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MARINE PROPELLER VIBRATION RESEARCH AT THE UNIVERSITY OF ADELAIDE

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AT THE UNIVERSITY OF ADELAIDE

by D.H. NORRIE B.E.(Hons)., B.Sc. *

ABSTRACT

This paper surveys the work which has been carried out and is still in progress in the Department of Mechanical Engineering at the University of Adelaide, on the problems of marine propeller vibrations. The paper is divided into four sections. The first section deals with the problem of ship vibration, its causes, characteristics and the adverse effects which these have on personnel and equipment. The second part of the paper outlines the design and development of the model propeller research facilities. The third section deals with the full-scale ship tests which have been carried out in recent years and presents some of the results obtained. The fourth section deals with investigation into possible configurations of displacement vessels which would have inherently low vibration levels.

1. THE PROPELLER VIBRATION PROBLEM.

The first recorded reference to a fluctuating force being produced by a screw propeller is in John Bourne's treatise on screw propellers, published in 1845. Since that time, there has been an increasing number of references in the literature to the problems associated with such fluctuating forces.

Vibrations on ships can have two adverse effects, firstly on machinery causing excess wear or failure, and secondly on personnel, inducing fatigue and loss of efficiency.

Until recently, the investigations associated with propeller vibrations had perforce emphasised the adverse effects on the various components of the propulsion system and hull of the vessel. There are many instances in the literature of failure of tail shafts due to excessive vibrations in the longitudinal, torsional or lateral directions, excited by the propeller forces. Failure of other components and excessive wear have also been recorded. Investigations showed that in almost every such case the basic cause was that the natural frequency of vibration of the component was of the same order as one of the exciting frequencies due to the propeller, and the well-known phenomenon of mechanical resonance was occurring with resulting excessive fluctuating stresses. Fortunately, the fluctuating propeller forces occur at discreet frequencies related to the shaft speed and number of propeller blades, and the solution to the problem is therefore to design the components in which trouble is likely to occur so that their natural frequencies are well away from the frequencies of the propeller exciting forces generated at the normal operating speeds. In these days, calculations to ensure this are carried out as a routine matter for each new ship and the number of failures associated with propeller excited vibrations has decreased to a very small percentage.

In recent years, however, the other aspect of vibration, that is, the adverse effect on personnel, has been steadily achieving increased prominence. In addition, many ships now carry comparatively delicate equipment and instrumentation whose performance and life can be impaired by excessive vibration. The level of vibration which is regarded as tolerable has thus been reduced in recent years. Unfortunately, over the same period, the power loading on propellers has been increasing, as ship power, speed and size has been increasing and this has resulted in increased levels of propeller exciting force. The vibration problem has thus been becoming increasingly acute and this trend can be expected to continue.

Investigations on various aspects of the propeller vibration problem are proceeding in many establishments. These researches may be divided into several groups:

(a) Experimental investigations aimed at obtaining a better understanding of the physical phenomena which occur.

(b) The development of satisfactory theories so that the effect of changes in configuration can be estimated for any future design, and the levels of vibration which will occur can be calculated.

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(c) Experimental investigations aimed towards obtaining, for design use, numerical data on the various parameters involved in the phenomena.

(d) Theoretical and experimental investigations of alternative propulsion systems which have inherently lower vibration levels.

1.1. The Origin of the Vibrations.

The vibrations which occur in a ship are due to exciting forces causing natural or forced vibrations of the hull and its contents. The exciting forces may be divided into those which are periodic and those which are transient, as shown in Fig. 1.

The transient forces are associated with the sea-way and the resultant unsteady motion of the ship. These forces can be of very high levels, the forces resulting when the hull "slams" into a heavy sea or when the propellers partly emerge from the water during rough conditions, being two examples.

The periodic forces are associated with either the propeller or the machinery of the ship. Such forces have a fundamental component proportional to the operating speed of the propeller or machine, with higher harmonics equal to multiples of this frequency. Vibrations originating in machines are often due to static or dynamic unbalance.

Vibrations associated with the propeller may be devided into two groups. In the first group, the fundamental frequency is at shaft frequency, i.e. once per revolution. Such forces originate from either mechanical unbalance (static or dynamic) or hydrodynamic unbalance (due to each blade of the propeller not being identical). In the second group, the fundamental frequency is at blade frequency (number of blades times rotational speed).

