Flexible Couplings for Marine Installations—Testing and Application

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The authors have restricted the paper to transmission couplings of which they have first hand knowledge, and, in view of their special interests in torsional vibration problems, have paid particular attention to this aspect of the subject.

Laboratory tests of the stiffness and damping characteristics of two rubber couplings are discussed, together with those of two more-recently developed couplings which are given in greater detail.

The use of a mechanical analogue for the qualitative study of non-linear coupling behaviour is also included.

The application of flexible couplings for marine propulsion is discussed for direct drive and geared installations. A number of actual examples is quoted in which the merits of flexible couplings in the solution of some torsional vibration problems are clearly illustrated with the aid of comparative diagrams.

INTRODUCTION

Power transmission from the ancient prime-movers, the water wheel and the windmill, was effected via crude wooden cogwheels of pins and grooves, which in this context can be regarded as one of the earliest forms of transmission couplings.

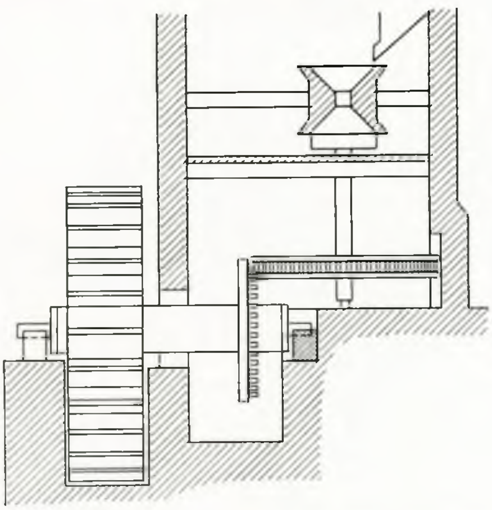

FIG. 1—*Early Roman water wheel driving a mill via wooden cogwheels*

In this simple device, the angle of drive and the speed ratio could be adjusted to suit requirements, and thus alignment presented no difficulty.

Modern flexible transmission couplings are more prolific in shapes and sizes and principles of operation put into use. In fact, there appears to be a quest for the "ideal" coupling that can not only permanently accommodate all the misalignment likely to be encountered in service, but also "cure" all torsional vibration difficulties and, at the same time, be available from stock in all sizes at a nominal cost.

Flexible coupling, however, is a loose term which should be qualified by indicating the sort of flexibility provided, i.e. axial freedom, radial and/or angular misalignment, or torsional elasticity, or a combination of these, and in what proportions.

In general, flexible couplings are necessary for alignment purposes where independently supported machines are coupled by a short length of shafting which is not capable of accommodating the residual misalignment or where excessive movement or wear in service cannot be ruled out.

There are two basic types of misalignment, usually present together in various proportions:

1) angular, where the two shafts meet, but are not parallel;
2) radial, where the shafts are parallel, but are offset.

Similarly, alignment couplings can be broadly divided into two groups according to the corrective duty intended:

a) those primarily designed for angular out-of-line, such as the pin and bush or flexible disc types, which, although efficient for the duty intended, can accommodate only a limited amount of lateral displacement resulting from radial misalignment, depending upon the volume of the bushes;
b) those capable of accommodating both angular and a considerable amount of radial misalignment, such as the link types, double-disc and gear types, including, of course, the cardan shaft and the quillshaft, and rubber tyre type couplings, all of which can also be made to accommodate some degree of axial movement.

Alignment couplings of this second category are necessary in case of flexibly mounted engines which are subject to movements in all directions. The fitting of such a coupling, however,

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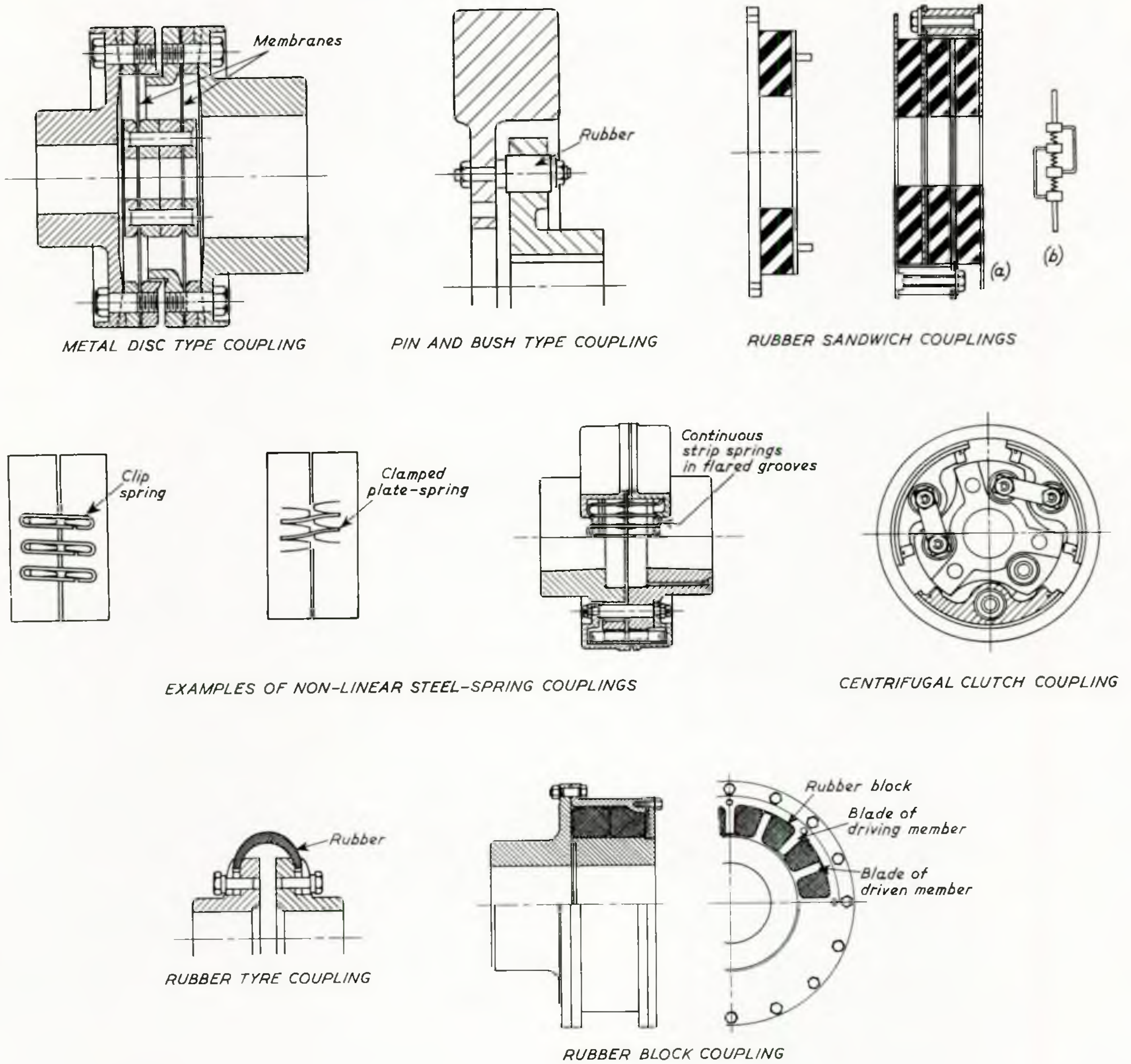


FIG. 2—*Various types of flexible couplings*

is not a substitute for careful lining up on installation as the initial and service out-of-line could overtax the coupling.

Fig. 2 shows a number of different types of flexible couplings which possess to a varying degree the characteristics mentioned in the foregoing. However, in view of the multiplicity of flexible couplings commercially available at the present time, there is a need for a complete list of such couplings together with comprehensive details of their alignment and torsional characteristics.

Further, as accommodation of misalignment subjects the coupling to alternating loading, there is also need for wider fatigue testing under controlled conditions in the laboratory.

The interposition of a flexible coupling for alignment purposes will usually affect also the torsional characteristics of the system. The latter consideration is particularly important in marine installations where torsional vibration may be excited by the engine and/or propeller. The choice of a coupling must therefore be considered not only in the light of its suitability to accommodate the misalignments and movements which may take place in the transmission system, but also in regard to the effect on the torsional characteristics of the complete installation. On the other hand, by careful selection of a coupling of suitable stiffness characteristics it is possible to influence the position of torsional critical speeds of the system and usually arrange these so as not to conflict with the normal running speeds. In other words, the critical speeds can usually be "tuned out".

In some cases where this is not practicable owing to the need for a wide range of service speeds, or when the installation appears to have more than its share or torsional criticals, it may be possible to select a coupling having enough damping to keep the vibration arising from the less severe criticals within acceptable limits. The latter course was often the only solution possible before the gradual development of the highly flexible couplings since the last war.

This paper discusses in detail, types of couplings of which the authors have had experience, both in regard to normal service or in laboratory tests in rigs which simulate as far as possible the service conditions.

For practical reasons this paper is not intended to form a design manual or a catalogue of all transmission couplings, but concerns itself with some of the problems encountered in the laboratory testing and service application of torsionally flexible couplings for main propulsion marine installations of which the authors have first hand knowledge.

Since rubber is used as the resilient element in many couplings, it is perhaps worthwhile including here some remarks on the general properties of rubber.

The term "rubber" when used in modern context not only includes the natural product of certain trees, shrubs and other plants, but also "synthetic" rubbers. All have the one common characteristic of high elasticity. The rubber normally encountered in engineering, whether basically natural or synthetic, is one which is formed by the compounding of the raw material with substances to give the desired properties and to facilitate vulcanizing which is achieved by addition of sulphur to the rubber at temperatures of 250-340 deg. F. (120-170 deg. C.) according to the requirements of the finished product. The compounding materials used are carbon-black, zinc oxide, clay, etc., and these give the stiffness and damping characteristics to the mix.

The main groups into which the resulting rubbers fall are generally identified by hardness which is often measured by an indentation method known as the Shore Durometer method. It is, however, useful for the engineer to relate the hardness value to the moduli of elasticity and rigidity, and Fig. 3 shows a typical relationship of Shore hardness to modulus of rigidity.

Another characteristic in which rubber differs from meta s is that, owing to its peculiar molecular structure, the static moduli do not correspond to the dynamic moduli when the rubber is subject to cyclically varying loads, and the relation of static to dynamic shear modulus can be seen in Fig. 3. Also shown in the figure is the variation of damping properties of rubber for different hardnesses. The typical relationships shown in this figure are valid for one frequenecy of applied load, but the advisability of dynamic tests is clearly indicated here. Part I of this paper discusses some tests carried out at the B.I.C.E.R.I. laboratory on torsional couplings using rubber as the flexible element.

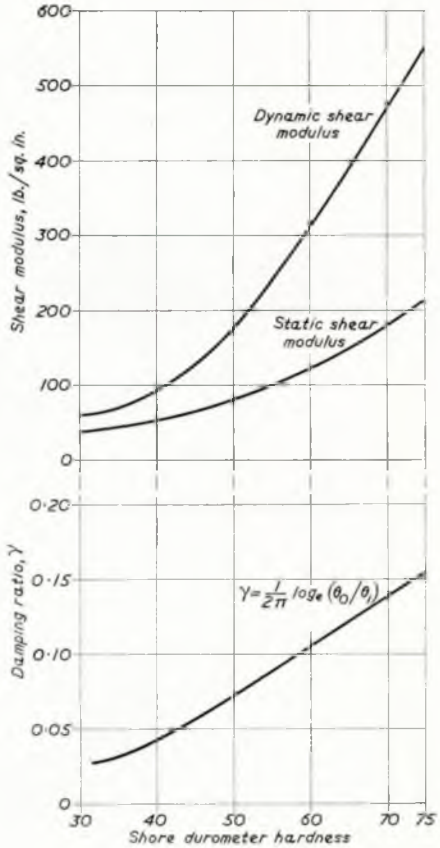


FIG. 3—*Typical relationship of Shore hardness to dynamic modulus of rigidity and damping properties of rubber*

PART I

LABORATORY TESTS TO ASSESS THE TORSIONAL CHARACTERISTICS OF COUPLINGS

It is necessary for coupling manufacturers to supply sufficient basic information to allow an assessment to be made of the torsional characteristics of a transmission system which is to incorporate a coupling of a particular design. Usually the information concerns physical properties of the coupling which are based on static considerations. For example, in the case of couplings incorporating rubber as the resilient element, which may be in the form of blocks, bushes, or bonded layers, it is usual to quote a torsional stiffness based on static tests of the whole coupling or perhaps even of one rubber element only. The following section of this paper gives the results of both static and dynamic tests.

Effect of Vibration Frequency on Torsional Stiffness of Solid Rubber Disc Couplings

Fig. 4 gives the torque-deflexion curves of a solid-rubber disc coupling of about 13½ inches in diameter under static loads with load changes being made at one minute and quarter minute intervals. These tests can be regarded as "zero frequency" tests and the increase of stiffness for the shorter interval load changes can be clearly seen. Using the mean slope of the hysteresis loop as a representation of the coupling stiffness, the value for load changes at one minute intervals is $4{\cdot}08 \times 10^5$ lb.in./rad. and $4{\cdot}39 \times 10^5$ lb.in./rad for 15 sec. intervals. The stiffness increase is thus about 7·6 per cent for shorter interval load changes. The same coupling was subjected to dynamic tests at different frequencies. This was done by suddenly releasing from the coupling an applied torque and measuring the frequency of the decaying vibration. The use of different inertia discs attached to the coupling gave different decaying vibration frequencies and the effective coupling stiffnes was obtained from the relation:

$$K = \omega^2 J$$

where J is the mass moment of inertia of the torque arm plus the added inertia weights (lb. in. sec.^{2}) and ω the measured frequency of decaying vibration (rad/sec.).

In three tests at frequencies of 275 v.p.m., 339 v.p.m. and 451 v.p.m. the corresponding stiffnesses were $3{\cdot}96 \times 10^5$, $4{\cdot}24 \times 10^5$ and $4{\cdot}24 \times 10^5$lb. in/rad. These dynamic tests therefore indicate a similar stiffness-increasing tendency to that shown by the "zero-frequency" tests.

Dynamic tests were later made on a coupling of similar construction but of somewhat smaller size, about 7in. diameter, over a frequency range of "zero-frequency" to 13,000 v.p.m. The results of these tests, in each of which the coupling had

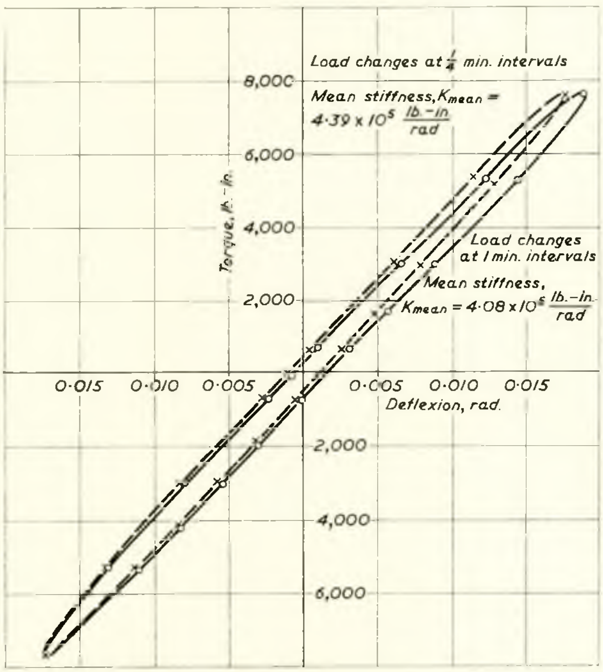


FIG. 4—*Torque-deflexion characteristics of rubber disc coupling*

approximately the same initial deflexion, are given in Fig. 5 and show that the stiffness increases to a maximum of more than double its static value at frequencies of 2,500-5,000 v.p.m., and then reduces at higher frequencies to a constant value at about 8,000 v.p.m. and above.

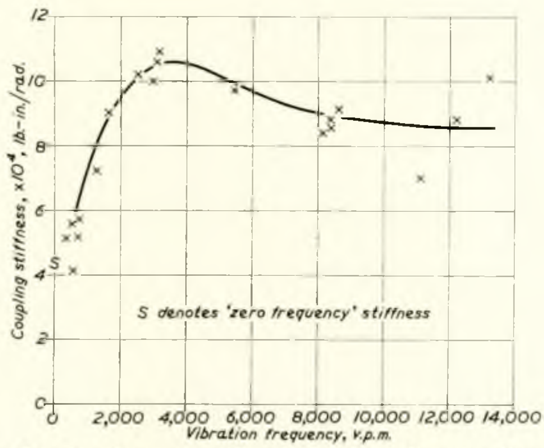


FIG. 5—*Dynamic stiffness of rubber disc coupling*

These tests clearly indicate the advisability of determining wherever possible by dynamic tests the stiffness of couplings which utilise rubber as the restoring force medium.

If no such results are available it is prudent to allow a wider "ignorance" margin in the application of the coupling to any particular problem.

Damping Properties of Solid Rubber Couplings

With the smaller of the two rubber couplings for which stiffnesses were obtained in the frequency range zero to 13,000 v.p.m. damping ratio values were obtained from the ratio of the amplitudes of two successive cycles, of the oscillogram for the decaying vibrations, from the expression

$$\text{Damping ratio, } \gamma = \frac{1}{2\pi} \times \log_e (\theta_1/\theta_2)$$

where θ_1 and θ_2 are the maximum amplitudes of vibrations.

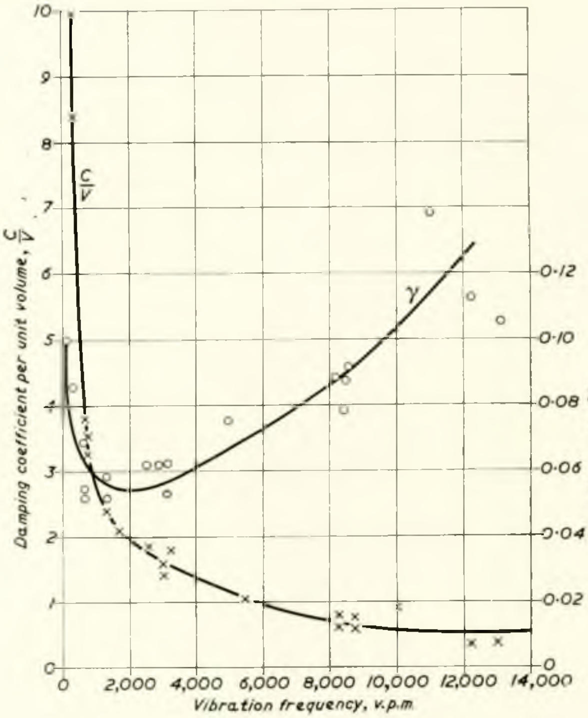


FIG. 6—*Damping characteristics of rubber disc coupling*

The results given in Fig. 6 show that damping ratio drops very steeply at first with increasing frequency, and then rises steadily thereafter. Fig. 6 also shows the damping coefficient of rubber couplings per unit volume from the expression:

$$\frac{C}{V} = \frac{2\ J\omega\gamma}{V}\ \frac{\text{lb.sec.}}{\text{in.}^2}$$

where J is the moment of inertia of the torque arm and γ is the damping ratio previously obtained. This curve shows that the damping coefficient falls rapidly at first with increasing frequency but tends towards an asymptotic value at 10,000-12,000 v.p.m.

Dry Fluid Coupling

A recent newcomer to the field of transmission couplings is the "dry fluid" coupling, referred to also as the "powder coupling"(1). This coupling (sketches of which are given in Figs. 7 and 8) consists of a light alloy casing containing a concentrically located rotor in the form of a hardened steel disc which is corrugated or circumferentially waved. The casing is bolted to the driving unit and the rotor to the driven machine. The transmission medium is hardened and tempered spherical

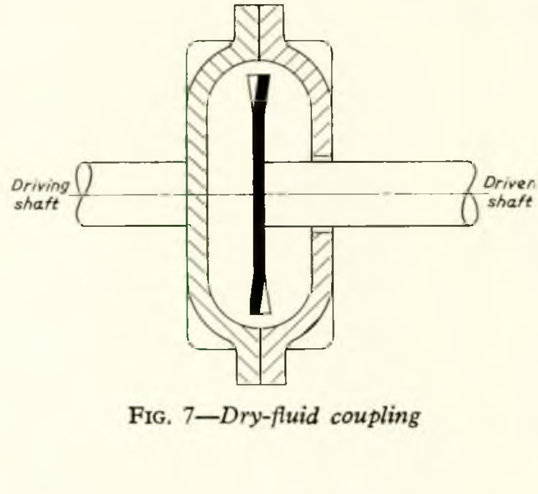


FIG. 7—*Dry-fluid coupling*

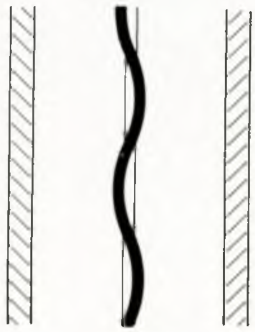


FIG. 8—*Rotor of dry-fluid coupling*

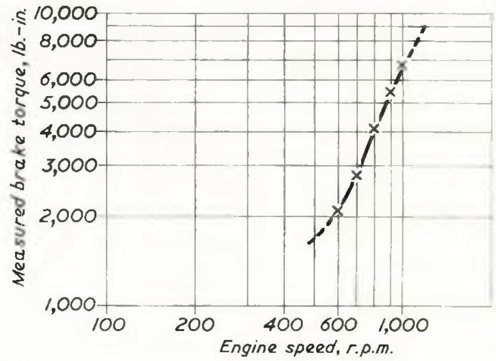


FIG. 10—*Torque-speed characteristics of dry-fluid couplings with 0·5 per cent slip*

steel shot, with graphite for lubrication purposes. The gauge of the steel shot may be selected to suit the type and size of the coupling, which in turn depends upon the power characteristics of the machinery in which the coupling is to be used. Suitable seals are supplied to ensure that there is no loss of the charge during running.

The action of the coupling is as follows. Upon starting and during acceleration, centrifugal forces acting on the shot create increasing peripheral packing of the shot, and this packing action gradually accelerates the corrugated rotor until eventually the driving and driven speeds are synchronized.

The coupling is normally supplied with stiff rubber bushes in the casing cover to allow angular, and a limited amount of radial and axial misalignment.

Laboratory tests were made on an engine and brake system designed to simulate a marine installation to assess the ability of this coupling to damp torsional vibration. For this purpose a 100 h.p. Diesel engine was coupled to a water brake through an additional pin and bush coupling of very low stiffness, the system being tuned to give reasonable amplitude of torsional vibration at the brake free-end. The amplitudes of a 3rd order critical indicated by torsional vibration pick-up at the brake shaft free-end are shown in Fig. 9, for three different transmitted torques. The dry-fluid coupling with its integral stiff rubber-bush coupling was then connected to the shafting between the engine and brake, retaining the very low stiffness rubber coupling. The effect of the rubber-bush coupling integral with the dry-fluid coupling on the torsional characteristics of the system was negligible owing to its relatively high stiffness.

The charge of the powder coupling was adjusted so that the full power of the engine could be transmitted at the engine rated speed without slip; it was then possible to induce coupling slip in the mid-speed range by increasing the brake load to give higher transmitted torques. As the torque-carrying capacity of the coupling is characteristically similar to that of the brake, that is, the torque varies as the shaft speed squared, it was possible to adjust the brake to give a particular coupling slip condition, for example 0·5 per cent, and then to traverse the engine speed range in the vicinity of the critical by varying the fuel rack setting, thus holding the coupling slip to approximately a constant value of 0·5 per cent. Fig. 10 shows the torque-speed characteristic of the powder coupling for 0·5 per cent slip. The coupling slip was measured by observing the coupling with a stroboscope and counting the difference in input and output revolutions against a stop watch.

The results of torsiograph tests for these different slip conditions are given in Fig. 11. The double amplitude measured at maximum transmitted torque was 0·6 degrees, giving an amplitude reduction of slightly greater than 3 : 1. The speed ratio of output/input shafts was 99·4 per cent, and it was observed that after half an hour's running with this slip the powder temperature, measured immediately after stopping, had risen from 72 deg. F. to 95 deg. F. (22 deg. C. to 35 deg. C.).

At the torque of 4,113 lb.in., the total swing (double amplitude) was 0·45 with the powder coupling, compared with 2·0 degrees for substantially the same torque, namely 3,930 lb.in., without the powder coupling. The ratio of output/input shaft speeds was 99·6 per cent.

In a final test the torque value was reduced progressively until the point occurred when no slip was visible with the aid of the stroboscope. At this torque of 2,830 lb.in. the total swing was 1·35 degrees. Interpolation of amplitudes and torques for the system without powder coupling indicated that a value of 1·9 degrees could have been expected at 2,830 lb.in., giving a corresponding amplitude reduction of 1·4 : 1.

These tests show that the dry fluid coupling could be used to good advantage for certain marine applications where

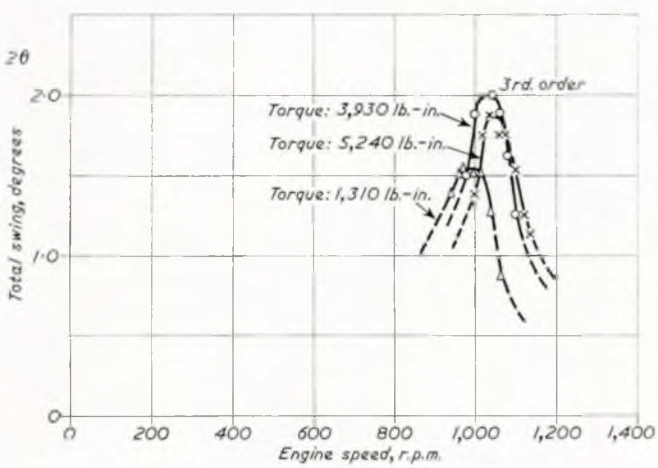


FIG. 9—*Third order torsional vibration amplitudes without dry-fluid coupling*

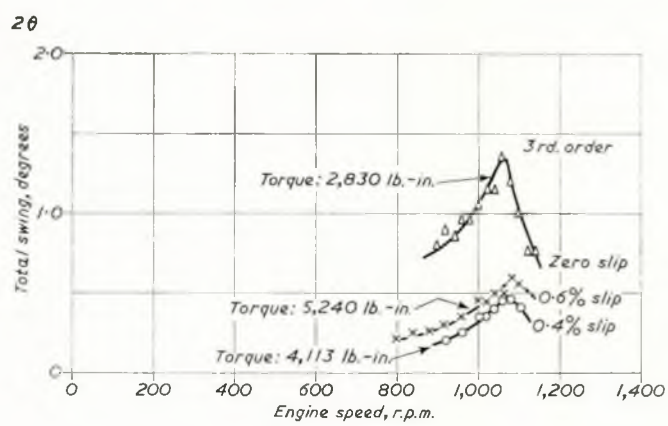


FIG. 11—*Third order torsional vibration amplitudes with dry-fluid coupling*

a torque-limiting characteristic is desired and a limited amount of damping is required in passing through critical speeds.

"Half-moon" Coupling

A coupling with the interesting characteristic of zero stiffness under some conditions has recently made its appearance in the field of transmission couplings[2]. Intended mainly for marine Diesel applications, it has special advantages in systems where transmitted torque varies as the square of the speed or in which a very low stiffness is required.

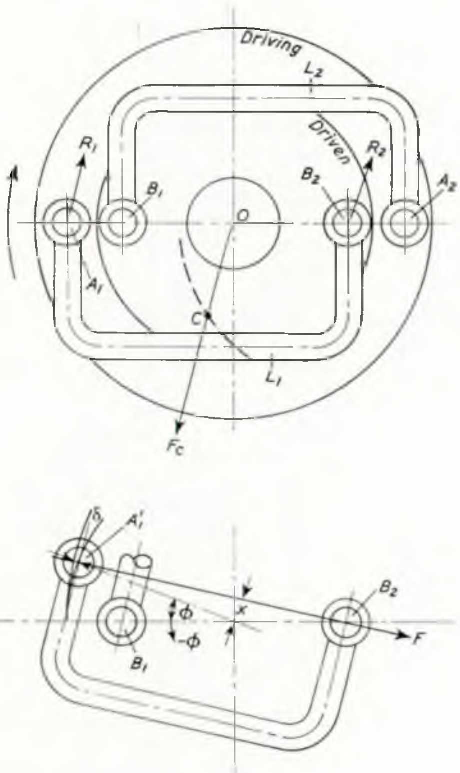


FIG. 12—*Diagrammatic arrangement of "half-moon" coupling*

Fig. 12 shows a diagram of this coupling which consists of a driving member carrying two pivot pins A_1 and A_2 diametrally opposed to one another and equidistant from the centre of rotation O, and a driven member carrying two similar pins B_1 and B_2 also diametrally opposed but at a smaller radius. The driving and driven members are coupled together (A_1 to B_2 and A_2 to B_1) by two cranked links L_1 and L_2 with bored holes at their extremities such that they can pivot on the pins. The centre distances of the pins and holes are so arranged that when the halves of the coupling are coupled together with a common rotational centre O, the pins are all in line. Assuming that the driving member rotates in a clockwise direction then the pin A_1 will swing around the centre O causing the link L_1 to pivot about the centre B_2. Assuming also that the driving member moves through an angle $\phi°$ so that pin A_1 reaches the position of A'_1 shown in Fig. 12, without any change in the position of B_2, then the distance between A'_1 and B_2 is less than the original distance A_1 B_2 by an amount δ. The link L_1 is in effect a very stiff C-spring which therefore exerts a force F in the direction A'_1B_2. This produces a torque Fx in the driven member; a similar torque is produced by the link L_2 pivoting about B_1. This total torque 2Fx will increase until it equals the reaction torque of the driven system at which point no further angular displacement takes place between the driving and driven members.

