten and refer to the properties of certain mixtures of sand, gravel, cement and water and of salt and water. One infers from these analogies that an additive is a substance which necessarily affects the bulk properties of an oil, while in fact, many types of additive affect only the interfacial properties of the oil, which latter may, as far as the particular function is concerned, be regarded purely as a medium. While the water used for wetting the surfaces of sand and gravel is a poor analogy for an additive, it is perhaps a rather better one than the formation of concrete by the addition of water to a cement mixture or the dissolution of salt in water to form a saline solution.

Presented by the Author.

Oil Firing for Trawlers. Reprint of Lecture given on 7th November, 1945, by A. G. Dobbs, M.I.Mar.E., A.M.I.N.A., to the Hull Association of Engineers.

Purchased.

Reports of the Intelligence Objectives Sub-Committees. 9½in. by 7¼in., 1945, H.M.S.O., London.

Item No. 29. File No. XXX-33. Interview with Mr. Eric Schneider, Techical Director, North German Lloyd Line, 16pp., 2s. net.

Machinery's Handbook. (Twelfth edition). By Oberg and Jones. The Industrial Press, New York. Sole Distributors for the British Empire: Machinery Publishing Co., Ltd., 17, Marine Parade, Brighton, 1. 1945, 1815pp., profusely illus., 38s.

The new material covers a large variety of subjects that are important to designers and builders of everything mechanical. Recent or revised engineering standards are included, together with a large amount of general information and mechanical data representing the latest designing and manufacturing practice.

MEMBERSHIP ELECTIONS.

Date of Election, 7th May, 1946.

Members.

George Webster Atherton.
John Bradford Black.
Alfred Leslie Clayton.
Hugh Dean.
Thornton Ernest Draggett.
George Henry Fairlem.
Joseph Patrick Gateley.
William Green.
William Henry Hewitt.
Herbert Arthur McConochie.
Archibald Victor Monk.
William Oswald Nicholson.
John Oliver.
William Brass Olsen.
Arthur George Porter.
Peredur Wynn Thomas.
George Torrie.
Roderick John McKenzie Shirrefs, Sub. Lieut.(E.), R.N.R.
William George Smith, Lt.-Com'r.(E.), R.N.V.R.
William John Smyth.
William Frederick Springgay, Lieut.(E.), R.N.V.R.
Herbert Edmund Tune.
John Irwin Walker.
Richard Wanless.
John Downing Willcocks.
Hugh Wynn-Davies.

Associate Member.

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Associates.

Alan Armer.
Edward Alexander Crowther Doyle.
Edward James Edgley, Lieut.(E.), R.N.R.
Jack Donald Fletcher Elvish.
George Dickie Fleming.
Charles Niesh Duncan Riddell.
William Henry Percival Robey.
Arthur Noel Savage.

Graduates.

Paxton South.
Leonard Teasdale.

Transfer from Associate to Member.

Henry Noel Buddle.

Transfer from Graduate to Associate Member.

Reginald Sydney Brett.

Transfer from Graduate to Associate.

John Colville Grimmett.

Transfer from Student to Graduate.

Shyam Uttamsingh, Lieut.(E.), R.I.N.

PERSONAL.

Professor L. C. Burrill, M.Sc., Ph.D. (Member), Engineer Vice-Admiral Sir John Kingcome, K.C.B. (Vice-President), and

DR. S. LIVINGSTON SMITH, F.C.G.I. (Member), have been elected Members of Council of the Institution of Naval Architects.

W. B. ANNING (Member) has retired from his appointment as engineer surveyor with the General Accident Assurance Company.

THOMAS H. G. BRAYFIELD (Member) is at present in Vancouver completing his recovery from nearly four years internment by the Japanese at Stanley Internment Camp, Hong Kong. Mr. Brayfield has severed his former connection with Messrs. Carmichael & Clarke, but is returning to Hong Kong at the end of July next.

H. N. BUDDLE (Associate) has joined the staff of Messrs. R. A. Lister (Marine Sales) Ltd. of Dursley.

W. G. C. BUTCHER (Member) has relinquished his post of Controller of Ship Repairs on being demobilized from the Royal Australian Navy.

F. D. CLARK (Associate Member), who has been elected an Associate Member of the Institution of Naval Architects, is shortly to take up an appointment with The British Mexican Petroleum Co., Ltd.

ERIC DAVIES (Member) has re-opened his office in Shanghai after release from a Japanese internment camp. Mr. Davies had an adventurous career in the Far East during the war, and his experience and sufferings were of an exceptional character.

P. J. FALLON (Associate) has been appointed outside manager with Messrs. Guthrie, Murdoch & Co., of Antwerp.

J. W. FIRTH, O.B.E. (Member), has left the service of Messrs. Grayson, Rollo & Clover Docks, Ltd., to take up an appointment as chief engineer with Messrs. Saguenay Terminals, Ltd.

J. N. GALLEY (Associate) has taken up an appointment with the Ministry of Works as Fuel Economy Officer and Boiler Inspector for the Birmingham district.

M. F. HEPTON (Member) has now been demobilized and is returning to his own firm, Messrs. Townhill & Hepton of Hull.

W. N. IMRAY (Member) is retiring after lengthy service with the Union Castle Line. He was for many years in charge of the Company's repair works at Blackwall.

A. MCDOUGALL (Member) has relinquished his post with the Ministry of War Transport as senior inspecting officer, Sea Transport, and director of merchant ship repairs, M.E.

J. S. MASON (Member) has commenced in private practice as a partner in the firm of consulting engineers, naval architects and marine surveyors of Messrs. J. S. Mason & Partners, Westminster.

ENG. LT.-COM. J. M. MUIR, R.N.R. (Member)) has resumed his position as branch manager with Messrs. L. Sterne & Co., Ltd., of Glasgow, on his return from captivity in Germany and demobilization from the Royal Navy.

D. L. NEOGY (Associate Member) has been appointed marine superintendent of the Bengal Assam Railway.

S. A. SHINGLER (Member) has taken up an appointment in South Georgia in the service of the Ministry of War Transport.

S. J. SMITH (Associate Member) has been appointed London office manager of Messrs. Priestman Brothers, Ltd., Holderness Foundry, Hull.

R. STIRLING (Member) has resumed his sea-going career with his old Company, The Donaldson Line of Glasgow, by whom he has been appointed 2nd engineer. Mr. Stirling held the rank of Lt.-Com. (E.) in the Royal Naval Reserve.

LIEUT. (E.) A. J. TAIT, R.N.R. (Associate) has been appointed an assistant technical officer with the I.C.I., Billingham Division.

D. J. B. TAIT (Student) has been promoted to the rank of Captain, R.E.

W. TILBY (Member) has returned home after 4½ years internment in Germany.

A. N. TIPPER (Member), who was mentioned in despatches in December, 1945, has been promoted to the rank of Lt.-Com. (E.), R.N.R.

A. N. TODD (Associate Member) has been demobilized from the Royal Navy with the rank of Lieut. (E.).

J. H. TODD (Member) has terminated his association with The Todd Shipyards Corporation and has opened offices in New York on his own account.

H. WAITE (Associate) has recently been demobilized from the Royal Naval Reserve with the rank of Sub.-Lt. (E.).

The following Members have gained the awards named for papers presented in recent sessions to the North-East Coast Institution of Engineers and Shipbuilders :—

J. L. ADAM—The Shipbuilding Medal, 1942-43.
PROF. L. C. BURRILL—The Engineering Medal, 1943-44.
C. C. POUNDER—The Engineering Medal, 1944-45.
W. H. PURDIE—The James Memorial Medal, 1943-44.

The INSTITUTE of MARINE ENGINEERS

Founded 1889. Incorporated by Royal Charter, 1933

SESSION 1946. *Transactions* Vol. LVIII. No. 5.

Patron: HIS MAJESTY THE KING. President: SIR AMOS L. AYRE, K.B.E.

Oil Engine Dynamics with Special Reference to the Opposed-Piston Engine.

By *W. KER WILSON, D.Sc., Ph.D., Wh.Ex., M.I.Mech.E.

Read on Tuesday, March 26th, 1946, at 5.30 p.m. at 85, The Minories, E.C.3.

Chairman: Mr. J. A. RHYNAS (Chairman of Council).

Synopsis.

The outlook on engine dynamics has changed considerably since the days when the determination of forces and couples arising from motions of the masses and investigation of methods for reducing their numerical values within the limits imposed by other design considerations was regarded as the high water mark of achievement. In recent times remarkable strides have been made in theoretical studies, assisted by no less important advances in instrumentation. In consequence there has been an increasing tendency to regard vibration study as a necessary accompaniment of sound fundamental design.

Since the earliest times marine engineers, naval architects, and their associates have made important contributions towards present-day knowledge, and in several instances their work has been directly responsible for stimulating new lines of thought, leading to major developments. As long ago as 1886, for example, the importance of crankshaft flexibility in preventing failure was recognized, while in 1897 Macalpine published a series of articles on vibration in steamships which were well ahead of contemporary thought and, indeed, anticipated ideas which appeared many years later. In the same year he produced a comprehensive analysis of the inertia forces of the moving parts of reciprocating engines. The Yarrow, Schlick and Tweedy patents of 1894/1895 represented an outstanding contribution on engine balancing, while between 1900 and 1902 Bauer and Frahm were amongst the first to direct attention to the serious nature of torsional vibration in engine shafting, with special reference to marine applications.

The present paper is an attempt to summarize in an interesting manner the problems on engine dynamics which were encountered during the development of the Doxford opposed-piston marine oil engine. The principal troubles which occurred at various stages in the development are described together with the theoretical and practical work which was undertaken to effect a cure. In several instances the corrective measures led to important design improvements.

The topics discussed in the paper are:—General problem of vibration; methods of dealing with vibration; hull vibration; the balancing of opposed-piston engines; practical experience on engine balancing; the evolution of the balanced opposed-piston engine; propeller vibration; vibration of engine framing; synchronising devices; torsional vibration; influence of changes of mass distribution and changes in shafting stiffness; influence of firing order; damping devices; practical experience on torsional vibration; and the evolution of the Doxford-Bibby detuning flywheel.

No attempt is made to deal with the mathematical details involved in the study of torsional vibration problems. An extensive literature on this subject has accumulated and a few selected references are given in a bibliography.

* Chief Research Engineer, The de Havilland Engine Co., Ltd., London; formerly Assistant Chief Designer, Messrs. Wm. Doxford & Sons, Ltd., Sunderland.

At one time determination of forces and couples arising from motions of the masses, and investigation of methods for reducing their numerical values within the limitations imposed by other design considerations were regarded as the high water mark of achievement in practical engine dynamics.

The outlook has, however, changed considerably, even since the days when the full significance of torsional vibration in engine shafting began to be appreciated.

It is now generally conceded that mere evaluation of applied loads has little significance if knowledge of the response of the structure and its surroundings is lacking. The full solution of the problem requires the determination of the dynamic characteristics of the whole system, a task which in many cases can only be performed by experimental methods. Neglect of this fundamental truth has been responsible for many failures, of which an example is the serious consequences which might follow the fitting of crankweb balance weights or a heavy flywheel without proper regard to the effect of such additions on torsional vibration.

It is perhaps natural to find many recent contributions on engine dynamics concerned with developments in the aircraft industry where performance requirements are severe and power plants are highly developed. At the same time, however, a review of papers on this subject read before senior technical institutions gives some impression of the immense contribution made by marine engineers, naval architects, and their associates. In many cases this work represented a pioneering contribution to present-day developments.

In this connection it is perhaps not without significance that the relatively large dimensions and slow speeds of marine power plants facilitate study of fundamental phenomena under what may be termed "slow-motion" conditions, with the aid of comparatively simple recording equipment.

Under these conditions there is little difficulty in observing the practical results of design changes, and investigations on marine installations have served as classical illustrations of principles underlying engine balancing and vibration theory. A few early contributions in this field are worth recalling.

In 1886 the author of a paper read before the Institution of Naval Architects(1) directed attention to an improved construction for crank and screwshafts which aimed at providing greater flexibility (see diagram *B* in Fig. 1). In his concluding paragraphs the author, Mr. J. F. Hall, wrote:—

> "In conclusion it goes without saying that a crank and propeller shaft must be one of two things, either completely rigid, or completely flexible. So long as half-hearted shafts are used, which are neither one thing or the other, so long will they continue to wriggle against the inevitable in the vain attempt to retain their true form at the expense of their vitality, decay and final collapse. All that was feasible and practicable without regard to cost has been done to make the rigid shaft a success in some of the finest vessels recently floated. But the fact that they have constantly to be lined up in their bearings is evidence that

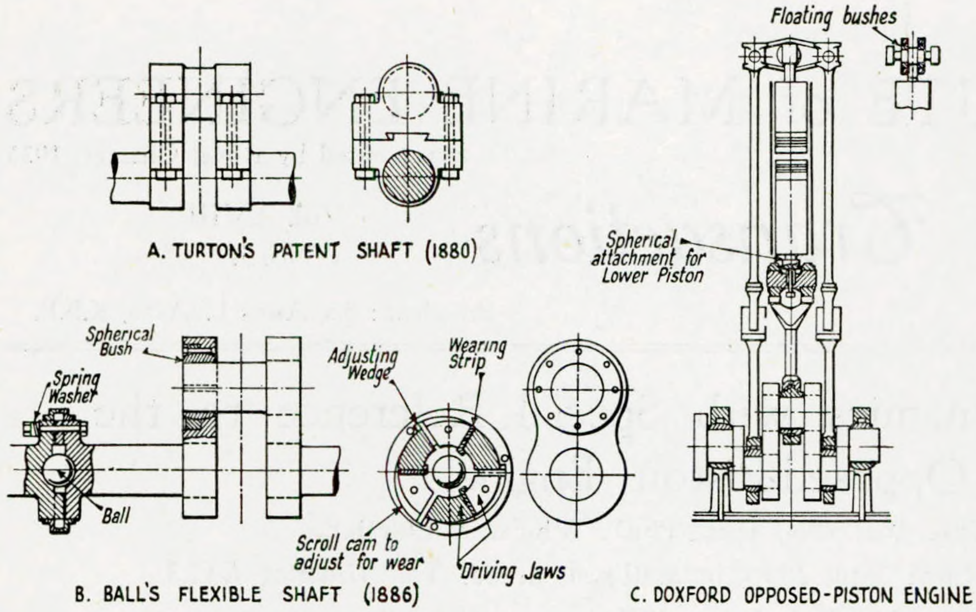

FIG. 1.—*Flexible shafts.*

the shafts run the risk of giving way, or have actually been bent at some point or other. Hence it is certain that rigid shafts will still continue to fail so long as bearings wear unequally and hulls strain. Also as bearings that will not wear, and hulls that will not strain, are unobtainable, the only way of consigning broken shafts to things of the past is to elude the evils by using flexible shafting".

In the earlier part of his paper, Mr. Hall wrote in favourable terms of the Turton patent built-up shaft, the manufacture of which was undertaken by Messrs. Wm. Jessop & Sons Ltd. of Sheffield about 1880. The idea underlying this shaft—see sketch *A* in Fig. 1—was the facility with which the crankpin portion could be replaced, an operation which it was hoped could even be carried out at sea. Although, according to Hall, this desirable feature was not realised, the shaft had a virtue in its additional flexibility which permitted some distortion under load without producing the fractures which were fairly common with more conventional designs at that time.

Over twenty-five years later, in 1912, Professor Junkers expressed similar sentiments when he wrote as follows concerning the three-throw crank element of his opposed-piston engine:—

"The only effect torsional vibrations have on the crankshaft is to subject it to a twisting moment tending to bend the webs and twist the crankpins and journals. The resultant strains produce a parallel translation of the shaft journals and should the bearing prohibit this movement corresponding loads and inclinations in the journals result. Since the effects of the spring in the shaft can readily be avoided by using spherically seated bearings, the advantage of the flexibility of the three-throw crank elements of the Junkers engine becomes quite obvious inasmuch as the loads it produces on the bearings are smaller than those in the crankshaft of a Diesel engine. It may be noted that the spherical supported bearings seem rather advantageous because they permit the shaft to take up positions peculiar to respective points in the revolution. In any case a shaft of this nature seems more congenial to the constructive sense of the engineer than a hardly flexible, stiffly supported shaft in rigid bearings with its inevitably heavy loads".

Although this quotation suggests that Junkers was a keen advocate of spherical bearings, it is interesting to recall that the Junkers marine engines built in 1910/1912 had plain bearings throughout. This might well have been one of the design features contributing to the failure of these ventures.

In the earlier Oechelhaeuser gas engine not only were plain bearings used throughout, but the side rods were rigidly connected to the transverse beams, thus preventing the automatic compensation for unequal straining of the two rods obtained with hinged connections. The latter feature resulted in considerable trouble due to failures of side rods.

In recent times innumerable laboratory and full-scale tests have shown that very high local stresses can occur at points in a structure subjected to dynamic loads where there is a sudden change of section, i.e. at points where the flexibility changes suddenly.

Spherical bearings have been a feature of the Doxford opposed-piston engine crankshaft assembly from the beginning; while in later designs provision has also been made to permit the pistons to run freely in the cylinder bore even when the guide shoes are not in correct adjustment or when there is some slight distortion of the framing such as might occur during warming up. These features are shown in sketch *C* of Fig. 1. The self-aligning characteristic of the lower piston is obtained by a spherical attachment to the centre cross-head, and in the case of the upper piston by floating bushes in the bores of the transverse beam. The effectiveness of these precautions has been confirmed by running experience, which indicates that given proper maintenance and freedom from accident, the original white-metal linings of the main bearings should last the life of the ship (about 20 years); while the introduction of self-aligning pistons is one of the factors contributing to the reduction of cylinder liner wear to the point where a life of eleven years is expected from each liner.

In 1897[2] Macalpine published a series of articles on vibration in steamships, which were well ahead of contemporary thought, describing various forms of vibration absorbers capable of dealing with harmonic as well as fundamental disturbances. In the same year he also published a comprehensive analysis[3] of the inertia forces of the moving parts of reciprocating engines.

The Yarrow, Schlick and Tweedy patents of 1894/1895 represented an outstanding contribution on engine balancing, and provided an elegant and practical solution of the problem of balancing multi-crank engines which is still in use to-day. Between 1900 and 1942, Bauer[4] and Frahm[5] were amongst the first to direct attention to the serious nature of torsional vibration in engine shafting, with special reference to marine applications.

These and other important contributions indicate that marine engineers have taken much more than a superficial interest in engine dynamics, and in several instances have been directly responsible for stimulating new lines of thought, leading to major developments.

Vibration is by no means the malignant influence which it appears to be if judged solely by the records of failures from this cause. On the contrary there is little doubt that few structures would withstand repeated application of live loadings if they had not the capacity

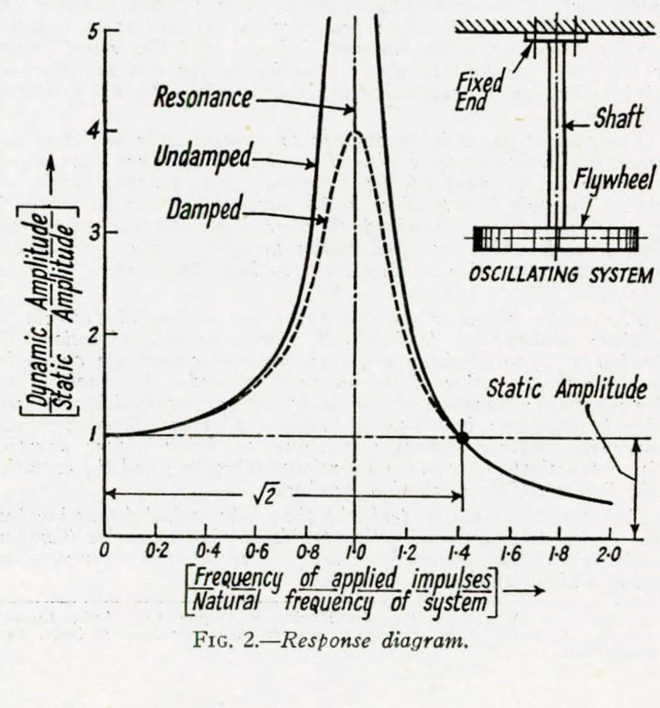

FIG. 2.—*Response diagram.*

for absorbing the impulses by elastic recoil followed by gradual dissipation of strain energy by vibration.

Nevertheless, when pulsating forces are applied to an elastic structure care is necessary to make sure that they do not, through the phenomenon of resonance, excite vibrations of sufficient amplitude to be destructive.

Resonance is such an important phenomenon in vibration engineering that a brief discussion of its cause and effect is necessary for proper appreciation of the remaining portions of this paper.

Referring to Fig. 2, consider an oscillating system comprising a heavy flywheel attached to one end of a length of flexible shafting firmly clamped at the other end. This arrangement represents a simple torsional pendulum.

If a steady torque is applied to the flywheel, it will cause the flywheel to rotate through a small angle until the restoring couple due to the internal elastic forces of the twisted shaft balances the applied torque. The work done against the elastic forces while the shaft is being twisted is stored as strain energy in the shaft.

The angular amplitude through which the shaft is twisted by the application of a steady torque is called the "static" amplitude to distinguish it from the "dynamic" amplitude discussed in later paragraphs. The stress corresponding to this static amplitude is called the static stress.

When the external torque is suddenly removed, the elastic forces begin to restore the system to its original configuration. The strain energy, stored in the shaft when it was twisted, is converted into kinetic energy which is stored in the flywheel when it begins to move as the shaft untwists.

When the flywheel reaches its original position of equilibrium, the shaft is, of course, completely untwisted and the whole of the strain energy imparted during the initial twist has been converted into kinetic energy, stored in the flywheel, which is then moving with its maximum angular velocity. The flywheel therefore continues to move beyond the original unstrained position until a strain equal to the original strain, but in the opposite direction, is imparted to the shaft, at which instant the whole of the energy imparted by the original twist is once again stored as strain energy in the shaft.

In a system without any damping or retarding forces, this sequence of events, once initiated, would continue indefinitely with the flywheel swinging from one extreme position to the other with a regular rhythmic motion. At each extreme position the flywheel would come momentarily to rest and the whole of the energy imparted by the initial twist would be stored as strain energy in the shaft: while at the instant of passing through the original position of equilibrium, now the mid point of the motion, the energy would be wholly kinetic and stored in the flywheel. At intermediate positions the energy would be partly strain energy stored in the shaft, and partly kinetic energy stored in the flywheel, but at every point in the motion the total energy would remain equal to the energy imparted to the system by the initial twist.

In all practical systems, however, there is always some resistance to the motion, i.e. some damping influence, which prevents the motion from continuing indefinitely. This damping influence might, for example, be due to air or some other frictional resistance at the flywheel, or hysteresis loss in the material of the shaft, or a combination of both. These losses gradually dissipate the vibrational energy with the result that the amplitude of the motion gradually diminishes until eventually the system comes to rest at its original position of equilibrium.

Vibration may therefore be regarded as the process whereby an elastic system dissipates the strain energy imparted to it when its equilibrium is disturbed.

A vibration cycle is defined as a movement of the flywheel from one extreme position to the other and then back again to the starting point, the amplitude of the vibration being defined as one half the total movement between the extreme positions. For a given oscillating system it will be found that the time required to complete each cycle is the same, i.e. the flywheel vibrates freely at a fixed number of cycles per minute after the disturbing torque is removed. This frequency is termed the natural or free vibration frequency of the system, and is determined solely by the inertia of the flywheel and the stiffness of the controlling shaft. Thus a heavy flywheel on a relatively flexible shaft will vibrate with a low natural frequency, whereas a light flywheel on a relatively stiff shaft will vibrate with a high natural frequency.

The foregoing discussion refers solely to the case where an oscillating system is left in a state of free vibration following the removal of a steady disturbing force. The case where a sustained pulsating load is applied to the system is more complicated since the resultant motion depends on the relationship between the frequency of the applied impulses and the natural frequency of the system.

When the pulsating load is first applied to the system, the resultant motion consists of two superimposed vibrations, a damped natural vibration at the natural or free vibration frequency of the system, and a damped forced vibration at the frequency of the applied or forcing impulses. The amplitude of the free vibration diminishes more or less rapidly according to the amount of damping in the system, and eventually the motion settles down to a steady state in which the amplitude is that corresponding to the damped forced vibration alone. During the starting period a transient vibration is experienced, the amplitude of which depends on the conditions at the moment of starting, and may be as much as double the amplitude of the steady state vibration to which the motion eventually settles down.

Transient vibration amplitudes are, however, not usually very troublesome in mechanical engineering applications, and in the majority of cases the transient stage is disregarded.

The diagram in Fig. 2 shows the steady state response of a simple oscillating system under a simple pulsating load. When the frequency of the applied impulses is a small fraction of the natural frequency of the system, the amplitude of the forced vibration differs very little from the deflection which would be produced by static application of the maximum value of the pulsating load to the system. In other words, the *dynamic amplitude* is only slightly greater than the *static amplitude.*

When, however, the frequency of the applied impulses becomes greater than one-half of the natural frequency, there is a rapid increase in the dynamic amplitude until, in the absence of damping, the dynamic amplitude approaches infinity when the applied frequency approaches the natural frequency. This phenomenon is termed resonance, and the speed at which the frequency of the applied impulses is equal to the natural frequency is termed the resonant or critical speed.

At impulse frequencies above resonance the dynamic amplitude begins to diminish and falls rapidly until it becomes equal to the static amplitude when the applied frequency is $\sqrt{2}$ times the natural frequency. Thereafter the dynamic amplitude diminishes at a decreasing rate and approaches zero as the applied frequency approaches infinity. It is of interest to note that for impulse frequencies greater than $\sqrt{2}$ times the natural frequency, the dynamic amplitude is less than the static amplitude. Many proposals for reducing vibration have been based on attempts to exploit this characteristic.