Forces and moments in the second group may be arbitrarily divided on the basis of the mechanism of transmission to the hull, into surface, bearing, and shaft forces. Surface forces are those due to pressure forces on the hull surface. Bearing forces are those transmitted to the ship via the propeller stern-tube bearing. Shaft forces are those transmitted to the ship by the shaft after passing through the stern-tube bearing and thus comprise the torsional and axial forces. An alternative classification of the second group retains the class of surface forces as defined above, but groups the bearing and shaft forces together under the heading "Shaft Forces". The "shaft forces" are then divided up into torsional moment, axial force, and bending moment.

It is often convenient to regard the fluctuating forces associated with the propeller as due to two separate causes, although, in reality, both phenomena occur together and are inter-related. On this basis, the forces can be divided into two types, the first type having their origin in the variation of the wake and the second type being due to the presence of the hull after-body and appendages such as the rudder. These two types will now be discussed separately.

1.1.1. Forces Originating from Wake Variation.

These fluctuations are due to the propeller operating in a non-uniform flow stream, the wake of the vessel. This wake may be regarded as having two components, firstly, that due to the potential flow field around the ship, known as the potential wake, and secondly, that due to the viscous forces acting in the boundary layer of the ship, called the frictional wake. Thus, because of the shape of the ship and the presence of viscosity, the velocity field at the propeller varies axially, radially and circumferentially. During rotation, therefore, an element of the propeller blading experiences a relative velocity which varies in magnitude, direction and angle of attack. The static pressure of the element also varies during rotation due to the depth of immersion changing. The force on the element and its magnitude and direction thus varies cyclically during each revolution. The sum of these blade-element forces, when integrated over the whole propeller, may be resolved into a resultant thrust, torque, and bending moment. (The bending moment is due to the net force being eccentric). The resultant thrust, torque, and bending moment will vary cyclically, with a period of one revolution. The fundamental frequency is at blade frequency (number of blades times shaft speed), with higher harmonics of multiples of this frequency.

The greater the assymetry in the wake, the larger will be the fluctuations in the thrust, torque and bending moment. The degree of assymetry of the wake is dependent on the shape of the after-body of the ship and the position of the propeller in relation to the hull. If the vessel has more than one propeller, the shape and position of the stern brackets or A-frames, and the relative positions of the propellers also influence the wake. Also, when the vessel is turning, the wake pattern in the vicinity of the propeller disc changes considerably and the fluctuations in thrust, torque, and bending moment may be greatly increased, depending upon the amount of side slip, the position of other propellers, and the angle of the rudder.

As an example of the effect of after-body shape, the wake distribution behind a typical "V" section after-body and a typical "U" section after-body is shown in Fig.2. It will be seen that in both cases (and this is generally true for all single-screw configurations) there is a minimum wake velocity in the vertical positions above and below the propeller, at 0 and 180°. As a propeller blade passes through these positions, the angle of incidence of the relative velocity will thus increase and the lift force on blade elements will increase correspondingly. There will thus be a maximum in the resultant force on a blade at the 0 and 180° positions. If the propeller has an even number of blades, the maxima of blade forces for opposite blades will thus be in phase and the resulting propeller vibratory forces will therefore be greater than for a propeller with one less blade. This phasing effect is slightly modified depending on the skew and rake of the propeller blades.

The varying forces on each blade element can excite the blades themselves into various complex modes of vibration. These vibrations help to induce localised cavitation on the blades, especially those which are heavily loaded and located near the surface of the water. When cavitation of this type exists, it generates very high-frequency vibrations which are usually insignificant, insofar as stresses are concerned, although the noise (often of a banging nature) may be troublesome. Since in large prototype propellers, the natural frequencies of the various modes of vibrations of the blades themselves are very high, compared with the natural frequency of the shaft system, this type of blade vibration does not have an appreciable effect on the dynamics of the complete system. In model studies, however, this effect can be significant and alter the characteristics of the system considerably.

1.1.2. Forces Originating from the Presence of the After-body and Appendages.

Associated with each blade of the propeller is a pressure and velocity field which rotates with the blade, and which varies as the blade force changes. If there is a solid surface near the rotating propeller, its presence will cause the pressure and velocity field to vary as the blade changes its distance from the body during rotation. The effect will be greatest when the blade approaches close proximity to the body. The change in the pressure and velocity field due to the presence of the body means a corresponding change in propeller force, as well as a variation in the surface force on the body.

As the pressure field varies, the intensities of pressure at points on the body will change. The resultant force on the body, obtained by integration of the surface pressure forces, will vary in position, magnitude and direction as the propeller blade rotates. The total surface force due to the whole propeller will clearly be cyclic with a period of one propeller rotation. It will have a fundamental frequency equal to blade frequency, and higher harmonics equal to multiples of the blade frequency.