For very small values of ϕ the deflexion δ is negligible so that F and x are negligible. This is the condition for very low torque transmission and the coupling has virtually zero stiffness. As ϕ increases the values of δ, F and x increase, and the coupling has the torque-deflexion characteristic shown in Fig. 13.

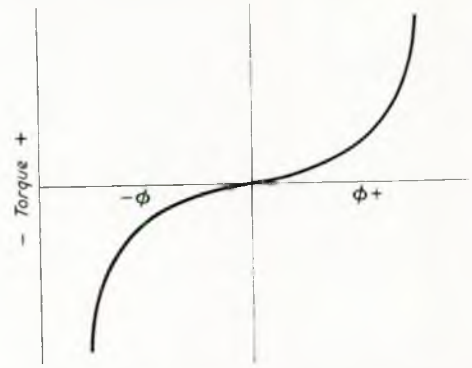


FIG. 13—*Torque-deflexion characteristics of "half-moon" coupling*

This coupling is, however, subjected to centrifugal forces during rotation which act upon the links L_1 and L_2. If the centre gravity of the link L_1 is at a point C (see Fig. 12), then centrifugal force F_c acts through this point from the centre of rotation and imposes reaction forces R_1 and R_2 on the pins A_1 and B_2. The resulting torque on the driving member is anti-clockwise with a clockwise torque of equal value for the driven member. Similar torques occur for centrifugal forces acting on the link L_2.

Under conditions where the driving torque is equal to the centrifugal torque no relative torsional displacement of the coupling will occur during driving, and the pins will remain in line. When the driving torque exceeds the centrifugal torque the coupling will deflect through an angle ϕ shown in Fig. 12. If, however, the driving torque is less than that produced in the coupling by centrifugal force then the angle becomes negative. As the coupling deflects, the centres of gravity of the links move through arcs as shown in Fig. 12, and with appreciable positive values of ϕ the centre of gravity of the link approaches centre O, and moves away from O for negative values. This centrifugal torque progressively diminishes as the coupling deflexion moves from ϕ negative to ϕ positive, so that,

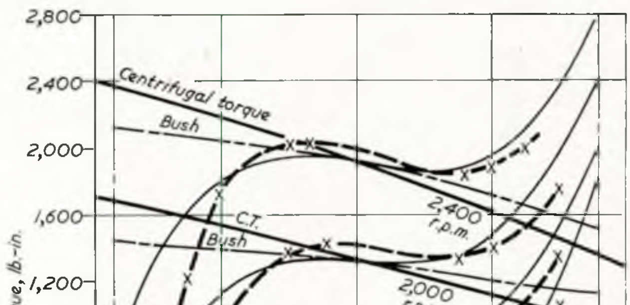


FIG. 14—*Effective torsional stiffness curves of "half-moon" coupling*

for example, at a constant speed of 2,000 r.p.m. further increases of transmitted torque cause a decrease of centrifugal torque. This characteristic is shown in Fig. 14, and the increasing negative slope of the torque-deflexion lines with increasing rotational speed is clearly seen. Fig. 14 also gives the centrifugal torque of the coupling at various speeds, with the addition of the transmitted-torque deflexion characteristic at each of these speeds. It can be seen that the slope of the torque-deflexion curve (that is, by definition, the stiffness of the coupling) is negative for coupling displacement around zero to $+5$ degrees. This negative slope can be reduced by fitting rubber bushes at the pivot points, thus creating a torque which opposes the centrifugal torque. Torque-deflexion lines for the bushes are included as broken lines in Fig. 14. The influence of the bush-torque on the centrifugal torque will depend upon the stiffness chosen for the bushes, while the value of the centrifugal torque will depend upon the mass and shape of the links. Thus the coupling can be disposed to have zero stiffness at $\phi = 0$ for any desired speed and torque. In practice, however, it is not possible for a coupling to have a true zero stiffness, that is, infinite displacement for negligible torque. At the design stage it was appreciated that the coupling would tend towards small positive or negative values of displacement rather than remain at the zero deflexion point, but stroboscopic tests were planned to clarify this point.

A system was calculated which would allow the coupling to be tested under laboratory conditions and to compare results with some system incorporating a known conventional coupling. A torsional system was chosen such that both one-node and two-node modes of vibration would be present, with nodes in the driven machinery. A mass was arranged to represent a gearbox inertia, with a water brake simulating propeller inertia, since the water brake has a torque-carrying characteristic similar to that of a propeller. The engine chosen was a Perkins 4·236 engine and the whole system thus simulated, albeit in a small scale, a Diesel geared marine drive. This is shown in Fig. 15.

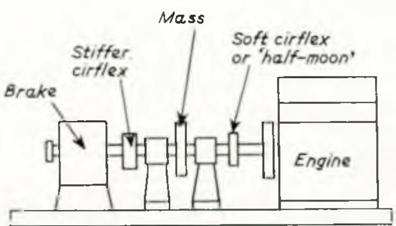


FIG. 15—*Diagrammatic arrangement of "half-moon" coupling test rig*

The system for the first tests against which the performance of the half-moon coupling was to be assessed was arranged to have a two-node frequency at 4,000 v.p.m. and a one-node frequency of 1,200 v.p.m. This would give a major two-node mode second order critical around 2,000 r.p.m. and one-node around 600 r.p.m. both of which are within the normal operating speed of the engine. This was achieved with two conventional couplings in the system, the softer being between the engine and "gearbox mass", and having a stiffness of $0{\cdot}0325 \times 10^6$ lb.in./rad. Torsiograph tests of the system were made in which the amplitudes and corresponding frequencies were measured at the brake end. The softer coupling was then replaced by the "half-moon" coupling, and the torsional tests were repeated, with additional observations made by the stroboscope of the deflexions of the "half-moon" coupling. The results of both series of tests are shown in Figs. 16 and 17, Fig. 16, giving amplitudes for the two-node mode frequency, shows that the largest amplitude (2nd order) is reduced from $\pm 0{\cdot}26$ to $\pm 0{\cdot}04$ deg., that the $1\frac{1}{2}$ order has a similar reduction from $\pm 0{\cdot}2$ deg. to $\pm 0{\cdot}03$ deg., and that the 4th order is reduced from $\pm 0{\cdot}18$ to $\pm 0{\cdot}07$ deg. The two-node frequency is reduced from 3,550 v.p.m. to 3,200 v.p.m. Considering now the results for the one-node mode given in Fig. 17, it can be seen that the one-node frequency has been lowered from about

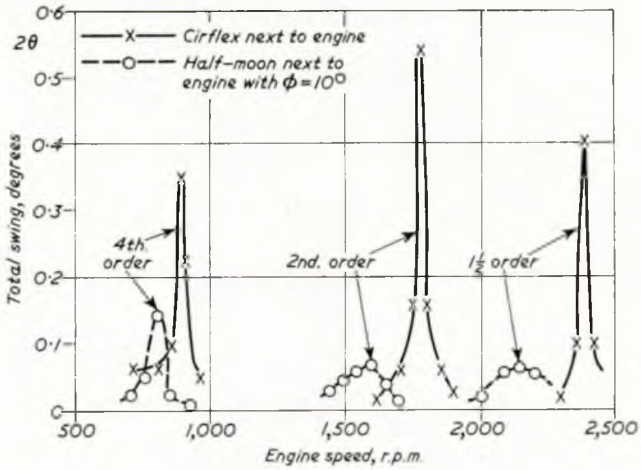


FIG. 16—*Two-node mode torsional vibration amplitudes*

1,050 v.p.m. to 750 v.p.m., with the result that the very large 2nd order, for which the peak amplitude is estimated at ± 5 deg., is now below the idling speed of the engine. The amplitude of the 1st order was, however, reduced from $\pm 2{\cdot}3$ deg. to $\pm 0{\cdot}6$ deg. It should be noted that these one-node frequencies only occurred when the torques were so adjusted that the steady deflexion of the coupling, observed by stroboscope, was about $\phi = +15$ deg. At smaller deflexions the stiffness was so low as to give frequencies well below the lowest running speed; in the case of the coupling with deflexion around zero, no clearly defined one-node resonance was observed, but swings up to ± 6 deg. were recorded at 750 r.p.m.; these are believed to be due to the coupling changing position over its substantially zero stiffness range as a result of engine hunting at very low loads. Stroboscopic observation of the coupling deflexion under the various conditions of speed and transmitted torque verified the expectation that the coupling deflexion would move away from the values around the zero stiffness positions. For example, Fig. 14 shows that at 2,000 r.p.m. (where the negative torque-deflexion characteristic of the coupling was most marked) deflexions of decreasing negative values with increasing transmitted torque were observed until a coupling deflexion of -3 deg. occurred, after which the coupling swung across to $+8$ deg. Also included in Fig. 14 are torque-deflexion curves for the coupling (shown in solid lines) from calculations. The agreement between calculated and test results is faily good.

The Study of Non-linearity of Couplings using a Bench Model Rig

It has often been claimed that under certain circumstances

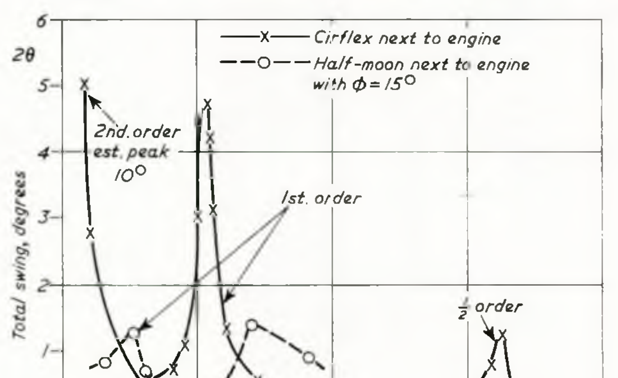


FIG. 17—*One-node mode torsional vibration amplitudes*

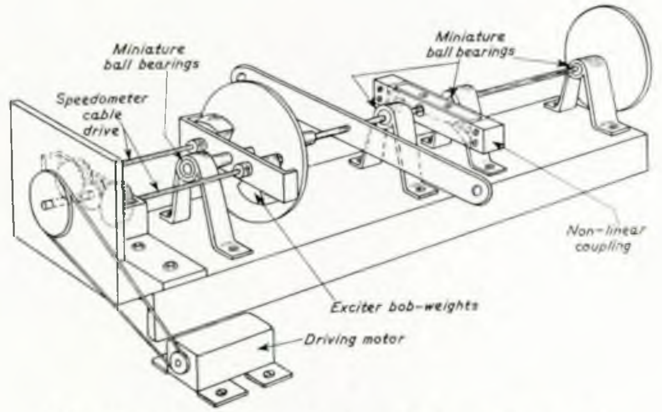


FIG. 18—*Non-rotating bench model coupling test rig*

there were considerable advantages to be gained in the use of non-linear couplings in transmission systems. In order to study the possible detuning effects of couplings with non-linear stiffness characteristics a bench model rig was built at the B.I.C.E.R.I. laboratory. The model, shown in Fig. 18, was intended to simulate a typical torsional system of engine and driven machine connected by a flexible coupling. The system, a non-rotating one, was excited by a pair of bob-weights geared together and phased to give a cyclically varying torque at one end mass representing cylinder masses. Damping was kept to a minimum by the use of precision miniature ball journal bearings for supporting the oscillating shafts, and drive from a variable speed electric motor to the bob weights was made flexible to allow the system to swing with maximum freedom. Resonant torsional deflexions at the coupling were measured electronically with a multi-vane trimming condenser which was mounted on the driving side of the coupling, with the operating spindle of the moving vanes attached to the driven side of the coupling.

The system was arranged with a linear coupling for the first tests. The linear coupling consisted essentially of a driving and driven member connected by a single strip of spring steel blade. Fig. 19 gives torsional vibration amplitudes and frequencies measured across the coupling for the model system with two different inertia masses representing the driven machine, that is, at the mass remote from the bob-weights.

The linear coupling was then replaced by one having the non-linear characteristic shown in Fig. 20. This coupling was also one in which a single strip of steel spring was used

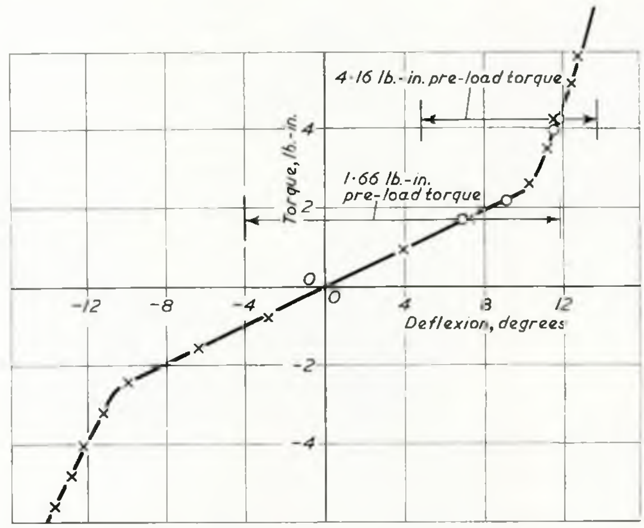


FIG. 20—*Torque-deflexion characteristics of the non-linear model coupling*

to connect the coupling halves, but in this case as the blade deflected with increased torque beyond a certain point, it abutted against a stop which effectively shortened its length and correspondingly increased its stiffness, giving it, in effect, the two-rate stiffness shown in Fig. 20.

Tests of this coupling were made in which the coupling was allowed to swing about its zero position (corresponding to no preload torque) and also about positions of initial deflexion caused by static preloads which simulated steady torques across the coupling. The results of these tests with the same two free-end masses as for the linear coupling are given in Fig. 21 and Fig. 22.

The tests of the coupling with zero and low preload torques show that the critical speeds with the two different end masses are extended considerably when running up through the speed range and that there is a tendency for the system to cling to the resonance. During decreasing speeds the critical range and amplitudes are smaller.

Where the preload torque is sufficient to deflect the coupling initially to about the position of greater stiffness of the two-rate coupling, the frequency spread and amplitude increase are less with increasing speeds, but greater during decreasing speeds, with the system tending to cling to the critical as the speed is reduced through the speed range.

It can be seen that the amplitude at which the resonance curve hooks over with zero preload torque corresponds to the stiffer position of the two-rate coupling. Thus it appears that

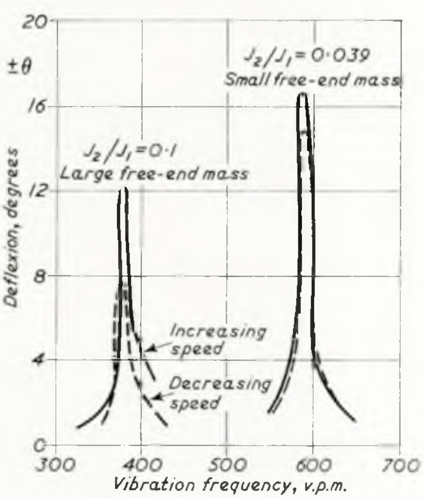


FIG. 19—*Torsional vibration amplitudes on the test rig with linear coupling*

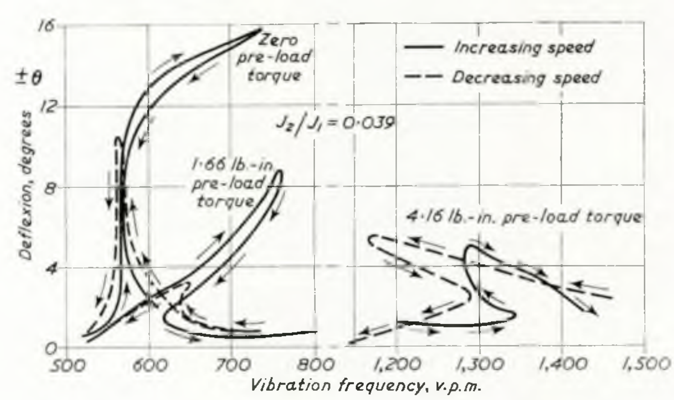


FIG. 21—*Torsional vibration amplitudes on the test rig with the non-linear model coupling*

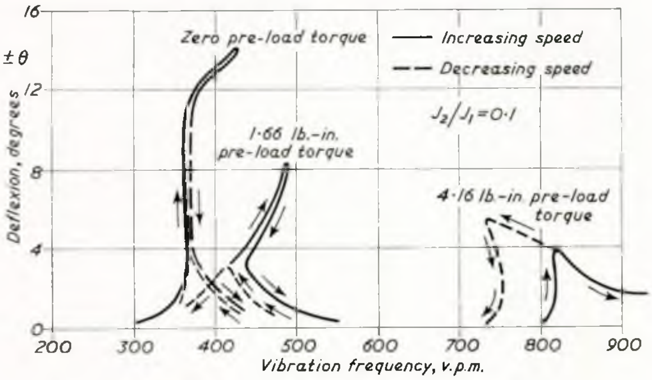


FIG. 22—*Torsional vibration amplitudes on the test rig with the non-linear model coupling*

the system resonant frequency increases as the excitation frequency rises so that the critical speed range is extended. This continues until a break-away point is reached at which the coupling amplitude limit is reached and the amplitude suddenly diminishes to a very small value; thereafter the energy produced by the exciter driving motor and absorbed in maintaining the large amplitude of swing of the system now becomes available to rapidly accelerate the exciter bob weights to a much higher speed.

The resonance curve shown in Fig. 23 is the result of a detailed study of the break-away phenomenon, in which the exciter speed was indicated on an oscilloscope simultaneously with the rapidly diminishing vibration amplitudes. It can be seen that during this very short period of time the system frequency decreases (due to the coupling reverting to the low stiffness characteristic with small amplitude) during which time the exciter speed increases slightly. With further speed increases the system becomes non-resonant and vibrates at the forcing frequency corresponding to the exciter speed.

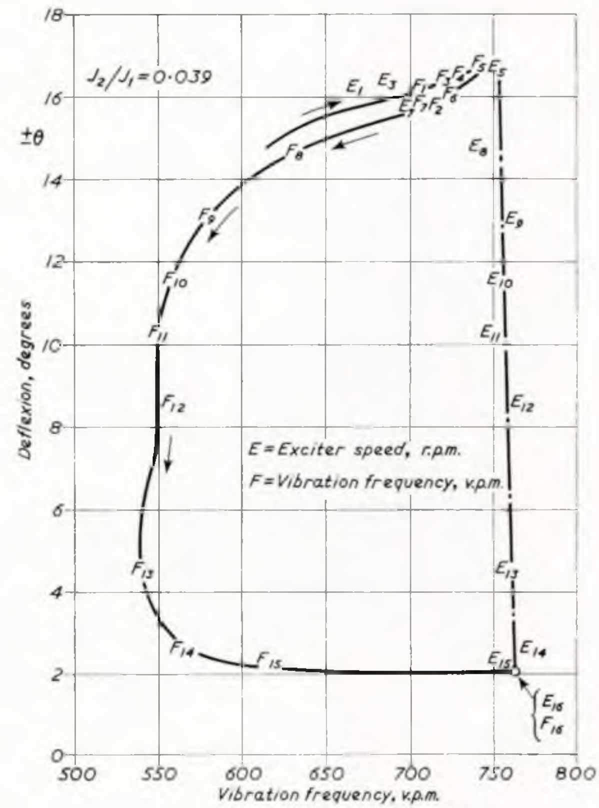


FIG. 23—*Non-linear resonance curve showing the break-away phenomenon*

The hook-shaped resonance curves are very similar to those shown by theory[6].

PART II

APPLICATION OF FLEXIBLE COUPLINGS FOR MARINE MAIN PROPULSION

Direct Drive Installations

Whereas some form of flexible coupling is almost invariably interposed between a Diesel engine and reverse/reduction gearing, the use of flexible couplings in direct drives is an unusual feature dictated by some specific considerations. Torsional vibration is generally to blame, and some circumstances are outlined where the application of flexible couplings is advantageous in solving torsional vibration difficulties.

The installations considered fall mainly in the higher power low and medium speed group where the cost of a flexible transmission coupling is not insignificant and its use must be justified also on economic grounds.

In ships having the main engines placed amidships the long line shafting is torsionally "soft", and, as a result, the one-node, i.e. the propeller-mode, criticals resonate low down in the speed range and are well separated from the two-node or "crankshaft" mode criticals.

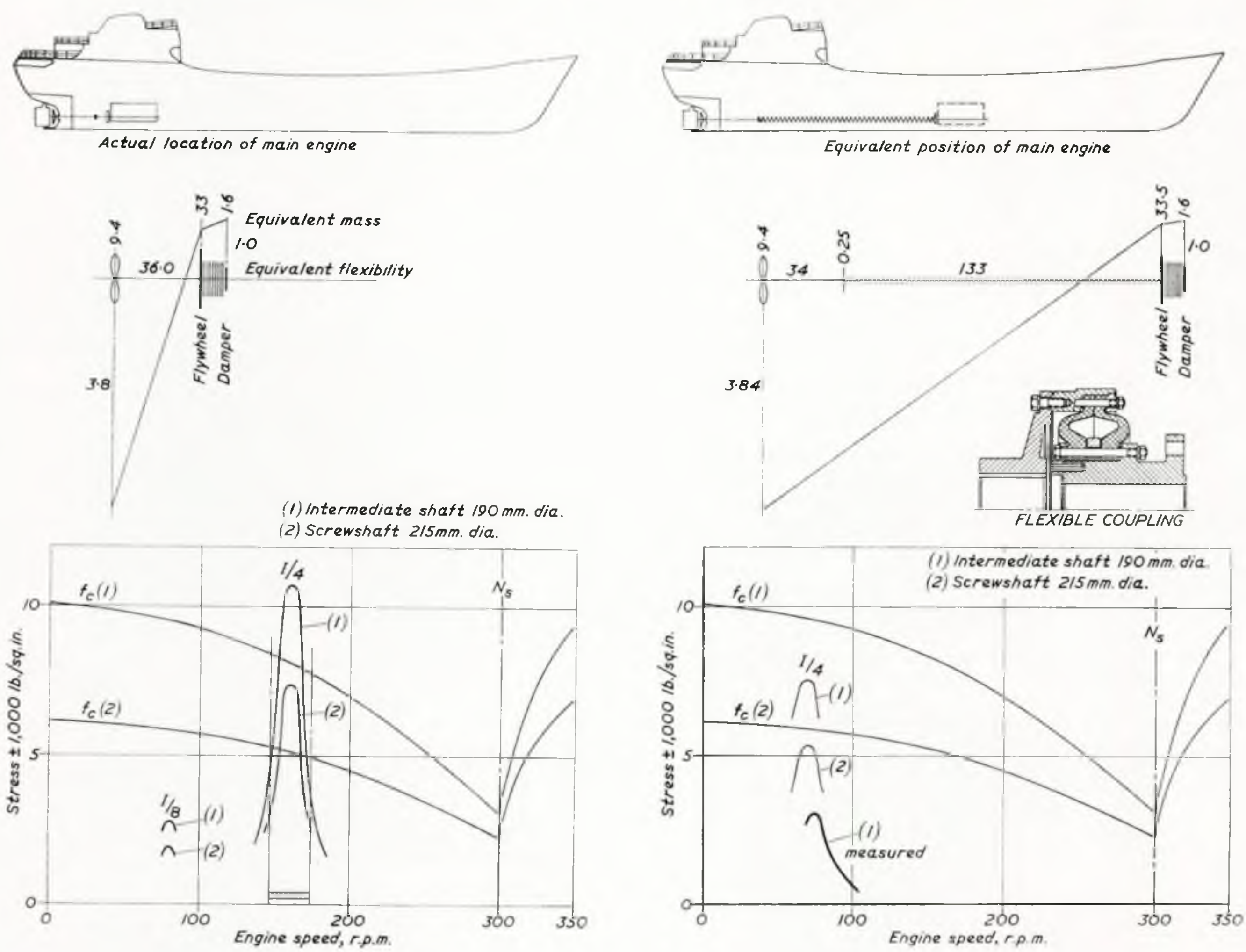
Such separation of different modes of vibration is advantageous in permitting each mode to be considered almost independently, and is equivalent to the "isolation" of the driven from the driving machines. In amidships installations there is already ample torsional flexibility in the system and the introduction of a flexible coupling would be of little advantage.

Where, however, the engines are situated aft or "three-quarters aft" (i.e. with one hold aft of the engine room), the shafting is short and comparatively stiff. The major one-node criticals tend to resonate at about 60-80 per cent of the service speed, depending on the number of cylinders and the length of shafting, the diameter of which is assumed here to be approximately as required by the Rules of the Classification Societies. The resulting stresses can in some cases be very high, necessitating a wide speed restriction, and are sometimes excessive even for passing through. In any case criticals of such magnitude are generally unacceptable in the upper speed range.

It has become the usual practice in such cases to employ "oversize" or "stiff" shafting with a view to placing the offending criticals clear above the service speed. The main advantages of such a solution are its simplicity, and that, except for the flank of the major critical, the remainder of the speed range is comparatively free from critical speeds. This is clearly illustrated later in Figs. 24(c) and 25(b).

There can, however, be serious disadvantages in this approach, especially where greatly oversize shafting is necessary, as indicated in the following comprehensive list, although all of these need not assume prime importance in each case:

a) The node for the first mode of vibration moves close to or even into the crankshaft which often has smaller diameter than the oversize line shafting. Crankshaft



(a) With rule size shafting

(b) With flexible coupling

FIG. 24(a) and (b)—*Example 1: 260ft. cargo ship powered by a four-stroke, eight-cylinder engine rated at 1,530 b.h.p. at 300 r.p.m.*

stresses, therefore, become the limiting criterion. This is more pronounced where a heavy propeller and a light flywheel are used.

b) Minor one-node criticals increase in magnitude owing to greater relative twist in the crankshaft and in some cases can become the limiting factor when added to the flank of the major critical.

c) The shafting and crankshaft can be subjected to high vibration stresses during racing; this is particularly important in ships less amenable to ballasting, e.g. bulk carriers, and in this connexion similar problems encountered in the Liberty type ships may be recalled[3].

d) The flank stresses are active during the whole working life of the machinery, and, in corrosive environment, may lead to corrosion fatigue, e.g. at the screwshaft forward coupling, or at the screwshaft cone due to leakage at the propeller seal.

e) The torsionally stiff shafting is also stiff in bending and thus more sensitive to any misalignment, either permanent or that occurring in a seaway.

f) There may be some loss of propulsive efficiency owing to the large diameters of the propeller boss and stern frame required in the case of greatly oversize shafting.

This formidable list of disadvantages ought to be weighed up before adopting the oversize shafting solution, and alternative solutions should also be considered.

In the authors' opinion the advantages of the stiff shafting solution are more likely to outweigh the disadvantages in cases where only moderate oversize is necessary. Where, however, with rule size shafting the major criticals resonate below about 70-75 per cent of the service speed, it can be more advantageous to employ alternative solutions based on the introduction of flexibility into the system with a view to placing the critical lower down in the speed range.

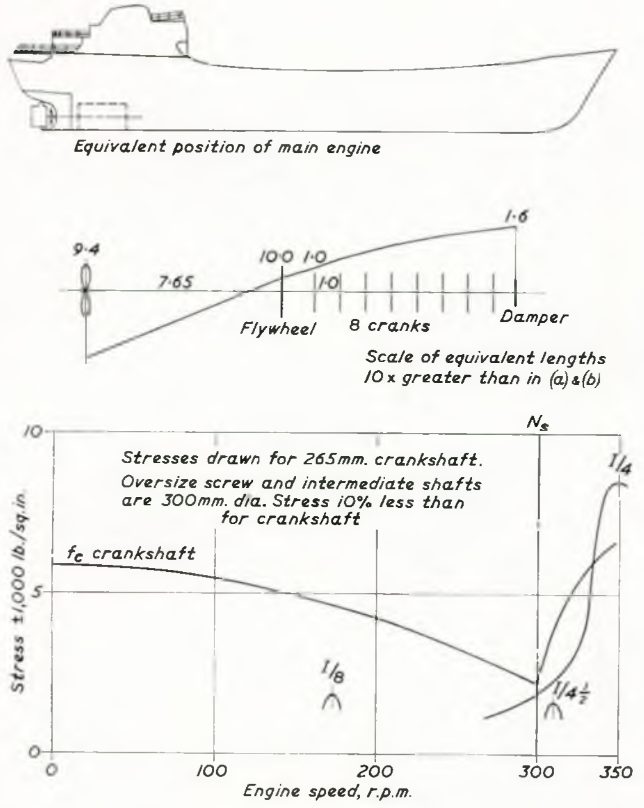
The availability of highly flexible couplings of high torque rating enables wider application of this solution. Three examples are given of different sizes and types of ships where the merits of each solution are discussed in detail for each case, viz:

Example 1. 260ft. Cargo ship powered by a four-stroke eight-cylinder engine rated at 1,530 b.h.p. at 300 r.p.m.