Although in all practical cases the existence of some degree of damping prevents the dynamic amplitude at resonance from becoming infinite, the dynamic amplitudes may nevertheless reach values which are 100 or more times the static amplitudes. Thus, in a badly tuned system it is possible for quite small exciting loads to produce dangerously high dynamic stresses. On the other hand, in a well tuned system the dynamic stresses may well be within acceptable limits even though the exciting loads are quite appreciable.

In this connection, however, it is well to bear in mind that modern requirements are not necessarily fulfilled by designs which merely provide freedom from mechanical failure. Most engineers now and then experience the satisfaction of handling a machine in which the different design factors are so well matched that the result is a product of outstanding merit. Even to-day, however, such examples are comparatively rare, and in many cases are confined to individual specimens from a production batch.

The achievement therefore appears to be largely the result of a happy accident which, somehow, provides a product which is neither too sluggish and lifeless, nor too lively and over-sensitive. This suggests that an important contributory factor towards the achievement of these ideal results is accidental realisation of the correct solution of the vibration problems associated with each case. Vibration study is therefore likely to continue to exert considerable influence on engineering design and development, and more and more attention is likely to be given to the proper control of vibration characteristics while a project is in the design stage.

There are three principal methods of dealing with vibration, namely:—

(1) Tuning the system so that there are no important critical speeds in the running range.

This is not an easy task in installations which are required to operate over a fairly wide range of speeds, especially if the oscillating system contains a large number of principal inertias and flexibilities, and is excited by a forcing load which contains many harmonic components.

In such cases there may be two or more significant modes of free vibration each having a different natural frequency, and a whole series of significant components of the exciting load each having a different impulse frequency. Thus, there may be several important resonant speeds at which the frequency of a significant mode of natural vibration coincides with the impulse frequency of a significant component of the exciting load.

A great deal can be done in the design stage, however, by careful adjustment of the principal inertias and flexibilities, to obtain a favourable disposition of critical speeds.

During recent years much use has been made of vibration absorbers for producing a favourably tuned oscillating system.

These devices consist essentially of a supplementary mass or masses flexibly connected to the main oscillating system at a point or points where their influence will have most effect in reducing vibration amplitudes. The much-discussed rotating pendulum vibration absorber can be included in this category.

It should be noted that besides changing the natural frequencies of the system, an alteration in the magnitudes or the disposition of the principal masses and elasticities may have an important influence on the vibrational energy input, where exciting loads are applied at several points in the system. This is because a change in the tuning of the system may produce an appreciable change in the relative amplitudes of vibration at the various points of application of the exciting loads.

Similarly, a change in the tuning of the system may have an important influence on the damping charactistics of the system by changing the relative vibrating amplitudes at the damping points.

(2) Reducing the magnitude of the exciting impulses.

Consideration should always be given to the possibility of reducing the magnitude of the excitations, since this attacks the problem at one of its roots.

Although, as already stated, even a small exciting load is capable of producing dangerously large vibrations through resonance, nevertheless a reduction in the magnitude of the excitation does reduce the vibration amplitude and this may either bring the amplitude within permissible limits or reduce the amount of supplementary damping required.

Thus, excitations arising from the motions of the reciprocating and revolving masses of a reciprocating engine can be reduced or even eliminated by careful balancing; while the amount of vibrational energy going into the system can be appreciably reduced in certain cases by an appropriate selection of the firing order of a multi-cylinder engine.

(3) Introduction of supplementary damping.

The need for special devices which aim at augmenting the inherent damping of the system should rarely arise if the methods discussed in the preceding sections *(1)* and *(2)* are properly applied.

Indeed, the use of special damping devices is to be deprecated since they are apt to behave erratically in service through over-heating and wear caused by friction. In addition, such devices represent an energy loss which is not incurred with other methods.

Energy absorbing devices may be useful, however, where, for example, it is necessary to reduce the amplitude of a critical speed lying below the operating range of speeds to enable the engine to be started or stopped without risk of failure.

The Balancing of Opposed-Piston Engines.

Prior to 1926 the strokes of the upper and lower pistons of all slow-speed Doxford engines were equal, and since the weight of the reciprocating parts of the upper piston assembly was about twice that of the lower piston there was an appreciable unbalanced reciprocating weight at each cylinder. Under running conditions this produced a pulsating force along the line of stroke of each cylinder which varied from a maximum to a minimum during each revolution. This once-per-revolution disturbance is known as *primary unbalance.*

In multi-cylinder engines, with evenly spaced cranks and cylinders in line, the inertia forces acting at the individual cylinders cancel one another, so that there is no residual inertia force for the engine as a whole. There is, however, an unbalanced couple tending to rock the engine about its centre of gravity at a frequency equal to the r.p.m. of the engine. This couple is due to the greater leverages of the primary inertia forces acting at cylinders remote from a given reference plane.

Now the application of pulsating force or couples to an elastic structure causes deformations which absorb a certain amount of energy and results in vibration which dissipates the input energy as heat generated by inter-molecular friction in materials or mechanical friction at rubbing surfaces.

Even so, the response of a structure to quite large disturbing forces or couples is often surprisingly small when, as already explained, the vibrating system is favourably tuned with respect to the frequency of the exciting impulses.

On the other hand, the vigorous response of an unfavourably tuned structure to quite small excitations when resonance occurs between the natural and excitation frequencies is equally surprising.

The powerful sympathetic response under resonant conditions is indeed at the root of most vibration problems encountered in mechanical engineering, and is not confined to vibration of the structure as a whole. In general every pulsating force searches the whole of the structure and its surroundings for points of sympathetic response and this may result in severe local resonance conditions becoming established even though the structure as a whole is relatively quiescent.

The hull of a ship is capable of vibrating in several principal ways or modes, including flexural vibration about the two principal axes of inertia, torsional vibration, and local vibration such as the rocking of the engine on its seating in the transverse or longitudinal directions.

The problem is extremely complex since a ship's hull represents an oscillating system having distributed mass and flexibility so that there is a whole series of different normal modes corresponding to each class of vibration. In most cases, however, the significant modes of vibration are vertical flexural vibration with two or three nodes, horizontal flexural vibration with two or three nodes, and torsional vibration with one or more nodes.

The problem is further complicated by the fact that there is an appreciable variation of natural frequency with the load condition of the ship, and with the depth of water under the hull.

From the point of view of engine balancing the unbalanced inertia forces and couples caused by the motions of the reciprocating and revolving parts of the engine are capable of producing severe response from the hull if their frequency happens to coincide with the natural frequency of one of the modes of vertical or horizontal flexural vibration.

Vertical vibration results from response to unbalance of both reciprocating and rotating parts, while horizontal vibration results mainly from response to unbalance of rotating parts only.

The earlier types of Doxford opposed-piston engine were mostly of the slow-speed type for installation in single-screw ships, and therefore the possibility of encountering severe resonant conditions was rather remote.

Thus, the engines installed in the first two motorships were of the equal stroke type with four cylinders 580 mm. bore by 2 by 1,160 mm. stroke, operating at a maximum speed of 77 r.p.m. Due no doubt to the low number of revolutions per minute, there was a remarkable absence of vibration, despite the fact that in addition to the unbalanced primary and secondary couples originated by the motion of the main cylinder masses there were unbalanced primary and secondary forces originated by the reciprocating and rotating parts of the centrally situated crank-driven scavenge pump. Apart from the inherent self-balancing characteristic of the opposed-piston mechanism, no special provision was made for balancing the moving parts.

During 1923 and 1924 the operating speed of engines of substantially the same design was increased to 87 r.p.m. Again no special provision was made for balancing the moving parts, and since the scavenge pump design was unaltered, the increased speed represented an increase of 28 per cent. in the magnitude of the unbalanced forces. In the case of the main cylinders, however, experience on earlier engines had enabled a reduction to be made in the weights of the reciprocating parts of the side and centre drives with the result that the increase in the magnitudes of the unbalanced couples was only 14 per cent.

The first ship to be completed showed perceptible vertical vibrations which were attributed to the primary vertical forces originated by the unbalanced reciprocating and rotating parts of the scavenge pump. It was therefore decided to fit balance weights of the type shown in the upper diagram of Fig. 3 to the scavenge pump crankwebs. In later engines the shape of the scavenge crankwebs was altered as shown in the middle diagram of Fig. 3 to facilitate machining and assembly.

The balance weights were made of cast steel with lead fillings to enable the required mass to be accommodated inside the engine crankcase. They were secured to the crankwebs by long studs, with dowel pins for axial location.

Some trouble was experienced through loosening of the lead filling, but this was overcome by improving the passages for escape of air during pouring and by pulling the sides of the cast steel shell together with a series of bolts so that they gripped the lead filling securely. The pinching bolts were then removed one at a time

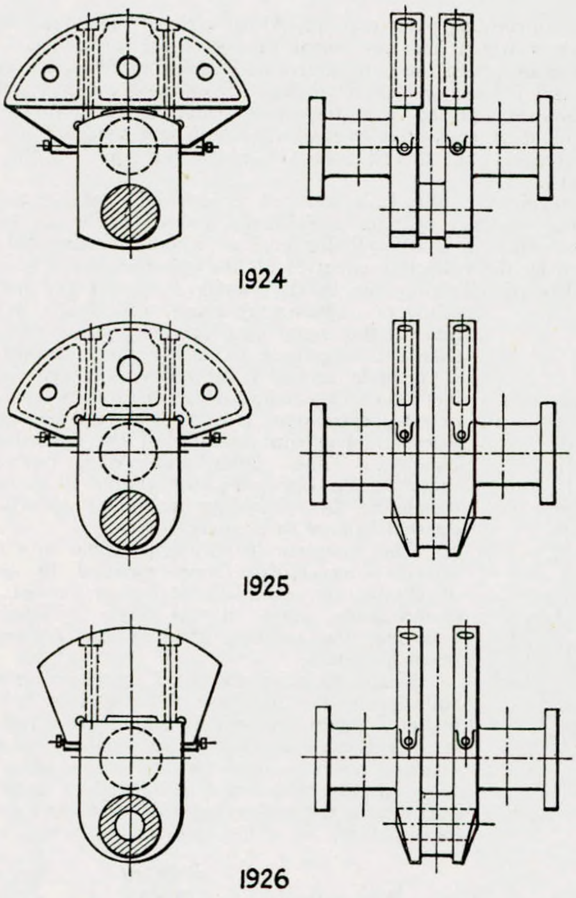

FIG. 3.—*Scavenge crank balance weights.*

and replaced by steel stays screwed right through the steel shell and lead filling and firmly riveted in place.

The balance weights were designed to eliminate the primary vertical force of the scavenge pump at the expense of overbalance in the horizontal plane, because it was considered that there was less likelihood of resonant conditions becoming established with the natural frequency of a horizontal mode of hull vibration. The introduction of these scavenge crankweb balance weights enabled the engines to operate at the designed speed without perceptible vibration.

In 1926 the construction of a series of four-cylinder engines with central crank-driven scavenge pump and with much larger cylinder dimensions was undertaken. These engines had cylinders 680 mm. bore by 2 by 1,360 mm. stroke and were designed to operate at 90 r.p.m. Due to the larger cylinder dimensions and the increased speed, the unbalanced couples originated by the main cylinder masses were more than double the values for the smaller engines, but no special precautions were taken to minimize these couples because it was considered that unbalanced couples were usually much less troublesome than unbalanced forces, especially in installations with the engines amidships. At the same time it was appreciated that every effort must be made to minimize unbalanced forces, and since these were originated solely by the scavenge pump the whole of the running gear of the pump was redesigned. The design changes consisted of substituting a light bronze piston for the heavy cast-iron piston previously employed; an aluminium piston trunk for the previous cast-iron trunk; and valve guards of aluminium instead of bronze. The scavenge pump piston rod, connecting rod, and crank-pin were made hollow instead of solid. These alterations resulted in reductions in the weight of the reciprocating and revolving parts of the scavenge pump of 50 per cent. and 10 per cent. respectively. Thus, despite the larger size and higher speed, the unbalanced inertia forces originated by the pump were no greater than they had been for the smaller, slower-running, engines.

Incidentally, these reductions in the weights of the running gear of the scavenge pump enabled the simple solid cast-iron balance weights shown in the lowermost diagram of Fig. 3 to be employed.

Scavenge pumps with lightened reciprocating and revolving parts were adopted for all subsequent engines employing centrally situated crank-driven pumps, as a precaution against the possibility of resonance between the primary inertia force of the pump and the natural frequency of the fundamental vertical mode of hull vibration. In later designs a further small reduction of the reciprocating weight of the pump was obtained by using welded steel fabricated construction for the piston instead of cast bronze, and it is of interest to note that piston rings were not used with either the bronze or the fabricated steel pistons. The elimination of the piston rings had no noticeable influence on the performance of the pump, which is perhaps not surprising bearing in mind the large diameter of the piston and the low working pressure—about 2lb. per sq. in.

During the maiden voyages of the first two ships fitted with these larger engines considerable trouble was experienced through loosening and breakage of the crankshaft coupling bolts. The crankshafts were of the original design with separate three-throw assemblies for each working cylinder, the four cylinder assemblies and the scavenge crank being connected by bobbin pieces which also formed the main bearing journals.

The nature of these failures indicated that they had probably originated through overstrain caused by excessive flexure. Subsequent investigations indicated that resonance could occur within the operating speed range between the unbalanced primary couple originated by the four main cylinders and the fundamental mode of vertical hull vibration. The precise location of the resonant speed depended on the loading of the ship and whether it was operating in deep or shallow water. The flexing at the bobbin-piece flanges had been so severe that ridges had been formed on the bolt bodies.

This problem was first attacked by removing all the bobbin pieces except the one at the forward end of the crankshaft, and reducing the thickness of the flanges by about 15 per cent. At the same time high-tensile steel bolts were fitted and the orientation of the crankshaft elements was altered so that the firing order was changed from 1-3-2-4 to 1-3-4-2, thus placing the cranks of adjacent cylinders at 90° instead of 180°. These changes aimed at reducing the stress on the couplings by reducing the flexural stiffness of the crankshaft. After the alterations had been made the ships operated without further trouble until 1930, a period of about three years. In 1930, however, fatigue cracks were found at the oil holes in the four bobbin-piece journals of both engines, counting from the after end of the engine. An investigation of the torsional vibration characteristics of the installation showed that with the original firing order 1-3-2-4 there had been a powerful sixth order two-node critical speed in the neighbourhood of the maximum running speed, whereas with the modified firing order 1-3-4-2 the peak amplitude of this critical was considerably reduced. It therefore appeared that this torsional vibration critical had been the main cause of the early failures experienced with these engines, and that flexural straining had been a secondary cause.

Now although the change of firing order from 1-3-2-4 to 1-3-4-2 had considerably reduced the amplitude of the torsional vibration critical, it had unfortunately produced a three-fold increase of the magnitude of the unbalanced primary vertical couple. Thus, despite the considerable reduction of torsional vibration stress and the reduced flexural stiffness of the crankshaft, the increased magnitude of the unbalanced vertical couple had been sufficient to cause failure by flexure rather than torsional vibration after a sufficient number of stress reversals.

This case is interesting since it supports a conclusion deduced from observation of many different engine installations that crankshaft failures from torsional vibration generally occur within a short period after the engine has been put into service, whereas failures from over-stressing in flexure usually take longer to develop. The precise interval depends on the magnitude of the over-stress and the time required to complete the number of stress cycles to cause failure.

About the time the first of these large four-cylinder equal-stroke engines was completed, considerable attention was being paid to balancing questions, because it was realized that complete balance of reciprocating and revolving parts would be an important design improvement since it would enable engine speeds to be increased and structural weight reduced without risk of serious vibration trouble from free forces and couples. Furthermore, interest in this subject was stimulated in 1926 by the troubles just described, and by the loss of an important contract mainly as the result of some criticism of engine balance. This problem was solved so effectively that in the same year the engine was selected for powering the quadruple-screw luxury liner "Bermuda", a contract which placed special emphasis on freedom from vibration.

The steps which were taken to achieve a balanced engine appear in retrospect to be fairly obvious and simple, but at the time much thought was given to the problem before a final solution was obtained.

The first step was to re-design the piston assemblies so that the primary inertia force of the lower-piston assembly balanced the primary inertia force of the upper-piston assembly. This was achieved by adopting the now familiar differential stroke arrangement which had been proposed by several patentees prior to 1900 and had been used by Doxford's themselves in a small experimental submarine engine built in 1915. In 1919, when the first full-scale Doxford marine engine was constructed, however, the objections to the use of differential strokes on slow-speed engines were reluctance to provide two types of piston skirt, and the fact that for a given total stroke the overall height of a differential stroke engine is greater than that of an equal stroke engine by an amount equal to the difference of the strokes of the upper and lower pistons. By 1926 the increasing speeds of the engines and the more insistent demand for the greatest possible smoothness presented problems which overshadowed any small manufacturing inconvenience; while by that time the fallacy of criticisms based on allegations of excessive height had been exposed so that it was no longer necessary to seek to minimize overall height to the detriment of other desirable features.

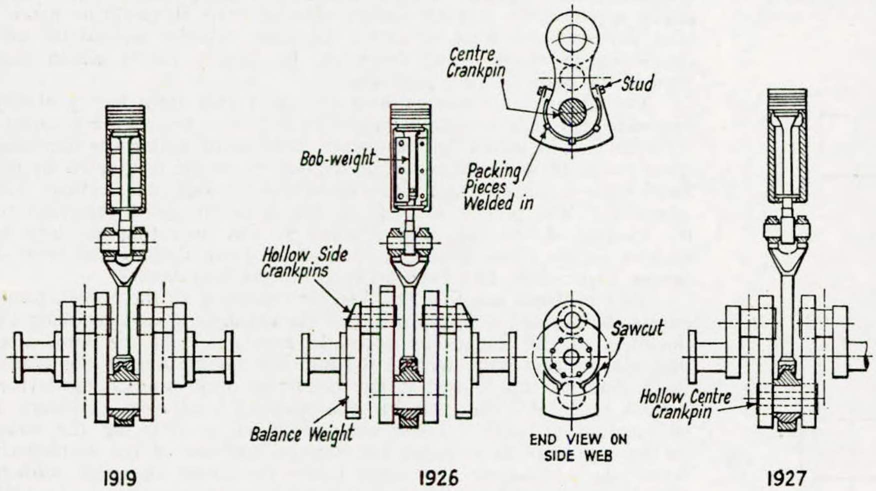

FIG. 4.—*Evolution of balanced engine.*

Primary balance of the reciprocating parts was obtained partly by adjusting the strokes of the upper and lower pistons, and partly by adjusting the weights of the respective reciprocating parts, so that the product of the weight of the upper-piston reciprocating parts and the stroke of the upper piston was equal to the product of the weight of the lower-piston reciprocating parts and the stroke of the lower piston. After some preliminary trials a satisfactory practical compromise was achieved by reducing the upper piston stroke by about 15 per cent. and increasing the lower piston stroke by about the same amount. This enabled the upper piston reciprocating parts to be reduced by about 14 per cent. leaving about 28 per cent. to be added to the weight of the lower piston to achieve final balance. It was found that the required increase in the weight of the lower piston could be obtained very simply by increasing the thickness of the cast-iron piston skirt.

Primary balance of the rotating parts was readily obtained because the differential strokes produced crank-throws which, with the revolving parts of the connecting rods, were so nearly in balance that the only adjustment required was the boring of a hole of the necessary size through the centre crankpin.

The evolution of the balanced engine is shown in Fig. 4, where the left-hand diagram represents the original equal-stroke design produced in 1919, and the right-hand diagram shows the differential-stroke balanced design produced in 1927.

The centre diagram shows the means adopted for balancing the large four-cylinder equal-stroke engines which, as already described, gave some trouble through failure of coupling bolts in 1926.

The rotating parts were balanced by forging extensions of the required size on each of the side crankwebs, the dimensions of these extensions being kept as small as possible by boring a hole through each side crankpin. The stiffening effect of the extensions was minimized by saw cuts at each side which left just sufficient metal to withstand the centrifugal loading.

The reciprocating parts were balanced by attaching a cast-iron bob-weight weighing about 1·5 tons to the lower piston rod. This was the largest mass which could be accommodated inside the piston skirt. With this arrangement the rotating parts were completely balanced while the unbalanced inertia force in the vertical plane due to the reciprocating parts was only 40 per cent. of the amount existing on engines which had no special provision for balancing.

In cases where the crankshafts had been completed, the required crankshaft balance weights were made from steel castings welded to the crankwebs as shown in the small sketch at the top of Fig. 4. The weight of each steel casting was about one ton, and it exerted a centrifugal pull at full speed of about 8·5 tons on the centre crankweb.

Returning to the fully-balanced differential-stroke engine; the balancing of each cylinder individually proved to be an excellent solution, since in multi-cylinder engines where primary balance is obtained by the collective effect of all the cylinders there is still the possibility of vibration due to the inertia forces at the individual cylinders. These may cause local elastic deformations of the engine structure and its surroundings of sufficient magnitude to expose the structure to an appreciable input of destructive energy. Even if this internal unbalance does not produce troublesome external symptoms, it may be a potential cause of flexure and eventual failure of the crankshaft and framing. The differential-stroke method of balancing the opposed-piston engine eliminated this possibility by providing complete external and internal balance of primary forces.

The complete balancing problem was not yet solved, however, for there remained the questions of eliminating secondary forces originated by the reciprocating parts of the main cylinders, and balancing the running gear of the engine-driven scavenge pump.

The crank sequence of all four-cylinder engines built prior to 1926 was 1-3-2-4, this being the sequence commonly used for single-piston two-stroke engines because it yielded the smallest unbalanced primary inertia couple for the engine as a whole. Now in all conventional reciprocating engines the obliquity of the connecting rods produces a pulsating inertia force at each cylinder with a frequency

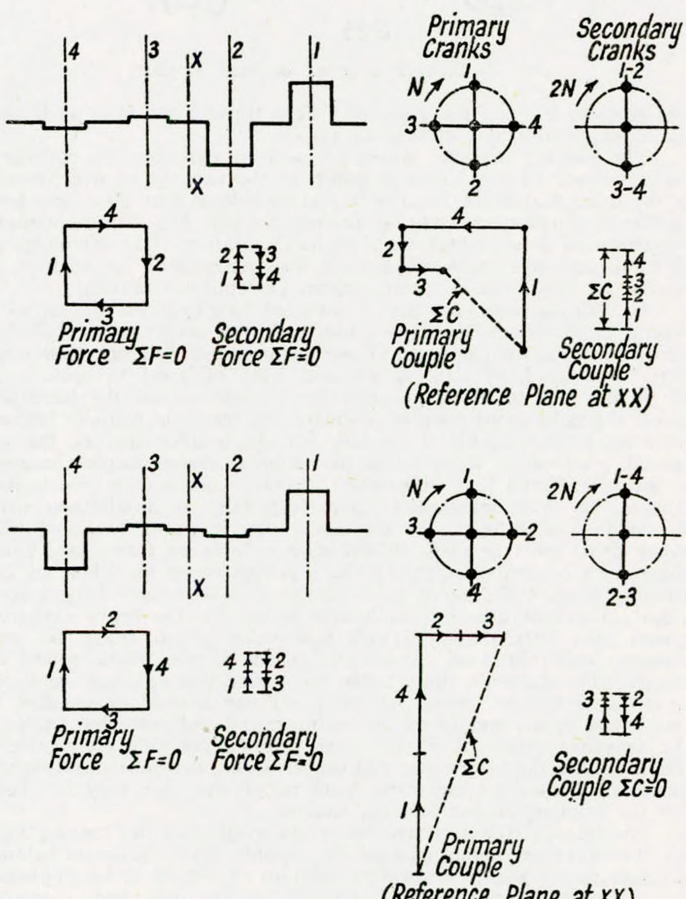

FIG. 5.—*Balancing diagrams.*

equal to twice the revolutions per minute of the crankshaft. Thus in an opposed-piston engine where the side and centre crankpins are at 180° the secondary force due to the lower piston is in phase with the secondary force due to the upper piston, i.e. these two forces must be added to obtain the total secondary force for each cylinder.

Furthermore, in a four-cylinder engine with crank sequence 1-3-2-4, although the secondary forces for the individual cylinders cancel so that there is no residual secondary force for the engine as a whole, the secondary couples do not cancel but leave a residual longitudinal couple which tends to rock the engine about its centre of gravity with a frequency equal to twice the revolutions per minute. Although the magnitudes of secondary forces and couples are considerably smaller than the magnitudes of the corresponding primary forces and couples, the possibility of troublesome vibration due to resonance with a natural frequency of the hull cannot be disregarded. Moreover, owing to their higher impulse frequency there is usually more likelihood of secondary effects producing troublesome local vibration through finding some point of sympathetic response in the engine or hull structure.

For these reasons, the elimination of secondary effects was considered to be just as important as the elimination of primary effects. This was accomplished by changing the crank sequence to 1-3-4-2, a sequence which had already been used for another purpose, namely, to reduce the flexural stiffness of the crankshaft as previously mentioned.

With this sequence the residual secondary forces at cylinders 1 and 4 are balanced by those at cylinders 2 and 3, while the secondary couple originated by the forces at cylinders 1 and 2 is balanced by the couple originated by the forces at cylinders 3 and 4. The change of crank sequence from 1-3-2-4 to 1-3-4-2 did not affect primary balance because primary inertia forces were balanced at each cylinder individually.

It should be noted, however, that this change cannot be used for producing secondary balance of a single-piston engine because with sequence 1-3-4-2 the primary inertia couple for the engine as a whole would be about twice the value with sequence 1-3-2-4.

In Fig. 5 the balancing characteristics of a four-cylinder single-piston engine are exhibited by vector diagrams in accordance with conventional graphical methods of solving engine balancing problems. The primary crank diagram is simply the crank sequence diagram for the engine, while the secondary crank diagram, which gives the directions of the secondary force and couple vectors, is obtained by doubling all the crank angles of the primary diagram since each secondary vector rotates at twice the speed of the corresponding primary vector.