1.2. The Levels of Ship Vibrations and Their Effects on Personnel.

Presented in Table 1 are the results collected from various sources of the torque, thrust, and moment fluctuations for a variety of ship types and configurations. The predominant influence of after-body shape on the magnitude of the fluctuations will clearly be seen. It will also be noted that, on the model scale at least, the use of a wake-adapted propeller* reduces the magnitude of the fluctuations very considerably. Even for the most favourable configurations, however, it will be seen that the magnitude of fluctuations is still comparatively large.

Further information on the magnitude of the fluctuations is given in Table 2, which also shows the division into surface and bearing forces. The torque and thrust fluctuations were not quoted in the reference from which Table 2 is adapted. It will be noted that the surface and bearing forces are of approximately equal magnitude.

*(A wake-adapted propeller is one which has been designed on the basis of a known wake distribution at the propeller, obtained from model tests. The shape and incidence of each blade element are chosen so that the best performance compromise is obtained as the blade element sweeps around its circumferential path passing through the known variation of relative velocity).

Table 3 presents data on the frequency and displacement of the vertical and athwartships vibrations for a number of vessels, mainly those for military application. It will be noted that in exceptional cases the vibration amplitude at the stern is equivalent to 1-2 millimetres. Even larger displacements have been recorded, one well-known passenger liner having displacement amplitudes at the stern of 2-3 millimetres.

Whether a certain vibration level is regarded as acceptable or not will depend upon the criterion used. This criterion will depend upon the use to which the ship is put, military applications usually allowing higher vibration levels than civilian uses such as passenger liners. One set of criteria based on a single parameter, displacement, is given in Table 4, below.

TABLE 4. VIBRATION DISPLACEMENT CRITERIA

RATING	HULL DISPLACEMENT AMPLITUDE (AT SHAFT OR PROPELLER BLADE FREQUENCIES)
Excellent	Less than 0.010 in.
Satisfactory	Less than 0.020 in.
Unsatisfactory	Greater than 0.020 in.

More refined criteria for vibration limits are available, based on two parameters, either acceleration and frequency, or amplitude and frequency. One set of such criteria in terms of allowable acceleration against frequency shown in Fig. 3.

In Fig. 4 is presented a set of curves based on vibration tests on more than 100 ships and on comments from the persons aboard. "The limit of permissible vibrations" shown on this curve, in terms of amplitude versus frequency, corresponds to the vibration level above which most people would feel irritated or annoyed. The author of the paper from which Fig.4 is taken, Professor E.A. Kjaer, has defined a vibration-factor which may be used in conjunction with this curve. This vibration-factor is defined as the ratio of the actual amplitude to the maximum permissible amplitude as determined from Figure 4, at the same frequency. Values of vibration-factor less than 1 are satisfactory, greater than 1 are unsatisfactory. In the same paper, Kjaer showed that in many cases, in ships, the vibration-factor considerably exceeds 1.

The effect of number of propeller blades is usually four or five as, with fewer or greater numbers of blades the propulsive efficiency tends to decrease. In Table 5 is shown a comparison of the vibration forces at both fundamental and second harmonic of blade frequency, for four- and five-bladed propellers. It will be seen that although the four-bladed propeller has more severe torque and thrust fluctuations than the five-bladed propeller, it has very much smaller vending fluctuations.

2. MODEL PROPELLER RESEARCH.

At the University of Adelaide, the research on model propellers is carried out in the 18" Propeller Tunnel. This tunnel was originally of the non-circulating type supplied from a gravity feed tank. The flow was from an elevated tank through a stilling section, a 16 to 1 contraction, and an 18" diameter working section, finally discharging into the laboratory sump. The design of this tunnel is described in Reference 10. In this form, the tunnel was restricted to a maximum water velocity of 8 feet per second. During 1962, the circuit was converted to a closed type incorporating two 125 h.p. centrifugal pumps. When this conversion has been completed much higher velocities will be obtainable.

A dynamometer with variable-speed drive is fitted into the working section so that investigations may be carried out on model propellers under various conditions of rotational speed, approach velocity and loading. The standard size adopted for model propeller work in the tunnel has been 8" diameter. The tunnel was first run in 1956 and during that year Tostevin (Ref.11) developed and tested a strain-gauge propeller dynamometer. Although it quite successfully measured mean values of torque and thrust, difficulty was experienced in resolving the small fluctuating forces at the propeller because of excessive background noise and the low natural frequencies of the mechanical system incorporated in the dynamometer. In 1958, a second design of dynamometer was constructed by Watkins (Ref.12). This used resistance strain-gauges cemented to thin shells as the measuring elements and incorporated a mechanical system using diaphragms and thin rods to separate the torque and thrust, before measuring them. Further development of the dynamometer has since taken place and the present version is described below.