Example 2. 379ft. Admiralty supply ship powered by a two-stroke five-cylinder engine rated at 5,500 b.h.p. at 135 r.p.m.

Example 3. 525ft. Ore carrier powered by a two-stroke six-cylinder engine rated at 7,500 b.h.p. at 115 r.p.m.

For ease of comparison among the different installations the mass-elastic systems of the main propulsion machinery are reduced, in effect, to non-dimensional units. The inertia per cylinder line and flexibility per crankthrow are taken as unity, the remainder of each mass-elastic system being expressed in



(c) With oversize shafting

FIG. 24(c)—*Example 1: 260ft. cargo ship powered by a four-stroke, eight-cylinder engine rated at 1,530 b.h.p. at 300 r.p.m.*

those units. Then, for each solution considered, the flexibilities, or equivalent lengths separating the different masses, are drawn to scale and the actual and equivalent positions of the main engine in the ship are superimposed on the longitudinal outline section. It is not practicable, however, to represent the relative inertias of the various masses accurately, e.g. by the length or thickness of the lines or circles drawn without loss of clarity and therefore the values of the more important masses are marked.

The recently published Lloyd's Register Guidance Notes on Torsional Vibration Stresses and Critical Speeds[4] are used as criteria of acceptance of the torsional vibration characteristics, although the examples quoted were originally considered on the basis of the Guidance Notes and Rules for shaft scantlings then in force. The differences, however, are comparatively small and do not alter the conclusions reached.

Finally, only modes of vibration which are influenced by flexible couplings have been considered in this paper.

Example 1. 260ft. Cargo Ship Powered by Four-stroke Eight-cylinder Engine rated at 1,530 b.h.p. at 300 r.p.m. [see Figs. 24(a), (b) and (c)]

Here the engine is situated aft, and, with approximately rule size shafting of 190 and 215mm. (about 7½in. and 8½in.) for the intermediate- and screwshafts respectively, the one-node 4th order critical is calculated to resonate at 160 r.p.m. See Fig. 24(a).

Stresses arising from this critical exceed the maxima acceptable for continuous operation by a wide margin, in about equal proportions for both the intermediate- and screwshafts, but are acceptable with suitable speed restrictions. The remainder of the running range is free from excessive one-node criticals.

In this case, however, a speed restriction at about half-speed was not acceptable to the owners, and accordingly alternative solutions were investigated.

A solution based on the introduction of flexibility into the system is more promising since with rule size shafting the major critical resonates at only about half-speed. A flexible coupling of the rubber tyre type was used having a nominal dynamic stiffness of 4·1 × 10^6lb.in./rad. This is equivalent to an additional 72ft length of rule size intermediate shafting, and could be comfortably accommodated in the ship if the main engine were placed amidships. See Fig. 24(b), which also includes a sketch of this type of coupling.

In this arrangement the natural frequency is more than halved, the 4th order critical being placed at about 70 r.p.m. The resulting vibration stresses are acceptable for continuous operation for both screw and intermediate shafts even without any allowance for the coupling damping.

There are no other significant one-node criticals in the running range.

A coupling twist of ±10·5 deg. at resonance exceeds somewhat the coupling capacity, and as usual in such cases torsiograph records were taken to confirm the calculations.

The measured results show that the critical resonates at near enough the calculated r.p.m. Stresses, however, are considerably lower giving an overall dynamic magnifier of 5·4 at resonance as against a calculated value of 13·7 based on engine and propeller damping only. The dynamic magnifier for the coupling alone is quoted in the coupling catalogue as 6·0, which agrees closely with the measured value when allowance is also made for the engine and propeller damping in addition to the coupling damping using the empirical relationship[5]:

$$M\ (\text{total}) = \frac{1}{\sqrt{\frac{1}{M_1^2} + \frac{1}{M_2^2} + \frac{1}{M_3^2} + \ldots}}$$

The makers recommendation for avoiding prolonged operation under the foregoing conditions is considered prudent, although the measured vibration stresses and coupling twist are comparatively low. It is understood this coupling has been operating satisfactorily since 1957.

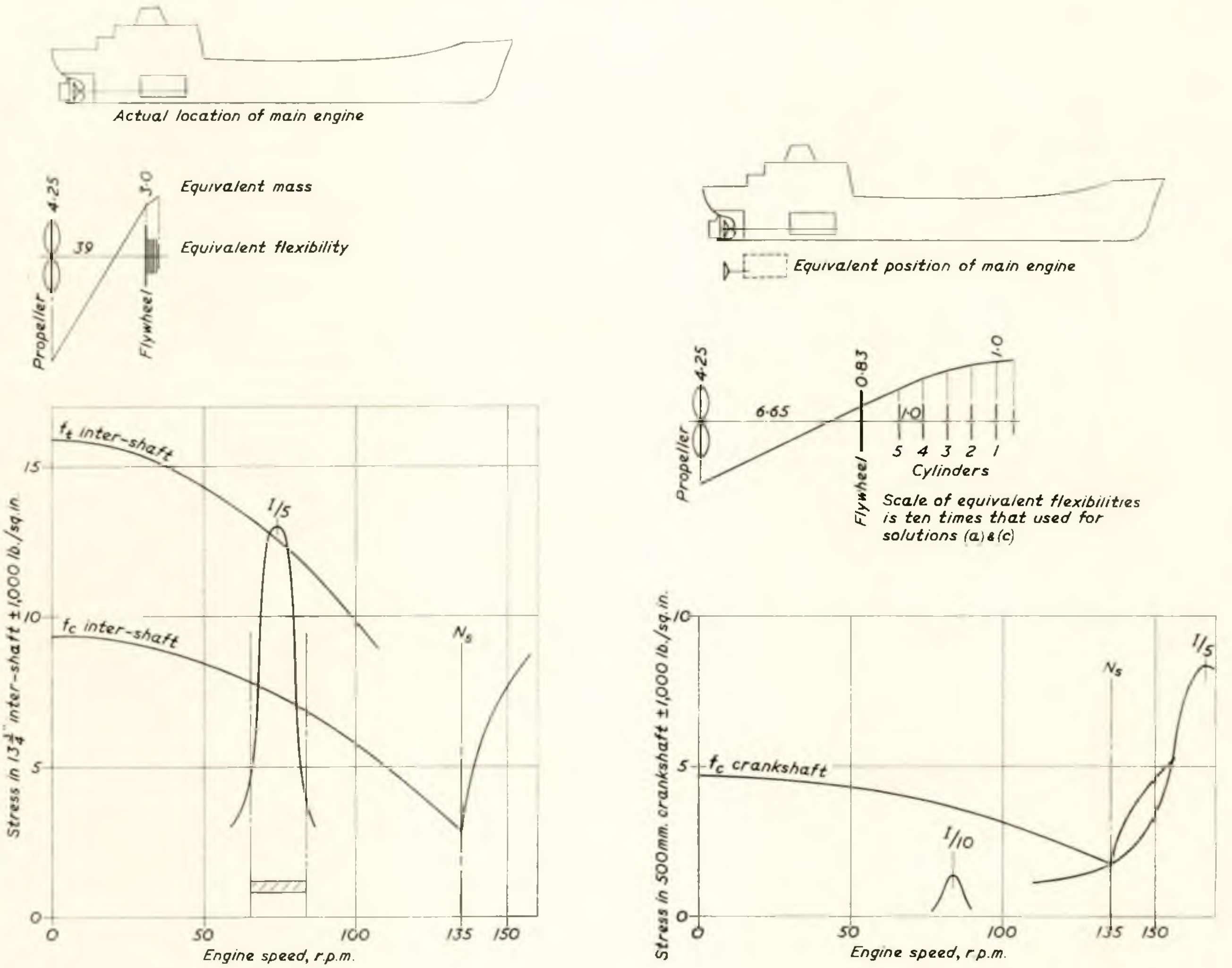
The solution based on stiff shafting proved impracticable, as excessively oversize shafting would have been necessary, and/or greatly reduced propeller inertia, if the installation were to be satisfactory for the full service speed of 300 r.p.m. See Fig. 24(c), where for clarity the flexibilities in this mass elastic diagram are drawn to a scale ten times larger than for the other solutions; in this connexion it should be noted that the overall equivalent length between the propeller and the free end of the crankshaft in the flexible coupling solution is almost exactly ten times longer than for the stiff shafting solution.

With the maximum practical diameter of 300 mm. (11·8in.) for both intermediate and screwshafts and a minimum flywheel inertia the one-node 4th order critical is calculated to resonate at 348 r.p.m., i.e. 16 per cent above the service speed, giving rise to a stress of ± 8,500 lb./sq. in. in the 265 mm. diameter crankshaft. Again the crankshaft becomes the most highly stressed section, whilst stresses in the 300mm. diameter line shafting are about 10 per cent lower.

This solution is not suitable for 300 r.p.m. taking also into consideration the minor 4½ order critical at 310 r.p.m., but would be satisfactory for a somewhat lower service speed.

Example 2—379ft. Admiralty Supply Ship powered by a Two-stroke Five-cylinder Engine rated 5,500 b.h.p. at 135 r.p.m. [see FIG. 25(a), (b) and (c)]

The main engine is situated aft, and with rule size shafting of about 13½in. and 16½in. for the intermediate and screwshafts respectively, the one-node 5th order critical would resonate at about 78 r.p.m., giving rise to excessive stresses of ±16,000 lb./sq. in. in the intermediate shaft and ±9,000 lb./sq. in. in the screwshaft. This stress level could be reduced to limits acceptable for transient operation, but with a wide speed restriction, by using a heavier flywheel and a lighter propeller. The



(a) With rule size shafting

(b) With oversize shafting

FIG. 25(a) and (b)—*Example 2: 379ft. Admiralty supply ship powered by a two-stroke, five-cylinder engine rated at 5,500 b.h.p. at 135 r.p.m.*

improved condition is shown in Fig. 25(a) and is about the best that could be achieved with rule size shafting.

Such a speed restriction, however, was not acceptable to the owners, and a solution was put forward based on oversize shafting of 22¼in. diameter for both screw and intermediate shafts in an attempt to place this critical sufficiently above the service speed. (See Fig. 25(b).)

As is usual in cases where greatly oversize shafting is necessary, the most highly stressed section moved into the crankshaft. In this case the stress in the 500mm. (19·67in.) diameter crankshaft is 1·38 times greater than in the 22¼in. diameter line shafting, and is higher than desirable for the full service speed of 135 r.p.m. Additional disadvantages of this "stiff" shafting solution would be, as already indicated, some loss of propulsive efficiency due to the large diameter of the stern frame, and possible service difficulties with alignment[7].

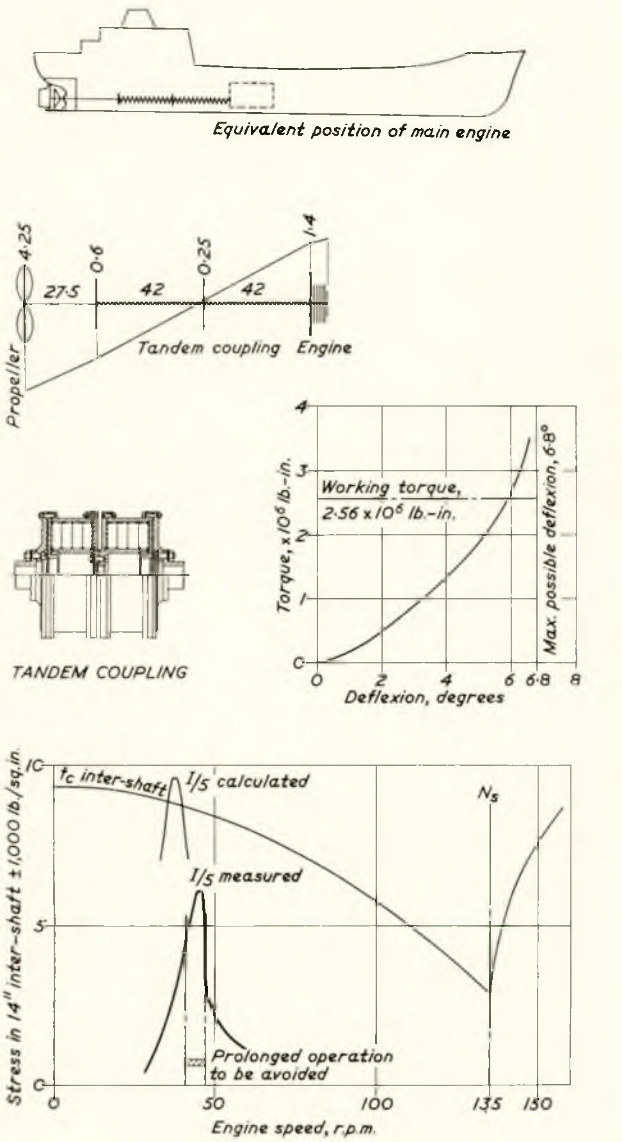
An alternative solution, put forward by Lloyd's Register of Shipping, acting in a consultative capacity, was based on the introduction of additional flexibility into the mass-elastic system, with a view to placing the major one-node critical below the minimum operating speed.

Various types of flexible couplings were considered, but only a tandem rubber-block coupling designed for maximum flexibility proved feasible for the given torque rating and clearances available. The coupling however is not capable of transmitting thrust and required a separate thrust shaft (see Figs. 25(c) and 26, in which the coupling and its torque-deflexion characteristics are shown).

The need for a separate thrust shaft is often of considerable disadvantage in the application of flexible couplings in view of the large number of engines having thrust blocks integral with the crankshaft.

The coupling, fitted with natural rubber blocks of 60 degrees Shore hardness, is 5ft. 6in. outside diameter and 4ft. 0in. long between flanges, having a dynamic stiffness of $40{\cdot}3 \times 10^6$ lb.in./rad, based on the measured position of the critical r.p.m. This stiffness is equivalent to an 85ft. length of 13¾in. diameter intermediate shafting, i.e. 21 times longer than the physical length of the coupling itself. It is worth noting here that the actual length of the remaining shafting is reduced by the length of the coupling itself, and that there is some further loss of flexibility due to the need for a separate thrustshaft, which in all could amount to a loss of flexibility of some five to ten per cent.

With this flexible coupling the one-node 5th order critical was calculated to resonate at 38 r.p.m. giving rise to a stress of ±9,600 lb./sq. in. in the intermediate shaft, the corresponding coupling twist being about ±8 degrees. Torsiograph records



(c) With flexible coupling

FIG. 25(c)—*Example 2: 379ft. Admiralty supply ship powered by a two-stroke, five-cylinder engine rated at 5,500 b.h.p. at 135 r.p.m.*

taken by the builders, and strain gauge records taken later by Lloyd's Register Engineering Investigation Department substantially confirmed the calculations, although the critical resonates at a somewhat higher speed, namely 45 r.p.m., and the measured stress of ±6,100 lb./sq. in. is somewhat lower. This stress corresponds to a coupling twist of ±5½ degrees, and an overall dynamic magnifier of 7·0 compared with the calculated value of 10·0 based on engine and propeller damping only. From the foregoing a coupling magnifier of 9·7 is deduced using the empirical relationship given in Example 1. The measured stress of ±6,100 lb./sq. in. in the intermediate shaft is equivalent to a vibration torque almost 30 per cent greater than the full load transmission torque. Accordingly it was recommended that prolonged operation should be avoided at the resonant speed, having regard to the life of the rubber blocks.

During the strain gauge investigation continuous recording was also made of the temperature inside the rubber blocks with a view to assessing the coupling damping. This was achieved with the aid of resistance thermometers inserted into the centre of the rubber block, the signals being conveyed via slip rings. After about six minutes continuous running at resonance the temperature rise was less than 10 deg. C. (18 deg. F.) indicating only a small amount of hysteresis damping. The test, however, could not be continued owing to excessive build-up of axial amplitude, causing thrust hammering, not previously apparent during the usual periods of recording. The recommendation for avoiding prolonged operation in the region of the 5th order resonant speed was retained, and was accepted by the owners. The couplings have now been operating satisfactorily in two sister ships since 1962.

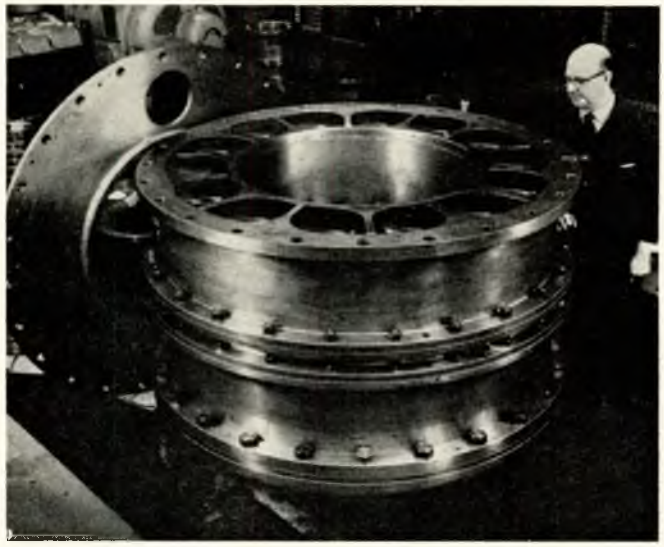


FIG. 26—*Tandem rubber-block flexible coupling used in Example 2—See Fig. 25(c) for its torque-deflexion characteristics*

Example 3—525ft. Ore Carrier powered by a Two-stroke Six-cylinder Engine of 7,500 b.h.p. at 115 r.p.m.
[see Fig. 27(a) and (b)]

This installation represents the largest application of a flexible coupling to main propulsion in the authors' knowledge.

The "natural" one-node major critical, i.e. with about rule size shafting and standard moment of inertia of the flywheel and propeller, resonates at 69 r.p.m. giving rise to a stress of ±6,400 lb./sq. in. in the screwshaft and necessitating a speed restriction [see Fig. 27(a)].

A rubber-block flexible coupling was used to place this critical at lower r.p.m. with a view to eliminating the speed restriction at about two-thirds speed [see Fig. 27(b)]. The coupling stiffness, based on measured results, is 72 × 10^6lb. in./rad, being equivalent to an additional 94ft. length of 315mm. (16.32in.) diameter intermediate shafting, which is sufficient to lower the one-node 6th order critical to 35 r.p.m. The measured stress of ±2,800 lb./sq. in. in the screwshaft is equivalent to a vibration torque only about ten per cent less than the full load transmission torque, and again prolonged operation at the resonant speed is being avoided with a view to protecting the coupling. It is understood this coupling has been operating satisfactorily since January 1963.

The example illustrates that flexible couplings can be used even in quite large main propulsion installations for the purpose of removing an undesirable critical from the working speed range and placing same at a less inconvenient speed. It is more difficult, however, to dispense altogether with a speed restriction, unless the critical can be placed below the minimum engine idling speed.

The oversize shafting solution would obviously be unsuitable in this case and was not investigated.

Flexible Couplings in Marine Geared Installations

In tugs, trawlers and coasters, the high or medium speed

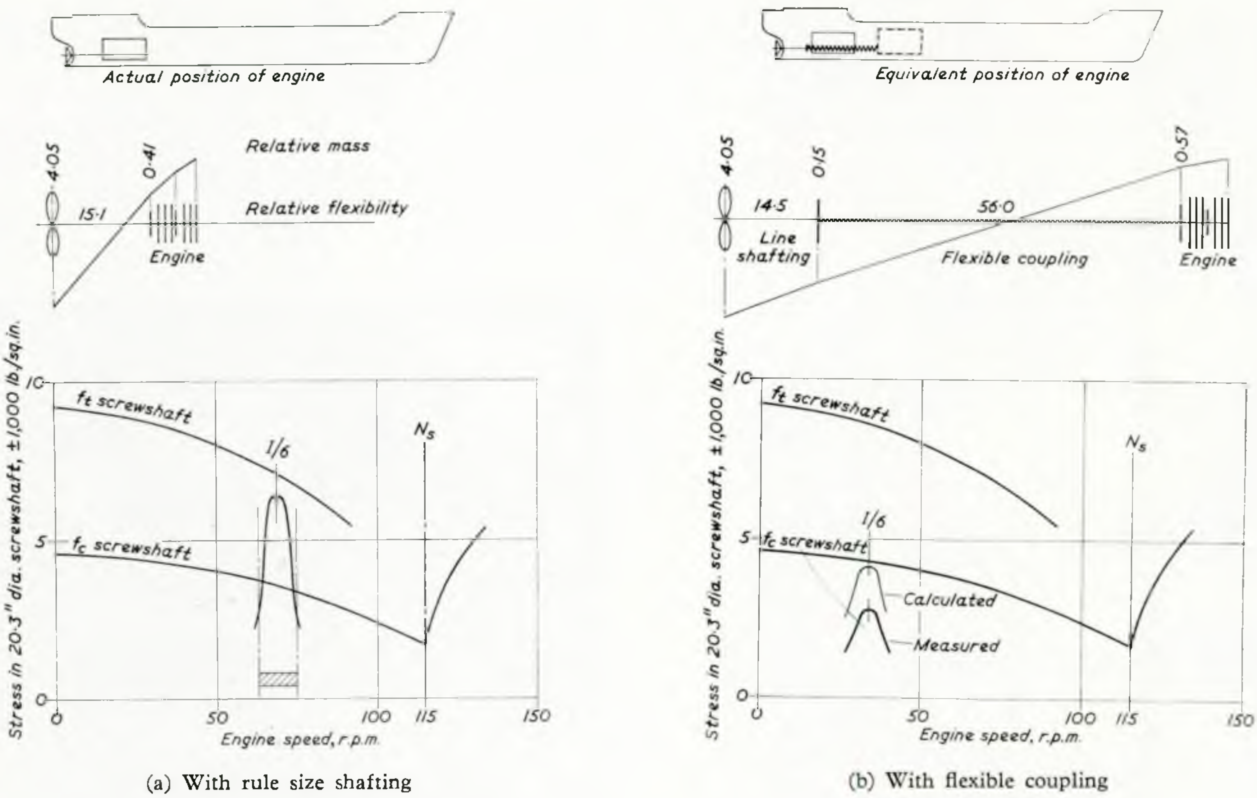


FIG. 27(a) and (b)—*Example 3: 525ft. ore carrier powered by a two-stroke, six-cylinder engine rated at 7,500 b.h.p. at 115 r.p.m.*

heavy-oil engine fitted aft and geared down to an efficient propeller speed is now the general rule. Some form of a flexible coupling between the engine and the gearbox is necessary for alignment purposes and to smooth out, to some extent, sudden torque changes.

Consideration of the torsional vibration characteristics, however, is generally the controlling factor in the selection of a particular type of coupling, as torsional criticals may easily reduce the useful working speed range and subject the gearing to excessive tooth loading.

The usual manifestation of a torsional critical in a geared installation is as audible "gear hammer", which occurs when the mean transmission torque is exceeded by the vibration torque. The resulting reversal of loading at the gear mesh causes the teeth to separate and subjects them to impact, its intensity being reflected in the severity of the gear hammer. The dynamic loading on the gear teeth depends, amongst other factors, on the magnitude of the critical and the backlash present in the gearbox. This criterion of torque reversal is an additional complication not met with in the directly driven installation, and is, on occasions, difficult to avoid, especially at lower revolutions.

Lloyd's Register guidance notes on torsional vibration recommend that in all cases where there is possibility of gear hammer, the backlash in the gears should be kept to a minimum.

Gear hammer as such is audible when the frequency of tooth separation is less than about 20 c/s (1,200 cycles/min.). At higher frequencies the individual impacts cannot be distinguished and are heard as one note.

Tooth separation is particularly likely to occur in the lower speed range where transmission torque is low and can easily be exceeded by even comparatively low vibration torque. Thus, the presence of some gear hammer near the idling speed need not necessarily be indicative of high tooth loading and need not cause undue alarm. Severe gear hammer, on the other hand, should always be avoided.

In some cases where it is not possible to remove or eliminate all the offending criticals in the speed range, within the framework of practical and economic considerations, some limitation has to be accepted on the working range of engine revolutions. Here the controllable pitch propeller has advantage over the fixed pitch in enabling any ship speed to be obtained from a limited range of, or even constant, engine r.p.m.

The design tooth loading can also be exceeded without the audible warning of gear hammer when the sum of the mean transmission and of the vibration torques is greater than the rated torque, whilst at the same time the vibration does not exceed the mean torque. Theoretically such a condition is possible where the mean torque is greater than one-half of the full load torque, i.e. at speeds higher than about 70 per cent of the full load r.p.m., in installations where the propeller torque is proportional to the square of the r.p.m.

In practice, however, the criticals that prove troublesome in this respect resonate mainly close to or above the speed corresponding to the full load torque.

Lloyd's Register latest guidance notes on torsional vibration[4] recommend with reference to reverse-reduction gearing that for critical speeds near the maximum speed the vibratory torque should not, in general, exceed one-third of the full transmission torque.

In cases where such conditions are likely to arise and it is difficult to tune out the offending criticals, torque-limiting couplings of the centrifugal type are sometimes used, being particularly advantageous where it is possible to arrange the dynamic system so that the vibration torque on the coupling is greater than on the gear mesh. In this way the excess vibration torque causes the coupling to slip without subjecting the gearing to excessive loading.

The problem, however, can be more complicated in tugs, which under towing conditions, develop the full load torque at considerably lower r.p.m. than when running free, or where a controllable pitch propeller is fitted, and in such cases the slip coupling must be designed for the highest torque rating.

The foregoing notes generalize some of the considerations involved in the application of flexible couplings in geared main propulsion marine installations. More detailed considerations are illustrated by the following example in which the effects of flexible couplings of widely different characteristics have been investigated with a view to obtaining the most satisfactory solution of the torsional vibration characteristics.

It can also be noted here that in aft end installations torsional vibration criticals are usually more difficult to deal with than when the engines are placed amidships. This is because with a short and thus comparatively stiff line shafting the different modes of vibration are separated in frequency to a lesser extent and may interact one with another.

In some instances this has been overcome by fitting a flexible coupling in the line shafting. In considering this solution, however, for geared installations, it should be noted that the size and cost of the coupling are considerably greater than those of couplings fitted between the engine and gearbox. This is due to the lower running speed of the line shaft, and also to the need for the coupling to be capable of transmitting the ahead thrust and astern pull. Further disadvantages are that the range of flexibilities available for such couplings is considerably less extensive than for non-thrust couplings, and that their moment of inertia is comparatively high.

Example of a Typical Geared Marine Installation

This example of machinery for an 85ft. twin-screw tug, comprising a six-cylinder, two-stroke oil engine, developing 690 b.h.p. at 425 r.p.m. and driving a propeller via reverse/reduction gearing of 2:1 ratio, is typical of conventional aft-end geared marine propulsion systems in the medium speed range.

Such systems commonly exhibit three predominant modes of torsional vibration, associated characteristically with the propeller-shafting, gearing, and crankshaft regions respectively, and usually in that order of ascending frequency.

As initially designed, this particular installation incorporated a pin-and-bush type flexible coupling between the flywheel and gearing, primarily to accommodate alignment errors and consequently of relatively high torsional stiffness comparable with the stiffness per crank throw. This feature is demonstrated in Fig. 28(b), where the equivalent shaft lengths in the mass-elastic system are referred to crankshaft speed and drawn to scale. Fig. 28(a) shows the actual position of the engine in the ship.

Estimates of vibration torque in the gearing are shown in relation to the mean transmission torque curves based on the propeller law for the towing and free running conditions, in Fig. 28(d). The major critical of the one-node mode (the shafting-propeller mode) resonates comparatively low in the speed range and is satisfactory, but the two- and three-node modes (the gearing and crankshaft modes) 6th order major criticals are calculated to resonate rather close together in the upper speed range and give rise to severe vibratory torques in the crankshaft and gearing, in spite of a large viscous damper fitted at the crankshaft free end. This solution, therefore, is considered unsatisfactory, especially for a tug.

As a first step towards improving this system, the effect of increasing the torsional flexibility of the coupling was investigated. It was found that the frequencies of the 1st and 2nd modes ("shafting" and "gearing") could be reduced progressively, leaving the 3rd mode relatively unchanged in frequency and almost purely "crankshaft" in character. However, in order to place both one- and two-node, 6th order criticals satisfactorily below the minimum operating speed, a thirty-fold increase in the coupling flexibility was found to be necessary. This presented no difficulty, the required dynamic stiffness of $2 \cdot 0 \times 10^6$ lb.in./rad being obtainable within the range of modern highly flexible couplings.

In terms of shafting lengths, similar flexibility could be obtained from a 19½ft. length of approximately rule size (4½in. diameter) shafting running at engine speed. The relative position of the engine in the ship if such a shaft were used instead of the coupling is shown in Fig. 29(a). If a quillshaft of higher U.T.S. material were substituted, this length could be approximately halved. Nevertheless, this solution was not entirely satisfactory in other respects as reference to the torque diagram Fig. 29(d), shows. Here the two-node 2nd order critical excited by the reciprocating scavenge pump is almost coincidental with the one-node 4th order propeller-excited critical, and jointly could induce excessive vibratory torque in the gearing.

Such conditions are liable to arise when, for example, a four-bladed propeller is used in conjunction with an approximately 2:1 reduction gear ratio, giving in effect 2nd order excitation at crankshaft speed which could become additive to engine excitation of the same order. Comparable conditions could arise with a three-bladed propeller and 3:1 reduction ratio giving in effect 1st order propeller excitation at crankshaft speed, and could lead to difficulties if there was also a significant 1st order engine excitation.