The vector diagrams in Fig. 5 show that primary and secondary forces are balanced for both crank sequences. With sequence 1-3-2-4, however, there are unbalanced primary and secondary couples; and with sequence 1-3-4-2 there is no secondary couple, but the magnitude of the primary couple is doubled.

The introduction of the differential strokes and the change to crank sequence 1-3-4-2 provided a four-cylinder opposed-piston arrangement with complete primary and secondary balance of the main cylinders, the primary balance being achieved both externally and internally. There remained, however, an unbalanced primary and secondary force due to the moving parts of the centrally situated crank-driven scavenge pump. As already explained, the unbalance due to the scavenge pump was reduced to a small residual secondary force acting in the vertical plane and a residual primary force acting in the horizontal plane by fitting balance weights to the scavenge crankwebs and by making drastic reductions in the weight of the reciprocating parts of the pump.

These changes produced such a remarkable improvement in the running characteristics of the engine that all Doxford engines built after 1926 were of the balanced differential stroke type. A rather striking demonstration of the effectiveness of the balancing principles employed was carried out on the test-bed shortly after the balanced engine had been introduced. A balanced engine having a normal rating of 2,400 b.h.p. at 90 r.p.m. was operated at 140 r.p.m. and 4,000 b.h.p. with all holding-down bolts removed. The engine was photographed while running at this speed, and although an exposure of 3½ minutes was used, the definition of the stationary parts of the engine was absolutely sharp without the slightest blurring even at the handrails on the topmost platforms about 25ft. above the test-pit floor, as shown in Fig. 6. The balanced engines gave the impression of spinning round without apparent effort, whereas there was always

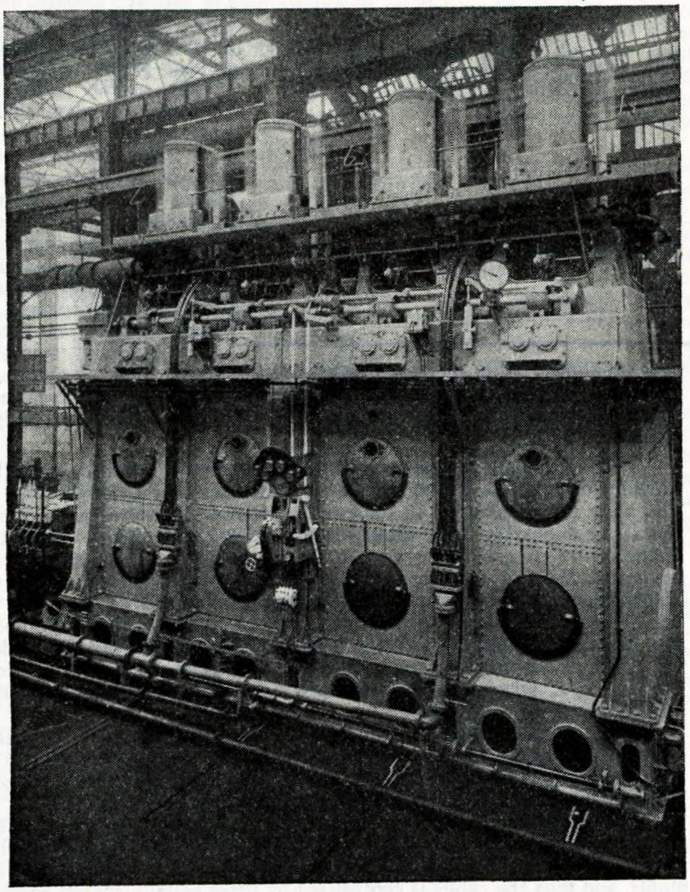

FIG. 6.—*Balanced Doxford engine.*

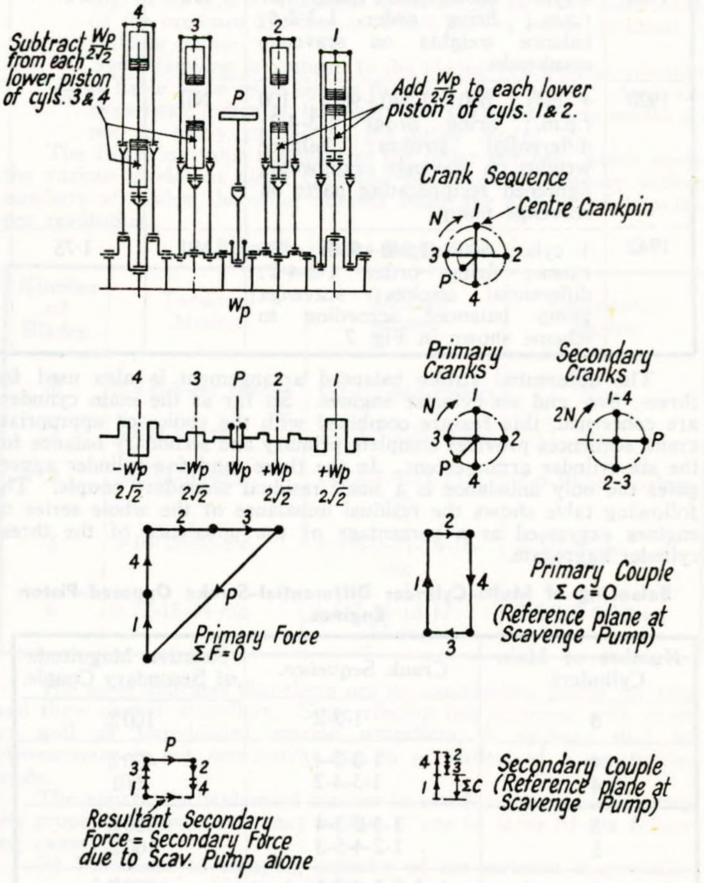

FIG. 7.—*Scheme for balancing scavenge pump of a four-cylinder engine.*

(2) Variations in the dimensions of the individual blades which, though small in themselves, are sufficient to produce a tuned system capable of responding vigorously to the external excitations.

(3) The existence of residual stresses in the blades which produce a stress pattern capable of tuning the system to an unfavourable natural frequency.

Recent experience in carrying out vibration tests on the individual vanes of multi-vaned high-speed supercharger impellers has indicated that such variations can have an important influence on the response of the system. In the course of these tests, which were carried out on a large number of impellers, two or three cases were found where an individual vane continued to vibrate with a loud ringing tone for an appreciable time after the excitation had been removed, whereas in the majority of cases the vibration ceased almost as soon as the excitation was removed.

Transverse Vibration of Engine Framing.

The principal working loads on the frame of an opposed-piston engine are the side thrusts on the side and centre crosshead guides and the equal and opposed side thrusts on the main bearings. These produce a torque reaction couple equal in magnitude but acting in the opposite sense to the driving torque of the engine. If the crankshaft was prevented from rotating while the engine framing and cylinders were free to turn about the crankshaft axis, the torque reaction couple would tend to rotate the structure in the opposite direction to that of the crankshaft.

Since the driving torque contains a whole series of pulsating components, these fluctuating loads are also impressed on the guide faces so that in addition to the steady torque reaction couple corresponding to the mean driving torque, there is a whole series of pulsating torques tending to produce rocking motions of the engine on its seating. Now although the actual magnitudes of the pulsating pressures on the guides are quite small, there is nevertheless a possibility of troublesome vibration from this cause should sympathetic response or resonance be established between the frequency of one of the significant components of the guide pressure and one of the significant natural frequencies of the structure.

This type of excitation, for example, is capable of producing torsional vibration of the hull with one or more nodes, transverse rocking vibrations of the engine on its seating, and local vibration due to resonance with parts of the engine structure or its surroundings.

In multi-cylinder engines with equally spaced cranks, many of the harmonic components of the torque reaction couple acting at each cylinder cancel out amongst the various cylinders leaving no external resultant, provided it can be assumed that the engine framing is sufficiently rigid to prevent elastic deflections of appreciable magnitude between one cylinder and another.

Under these conditions the only unbalanced components for two-stroke cycle engines are those which are integral multiples of the number of cylinders. Thus in a four-cylinder two-stroke cycle engine the unbalanced components are the 4th, 8th, 12th, etc. orders having impulse frequencies equal to 4, 8, 12, etc. impulses per revolution of the crankshaft. The frequency of even the lowest of these impulse frequencies is considerably higher than that due to primary and secondary mass unbalance so that, broadly speaking, resonance with the fundamental modes of hull vibration is unlikely.

The impulses do, however, search the engine structure and its surroundings for places of sympathetic response, and if such a region is found, troublesome local vibration may be experienced. This is liable to occur if the engine structure or its seating are unduly flexible, due, for example, to weak design or poor fabrication. In this connection, since cancellation of the lower order components of the torque reaction couple depends on adequate frame rigidity, any weakness in the engine structure will tend to aggravate the picture by producing externally unbalanced lower order components, thus introducing the possibility of resonance with the fundamental natural frequencies of the hull.

The natural frequency of transverse rocking vibrations of the engine on its seating depends greatly on the elasticity of the seating itself. Normally this frequency is sufficiently far above the fundamental impulse frequency to avoid resonance effects in the operating range of speeds; while the higher order impulses are usually too weak to cause serious disturbances even in the event of resonant conditions becoming established in the operating range. If the engine seating is too flexible, however, the natural frequency may be reduced to a value which produces resonance with the fundamental impulse frequency within the running range.

An interesting account of the troubles experienced with engine seatings which had insufficient stiffness is given in a paper read in 1931[10].

The advantages of adequate structural rigidity in both the engine framing and in the engine seating were fully appreciated from the start of Doxford engine development. The engine was supported on a double bottom of adequate depth with the flat-bottomed bed-plate bolted directly to specially thickened tank-top plates. Intercostals were provided in way of the holding down bolts and care was taken to ensure that the stiffening of the hull structure in way of the engine did not end abruptly at each end of the bed-plate.

The evolution of the main structure of the Doxford engine is shown in Fig. 9. In the first engines to be constructed, in 1919, the framing was given considerable transverse rigidity by providing four separate cast-iron columns for each of the four cylinder assemblies, and by making the cast-iron bed-plate rather wide. The actual ratio of engine height to bed-plate width was 2·2. The columns were not provided with transverse cross-bracing, however, because it was considered desirable to have free access to the crankshaft in the event of trouble.

By 1924 sufficient service experience had been obtained to indicate that the engine framing possessed more than adequate rigidity, and furthermore the special provision for access to the crankshaft had never in fact been utilized.

A substantial reduction of weight was therefore obtained by reducing the number of main columns and by increasing the ratio of engine height to bed-plate width to 2·5. The number of columns used on a four-cylinder engine was reduced from sixteen to twelve, and again no transverse cross-bracing was provided. At first the modified framework appeared to have adequate rigidity but, as speeds were increased, noticeable transverse vibration began to appear, and in 1926 substantial cross-bracing was provided as shown in Fig. 9.

In 1933 the cast-iron construction was abandoned in favour of

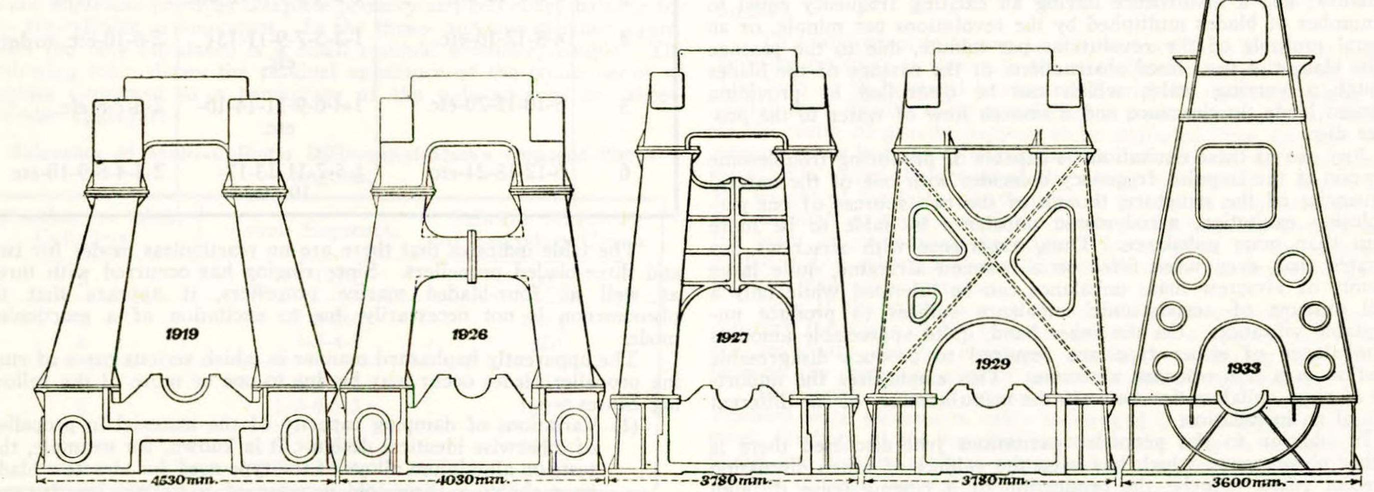

FIG. 9.—*Engine framing.*

the now familiar fabricated steel construction, in which the ratio of engine height to bed-plate width is about 2·7.

Throughout the development period the only serious trouble due to transverse vibration occurred in 1929 on a twin-screw installation with four-cylinder engines having cast-iron columns of the heavily cross-braced type shown in Fig. 9. In this case the vibration was of the "tuning fork" type in which the two engines vibrated like the prongs of a tuning fork.

Attempts to cure this trouble by connecting the top of each engine to the adjacent side of the ship by strong tie beams were unsuccessful, and merely resulted in broken tie-beam attachments accompanied by damage to the side plating. The real cure was effected by introducing strong beams to connect the tops of two engine entablatures together, using carefully fitted bolts for securing the ends of the beams to the engine structures. This is now a well-known device for dealing with tuning-fork vibration, and in the case under discussion was quite effective so far as this type of vibration was concerned.

Unfortunately, in this particular case the main trouble was more deeply rooted, and after the transverse vibration had been cured, serious failures of the shaft system developed due to torsional vibration, a subject which is discussed later. Nevertheless, the experimental work on transverse vibration did indicate the effectiveness of tie-beams between the two engines, and these are now a standard feature of twin-screw installations.

There is no simple method for neutralising torque reaction couples in reciprocating engines, though various schemes have been proposed from time to time which aim at producing a considerable reduction in the magnitude of the disturbance. The problem is particularly difficult in single-screw installations and up to the present time engine builders have preferred to rely on the provision of adequate structure rigidity in the engine framing and in the seating rather than the introduction of more or less complicated compensating mechanisms.

As far back as 1897 Macalpine[2] put forward a series of suggested mechanisms comprising various arrangements of cams, bob-weights and links for neutralising vibratory disturbances arising from fundamental and higher order impulses originated by moving parts, and torque reaction couples whether due to inertia or gas pressure. His proposals, though sound enough in theory, were not so attractive mechanically on account of the large number of moving parts.

Later, in 1927, Professor F. M. Lewis[8] suggested the use of gear-driven counterweights. This scheme employed a pair of rotating counterweights mounted on layshafts symmetrically disposed on either side of the centre line of the engine. The layshafts were driven by gearing from the crankshaft, the counterweights being set at 180° phase to one another, so that the centrifugal forces originated by their rotation produced a pure couple acting on the engine frame. By adjusting the weights so that this couple was in correct phase relationship with the crankshaft, a considerable reduction of the torque reaction could be obtained.

It should be noted, however, that the magnitude of the couple produced by the counterweights varied as the square of the r.p.m., whereas the torque reaction couple varied to a much smaller extent with r.p.m. Hence, correct balance could only be obtained at one particular speed. The device would therefore only be fully effective on constant speed machines, and even then the balance would not remain absolutely complete, due to variations in operating conditions in the engine cylinders. Nevertheless, there would always be a useful reduction in the magnitude of the couple tending to rock the engine.

Comparatively small weights would be required due to the relatively high rotating speed. Thus, in a four-cylinder two-stroke cycle engine where the lowest unbalanced order of the torque reaction couple was the fourth, i.e. four impulses per revolution, the counterweights would run at four times crankshaft speed. From the point of view of obtaining the best running conditions at any given speed, there would appear to be advantages in providing means for adjusting the out-of-balance of the weights and their phasing while the engine was running. It would also appear advantageous to run the two layshafts along the full length of the engine with separate pairs of counterweights at each cylinder. In this way the couple at each cylinder would be balanced individually and this would provide internal as well as external balance, thus relieving the engine structure of a considerable part of the dynamic load. Mechanically, however, this arrangement might prove difficult, especially if it was desired to provide means for varying the adjustments of the several weights while the engine was running.

No record has been found of any actual installation in which this scheme was employed.

In twin-screw installations it is necessary to distinguish between "tuning-fork" vibration in which one engine vibrates against the other, and rocking vibration in which the two engines vibrate against the hull with a transverse rocking motion. As already mentioned, "tuning-fork" vibration can be controlled by the simple expedient of tying the tops of the two engines together, but this device is not effective against transverse rocking vibration.

Various schemes have been put forward from time to time for controlling vibration due to torque reaction in twin installations by using the couples originated by one engine to neutralise those originated by the other.

In principle, this idea is extremely simple since it requires no more than some means for connecting the two crankshafts so that they rotate in opposite directions at the same speed and in the correct phase relationship for neutralising the torque reaction couples. This could be achieved, for example, by a simple gear train between the two shafts, and then the only source of disturbance would be variations in the operating characteristics in individual cylinders.

Several ingenious synchronising arrangements for twin-screw oil-engine installations are discussed in an article by D. W. Rudorff[15]. These include rigid mechanical coupling by means of a cross shaft connected by bevel gears to the two main shafts. A planetary bevel gear assembly of the type used for the differential gears of motor-cars is inserted in the cross shaft and so long as the outer casing of the differential is free to rotate the effect of starting one engine with the other stationary is to cause the differential casing to rotate at half the speed of the active crankshaft; while if one engine is running ahead and the other is running astern at the same r.p.m., i.e. both sections of the cross shaft running in the same direction in a twin-screw arrangement with right- and left-hand propellers, the differential casing will also rotate in the same direction and at the same speed as the cross shaft. Under normal cruising conditions the differential casing is locked so that the two crankshafts are forced to rotate in opposite directions at the same speed. Furthermore, by providing means for varying the position at which the casing is locked, for example by a worm and wheel, the phasing of one crankshaft relative to the other can be varied.

In this way a gear is provided which synchronises the crankshafts for both speed and angular relationship. At the same time, by providing means for unlocking the differential casing, full flexibility in starting and manœuvring can be obtained. Interlocking gear or clutches would, of course, be desirable to prevent damage through attempts to carry out manœuvres with the casing locked.

An alternative method is synchronisation by electrical means, an arrangement which has been used successfully on a number of ships.

In the electrical system, an alternator is mounted on each shaft and the two alternators are connected in parallel so that they maintain both the r.p.m. and the angular relationship of the two crankshafts constant. Normally the alternators do not develop any power, but if the speed of one engine tends to drop, power is developed by the alternator on the leading engine, and is transmitted to the alternator on the lagging engine so that it becomes a motor tending to assist the lagging engine to maintain the set speed.

According to Rudorff the mechanical gearing or the alternators in these direct-coupling schemes must be designed to carry about one-half the horse power of one engine and this represents a considerable addition to the cost and weight of the installation. Furthermore, they represent an additional source of loss which reflects on overall economical performance.

These criticisms are overcome in a third scheme in which the direct mechanical coupling arrangement is replaced by a similar mechanism but of much lighter construction. In this scheme the cross shaft is used for regulating the speeds and relative angular settings of the two crankshafts by connecting its differential casing to a pilot valve controlling the flow of operating fluid to a relay cylinder. The piston of the relay cylinder is connected to the fuel control valve of one engine so that if there is any tendency for the engines to get out of step, corrective action is automatically applied. In an alternative arrangement one of the engines drives a variable pitch propeller and the relay cylinder controls the pitch of this propeller in accordance with requirements.

An alternative method of synchronisation based on the same fundamental principle is described by de Vos[13]. In this arrangement the mechanical system described by Rudorff is replaced by a hydraulic system in which each engine drives a rotary sleeve valve, the two sleeves being housed in tandem in a common casing. The two sleeves rotate in the same direction and are provided with ports which ensure that equilibrium conditions are only maintained when the two crankshafts are running at the same speed and in a predetermined angular relationship. Any deviation causes pressure oil to be supplied to a servo-cylinder on one of the engines which restores the equilibrium condition by an appropriate adjustment of the fuel supply. Successful results were obtained with this system on a single-screw installation where the propeller was driven through simple reduction gearing by two six-cylinder two-stroke cycle engines each developing 3,300 s.h.p. at 215 r.p.m., each engine being connected to its associated pinion by a Vulcan hydraulic coupling.

With all types of synchronising gear there is a tendency for the inboard engine to be seriously overloaded when the ship is turning, but this difficulty could no doubt be overcome by suitable elaboration of the equipment.

An interesting twin-screw arrangement is shown in Fig. 10, which is a cross-section of the Sun-Doxford opposed-piston engine built in

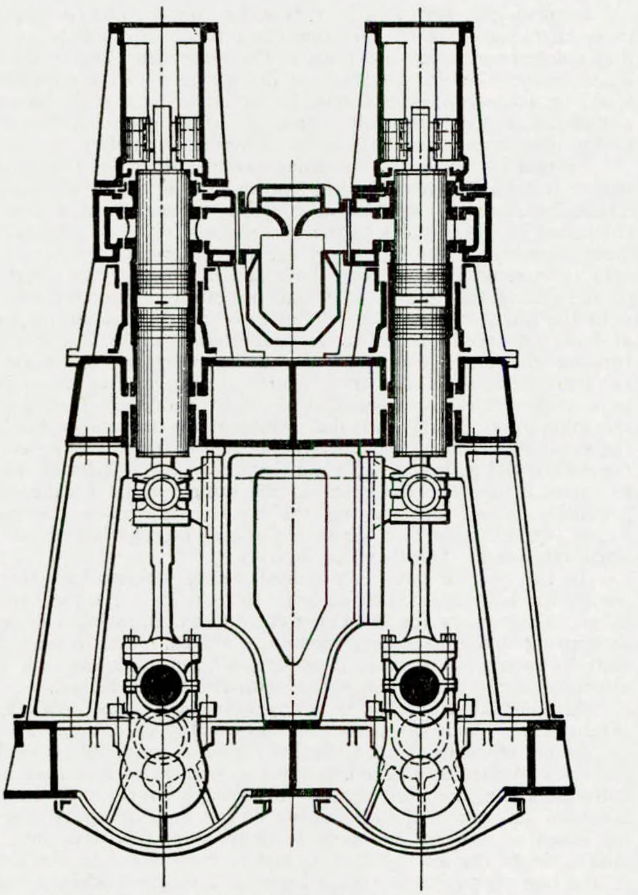

FIG. 10.—*Sun-Doxford engine in m.y. Sialia.*

1925 for Henry Ford's yacht *Sialia*. The engine consists of two rows of four cylinders built on a common bedplate, with each row of cylinders driving a separate propeller. The cylinders are 13in. bore×(22in.+17in.) stroke, and the designed rating is 750 b.h.p. per shaft at 200 r.p.m. with a brake mean effective pressure of about 72lb. per sq. in. The columns and entablature are of cast aluminium alloy. Although the engine is of the differential stroke type, the crank sequence is 1-3-2-4, which implies that there is a small residual secondary couple for each row of four cylinders. It has not been possible to determine whether a synchronising mechanism is used between the two crankshafts.

The *Sialia* has a rather interesting history. Built in 1913 by Pusey and Jones, Wilmington, Delaware, as a steam yacht, she was refitted with Sun-Doxford oil engines in 1925. Shortly after this conversion, Ford decided that she was not quite long enough. The oil engines were therefore removed and her length was increased from about 200 to 220ft. by cutting the hull in two and inserting a new midship section. The cost of this operation, which was carried out with complete success, was about £130,000, or about £10,000 more than the price which Ford had paid for her in the first place.

Subsequently she was sold by Ford, and after passing through the hands of three different private owners, was purchased by the Clipper Line in 1938 and renamed the *Yankee Clipper*. In 1941 she was taken over by the U.S. Navy as a patrol vessel, and renamed the *Coral*. This novel Sun-Doxford engine is still operating satisfactorily.

An unusual method of synchronising twin-screw engines is described by Woodman in Patent No. 502,747 of 1936. The inventor claims that the two petrol engines in a twin-screw motor-boat installation can be made to run synchronously even under considerable load variations by connecting the ignition circuits so that the distributor on one engine serves the ignition plugs on the other. Thus, if one engine slows down, the ignition timing on the other engine is automatically retarded and so its speed is also reduced. A change-over switch is provided to enable the engines to be operated independently until they reach the same speed, and a synchroscope may be used for indicating when the two engines are in phase.

Alternatively it is suggested that the correct moment for synchronising can be judged by listening to the beat of the engines.

It is reported that some years ago promising results were obtained with this scheme by Gar Wood, the American racing motor-boat enthusiast.

In direct-coupled installations serious vibration due to torque reaction is comparatively rare and when it does occur is usually confined to areas of the engine or hull structure where local resonance conditions have been established. Such cases usually yield readily to treatment by local stiffening. It should be noted, however, that transverse and vertical vibration of the engine and hull structures can originate from torsional vibration of the shaft system. Indeed, the only serious case of transverse vibration encountered during the development of the Doxford opposed-piston engine occurred in a twin-screw installation where there was a severe torsional critical at the normal running speed.

In geared installations, on the other hand, the torque variations originated both by the engine and by the propeller are transmitted directly through the gear casing to the hull.

In examples where the input and output shafts at the gearbox rotate in opposite directions the torque reaction transmitted by the gearbox to the hull is the sum of the input and output torques, whereas if the input and output shafts rotate in the same direction, it is the difference.