Much of the effort on the model facility has been directed towards the development of a satisfactory dynamometer and also towards obtaining suitably uniform flow conditions in the working section of the Water Tunnel. Satisfactory solutions to both these problems were obtained by mid-1961. Since that time an investigation has been in progress to determine the effect of parameters such as pitch/diameter ratio, amplitude, frequency, and loading on the entrained mass, virtual inertia and damping of a model propeller, all factors of primary importance in the dynamic response of the propeller and shaft system.

2.1. The Propeller Dynamometer.

The general arrangement of the propeller dynamometer is shown in Fig.5. The dynamometer unit, on which the model propeller is mounted, is housed within the drive-shaft, this latter supported at the propeller end by a pressurised water bearing and at the drive end by a double-conical oil bearing. The drive-shaft is rigidly connected to the flywheel which has sufficient mass and inertia to be regarded as fixed when considering torsional and axial vibrations of the measuring system. The fly-wheel is driven through a shaft and coupling system by a variable-speed d.c. motor. The torque and thrust forces on the propeller are separated mechanically within the dynamometer and then measured electrically by the strain gauges cemented to the thin strain shells within the dynamometer. The electrical signals from the strain gauges are carried through shielded wires inside the drive system to the transistor pre-amplifiers located within recesses in the fly-wheel. The amplified signals are then transmitted through slip-rings to the recording and measuring instruments. The signals from the strain gauges are small and without the pre-amplifiers would be seriously interfered with by the brush noise at the slip-rings.

In Fig.6 is shown the means by which the thrust and torque are mechanically separated within the dynamometer. The central rod, rigidly connected to the propeller end-shaft, is supported by flexible diaphragms at two positions some distance apart. The diaphragm at the propeller end, being torsionally stiff but axially soft, allows the thrust force on the propeller to be transmitted down through the central rod to the thrust measuring shells, and allows the torque applied to the propeller to be transmitted through the short outer cylinder to the torsion measuring shell. Since the central rod is torsionally soft but axially stiff, the torque would be resisted by the flexible diaphragm which supports the end of the central rod between the two tension measuring shells. The proportion of the sizes of the components has been chosen so that the dynamometer is comparatively stiff in bending. The inner dynamometer unit incorporating the torsion and thrust measuring system is supported at the propeller end by a close-clearance pressurised aerostatic bearing, and at the other end by a flexible cylinder. The aerostatic bearing allows measurement of the torque and thrust without hysteresis due to mechanical friction. The flexible cylinder has been designed to have low torsional rigidity, low axial stiffness and low bending rigidity, so that the inner dynamometer unit can "float" very slightly within the aerostatic bearing and centralise itself. This was made necessary by the close clearance (0.0005 in. diametral) of the aerostatic bearing, and the resulting difficulty of holding machining tolerances to a sufficiently high accuracy to allow centralisation of the inner dynamometer unit, had it been rigidly attached to the drive shaft. The design is such that the flexible cylinder may be removed and replaced by a softer or stiffer cylinder quite simply. Since the cylinder is the component having the lowest torsional rigidity in the torque path from propeller to drive shaft, and the component with the lowest axial stiffness in the thrust path, a change in the characteristics of this cylinder allows the torsional and axial natural frequencies of the dynamometer system to be easily altered. The only significant mode of vibration for the dynamometer in both the torsional and axial directions is the first mode, in which the propeller oscillates on the flexibility of the dynamometer, the fly-wheel being the nodal position.

2.1.1. The Torsion Cell.

The usual arrangement of resistance gauges in a fully-active torsional bridge relies on the symmetry of gauge outputs to produce zero resultant output when any force other than torsional is applied to the dynamometer element. Lack of perfection in the symmetry, which may result from inaccurate positioning of the gauges, differences in gauge factors or resistances, or unsymmetrical stresses in the element, will produce an unwanted output voltage which will vary as the direction of the applied forces changes. With a rotating element, and a force which is fixed in direction, the resultant bridge output would be cyclic with frequency equal to the shaft rotational speed or a multiple of this.

The arrangement of eight gauges used in the torsion cell in the dynamometer, and shown in Fig.7, seeks to minimise the effects of symmetry errors by placing the gauges in such a way that the two gauges in each leg of the bridge are on opposite sides of the cylinder. As indicated, such a bridge will have zero output if a thrust or bending moment or side-thrust is applied.