The final stage in the design process of producing a system having acceptable vibratory characteristics throughout the running range was achieved by resort to a "two-rate" flexible coupling.

Such couplings have low torsional stiffness, based on rubber in shear, which remains effective up to a given value of coupling twist and therefore transmission torque. Above this value any further deflexion is constrained by additional compressive rubber buffer elements which increase the coupling stiffness. The torque deflexion characteristics of a coupling of this type and the coupling itself are shown later in Figs. 31(e) and 32.

It is common practice to arrange for the transmission torque at which change-over from low to high stiffness occurs to be in the region of 25 per cent full load torque or 50 per cent of maximum revolutions, based on the propeller square law. In tugs additional account has to be taken of the towing and free running torque rating, which gives in effect a range of change-over speeds. Comparable conditions exist also where a controllable pitch propeller is fitted.

Effectively, then, because of the change in the stiffness of the coupling at a given value of mean transmission torque, different natural frequencies exist on each side of the change-over speed and in consequence it is possible to eliminate the more important criticals from the whole operating speed range. Placing of significant criticals in the change-over region, however, should preferably be avoided.

Returning to the example under consideration, in the "low" coupling stiffness condition, illustrated in Fig. 30, the major one-node and two-node criticals resonate well below the minimum engine idling speed. The two-node 2nd order critical which would resonate just below the service speed with the "low" stiffness is non-existent as at that speed the coupling is already working in the "high" stiffness range.

It is worth mentioning here that the "low" coupling stiffness of $0 \cdot 38 \times 10^6$ lb.in./rad represents a very high degree of flexibility, being equivalent to the flexibility of 102ft. length of rule size shafting, which is obviously impossible in this ship as is clearly evident from diagram Fig. 30(a).

Fig. 31 shows the corresponding conditions for the upper speed range where the "high" coupling stiffness is effective. Here again it will be seen that all major criticals likely to create objectionable conditions in the gearing have been eliminated, leaving only the combined one-node, 2nd engine/4th propeller order, and the two-node 3rd order as criticals of any consequence in the range.

From the calculations it can be predicted with confidence that neither of these criticals will be severe enough to cause torque reversal at the gear mesh, or be otherwise objectionable. However, from past experience of comparable installations where light gear hammer had in fact occurred, this had been eliminated by the fitting of telescopic dampers to the coupling.

FIGS. 28-31—*Example of typical geared marine installation—An 85ft. twin-screw tug powered*

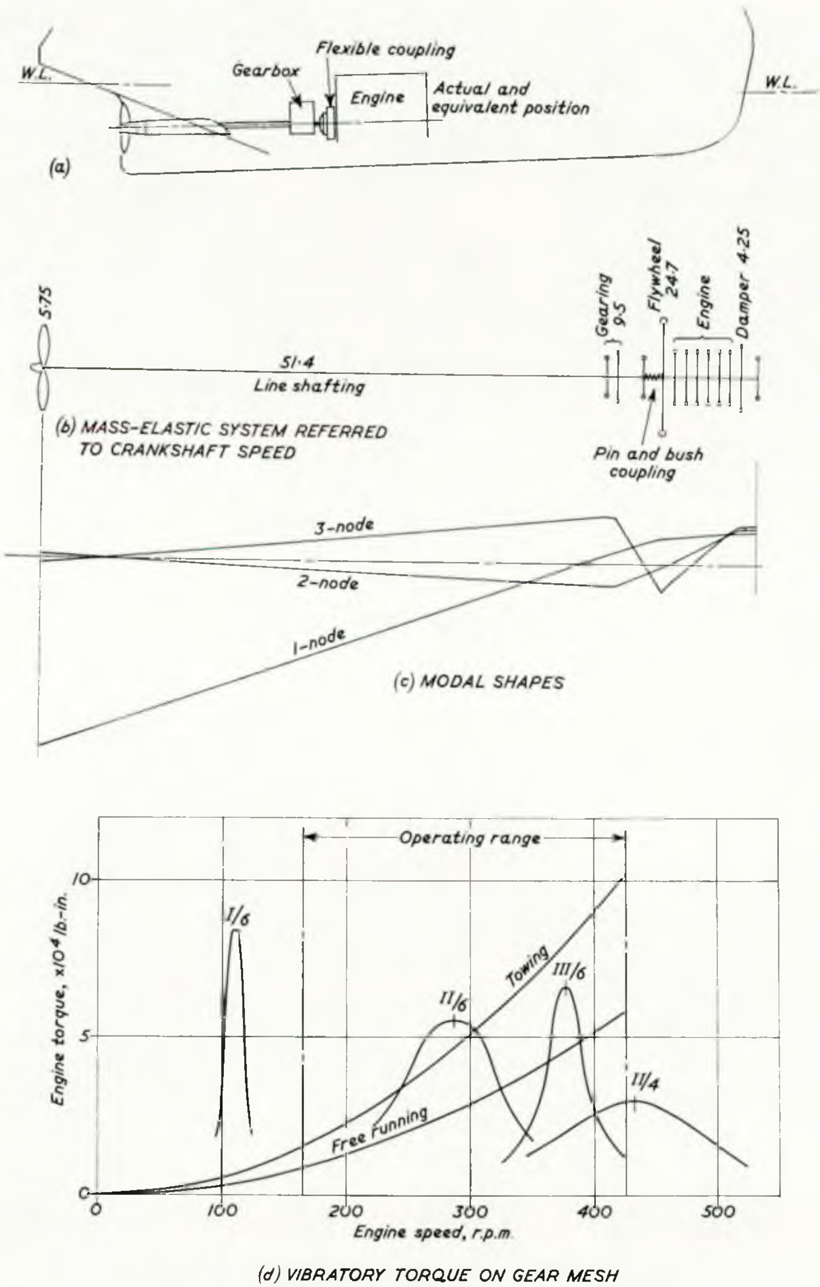


FIG. 28—*With a stiff pin-and-bush coupling having a dynamic stiffness of 62 × 10^6 lb. in./radian*

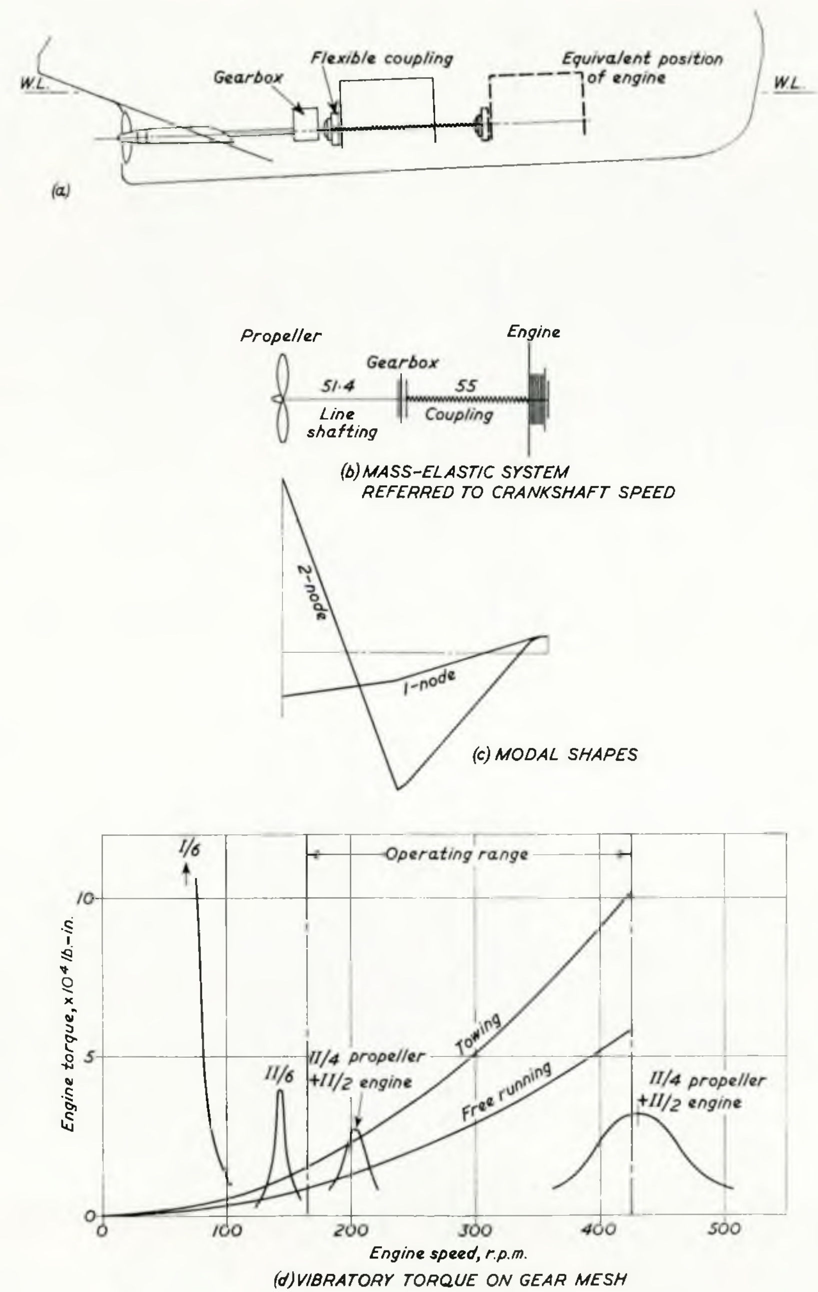


FIG. 29—*With flexible coupling having a dynamic stiffness of 2·0 × 10^6 lb. in./radian*

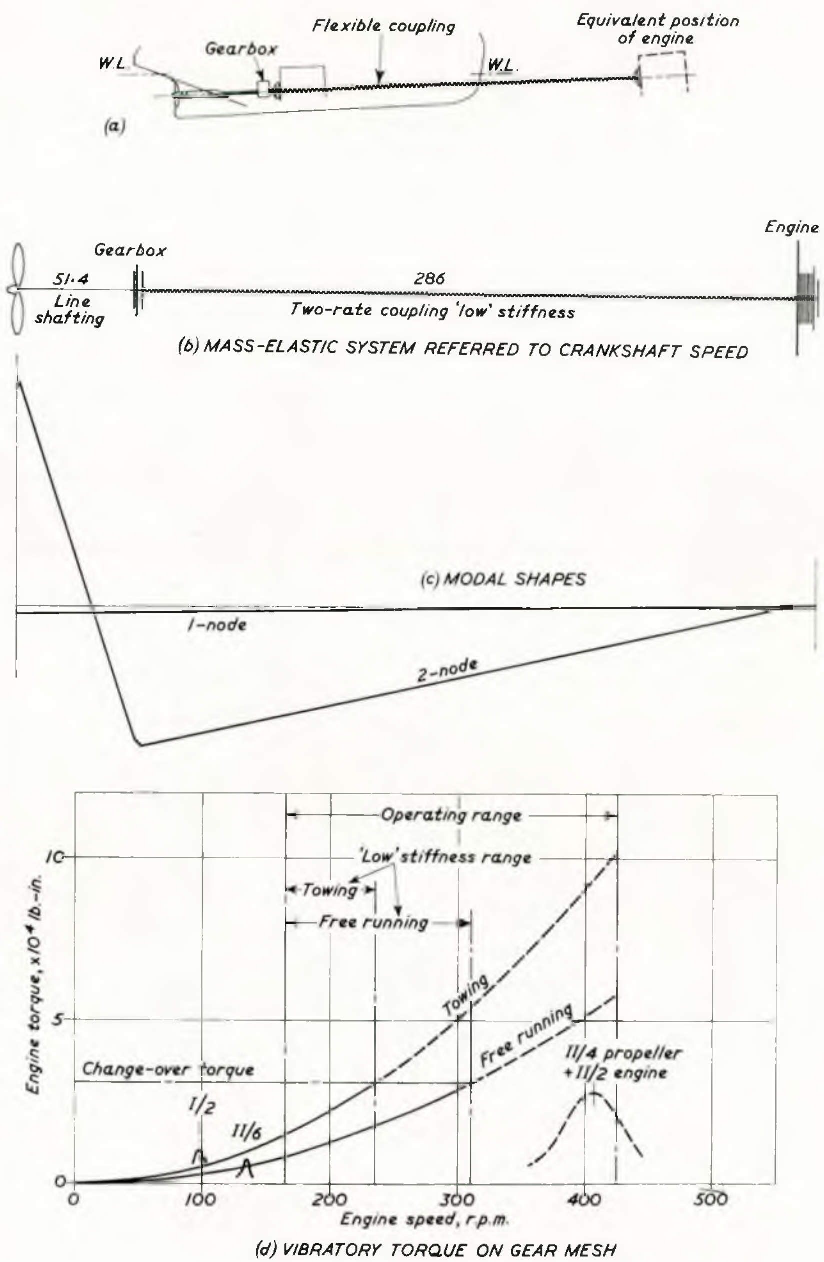


FIG. 30—*With two-rate flexible coupling; "low" dynamic stiffness of 0·38 × 10⁶ lb. in./radian (0·032 × 10⁶ lb. ft./radian)*

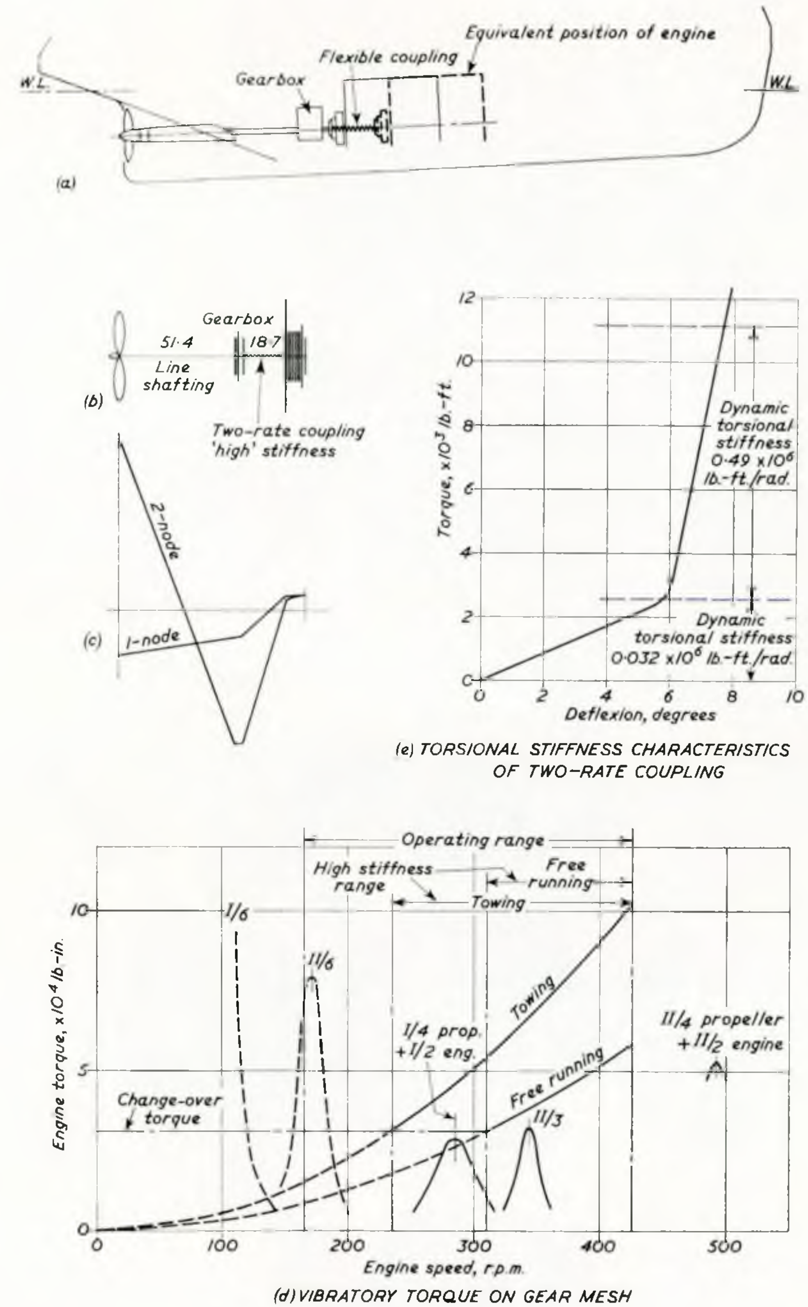


FIG. 31—*With two-rate flexible coupling; "high" dynamic stiffness of 5·9 × 10⁶ lb. in./radian (0·49 × 10⁶ lb. ft./radian)*

by a six-cylinder, two-stroke oil engine rated at 690 b.h.p. at 425 r.p.m. with 2 : 1 reduction gear

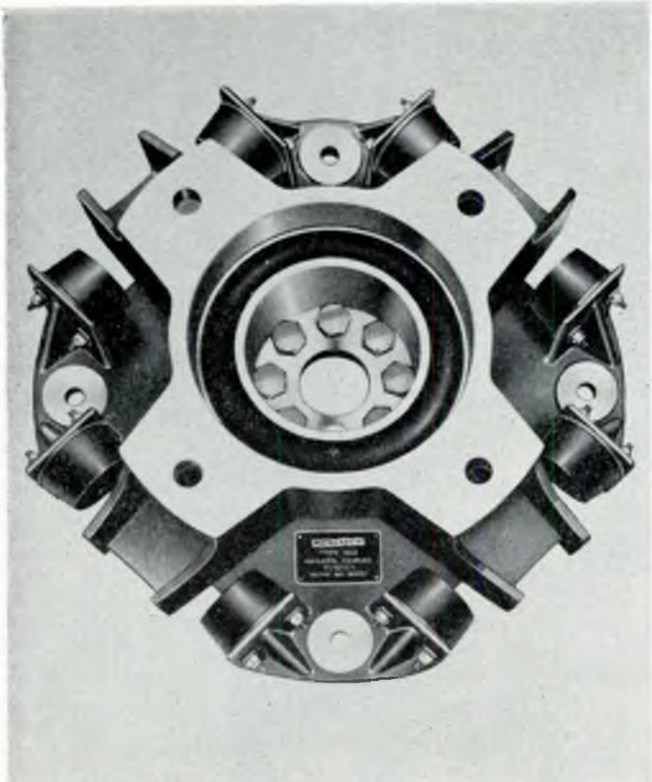


FIG. 32—*Two-rate flexible coupling described in the geared example—See Fig. 31(e) for its torque-deflexion characteristics*

CONCLUSIONS

The interposition of a flexible coupling for alignment purposes will usually affect also the torsional characteristics of the system. The choice of a coupling must therefore be considered not only in the light of its suitability to accommodate the misalignment and movements which may take place in the transmission system, but also in regard to the effect on the torsional characteristics of the complete installation. On the other hand, torsionally elastic couplings can also be used to good advantage purely for torsional tuning.

With this in view it is necessary for the coupling characteristics to be specified with reasonable accuracy if excessive factors of safety are to be avoided in the application of the coupling. The most effective means of obtaining such basic knowledge of the coupling characteristics is by laboratory testing under conditions controlled within the required limits.

The use of a mechanical analogue for the study of the behaviour of non-linear couplings gives qualitative results in agreement with theory for a system with the minimum of damping.

These results indicate that the indiscriminate use of non-linear couplings may not give the conventionally expected detuning effect, since this depends upon a combination of preload, vibratory torque and the direction of speed change in relation to the coupling characteristics.

It was shown in the examples selected that, in Diesel-engined direct drive installations, the introduction of a flexible coupling can provide the difference in torsional flexibility that exists between aft-end and amidships installations, and further, that torsionally flexible couplings can be used up to quite large torques.

These examples also show that in cases where the "natural" one-node major critical, i.e. with rule size shafting and standard moments of inertia of the flywheel and propeller, resonates below about three-quarters of the service speed, it is generally preferable to adopt the "flexible" solution as otherwise greatly oversize shafting would be necessary to place the critical sufficiently high above the service speed.

In the authors' opinion the application of the "flexible" shafting solution will become more widespread since the reduction of rule scantlings for intermediate shafts in the new Lloyd's Register Rule revisions will tend to lower the "natural" position of the one-node major criticals, and therefore, make this solution more attractive. In this connexion, where only a limited amount of flexibility is required, use is sometimes made of intermediate shafting of higher U.T.S. material. Where, however, a considerable amount of additional flexibility is necessary then recourse must be had to flexible couplings with rubber as the working medium, those employing rubber in shear being generally softer although limited to lower torque ratings.

Torsional vibration, both theory and practice, is now at a sufficiently advanced stage of development to enable a reliable assessment at the design stage of vibratory conditions which will occur in service. Provided such assessment is done at a sufficiently early stage, a wide field of possible solutions can be quickly studied with the aid of computers, and a sound and economically acceptable solution can be obtained in most cases.

ACKNOWLEDGMENTS

The authors express their thanks to the Committee of Lloyd's Register of Shipping and to the British Internal Combustion Engine Research Institute Ltd., for leave to publish this paper.

Thanks are due also to owners, builders and manufacturers for permission to include valuable information.

They also gratefully acknowledge the help given by their colleagues, especially Mr. D. H. L. Inns for his assistance with the geared example.

REFERENCES

1) WHITE, D. J. B.I.C.E.R.A. Research Report No. 62/4: "Torsiograph Tests of an Engine System Incorporating a Powder Coupling".
2) *Oil Engine and Gas Turbine*. June 1964. "Introducing a Zero-Stiffness Coupling".
3) ARCHER, S. 1949. "Screwshaft Casualties—The Influence of Torsional Vibration and Propeller Immersion". Trans. I.N.A., Vol. 91, p. J56.
4) "Guidance Notes on Torsional Vibration Stresses and Critical Speeds for Main and Auxiliary Oil Engines", Chapter R(E) of Lloyd's Register, 1965, Rules and Regulations for the Construction and Classification of Steel Ships.
5) KER WILSON, W. 1941. "Practical Solution of Torsional Vibration Problems", Vol. 2, Second Edition, Chapman and Hall.
6) B.I.C.E.R.A. 1958. "A Handbook on Torsional Vibration", Cambridge University Press (compiled by E. J. Nestorides).
7) ARCHER, S. "Marine Propulsion, With Special Reference to the Transmission of Power". The thirty-sixth Thomas Lowe Gray Lecture, I.Mech.E., Advance Copy.

Discussion

MR. E. W. CRANSTON, Wh.Sc. (Member) thanked the authors for inviting him to open the discussion on this most interesting paper, which gave a very clear explanation of the benefits to be derived from the use of flexible couplings for the transmission of power. With one or two exceptions, all the main engines provided by his company and the licensees were directly coupled to the propellers and no flexible couplings were necessary. The exceptions had been geared installations and Example 2 (pages 393 to 395 of the paper).

Most of the geared installations had had two or four engines per shaft and either hydraulic or electro-magnetic couplings had been preferred for fitting between the engines and gearbox. Two of the main reasons for this were dependent on the possibilty of uncoupling the engines easily by either emptying the hydraulic couplings or de-energizing the electro-magnetic couplings. With an installation with two engines per shaft, one engine could be run ahead and the other astern so that manœuvring could be carried out simply by driving the propeller through one coupling or the other without having to keep reversing engines. Also an engine could be easily disconnected from the drive for adjustments and overhaul whilst the ship was run at reduced speed.

For installations with one engine per shaft the advantages of these couplings did not apply and flexible couplings would be preferred because of their lower cost and simplicity of application. The paper described, on page 396, why the torsional vibrations must be considered in order to prevent gear hammer, but made no mention of cyclic torque variation of the engine which might be of importance at low speeds where there was a light flywheel and a small number of cylinders.

Most high powered direct drive engines nowadays had pulse type turbocharging and in order to obtain reasonable turbocharger efficiency the crank angles and firing order of the engine, whatever the number of cylinders, must be chosen carefully and only a limited number of arrangements was possible. These were further reduced by reasons of balance of the engine masses. There remained the question of obtaining satisfactory torsional critical speed conditions.

Usually it was possible to proportion the weight of the flywheel and the diameter of the shafting so as to produce the desired conditions. In some cases an additional flywheel might be required at the forward end of the engine and in others a damper of the viscous fluid type must be fitted.

In Example 2 (pages 393 to 395) it was clear that a large increase in shaft diameter was not desirable. The authors omitted to state the difficulties that would have arisen in lifting the propeller shaft inboard for examination because of the excessive weight and restricted space available in the ship.

The use of a damper in this case with rule size shafting was considered, but had little benefit because it would not reduce sufficiently the stress caused by the single-node 5th order critical vibration. No guarantee could be given that this stress would not exceed the transient limit, f_t, and it was obvious that another solution was necessary.

The suggestion to fit a flexible coupling, put forward by Lloyd's Register of Shipping was, in 1961, rather unusual for transmitting such a large torque under marine conditions. The solution was acceptable to all concerned and he could confirm that in service no troubles had been experienced.

Without the use of the flexible coupling, the shipowner would have had either to fit another type of engine requiring more space or to install the engine amidships with a loss of valuable cargo space.

Most of the flexible couplings mentioned made use of rubber stressed in various ways. When rubber aged it usually became harder and this would reduce the flexibility of the couplings. This factor should be taken into account when calculating the critical speed conditions, and if necessary the critical speeds could be checked on board ship after some years of service. In some cases it might be preferable to use couplings with replaceable rubber pads which would allow the couplings to be brought back to their original flexibility at any time.

One difficulty, which had been met more than once, was the reluctance of the shipowner or the consulting engineer to authorize the fitting of a flexible coupling for a direct drive. It was hoped that this paper would help to remove doubts on this matter and would explain to them some of the benefits which could be obtained from flexible couplings.

MR. R. W. ZDANOWICH, M.A., said that the authors were acknowledged experts in the field of torsional vibrations and he felt that he could speak for all present when he said that they welcomed such an authoritative paper on the subject of couplings, the more so since it described tests and experiments conducted by the authors themselves.

The question of the difference between static and dynamic value of coupling stiffness was first discovered as far back as 1936 while experimenting with various sizes and hardnesses of wedge type couplings fitted in the transmission systems of Napier Lion installations on board torpedo boats and rescue launches. This system was essentially simple and readily lent itself to the prediction of torsional frequency. For a "bare" installation there was a practically 100 per cent agreement between observed and estimated frequencies. Fitting of wedge type couplings resulted in serious discrepancies between the two values and eventually led to the evaluation of a stiffness coefficient which, for the type of coupling, grade of rubber and frequency in question, averaged about 1·65 for rubber of 49 S.H. and 2·65 for stiff rubber of 87 S.H. These results were subsequently communicated to Messrs. Metalastik, the suppliers of the couplings, who instituted research on the subject and confirmed the findings. Soon after all reputable rubber manufacturers acknowledged the phenomenon, now universally accepted.

Regarding the physical properties of rubber, those interested in the subject from the standpoint of application to engineering problems, were referred to the speaker's paper with Mr. Moyal in 1945*.

A useful expression had been derived:

$$\lambda = \frac{Cd}{C} = \frac{\eta}{G}$$

Where λ = Ratio.
Cd = Damping or internal friction coefficient.
C = Dynamic stiffness.
η = Cœfficient of normal viscosity (stress per unit gradient of velocity).
G = Shear Modulus.

* Zdanowich, R. W., and Moyal, J. E. 1945. "Some Practical Applications of Rubber Dampers for the Suppression of Torsional Vibrations in Engine Systems". Proc.I.Mech.E., Vol. 153 (War Emergency Issues 1-12), pp. 64-67.

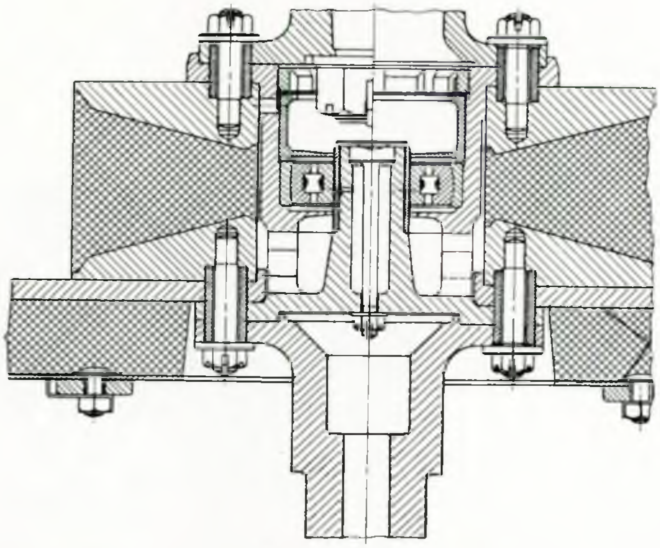


FIG. 33—*Power boat installation—Section*

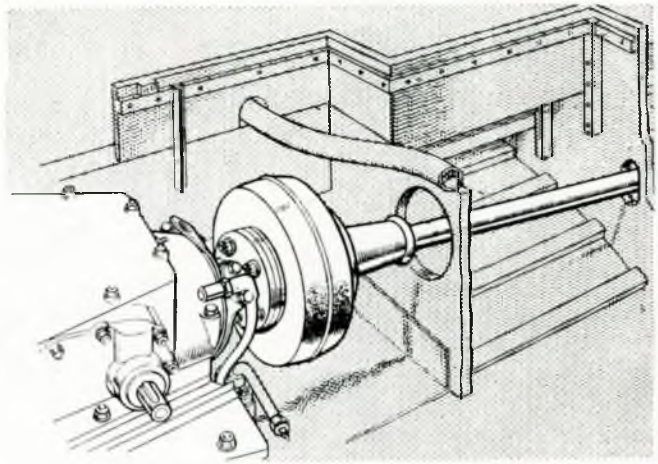


FIG. 34—*Power boat installation—Side view*

The paper in question also gave an expression for dynamic magnifier.