Torsional Vibration.

Although the full significance of torsional vibration was not fully appreciated until much later in the development of the Doxford opposed-piston engine, this troublesome phenomenon cast a slight shadow over the performance of the first Doxford motor-ship during its maiden voyage in 1921.

The camshaft drive on this engine was by helical gears and vertical shafts situated at the forward or free end of the crankshaft, mainly for accessibility and because the main helical wheel could be made in one piece. During the maiden voyage this drive vibrated considerably when the engine was running in the region of 70 to 72 r.p.m. On either side of this speed the drive was comparatively quiet. Nevertheless considerable wear occurred in the teeth of the helical gears, and this was attributed to movement of the forward crankshaft journal to which the main helical wheel for the camshaft drive was rigidly bolted. The wear was sufficient to necessitate replacement of the mating pinion after about 14,000 miles.

In an attempt to reduce the vibration the propeller shafting was disconnected at the after coupling and the tailshaft was rolled round one bolt pitch before recoupling. This moved the propeller blades to a different angular position relative to the engine cranks, but the change had no noticeable effect.

The immediate trouble was overcome by removing the main helical wheel from the crankshaft journal and mounting it on a separate layshaft. The forward end of this layshaft was carried in a journal bearing supported on a bracket bolted to the forward end of the main bedplate, while the after end was formed into a sphere which was carried in a spherical seating inserted at the end of a hole bored for the purpose in the forward crankshaft journal. The wheel was driven by a special bolt secured in one of the existing coupling bolt holes in the forward journal flange. The driving bolt had a spherical head which engaged a sliding block accommodated in a slot cut in the body of the wheel. This arrangement allowed the crankshaft journal to move freely without transmitting the movement to the camshaft driving gear.

At the first opportunity the camshaft drive was moved to the after end of the crankshaft and the helical gears were replaced by a bevel gear drive. This rearrangement cured the trouble and had the additional advantage that in the event of damage to the forward section of the crankshaft the camshaft drive would still be available for working the remaining cylinders.

Torsional vibration calculations, made several years later, provided the explanation for the serious vibration noticed in the region of 70 r.p.m.

Thus, referring to Case *A* in Fig. 14, it is seen that in this installation there was an 8th order two-node critical speed at 70 r.p.m. which, though not of sufficient amplitude to endanger the crankshaft, was sufficient to explain the trouble experienced with a camshaft drive located at the free end of the crankshaft where the vibratory amplitude for the two-node vibration is greatest. The removal of the camshaft drive to the after end of the engine placed it at a point near the crankshaft node where the vibratory amplitude at two-node criticals is relatively small.

In 1928, after about sixty ships had been put into service, the first serious crankshaft trouble was encountered. This occurred in two twin-screw vessels fitted with four-cylinder engines of the balanced type, having cylinders 600 mm. bore $\times$(980+1,340) mm. stroke, each engine designed to develop 3,300 b.h.p. at 98 r.p.m. As already mentioned in the section on transverse vibration, the first symptom that some serious vibratory condition existed on these ships was the presence of objectionable transverse vibration of the "tuning-fork" type of the engines on their seatings. In addition there was considerable clattering of the running gear of the upper piston and the camshaft driving gear when the engines were operated in the neighbourhood of the designed speed of 98 r.p.m. Below 97 r.p.m. and above 99 r.p.m. the engines operated quietly.

Both vessels completed the first outward voyages to Australia at an engine speed of 92 r.p.m. without mishap, vessel *A* preceding vessel *B* by two months.

The first homeward voyages were commenced with the engines operating at about 96 r.p.m., and although vessel *A* showed no symptoms of serious trouble, the forward side connecting rod of No. 2 cylinder on the starboard engine snapped without warning within six hours of commencing the voyage, at a point about one-third of the length of the rod from the crosshead end.

Vessel *B* on the other hand reported extremely rough running, the symptoms being clattering of the upper piston assemblies, rumblings of the camshaft driving gear, and general frame vibration. After 22 days running at about 96 r.p.m. the forward side rod of No. 2 cylinder on the port engine snapped without warning at substantially the same position as the failure in vessel *A*.

Fortunately no serious damage was done to the engine structure and after the following changes had been made, both vessels were able to complete their voyages with both engines in operation. In both cases the three connecting rods were removed from the cylinder concerned, No. 2; the lower piston and its crosshead were blocked up from the guide plate so that the piston sealed the scavenge ports; and the top guide and upper piston were replaced, the piston being lowered far enough into the cylinder to seal the exhaust ports.

Although physical tests of the connecting rod material revealed no peculiarities to account for the failures, and the calculated stress range at the point of fracture was well within permissible limits, new sets of rod with a 50 per cent. increase of sectional area were fitted to each vessel before the second outward voyage to Australia.

Since nothing unusual had been revealed by metallurgical examination and stress calculations, and bearing in mind the rather remarkable coincidence that both failures had occurred at the same point in rods occupying the same position in the engines, it was suspected that some obscure dynamic effect was responsible for the failures. The increased sectional area was therefore obtained by making an appreciable increase in the diameter of the rod body and boring a large hole right through the shank, with the object of increasing the natural frequency of the lateral mode of vibration of the rod to avoid any resonant condition which might have existed with the original rods. In addition, instructions were given for a careful watch to be kept for any unusual engine vibration.

The original connecting rods were fitted to engines of similar bore and stroke in 1930, and are still giving satisfactory service, indicating that there was no inherent weakness in the original design.

During the second voyages to Australia and back to the United Kingdom, vessel *A* reported comparatively smooth operation at 98 r.p.m., but slight vibration at 86 r.p.m., observed mainly as a rattle in the camshaft driving gear.

Vessel *B*, on the other hand, reported extremely rough running in the neighbourhood of 98 r.p.m., the principal symptoms being fuel pressure variations of the order of 1,000lb. per sq. in. and general engine vibration, particularly in the neighbourhood of No. 4 cylinder at the after end of the engine. There was also considerable vibration in the bridge structure of the ship at this speed.

During the voyage the coupling bolts in the vertical shaft of the camshaft drive on the port engine sheared twice. In general it was considered that the vibration had increased considerably after the installation of the modified side connecting rods.

When vessel *B* reached the United Kingdom, the engine builders took the opportunity to make some coastal trips with the object of investigating the vibration which had been experienced at 98 r.p.m. Various expedients were tried, including removal of all bolts from the cross-ties between the engine columns, to determine if this had any influence, good or bad, on the vibrations of the engine framing; alteration of the phasing of the propeller blades relative to the engine cranks by rolling the propeller shaft through one coupling bolt pitch; and fitting an intermediate bearing on the vertical shaft of the camshaft drive. None of these changes had any appreciable effect.

Finally, as a last resource, when the vessel was in Hamburg, a Geiger torsiograph was borrowed from Messrs. Blohm & Voss, mainly with the object of proving that the trouble was not due to torsional vibration. Torsiograph records were obtained from the propeller shafts of both port and starboard engines at a point about 22ft. aft of the centre line of the engine flywheel, during the passage to London on the 11th December, 1929. These records are shown in Fig. 11, the outstanding features being a 4th order critical speed at 56 r.p.m., an 8th order critical at 86 r.p.m. and a 7th order critical at 98 r.p.m.

The 4th order critical corresponded to a natural frequency of 224 cycles per minute, while the 8th and 7th order criticals corresponded to a frequency of about 687 cycles per minute. The measured amplitude of the 7th order vibration at 98 r.p.m. was $\pm 0{\cdot}7°$ at the measuring point, or $\pm 0{\cdot}75°$ when referred to the free end of the crankshaft; the corresponding movements of the crankpins of No. 1 cylinder being about $\pm 0{\cdot}25$in. for the side and $\pm 0{\cdot}35$in. for the centre crankpins.

A further point of interest is that an 11th order critical can be discerned at 62 r.p.m. on the port engine record only, the frequency being 682 cycles per minute.

Although in 1928 the torsional vibration characteristics of machinery installations were not so closely studied as they are to-day, it was Doxford practice to calculate the one- and two-node natural frequencies of each engine-propeller system. The calculated frequencies for the installations under discussion were 224 and 686 cycles per minute for the one- and two-node modes respectively. It was therefore rather pleasing to note that the Geiger records showed remarkable agreement with these calculated values, the 4th order at 56 r.p.m. confirming the one-node value of 224 cycles per minute, and the 11th, 8th and 7th orders at 62, 86 and 98 r.p.m. respectively confirming the two-node value of 686 cycles per minute.

The large amplitude recorded for the 7th order two-node critical was, however, very puzzling because it was generally believed at that time that serious torsional resonance could only occur at speeds where the principal engine impulse frequency or an integral multiple thereof coincided with the natural frequency of one of the principal modes of vibration.

Thus, in a four-cylinder two-stroke cycle engine, where the fundamental engine impulse frequency is four impulses per revolution, the ideas held at that time suggested that trouble was only likely to occur at 224/4, 224/8, 224/12, etc., r.p.m., and at 686/4, 686/8, 686/12, etc. r.p.m., i.e. at 56, 28, 19, etc. r.p.m. and 172, 86, 56, etc. r.p.m. For this reason the possibility of a severe 7th order torsional vibration critical had been discounted, until the torsiograph records shown in Fig. 11 showed without doubt that such a condition could and did exist.

While these investigations were in progress, vessel *A* had started on the third outward voyage to Australia at 98 r.p.m., happy in the belief that whatever might be troubling vessel *B* at that speed did not appear to concern vessel *A*. Her engines appeared to run quite quietly at 98 r.p.m. though there was a slight disturbance in the neighbourhood of 86 r.p.m.

Nevertheless, it was noticed during this voyage that the engines were not running quite so smoothly as before, and on arriving in Australia criss-cross torsional fatigue cracks were discovered originating from the oil holes in the side crankpins of both engines. The cracked pins were those nearest to the crankshaft node for the two-node mode of torsional vibration, which appeared to confirm that the trouble was due to the severe 7th order critical speed disclosed on the torsiograph records obtained from vessel *B*.

Vessel *B* had a fortunate escape since from the beginning there had been sufficient roughness when running at 98 r.p.m. to cause the engineers to avoid running continuously in the noisy region.

This case is an interesting example of the danger of accepting apparently smooth running conditions as a guarantee of freedom from severe torsional vibration. It also illustrates the thoroughness with which vibration energy searches an elastic structure for points of sympathetic response. There is little doubt that the failures of the side connecting rods on both vessels during their earlier voyages were due to sympathetic response of the rods to the impulses transmitted to them by torsional vibration of the crankshaft. When the original rods were replaced by a stiffer design, which destroyed this sympathetic response thus eliminating one avenue for dissipation of vibrational energy, the load on the crankshaft itself was correspondingly increased and resulted in failure of the shafts on vessel *A*.

A satisfactory theoretical explanation of the rod failures was never obtained, but calculations of the natural frequency of transverse or whipping vibration of the rod, considered as a beam supported at each end, gave values between 720 and 900 cycles per minute

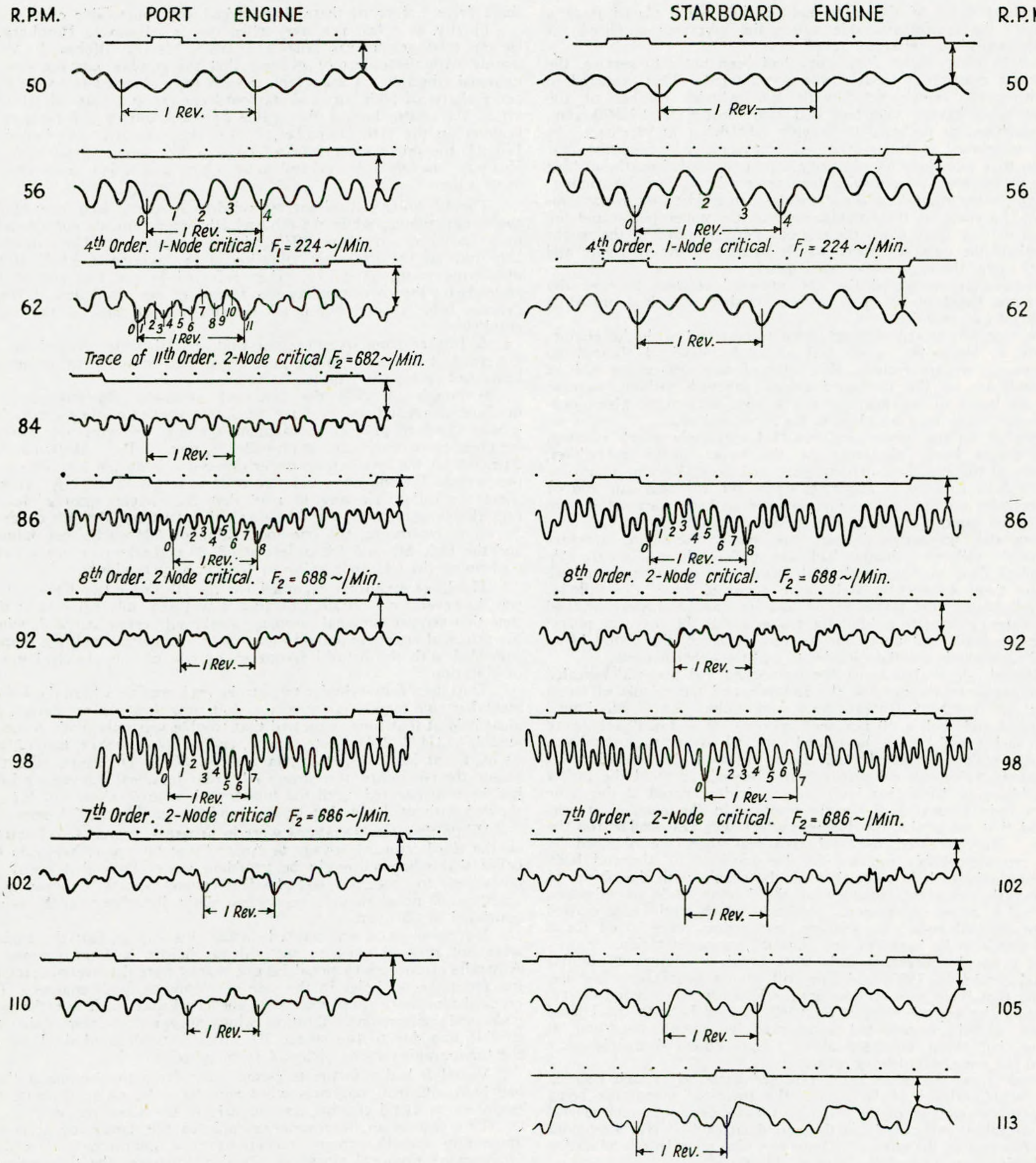

FIG. 11.—*Torsiograph records from propeller shafts of twin-screw motor ship.*

according to the assumptions made in performing the calculation. A practical determination of the frequency, in which a bare rod was struck with a heavy hammer while resting in a horizontal position on the ground, gave a value of about 1,000 cycles per minute. It is therefore conceivable that under operating conditions, bearing in mind the additional flexibility imparted to the system by the crankthrows at the large end of the rod and the engine structure at the small end, the modifying influence of the weight of the upper piston assembly, and the working loads imposed on the rod, the natural frequency of the particular rods which failed was detuned to 686 cycles per minute, i.e. to a frequency which coincided with the natural frequency of the crankshaft system. This coincidence was, of course, as remarkable as it was unfortunate, though in this connection it should be borne in mind that the vibrational energy in seeking an outlet had eight different rods from which to choose. It is probable that each one of the eight was tuned to a slightly different frequency according to the operating conditions and crankshaft flexibility in its neighbourhood. Thus, it only required one out of the eight rods to have a tuning which approached that of the crankshaft system to produce vigorous response.

These experiences had the effect of initiating a thorough study of torsional vibration. The first practical step was to carry out a complete investigation of the torsional vibration characteristics of vessel *B* during its third outward voyage to Australia. These records were obtained during the passage from London to Port Said in December, 1929, the work being considerably assisted through the courtesy of Messrs. W. H. Allen Sons & Co., Ltd. who not only provided the instruments used during the investigation, but also the services of a member of their engineering staff, Mr. L. C. Leigh, who had considerable experience in the operation of torsiograph equipment and in interpreting torsiograph records.

As a result of this investigation it was decided to replace the ten-ton flywheels fitted at the after end of these engines by small turning wheels weighing about one ton. This alteration raised the two-node frequency to about 950 cycles per minute, so that the running speed was located between the 9th order critical at 106 r.p.m. and the 11th order critical at 86 r.p.m. Subsequent trials showed that the stresses at these two criticals were well below the permissible value.

As soon as the full significance of torsional vibration was realised, all existing installations were re-examined and in many cases theoretical studies were followed by torsiograph investigations under service conditions.

This work resulted in a substantial improvement of the torsional vibration characteristics of a number of installations built prior to 1928, in most cases by the simple expedient of making a substantial reduction in the weight of the engine flywheels.

Fig. 12 illustrates some of the fundamental considerations underlying torsional vibration in marine installations.

The actual oscillating system in Fig. 12 comprises a four-cylinder opposed-piston engine and flywheel, directly coupled to a propeller. The first step in investigating torsional vibration characteristics is to reduce the actual to an equivalent oscillating system. The principal masses are replaced by a series of rigid flywheels having polar moments of inertia equivalent to those of corresponding masses in the actual system, connected by lengths of weightless shafting having torsional rigidities equivalent to those of corresponding sections of shafting in the actual system.

This equivalent oscillating system is capable of vibrating in several different modes, each having a different natural frequency; but broadly speaking, only the fundamental and the next higher mode are usually of serious significance in marine applications. The fundamental mode is usually referred to as the one-node mode, since the corresponding swinging form, or normal elastic curve, is characterized by the swinging of the engine and flywheel masses in one direction while the propeller mass swings in the opposite direction. Thus, there is a single stationary or nodal point in the section of shafting between the flywheel and the propeller. The next higher mode is usually referred to as the two-node mode, and is characterized by the swinging of some of the engine masses and the propeller mass in one direction while the remaining engine masses and the flywheel mass swing in the opposite direction. Thus there are two stationary or nodal points, one—termed the propeller shaft node—in the shaft section between the flywheel and the propeller, and the other—termed the crankshaft node—in the crankshaft.

Cases do occur where the second higher mode, usually termed the three-node mode, is of practical importance, but these are the exception rather than the rule.

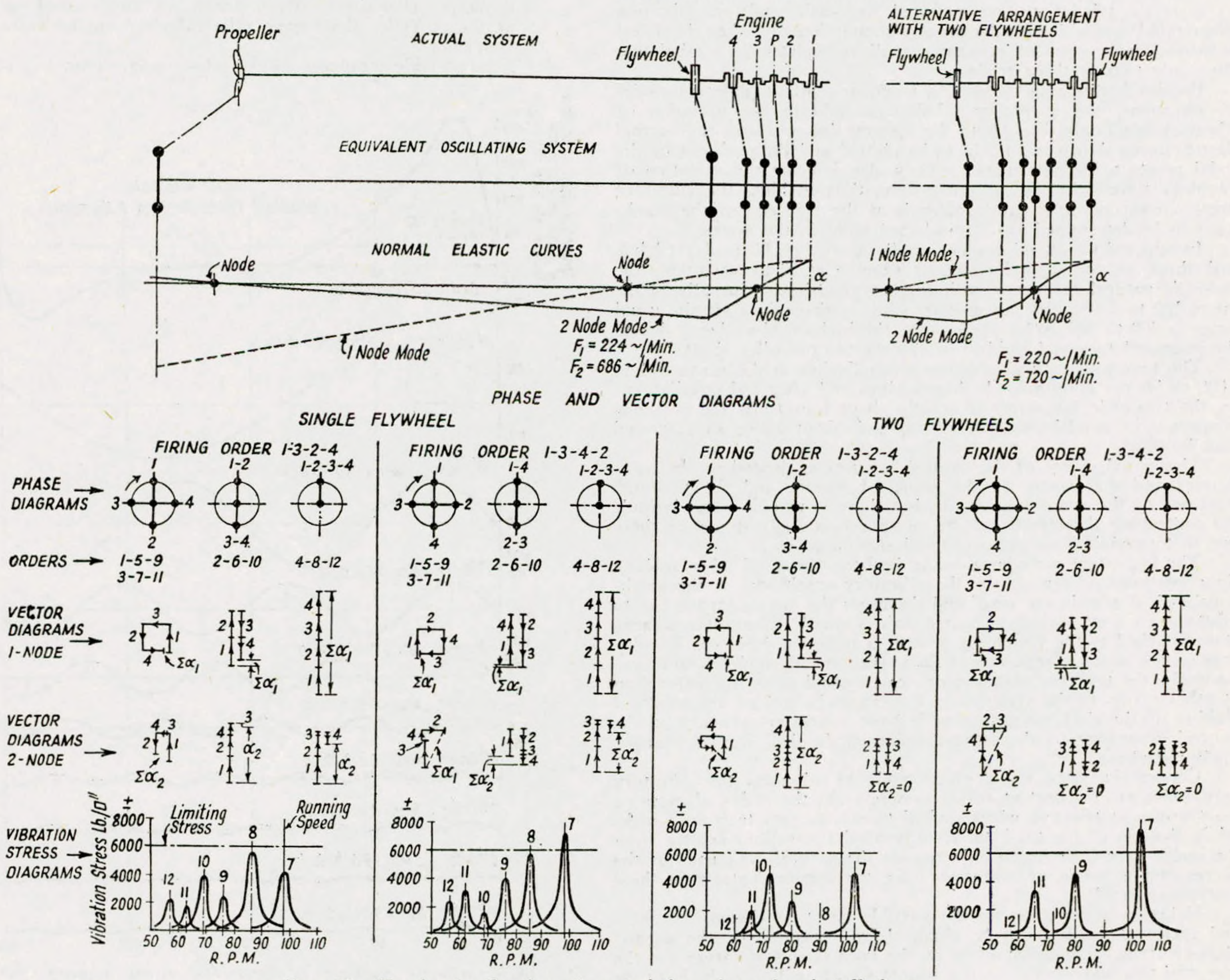

FIG. 12.—*Torsional vibration characteristics of marine installations.*

The second step in the investigation is to determine the natural frequencies of undamped torsional vibration for modes likely to be of practical significance. Since a system executing undamped natural vibrations requires no external force or couple to sustain the motion, the natural frequencies are obtained by finding the frequencies at which the summation of the maximum inertia torques at the various masses is zero. The most convenient method of performing these calculations is to commence with an assumed value for the amplitude at the free end of the crankshaft. Since the natural frequencies are independent of amplitude, unit amplitude is chosen for convenience.

A likely value for the frequency is then chosen from previous experience on similar installations and the inertia torque due to oscillation of the first mass through unit amplitude at the chosen frequency is calculated. Since this is the torque transmitted through the first section of shafting to the second mass, the torsional deflection between the first and second masses can be calculated from the known torsional rigidity of the first section of shaft. The vibratory amplitude at the second mass is then obtained by subtracting the torsional deflection in the first section of shafting from the assumed amplitude at the first mass, i.e. from unity. The inertia torque at the second mass due to oscillation of this mass at the chosen frequency through the calculated amplitude can then be obtained. The calculation proceeds in this way from mass to mass until the propeller end of the system is reached. If the chosen frequency is one of the natural frequencies of the system, the summation of the torques at all the masses will be zero. It is unlikely that the first attempt will fulfil this condition, in which case a second attempt is made with a new value for the frequency. The calculation is best performed by tabulation, the table being termed a torque summation table.

Separate tabulations are required for each mode of vibration investigated, and although the process sounds lengthy when described in words, an experienced operator usually completes each tabulation after only two or three trials.

Besides determining the natural frequencies of the different modes of vibrations, the frequency tabulations indicate the variation of vibratory amplitude throughout the system, thus enabling the normal elastic curves shown in Fig. 12 to be plotted and the positions of the nodal points to be determined. They also indicate the variation of vibratory torque throughout the system, thus enabling the vibratory stress corresponding to unit amplitude at the free end of the crankshaft to be determined for any selected point in the system.

In opposed-piston oil engine installations of 2,000 to 4,000 b.h.p. with three- or four-cylinder engines running at 90 to 100 r.p.m., the one-node natural frequency with engines amidships is usually in the range 150 to 250 cycles per minute, while with engines aft the usual range is 300 to 500 cycles per minute, the higher values being due to the increased torsional rigidity of the shorter propeller shaft.

The two-node natural frequency usually lies in the range 600 to 1,000 cycles per minute for both amidships and after-end installations, i.e. the two-node frequency is usually about four times the one-node frequency in amidships installations, and about twice in after-end installations.

The actual values of the natural frequencies depend on the magnitudes and disposition of the principal masses and the torsional rigidities of the connecting shafts, but there is usually some latitude for controlling the frequencies by an appropriate adjustment of these two factors while a project is in the design stage.

The important considerations to be borne in mind when making these adjustments are, firstly, the vibratory amplitudes in the neighbourhood of a node are small and therefore the inertia torque due to vibration of a mass situated near a node is small. Thus, even a large mass attached to the system at or near a node has little or no influence on the natural frequency of the corresponding mode of vibration. Secondly, the torsional strain at or near a nodal point is greater than at other points in the system and therefore changes of the torsional rigidity of the shafting near a node have a greater influence on the natural frequency of the corresponding mode than similar changes made elsewhere.

Conversely, since for a given mode of vibration the vibratory amplitudes are greatest at points remote from the nodes, changes of mass are most effective when applied to masses remote from a node, while changes of the torsional rigidity of shaft sections remote from the nodes have little or no influence on the natural frequency of the corresponding mode of vibration since the torsional strain in these sections is small.