The probability that the errors mentioned above will be reduced by cancellation increases with the number of gauges, and was a factor influencing the decision to use eight gauges.

2.1.2. The Thrust Cells.

To avoid the errors which would be associated with buckling of thin cylinders in compression, the thrust measuring unit makes use of a pair of pre-tensioned thrust cells, as shown in Fig.7. These cells are arranged to operate in push-pull, an axial force applied to the centre of the thrust unit causing an increase in tension stress in one cell and an equal reduction in tension stress in the other cell. The arrangement of strain-gauges on the cells is such that both changes of stress cause additive changes in the output voltage. The eight gauges are arranged so that minimum sensitivity to forces other than thrust is obtained, as indicated.

2.1.3. The Transistor Pre-Amplifiers.

The strain-gauge system used on the dynamometer is of the direct current type. There was little difficulty in obtaining the desired sensitivity, but some trouble was experienced in obtaining the requisite stability, especially in the amplifiers. For reasons of size and stability, the pre-amplifiers in the fly-wheel are of the transistor type. The use of push-pull stages throughout the amplifier, coupled with the use of matched transistor pairs, as described below, has enabled satisfactory stability to be achieved.

Zero-stability is of primary importance in a d.c. straingauge amplifier, if mean values of load are to be measured. Drift of the zero may result from a change in the temperature or supply voltage. It is a simple matter to hold the supply voltage constant but it is not practical to hold the ambient temperature constant. Transistors have their performance greatly affected by temperature changes. Careful selection, however, makes it possible to obtain pairs which behave almost identically over a limited range of operating conditions, including temperature. If these matched pairs are used for push-pull stages, any drift that occurs in one will also occur in the other, provided that they are held at the same temperature by a suitable heat sink. In an ideal push-pull pair, drift voltages would have equal magnitude and the same phase, and hence would not cause any change in signal voltage. In practice, there is always some residual drift.

Other measures to reduce zero drift were adopted. In the amplifier first stage, resistors in the emitter circuits cause a large reduction in both zero drift and gain. A low-valued resistor between the two emitters restores the gain to the desired amount, but leaves the drift stabilisation substantially unaffected. Over the whole amplifier, negative feed-back is used to further control the gain, to improve the stability of the second and third stages and the linearity of all stages. As far as drift is concerned, the first stage does not benefit from the negative feed-back. High-stability resistors were used throughout the circuit and the first-stage resistors carefully balanced to reduce drift due to resistance changes. A wiring layout which is symmetrical is an advantage in rotating amplifiers because the voltages generated by the cutting of stray magnetic fields will cancel within the circuit, and such a layout was adopted as far as was possible.

2.1.4. The Characteristics of the Dynamometer.

The dynamometer was designed so that the maximum values of torque and thrust which could be measured would be respectively 50 lbf-in. and 50 lbf. The sensitivity of the system can be adjusted by changing either the amplifier gain or, to a smaller extent, the bridge supply voltage. For the settings that have been adopted as standard, the sensitivities for torque and thrust are respectively 9.35 lbf-in./volt output, and 9.04 lbf/volt output. Static calibration over the ranges of 0-25 lbf-in. torque and 0-25 lbf. thrust, showed the response of both systems to be linear. A dynamic calibration carried out by measuring the natural frequency over a range of vibration amplitudes, with discs of known inertia replacing the propeller, showed the torsional stiffness of the system to be 7,000 lbf-in./radian, with a slight variation depending on amplitude of oscillation. The torsional natural frequency (in air), when the variable-pitch model propeller was fitted, was 159 cycles per second, with a very slight variation depending on the amplitude.

A series of tests was carried out to determine whether there was any cross-coupling between torque and thrust. No such cross-coupling could be detected, the torque output being independent of the applied thrust, and the thrust output being independent of the applied torque. The sensitivity of the torque and thrust to bending forces on the propeller was found to be extremely small.

2.2. Investigations on Entrained Mass.

An investigation is in progress to determine the dependence of the entrained mass and virtual of a propeller on the frequency and amplitude of vibration, and the propeller geometry and loading. The results of this work will be correlated with the values obtained from the investigations on several full-scale vessels. The results obtained from preliminary model studies on virtual inertia are described in Ref.13. In these tests, a model propeller was tested over a range of operating conditions. The blades of the propeller were removable and could be set at any desired pitch.