The "half-moon" coupling sounded interesting and the results quoted showed promise. It was certain that most of those concerned would impatiently look forward to qualitative results of tests under actual service conditions. The question of reliability was, of course, of prime importance and, providing that potential users could be satisfied on that score, the coupling might well constitute an answer to the proverbial "maiden's prayer".

One type of coupling which deserved a mention was a rubber "wedge" or "vee" type, already touched upon (see Figs. 33 and 34).

Such a coupling had several advantages compared with other types. First, owing to its construction, the stress in rubber was constant from the inside to the outside radius. The design permitted angular mal-alignment without undue stress either in the bond or in the rubber itself. Some radial mal-alignment was also permissible. The centralizing bearing was fitted since the coupling had to transmit the propeller thrust. Owing to the large volume of rubber present it also acted as an absorber. In the case of the installation shown the reduction of the torsional severity was of the order of 30 per cent for the engine flywheel mode, not directly affected since the coupling was fitted in the propeller shaft as shown. There was also a marked decrease in noise. One disadvantage of this type was its weight and bulk, essential to keep the stress to within 60lb./sq. in. for long life. The coupling was, thus, mainly suitable for low to medium powers developed at high engine speeds.

The so-called non-linear steel spring couplings deserved particular mention. Three examples were given in Fig. 2 on page 384 of the paper. Of these one needed only to consider the third, incorporating the continuous grid spring, as developed and successfully applied to so many installations by the late Mr. James Bibby. It was almost universally described as a non-linear stiffness coupling and this characteristic was

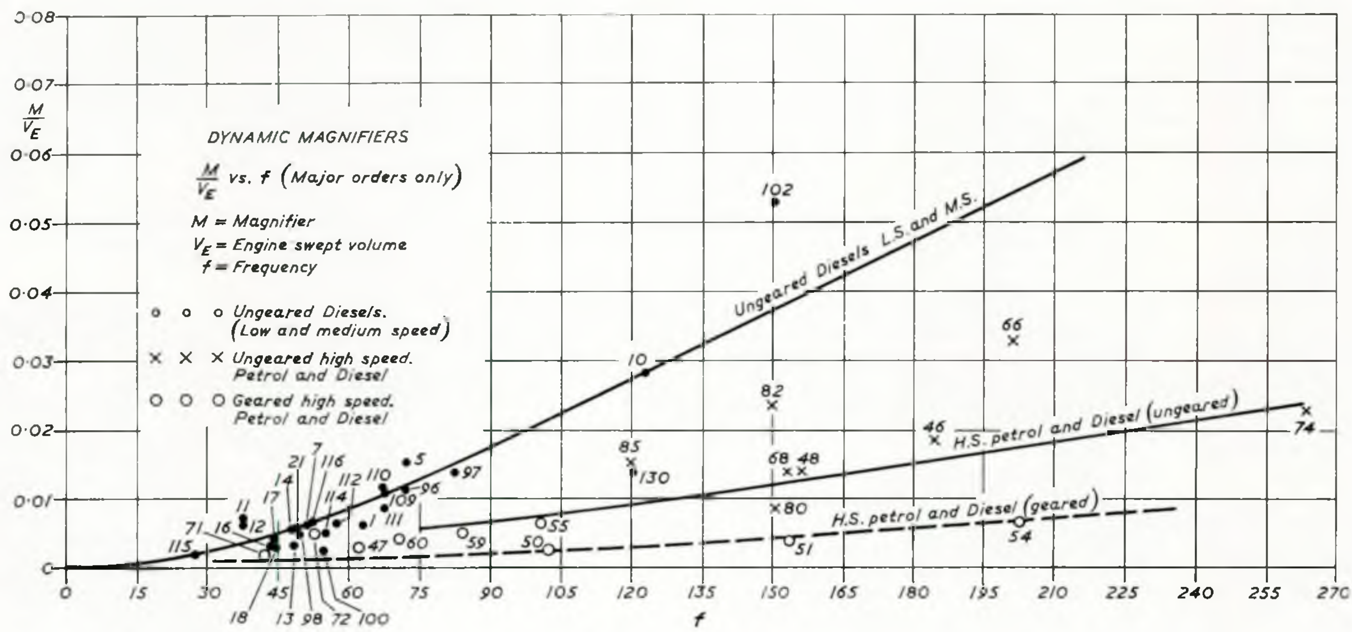


FIG. 35—$\frac{M}{V_E}$ *vs. f*

supposed to "detune" the system. While this might well occur in some installations, in the speaker's opinion, the coupling was primarily a transmission damper and a "shock absorber". In support of this statement let us consider the effect of gearing in the transmission (see Fig. 35).

$\frac{M}{VE}$ versus f (existing)(see Fig. 35)

This figure had been already shown before* (see *Trans. I.Mar.E.*, April 1964, p. 111) and while the relationship taken was open to argument, the relative effect of the introduction of gearing was unmistakable and showed that, other things being equal, the gearing more than halved the severity of the vibration condition. As stated before, the large extra damping was mainly due to the shearing of oil film between the teeth of the gears. As was, of course, well known, in spur gears there were seldom more than about $1\frac{1}{2}$ pairs in mesh, yet even that was sufficient, other things being equal, to halve the severity.

In the Bibby type coupling there were many "pairs in mesh", the shearing action being due to the relative angular movement of the two halves of the coupling and consequent substantial sliding motion between each portion of the spring and flared groove (see Fig. 36).

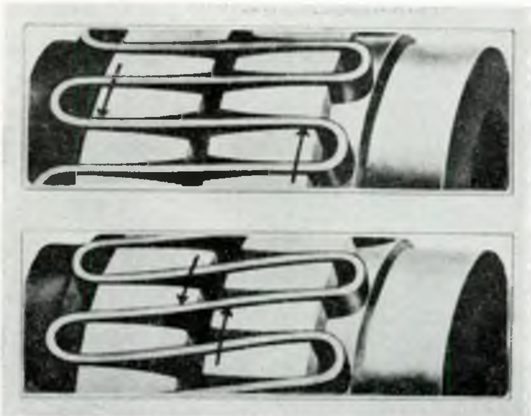


FIG. 36

This at once stressed the importance of the end loops of the continuous grid spring. If these loops were removed and each portion of the spring clamped at the end, this important characteristic would be lost and the coupling would act mainly as a detuner.

Of course, by virtue of its construction, the coupling also acted as a successful shock absorber, which explained why so many of them were incorporated in the gear drives of rolling mills in steel works. The cushioning effect of the coupling appreciably prolonged the useful life of the driving gears and frequently enabled gears previously discarded to be used again.

Another promising field of application was in cases when a small engine was called upon to drive a very large mass like a large propeller or generator. It had been found by experience that whenever the inertia of the latter was more than three times greater than that of the engine, trouble could be expected.

While the actual severity of the conditions would depend on the specific output, frequency, top speed and other factors which might override the basically unsatisfactory conditions, in case of difficulties, a Bibby type coupling was the quickest and cheapest remedy, far more satisfactory than the lengthy and costly development work and far more reliable than most types of damper known at present.

Fig. 37 showed a very successful application of a Bibby coupling to a single-cylinder compound test unit.

The unit was designed for an unusually high output but suffered from such severe torsional conditions that it never attained more than about half the rated output and even then kept on smashing itself up. This went on for a long time, until the fitting of Bibby couplings, one between each crankshaft and its output gear, completely cured the trouble and for the first time it was possible to develop full power without further mishaps. In addition to that, the unit became incomparably quieter.

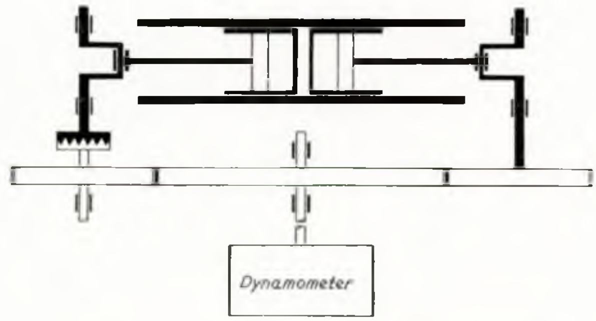


FIG. 37—*Diagrammatic layout of single-cylinder compound test unit—Bibby coupling*

There were, of course, certain conditions which must be rigidly observed for a successful application. Indiscriminate fitting could only lead to difficulties and was bound, in the long run, to bring the coupling into disrepute—a fate which so frequently overtook many otherwise excellent devices. In cases like these it was the technician or the management who were at fault, or both, not the coupling. It was incredible that in some quarters even a mention of a Bibby coupling provoked merriment, just as if the whole thing were a joke. Bibby left £496,000—starting with a bank loan—one did not leave such a tidy sum just by selling jokes, after all it was quite impossible to mislead all the people all the time.

The authors had made a thorough study of the effects of non-linearity. These constituted a valuable addition to the store of knowledge but, in the speaker's opinion, did not add much to the understanding of the functioning of the Bibby type coupling, the main difference being that in the model shown on page 390, Fig. 18, the powerful shearing action between the springs and the flared grooves was absent. It was true that the tip ends of the leaf-spring, placed between the two shaped profiles, could slide in the fixing slots provided, but the effect was too slight to simulate the functioning of the Bibby coupling. It would seem that the action was more nearly akin to the type which consisted of a number of independent clamped plate springs. It would be interesting to know what would happen if two or more springs were fitted in the model described. If the friction between such springs should prevent the building up of peaks of resonance, it would constitute a further proof of the Bibby coupling acting as a transmission damper. Perhaps the authors would care to consider this point in some detail.

The example of the application of flexible couplings to geared drives was interesting and showed how much extra scope was now in the hands of the marine technician. The aircraft industry was and had been fully aware of the advantages and one frequently saw a flexible connexion between the crankshaft and output gears—often in the shape of a quill. Moderate torque reversals due to vibratory conditions had been and were quite common and a surprising number of aero-engines, even of the "household word" variety, suffered from them without any apparent ill-effects.

Regarding the two-rate rubber flexible coupling, the speaker felt that similar results would follow an intelligent application of a Bibby coupling with the added advantages of large extra damping and of more exact prediction of the conditions. The last point was important since the physical properties of rubber could not be guaranteed to within narrow limits, and, if overstressed or overheated, were liable to change.

The references to geared engines, even though confined mainly to small scale installations, were gratifying since they

* Zdanowich, R. W. Contribution to discussion on "Some Factors Influencing the Life of Marine Crankshafts". Trans.I.Mar.E., Vol. 76, p. 111.

showed that geared drives were slowly but surely being accepted in marine engineering. The aircraft industry had been using gearing extensively for many, many years and for quite large powers. Thanks to the pioneering work of Dr. Merritt and others, the technique of gear design and manufacture was so satisfactory that gears seldom gave any trouble, in fact they gave less trouble than many other components.

It was to be hoped that this outlook would be eventually assimilated by the marine industry since a high powered high speed geared engine offered great advantages. An engine developing several thousand horsepower for a total weight of, say, about 6 tons, overall length a little over 12ft. and height of only 7ft. could be installed in an odd corner thus enabling the saving of space to be used for carrying extra payload. In some cases the gain in space and weight made it possible to carry so much additional cargo that the extra profits were sufficient to pay the wages of the crew. It was hoped that this lesson would not be ignored by the British shipowners and shipbuilders. The shipping industry did not appear to be in a very healthy state at present and competition from Germany and Japan seemed to be a little too much for it. The fact that ships with small geared engines were more economical and could earn more would tend to counteract the effects of higher first cost and uncertain delivery dates.

It made little sense to talk about increasing exports and closing the trade gap if lessons of that kind were not taken to heart.

Admittedly high powered geared Diesel drives would present the marine engineer with less familiar matters of design, installation, performance, etc., in addition to the basic vibrational effects like gear detuning, non-linearity and increased damping.* The classification societies likewise would be faced with many new problems and these would need consideration when the essential revision of the Rules became necessary.

Mr. C. W. Chapman said that the paper was not only of great interest but most timely, for these days flexible couplings were becoming more and more important in both marine and all other engineering applications.

He wished to say something about the "half-moon" coupling mentioned in the paper. He had always felt that if one could only get a coupling with no stiffness and yet capable of transmitting torque, torsional vibrations would be a thing of the past. But, of course, as had been said in the paper, the true zero stiffness coupling over its whole deflexion range was if not actually impossible, certainly undesirable because of its instability. This "half-moon" coupling was worked out on paper in the first place for powers varying from 30 to 50 h.p. at 2,000 or 3,000 r.p.m., and from 2,000 to 3,000 h.p. at 100 to 400 r.p.m., and it seemed to be theoretically possible in reasonable dimensions, and comparatively inexpensive. As however, the principle was new and untried, Mr. Chapman and his colleagues had contacted B.I.C.E.R.I., and asked them to look into the theory, see if they could make a model rig and find out whether the effect was as had been envisaged. That was all described in the paper. Fig. 14 showed the comparison between the actual recorded results and the theoretical results. The test results should really be moved about 2½ deg. to the left because this coupling, at neutral position, was so extremely soft that even the friction of the engine bearings could deflect it several degrees, with the result that, when the test was over and it was checked, it was found that the zero mark for the stroboscopic recordings was about 2½ deg. out.

It was interesting to note how using so soft a coupling had so effectively reduced the vibration amplitudes. In the figure that showed the two-node vibrations, where the coupling was running 10 degrees away from its neutral position, the stiffness was approximately 2,400 lb.in./radian as against 32,500 lb.in/radian for the already very soft known coupling.

However, this was only a small scale laboratory test and everything else was just on paper, so a full scale "half-moon" coupling was fitted and tested on a six-cylinder engine, of 797 h.p. at 600 r.p.m., in a tug, where the conditions would be most arduous. The coupling had to be made strictly interchangeable with the existing coupling for the job, and according to calculations there would be a two-node 3rd order even with coupling stiffness zero, at about 233 r.p.m. in the running range. Just what effect this would have was not known, so dampers that could be let in or taken out of action quickly were incorporated in case the two-node 3rd order had to be damped. He understood that the dampers were found to be completely unnecessary. He had not seen any figures, but was given to understand that no very noticeable vibrations were recorded anywhere in the speed range. He further understood that both authors were present at the trials, both the basin trials and the sea trials, and that Mr. Chartan, to be on the safe side and to make really sure what was happening, asked for further tests with strain gauges on the gearbox input shaft and also on what they called the "half-moon" springs. The final figures had not yet been worked out, but he hoped that when they had, Mr. Chartan would be able to incorporate them in the reply to the discussion.

Mr. A. O. Hunt said that he found the paper most interesting, particularly as it gave first-hand information on solutions and their results for installations which had been operating for a number of years. The authors' experience with first-hand knowledge of laboratory tests on a broad range of couplings working on different principles was most valuable. It was obvious that the authors did not have the time or space in this paper to deal with each type of coupling completely, as the subject was too vast. Referring to the powder coupling, many new theories were involved which he found most interesting, and in view of this he wished to enlarge upon certain aspects; the authors' views on these points would be appreciated.

The transmission medium between the outer driving casing and the driven rotor was spherical steel shot, 0·024in. in diameter. When the shot was distributed around the casing each ball had single-point contact with the adjoining ball and therefore the mass lacked elasticity. Should a torsional vibration be transmitted to the driving housing, it would be absorbed in the mass of shot and was considerably reduced on the rotor. For example, to transmit 1,500 b.h.p. at 500 r.p.m., 200lb. of shot were required, and it would be appreciated that, with such a large mass, the effective damping was achieved with very slight slippage.

The authors referred on page 387 to peripheral packing of the shot. Whilst this might be so he felt it was of sufficient importance to mention that there were findings indicating that the shot did not revolve as a solid mass; it separated out into layers, the number of which depended on the diameter of the shot and its depth between the casing and the internal radius of the rotor. Each layer revolved at different speeds, and due to this the shot therefore became viscous and could be thought of as a fluid.

On page 387 it was stated that after half an hour's running with 0·6 per cent slip and a reduction in amplitude exceeding 3-1 the rise in temperature of the coupling flow charge was from 72 to 95 deg. F. (22 to 35 deg. C.). Tests and practical experience indicated that when the coupling commenced to slip and operated as a damper there was a steady rise in temperature. However, the temperature gradient levelled off and became constant with no further rise. The heat was dissipated through the coupling fins and these were cast radially around the aluminium driving housing. Under normal conditions, the damping characteristics were not altered by heat and a temperature rise of up to 392 deg. F. (200 deg. C.) could be tolerated without difficulty. The shot was retained in the housing by a rubber seal which was located in the bore of the coupling and was not in direct contact with the rotor, and this seal could operate satisfactorily at 392 deg. F. (200 deg. C.).

Calculations regarding the transfer of heat from the steel

* Zdanowich, R. W. Contribution to discussion on "Some Factors Influencing the Life of Marine Crankshafts". Trans.I.Mar.E., Vol. 76, p. 110, p. 112.

shot to the casing could not be based on the normal heat conductivity standards of steel, as the balls of shot were only in point contact with each other, there being small pockets of air, and, in addition, the shot was coated with an insulating barrier of graphite which was used for lubrication purposes. Bearing these points in mind, it could be established that the coupling would operate for indefinite periods as a damper at critical speeds with no time restriction imposed.

On page 387, Fig. 10, an interesting curve was detailed showing the torque-speed characteristics with 0·5 per cent slip and the torque varied as the shaft speed squared. This curve could be considered as the normal torque curve. The characteristics of a powder coupling were set by adjusting the weight of steel shot, and on installations where the propeller torque was proportional to the square of the r.p.m., full advantage of the powder coupling could be made as the unit followed the propeller torque requirement throughout the operating range. The coupling was normally filled with shot to allow operation under normal conditions with one revolution slip in 10,000. Should there be torsional vibrations in the engine speed range, the coupling would slip and reduce amplitudes by as much as 4-1 with very little slip at any speed. In the event of a sudden overload, such as the propeller striking a submerged object, the coupling would break away and slip 100 per cent. The minimum torque for break-away on overload was 1·2 times the normal torque. When stalled, the coupling housing, which was connected to the engine, continued to rotate, and the rotor, which was connected to the propeller shaft system, became stationary. Under this condition the corrugated rotor cut a groove through the shot and was only touching the shot at the extremities and not on the inclined faces. Therefore, as the area of contact of the transmission medium and the rotor was reduced, the torque was correspondingly reduced, and tests showed that this was approximately 80 per cent of normal torque. It therefore followed that when assessing characteristics of this type of coupling it was important to bear in mind three torque curves.

The first torque curve was the normal curve, as shown in the paper; the second was the break-away curve on overload, which was at least 1·2 times normal; the third curve was the stalled, which was approximately 80 per cent of the normal torque.

He wished to comment next on what the authors had stated in their penultimate paragraph on page 387. The torque value on test was reduced progressively until a point of no-slip occurred. The torque was 2,830 lb. in. and the total swing 1·35 deg. Under these conditions the powder coupling was only transmitting 60 per cent of the engine torque and, in similar tests on an omnibus application, it was found that there was a 4·02 per cent saving in fuel compared with conventional types of fluid coupling.

The authors stated in the final paragraph, dealing with the powder coupling, that it could be used to good advantage for certain marine applications, and he wished to contribute to this statement by saying that where manœuvrability was of primary importance the coupling enabled from full ahead to full astern, with a direct reversing engine, to be achieved in less time than expected with a direct shaft drive. The break-away characteristics at 1·2 times normal torque could be extremely useful on geared installations.

Referring to Lloyd's Register of Shipping latest guidance notes, it was recommended that, for critical speeds near the maximum speed, the vibratory torque should not in general exceed one-third of the transmission torque and, under such conditions, the coupling would slip, giving effective damping.

MR. J. WRIGHT, referring to the introduction, page 384, said that mention was made of the desirability of having available full information on flexible couplings. Work was well advanced concerning these characteristics on the double-tyre type of coupling mentioned in Part II, Example 1, Fig. 24(b). This was borne out by the close relation between calculated and measured frequency and the accurate prediction of coupling dynamic magnifier, shown by Example 1.

With regard to coupling damping properties, mentioned in Part I, on page 386, the double-tyre type relative damping had been shown to be mainly dependent upon pre-load (steady transmitted torque) and was practically independent of vibratory torque and frequency.

With this type of coupling it had been shown that the ratio of dynamic stiffness over static stiffness depended both upon coupling damping and upon frequency, and increased, with an increase in damping and in frequency, up to a maximum of 2·2 at a maximum frequency of 3,000 v.p.m. (this was the maximum frequency at which the trials were carried out).

When the double-tyre type of coupling was compared with a rubber-sandwich type of coupling, the double-tyre type had three times the relative damping at low pre-load and 1·8 times at high pre-load. In this respect it was important, in marine applications, to obtain efficient damping at low pre-load torque because it was under these conditions, i.e. at low speed, that most critical speeds occurred.

With regard to the dry-fluid centrifugal coupling mentioned on pages 386 and 387, with this design he had a feeling that mal-alignment, sealing and fusion of the shot could be severe problems, but in the general terms of centrifugal clutch coupling it should be pointed out that a coupling of this type, shown on page 384 of the paper, had been in use in marine propulsion applications for many years, and this clutch coupling had built into it inherent mal-alignment flexibility. This clutch coupling had been used for many years to damp torsional vibration, to provide controlled overload protection and to cater for mal-alignment. These units had been installed in most cases in small to medium size geared commercial type vessels, and many hundreds of these units in cardan shaft form had been employed in naval vessels.

With regard to Example 2 in Part II of the paper, this referred to an installation of an Admiralty supply ship, five-cylinder engine, 5,500 b.h.p. at 135 r.p.m. The double-tyre type of coupling, mentioned in Example 1, was now available in tandem form to cater for a transmitted torque of $2{\cdot}56 \times 10^6$ lb. in. A coupling to suit this transmitted torque would have an overall diameter of about 3ft. 8in. and an overall length of about 4ft.

Tests on dynamic torsional stiffness and damping characteristics had not yet been finalized, but, for a frequency of around 150 v.p.m., these values would be:

Dynamic torsional stiffness — 20×10^6 lb.in./rad., approximately

Dynamic magnifier, M — 6, approximately

Applying these values to Example 2 gave:

One-node frequency — 134 v.p.m. approximate. The 5th order resonance would thus occur at about 27 r.p.m.

Based on the empirical relationship for combining dynamic magnifiers given on page 393 of the paper

$M_{\text{coupling}} = 6$

$M_{\text{engine + prop.}} = 10$

$M_{\text{total}} = 5{\cdot}1$

(M_{total} from Example 2 $= 7{\cdot}0$)

Intermediate shaft measured stress was $\pm$ 6,100 lb./sq. in.

Multiplying by a simple ratio of the magnifiers of $\frac{5{\cdot}1}{7{\cdot}0}$ the stress of $\pm$ 6,100 would be reduced to $\pm$ 4,450 lb./sq. in. This is equivalent to about 95 per cent of the full load torque. The substantial reduction in the critical speed from 45 r.p.m. to 27 r.p.m. would provide a welcome increase in operating speed range and the reduction in stresses also constituted a worth while improvement.

Coming to Example 3, a 525ft. ore carrier powered by a two-stroke, six-cylinder engine rated at 7,500 b.h.p. at 115 r.p.m., there was now a design of highly flexible coupling employing inflatable rubber air-bellows available for such high capacity drives. For the drive in question a dynamic torsional stiffness of nominally 50×10^6 lb. in./rad. could be provided by such a coupling; the feature of torsional stiffness being variable over a wide range, according to the air pressure employed, was useful in enabling various values to be tried in practice. Also, this design of coupling could accommodate very great misalignment, in any or all directions, while imposing minimum reaction forces on adjacent bearings. This last mentioned point was not given enough emphasis when considering various types of flexible coupling, not only for their torsional vibration properties, but also for misalignment characteristics. It was not only a matter of how much misalignment the coupling would cater for but how much loading it imposed while catering for this misalignment.

In conclusion, he congratulated the authors on their paper, which was both informative and interesting, and would do a great deal to promote the acceptance of flexible couplings for marine propulsion drives, especially the larger powered direct drive types of application, where the use of flexible coupling was particularly important with regard to aft mounted engines.

Mr. E. J. Nestorides said that a point, amply demonstrated by the paper, was that the choice of a coupling for a particular installation was definitely a subject for specialists, and there would be a considerable possibility of risk in continuing the practice of the "good old days", when a non-specialized installation manager could say "Just put in some soft coupling—this should settle our problem", without allowing for a detailed study of the behaviour of the complete installation, with and without a coupling. Whether the specialist was in the users' works or at the coupling makers, or elsewhere, the main thing was to make full use of his experience.

There were three points in this extremely useful paper on which he wished to comment. Firstly, what were the claims in favour of variable stiffness couplings? In the past, a detuning action was considered to be the cause of reduced amplitudes with such couplings. As distinct from transient running through a critical speed, "detuning" and "retuning" during a vibration cycle did not produce amplitude reduction —not with the analogue model described in the paper and not in theory. All the amplitude reducing couplings which had so far come to his knowledge operated either by permanently shifting away the resonance or by damping. It would be interesting to have data on any such design for reducing amplitudes without damping.

The second point concerned gears. He felt that "gear hammer" was a somewhat subjective term. In a borderline case for vibratory torque, if the gear cover were provided with an anti-drumming compound, would this reduce hammering to acceptable levels? The overall stiffness of the gears, the gear-shafting and its bearing supports could come into consideration in some cases. Possibly accelerometer measurements could give additional information.

Thirdly, some very interesting applications of "two-rate" couplings had been seen. These were also variable-stiffness couplings. As evidenced by Fig. 28 in the authors' paper, the low-stiffness value in itself would satisfy all the requirements with regard to the positions of the critical speeds. But a higher spring rate was necessary for full-load torque, to avoid overstressing the coupling. This feature was also present in floating-spring-grid couplings and C-link couplings. It seemed, therefore, that the main practical purpose of variable stiffness was to provide increased load-carrying ability. This was successfully achieved in specialized designs. The damping available was used to provide amplitude reduction, as in constant-stiffness couplings. Comments on this tentative assessment would be welcomed.

Correspondence

Mr. A. J. Young (Member) wrote to thank the authors for presenting this paper at a time when many were interested in the problem of using solid rubber block couplings in main and auxiliary drives.

With reference to Fig. 25(c) and the rubber block coupling shown, the stiffness, according to the accompanying graph of deflexion against torque, appeared to be distinctly non-linear.

Mr. Young asked whether, in a simple branched system with such a coupling in the auxiliary drive and also a smaller similar coupling in the same drive at higher r.p.m., the authors would be surprised if, on test, the lowest estimated critical of the branch appeared to be non-existent, apparently due to detuning.

The B.I.C.E.R.A. Handbook on Torsional Vibration (p. 111) seemed to suggest that this was possible. In any case if, at about 360 v.p.m., the damping bore any resemblance to that shown in Fig. 6, the combined effect of damping and detuning would appear to be quite powerful.

Would the authors say what they considered the relative effects of damping and detuning on the reduced measured amplitudes to be, as compared with the calculated amplitudes as shown in Figs. 25(c) and 27(b).

Mr. J. C. Crum wrote that probably the greatest demand on anyone who became associated with torsional vibration problems was made on their reserves of faith. These reserves were always strengthened by test results, and in this way the present paper was very generous.

It seemed probable that the problems of detuning torsional vibrations and providing suitable damping would always be with us. The use of a freewheel would be most attractive, in that any vibration energy extracted would be turned into useful work, a destination without any obvious limits of capacity. Probably the nearest approximation to this which was currently available, was the centrifugal clutch coupling, particularly the type fitted with leading shoes.

The use of a really soft coupling which virtually isolated the engine from the rest of the system was obviously very attractive. However, when using such a coupling on auxiliary engines, the unexpected difficulty of governor surge had been met, caused because the I-mode frequency came close to the "engine instability frequency". The manufacturer of the governor stated that frequencies less than 1,200 v.p.m. should generally be avoided. This limit must be in some way connected with the natural frequency of the governor, but the relationship was not immediately obvious, being also dependent on the number of cylinders and other factors.

One of the laws which became clear from the first part of the paper was that, when using a double-rate coupling, it was essential to arrange that the change from one rate to the other occurred at a speed where there were no significant vibrations. Fig. 31 showed however, that in the last example considered in the paper, this change was made close to the I-mode 2nd-order vibration, especially if the tug was towing a light load. As pointed out by the authors, this vibration was quite small, and the presence of telescopic dampers in the coupling was doubtless of assistance, but the law had been transgressed. One was left wondering how often it was possible to keep it implicitly.

Mr. T. W. Spaetgens wrote that, in this paper, appropriate emphasis was placed on the importance of accurate coupling stiffness data, and the usefulness of laboratory testing in acquiring this was convincingly demonstrated. The breakaway phenomenon of a non-linear coupling was very effectively illustrated.

The authors' studied opinion that the "flexible" solution was generally preferable was a matter of delight to this correspondent, long an advocate of low tuning in variable speed systems. This persuasion in fact led to the development of a helical-spring coupling of high torsional flexibility and, in relevance to the subject of this paper, wherein various types of flexible couplings were illustrated, a brief description of this coupling would follow this discussion.