Valuable information on the possible effect of changes of mass and torsional rigidity can be obtained by inspection of the normal elastic curves. Thus, inspection of the normal elastic curves for the one- and two-node modes of the system shown in Fig. 12 leads to the following conclusions :—

(*a*) The one-node frequency can be altered appreciably by changing the torsional rigidity of the propeller shafting or by changing the polar moment of inertia of the propeller or engine masses. An increase of shaft diameter or a reduction of the polar moment of inertia of the propeller or engine masses increased the one-node frequency. Conversely a reduction of propeller shaft diameter or an increase in the polar moment of inertia of the propeller or engine masses will reduce the one-node natural frequency.

It should be noted, however, that a change in the polar moment of inertia of the flywheel alone has usually only a small influence on the one-node frequency, because the flywheel inertia is generally only a small proportion of the total engine inertia in present-day opposed-piston engine installations.

(*b*) The one-node frequency is not appreciably altered by changes in the torsional rigidity of the crankshaft because the crankshaft is much more rigid than the propeller shaft and the torsional strain in the crankshaft for the one-node vibration is small. This implies that the one-node torsional vibration stress is greatest in the propeller shaft, and is negligible in the crankshaft.

(*c*) The two-node frequency is not appreciably affected by changes in the polar moment of inertia of the propeller, since the two-node vibratory amplitude at the propeller is small compared with the amplitudes of the engine and flywheel masses.

(*d*) Since the torsional rigidity of the propeller shaft is small compared with that of the crankshaft, the torsional strain in the crankshaft for the two-node mode is much greater than in the propeller shaft. Thus, changes in the torsional rigidity of the propeller shaft have little influence on the two-node frequency.

(*e*) The two-node frequency can be altered appreciably by chang-

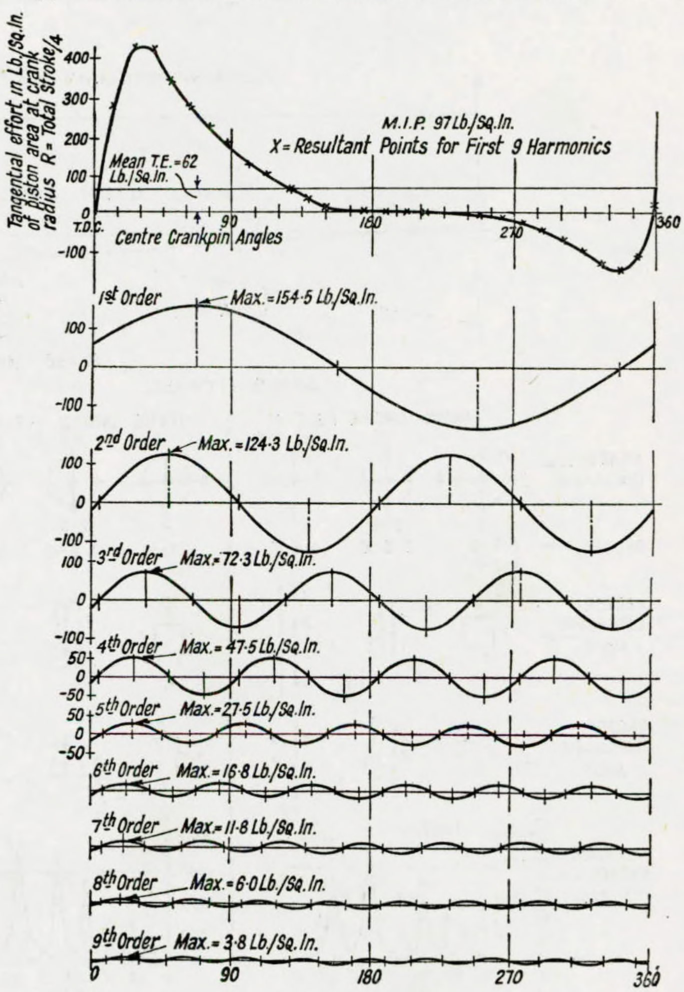

FIG. 13.—*Harmonic analysis of tangential effort diagram for an opposed-piston engine.*

ing the torsional rigidity of the crankshaft, particularly the sections in the neighbourhood of the node, or by changing the polar moment of inertia of the crankshaft masses, particularly masses most remote from the crankshaft node.

(*f*) The foregoing considerations indicate that an alteration in the polar moment of inertia of the propeller or the torsional rigidity of the propeller shaft will change the one-node natural frequency appreciably without having much influence on the two-node frequency. On the other hand, an alteration in the polar moment of inertia of the flywheel or engine masses, or in the torsional rigidity of the crankshaft, will change the two-node natural frequency appreciably without having much influence on the one-node frequency. In this way it is possible to change the tuning of the one- and two-node modes independently.

Next, it is necessary to examine the oscillating system for disturbing forces which represent potential sources of resonant torsional vibration. The two principal sources of excitation are the propeller and the engine. The principal torsional excitations from the propeller have frequencies equal to the revolutions per minute of the propeller multiplied by the number of blades, or an integral multiple of the number of blades; but these disturbances are usually so small as to be of little practical significance compared with engine disturbances.

Fig. 13 shows the variation of the tangential effort over one revolution for one cylinder of a two-stroke cycle opposed-piston engine. This diagram can be regarded as composed of the mean tangential effort which represents the constant driving torque transmitted to the propeller shaft, and a whole series of sinusoidal curves having 1, 2, 3, etc. complete cycles per revolution of the crankshaft. These pulsating components are termed "harmonics" and the first nine are shown. The harmonic components are distinguished by order numbers, the order number being the number of complete cycles per revolution of the crankshaft.

Thus the excitation frequency of a given harmonic is its order number multiplied by the r.p.m., and resonance occurs whenever an excitation frequency coincides with the natural frequency of one of the modes of vibration of the system. The resonant or critical speeds corresponding to a given harmonic order are therefore determined by dividing the natural frequency by the order number.

In the system shown in Fig. 12, for example, where the one- and two-node natural frequencies are 224 and 686 cycles per minute, the critical speeds are as follows:—

Order Number.	Critical Speeds	
	One-Node.	Two-Node.
1	224/1=224 r.p.m.	686/1=686 r.p.m.
2	224/2=112 „	686/2=343 „
3	224/3=75 „	686/3=229 „
4	224/4=56 „	686/4=171 „
5	224/5=45 „	686/5=137 „
6	224/6=37 „	686/6=114 „
etc.	etc.	etc.

Although this table indicates that theoretically there are a great many critical speeds, many are of no practical significance either because they are well outside the operating speed range or because the amplitude is small.

In multi-cylinder engines the relative importance of the various critical speeds depends on the shape of the normal elastic curve for the mode of vibration under consideration, the firing order of the engine, and the relative magnitude of the harmonic impulses.

The firing order of the engine determines the phase relationship between the harmonics of a given order originated by the several cylinders. This phase relationship is obtained from phase diagrams of the type shown in Fig. 12, where the first order phase diagram for a two-stroke cycle engine is simply the firing order or crank sequence diagram since the first order harmonic completes one cycle per revolution, i.e. the first order harmonics for the various cylinders follow one another in the same sequence and in the same angular relationship as the engine cranks.

The second order harmonic completes two cycles during one revolution of the crankshaft, i.e. twice the phase angle of the first order harmonic in the same time interval. Thus, the second order phase diagram can be represented by an imaginary crank diagram in which the cranks rotate at twice the speed of the crankshaft, and the angles between the cranks are double those in the first order diagram. Similarly the 3rd, 4th, etc. order phase diagrams are obtained from the first order diagram by multiplying all the crank angles in the first order diagram by 3, 4, etc., as shown in Fig. 12.

When constructing the phase diagrams it will be found that for order numbers which are integral multiples of the number of cylinders the imaginary cranks are all in-line, i.e. the impulses from the various cylinders are all in phase. It will also be found that each phase diagram represents a whole series of order numbers, since the imaginary crank diagrams either repeat themselves or form mirror images for some of the higher orders. Thus, in the four-cylinder engine shown in Fig. 13, three phase diagrams are sufficient to determine the phase relationships for all harmonic orders. The three groups comprise, firstly, all odd orders (1-5-9-etc. and 3-7-11-etc.); even orders which are integral multiples of the number of cylinders (4-8-12-etc.); and the remaining even orders (2-6-10-etc.).

The phase diagrams indicate that in general there is a phase relationship to be taken into account when determining the resultant vibrational energy which is fed into the oscillating system by a group of cylinders. The energy imparted by any one cylinder of the group is proportional to the maximum amplitude of the harmonic torque at the assumed operating conditions, for the particular harmonic under consideration, multiplied by the specific amplitude on the normal elastic curve at the cylinder in question.

The energy input from the whole group of cylinders is therefore proportional to the resultant of a vector diagram in which the energy vector for each cylinder is drawn parallel to the imaginary crank for that cylinder in the appropriate phase diagram. The length of the vector is the product of the maximum amplitude of the harmonic impulse under consideration and the specific amplitude on the normal elastic curve at the cylinder in question, taking care to regard amplitudes above the shaft axis as positive and those below as negative.

In practice it is more convenient to defer multiplying by the amplitude of the harmonic impulse until the vector resultant has been found, i.e. the lengths of the vectors are usually made equal to the specific amplitudes on the normal elastic curve and only the vector resultant is multiplied by the amplitude of the harmonic impulse. This avoids having to draw a separate vector diagram for each of the harmonic orders covered by each phase diagram.

The actual amplitudes of the vibratory stresses at the various critical speeds can only be determined if the magnitudes of the damping torques acting on the system are known. Inspection of the normal elastic curves indicates that for one-node vibrations the amplitude is much greater at the propeller than at any other point in the system. Furthermore, the damping action of a marine propeller is very much greater than damping due to hysteresis in the material of the propeller shaft, while the engine amplitudes are relatively so small that engine damping has only a minor influence.

Thus, it is usually sufficient to regard the propeller as the sole source of damping of one-node vibrations. Since fairly reliable data is available on propeller damping there is usually good agreement between calculated and observed amplitudes at one-node critical speeds.

The vector diagrams in Fig. 12 show a large resultant for order numbers which are integral multiples of the number of engine cylinders, while for all other orders the resultants are so small that they can be disregarded.

Thus, in a four-cylinder engine the only significant one-node criticals are the 4th, 8th, 12th, etc. orders, and since the vector resultants for these orders are the same for all firing orders, changes of firing order have no influence on one-node torsional vibration stresses.

As previously stated, the one-node natural frequency for installations of the sizes under consideration usually lies in the range 150 to 250 cycles per minute with engines amidships. Hence the lowest order one-node critical speed for a four-cylinder engine lies between 40 and 60 r.p.m., i.e. well below the cruising speed range. All other significant one-node criticals are even more remote from the cruising range. Nevertheless, it must be borne in mind that this critical has to be negotiated whenever the engine is started or stopped, and for this reason it is necessary to make sure that the vibration stress is within safe limits. Now it has already been mentioned that the maximum vibratory stress for one-node vibration occurs in the propeller shafting so that the only precaution necessary so far as one-node criticals are concerned is to see that the propeller shaft is of sufficient diameter to keep the stress within safe limits.

In cases where the required increase of propeller shaft diameter would bring the one-node critical speed too close to the cruising range, the vibratory stress can be reduced by increasing propeller damping. This can be done by increasing the relative vibratory amplitude at the propeller either by reducing the polar moment of inertia of the propeller or by increasing the polar moment of inertia of the engine masses. The effect of either change is to move the node away from the propeller and nearer to the engine, thus increasing the relative amplitude at the propeller. This not only increases propeller damping but at the same time reduces the energy input from the engine, both effects tending to reduce stress amplitudes.

When the engine is installed at the after end of the ship, the one-node natural frequency lies in the range 300 to 500 r.p.m., i.e. the lowest one-node critical speed for a four-cylinder engine lies between 75 and 125 r.p.m. In such cases the one-node critical may occur within the cruising range with shafting proportioned in accordance with normal classification requirements. This is not desirable even though the stress can be kept within safe limits. It is better practice to raise the one-node natural frequency by an appropriate increase of propeller shaft diameter, so that the lowest order one-node critical speed occurs at least 30 per cent. above the maximum running speed. When this is done only higher order one-node criticals will be present in the lower speed range and these will not only be below the cruising range, but will be of small amplitude, due to the reduced magnitude of the forcing torques.

The above discussion indicates that, broadly speaking, one-node torsional vibration is not a major problem in marine installations of the type under discussion. This aspect of the subject will therefore not be pursued further.

The determination of the vibration stresses at two-node critical speeds is usually the crux of the torsional vibration problem in installations of the type shown in Fig. 12, and is a more difficult problem than the determination of one-node stresses. The following are the main characteristics of two-node vibrations :—

(*a*) The vibratory stress is greatest in the crankshaft section of the oscillating system and is practically negligible in the propeller shaft.

(*b*) Since the relative amplitude at the propeller, as indicated by the two-node normal elastic curve, is very small, propeller damping is of little practical significance. This implies that the principal damping of two-node vibrations occurs in the engine portion of the system.

Engine damping is due to several causes, including external friction at rubbing surfaces and at mechanical joints such as flanged couplings; internal friction due to elastic hysteresis in the materials of the shafts and running gear; energy losses caused by impacts between moving parts; and the effects of cyclic variations of speed, cyclic changes in the moments of inertia of heavy masses due to their irregular motions, and changes of torsional rigidity of the shafting due to straining under load, all of which tend to prevent synchronism between the frequency of the applied impulses and the natural frequency of the system.

The estimation of the overall damping of two-node vibrations is therefore very difficult, and reliance has so far been placed on empirical rules devised by each manufacturer to suit the characteristics of a particular engine type. These empirical rules are based on comparisons between stress amplitudes calculated according to a selected formula and the results actually obtained from torsiograph tests on the complete installation. In this way it has been possible, in the case of the opposed-piston engines under discussion, to evolve rules which do give a fairly reliable indication of the stress amplitudes likely to occur at the various two-node criticals in a new installation.

(*c*) Inspection of the normal elastic curves in Fig. 12 show that whereas for the one-node mode the various cylinders vibrate with practically the same amplitude, in the case of the two-node mode there is considerable variation between the amplitudes at the various cylinders. This implies that the vector diagrams may show appreciable resultants for all harmonic orders, so that in general there will be several significant critical speeds in the cruising range. Thus, the simple expedient of adjusting the tuning of the system by a change in the torsional rigidity of a convenient shaft or the polar moment of inertia of a convenient mass may not suffice to place all significant two-node criticals outside the cruising range.

(*d*) Since the magnitude of the vector resultant depends on the phasing of the vectors for each cylinder of a group, i.e. on the firing order of the engine, the relative amplitudes of the two-node vibratory stresses at the various criticals can be changed by altering the firing order. In the case of two-stroke cycle engines, this implies a change of crank sequence. It should be noted, however, that changes of crank sequence can have no effect on harmonic orders which are integral multiples of the number of cylinders, a point which is clearly brought out in Fig. 12.

Furthermore, it should be borne in mind that in single-piston engines where primary balance of the moving parts is achieved by collective balancing, a change of crank sequence to reduce torsional vibration may result in unpleasant vibration due to unbalance. This handicap is not present in opposed-piston engines of the balanced type, where primary balance is achieved independently of crank sequence.

(*e*) The magnitudes of the vibratory stresses at two-node criticals can be changed by altering the disposition of the engine masses, for example by replacing a single flywheel at the after end of the crankshaft by a flywheel at the forward end; by replacing the single flywheel by two smaller flywheels, one at each end; or by distributing the flywheel inertia in the form of balance weights on the crankwebs. Changes of this type alter the relative amplitudes at the various cylinders, and thus alter the magnitudes of the vector resultants for the various harmonic orders.

The examples shown in Fig. 12 are based on the two twin-screw opposed-piston engine installations already discussed, on which the first serious torsional vibration trouble was experienced. These engines were amongst the first of the unequal-stroke balanced design with firing order 1-3-4-2, whereas previous engines had been mostly of the equal-stroke type with firing order 1-3-2-4.

The vibration stress diagrams for these two firing orders and for two different flywheel arrangements are given at the bottom of Fig. 12.

These diagrams show that with the single after end flywheel originally used on these engines, there were two significant two-node criticals, namely a 7th order critical at the designed running speed of 98 r.p.m. and an 8th order critical at 86 r.p.m.

Now the phase and vector diagrams show that the amplitudes of the 4, 8, 12, etc. order two-node criticals are the same for both firing orders, whereas the amplitudes of the remaining *even* order criticals are small for firing order 1-3-4-2 and large for firing order 1-3-2-4. The amplitudes of the *odd* order two-node criticals, on the other hand, are large for firing order 1-3-4-2 and small for the firing order 1-3-2-4.

Thus the introduction of the balanced-type engine with firing order 1-3-4-2 produced a large increase in the amplitude of the odd order criticals. Since in the particular installations under discussion the 7th order critical speed happened to coincide with the designed operating speed, resonant torsional vibration of sufficient amplitude to cause failure of the crankshafts was produced. This was undoubtedly the root cause of the vibration trouble experienced on these ships.

The peak stresses at the various two-node criticals shown in the vibration stress diagrams in Fig. 12 are based on an empirical rule developed from torsiograph measurements made on a large number of opposed-piston engine installations. The limiting stress shown in these diagrams is based on a torsional fatigue limit of ± 7 to ± 8 tons per sq. in. for laboratory specimens of 28 to 32 tons per sq. in. mild steel, allowing a factor of from 2·5 to 3 to cover stress concentration effects at abrupt changes of section such as occur at crankpin and journal fillets and oil-holes, and the effect of size on the metallurgical properties of the material. Experience of failures on large oil-engine crankshafts appears to confirm that shafts made from 28 to 32 tons per sq. in. mild steel are in danger of failure when the nominal torsional vibration stress in the shaft exceeds $\pm 6{,}000$lb. per sq. in. Additional confirmation of the reliability of this criterion will be found in an article by Lehr and Ruef[16].

This article describes full-scale torsional fatigue tests on a crankshaft having 9·5in. diameter crankpins and journals, made from 35 to 45 tons per sq. in. carbon steel. The torsional fatigue limit for laboratory specimens of the material was ± 10 tons per sq. in., while the value for the actual crankshaft was only $\pm 2{\cdot}7$ tons per sq. in., i.e. the crankshaft showed an overall stress concentration factor of about 3·5. The shaft tested was of the solid-forged type and it was found that specimens with metallurgical flaws had practically the same fatigue strength as specimens without flaws, which suggests that the influence of the geometrical design of the shaft was greater than that of metallurgical imperfections.

There is no doubt that careful design aimed at providing the smoothest possible contours at transition points, followed by careful control of all manufacturing processes, represents the most effective means for obtaining the highest fatigue strength from a crankshaft

The diagrams at the right-hand side of Fig. 12 show the effect of replacing a single flywheel located at the after end of the crank-

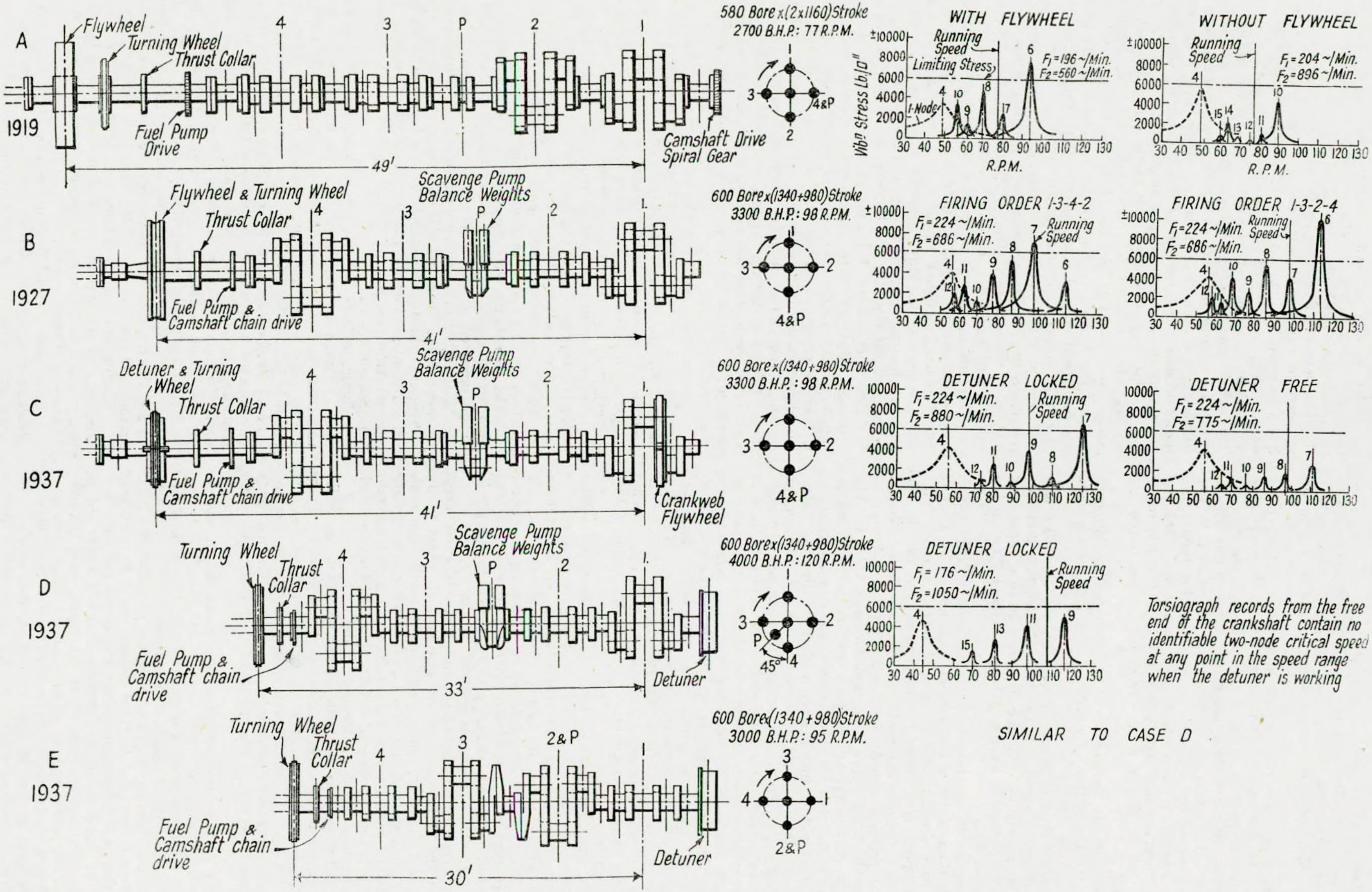

FIG. 14.—*Torsional vibration characteristics of four-cylinder opposed-piston marine oil engines.*

shaft by two smaller flywheels, one at each end.

In the particular case illustrated the polar moments of inertia of the flywheels were adjusted so that the crankshaft node for the two-node mode of vibration was located at the centre of the group of cylinders.

As previously stated, this change has little or no influence on one-node criticals. In the case of two-node criticals, however, the effect of moving the crankshaft node to the centre of the cylinder group is to produce a symmetrical normal elastic curve in this region, and this results in cancellation of certain two-node criticals. The 4, 8, 12, etc. order two-node criticals are cancelled for both firing orders, while the remaining even order criticals are cancelled for firing order 1-3-4-2 but not for firing order 1-3-2-4. The odd order two-node criticals are small for firing order 1-3-2-4 and large for firing order 1-3-4-2.

These arrangements are included to show the important influence which changes in the disposition of engine masses can exert on the magnitudes of two-node criticals.

Fig. 14 illustrates the development of the Doxford opposed-piston engine from the point of view of crankshaft construction and torsional vibration characteristics. All the diagrams in this illustration relate to four-cylinder engines having a bore of about 23in. and a total stroke of about 92in.

Diagram A applies to the prototype engine built in 1919 which had four separate three-throw crank elements connected by a short scavenge crank, and separate thrust and flywheel shafts. This engine had equal upper and lower piston strokes, and due to the imperfect balance of the moving parts of the individual cylinders there was residual primary couple unbalance which was kept to a minimum by using the firing order and crank sequence 1-3-2-4. The engine was rated at 2,700 b.h.p. at 77 r.p.m., and the torsional vibration stress diagram shows that for this condition the vibratory stresses at the significant two-node criticals were within the limiting value. No serious torsional vibration troubles were experienced with engines of this type and rating, although, as previously mentioned, the camshaft drive had to be moved to the after end of the engine because of excessive vibration which caused deterioration of the toothed gearing. This indicated that although the torsional vibration stresses were below the limiting value, the vibratory amplitudes in the neighbourhood of the 8th order two-node critical were undesirably high and were having a deleterious effect on engine performance. It is also of interest to note from the torsional vibration stress diagram that the presence of a serious 6th order critical at about 93 r.p.m. would have been a limiting factor on attempts to increase the engine rating by an increase of r.p.m.

As soon as the full significance of torsional vibration was appreciated, by which time experience had shown that a large flywheel was not only unnecessary for starting and slow-running, but might be a serious menace from the point of view of torsional vibration, the system shown in Diagram *A* was retuned by removing the heavy flywheel. This produced a considerable increase of the two-node natural frequency, and also moved the crankshaft node to the neighbourhood of the centre of the engine cylinder group.

These changes resulted in the very favourable vibration stress diagram shown at the right-hand end of Diagram *A*. Due to the increased two-node frequency, only high-order small amplitude impulses occurred in the operating speed range, while, due to the shift of the crankshaft node towards the centre of the cylinder group, the only significant criticals were the 10th and 14th orders, both having peak stresses well below the limiting value.

Diagrams B and C apply to the twin-screw installations on which the first serious torsional vibration trouble was experienced. These engines were of the unequal-stroke balanced type with crank sequence and firing order 1-3-4-2. The crankshaft consisted of two sections, each containing two three-throw crank elements, connected by a short scavenge crank section. The flywheel was mounted on the thrust shaft.