Further test data will be required before reliable conclusions can be reached, but the following trends were observed:

1. The dependence of virtual polar inertia on the propeller loading, as expressed by the advance ratio J, is very small.

2. The dependence of the virtual polar inertia on amplitude of vibration appears to be small at normal values of pitch/diameter ratio. However, there appears to be a significant dependence on amplitude at high values of pitch/diameter ratio.

3. The primary parameter on which the virtual polar inertia depends, for a given blade shape, is the pitch/diameter ratio.

4. The values of virtual polar inertia obtained are of the same order as commonly accepted values, for medium values of pitch/diameter ratio, but differ considerably from these values at high pitch/diameter ratios. For example, the value of virtual polar inertia is often quoted as being approximately 25% of the polar inertia of a bronze propeller. For a medium value of pitch/diameter ratio, values of 30-35% were obtained for the test propeller. However, for twice this pitch/diameter ratio, the values obtained were between 70 and 90%.

3. FULL - SCALE PROPELLER RESEARCH.

In order to obtain data on shaft vibrations in full-scale ships, several investigations have been carried out by the University of Adelaide. The first of these were on auxiliary-powered schooners which were experiencing some difficulties with tail-shaft failures. The result of these tests, which were carried out in 1954, will be found in Ref.14. In recent years, investigations have been carried out on two diesel-powered tugs and on two 19,000 ton turbine-powered bulk ore-carriers. The most extensive tests were those on the ore-carriers. These were carried out at various ship velocities and shaft speeds under various conditions of draft, water depth, and seaway, and are described in further detail below. Three series of tests were carried out on the ore-carriers:

Series A: On ship A during maker's sea-trials.

Series B1: On ship B (a sister ship to A) during maker's sea-trials.

Series B2: On ship B during a 6,000 mile voyage around the coast of Australia.

3.1. The Nature of the Ore-Carrier Investigations.

Measurements of the instantaneous torque and thrust in the tail-shaft were made over a range of operating conditions, by using electrical resistance strain-gauges cemented to the tail-shaft surface and suitable equipment to amplify and record the electrical output from the gauges, as the tail-shaft stresses varied. The purpose of these tests was:

1. To investigate the effects of factors such as ship draft, speed, water-depth under the keel, and sea-way conditions on the torque and thrust stresses in the tail-shaft.

2. To obtain an estimate of the vibratory forces applied to the propeller under these varying conditions, to be calculated from the data recorded and a knowledge of the physical dimensions and characteristics of the propellershaft system.

3. To obtain information on the virtual inertia and damping of the propeller-shaft system, by recording the behaviour at and near resonance, for comparison with results from the model propeller facility. It was intended only to obtain data on torsional virtual mass and damping, since the torsional natural frequency of the system was within the operating range. The axial natural frequency was well above this range.

4. To obtain information on the flexibility of the thrustblock anchorage from measurements of the movement of the thrust-block.

3.2. Description of the Equipment Used.

The equipment used in the tests on ore-carriers A and B was in essence the same. However, based on the experience from the tests on ship A, certain modifications and improvements were incorporated in the equipment used for the tests on ship B. The description below applies to the equipment as used on ship B.

Electrical resistance strain-gauges were cemented to the surface of the tail-shaft. The output from these gauges was amplified by transistorised pre-amplifiers attached to the rotating shaft. The amplified output was then transmitted through slip-rings to the recording equipment.

To measure the thrust in the tail-shaft, four straing-gauges were used, two being placed parallel to the shaft axis and two at right-angles to it, this arrangement ensuring that signals due to bending stresses would be cancelled out by the bridge configuration and not recorded. For the measurement of torque, strain-gauges were placed at angles of 45° and 135° to the shaft axis, that is, along the lines of the principal tension stresses. Eight torque gauges were used, two in each leg of a fully-active bridge, a symmetrical arrangement being used so that signals due to bending stresses would be cancelled out. Under normal operating conditions the stresses due to the torque in the shaft, although small, were within the range for which strain-gauges are commonly used without undue difficulty being experienced by drift effects and temperature errors. The stresses associated with the thrust in the shaft were much smaller and special precautions against drift and temperature errors had to be adopted. The same measures used in the thrust measuring system were also, as a matter of standard procedure, adopted for the torque measuring system. The most important of these measures were:

1. Gauges were matched as closely as possible for gauge-factor and temperature-coefficient by using only gauges from the same manufacturing batch.

2. Gauges were carefully matched for resistance, by measuring their resistance to the fifth figure under constant temperature conditions.