The authors listed six disadvantages of the "stiff" solution. Would they consider that a further disadvantage was the additional design study required (along with the attendant possibility of design changes) to examine into the "crankshaft" mode which—by virtue of the stiff shafting—had been altered considerably from the "engine alone" or "stock engine" condition? The "soft" or "flexible" solution greatly enhanced the inherent adaptability of the engine to the propulsion system, "immunizing" it from its environment. The authors made some reference to this aspect in the section "Example of a Typical Geared Marine Engine".

Mr. Spaetgens would value the authors' comments relative to their experience with low-tuned systems where irregular-, unbalanced- or mis-firing conditions in the engine cylinders introduced low-order criticals, particularly in the case of two-cycle engines. He had encountered a few problems along this line and had now adopted the policy of pointing out the location of any such resonant zones and the added desirability of keeping fuel injection systems properly serviced. In the authors' Example 2, Fig. 25(c), for instance, a 2nd order critical speed would exist at ($6 \times 45/2 =$) 135 r.p.m., exactly the service r.p.m., due to possible harmonic unbalance developing at some time.

Fig. 25(c) indicated a rather large (15 per cent) discrepancy between calculated and measured I-mode frequency. While this was possibly typical of rubber couplings with their inherent variations and dynamic factors, was the influence of propeller WR^2 error ruled out by virtue of inertia testing?

Turning to geared drives, would the authors not consider it worthwhile to have mentioned some other damaging effects of high vibratory torque across gearcase meshes (aside from excessive tooth loading) such as drastic reduction on the life of bearings, loosening of hub fits, failure of splines and shafts, and the shaking moment on the gearcase? Many case histories existed which showed bearing failure repeatedly occurred in a few hundred hours and where the fastenings attaching gearcases to their beds or to the engines, had fatigued before any damage occurred to the gear teeth. Most of these problems occurred at criticals where vibratory torque was not sufficient to cause teeth separation and to provide the attendant audible warning.

Calculations and subsequent tests had shown that, where flexible couplings were disposed between the engine and gearcase, for purposes of low tuning, a tuning flywheel mounted on the gear side of the flexible coupling (or on the pinion shaft) would not only assist in the overall tuning of both I and II modes, thereby making coupling selection easier, but would reduce the resonant vibratory torques applied to the gear mesh.

In Fig. 29(d), the I/6 and II/6 criticals were shown independently. An interesting aspect brought out by the proximity of these two criticals in geared drives was the interaction of the torque effects at the gear mesh. In between the two criticals, the gear mesh torques from each critical were usually additive, while above the II/6 critical, these torques subtracted, giving actually a total torque curve something like that shown in Fig. 38.

This nearly vertical portion of the torque curve above II/6 (the proximity of the criticals governs the vertical effect) had an interesting and practical application. With a linear coupling, the teeth separation point "a" under towing conditions would be practically identical to the teeth separation point "b" under free running conditions. Thus the lower end of the usable speed range could be fixed at the same r.p.m. value (some distance above the separation point) for both towing and free running conditions. Many tests on flexibly-connected engine/gear/propeller drives, by the writer, confirmed that this separation point was instantaneous (denoting a vertical torque curve) and that its calculated location could be determined to within one or two per cent of the measured location.

In cases where the major II-mode (gear/propeller) critical could not be located down below the minimum speed due to limitations on the minimum size of shafting, Mr. Spaetgens felt that it was better to shift it further up into the speed

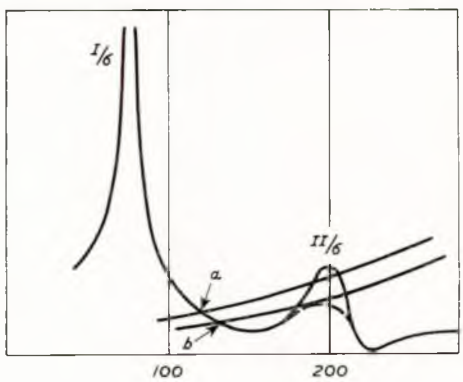


FIG. 39

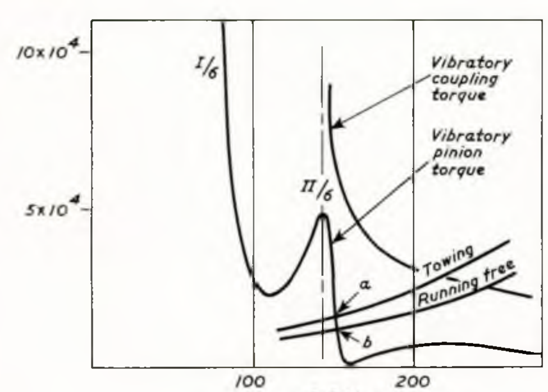


FIG. 38

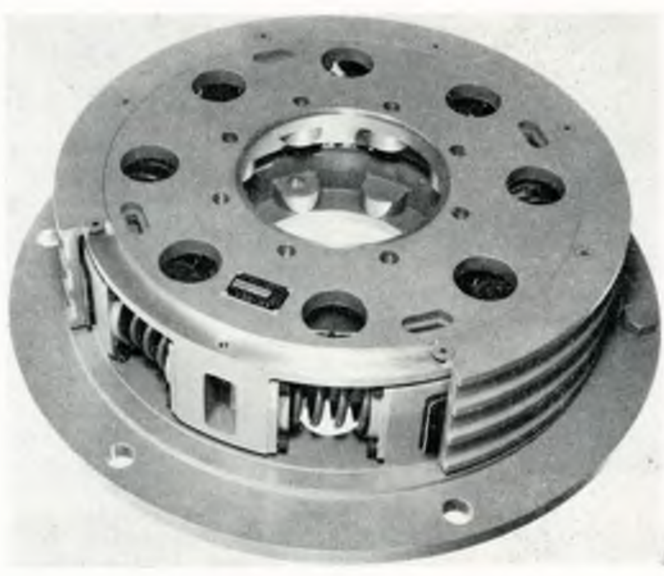


FIG. 40—*Linear flexible coupling*

range by using slightly stiffer shafting. A condition illustrated by Fig. 39 then applied, and generally tooth separation could be avoided at the II-mode critical by higher ratios of gear/propeller inertia. Generally speaking, with flexibly connected gearing, the higher the II (gear/propeller) mode, the less sensitive the II mode was to engine excitation. The foregoing discussion applied, of course, to cases where the engine/gear frequency was below the gear/propeller frequency.

A linear flexible coupling which utilized helical coil springs and had a low, constant stiffness was shown in Fig. 40. Rubber or composition limit stops prevented springs going solid during critical traversals thus minimizing noise and over-stress. Friction dampers with adjustable tensioning provided adequate damping to reduce intensity of engine/gear or engine/propeller criticals without affecting the coupling stiffness. The constant and accurately-controlled torsional stiffness (typical error three per cent) and inherent damping facilitated simple and accurate torsional calculations of the system.

Mr. T. Hindmarsh (Member), in a written contribution, felt that the paper dealt very admirably with torsional stiffness of flexible couplings, but did not appear to deal with the axial stiffness of these couplings and it was suggested that any information the authors could include, in their replies to the discussion, on this matter, would be a useful addition to the information in the paper. This suggestion was made because it was thought that the axial stiffness of the coupling in some cases was very important, for two reasons:

1) the possibility of axial vibration in the engine/gearbox system causing gear pitting;
2) the difficulties experienced with some engine room arrangements for providing location of the engine crankshaft and the gearbox input shaft without imposing undue loads on the locating bearing. For example, some engines had only a very small axial float in the crankshaft and this must not be exceeded. Those engines, on occasions, were coupled to a gearbox having a generator interposed between the engine and the gearbox. It might also be necessary to have a very limited axial float on the gearbox input shaft and, therefore, due to expansion in the shafting between the engine locating bearing and the gearbox locating bearing, some provision must be made in the flexible coupling to absorb this expansion without undue thrust on the engine and gearbox locating bearing.

Mr. F. E. B. Webb wrote that on page 384, in the paragraph referring to Fig. 2, the authors said that there was a need for a complete list of flexible couplings. A survey entitled "Flexible Couplings" by the editorial staff of *Engineers' Digest*, was published in 1957 and was believed to be still available. All the types shown in Fig. 2 (except the centrifugal clutch coupling) and many others were fully described.

On the subject of couplings incorporating rubber elements, it was widely believed that it was difficult to obtain repeatable results from rubber mixes of the same specification made at different times, so that a number of couplings, nominally similar, might have very different torsional characteristics from one another. Another property of rubber products, which made many designers wary of them, was the way their characteristics varied with age. Clearly a coupling, the torsional stiffness of which changed gradually throughout its working life, could be dangerous.

These two disadvantages were not present in the "all metal" coupling, but the types which were now commercially available were usually too stiff torsionally. It had been found that a suitable torsional stiffness resulted in a deflexion of about 4 deg. under full-load torque. The type of coupling, incorporating a number of helical compression springs arranged circumferentially, had been employed satisfactorily in a number of cases. The stiffness of this type of coupling was easily calculated and remained constant. If necessary the torsional stiffness of the coupling could be varied, within limits, by changing the rate of the springs.

The authors' views on fluid couplings would, most probably, have been interesting. These were often claimed to isolate driven machinery completely from the torsional vibration of the prime mover. However, it would seem that any coupling, which could transmit power, must also transmit vibration to some extent.

Mr. E. W. M. Britain wrote that he had found the paper a most interesting and valuable contribution to the understanding of the characteristics of flexible couplings, a field too often ignored by investigators, in his opinion.

While he did not feel competent to comment on many of the theoretical and experimental findings, he was particularly interested in the "half-moon" coupling and wondered whether the authors could add some details of service experience with this coupling. It seemed to him to be particularly useful where there was some doubt as to the reliability to be expected from a rubber-in-shear coupling, because of high temperature for example. The ability to modify considerably its natural characteristics by change of dimension of the "half-moon" links would seem to be particularly valuable.

The duty to be performed by these links, however, did seem to be rather severe, and some special techniques of manufacture and some design precautions might have been taken to prevent premature failure. Although the bending fatigue experienced by these links was really no different to that undergone by the spring member of a Bibby coupling, for a given size of device, they were larger and heavier, and more complex in shape, and he would be very glad to hear of any particular requirements of the design of such members, for example shot peening or other surface treatment, that might be considered necessary.

The fact that it could be made very short in overall length was particularly useful in cramped spaces.

Finally, Mr. Britain asked the authors whether they considered that it would be suitable to interpose between the engine and generator of a Diesel generator set, particularly a multi-crank Diesel generator set, where torsional vibration characteristics might demand a very large reduction of torsional stiffness in the shafting. In such a case, the coupling would be buried inside the carcase of the direct-coupled generator, or within the engine flywheel, and complete reliability with no maintenance at all, together with unchanging characteristics over the whole life between overhauls would be essential. If these conditions could be met, such a device might possibly be used in preference to a silicone fluid damper, which was not completely reliable in his experience.

Mr. E. C. Garratt (Member) wrote that the authors' comparison of advantages and disadvantages of using a form of flexible coupling or of increasing shafting diameters to overcome a torsional vibration problem was particularly interesting.

It was also depressing as it showed the limitations of both methods when one was confronted with this type of problem, which unfortunately was bound to occur more frequently with present trends of design, with such a high proportion of machinery being placed aft and the power required also increasing.

The practical disadvantages of excessively large diameter shafting were obvious.

It was not so clear why there seemed to be a power limitation on flexible couplings which one inferred from the fact that 7,500 b.h.p. was the largest power application of which the authors had knowledge.

Surely there were far larger shore installations embodying flexible couplings and was there any real difference between a shore and a marine installation in this respect?

Mr. Garratt sometimes felt that the engine designers could do more towards reducing the criticals, and that consideration of the installation as a whole was so often left too late.

One was confronted with a problem which could only be dealt with by a flexible coupling or increasing shafting diameters, the main engine being already under construction, and any modifications to it apparently impossible.

Mr. G. Broersma wrote that he came from a family of harbour and canal builders and, in this respect, was conversant with coupling difficulties in the drive of suction pumps of suction dredgers. Fig. 41 showed what a suction pump might bring up in 1921 and why it stopped when these stones finally arrived in the pump inlet.

Fig. 41—*Stones sucked into suction pump of suction dredger*

Initially, the writer's father used steam power, which was inherently flexible, but which became too bulky in dredgers, with increasing size of the suction pumps.

After a short period of using Diesel power, he was probably the first to introduce electric power in a suction dredger which was used for the first time in digging the Laak harbour in The Hague (1921-1922).

Fig. 42 showed this installation. On the left hand 200 h.p. electric motor, one might have read the notice: "Hoogspanning. Voorzichtig. (High voltage. Careful) for 2,000 volts. On the right was a 100 h.p. electric motor for driving another installation. Electric current was supplied from the quayside by a small power station built for the occasion.

In the larger pumping installation, which was shown in full view, a simple coupling could be noted with a single intermediate disc bolted alternately to the flanges of electric motor and pump shaft flanges, in order to obtain flexibility. In cases where the suction pump stopped suddenly, through obstructions entering it, either the coupling bolts failed or the peak load on the electric motor caused one or more blown fuses.

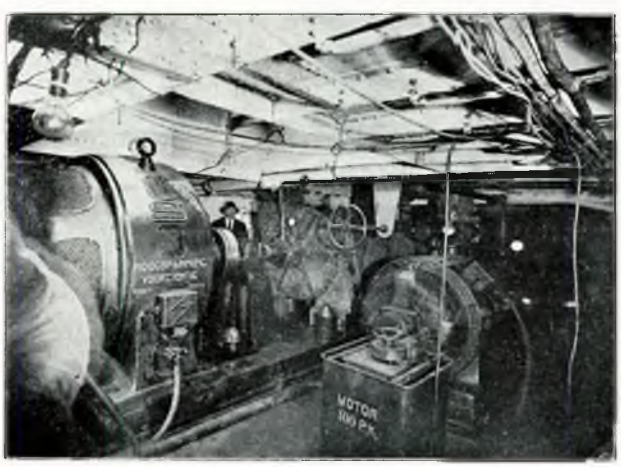


Fig. 42—*Electric drive of suction pumps in suction dredger (1921)*

Did the two authors of the present paper ever measure extreme shock loads in their tests? In case of sudden stopping, what happened then to the bonding between metal and rubber, and/or to the rubber itself?

Some years ago, the writer managed to develop successfully a middle vacuum gear pump for chemically aggressive fluids. The gears and its housing were manufactured to a high degree of accuracy and finish, with minimum values of clearances. Standards were quite exceptional for the stainless steels used in all parts.

However, failures occurred in the laboratory stage for some time, before the right type of design was developed with the right clearances, and, also, before a manufacturer was found who could and would produce a pump of this type.

Between drive and pump, a simple rubber coupling was introduced, which never failed—neither did the electric motor which drove the pump. In the coupling, a single rubber disc was introduced between the flanges of drive and gear pump, and bolted alternately to these flanges.

In chemical engineering, the life of a drive was only a few years, in view of the economic life of most chemical plants. No difficulties were to be expected in rubber couplings, therefore, even in the chemically aggressive atmosphere in which they had to work.

However, what did the two authors consider as the life expectancy of a coupling in a marine main propulsion system?

Finally, the authors produced damping characteristics of a rubber disc coupling in their Fig. 6—damping coefficient per unit of volume as a function of vibration frequency.

In his doctor's thesis, Kosten* had shown that these characteristics varied with the area of bonding between rubber and metal. That was to say, if the bonding started to fail, resonance speeds were shifted.

Did the authors have any evidence from their tests, where this happened inadvertently?

Mr. J. Jones wrote that the paper highlighted the increasingly important role played by flexible couplings in marine Diesel transmission and the need to accurately predict their performance before installation.

The curves in Figs. 21 and 22, for the laboratory rig nonlinear stiffness couplings were most interesting, but it was noted that there was no marked reduction in peak amplitude from that obtained with the linear coupling (see Fig. 19); could this be because the coupling had two linear stiffness rates instead of the more usual characteristic of a constantly increasing stiffness?

It was also interesting to note that the test rig coupling torque deflexion characteristic was similar to the two-rate flexible coupling analysed in Figs. 30 and 31. This proprietary design of coupling had been proved in service to be entirely satisfactory when correctly applied and, therefore, perhaps the rather ominous looking deflexion characteristics shown in Figs. 21 and 22 were not realized on a full size actual installation.

With reference to direct drive installations, his company had often employed flexible couplings to deal with offending critical speeds, but noted that sometimes it had been found necessary to increase the ratio of engine flywheel WK^2/ propeller WK^2 in order to reduce amplitude of the engine in way of critical speed flanks near idling speed.

Regarding the reference to gear backlash, his company had had recent experiences where this had varied considerably between gears of identical size, type and manufacture; did the authors consider that values of maximum permissible backlash

* Kosten, C. W. 1942. "Over de elastische eigenschappen van gevulcaniscerde rubber (On the elastic properties of vulcanized rubber)". Thesis, University of Technology, Delft. Waltman (A. J. Mulder), Delft, 1942.

could be compiled, varying with, say, diametrical pitch, and issued in the form of a guidance note.

The note on page 396, regarding maximum vibration torque on gears at maximum speed, was considered a most important recommendation and should have proved always possible to achieve by suitable size of engine flywheel and/or making the flexible coupling torsionally "soft" enough.

It would be interesting to have the authors' opinion on whether the figure of one-third full transmission torque could be incorporated in the rules and recommendations for gear design; recent enquiries regarding a proposed land installed geared pump drive revealed that the gearbox quoted could withstand a vibration torque loading at maximum speed of 20 per cent full transmission torque.

With reference to the comments that the steel spring coupling was normally torsionally much stiffer than the rubber type, the "Geislinger" coupling was torsionally very flexible and had been satisfactorily applied to a number of geared drives. This coupling was a leaf spring spoked coupling which had the additional feature of oil dashpot damping.

Mr. A. C. Grant wrote that his company had produced a compact high torque flexible coupling for marine and heavy engineering power transmission shafting with rubber or similar used for the cushioning medium (see Fig. 43).

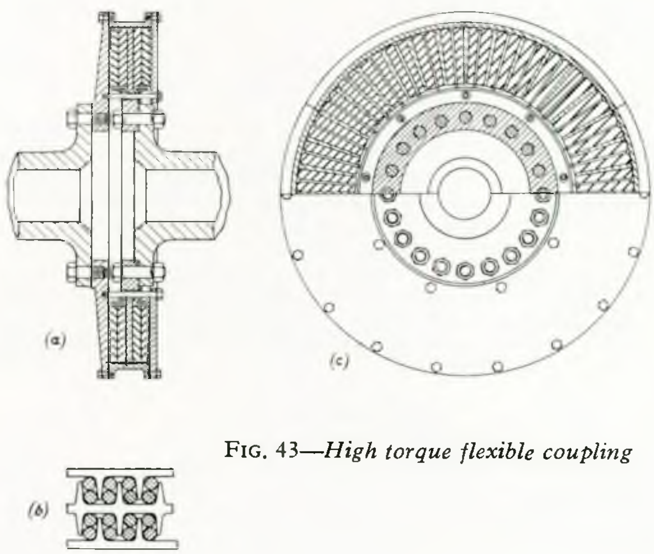


Fig. 43—*High torque flexible coupling*

In the past, high powered shaft couplings, utilizing rubber, had been made with:

1) the rubber in shear for torsion, usually also necessitating bonding of the rubber to the associated metal parts;
2) the rubber in compression with the rubber elements disposed on as large a mean radius as possible to achieve flexibility with torque-carrying capacity;
3) a combination of rubber in shear and rubber in compression, in some cases used in a two-rate set-up.

When high torques were involved associated with marine shafting, the dimensions of the types of coupling just mentioned became excessive. This was mainly due to the fact that in all rubber-compression couplings, the ratio of rubber volume to actual coupling volume was very low and in the case of the bonded rubber in shear variety, the shearing force for unit area had to be kept low.

To overcome these difficulties the company produced a coupling, using radial rubber cylinders associated with radial meshing vanes. By this, they achieved a coupling with a rubber volume/total volume ratio much higher than before. The rubbers were always substantially in compression, therefore high loading was possible. This coupling was approximately four or five times shorter than other types of the same diameter.

With radial rubber pieces they had been able to arrange for inspection or renewal of these pieces without uncoupling the main flanges on to the coupling. This was done by using a segmented outer cover.

By using circular-sectional radial cylinders of rubber, which tended to compress into square sectioned cylinders when under load, they were able to give the coupling a stiffness characteristic which increased slowly at first and progressively much faster as the load was increased. The coupling therefore could act as a low frequency damper at relatively low loads, a desirable feature in marine shafting especially when using direct Diesel drive.

By choosing a suitable stiffness, or grade of rubber element and a voidage space into which the rubber could be partially or totally deformed, then it was possible to obtain a wide range of stiffness or damping characteristics.

A further advantage of the new coupling compared with other varieties was that it had stiffness in every possible direction, including the axial direction. It was found that this coupling was standing up very well to service conditions, and there was no reason why this coupling should not be adopted for low powered applications.

Authors' Reply

The authors gave their thanks to those who had contributed to the discussion of the paper, either verbally or in written correspondence. The result of the various contributions had been considerably to widen the scope of the paper, and many interesting points had been brought out in the discussion.

Mr. Cranston had made informative remarks on the application of electro-magnetic and hydro-dynamic couplings in multi-engined marine installations. Such couplings were claimed to isolate the driven from the driving systems, in so far as torsional vibration was concerned, but at a cost of permanent slip in service.

With regard to the firing orders: at one time the choice was predominantly influenced by torsional vibration and balancing problems but, as pointed out by Mr. Cranston, other considerations such as pulse type turbocharging now took priority. Fortunately torsional vibration technique was keeping pace with the developments and corrective measures could generally be taken with confidence.

Regarding Mr. Cranston's remarks on the cyclic torque variation, the authors believed that what was, in some cases, passed off as cyclic irregularity was in effect the interaction of the adjoining modes of vibration described in the written contribution by Mr. Spaetgens.

The point which Mr. Cranston brought out with regard to the aging of rubber was a valid one, and it would be useful to obtain more data on the incidence of replacement of elements. It was the authors' experience that couplings operating under design conditions did not show deterioration.

The authors supported Mr. Zdanowich's opinion that the Bibby coupling functioned principally as a transmission-vibration damper and shock absorber, and not as a system detuner. The efficiency of the coupling, therefore, in controlling torsional vibration criticals would depend on the modal twist across the coupling, and, in order to achieve a relatively high degree of twist, the coupling must be comparatively flexible. It followed also that the coupling would be effective in controlling a mode of vibration whose frequency was significantly affected by the introduction of the coupling.

Mr. Zdanowich pointed out that indiscriminate fitting of flexible couplings was bound to lead to serious trouble, sooner or later. The authors fully endorsed this view and advocated theoretical investigation of all service troubles, as far as circumstances permitted, even if a practical solution was found meanwhile, with a view to obviating future difficulties by fuller understanding of the problem.

To answer Mr. Zdanowich's point regarding the question of damping in the study of non-linearity effects of a coupling

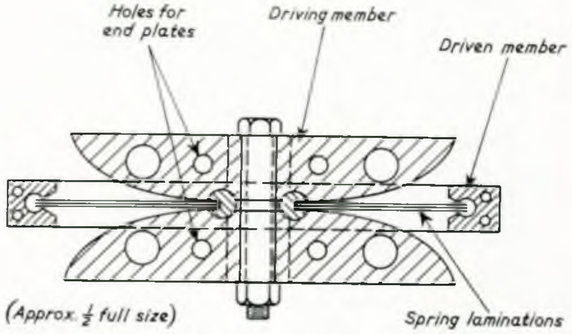


FIG. 44—*Variable stiffness coupling*

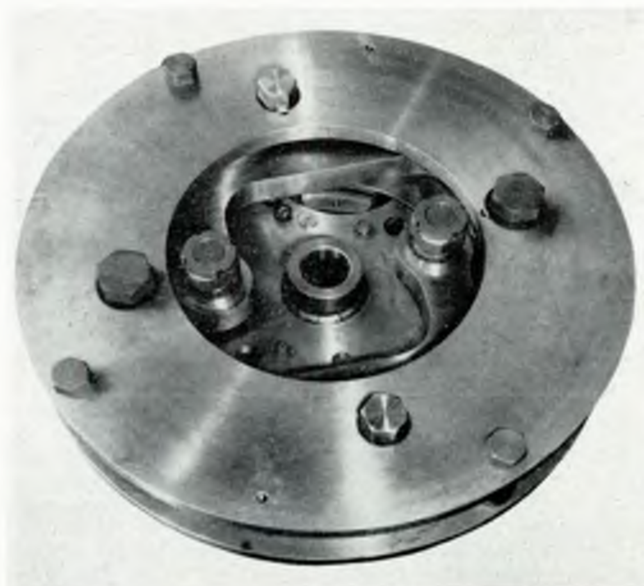


FIG. 45a—"*Half-moon*" *coupling*

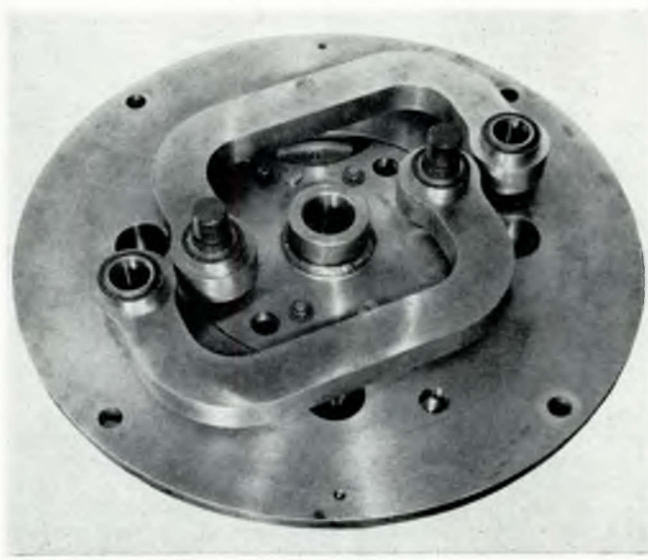


FIG. 45b—"*Half-moon*" *coupling with top cover removed*

using a mechanical analogue, it should be emphasized that damping influences were deliberately reduced to the minimum possible amounts. During the early stages of the development of the rig a multi-blade spring was used of the type illustrated in Fig. 44. Although the static stiffness characteristic of the coupling was similar to that of the single-bladed coupling described in the paper, the damping was so large in relation to the excitation torque that only small amplitudes of swing were obtained, about one-tenth of those shown in Figs. 21-22. These amplitudes were so small compared with the static deflexion characteristic of the coupling that the coupling behaved, at each preloaded condition, as a linear coupling having a static stiffness characteristic according to the preload applied. This coupling operated effectively as a damper, and it was likely that the Bibby coupling operated in a similar manner, as suggested by Mr. Zdanowich.

Mr. Chapman had mentioned a $2\frac{1}{2}$ degree shift in the small "half-moon" coupling tested at B.I.C.E.R.I. laboratory. A similar shift was later observed on the full size coupling fitted in the tug. This was probably due to the very low stiffness of the coupling at this small deflexion, such that even the very small frictional torque of the gearbox, with the clutches disengaged, was sufficient to cause the shift.

Mr. Chapman had asked for details of the tests carried out on the tug: briefly, the installation comprised a six-cylinder four-stroke engine developing 785 b.h.p. at 600 r.p.m. and driving a propeller via a 3·5:1 reverse-reduction gearing. The "half-moon" coupling, shown in Fig. 45 (a) and (b) was interposed between the engine and gearbox.

The purpose of the investigation was to evaluate the performance of the coupling and to measure stresses in the "C" springs or links under service conditions. With this in view strain gauges were affixed to the gear input shaft and to the "elbow" of the coupling link. The shaft gauges were orientated at 45 degrees to the centre line so as to be sensitive to torsional strain only, and those on the link were in line so as to measure bending stresses. Signals from the gauges were conveyed to the amplifier and recorder via slip rings attached to the pinion shaft hub. Due to limitation of space, only one set of gauges could be connected at one time, and for this reason the tests were repeated for all conditions of loading in order to measure both sets of stresses. Timing marks at half-second intervals and revolution marks once every two engine revolutions were recorded simultaneously with the strains.

The results showed that no definite torsional vibration resonance occurred in the whole running range. There was some 3rd order vibration at 160-170 engine r.p.m., which was below the idling speed of about 200 r.p.m., and a trace of 3rd order was also discernible at about 500-600 r.p.m. The maximum alternating torsional vibration stresses in the working speed range were less than $\pm$ 500 lb./sq.in. in the pinion input shaft, and there was no gear hammer present in the working speed range.

Stresses in the "elbow" of the link varied between about 6,000 lb./sq.in. tensile and 7,000 lb./sq.in. compressive when the links were in contact with the stops. The engine r.p.m. at which zero stress and, therefore, zero coupling stiffness occurred depended on the transmitted torque which in turn depended on such factors as the towing conditions and the speed of the ship. It was found almost impossible, therefore, to repeat tests for any particular condition of load and speed, but a total stress range of about 14,000 lb./sq.in. might be regarded as typical.