This case has already been discussed in some detail, and as shown in the torsional vibration stress diagram for case *B*, the cause of the trouble was the presence of a large 7th order two-node critical in the neighbourhood of the designed running speed. The vibration stress diagram at the right-hand end of diagram *C* shows the effect of changing to firing order 1-3-2-4. This change would have reduced the 7th order peak stress quite appreciably, but would not have provided a satisfactory solution, firstly because it would have had no effect on the large 8th order critical at 86 r.p.m., and secondly because it would have introduced an unbalanced secondary couple.

The cure adopted at the time of the trouble was to replace the original heavy flywheel by the lightest possible turning wheel.

This raised the two-node frequency to 950 cycles per minute so that the running speed occurred between the 9th order critical at 106 r.p.m. and the 11th order at 86 r.p.m.

Later, in 1937, a further improvement in the torsional vibration characteristics of these installations was obtained by the introduction of a detuning flywheel.

The detuning flywheel method of dealing with torsional vibration was carefully studied at Doxford's in the early part of 1934, after over five years spent in research on torsional vibration problems, during which a large number of experimental investigations were carried out on various types of installation both on the test-bed and at sea. This work culminated in the evolution of an oscillating system in which a conventional single heavy flywheel was replaced by two light flywheels, one at each end of the crankshaft, proportioned so that the crankshaft node for the two-node mode of vibration was located at or near the centre of the cylinder group.

In this way it was possible to raise the two-node frequency to a value which ensured that only high-order and, therefore, comparatively weak exciting impulses were present in the operating speed range. The location of the node at the centre of the engine produced a symmetrical normal elastic curve, thus bringing about cancellation of certain orders, as previously explained.

This provided much wider ranges of speed free from torsional resonance than had previously been possible, while the actual magnitudes of the vibratory stresses at the remaining critical speeds were usually well below the limiting value.

Nevertheless, it was considered that the work on torsional vibration would not be complete until a method had been found which would reduce torsional vibration to a negligible quantity over the whole speed range. This viewpoint was strengthened by the appearance in some engine specifications of a clause which stipulated freedom from torsional criticals at all speeds as a contract requirement.

Two methods of bringing about a still further reduction of torsional vibration were considered. Firstly, the introduction of supplementary damping to augment the inherent damping capacity of the engine system; and, secondly, the introduction of a device which would automatically produce a cyclic variation of the tuning of the system whenever the vibratory amplitude tended to build up at resonance. Energy absorbing dampers were not regarded with favour on account of mechanical difficulties, and because it was thought that they might prove unreliable in operation and expensive and troublesome in maintenance.

Devices for interfering with the tuning of the system were not new, but had appeared in various forms many years previously. Thus, in 1922 Gümbel(7) described two methods of changing the natural frequency. In the first arrangement a supplementary mass was connected to the free end of the system, i.e. at a point remote from a node, using a friction or hydraulic clutch so that the mass could be connected or disconnected as required. In the second arrangement a special section of shafting was inserted in the neighbourhood of a node where the torsional strain was greatest. This special section of shaft comprised a thin quill shaft telescoped inside a tubular shaft of much greater torsional rigidity.

The two shafts were connected to the system through a friction or hydraulic clutch so that the stiff tubular section could be connected or disconnected as required.

With either of these arrangements two natural frequencies could be selected, the lower with the supplementary mass connected or the stiff tubular shaft disconnected, and the higher with the supplementary mass disconnected or the stiff tubular shaft connected. These schemes suffered from mechanical difficulties and the fact that their action was not automatic.

Again, in 1930, Kjaer patented a number of devices whereby the polar moment of inertia of one of the principal masses, or the torsional rigidity of one of the principal elastic members could be varied periodically and automatically by the rotation of the shaft (British Patent No. 359,517). In one of the variable inertia devices a heavy flywheel rim was periodically connected to the shaft and then disconnected by a system of links and pistons operated by oil pressure controlled by ports cut in the shaft. In one of the variable stiffness devices a heavy flywheel rim was connected to the shaft by a series of spokes of rectangular cross-section, the spokes being arranged so that they turned about their radial axes as the shaft revolved. Thus, the flywheel rim was alternatively stiffly and flexibly connected to the shaft according to whether the longer or shorter side of the rectangular cross-section of the spokes was in the plane of rotation.

Kjaer's interest in devices for periodically varying the natural frequency of the system was doubtless stimulated by the experimental work which he described in a paper read before the Royal Technical University, Copenhagen(9).

This work showed that appreciably lower vibratory stresses were obtained in otherwise similar engines when the cyclic fluctuation of speed was increased. It also indicated that cyclic variation of natural frequency due to variation of the polar moment of inertia of the reciprocating masses with crankpin position could, in certain cases produce an appreciable reduction of the peak stresses at critical speeds.

From this study of available literature it was concluded that if the rim of a flywheel at either the forward or after end of the crankshaft was connected to the shaft by a flexible coupling which automatically changed the torsional rigidity of the connection when the vibratory amplitude began to increase in the neighbourhood of a critical speed, then a useful reduction of peak stresses might be obtained. Furthermore, it was known that the continuous-rung type of spring used in the Bibby flexible coupling offered one means of obtaining the required performance. Couplings of this type had been in use for many years on a large variety of installations and operating experience had been very satisfactory.

An interesting example was the fitting of Bibby couplings in the airscrew shafts of the Graf Zeppelin after it had made a forced landing at Toulon with four broken crankshafts at the commencement of its first voyage to America in 1928.

The early development of the Bibby coupling was based on its use as a transmission member, the frequency changing characteristic being obtained by using one of the principal inertias as the co-operating mass. There was, therefore, no real novelty in the proposal to use a special supplementary flywheel as the controlled inertia and it was not surprising to find, when the subject was discussed with Mr. Bibby in 1934, that he had already been actively working on a similar proposition (see British Patent No. 427,138), which he had named a "detuning flywheel" or "detuner".

The first detuning flywheel fitted to an opposed-piston engine was tested on the test-bed at Doxford's works during April and May, 1934. The engine had three cylinders 400 mm. bore×(540+760) mm. stroke, and the trials were carried out with the dynamometer adjusted to full-load conditions, namely, 1,250 b.h.p. at 240 r.p.m. With the normal flywheel the one-node frequency on the test-bed was 1,350 cycles per min.—this corresponded approximately to the two-node frequency of the corresponding ship installation—and there was a serious 6th order resonance at 225 r.p.m. having an amplitude at the free end of the crankshaft of $\pm 0{\cdot}7°$ or a peak vibration stress in the crankshaft of ±9,500lb. per sq. in.

There were also less serious 7th, 8th and 9th order criticals at 193, 170 and 150 r.p.m., having an amplitude of about $\pm 0{\cdot}3°$ at the free end of the crankshaft or a crankshaft stress of about ±4,000lb. per sq. in.

The locations and severities of all these criticals were carefully determined by torsiograph tests, after which the detuning flywheel was installed in place of the normal flywheel. The detuning flywheel and its control spring were proportioned so that the mean value of the natural frequency was about the same as that of the original system, namely, 1,350 cycles per min.

Torsiograph tests with the detuning flywheel in operation showed the presence of a small 6th order critical at 225 r.p.m., but its amplitude was so small that it could only be determined by harmonic analysis of the Geiger records.

This analysis yielded an amplitude of $\pm 0{\cdot}08°$ at the free end of the crankshaft, or a crankshaft stress of under ±1,000lb. per sq. in. No trace of the other criticals could be found on the records.

These tests amply confirmed Mr. Bibby's claim that with proper design and in appropriate applications a detuning flywheel at the after end of the crankshaft could reduce the crankshaft stresses to about 10 per cent. of their value with a normal flywheel.

Further tests were carried out during December, 1934, and April, 1935, this time on an engine which had provision for mounting the detuning flywheel at the free end of the crankshaft. The engine had three cylinders, 500 mm. bore×(1,200+880) mm. stroke, and the dynamometer was again adjusted to full load conditions, namely, 1,880 b.h.p. at 120 r.p.m.

The engine was a normal three-cylinder unit having a moderate-size flywheel at each end of the crankshaft, the oscillating system being adjusted so that the crankshaft node was at the middle of the cylinder group.

The detuning flywheel was secured to the front end of the forward flywheel, and since its polar moment of inertia was only about 3 per cent. of the polar moment of inertia of the engine masses, the natural frequency of the system with the detuning flywheel rim locked was not greatly different from the value with the normal flywheel alone.

Geiger torsiograph records taken with the detuning flywheel rim locked indicated that the only important critical was the 7th order at about 136 r.p.m., the amplitude being about $\pm 0{\cdot}2°$, corresponding to a crankshaft stress of ±2,180lb. per sq. in.

With the detuner in action, the 7th order was just discernable at about 130 r.p.m., the amplitude, determined by harmonic analysis of the Geiger records, being about $\pm 0{\cdot}03°$, or a crankshaft stress of about ±330lb. per sq in. Thus, the effect of the detuner in this case was to reduce the peak stress to about 15 per cent. of its value with the normal flywheels.

The only mechanical trouble encountered during the trials was a tendency for the spring segments to bind in the axial direction when the assembly was first tested. This prevented the detuning mass from swinging freely on the springs and was due to no fault in the detuner itself but resulted from an oversight in not allowing sufficient axial float of the spring elements—usually about 0·012in. The trouble disappeared as soon as the specified end float was provided.

Although the tests just described represented the whole of the experimental work carried out on the detuner by Doxford's, the results were considered sufficiently good to justify a decision to fit the device on the engines of a twin-screw installation then under construction at the works of Messrs. Barclay, Curle & Co., for installation in The British India Steam Navigation Company's motor liner *Dilwara*.

These engines were the first five-cylinder Doxford engine units to be constructed and had cylinders 560 mm. bore×(980+700) mm. stroke, with a maximum rating of 3,265 b.h.p. at 128 r.p.m. per engine.

The detuning flywheel was bolted to a flange at the forward end of the crankshaft, and was arranged so that the detuning rim could be locked to the hub when required. Test-bed trials were carried out at Messrs. Barclay, Curle's works in September, 1935. The records obtained with the detuner locked showed well-defined resonant conditions at the 6th, 7th, 8th and 9th order criticals, i.e. at about 165, 142, 125 and 110 r.p.m. respectively. Of these the 6th order had a recorded amplitude of $\pm 0{\cdot}20°$ at the free end of the crankshaft, while the remainder had an amplitude of about $\pm 0{\cdot}1°$.

With the detuner in action there was no clearly defined critical at any speed. Careful examination of the records, however, revealed a small 6th order at about 155 r.p.m., with an amplitude, determined by harmonic analysis, of $\pm 0{\cdot}06°$. Thus the detuner had reduced the 6th order vibration stress to about one-third of its value with the detuner locked. These results were obtained with the dynamometer disconnected so that the engine was running light. Consequently the critical stresses with the detuner locked were not as large as they would be under full load conditions. This in turn implied that with the detuner working the springs were not being subjected to their full designed duty, and therefore the detuning action was less than had been found on previous tests. Nevertheless, the results indicated that even under light load conditions the detuner was capable of exerting a powerful restraining action on the building up of resonant peaks.

The mechanical construction of present-day detuning flywheels is shown in Fig. 15.

The essential components are a cast-iron driving member, which itself represents a light flywheel, bolted to the crankshaft, a cast-iron flywheel member or detuning mass mounted on a journal carried

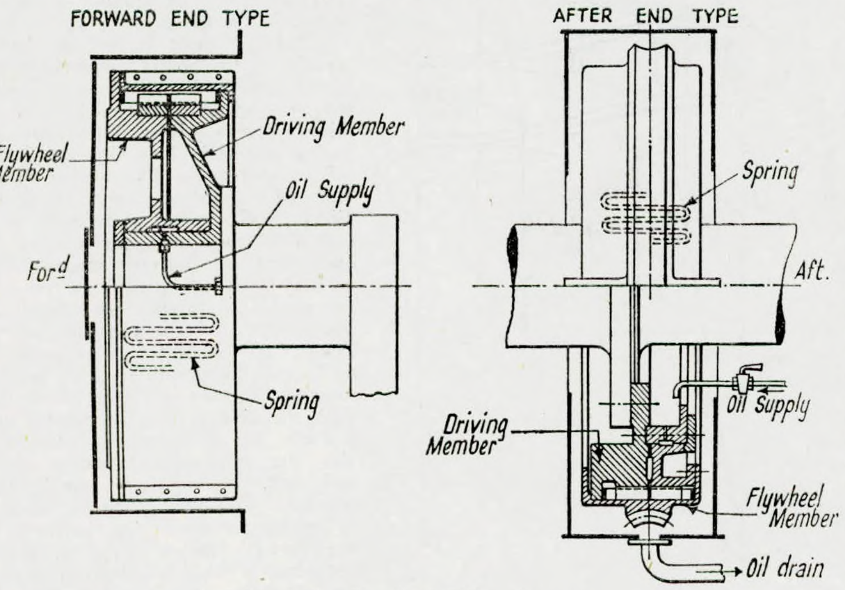

FIG. 15.—*Doxford-Bibby detuner.*

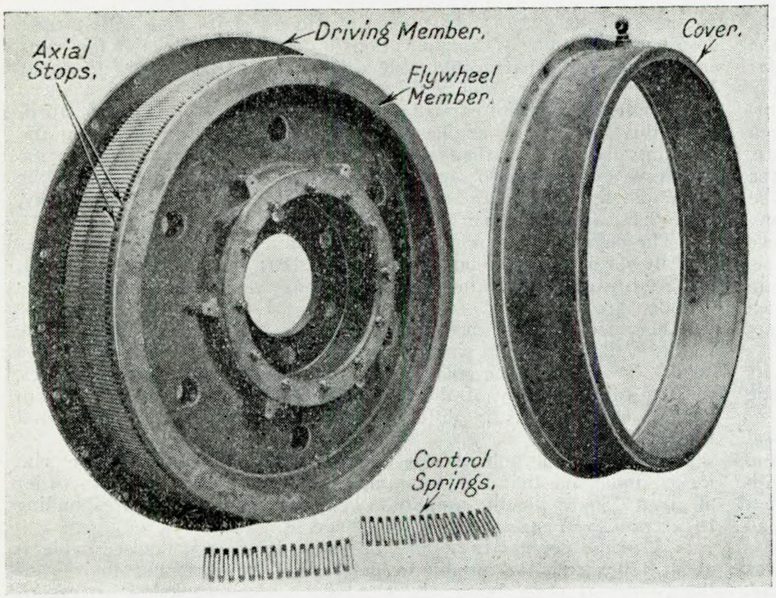

FIG. 16.—*Doxford-Bibby detuning wheel as fitted to twin-screw Doxford engines of 7,500 h.p.*

by the driving member so that the detuning mass is free to perform angular oscillations relative to the driving member; and the steel control springs which connect the driving and flywheel members.

The control springs, which are made from steel strip folded into a continuous zig-zag ribbon, are accommodated in slotted rings of forged steel shrunk on to the peripheries of the driving and flywheel members, and are retained in position by an outer casing bolted to the flywheel member. The outer casing is divided across a diameter to facilitate removal.

Lubricating oil is supplied, through suitable drillings, to the centre of the journal bearings, and the leakage oil flows radially into the spring pockets.

In the forward-end type the driving and flywheel members are each made in one piece, the journal being cast integral with the driving member.

The lubricating oil supply is taken from the crankshaft system and is drained back to the engine sump through drain holes of ample size in the bedplate girders.

In the after-end type the driving member is bolted to a collar formed on the thrust shaft and is made in halves to facilitate assembly. The need for separate forged steel rings for the slotted members may be avoided by making both the driving and flywheel members of cast steel, a cast-iron journal ring being bolted to the driving member. Lubricating oil is supplied to an annular space, formed by a flange on the journal ring, from which it flows through suitable drillings to the journal and thence outwards to the spring pockets. The detuner is enclosed in a sheet metal casing provided with an oil drain pipe of ample size.

Fig. 16 is a photograph of an earlier type of detuner in which the outer casing was made in one piece. A point of interest are the axial stops which were provided in some early types of limiting axial movements of the spring segments. These stops are not an essential feature of the detuner, but are a refinement fitted only in cases where exceptionally silent operation is demanded.

The action of the spring system is shown diagrammatically in Fig. 17, where the left-hand diagram shows the spring rungs in the unloaded condition.

If one of the members of the coupling is fixed and a twisting moment is applied to the other, then equal and opposite circumferential loads P are applied to each spring rung at the points of contact with the sides of the slot in which the rung is housed. The rung is maintained in equilibrium by equal and opposite axial loads Q applied by adjacent rungs at the centre of the end loops. Thus, there is no unbalanced end reactions tending to cause axial movements of the spring elements or the coupled members.

This construction overcomes the difficulty which would be experienced in anchoring the ends if individual spring steel strips were used instead of a continuous grid. Furthermore, the grid formation ensures that each rung bears only on one side of each groove for a given direction of torque loading, thus avoiding any tendency for the rungs to become locked in the grooves.

But the most important characteristic of the continuous grid spring is the greatly increased resilience or energy storing capacity which is obtained for a given expenditure of spring material. This high volumetric efficiency reduces the overall dimensions of a coupling of given flexibility. Since resilience is proportioned to the square of the stress, an essential requirement for high volumetric efficiency is that the spring material must be of high quality so that high operating stresses can be carried safely, and it must be stressed as uniformly as possible. The bending moment diagram in Fig. 17 shows that with the grid spring the greater proportion of the looped ends carries the maximum working stress, and only a relatively small proportion of the spring material is understressed. An increase in the energy storing capacity of the coupling is readily obtained merely by extending the overhang of the end loops.

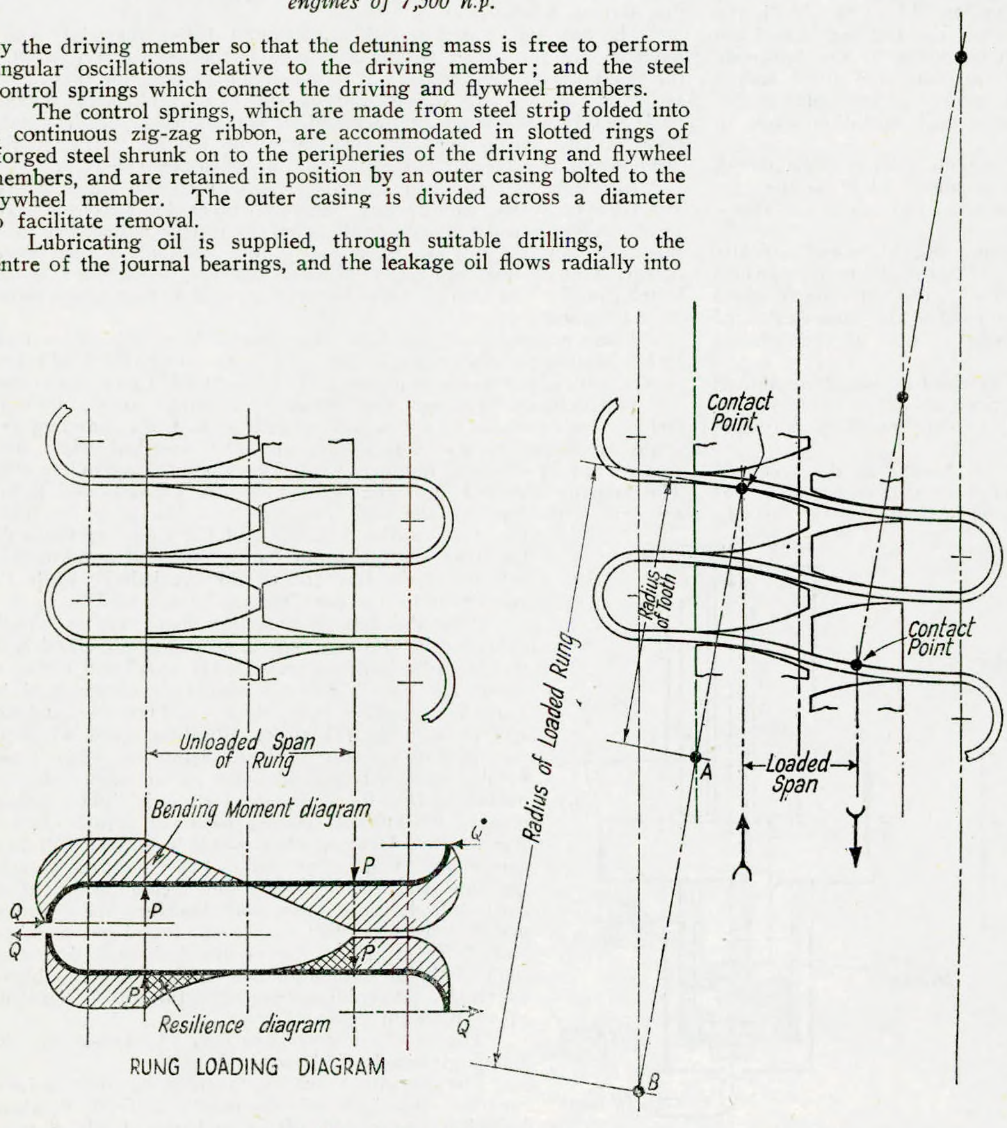

FIG. 17.—*Action of detuner springs.*

With individual rungs, on the other hand, the majority of the spring material is understressed, and consequently the resilience is reduced to the double cross-hatched area shown in Fig. 17.

The action of the grid spring under load is shown in the diagram at the right-hand side of Fig. 17. The flank of each tooth is formed by a radius struck from centre *A*. When one member of the coupling is twisted relatively to the other, the points of contact of each rung move along the respective tooth flanks, thus shortening the loaded span and increasing the torsional rigidity of the coupling.

When the contact points are near the tips of the teeth, the coupling is practically torsionally rigid.

For any given configuration the radius of curvature of the rungs is obtained by projecting a line passing through the point of contact and the centre *A* from which the tooth flank radius is struck until it intersects a line drawn through the centres from which the radii of the end loops are struck. This point of intersection is shown at *B* in Fig. 17, and is the centre of curvature of the rung. Thus point *B* will occupy a different position on the line joining the radii of the end loops for each different configuration of the rungs. For each configuration there are only two points of contact between any individual rung and its co-operating teeth. This implies that the movement of the rungs is a rolling and not a wrapping action, a feature which accounts for the remarkable freedom from wear of grid spring couplings, even when lubrication is indifferent.

From the point of view of the detuning flywheel, the fact that the rungs function by rolling is advantageous because it reduces frictional resistance to a minimum, thus permitting the detuning mass to swing freely through relatively large amplitudes without overheating. In practice the detuners remain quite cool, indicating that frictional losses are small.

A further point to be noticed in connection with the action of a grid spring is that the rung curvature is always greater than the radius of the tooth flank. Hence, if the flank radius corresponds to a spring stress which is within permissible working limits, the spring can never be overstressed. The teeth on one member are sometimes made rather longer than those on the other so that the rungs cannot contact the tips of both sets of teeth simultaneously.

This avoids any tendency to nip the rungs at the centre when occasional overloads are imposed. Only one or two of the earlier Doxford-Bibby detuners had this feature, however, the standard practice over the last nine years being to use the same length of teeth on both the driving and the floating members.

The detuning action of the grid spring is due to automatic change of span, and therefore of torsional rigidity, with change of vibratory amplitude.

It should be noted, however, that the detuning flywheel is only effective for modes of vibration where the torsional flexibility of the controlling springs is large compared with the flexibility of other significant spring elements in the system. This condition can usually be satisfied, however, for any particular mode of vibration by correctly matching the flexibilities.

A detuning flywheel is now a standard feature of Doxford opposed-piston engines having four or more cylinders. In three-cylinder engines the crankshaft is relatively short and stiff, and this enables the oscillating system to be tuned so that the two-node frequency is fairly high. Furthermore, a flywheel is fitted at each end of the engine, and the polar moments of inertia are adjusted so that the crankshaft node for the two-node mode of vibration occurs at or near the centre of the cylinder group, thus cancelling all criticals corresponding to harmonic orders which are integral multiples of the number of cylinders, i.e. the 3rd, 6th, 9th, etc. orders.

In this way it is possible to ensure that only high-order and therefore weak criticals occur in the running range. For example, in a typical amidships installation the lowest order is usually the 10th, while in after-end installations it is usually the 13th. In these circumstances the maximum vibration stress is never much greater than ±2,000lb. per sq. in., so that there is hardly sufficient to be gained to justify the cost of fitting a detuning flywheel on these three-cylinder engines.

With the node at the centre of the cylinder group, the relative vibratory amplitude is the same at both ends of the engine. This has no adverse effect on the camshaft drive—which is situated at the after end—because the amplitudes at criticals in the running range are small.

From the points of view of starting and slow-running, a heavy flywheel is not required on opposed-piston engines having four or more cylinders, and would indeed only serve to make the torsional vibration problem more difficult.

The usual arrangement on such engines is to provide a light turning wheel at the after end of the crankshaft and a detuning flywheel, having about the same total moment of inertia as the turning wheel, at the forward end.

The two flywheels are made no larger than is necessary to comply with mechanical and detuning requirements, and their polar moments of inertia are adjusted so that, with the detuning mass locked to the driving member, the crankshaft node of the two-node mode of vibration is at or near the centre of the cylinder group. This ensures that the two-node natural frequency has the highest practicable value and also provides for cancellation of certain criticals, as previously explained. In this way it is usually possible to produce an oscillating system in which only relatively high order two-node criticals occur in the operating range with stress amplitudes within the permissible limit.

These adjustments lessen the duty required from the detuner, enabling it to be made smaller and lighter than would otherwise be necessary.

Experience has indicated that in the installations under discussion satisfactory detuning action is obtained by proportioning the detuning mass so that its polar moment of inertia is about three per cent. of the total polar moment of inertia of the engine masses. The actual value, however, is adjusted to suit the nearest standard spring assembly, and the suitability of a given mass/spring combination can be roughly checked by calculating the natural frequency of the detuning mass on its control spring, considered as a simple torsional pendulum. This frequency should be of the same order as the two-node natural frequency of the engine system with the detuning mass locked.