3. In the later series of tests, the shaft section where the gauges were located was covered with a thick layer of foam plastic to ensure that, as far as possible, all gauges attained shaft temperature.

4. D.C. amplifiers with a low-drift were used on the rotating shaft. The amplifier output, with no power supplied to the gauges, was measured at intervals during the tests, so that a continuous record of such amplifier drift as did occur could be deduced and corrected for.

5. The overall gain of the system was checked at intervals throughout the tests by logging the increase in system output when a high-stability calibrating resistance was connected into the bridge, remote control being used to carry out the switching on the shaft.

The transistorised pre-amplifiers were designed to have very low drift, and incorporated carefully selected and matched pairs of transistors, as well as high-stability resistors. Provision was made for changes of gain and for balance, by multi-position switches.

The slip-rings were of brass and were embedded in an epoxy-fibreglass annulus retained on the shaft by a duralumin clamping ring. The slip-ring assembly was split on a diametral plane to enable it to be attached to the shaft. Two sets of silver-carbon brushes were used, held in perspex brush-boxes A and B at approximately 120° apart around the shaft, to eliminate effects due to the end gaps of the half slip-rings. The split-plane of the slip-ring assembly was positioned to coincide with brush-box A when the axis of a certain blade was vertical. This enabled both the propeller blade position and the shaft rotational speed to be recorded by supplying a half-slip-ring with an electrical voltage from an adjacent slip-ring. Since a single brush (in brush-box A) was used to pick up output from this half-slip-ring, a signal was obtained only during the period that the half-slip-ring was in contact with the brush, that is, during the 180° after the chosen blade had passed through the vertical position.

Both the torque and thrust output from the slip-rings were recorded in two forms. The total output for either torque or thrust can be regarded as the sum of two components, a mean d.c. value and an a.c. component superimposed upon the mean. The mean value was recorded by a high-impedance meter which was heavily damped. The a.c. component was separated from the mean value by means of a specially designed a.c. coupler, then further amplified and recorded. The a.c. coupler was basically a cathode-follower amplifier with a gain of 0.85, which blocked the mean d.c. value but transmitted the a.c. waveform without change of phase. The total wave-form was also recorded, at a lower level of amplification.

The recording unit was a pen-type recorder (8 channel) with variable paper-speed.

Associated equipment included a time-pulse unit, constant-voltage supply units and a strain-gauge displacement meter for measuring thrust-block deflection.

3.3. Analysis of Full-Scale Results.

The analysis of the full-scale results is still in progress. However, some of the information from a preliminary analysis of portions of the data is shown in Figs.8,9,10.

Fig.8 shows the torque fluctuations as a function of propeller rotational speed, obtained from test series A1. For steady-state conditions it will be noted that the peak-to-peak fluctuations as a percentage of the mean torque were only 2½% at the higher service speeds but increased rapidly at the lower speeds as the torsional critical speed of 51 r.p.m. was approached. At 60 rev/min (which is the lowest speed above the torsional critical speed at which the ship is operated under steady-state conditions) the peak-to-peak fluctuations were 23% of the mean torque. Also shown on Fig.8 are the peak-to-peak torque fluctuations for a normal acceleration of the ship from stationary to the velocity corresponding to 100 rev/min. The shaft was accelerated rapidly through the region of torsional resonance and it will be seen that the maximum fluctuations under these conditions did not exceed 26% of the mean torque.

Fig.9 shows the large-amplitude torsional vibration which resulted from the emergency closing of the main steam stop-valve. During the period shown, the ship's forward speed would have changed very little. There would thus have been comparatively little change in the wake pattern over this period. As the shaft rotational speed decreased, the applied fluctuation forces on the propeller decreased correspondingly. However, the value of the mechanical dynamic-magnifier increased greatly as the rotational speed approached the torsional critical. The net result was that the torsional fluctuations were greatly increased as the shaft speed apprcached the critical, the maximum fluctuations recorded being approximately 50% of the normal 100 rev/min mean torque. It will also be noted that the torque passed through the zero value on a number of successive occasions, and would have thus caused gear-hammer.

In Fig.10 is shown the variation in the wave-form of thrust fluctuations at various steady-speed conditions. The curves for different speeds have been reduced to approximately the same peak-to-peak amplitude to facilitate comparison.

To facilitate the analysis of the data by digital computer, an Automatic Digital Curve Reader is being developed. This instrument, by photo-electrically scanning the pen traces on the recording paper, will print the digital values of successive ordinates on paper tape. As the requirement for converting an analogue-recording on paper to digital form is a common one, the Curve Reader is being constructed so that it can be easily adjusted to handle the records from any given pen recorder.