Mr. Hunt's description of the operation of the powder coupling was interesting, particularly the suggestion that the shot did not revolve as a solid, but rather as separate layers revolving at different speeds. Regarding the temperature of the shot, under normal conditions of operation when slip was limited to not more than 0·5 per cent, the average temperature in the shot never rose above 95 deg. F. (35 deg. C.) during the tests carried out at B.I.C.E.R.I. The authors, however, took the conservative view that it would be prudent gradually to uprate the coupling on to more onerous duties until adequately backed by service experience, and they looked forward to more applications in marine service.

Mr. Hunt had mentioned that the powder coupling under normal conditions would be suitably charged with shot so as to give a slip of approximately 1 revolution in 10,000, and that, should torsional vibrations occur in the engine speed range, slip would occur and amplitudes would be reduced by as much as 4-1 with little slip at any speed. It was, however, worth remembering that the tests referred to in the paper showed that the larger amplitude reductions were associated with the larger coupling slip conditions. It seemed that the extent to which vibration amplitudes were reduced was dependent upon the ratio of vibratory torque to transmitted torque capability of the coupling. It thus appeared that Mr. Hunt should add vibratory torque to the three torques which he suggested should be considered when assessing the characteristics of this type of coupling.

Mr. Wright had made a valuable contribution to the damping properties and dynamic stiffness of the double-tyre type flexible couplings. It was interesting to note that the dynamic stiffness to static stiffness ratio quoted by Mr. Wright for the double-tyre type coupling (2·2 at a frequency of 3,000 v.p.m.) was very similar to that shown in Fig. 9 of the paper for the bonded rubber disc coupling at the same frequency.

Mr. Wright had made a comparison between the dry-fluid centrifugal coupling and the centrifugal clutch, and there was no doubt that these two had similar characteristics. He had mentioned that fusion of the shot might be a problem, but it seemed fair to suggest that if slip should occur to such an extent that fusion did take place, then surely in similar circumstances a clutch coupling might suffer through lining wear and overheating. In any event, such severe slip would only occur in the presence of severe overload at the coupling, not catered for in the design, with the slipping coupling supplying the protection for which it was intended.

The presence of rubber bushes in the dry-fluid coupling catered for misalignment in a similar manner to those fitted in the centrifugal clutch coupling, and provided that misalignment values did not exceed manufacturer's recommendations, no problems should arise.

In regard to the question of misalignment, emphasis should be given to Mr. Wright's point that this should always be carefully considered in relation to the loads imposed upon adjacent bearings and housings. The inflatable air-cushion type coupling which he mentioned would undoubtedly accommodate large axial displacements with minimum reactions on the adjacent shafts, but it must not be forgotten that these misalignments would produce cyclically varying torques in the system which could produce resonances under the right conditions.

The new developments in the field of highly flexible couplings for high torque rating were welcome. In this connexion the authors had already stated their views that judicious application of such couplings could solve some otherwise difficult or expensive torsional vibration problems.

The authors were glad to have Mr. Nestorides' support of their views on the detuning.

Regarding the advantages of a two-rate flexible coupling, although, as pointed out by Mr. Nestorides, the main practical advantage of such a coupling was its increased load-carrying capacity, nevertheless in the geared example quoted, see Fig. 30 (d), the II/4 propeller + II/2 engine critical was calculated almost exactly at the service speed if the coupling had only its "low" stiffness characteristics. This would have been an undesirable feature which was eliminated by the "high" stiffness rate present at this trial. As a matter of interest, torsiograph records taken on trials held since the paper went into print showed, somewhat unexpectedly, that this critical could in fact be occasionally detected for a period of one to three seconds when slowing down on one engine and thus reducing the transmission torque on the coupling down into the "low" stiffness range. The magnitude of this critical was measured at about two-thirds of the calculated value and its position in the speed range about 20 r.p.m. higher than calculated.

No other criticals could be detected in the working speed range, the gearbox being exceptionally quiet.

The authors agreed with Mr. Nestorides that "gear hammer" was a somewhat subjective term, but nevertheless there was little difficulty in distinguishing severe gear hammer, irrespective of any measures taken to reduce gearcase resonance.

In reply to Mr. Young, the authors would not be surprised if some of the calculated critical speeds in a branched system described by Mr. Young appeared non-existent. Experience had shown that the excitability of some modes of vibration was very much less than expected.

Referring to Figs. 25(c) and 27(b) Mr. Young had asked about the relative effects of damping and detuning. Certainly the shape of the resonance curve in Fig. 25(c) showed non-linearity to be present in the coupling, but only to a limited extent.

The shape of the resonance curve in Fig. 27(b) did not indicate a non-linear characteristic. The authors were of the opinion that the lower measured amplitudes in Fig. 25(c) and 27(b) were due to higher damping than had been calculated, occurring in the system.

Mr. Crum and Mr. Spaetgens, although obviously advocates of the use of low-stiffness couplings, had interposed a timely word of warning in respect to possible excitations due to governor surge or unbalanced firing conditions. The authors were aware of this phenomenon, usually associated with ½-order excitation in four-stroke high speed engines in conjunction with sensitive governing response, and were of the opinion that thorough investigation was highly desirable, particularly in view of the increasing popularity of flexibly coupled engine-generator drives.

The authors were delighted that Mr. Spaetgens also favoured the "flexible" solution, especially as there was still a hard core of designers who preferred the traditionally "stiff" solution.

With regard to the 2nd order excitation, this was generally taken into consideration, particularly where reciprocating scavenge pumps were fitted. In Example 2, Fig. 25(c), the I/2 critical worked its way up to 135 r.p.m. as a result of the rise in the natural frequency, but fortunately this critical had not shown up on any of the torsiograph and strain gauge records taken on two sister ships on trials and after a period in service.

The propeller inertia in this example had not been verified by "swinging", but from experience of comparable propellers there was no reason to doubt the calculated value.

The authors were very glad to receive further confirmation of the undesirability of excessive vibratory torque on the gear mesh, particularly at speeds in close proximity to the normal service speed. Comments to this effect were also made by Mr. Jones in his written contribution. With this problem in view Lloyd's Register Guidance Notes on Torsional Vibration strongly recommended that the vibratory torque should not, in general, exceed one-third of the full transmission torque.

The authors were in agreement that in some cases flanks of the adjoining mode criticals, the I/6 and II/6 in this case, ought to be considered jointly, especially where experience indicated a considerable amount of interaction between the modes, but for the sake of simplicity this treatment was not pursued in this paper. Nevertheless a computer programme was in use for forced-damped frequency tabulations which were necessary if the resultant amplitudes and torques were to be estimated with any degree of accuracy.

This approach was also applicable to cyclic irregularity, raised by Mr. Cranston in his opening remarks.

Mr. Hindmarsh had remarked on the need to consider also the axial stiffness characteristics and asked for information. There was a paper* on a rubber flexible coupling, in which axial, radial and angular stiffness characteristics had been investigated in detail in addition to the torsional stiffness characteristics.

Such data was valuable in assessing loads imposed on the bearings due to misalignment, which point was also raised by Mr. Wright in the discussion. It could be fairly stated, however, that manufacturers did not normally give values for axial stiffness. There were, nevertheless, many couplings available which had freedom of axial movement (spring-grid "Bibby" types, sleeve-spring "Renk" types, gear couplings, etc.) and others which had very low axial stiffnesses, such as the rubber tyre types.

The authors were aware of an additional complication that could arise in some disc type flexible couplings used primarily for angular misalignment, namely that their torsional stiffness could be significantly reduced when the discs were deflected under axial load. In such cases care needed to be taken to ensure that the maximum permissible axial misalignment for the coupling was not exceeded.

Mr. Webb had pointed out that a survey of "Flexible Couplings" was published in 1957 by the *Engineers' Digest* and was still substantially up to date. The authors were aware of this valuable survey, but felt that there was also a need for a publication more in the nature of a design manual, including graphs of stiffness characteristics, moments of inertia, etc., in addition to the full range of torque and r.p.m. available for each coupling.

With regard to the stiffness of couplings employing rubber elements, it was agreed that the stiffness characteristics of such couplings would not be as constant as in all-metal couplings, and it would be helpful to the user if the expected range of variation was stated by the manufacturers. As indicated previously, the authors would generally allow a wider margin of error in natural frequency calculations to accommodate such variations in stiffness. When this was not practicable, recourse to all-metal couplings gave a greater degree of confidence in the frequency calculations.

As regards conventional fluid couplings, the powder coupling, for which tests had been described in the paper, was termed a "dry-fluid" coupling. The tests showed that although the coupling operated effectively as a damper, it did not isolate low frequency vibrations. The authors were also aware of a system incorporating electro-magnetic slip couplings in which a low frequency vibration caused gear hammer.

On the question by Mr. Britain of service experience with the "half-moon" coupling, none was yet available as only prototypes had been tested. A detailed description of the tests carried out on the tug was given in the reply to Mr. Chapman who designed the coupling.

In reply to the specific queries raised by Mr. Britain it could be said that the duty performed by the links under those conditions was not particularly arduous, and that no special precautions would be required beyond those of normal engineering practice.

Mr. Britain had asked about the suitability of the "half-moon" coupling for interposition between the engine and generator of a Diesel-generator set. The type which he was considering was undoubtedly of the single-bearing generator type with generator and engine casing virtually a single unit. The torsional characteristics of this type of machine could often be improved with a low stiffness coupling, but the disadvantage with a conventional rubber coupling in this application was its susceptibility to heat and oil mist. The "half-moon" coupling was normally supplied with rubber bushes to serve as pivots for the links, to give restoring torque and to cater for misalignment. In the case of the single-unit engine-generator set no misalignment problem existed, and it seemed to the authors that the rubber bushes of the "half-moon" coupling could be replaced by self-lubricating bearings, or possibly high-temperature rubber bushes could be used. A pilot bearing would, of course, be required to locate the inboard end of the generator shaft. Provided that calculations showed the system to be tractable, it seemed that the "half-moon" coupling might be very suitable in this application, although this might not obviate the need for a crankshaft damper to deal with engine modes of vibration.

In reply to Mr. Garratt, there was no practical reason why flexible couplings could not be applied for powers in excess of 7,500 b.h.p. and the authors would be pleased to

* Pinnekamp, W., and Jörn, R. April 1964. *MTZ25 Jarhrg.*, Heft 4, pp. 130-135.

evaluate the merits of different solutions for any particular installation. Provided this was done early in the overall design stage, a satisfactory and economically sound solution could generally be arrived at without undue difficulty.

The authors were indebted to Mr. Broersma for his valuable contribution. The modern rubber to metal bonding was generally as strong as the bonded materials themselves. Nevertheless, where the coupling was likely to be subjected to such severe shock loading as describd by Mr. Broersma, the authors would recommend a slip coupling, e.g., of the centrifugal clutch type or the dry-fluid coupling, instead of, or in addition to, the rubber disc coupling, depending on the torsional vibration characteristics of the system. The life of a rubber coupling would depend on its duty in service. If the duty were well within the design characteristics for the coupling, there was no definite life limitation.

Mr. Jones had referred to the similarity of the torsional stiffness characteristics, shown in Fig. 20 for the non-linear coupling in the mechanical analogue, to the two-rate coupling in Fig. 31(e). The authors were of the opinion that had sufficient excitation been present in the geared marine installation to overcome the damping, a comparable non-linear form of the resonance curve would have been obtained.

The authors were very glad to receive further recommendation for keeping vibratory torques on the gear mesh as low as possible, especially near the service speed. This point was also raised by Mr. Spaetgens in his written contribution.

Mr. Jones had asked for guidance notes on the maximum backlash in gear mesh. However, as the primary function of backlash was to allow for differential expansion between gearcase and rotating elements, with such additional factors as misalignment, tooth thickness variation, pitch and helix angle variation, etc., the minimum value of backlash was more important. British Standard 1807 Part 1, Gears for turbines and similar drives, gave a formula, revised in 1962:

$$\text{Minimum normal backlash} = \frac{C \times \psi n}{60{,}000} + 0{\cdot}005\text{in.}$$

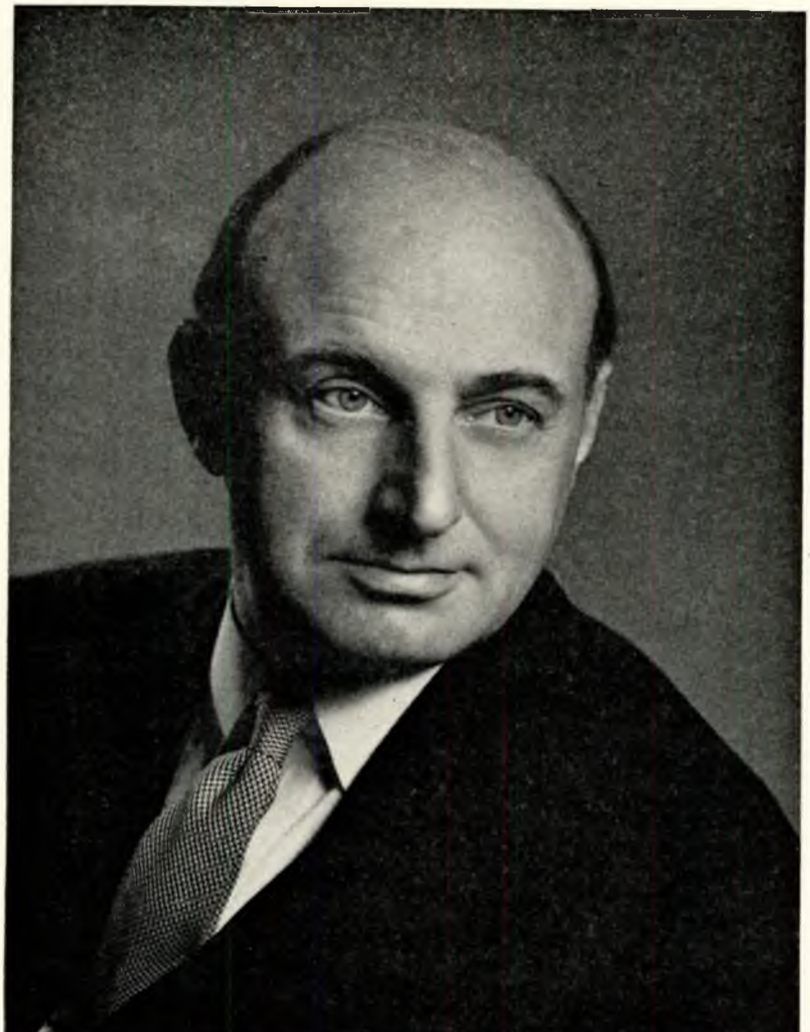
where C = centre distance of the pair of wheels, in inches;
ψn = normal pressure angle, in degrees.

It was generally accepted that this value could be halved for shaved or ground oil engine gears, and as indicated in the paper, the lower value should be aimed for when gear hammer was anticipated.

Reverting to the maximum backlash, B.S.S. recommended that in general the maximum backlash should not exceed three times that given by the above formula.

The authors were aware that the "Geislinger" coupling had a comparatively low torsional stiffness characteristic, but nevertheless the maximum twist in the later designs had been limited so as to prevent excessive spring stresses. Service experience of this relatively new coupling was still limited.

Mr. Grant had described, in detail, a new type of rubber-block flexible coupling: the authors would be glad to receive full details of all the stiffness characteristics, i.e., torsional, axial, radial and angular, and other design data with a view to evaluating the merits of this coupling for specific applications.



SIR JOHN HUNTER, C.B.E., B.Sc., J.P.

SIR JOHN HUNTER, C.B.E., B.Sc., J.P.

Sir John Hunter started his apprenticeship at the Wallsend Shipyard of Swan, Hunter and Wigham Richardson, Ltd., in 1930 on the sandwich system, and graduated in the Degree of B.Sc. in 1935. After leaving Durham University, he spent two years in the Drawing and Design Offices of Wallsend Shipyard, following which he joined the staff of Barclay, Curle and Co. Ltd., first in the North British Engine Works and subsequently at Elderslie Dry Docks, from 1937 until 1939.

In July 1939, Sir John returned to the Dry Docks Department of Swan, Hunter and Wigham Richardson, Ltd., as an Assistant Manager and he was appointed Assistant General Manager of the Department in 1941. In 1943 he was appointed General Manager and he became a Director in 1945, which positions he held when he was appointed Chairman of Swan, Hunter and Wigham Richardson, Ltd., in May 1957. In addition he is Chairman of M. W. Swinburne and Sons, Ltd., the Hopemount Shipping Co. Ltd., Brims and Co. Ltd., Barclay, Curle and Co. Ltd., the Wallsend Slipway and Engineering Co. Ltd., Merchandise Presentations Ltd., and Joint Chairman of Vickers and Swan Hunter Ltd. He is a Director of the Glasgow Iron and Steel Co. Ltd., the British Ship Research Association, Consett Iron Co. Ltd., the Shipbuilding Corporation Ltd., and the Mercantile Dry Dock Co. Ltd.

Sir John Hunter was elected a Member of the Institute in April 1949 and he is also a Member of the Institution of Engineers and Shipbuilders in Scotland and a Member of the Royal Institution of Naval Architects; a Past President of the North East Coast Institution of Engineers and Shipbuilders, the Shipbuilding Employers' Federation, the British Employers' Confederation, and the Grand Council of the Confederation of British Industry; a Past Chairman of the North East Coast Shiprepairers' Association, the Tyne Shipbuilders Association and the Dry Dock Owners and Repairers' Central Council; Vice-President of the Institution of Works Managers and Chairman of the Central Training Council. He is a member of the Research Council of the British Ship Research Association, the North Eastern Electricity Board, Lloyd's General Committee, the Executive Board of the Shipbuilding Conference, the Finance Committee of the National Association of Boys' Clubs, and the Council of the Friends of Scouting.

He was appointed a Companion of the Order of the British Empire by Her Majesty The Queen in June 1960 and created a Knight Bachelor in the 1964 New Year's Honours List. Sir John is a Justice of the Peace for the Borough of Wallsend, a Liveryman of the Worshipful Company of Shipwrights and a Freeman of the City of London.

PRESIDENTIAL ADDRESS

of

SIR JOHN HUNTER, C.B.E., B.Sc., J.P.

I feel very honoured to have been elected as your President and to have the opportunity of addressing you in that capacity tonight.

In opening my remarks to you as your President I would like, if I may, to refer to some remarks made by His Royal Highness The Duke of Edinburgh, when he spoke to you as your President three years ago.

At that time he said that the marine engineering profession must attract well trained and competent men with the vision to design for the future and with the urge to put new ideas into operation. He went on to say, that in his view, the most vital function of the Institute was the organization and supervision of the education of marine engineers at every level, and with characteristic frankness he urged the Institute to do something about it.

There are, I think, seven facets to the profession of marine engineering; Research, Design, Construction, Component Manufacture, Installation, Survey including quality control, and Operation.

In all of these departments of marine engineering will be found two sets of people, both of which share the title of Engineer, each complementary to the other, but each with widely different degrees of professional responsibility. These are the technologists and the technicians, and let me say at once, that neither can do without the other, and the success of industry depends as much on the one as on the other. It is, nevertheless, important to distinguish between the two. The formation of the Engineering Institutions Joint Council, now called the Council of Engineering Institutions (C.E.I.) of which this Institute is a Constituent Member, has resolved this problem by designating the technologist or professional engineer as a "Chartered Engineer" on the basis of an examination equivalent to a British University degree standard, together with membership of a professional institute such as ours.

Your immediate Past President went to a great deal of trouble to explain to members, throughout the Institute's branches, the significance of the Engineering Institutions Joint Council and the impact it will have on future membership.

The new requirements for full membership of the Institute, namely, an approved examination of University degree standard related to a syllabus common to all constituent members of C.E.I., together with requirements related to the specialized technical interest of the Institute concerned, set a formidable problem both to this Institute and the marine engineering industry generally.

Hitherto the shipbuilding and marine engineering industries have not demanded such a high general standard for their engineers and naval architects. True there has been plenty of room in design offices and research establishments for the graduate, but in general the industry has been content, and indeed well served by technologists of more modest academic education.

It is apparent, however, that in the light of increasing international competition and the high level of employment of technologists in many countries, the British industry will suffer if we fail to attract and employ those of at least equivalent standards. We need men with creative ability and forward-looking minds in our design offices, and we need people trained in applied sciences, able to think clearly and understand how to use the best management techniques to give the best results in production. This, of course includes the understanding of human relationships.

A young man with a high academic degree in engineering science is only of use to industry if he uses his well trained brain in that highly competitive industrial environment where costs, urgency and human relations are the predominating considerations.

If we assume, as we should, that in future all the professional posts in marine engineering should be filled by Chartered Engineers, what must be done to improve the attractiveness of the industry and to ensure an adequate flow of Chartered Engineers into it?

First, many more young men than hitherto must seek admission to a university or technical college course leading to an examination recognized by C.E.I. for Chartered Engineers. In my view, this means that the type of man who previously qualified for the Higher National Certificate will, in future, have to qualify to a degree standard. Hitherto, this type of man has been the corner-stone of our industrial management both on the shop floor and in the design office. This is no longer good enough and instead of the Higher National Certificate the status of Chartered Engineer is now required.

I was interested in the statistical analysis recently carried out by the Institute which attempted to assess the future needs of the British marine engineering industry for Chartered Engineers. This Institute has some 10,000 Corporate Members in the United Kingdom of which only about 3,200 are employed in the marine engineering industry ashore. This represents about 65 per cent of the total technologists and technicians employed in the industry, the remaining 35 per cent not being members of the Institute.

There are some 650 qualified technologists employed in this industry in the United Kingdom, and to maintain this position we would not need to recruit more than twenty technologists per annum for the next twenty years. However, many of the posts at present held by technicians, will, over the next twenty years, have to be filled by technologists, and to obtain an adequate recruitment the annual intake must steadily increase to at least fifty or sixty by the year 1986, even assuming no significant expansion in the marine industry.

This estimate was based on an assumption that the ratio of technician jobs to technologist jobs in the industry would remain roughly as it is today, in the ratio of two and a half to one, and assumed that eventually all technologist jobs would be filled by qualified technologists. It is my opinion that this estimate may prove to be conservative.

What is needed if our industry is to keep in the vanguard of technical progress is trained and well qualified engineers—Chartered Engineers. The rationalization of firms, be they shipyards or engine works, necessarily demands the employment of technologists if the greatest benefit is to be obtained from a streamlined industry set to regain world markets for their products. Research, design offices, management, supervisory, sales—these are the departments where technologists should be employed.

Any forward-looking scheme of training should aim at an output of at least fifty technologists per annum. This may take several years to reach but that is the target so far as can be seen at present.

Reverting to the Duke of Edinburgh's exhortation to the Institute to "do something", what indeed has been done, and what remains to be done?

The Institute has taken the first step by assessing the future needs of the industry. It now has an important role to play in helping to satisfy those needs in the field of training and education of marine engineers for technologist jobs. I see this as a joint operation between four parties, namely:

1) The marine engineering industry, consisting of engine builders, manufacturers, shipowners.
2) Universities and technical colleges.
3) The Institute.
4) The student.

The most important element of all is, of course, the fourth, and the pattern of training and career offered, must be attractive to the best type of student.

I am convinced, as anyone connected with the marine engineering industry must be, that the type of training best suited to our industry is that obtained from what is known as the "sandwich" system. It is the great merit of this system that the student does not pursue his studies isolated from the industrial environment of works and drawing offices. It is to be hoped that marine engineering firms will recognize their responsibility to support this system of technical training by providing the workshop facilities which it requires. This could be done under a scheme of training in which a number of firms together make this possible, and jointly finance students irrespective of where they may ultimately be employed.

It is essential that universities and technical colleges, from which it is hoped chartered marine engineers will be forthcoming, should also support this system. The academic course, both for the pass degree and the Diploma of Technology must be recognized by C.E.I. for the qualification for Chartered Engineer status, and these will henceforth comprise a mandatory qualification for corporate membership of this Institute.

In accepting constituent membership of C.E.I. involving the obligation to adhere to the agreed standards for the Chartered Engineer examination, the Institute undertook a serious responsibility.

The decision of the Council was wise and far-reaching in its effect, and if the benefits to the marine engineering industry are to be attained, the Institute has no alternative but to follow up its decision by taking steps to ensure a flow of candidates for corporate membership. Otherwise, there is no future for the Institute in its new constitution. What steps should the Institute take? First, it must subsidize financially marine engineering students taking a degree or diploma of technology course leading to the Chartered Engineer qualification. This is already being done. Secondly, it must seek to organize, in association with industrial firms, a post-graduate course of training which will prepare the candidates for a worthwhile career, and at the same time meet the special requirements for corporate membership of the Institute.

Whether such a course should be provided at a special post-graduate college, or whether existing colleges can provide suitable courses, are questions which will no doubt have urgent consideration by the Council. I need only emphasize that corporate membership of an Institute which is itself a constituent member of C.E.I. is an essential condition for the Chartered Engineer qualification. Young men are therefore bound to seek admission to those institutions which best facilitate their acquisition of this qualification.

It is hoped that firms will co-operate with this Institute by providing practical training and specialist instruction to graduates, and thereafter employ them in positions leading directly to major responsibility. In their own interest it will be necessary for firms to see that the higher posts are eventually filled by chartered engineers.

I imagine that the special requirements for corporate membership of this Institute will include some period of professional responsibility in industry. It is therefore essential that there should be good liaison between firms and the Institute since a candidate will have to obtain some measure of industrial experience before he can be accepted as a Corporate Member and registered as a Chartered Engineer.

A post-graduate course should cover such practical aspects of marine engineering as:

- Design
- Construction and Installation Methods
- Production, Management and Supervision
- Steelmaking and Testing
- Types of Machinery Installations
- Performance of Machinery and Assessment of Reliability.

The list is not exhaustive and no doubt others more competent that I can suggest a more detailed syllabus, but I do wish to emphasize that a post-graduate course of this kind to be of maximum benefit to industry, as well as to the student, should be directly related to the practical needs of the industry. To this end, much of the instruction given should be carried out by experts from industrial firms. The course might also include some expert evaluation of creative aptitude from which future design leaders might be spotted.

It may well be necessary that the Institute provide such a course, or take a lead in financing its operation. Your Council, I know, is fully aware of its responsibilities and enjoys the confidence of the members. I hope that my remarks will be echoed throughout the industry and that our Institute can count on the goodwill and co-operation of industrial firms, universities and technical colleges.

Another step which your Council is taking, and which will further enhance the profession of the marine engineer in industry, is in drawing together with the three other important learned societies, namely, The Royal Institution of Naval Architects, the Institution of Engineers and Shipbuilders in Scotland, and the North East Coast Institution of Engineers and Shipbuilders. An Inter-Institutions Panel has been formed to work out details of closer collaboration, and I wish it every success.

In this Address I have not so far referred to the remaining important element in marine engineering, namely, the seagoing engineers. They represent a small but important part of our membership. They are, in the main, a continually changing group as many obtain posts ashore.

Of the minority who make the sea their calling, the future may well offer a considerable number of high powered and highly automated ships propelled by modern machinery, where a chartered engineer might be in charge. By and large, however, it seems likely that ships will continue to be manned by technicians rather than by technologists. The Institute must provide a route towards the Chartered Engineer grade for such men. Further, the standard of technical competence of the seagoing engineer must continue to be a responsibility of Her Majesty's Government, which, through the Marine Safety Department of the Board of Trade, conducts the examinations for certificates of competency. It may be that the standards and system of certification need to be revised to suit the requirements of a variety of machinery installations, and it may well be that the entire system of shipboard manning needs modernizing. This is primarily the province of shipowners, government departments, seafarers' organizations and the marine colleges. It is sufficient for me to say that the seagoing engineers will always be encouraged to join our Institute which will provide a gateway to the Chartered Engineer qualification for those who can meet the required academic standard.

The raising of the standard of the professional marine engineer must result in the improvement of the standards of marine engineering generally. In this respect we must remember that the Institute is international in its operations, and is particularly concerned with marine engineers throughout the Commonwealth. Improved status based on improved education and training will prove of lasting benefit to all our members and the industry in which they serve.

The Institute has chosen its path, there can be no going back nor can there be any standing still. Responsibility has to be faced and I have indicated how this should be done. You will all endorse His Royal Highness Prince Philip's view that the most vital function of the Institute is the organization and supervision of the education of marine engineers. In making a plea, which I know will not go unheeded, to the British marine engineering industry to support the Institute in its efforts, may I also express the hope, which I am sure all will share, that your Council will press on to a successful conclusion the work it has so well begun.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 9th February, 1965.

An Ordinary Meeting was held by the Institute on Tuesday, 9th February 1965, at 5.30 p.m. when a paper entitled "Flexible Couplings for Marine Installations—Testing and Application" by C. Chartan, B.Sc.Tech., A.M.I.Mech.E. (Member) and D. J. White, was presented by the authors and discussed.

Mr. W. Young, C.B.E. (Chairman of Council), was in the Chair and ninety-six members and guests were present.

In the discussion which followed six speakers took part.

A vote of thanks to the authors was proposed by the Chairman and received by acclamation.

The meeting ended at 7.45 p.m.

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 28th September 1965.

An Ordinary Meeting was held by the Institute on Tuesday, 28th September 1965, at 5.30 p.m. Mr. H. N. Pemberton (Chairman of Council) was in the Chair, supported by Mr. R. R. Strachan (Vice-Chairman of Council), the Honorary Treasurer, Mr. J. Calderwood, M.Sc. (Honorary Vice-President), and Mr. J. Stuart Robinson, M.A. (Secretary).

One hundred and ten members and guests were present.