In common with dynamic vibration absorbers using linear spring elements, the detuning flywheel replaces each of the original critical speeds by two new criticals one a little above and the other a little below the original critical, i.e. there are two modes of motion of the detuner. In the first mode the detuning member swings in phase and in the second mode out of phase with the driving member of the detuning flywheel. The natural frequencies of both these modes of vibration must be calculated for various values of the control spring stiffness, commencing with the nominal value. The effectiveness of the detuning action can then be judged from the percentage frequency change produced by a given change of control spring stiffness.

Broadly speaking satisfactory detuning action will be obtained if the effect of doubling the control spring stiffness is to produce from 15 to 20 per cent. change in the natural frequencies. Under these conditions the vibratory stresses at all significant criticals is usually reduced to under ±1,000lb. per sq. in., and in some examples the recorded stresses have been under ±500lb. per sq. in. These particularly favourable results are usually obtained in cases where there are no important criticals corresponding to the higher frequency out-of-phase motion of the detuning mass. In such cases the characteristics of the system can be adjusted to favour the lower frequency in-phase motion by sacrificing to some extent the detuning action which would otherwise have been required for the out-of-phase motion.

Where important criticals of both types of motion have to be dealt with it is necessary to apportion the detuning action more or less equally between the two. In such circumstances the results obtained are still very favourable, even though the frequency change when the control spring stiffness is assumed to be doubled, is only of the order of 10 per cent.

The foregoing considerations indicate that the characteristics of the detuning flywheel must be properly adjusted to suit the dynamics of the system as a whole, otherwise a detuner might worsen rather than improve operating conditions.

Returning to the four-cylinder arrangement shown in Fig. 14, *diagram C* shows the final re-adjustment made in 1937 to the twin-screw engines which had given trouble from torsional vibration in 1928. The original arrangement is shown in *diagram B,* and as already stated the trouble was overcome at the time of its occurrence by replacing the heavy flywheel at the after end of the crankshaft by a light turning wheel.

The successful development of the detuning flywheel, however, enabled running conditions to be still further improved. The detuning flywheel could not be installed at the forward end of the crankshaft without considerable trouble and delay. Consequently the light turn-

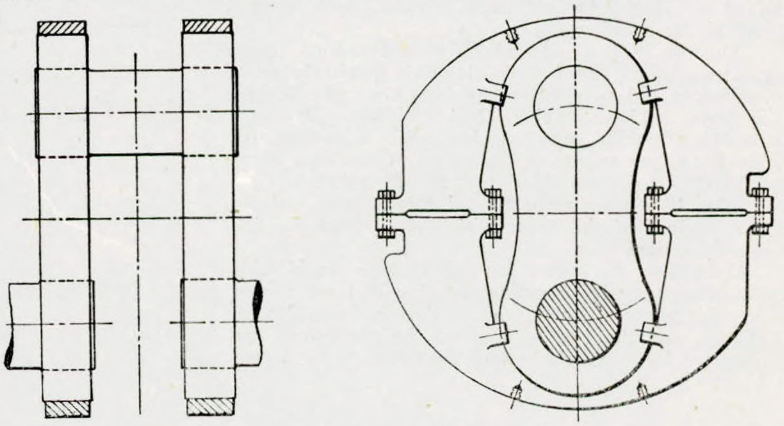

FIG. 18.—*Crankweb flywheel.*

ing wheel at the after end was replaced by a specially designed detuning flywheel with the teeth of the turning wheel on the outer casing, see Fig. 15.

The flywheel was made in halves to facilitate installation, and at the same time a flywheel mass, shown in Fig. 18, was secured to the forward crankweb of No. 1 cylinder to bring the crankshaft node of the two-node mode of the system with the detuning mass locked to a position near the centre of the cylinder group. This reduced the amplitudes of all even-order two-node criticals to negligible values as shown in the vibration stress diagram at the right-hand side of Fig. 14. The movement of the crankshaft node to the centre of the cylinder group had the further advantage of increasing the relative amplitude at the after end of the crankshaft, thus increasing the effectiveness of the detuner.

Torsiograph tests with the detuner in action showed that the amplitudes of the odd-order criticals in the operating speed range had been reduced to values under $\pm 1{,}000$lb. per sq. in.

In 1933 important changes were made in the design of the Doxford engine. The principal novelty was the introduction of electrically-welded fabricated steel construction for the engine superstructure and its supporting framework. The new system proved to be entirely successful and provided substantial reductions of weight and overall dimensions. Additional advantages of the fabricated construction were its greater rigidity, due to the higher modulus of elasticity of steel compared with cast-iron, and its high resistance to shock loads.

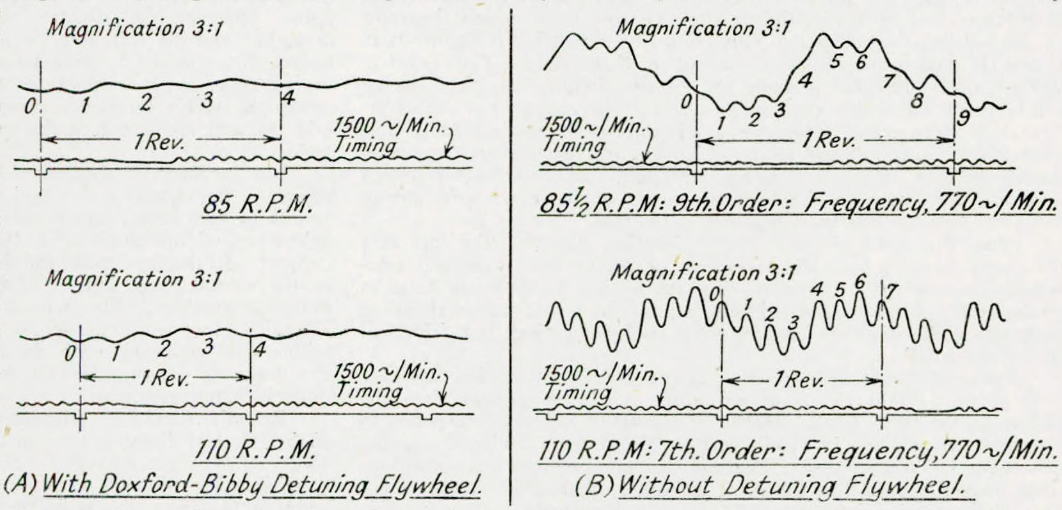

FIG. 19.—*Comparison of torsiograph records from a four-cylinder engine with and without a detuner.*

Diagram D in Fig. 14 shows the crankshaft arrangement for four-cylinder engines of this type. The thrust collar, camshaft and high-pressure fuel pump driving gear, and the turning wheel were all mounted on an extension of the after end of the crankshaft, thus dispensing with a separate thrust shaft and producing an appreciable reduction of overall length. In this connection a comparison of *diagram A* and *diagram D* shows the remarkable reduction which was achieved in the overall length of the crankshaft assembly between 1919 and 1933.

The only flywheel mass fitted to the 1933 version of this engine was a light turning wheel at the after end of the crankshaft. Though this arrangement gave satisfactory operating conditions from the point of view of torsional vibration, the conditions were made even more favourable by the addition of a detuning flywheel at the forward end of the crankshaft in 1937.

The vibration stress diagram at the right-hand side of *diagram D* shows that with the detuner locked there are no really dangerous criticals in the operating range. Nevertheless torsional vibration, even though the peak amplitudes are not in themselves dangerous, does imply engine roughness which causes wear and tear resulting in increased maintenance costs and reduced operational efficiency, which leads eventually to risk of mechanical failure through overstressing from deterioration in service.

Torsiograph records obtained from the forward end of the crankshaft with the detuner working contained no identifiable two-node critical speed at any point in the speed range.

During 1937 a pair of Fairfield-Doxford four-cylinder engines fitted with detuning flywheels were installed in the twin-screw Anchor Line motor ship *Circassia.* These engines had cylinders 725 mm. bore$\times$(1,300+950) mm. stroke, with a rating of 5,700 b.h.p. per engine at 120 r.p.m. On trial a total of 14,000 b.h.p. was developed at 126 r.p.m.

Fig. 19 shows some typical torsiograph records from engines of this type, with and without the detuner in action.

Diagram E shows an alternative crankshaft arrangement used for engines fitted with lever-driven scavenge pump. The pump is driven from No. 2 cylinder, and the two sections of the crankshaft are connected by a short coupling piece which forms the centre journal and is provided with flexible flanges to avoid undue flexural stiffness at the centre of the crankshaft.

Detuning flywheels are also fitted on all five- and six-cylinder Doxford engines, with the result that the torsional vibration stress at all two-node critical speeds is well below $\pm 1{,}000$lb. per sq. in

This enables the crank sequences of these engines to be selected without special regard to torsional vibration. Also, since primary inertia forces are balanced from each cylinder individually, there is no need to select the crank sequence from the point of view of primary balance. The decisive factor is therefore secondary balance, and in the case of five-cylinder engines this leads to the selection of sequence 1-2-4-5-3, since this provides the smallest residual secondary couple. In six-cylinder engines complete secondary balance is achieved with sequence 1-5-3-6-2-4.

Fig. 20 shows typical torsiograph records from a 6,800 b.h.p. six-cylinder engine with and without the detuning flywheel in action. Torsiograph records with the detuner working contain no identifiable two-node critical at any point in the speed range.

No attempt has been made in this paper to deal with the mathematical details involved in the study of torsional vibration problems. An extensive literature has accumulated on this subject and a few selected references are given in the bibliography. During recent years remarkable strides have been made in the theoretical studies, and these have been backed by no less important advances in instrumentation. However, even in relatively straightforward systems, which are substantially linear, much remains to be investigated.

Engine damping, for example, still awaits a satisfactory theoretical treatment, while comparatively recently published work on the coupling of torsional and flexural vibration in aero-engine/air-screw systems has opened up new prospects.

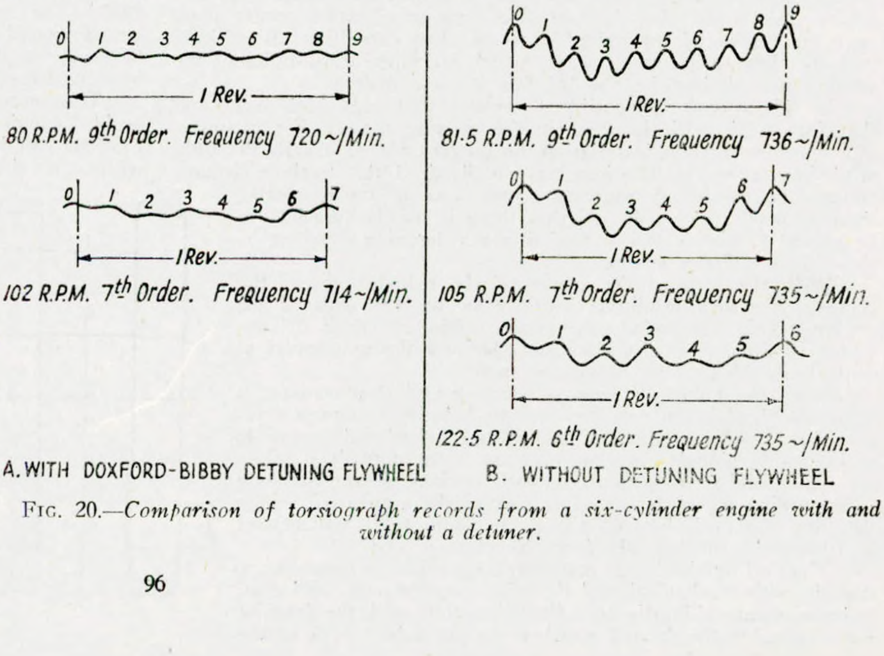

FIG. 20.—*Comparison of torsiograph records from a six-cylinder engine with and without a detuner.*

Acknowledgments.

The author gratefully acknowledges his indebtedness to Messrs. Wm. Doxford & Sons and Mr. W. H. Purdie, M.I.Mech.E., for assistance in preparing the manuscript and illustrations relating to the Doxford engine; and to Mr. James Bibby, M.I.Mech.E. for assistance in preparing the material on the Bibby detuner, and for much helpful criticism.

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[NOTE.—According to information recently obtained from M.A.N., Augsburg, the torsional fatigue strength of full-scale crankshafts, determined experimentally, was about ± 420 kg./cm.^{2} ($\pm 5{,}950$lb. per sq. in.), and M.A.N. practice was to allow a torsional vibration stress of ± 300 to ± 400 kg./cm.^{2} ($\pm 4{,}250$ to $\pm 5{,}650$lb. per sq. in.) under steady state conditions, and $\pm 1{,}000$ kg./cm.^{2} ($\pm 14{,}200$lb. per sq. in.) for transients.]

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Discussion.

Mr. L. C. Leigh (Member), opening the discussion, said that the paper was a very comprehensive study of the subject, clearly and honestly put forward. Each of the problems had been met and overcome, and the present-day Doxford engine was the product of many years' patient toil and development. The most important feature of this type of engine was the fact that the entablature was relieved of gas load. The scantlings could be determined by constructional limitations. Each cylinder was inherently in primary balance, and the upper and lower piston gas loads did not impose any bending moment on the crankshaft element.

The stiffness of a crankshaft was built up of combined torsional and flexural rigidities of the webs, crank pins and journals. Any crankshaft, if called upon to deliver pure torsion—that was, relieved entirely of gas, inertia and centrifugal loads—would deflect in greater or lesser degree according to those rigidities. All parts that made up the crankshaft would both bend and twist. The main bearings would tend to lag behind one another and would impose bending and torque reactions on the bearing housings.

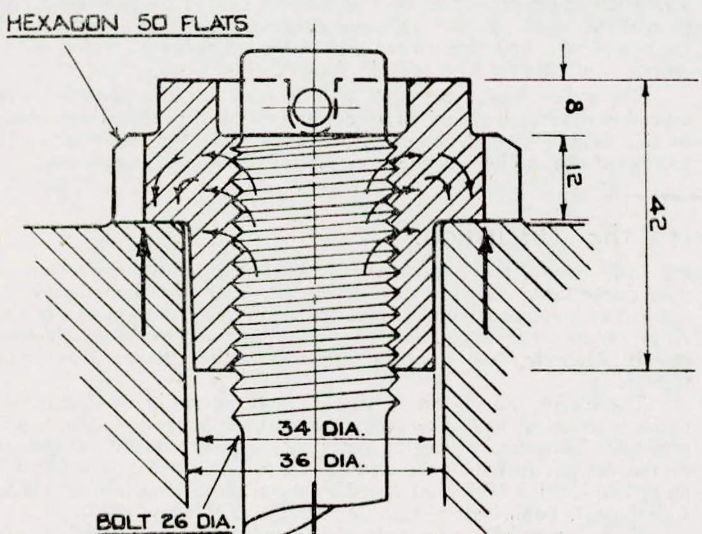

FIG. 21.—*Elastic type nut.*

He therefore agreed with the author that spherical bearings were a desirable feature in the Doxford engine where the flexibility changed. If connecting-rod big-end bolts were strong and the housing weak, the whole of the impulse loading would be carried by the bolts. The converse was also true. Hence the use of stretching length to provide this flexibility; but the bearing housings must be rigid to resist the shaft reactions. In support of this view, the speaker pointed to the same state of flexibility change observed in bolt threads and nuts. Fig. 21 showed a form of elastic nut, the object of which was to distribute the bolt load along the threads more evenly. It would be noted that the nut was provided with a circular portion projecting below the nut reaction face. This lower portion could yield as the bolt stretched in taking up the load and prevented concentration in the bottom threads.

He drew attention to another feature of the Doxford engine; that the axes of the side rod ends were at right angles. This gave some degree of universality of movement, as the top cross beam could compensate for length changes, and the crosshead end could accommodate slipper movements.

He opposed on principle the attainment of dynamic balance by increase in engine weights. He would much prefer to see it achieved, if possible, by the reduction of the masses. He was aware that in two-stroke engines the inertia loads tended to level out the gas pressure intensities. Nevertheless, in a crosshead engine running at 100-115 r.p.m. the maximum surface speeds and loadings of the principal bearings were approximately half those in medium speed engines at 600-1,000 r.p.m. Apart from other factors, this would have the immediate benefit of increasing the two-node natural frequency.

The shaft layouts in Fig. 14 clearly showed the enormous development that had taken place since 1919. The length of the engine had been reduced by over 30 per cent.—no mean achievement. The shaft length had been reduced, and the disposition and magnitude of the principal harmonics had been much modified. He noted that the fuel pump and camshaft chain drive were located at the after end of the crankshaft. He asked whether, seeing that the engine node was centrally situated, there were any objections to the central position, or any alternative positions for the drive.

The author had also shown the form of the torsiograph picture with the Bibby detuner fitted to the free end of the engine crankshaft. This was undoubtedly a most commendable achievement. When the figures obtained with the detuner were compared with those obtained without it, it was amazing to reflect on the enormous technical advance that had been made during the last ten years by the introduction of this appliance which was the acme of simplicity.

He asked the author to give a few engine weights, because all these new construction methods with fabrication had introduced reductions in size, length and weight.

In thanking the author for his paper, the speaker remarked that troubles were not often spoken of with such freedom or described with such analytical clarity.

Mr. C. H. Devitt (Visitor) described some improvised tests on the crankshaft of an engine at sea. Having no instruments, the engineer officers decided to work with glasses of water placed at three points in the ship, until they found the setting which gave the smoothest possible running. The results had been very similar to those given by the torsiograph. At 75 revolutions there was nothing wrong, as far as could be seen, with the behaviour of either the ship or the engine, but when the revolutions were increased to 82, the masts nearly came out and the funnel nearly fell overboard—the vibration almost shook everything in the ship to pieces. They had run the engine for several hours and had known exactly which old bolts to take out and renew. After a great deal of experiment they had found a good running speed between 86 and 88 revolutions and had afterwards had very little trouble.

None of the staff, although they were highly qualified, had known much about vibration. He pleaded for the better instruction of the young engineer, so that when he went to sea he would not only understand what the red line was for but what happened when it was crossed. Unluckily many of these data were locked away in the berth of the chief engineer; they should be available to everybody, if only because the chief might be in bed when they were badly needed.

Many engineers worked at various times with Doxford engines, but few of them knew much about their dynamics. This, he thought, was heading for disaster, because if an engine was not properly designed on the drawing board the engineer would be confronted with all sorts of complicated problems.

Mr. R. Coombs (Visitor) confirmed the great interest that was felt by most marine engineers in the subject of torsional vibration, and how little was generally known about it. A speaker had recently said on the wireless that even truth, if put clearly enough, would be believed. The Institute had heard a good deal of truth that evening.

Several years ago he had had the pleasure of taking one of the Doxford oil engines to sea, and his ignorance and that of his colleagues of torsional vibration had been very great, but the trouble had fortunately developed after he had left the ship. The keep of No. 1 cylinder had continually broken, and some time after that the upper pistons had started to break just near the gudgeon. The engineers had not known whether this had been due to torsional vibration or not and, in fact, had not cared, but had merely replaced the parts.

He asked for some guidance on how to tell whether a trouble was due to torsional vibration or to the ordinary troubles such as bending for which marine engineers looked.

Mr. T. A. Bennett, B.Sc. (Member of Council), proposing a vote of thanks to the author, testified to the difficulty of the troubles with which the lecturer had contended, and said he was sure they must have given him many headaches but also a great deal of satisfaction in their solution. The figures he had given for the final balancing of the Doxford engine showed the magnitude of the work he had achieved. His qualifications, though impressive, showed that he had not reached the summit of his profession, for he was not yet a member of the Institute. No doubt this was only a matter of time.

The vote of thanks was accorded with acclamation.

BY CORRESPONDENCE.

Mr. J. A. Rhynas (Chairman of Council): The author quoted a figure of eleven years as the life which was expected from each liner. It would be interesting to learn if he had any knowledge of a liner having been in service for that length of time and, if so, the extent of the cylinder liner wear.

In connection with vibration, the writer recently had a vibration problem in connection with a new Diesel generator installed in a ship, which he must admit was not solved yet. On starting up the engine the vibration was terrific, and it was therefore deemed advisable to stop the machine and investigate. Nothing amiss could be found. All holding-down devices were sound and the seatings and foundations as a whole had adequate strength. Further, there was no alteration in the loading of the vessel. Another start of the engine was made and the vibration was scarcely perceptible at full load. Later on, a further start of the machine was made and the vibration was as bad as ever, and so it went on. It would be interesting to have the author's opinion on this matter.

Mr. R. C. Manley, B.Sc.: The paper combined a very interesting historical account of the development of opposed-piston engines with a masterly summary of the main features of vibration problems in marine engineering. The latter was remarkable for the complete absence of the mathematical formulæ which normally made the study of vibration literature unnecessarily difficult, and the author had shown that it was possible to give an adequate critical discussion of the vibration problem in plain English.

The writer noted the absence of any specific reference to dynamic unbalance in the description of propeller excitations, and would like to know whether dynamic balancing was generally unnecessary, as might be implied by the statement that "mass unbalance can usually be controlled by statically balancing the propeller".

In the description of the various modes of propeller vibration, attention might usefully be drawn to the fact that, except in the reactionless modes, the severity of the resulting vibrations was affected by the inertia/flexibility properties of the shaft system, as well as by the actual dissipation of energy through damping in that system.

The author twice referred to the question of critical speeds below the operating speed range, which had to be negotiated in approaching or leaving the running condition. Presumably the severity of stresses caused by accelerating or decelerating through a critical speed was somewhat lower than that of the stresses caused by prolonged running at that speed, as the resonance effect did not have time to build-up completely, and the writer would be interested to know if the author could supply any reliable data on this point.

The paper would be read with interest by all connected with vibration engineering, and as an account of hard practical experience in this field it should be prescribed as a tonic for those who tend to regard the subject as the preserve of theoretical specialists.

The Author's Reply to the Discussion.

The author was in complete agreement with Mr. Leigh's remarks on the design considerations which endowed an engine structure with a special capacity for resisting dynamical loadings, and he was particularly pleased to observe that Mr. Leigh had come to the conclusion that many of these desirable features were to be found in an opposed-piston arrangement.

Mr. Leigh's comments on the loading of connecting-rod bolts were especially valuable, since it was important to appreciate that flexible bolts in a relatively rigid housing were essential for satisfactory performance under pulsating loads. With such a combination it was only necessary to make sure that the initial tightening of the bolts was in excess of the maximum value of the applied load—steady load plus maximum value of the pulsating load—to ensure that the bolts would be practically free from the fluctuating stress which was the main cause of fatigue failures. With relatively rigid bolts in flexible housings, on the other hand, a large proportion of the fluctuating stress would be experienced by the bolts. It was important in this connection, however, to appreciate that the bolts must be adequately tightened initially and it was no exaggeration to say that most failures of highly-stressed bolts, carrying loads which were mainly dynamic, had occurred through under- rather than over-tightening.

The elastic nut shown in Fig. 21 was interesting and appeared to be a practical embodiment of the uniformly-stressed design proposed by Timoshenko several years ago. Some useful information on nut design—including the type shown in Fig. 21—was contained in an article entitled "Effect of Nut Design on the Strength of Threaded Fastenings", published in *Machine Design*, February, 1943.

Everyone would agree with the soundness of Mr. Leigh's principle that dynamic balance should be achieved if possible without any increase of engine weight, and it would appear that an opposed-piston assembly came nearest to the realisation of that ideal. Apart from the desirability of minimizing structural weight, the addition of supplementary masses to achieve engine balance could have an adverse effect on other dynamical characteristics.

For example, several cases were on record where crankweb balance weights fitted to single-piston engines had to be removed shortly after the engines went into service, because of serious torsional vibration trouble.

In this connection there was no doubt that experience had taught a valuable lesson in indicating that the mere piling on of weight was not only uneconomical but might even fail to cure troubles originated by dynamic loadings. The evolution of the framing of the Doxford engine, shown in Fig. 9, afforded an interesting example of the way in which reduction of structural weight could be accompanied by improvement of dynamical characteristics. The measured natural frequencies of these different frame constructions and the critical speeds of transverse vibration of the framing, *i.e.* the 4th order criticals in the case of four-cylinder engines, were given in the following table.

Transverse Vibration of Engine Framing.

(Four-Cylinder Engines).

Year. (See Fig. 9).	Natural frequency of transverse vibration, cycles/min.	4th order critical speed, r.p.m.
1919	500	125
1926	380	95
1927	400	100
1929	440	110
1933	520	130

Since the natural frequency of transverse vibration was a measure of the rigidity of the engine structure, the values given in the table indicated that the fabricated steel construction of the 1933 engine, which had a total weight of 230 tons, was as rigid as the heavy cast-iron construction of the 1919 engine which had a total weight of 425 tons.

With regard to the position of the fuel pump and camshaft chain drive, it was undoubtedly true that considerable benefit could be derived from placing this drive at or near a nodal point in engines which were fairly lively from the point of view of torsional vibration. In the case of the engines under discussion, however, the torsional vibration problem had been solved so effectively that there was a free choice in positioning this drive, and the after end location had been selected purely from the point of view of installation convenience.

In response to Mr. Leigh's request for a few engine weights, the following table gave some figures for engines fitted with the crankshafts shown in Fig. 14.

Engine Weights.

Ref. in Fig. 14.	Engine rating.		Framing construction.	Total weight of engine.		*Weight of crankshaft.
	b.h.p.	r.p.m.		Tons.	lb./b.h.p	Tons.
A (1919)	2,700	77	Cast iron	425	350	72
B (1927)	3,300	98	Cast iron	360	245	56
C (1937)	3,300	98	Cast iron	355	240	50
D (1937)	4,000	120	Fab. steel	237	133	48
E (1937)	3,000	95	Fab. steel	230	174	45

*Includes flywheel, thrust shaft, etc., as shown in Fig. 14.

Incidentally, the space occupied by present-day engines of this type was from 2 to 3 cu. ft. per b.h.p., which was rather less than some recently-published figures for the space occupied by marine gas-turbine installations (see *Gas and Oil Power,* October, 1945, page 347).

Mr. Devitt's contribution to the discussion was a reminder that the solution of difficulties of the kind described in the paper depended greatly on the co-ordinated efforts of all concerned, whether their duties lay in the workshop, or in the office, or at sea. Mr. Devitt himself had considerable experience of these problems and he therefore spoke with authority.

There was no doubt that the diagnosis of a vibration trouble—which must precede any attempted cure—was greatly facilitated if reliable information was available on which to base a judgment, and no one was in a better position to collect this data than the engineer in direct daily contact with the troublesome machinery. In many cases considerable insight was obtained by making two relatively simple measurements, namely, the frequency of the troublesome vibration and the revolutions per minute at which this vibration reached its peak value. These measurements enabled the investigator to determine the number of vibration cycles per revolution of the main or auxiliary engines, and from this knowledge it was usually possible to obtain a shrewd idea of the source of the trouble. A few examples were given in the following table.

Diagnosis of Vibration Troubles.

	Frequency of vibration.	Possible source of vibration.
Main Engine.	0·5 × r.p.m.	Uneven combustion in the various cylinders: four-stroke engines only.
	1 × r.p.m.	Uneven combustion in the various cylinders: two-stroke engines only. Primary static or dynamic unbalance of the moving parts. Misalignment of a coupling. Bent shaft due, for example, to the wear down of a bearing.
	2 × r.p.m.	Secondary static or dynamic unbalance of moving parts.
	Number of cyls. × r.p.m./2.	Power impulses: four-stroke engine.
	Number of cyls. × r.p.m.	Power impulses: two-stroke engine.
Propeller.	1 × r.p.m.	Static, dynamic, or hydrodynamic unbalance.
	Number of blades × r.p.m	Tip clearance too small. Irregular flow to propeller disc.
Auxiliary Engine.	See main engine.	See main engine.

Although the examples given in the table by no means exhausted the possibilities, they did serve to show that a knowledge of the fundamentals of vibration theory could be a useful aid in the diagnosis of some of the simple everyday ailments to which running machinery was subject.

Incidentally there were available a number of simple and efficient mechanical instruments for measuring vibration frequencies with sufficient accuracy for making a preliminary vibration analysis. Indeed, it was not very difficult to make and calibrate a vibrating-reed type instrument which would detect vibration within the frequency range usually encountered on board ship, and a "home-made" vibrograph of this type was described by Professor Burrill in his paper "Ship Vibration: Simple Methods of Estimating Critical Frequencies", Trans. North-East Coast Institution of Engineers and Shipbuilders, Vol. LI, 1935.

In reply to Mr. Coombs, it could be said that the nature of a fracture usually conveyed a considerable amount of information to an experienced observer. Broadly speaking, failures resulting from vibration produced "fatigue" fractures which were characterized by absence of plastic deformation at the broken surfaces, and by the presence of a series of concentric markings which indicated the progress of the crack from its initiation to final rupture.

These markings converged towards the nucleus from which the crack had originated—usually a point of severe change of section, or a part of the surface where there was a mechanical defect such as a tool mark or other indentation, or a metallurgical defect such as an inclusion.

In the case of failures due to steady loadings the fractured surfaces usually showed considerable plastic deformation. These distinguishing features could be kept in mind by remembering that in a steady tensile test on a material, the specimen usually suffered considerable deformation by "necking" before rupture occurred. In a fatigue test, on the other hand, the specimen showed practically no deformation at the broken surfaces.

Some valuable information on the nature of fatigue failures was contained in an article by the late Professor Frederic Bacon entitled "Cracking and Fracture of Metals with Special Reference to Service Breakages", published in *The Iron and Steel Industry*, March and April, 1934.

The question whether the failure was due to torsional vibration or some other cause was sometimes more difficult to answer, particularly in the case of engine crankshafts where both torsion and flexure were known to be causes of fatigue failures. Usually, however, there were indications apart from those afforded by the fractures themselves, which enabled the experienced observer to trace the true cause.

With regard to the particular trouble mentioned by Mr. Coombs, there was no record of any case of a broken piston-rod keep. It was possible, however, that the incident mentioned was one of a few failures of the lower half bearing on the upper piston rod, which were experienced on some of the earlier engines. These engines were fitted with the original type of transverse beam and there was a large spigot to centralize the lower half of the transverse beam centre bearing on the upper piston rod. This spigot cut away a great deal of the bearing surface directly underneath the transverse pin, with the result that the metal spread due to the excessive bearing stress and allowed water to enter, thus initiating erosion. In addition, the bearing was too thin between the face and the bore. This trouble was cured by reducing the spigot diameter from 100 mm. to 40 mm.

In reply to Mr. Rhynas's query regarding the life which was expected from each cylinder liner, records, which had been very carefully maintained, showed that in two or three vessels built between 1937 and 1939 the liner wear was about 0·4 mm. per annum. It was therefore anticipated that the wear should be from 4·5 to 5 mm. after 11 years' life. One of the vessels was now in her tenth year and the liner wear had not yet reached 4 mm.

The vibration problem mentioned by Mr. Rhynas sounded very interesting, but it was impossible to attempt an analysis of the trouble without a great deal more information than was given in his description. In problems on mechanical vibration, the avoidance of resonance was the chief pre-occupation of the vibration engineer, and in marine installations there were—in theory—many possibilities in that direction. Some remarks on the diagnosis of vibration troubles had been given in the reply to Mr. Devitt, while some examples from practice were contained in the paper itself. In addition, the following table might be useful as giving a list of the more important possibilities.

Critical speed diagrams of the type shown in Fig. 22 were very

Resonant Vibration—Marine Installations.

Type of vibration.	Possible excitation.
HULL.	
2-node vertical. 2-node horizontal. 3-node vertical. 3-node horizontal.	Primary unbalance of main and auxiliary engines. Secondary unbalance of main and auxiliary engines. Mass or thrust unbalance of propeller. Excitations at propeller blade frequency.
1-node torsional.	Engine torque reaction. Primary unbalance of twin-screw main engines. Secondary unbalance of twin-screw main engines. Mass unbalance of twin-screw propellers.
ENGINE.	
1-node shaft torsional 2-node shaft torsional	Integral order harmonic components of engine torque curve: two-stroke cycle engines. Integral and half-order harmonic components of engine torque curve: four-stroke cycle engines.
Axial vibration of shafting.	Harmonic components of propeller thrust curve. Harmonic components of radial forces on crankpins. Coupling with torsional vibration of crankshaft.
Transverse vibration of engine frame.	Harmonic components of side thrust at engine crossheads.

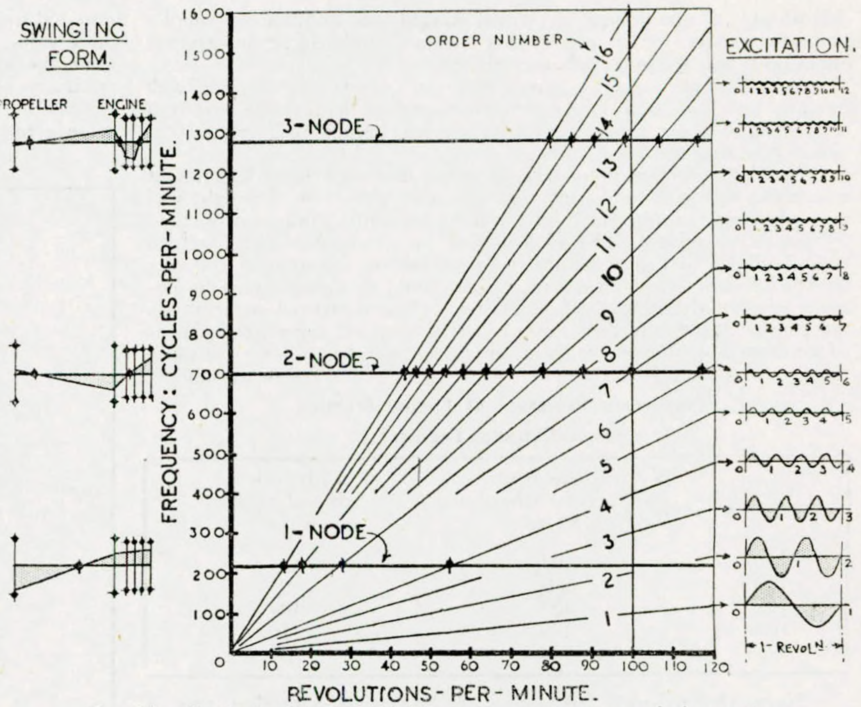

FIG. 22.—*Typical critical speed diagram—marine transmission system.*

useful for affording a "bird's eye" view of the possibilities of resonance.

That particular diagram was drawn to represent the torsional vibration characteristics of a four-cylinder two-stroke cycle marine oil engine installation. The base line represented revolutions per minute and the ordinates represented frequency. The heavy black lines drawn parallel to the base line represented the natural frequencies of torsional vibration of the shaft system with one, two, and three nodes, the swinging forms for each of the modes being sketched at the left-hand side of the main diagram.

The sloping lines radiating from the origin represented the various excitations which, in the case of a two-stroke cycle engine, had excitation frequencies equal to all integral multiples of the revolutions per minute.

Thus, with the engine running at 100 r.p.m., the excitation frequency of a first order excitation was ($1 \times$ r.p.m. $= 100$ cycles/min.); second order ($2 \times$ r.p.m. $= 200$ cycles/min.), and so on.

Theoretically, resonance occurred at each point where one of the excitation lines crossed a frequency line. Thus, there were a great many possible resonant or critical speeds, as indicated in Fig. 22. Fortunately, many of these were of no practical significance, either because they occurred outside the operating speed range or because the accompanying vibration was very small. A knowledge of the characteristics of the installation usually enabled the vibration engineer, from a background of previous experience, to make a shrewd estimate of the particular resonances which were likely to be troublesome. Diagrams of the type shown in Fig. 22 could be extended to include other modes of vibration and other sources of excitation merely by drawing these additional frequency and excitation lines in their appropriate places.

The particular problem mentioned by Mr. Rhynas appeared to be rather elusive since the results were not consistent from one occasion to another. This apparently erratic behaviour might, however, be due to a change in operating conditions between one trial and the next. For example, engine roughness could be experienced in an apparently haphazard manner if the fuel injection equipment was not in proper order, so that combustion varied from one cylinder to another; or if the cylinders were worn to the extent that there was a variation of compression from one cylinder to another. In this latter event the variation of compression might tend to be most marked at starting and would tend to even up when the engine reached its normal operating temperature. Faulty piston rings could, of course, produce a similar effect. Another possibility was that there was a loose coupling somewhere in the shaft system, or a flywheel loose on its shaft. This would introduce some non-linearity into the system so that there might be instability at certain speeds.

Mr. Manley raised one or two interesting points. With regard to propeller balancing, the usual practice in this country with high-class propellers was to carry out a static balancing operation, using conventional knife-edge equipment. In America, dynamical balancing was practised by the U.S. Navy. Over here, it was considered that the production of propellers with blades which were as nearly geometrically similar as possible was a better safeguard against vibration than a dynamic balancing operation, which in any case would have to be carried out in air so that the effect of entrained water would not be taken into account. Occasionally, however, the small diameter propellers used on high-speed craft were dynamically balanced.

In this connection it was of interest to recall the fact—mentioned in the paper itself—that experience with airscrews had indicated that errors in the geometrical characteristics of individual blades were a much more potent source of vibration than errors of mass balance, and this appeared to be a vindication of the policy of confining balancing operations on marine propellers to a simple static balance.

With regard to Mr. Manley's remarks on the various modes of propeller vibration, it was now well-known that in aero-engine-air-screw systems there was considerable coupling between the significant airscrew flexural and engine torsional vibrations, so that it was necessary to take airscrew blade flexibility into full account when determining the overall vibration characteristics of the system.

In the case of marine installations, however, the significant propeller blade frequencies were so much higher than the significant frequencies of the engine shaft system that for practical purposes the torsional vibration characteristics of the engine system could be investigated without regard to propeller blade flexibility, *i.e.* the propeller could be regarded as a rigid flywheel having the same polar moment of inertia. Similarly the propeller blade vibration characteristics could be investigated without regard to the engine masses.

Very little appeared to be known in this country about the severity of the vibration experienced when an engine was accelerated or decelerated through a critical speed which was otherwise outside the normal operating range of speeds. This problem, however, appeared to have received some attention on the Continent where it was customary to allow appreciably higher stresses under transient conditions than under steady state conditions. The practice of some of the leading Continental manufacturers of oil engines was to allow a torsional vibration stress of $\pm 4{,}000$ to $\pm 5{,}000$lb./in.^{2} for steady state conditions and $\pm 14{,}000$lb./in.^{2} for the transient condition experienced during starting and stopping, for shafts made of 35 to 45 tons/in.^{2} steel.

The diagram shown in Fig. 23 was a torsional fatigue curve, published by Lehr and Ruef(16), for the full-size crank element shown. Control tests carried out on plain specimens of the crankshaft material gave a torsional fatigue strength of $\pm 22{,}000$lb./in.^{2}, and this indicated a significant stress concentration factor for this particular crankshaft design of about $22{,}000/6{,}000 = 3{\cdot}66$.

The curve showed that the number of cycles for a stress of $\pm 14{,}000$lb./in.^{2} was about 800,000. This figure would enable the life of the engine under transient conditions to be determined if the frequency of manœuvring and the time required to negotiate the critical were known. As a general rule, the time required to accelerate through a critical was only from one-half to one-third the time required for deceleration, but the figures were best determined by actual trial.

Assuming, however, that in a particular case it required 2·5 secs. to accelerate and 7·5 secs. to decelerate and that the natural frequency of the relevant mode was 30 cycles per sec., then the number of cycles in the neighbourhood of the transient during each manœuvre was 300. Thus, the number of manœuvres which could be completed

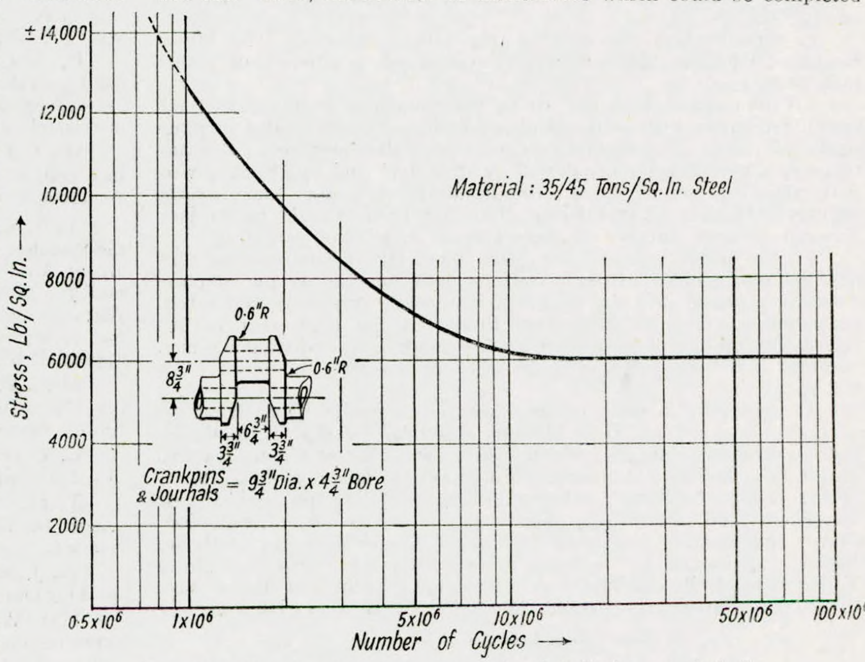

FIG. 23.—*Torsional fatigue strength of full-size crankshaft.*

before there was danger of failure due to traversing a transient having a peak stress of $\pm 14{,}000$lb./in.^{2} was 800,000/300 or, say, 2,700 manœuvres. Assuming also that two manœuvres were executed every week, then the expected life under these conditions would be 1,350 weeks, or 26 years!

The foregoing estimate was rather pessimistic because it assumed that the full peak stress would be reached and retained during the whole of the transient period. No reliable data on the peak stresses actually attained under transient conditions had been traced. It was possible, however, that in the case of really severe transients the peak value was reached, particularly during deceleration when the engine tended to dwell in the critical zone.

ADDITIONS TO THE LIBRARY.

Presented by the Publishers.

British Standards Institution.

PD. 477. Amendment No. 2 to B.S. 638: 1941—Welding Plant and Equipment.

PD. 497. Amendment No. 4 to B.S. 907: 1940—Dial Gauges for Linear Measurement.

PD. 476. Amendment No. 1 to B.S. 1042—Code for Flow Measurement.

PD. 500. Amendment No. 1 to B.S. 659: 1944—Light Gauge Copper Tubes.

PD. 496. Amendment No. 1 to B.S. 693: 1940—Oxy-Acetylene Welding in Mild Steel.

Transactions of the Liverpool Engineering Society. Volume LXXI. Session 1944-45.

Journal of the Institute of Petroleum. May, 1946, Vol. 32, No. 269.

The Journal of the Institute of Metals. May, 1946, Vol. 72, Part 5, and Metallurgical Abstracts, Vol. 13.

Forming of Aluminium Alloys by the Rubber Die Press. A.D.A. Information Bulletin No. 11, 32pp., illus., price 1s. Aluminium Development Association, 67, Brook Street, London, W.1.

Plastics Scientific and Technological. (Second edition). By H. Ronald Fleck, M.Sc., F.R.I.C. English Universities Press, Ltd., London, 1946, 361pp., 90 illus., 30s. net.

When a second edition of a book on plastics makes its appearance so soon after the first, one assumes that this has been impelled by the rapidity with which the field has expanded and that the opportunity has been seized to give prominence to the most recent developments. Such expectations cannot be said to have been fully satisfied by the second edition of *Plastics Scientific & Technological.*

New sections have been written on polymerisation reactions, polythene, polyvinylidene chloride and "heatronic" moulding. The data on melamine plastics have been transferred from the addenda to the main body of the book and the chapter on molecular weight determination has been completely recast, giving due attention to ultra-centrifugal methods. On the other hand, there is still no mention of recent additions to the plastics field such as allyl plastics, polychlorstyrenes, aniline-formaldehyde and furane resins whilst the section on silicones is unchanged, these products still being described as being mainly of theoretical interest.

As far as it goes, this work remains a very creditable effort to portray the plastics industry to the general reader. The early chapters on the chemistry of plastic materials show evidence of much painstaking effort. It is all the more regrettable that inaccuracies still appear in the text, *e.g.* the boiling point of styrene is given correctly in one section (p. 98) but incorrectly in another (p. 143), the softening point of polyvinylidene chloride is given as 185 - 200° C. (p. 108) whereas it is closer to 80 - 100° C. and it is still stated that casein is formalised in 40 per cent. formaldehyde solution (p. 129).

This book is useful as a broad introduction to the subject and for general reading, but is too condensed and insufficiently documented to be of great value to the expert.

The Motor Boat and Yachting Manual. (13th edition, 4th impression, 1946), by the Staff of "The Motor Boat and Yachting". Temple Press Ltd., Bowling Green Lane, London, E.C.1, 271pp., profusely illus., 7s. 6d. net.

In some respects the present (the 13th) edition of "The Motor Boat and Yachting Manual" may be considered as a new book rather than a revision.

All the engines described are of the types now manufactured and new illustrations with sectional drawings have been provided in practically all cases. The section on transmission has been re-written and there is a new chapter on British boatbuilders and repairers, giving particulars of practically all the builders of motor boats in the country. The list of motor-boat clubs has been brought up to date, although a large number of them are not at present operating.

All the other chapters have been completely overhauled and particulars are included of such modern developments as the employment of plywood and the adoption of prefabrication in motor-boat construction. It is believed, therefore, that the book represents a thoroughly up-to-date exposition of the motor boat and its machinery and that it will be of value to all who are interested in motor boats and yachts.

The nineteen chapters of the Manual deal with: First Principles of Boat Construction; Hull Design; Practical Details of Boatbuilding; Making the Moulds; Metal Hulls—Small Steel Craft; Typical Motor Boat Design; High-speed Motor Boats; Naval and Air Force Motor Boats; Navigation of Small Craft; Sailing and Sail Plans; Propellers and Propulsion Systems; Transmission; Installation; Petrol and Paraffin Engines; High-speed Diesel Engines; Outboard Motors; Outboard Motor Boats; Navigating Gear for the Motor Yacht; British Boatbuilders and Repairers; Yacht and Motor Boat Clubs, Associations for Private Members—Trade Associations.

MEMBERSHIP ELECTIONS.

Date of Election, 4th June, 1946.

Members.

Thomas Barrow.
Alexander MacDonald Bryan.
Alexander Campbell, B.Sc.
James Campbell.
Edmund Francis Kay.
John Grant Knowlton, A/Commodore(E.) R.C.N.
Trevor Owen Morris.
George Peddie.
David Mathewson Reid, Lieut.(E.), R.N.R.
Sidney Donald Shirley, Lieut.-Comdr.(E.), R.N.R.
Charles William Thomas.
John White, D.S.C., Comdr.(E.), R.N.
Kuldip Singh Oberoi.
Richard Redwood, M.B.E., Lieut.(E.), R.N.V.R.
Wilfrid Arthur Rhodes.
William Gerald Rhodes.
John Clive Thomason, Lieut.(E.), R.N.R.
Frederick Henry Warren.
Aleksander Wilcyznski, Lieut.(E.), P.N.

Associate Members.

Leslie Arthur Gotman.
Geoffrey Briscoe Penn, Lieut.(E.), R.N.

Associates.

Ronald Napier Arundel.
Geoffrey Joseph Atkins.
Arthur Buckley.
Robert Gardner Donoghue.
Henry John Godfree Giles.
Alexander Hamill.
Donald Eric Jameson.
Thomas Francis Johnson.

Graduate.

Alexander Dennis Taylor.

Student.

Kenneth Douglas Ayton.

Transfer from Associate Member to Member.

John Jeffrey Reed, Lieut.-Comdr.(E.), R.N.R.

Transfer from Associate to Member.

Frederick Bourne.
William Maddock.
James Rennie.
Albert Eric Robinson.

Transfer from Student to Associate.

John Patrick Niall Fox, Lieut.(E.), R.N.

PERSONAL.

T. E. Aitchison (Associate) has joined Messrs. Aitchison, Blair, Ltd., as a draughtsman.

R. Boyes, Jun'r. (Member) has been released from the Navy and has resumed his appointment as technical representative with The Vacuum Oil Co., Ltd., at Birmingham.

R. S. Brett (Associate Member) has accepted an appointment as marine representative of The Texas Oil Co., Ltd., for the North-East Coast area.

M. W. Butchart (Member) has been appointed engineer to the Hawarden & District Waterworks Co., near Chester.

E. N. Cady (Associate) has been appointed an engineer surveyor with the Ocean Accident & Guarantee Corporation, Ltd.

R. A. Collacott, B.Sc., Ph.D. (Associate Member of Council), has been released from his commission in the Royal Air Force and has been appointed by the Shell Petroleum Company as technical adviser, in control of research work in the application of lubricating oils.

Ian Cowie (Member) has now been released from the R.N.R. and has resumed his appointment as engineer surveyor to The Ocean Accident & Guarantee Corporation, Ltd.

Arthur E. Crighton (Vice-President) is retiring in August from the position of chief superintending engineer and naval architect of Royal Mail Lines, Ltd. after 36 years' service with that Company and the former Royal Mail Steam Packet Company, including 29 years in his present capacity. He is being retained as consultant to the Company. H. J. Wheadon (Member of Council) has been appointed to succeed Mr. Crighton on his retirement.

Miss V. A. Drummond, M.B.E. (Associate Member), has gained the Motor Endorsement of her M.o.W.T. Second-Class Steam Certificate of Competency.

E. Ellison (Member) has now returned to Hong Kong as surveyor of ships in the Harbour Dept.

T. H. Forbes (Member) has returned to Rangoon from Simla, where he had been carrying on his duties in the Burma Marine Service.

W. J. Hicks (Associate) has been appointed a technical assistant by The Dunlop Rubber Co., Ltd.

Maj. W. A. Hutcheon (Associate Member) has been appointed superintendent engineer with The United Africa Co., Ltd., on his release from the Army.

Lt.(E.) J. F. Ingham, R.N.R. (Member), has recently been demobilized from the Royal Navy.

A. M. Keith (Member) has been appointed an engineer surveyor with the Ministry of Works.

W. McDonald (Member) has been appointed mechanical engineer at the Burhar Colliery, India.

A. P. Miles (Member) has entered the service of Bureau Veritas as a surveyor in the United Kingdom.

Com.(E.) J. J. Murray, R.N.V.R. (Member), nautical adviser to the Government of Burma, has been awarded the O.B.E.

Captain R. C. Petter (Member) has resigned his position as managing director of British Oil Engines (Export), Ltd., but will remain on the board and act in an advisory capacity.

D. Rebbeck (Member) has been appointed to a seat on the board of directors of Messrs. Harland & Wolff, Ltd., and has obtained the degree of B.Litt. (Dublin), by thesis, at Trinity College, Dublin.

J. Roberts (Member) has taken up an appointment as fuel economy officer in the Notts area with the Ministry of Works.

P. J. Sims (Member) has been re-elected a member of the Council of the Society of Motor Manufacturers and Traders, Ltd.

W. H. Steer (Member) has taken up an appointment with the Ministry of Works.

J. O. Thomson (Member) has recently entered the service of the Ministry of Works.

R. R. Waddington (Associate), recently released from service as Lt.(E.), R.N.V.R., has become the London representative of Messrs. Ruston & Hornsby, Ltd. (Marine Division).

J. A. Watson (Member) has been appointed to the technical staff of The Aluminium Company of Canada, Ltd., Montreal.

D. A. Winton, B.Sc. (Graduate) has taken up an appointment with the British Iron and Steel Research Association on his release from the Royal Air Force.