The CHAIRMAN said that it gave him great pleasure to introduce the President, Sir John Hunter, one of the great leaders in the shipbuilding industry. Sir John was in the best position to put forward the policy which Council was trying to evolve. The Institute was at a crossroads in its history, and with Sir John's help they hoped, in the next few months, to put across to industry a new look and also to seek the help of industry in accomplishing it.

Sir John was particularly suited to be President because he was, in addition to having many other responsibilities, Chairman of the Industrial Training Council.

He then called upon Sir John Hunter to present his Presidential Address.

The PRESIDENT, Sir John Hunter, C.B.E., B.Sc., J.P., then delivered his Address.

The CHAIRMAN thanked the President for an inspiring address, in which he had not only emphasized the problems facing the Institute but had indicated a means of reaching a satisfactory solution to them. The Council would now go forward and seek the co-operation of industrial firms in the post-graduate training of engineers for the shipping and shipbuilding industries.

He regarded the Presidential Address not only as a rallying call to the Institute's own membership, but also to the industry as a whole.

He said it gave him much pleasure to propose a vote of thanks to the President.

MR. R. R. STRACHAN (Vice-Chairman of Council), in seconding the vote of thanks, said that the Address had been very interesting and informative on the role the engineer had to play in the future. The Address contained plenty of food for thought and it was perhaps unfortunate that any debate was precluded on such an occasion because he was sure a very interesting discussion would have ensued if this had been possible.

It was pleasing to note in the audience a number of students and young engineers. They would be aware of the fact that the Council was giving much thought to the problem of improving, by proper training, the role of the engineer in the modern world. It was up to everyone to see that everything was done to broaden the image of the engineer in the public eye.

He had great pleasure in supporting the vote of thanks to their President.

The vote of thanks was carried by acclamation.

The President then presented Parchments of Honorary Life Membership to the following:

Captain W. E. Dommett, elected 2nd March 1915.
Mr. F. J. Mayor, elected 14th April 1915.
Mr. E. M. Sellex, elected 22nd January 1914.

The meeting terminated at 6.30 p.m.

Section Meetings

Auckland

The Annual Dinner and Dance of the Section was held on Saturday, 18th September 1965, in the Berkeley Lounge, Mission Bay, Auckland, at 7.00 p.m.

Ninety-eight members and guests attended the function and among the guests were the Chairman of the Auckland Harbour Board, Mr. R. C. F. Savory, and Mrs. Savory, and Captain R. H. Carter, Harbour Master, and Mrs. Carter. Miss Mary O'Leary, a talented young singer, entertained the guests with several songs.

At the conclusion of the Dinner those present enjoyed dancing until midnight.

Ceylon

Student/Apprentice Works Visit

A student/apprentices works visit, sponsored by the Section, took place on Wednesday, 27th September 1965, at 9.00 a.m. Thirty-three student/apprentices of marine and allied engineering, visited the Colombo Port Commission Dry Dock where the Royal Ceylon Navy vessel H.M.Cy.S. *Gajabahu* was being refitted. Following the tour the party returned to the Voluntary Naval Force Headquarters, Kochchikade, Colombo, where they were given lunch.

In the afternoon the following films were shown: "Turbo Jet Propulsion", "Pride on Workmanship", "Unfinished Rainbow", "E.R.A. goes to Sea" and "The Marine Diesel Engines".

Dinner Dance

The annual social of the Section, in the form of a Dinner Dance, took place at "Little Hut", Mt. Lavinia, at 9.00 p.m., on Friday, 15th October 1965, attended by members, their ladies and guests, numbering fifty.

The guest of honour on this occasion was Mr. M. Chandrasoma, Chairman, Colombo Port Commission and the Port (Cargo) Corporation, who was accompanied by Mrs. Chandrasoma.

North Midlands

A general meeting of the Section was held on Thursday, 14th October 1965, at The University, Leeds, at 7.30 p.m., when a paper entitled "Some Factors Affecting the Selection of Systems for Automatic Control of Marine Machinery" by Ll. Young (Member), and P. J. Wheeler, B.Sc., was presented by the authors.

Mr. H. V. Campbell (Chairman of the Section) was in the Chair and thirty-five members and visitors were present.

Mr. Young explained that, the paper being a lengthy one, he would concentrate on the part dealing with combustion controls and with some excellent slides, he gave a very lucid description of both pneumatic and electrically-controlled systems.

Mr. Wheeler took over for the discussion which lasted over an hour, indicating the interest that had been aroused.

A vote of thanks to the authors was proposed by Mr. G. Prentice (Vice-Chairman of the Section) and carried by acclamation.

The Chairman closed the meeting at 9.40 p.m.

Northern Ireland Panel

Firth Brown Golf Trophy

The first competition for the Firth Brown Golf Trophy was played on Monday, 13th September 1965, at Clandeboye Golf Course, County Down.

Twenty-two members and visitors took part in the competition.

Mr. D. G. Newel, B.Sc. (Member), was the winner of the cup, the runner-up being Mr. R. Harrison (Member).

The visitors' prize was won by Mr. H. McClune, with Mr. J. Frazer as runner-up.

General Meeting

A general meeting of the Northern Ireland Panel was held on Tuesday, 26th October 1965, in the Millfield Building of the College of Technology, Belfast, at 7.00 p.m.

Chairman of the Panel, Mr. D. H. Alexander, O.B.E., F.C.G.I., M.Sc., Wh.Sc. (Local Vice-President) was in the Chair and sixty-eight members and visitors were present.

The speaker at the meeting was Mr. T. B. Hutchison (Member), who presented his paper entitled "30,000 s.h.p. Unitized Reheat Steam Turbine Propulsion".

Mr. C. C. Pounder (Past President) proposed a vote ot thanks to the author which was seconded by Mr. C. McD. J. T. Scott (Associate Member).

The meeting closed at 9.30 p.m.

Merseyside and North Western

A general meeting of the Section was held on Monday, 1st November 1965, in the Conference Room of the Mersey Docks and Harbour Board, Dock Board Building, Pier Head, Liverpool, 3, at 6.00 p.m. when a paper entitled "The Modern Manufacture of Steel for Marine Purposes" by I. M. Mackenzie, was presented by the author.

A brief summary of Mr. Mackenzie's paper follows.

The steel industry is continuously taking action to maintain and improve the competitive position of steel as a basic constructional material.

Apart from being an economic alternative to the open-hearth process, oxygen steel-making processes have special merits of their own with regard to steel quality. However, because of the very fast production rates possible, the oxygen processes have raised very difficult control problems. The vigorous reactions also cause rapid wear of the vessel lining, and this has presented a challenge to the refractory manufacturer. The introduction of oxygen processes has stimulated development of the basic open-hearth process which can now be operated at highly competitive production rates.

Techniques such as vacuum degassing, continuous casting and special heat treatment are increasing the range and improving the quality of steels available to the engineer. Developments in non-destructive testing have improved the efficiency of quality control procedures.

A great deal of effort has been directed to the development of new types of steel with special characteristics requested by designers. Many of the recently developed steels have yet to be exploited in the marine engineering field.

Approximately sixty-five members and guests attended the meeting and a most lively and interesting discussion, opened by Mr. J. L. Snowdon (Member), followed the presentation of the paper.

Scottish

A general meeting of the Section was held on Wednesday, 13th October 1965, at the Institution of Engineers and Shipbuilders in Scotland, 39, Elmbank Crescent, Glasgow, C.2., at 6.15 p.m.

In the absence of Mr. H. Brady (Chairman of the Section), Mr. T. Liddell (Vice-Chairman of the Section) presided, and after extending a welcome to the forty-eight members and visitors present, invited Mr. R. R. Pike (Member), to present the Chairman's Address entitled "A Square Deal for Coastal Shipping Will Relieve Road Traffic Chaos".

Mr. Pike preceded the Address with a short film entitled "Traffic", a film showing the traffic build-up on roads in, and around Glasgow. Mr. Pike then read Mr. Brady's paper.

The subject was very controversial and it was interesting to note that the contributors to the discussion were careful in their remarks. The principal criticism was that the author had chosen the ports of Glasgow and Liverpool and these could not be considered to be representative of vastly inferior conditions which could arise if, say, Liverpool and Hull had been considered. Mr. Pike hastened to say that the study was carried out on a basis of 200 miles travel and that inland waterways could be used rather than the sea for the half way around Great Britain. He did agree, however, that the use of inland waterways had not been covered in the paper but it must be realized that upon such an occasion, it was not possible to cover every aspect of the case.

The discussion could have continued for a considerable time if the contributors had not been guarded in their comments. The discussion was on a very high level indeed.

Mr. R. Beattie (Vice-President) proposed a vote of thanks to the speaker and while expressing some regret that Mr. Brady had had to undertake a business visit to New Zealand and Australia and been unable to present his Address, he congratulated Mr. Pike upon the way he had presented the paper and the very able way in which he had handled the discussion. The vote of thanks was carried unanimously.

The meeting closed at 7.40 p.m.

South Wales

A general meeting of the Section was held on Wednesday, 6th October 1965, at the South Wales Institute of Engineers, Park Place, Cardiff, at 6.00 p.m., when a paper entitled "The Development and Running of the Sulzer RD Engine" by E. T. Kennaugh (Member) was presented by the author. Seventy members and guests attended.

In the absence of Mr. T. C. Bishop (Chairman of the Section), Mr. T. W. Major (Vice-Chairman of the Section) took the Chair.

Prior to the lecture, Mr. Major reported on a meeting of the Chartered Engineers (South Wales) which he had attended as representative of the South Wales Section. He reported that as he had been instructed by the Section to act only as an observer, he was unable to vote on the proposition that the Chartered Engineers (South Wales) should make formal application to the Council of Engineering Institutions to become a local Committee of that Council. The proposition, Mr. Major said, was unanimously agreed upon by all present, with the exception of his abstention.

As Mr. D. Skae (Vice-President), had discussed this proposition in London recently, and the Institute had fully approved this step, Mr. Major wished to place before the meeting the proposition that the Section should be included in the formal application by the Chartered Engineers (South Wales). The meeting unanimously agreed to the proposition.

Mr. Major then introduced Mr. Kennaugh who presented his lecture on the Sulzer RD engine. The lecture was followed by a most interesting discussion amongst those present and a vote of thanks to Mr. Kennaugh, proposed by Mr. Wormald, B.Sc. (Member of Committee), was warmly supported.

Mr. Skae proposed a vote of thanks to the Chairman which received a warm response.

The meeting closed at 7.45 p.m.

West Midlands

A general meeting of the Section was held on Thursday, 21st October 1965, at the Engineering and Building Centre, Birmingham, at 7.00 p.m., when a paper entitled "The Development of the Hovercraft" by R. Stanton Jones, was presented by the author before an audience of fifty-nine members and guests.

The lecture was divided into three parts, the first dealing with the various designs of Hovercraft and their ways of working and the engineering and special features applicable to each.

Slides of the various air-cushioned vehicles were shown to demonstrate the different designs and these were supported by graphs indicating the relative efficiencies, pay load etc., for the various types and developments to date.

Following the showing of the slides, a 15mm. optical sound film was shown which clearly indicated the tremendous potential of these vehicles in almost all fields.

A very lively discussion took place and the meeting closed at approximately 9.00 p.m.

West of England

Annual Dinner and Dance

The Sixth Annual Dinner and Dance of the Section was held on Friday, 8th October 1965, at the Grand Hotel, Bristol. One hundred and forty members and their guests were received by the Chairman of the Section, Mr. J. P. Vickery. The principal guests were the Right Honourable, the Lord Mayor of Bristol, Alderman Thomas H. Martin, M.B.E., and the Lady Mayoress, Mr. H. N. Pemberton (Chairman of Council), and Mrs. Pemberton. Also among the guests were the Chairmen of the local branches of the Institution of Mechanical Engineers and the Institution of Electrical Engineers.

Following the Loyal Toast, the guests were warmly welcomed by Mr. Vickery who said that it was indeed an honour for him to welcome for the first time the Lord Mayor and Lady Mayoress of Bristol. It was a privilege to have amongst the members the chief citizen of such a great and ancient city. Not only was Bristol a great port, but it was also a city of modern engineering. At Portbury, new docks to take bigger vessels had been designed and were waiting to be built. This would be a challenge to engineering of all specialities and would provide facilities equal to those which had been built by Brunel and the great engineers of the past. The Bristol area also had three nuclear power stations, the Severn Bridge, and an aircraft industry, all of which were engineering projects as advanced as any in the world today.

The Chairman then proposed the toast "The City and County of Bristol". In reply, the Lord Mayor thanked the Section for inviting him and the Lady Mayoress to the function and said how much they had enjoyed themselves. Speaking of the new Portbury Docks Scheme, he said he hoped that it would not be long before the City Council were given final Government approval to go ahead. When this was forthcoming, he knew that the marine engineers, because of their "know-how" would help to make the scheme a success, and he looked forward to the day when this project became a reality. At this point the toast "The Institute of Marine Engineers" was given.

On behalf of the Institute and guests, Mr. Pemberton, replying to the toast, thanked the Committee for organizing such a wonderful evening. He wished the Section well for the future and hoped that it would keep up the same interest in the work of the Institute as in the past.

Photograph by Tudor, Facey and Miller Ltd

West of England Section

At the Sixth Annual Dinner and Dance of the Section held on Friday, 8th October 1965, at the Grand Hotel, Bristol. From left to right: Mr. M. R. Goodacre (Honorary Secretary), Mr. F. C. Tottle, M.B.E. (Local Vice-President, Bristol), Mr. H. N. Pemberton, (Chairman of Council), Mrs. H. N. Pemberton, Alderman Thomas H. Martin, M.B.E., Lord Mayor of Bristol, the Lady Mayoress, Mr. J. P. Vickery (Chairman of Section) and Mrs. F. C. Tottle

Dancing to the orchestra of Arthur Alexander followed the dinner and a number of novelty dances was included.

The dance ended at 1.00 a.m.

Election of Members

Elected on 18th October 1965

MEMBERS

Donald C. Barta, M.Sc. (Cornell University)
Donald Bradley
Ir. Nicolaas Dijkshoorn, M.Sc. (Delft)
Frank Cyril Fancey, Eng. Lt. Cdr., R.N.
William James Fielder
George Mervyn Hinds, Lt. Cdr., R.N.
Solomon Aaron Joseph
Raymond Edwin Lawrence, Eng. Lt. Cdr., R.N.
Peter Charles Lansdown
Malcolm Horace Lindsey
Denis Lucy
Thomas Michael McCann
Donald James McKenzie
William Richard Anthony Price
Edmund Robson
Leslie Sinclair
Norman Kenneth Thompson
Arthur Henry Wendt
Joseph Williams
Zbigniew Casimir Wojcik, M.Sc. (Danzig)

ASSOCIATE MEMBERS

John Lambert Ashton
Gerald Anthony Atkinson
Cyril Bond
Leonard John Breewood
Donald Graham Brown
Lawrence Alfred Caine
Michael John Conway
John Cecil Cowpe Craze
Stanley Dagg
Vernon Robert Dale, Eng. Lieut., D.S.M., R.N.
Mondraty Divakaram, B.Sc.
Graham Carrick Henderson
Roy Henson
Alan Hesketh
Balkrishna Chintaman Karve
Ronald Kerr
Abdul Hamid Khan
Dawood Abdul Latif Khatib, B.Sc. (Bombay)
John Lindsay
John Robert Lyth
George Elliot McConnachie
William Mackay
Kenneth MacLeod
Robert Edward Messenger
Surendra Kumar Misra
Ronald Hamilton Nash
Thomas Cooke Needham
Enn Part
John Stewart Elder Riddel
William Oldham Sharples
Soon Tet Loy, Lieut., R.M.N.
Peter Trevor Doughton Williams
Robert Owen Williams
Jan Henry Woodbridge

ASSOCIATES

John Nelson Coffill
Edward G. Noftall
John Oakes
Harry Taylor

GRADUATES

John Russell Acors
Feroze Akhtar Ansari
Bruce Stuart Birnie
Colin James Moncreiff Buckley
Peter Stanley Curtis
Leo Terence Joseph Fitzpatrick
Keith Grant
Albert Edward Isenhood, Jnr.
Roy Jeffery
Terence Edgar Noble
William Henry Pooley
Ian Andrew Ross
Kaushal Raj Sachar
Allan Maxwell Stephenson
Thomas Tate
Graeme John Vagg, Lieut., R.A.N.

STUDENT

Russell Halford Judge

PROBATIONER STUDENTS

Donald Marshall
Michael Albert Leroy Terry

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER

Gordon Beel
William Christian Birkett
James Roland Callow
Roland August DeCuyper
Kenneth Fettes
Guy Griffiths
Geoffrey Parker Jordon
Eric McEwan
Alexander Leitch McIntosh
George Malan
Norman George Margverie
Alexander Imrie Meikle
Geoffrey Colin John Moffatt
Frederick William Norton
Noel Slack Scott
John James Springate

TRANSFERRED FROM ASSOCIATE TO MEMBER

George Allan Anderson
David Frederick Brooke-Smith
James William Foster
James William Jardine

TRANSFERRED FROM ASSOCIATE TO ASSOCIATE MEMBER

Stanley George Bloomfield
William Whitley Grier

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER

Peter Fredrick Anthony
Nitte Subhas Chandra Bhandary
Bernard Donnelly
Felix Alexander D'Souza
Derek Clifford Fermor
Garneth Cyril Alexander Ellis
Keith William Ferguson
George Beckwith Holbourn
Roger George Alfred Hull
Walter George Vernon Lugg
Cedric Robert McGregor
John Anthony Rundstrom
James Runcie Troup
Kunhiramakurup Ravindran
Kadayam Viswanathan Srinivasan
Charles Ralph Willoughby

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER

Donald Frank Brown
Eric Cockerill
Hugh Alexander McDowall Cowan
Anil Walter John David
Peter Albert Dust
David Michael Fuller
William George Hayes

TRANSFERRED FROM PROBATIONER STUDENT TO ASSOCIATE MEMBER
Robert Cyril Crone
John Henry George Heffernan
John Anthony Mawer

TRANSFERRED FROM GRADUATE TO ASSOCIATE
Denis Reginald Hoare

TRANSFERRED FROM STUDENT TO GRADUATE
Michael Bruce, B.Sc. (Hons.) (Durham)
William Alfred Lawrence Cox
Thomas Wilson Steen

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE
Maurice Richard Husband
Alan John Smith
Graham Thomas Walton

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT
Roy Alan Frank Cumley
David Rowbotham

OBITUARY

Captain George Ian Dewart Hutcheson, C.B.E., B.E., R.A.N.

Appreciation by Captain R. G. Parker, O.B.E., R.A.N. (Vice-Chairman of the Sydney Section)

The sudden death on 1st September 1965 of Captain G. I. D. Hutcheson, Vice-President for Australia of the Institute of Marine Engineers, has taken from our midst one of Australia's most distinguished engineers and industrialists, and a man who will long be remembered amongst his friends and colleagues for his lovable personal qualities.

Captain Hutcheson, or "Hutch" as he was affectionately called, was a remarkable man in so many ways. He was born on 9th June, 1897, in Coleraine, Victoria, and attended Adelaide University where he graduated as a Bachelor of Engineering with First Class Honours in 1918. He then joined the Royal Australian Navy as an Engineer Sub-Lieutenant, serving in the First World War on board the battle-cruiser H.M.A.S. *Australia* and being present at the surrender of the German Fleet at Scapa Flow.

During his thirty years career with the Navy, he served in many ships, including H.M.A.S. *Geranium* and the destroyer *Stuart* as the Flotilla Engineer Officer of the "Scrap Iron Flotilla" which later played a major role in the convoys to Malta. He was Senior Engineer of H.M.A.S. *Canberra* from 1927 to 1929 and Engineer Commander of the cruiser H.M.A.S. *Sydney* from 1938 to 1939. In his early career he specialized in gun-mounting engineering, later becoming First Assistant and then Engineer-Manager, as Engineer Captain, at Garden Island Naval Dockyard, Sydney, from 1942 to 1945. From June 1939 to September 1942 he was Principal Naval Overseer at Cockatoo Island Dockyard, Sydney, supervising the construction of *Tribal* Class destroyers, frigates and sloops for the Royal Australian Navy and the conversion of many merchant vessels to transports and armed merchant cruisers. In November 1945, Captain Hutcheson became Director of Naval Engineering, stationed at the Navy Office, Melbourne.

During his period as Engineer-Manager at Garden Island, he was responsible for the repairs—many of them major—to Allied war vessels that had been damaged in the Pacific, in action with the Japanese. These included the refit of several American cruisers and landing ships that had been torpedoed in the Solomons battles, and later on the Royal Navy Fleet carriers, *Formidable, Indefatigable* and *Victorious* extensively damaged by kamakaze attacks. The speed and efficiency with which these repairs were carried out earned high praise from the Commander-in-Chief, Admiral Sir Bruce Fraser. In 1947 Captain Hutcheson retired to become Managing Director of Cockatoo Docks and Engineering Co. Pty. Ltd., a position he held until October 1962, when he became Chairman of all the Vickers Group Companies in Australia.

To the average man these appointments would have represented a full-time job, but Captain Hutcheson was so interested in the engineering profession and shipping generally that he made time to accept many honorary offices, which included:

Federal President of the Institution of Engineers, Australia, 1960; President of the Chamber of Manufacturers of New South Wales, 1953-1954; President of Metal Trades Employers' Association 1952-1953; Chairman of the Australian Shipbuilders' Association from 1962-1965; Chairman of the Standards Association of Australia from 1957 until his death. He also served on the Council of the University of New South Wales since its inception and was recently elected a member of the Australian Committee of Lloyd's Register of Shipping.

The C.B.E. was conferred on Captain Hutcheson in 1960 for his services to industry, and in 1962 he was awarded the Peter Nicol Russell Memorial Medal—the highest award of the Australian Institution of Engineers.

It was Captain Hutcheson who was largely instrumental in the formation, in 1948, of the Sydney Section of the Institute of Marine Engineers, where he served as Honorary Secretary until June 1954, when he was elected Local Vice-President and Chairman of the Section. In 1961 he became the first Vice-President for Australia and continued as Chairman of the Sydney Section until 1963. It was his tremendous drive and enthusiasm that enabled the Sydney Section, and also the new Sections in Melbourne and Perth, to flourish as they have done over recent years. The Institute of Marine Engineers owes him a very great debt in promoting its interests throughout Australia.

To us who served with him and knew him over so many years in the Navy, and later in industry, we feel that we have lost a great friend. Those of us who were privileged to enjoy his friendship will agree that among his outstanding characteristics were his wonderful sense of humour and the great enthusiasm and thoroughness with which he carried out any task he set himself. He had tremendous energy, capacity and wise counsel, and was unsparing in his efforts to produce efficiency. All those who were associated with him, both in Australia and overseas, will long remember him with deep respect and affection.

LIEUTENANT-COMMANDER COLIN ROY ATKINS, R.N. (Member 18740) died suddenly of a heart attack, on 30th September 1965, after a short illness.

Lieutenant-Commander Atkins, who was born on 6th July 1907, entered the Royal Navy as an artificer apprentice in 1923, having attended Medway Technical College for the three previous years. From 1928 to 1938, he was an engine room artificer, being promoted to chief engine room artificer in the latter year. He was Engineer Officer in charge of a watch at sea from 1940 to 1947, and then was Divisional Officer in naval barracks for two years. He returned to Engineer Officer duties in 1949, with the rank of Engineer Lieutenant, and in 1955, joined the staff of the Rear-Admiral for Engineering Duties, The Nore. He was awarded a First Class Service Certificate in 1957, the year in which he retired from the Navy.

On his retirement from the Navy, Lieutenant-Commander Atkins took up an appointment with Spearing and Partners, Ltd., engineers of London, and was concerned with the erection of power plants. In 1960, he moved to work nearer his home, in the production department of Farris Engineering Ltd., a subsidiary of Elliott Automation, at Rochester Airport. He was there until illness overtook him a few weeks before his death.

Lieutenant-Commander Atkins, who was elected a Member of the Institute in April 1957, was a keen member of the South East England Section and attended all the meetings, although he was unfortunately unable to take part in any of the social functions. He had a great gift for making friends and will be greatly missed.

Lieutenant-Commander Atkins is survived by his wife.

JOHN NUTT BLAIR (Associate Member 16420) died on 18th December 1964 at the age of forty-two years.

Having served a five-year engineering apprenticeship, Mr. Blair went to sea in 1944 as a junior engineer with Bibby Line Ltd., transferring later to the British Tanker Co. Ltd., with whom he was employed as third, second and chief engineer. He gained a First Class Combined Steam and Motor Certificate in 1955.

Mr. Blair was elected an Associate Member of the Institute in April 1955. He leaves a widow.

ROBERT GEORGE BRIDGES (Associate 8412) died on 5th November 1964.

Born in 1905, Mr. Bridges first went to sea in December 1926, with the British India Steam Navigation Co., with whom he served for eight years, reaching, by January 1935, the rank of third engineer, and having gained his First Class Steam Certificate. He then joined the Sunderland Oil Co. as engineer in charge of an oil installation.

Mr. Bridges, who is survived by his wife, was elected an Associate of the Institute in 1937.

FREDERICK JOHN CARGILL (Member 4548), a Member of this Institute since May 1922, died on 27th February 1965, aged eighty-five years.

After serving his apprenticeship with J. H. Wilson and Sons of Liverpool, he spent sixteen years at sea in the service of Alfred Holt and Co. Ltd., obtaining a First Class Board of Trade Certificate of Competency and achieving the grade of chief engineer. During the 1920s he left the sea to take up shore employment, one of the positions which he held being as overseer at the Taikoo Dockyard in China. However, he returned to his seagoing career and, in 1939 was serving with the Canadian Pacific Steamship Co. Ltd. He later joined the Haddon Steamship Co. Ltd. and was appointed chief engineer of their vessel *Empire Brutus* in 1947.

Mr. Cargill retired in 1948. He leaves a widow.

FRANK CHARLTON (Companion 17938) died in hospital on 22nd September 1965.

A native of Jarrow, Co. Durham, Mr. Charlton qualified as a chartered accountant in Newcastle and then came to London to join the staff of Price, Waterhouse and Co. After a few years with this firm, he joined the White Star organization in Liverpool, where he advanced rapidly, being appointed chief accountant and secretary at the age of thirty-one, and later becoming a director and general manager of the company. In 1928 he transferred to London.

He was prominently concerned with the merger of White Star Line with the Cunard Line, and was an original director of Cunard White Star and a director of the Cunard Steam-Ship Company.

In 1939 he rendered valuable service to H.M. Treasury in connexion with financial matters affecting shipping interests and later he accepted an invitation to join the Ministry of War Transport, where he remained until 1943.

In 1944, Mr. Charlton was elected a director of Furness, Withy, and Co. Ltd. and deputy chairman in 1948. In 1959 he succeeded Sir Ernest Murrant as chairman. He gave up the chairmanship in 1962, and retired from the board of this and five other companies in 1963.

RICHARD COSTIGAN, O.B.E. (Honorary Life Member 2634) died on 10th May 1964, aged seventy-eight years.

He served his apprenticeship, for five years, with Smith's Dock Co. Ltd., after which he went to sea. During the course of his seagoing career he gained a First Class Board of Trade Certificate of Competency. He was awarded the O.B.E. in the New Year Honours List for 1943.

Mr. Costigan was a Member of the Institute for over fifty years, having been elected in September 1912.

WILLIAM TURNER CROMBY (Member 6215) was born on 23rd January 1897. He served an apprenticeship with G. T. Grey and Co. Ltd. at South Shields, from 1912 to 1918, and then took up his first seagoing appointment as fourth engineer aboard s.s. *Brinkburn*, later becoming third engineer in the same vessel. He served in various other ships in the same capacity until 1921, when he joined Messrs. Butterfield and Swire in Shanghai. He remained with this company until 1929 and at the time of leaving their service was their assistant superintendent engineer at Shanghai. It was during this period that he gained his First Class Certificate of Competency.

Mr. Cromby then worked, as an engineering assistant, for Arnhold and Company of Shanghai, and was engaged in erecting boiler and oil engine installations in many parts of China. He left in 1931 to become a working partner in the company of Nielsen and Malcolm, acting as a consulting engineer and surveyor, and remained with the company until interned in Shanghai by the Japanese.

After the war, Mr. Cromby returned to England and for a short period worked as a draughtsman with the Wallsend Slipway and Engineering Co. Ltd. In 1947, he joined Lloyd's Register of Shipping as a non-exclusive surveyor and was posted to Gibraltar. While there, he also acted as non-exclusive surveyor for the American Bureau of Shipping and was machinery surveyor for the Colonial Government of Gibraltar.

Mr. Cromby retired in 1962 and returned to England to live. He died on 24th January 1965, leaving three children.

HOWELL PRICE JONES (Member 19544) died on 15th July 1965.

Born in 1904, Mr. Jones served an apprenticeship as an electrical engineer from 1921 to 1926 with Campbell and Isherwood Ltd., Bootle, Liverpool, and also studied at Liverpool Technical College.

He continued with Campbell and Isherwood as chief electrical tester and electrical and mechanical works foreman until 1934, when he became assistant works manager.

From 1939 to 1944 he was branch manager in Cardiff and then local managing director of the London and Thames area. He was later appointed a director of the company.

Mr. Jones was elected a Member of this Institute in December 1957 and was also an Associate of the Institution of Electrical Engineers and a Member of the Liverpool Marine Engineering and Naval Architects' Guild.

He leaves a widow.