4. RESEARCH ON LOW-VIBRATION CONFIGURATIONS.

For any vibration problem there are three possible solutions:

1. The design of the source can be changed to reduce the omitted level of vibration forces.

2. The transmission of the vibrations from the source to the persons, machines or vehicles whose operation they are disturbing, can be reduced by interposing between the source and recipient, vibration-absorbing barriers.

3. The person or machine being disturbed can be resiliently isolated.

In the case of ship vibrations, the third method is not applicable except in certain special cases. An interesting exception to this rule occurs on a certain modern liner for which it has been found that the vibration levels associated with the full service speed are such as to cause annoyance to a considerable proportion of the passengers. During daylight hours therefore, the ship's speed is reduced to a value at which the vibration level is acceptable, but during the latter part of the night and early morning when it is assumed that the passengers are resiliently isolated on their beds, the ship's speed is restored to maximum. The use of the second method to reduce the transmission of vibration is not practical in the case of the ship, because both the large magnitude of the transmitted forces and their comparatively low frequency would require that the barriers be heavy and cumbersome. The first method, that is, to reduce the vibrations at the source, while involving many difficulties, does appear to offer some promise. It would require a change in the configuration of the propulsion system.

Early in 1962, an investigation was commenced on possible configurations of propulsion systems which would have inherently low vibration characteristics. There are a number of requirements which must be satisfied by any such propulsion system. The propulsive efficiency must be at least comparable with those achieved by present conventional screw propellers. The steering power must be adequate. Last, but not least, the ability to maintain satisfactory performance in varying conditions of sea-way and the ability to withstand damage in excessive conditions, must be comparable with screw propellers.

A number of possible configurations are being considered and an attempt is being made to compare their characteristics. Those configurations which the preliminary study shows to be worthy of further consideration will be examined in greater detail, a project design study being made in each case. If necessary, model studies will be carried out. The construction of a 20 foot self-propelled model incorporating the system showing the most promise is contemplated.

4.1. Configurations Under Study.

Some of the configurations being considered as likely to have inherently low vibration characteristics are:

1. Conventional screw behind a hull with modified after-body shape.

2. Various configurations with the propeller mounted at or forward of the bow.

3. Various multi-hull configurations in which the propeller or propellers would operate in more uniform wakes than with the conventional system.

4. A submersible hull consisting of a cigar-shaped main body completely submerged with a narrow streamlined superstructure of uniform cross-section protruding above the waterline.

5. A vessel with propulsive units in pods on outriggers.

6. Various forms of internal duct systems (hydraulic jet propulsion).

Although the study has not progressed to the stage where definite conclusions may be reached, it does already appear that in particular specialised applications certain of the above systems may be advantageous.

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TABLE 1. - VIBRATION EXCITING FORCES AND MOMENTS

SHIP	MODEL OR PROTOTYPE	PROTOTYPE: Length (between Perpendiculars)	PROTOTYPE: Displacement	PROTOTYPE: S.H.P.	PROTOTYPE: R.P.M.	PROTOTYPE: Thrust	PROTOTYPE: Speed	NUMBER OF PROPELLERS	BLADES PER PROPELLER	$\frac{\Delta T}{T}$ % *	$\frac{\Delta Q}{Q}$ % ϕ	(HALF p-to-p FLUCTUATION) MEAN: Vertical Force (Surface) %	(HALF p-to-p FLUCTUATION) MEAN: Athwartships Force (Surface) %	(HALF p-to-p FLUCTUATION) MEAN: Couple (About Prop. Axis) %	REFERENCE	REMARKS
Maritime C-4 Class Vessel	M	528 ft.	18,610 ton	13100	96	191,900 lb	20 kt	1	4	5-8	12-15	5.3	15.5	68	1	Dry Cargo Ship
s.s. "Gopher Mariner"												7.0			1	
s.s. "President Hoover"	M							2	3		3.2	12.5			2	
	P										3.3				2	
Old "Colony Mariner"	M	523 ft.			103		21.3 kt	1	4			7			1	
	P	528 ft.			100		21.3 kt	1	4			5			1	
Single Screw Vessel	M						14 kt	1	4	16.0	7.4				3	Identical models but different measuring techniques
	M						14 kt	1	4	16.8	10.2				3	
"Hadrian"	P	215 m	60,119 c.m.		103	145 ton	17.5 kt	1	4	7-8	3.8				4	Tanker A.G. Weser (Cigar shape stern)
Series 60 Models																
Extreme U-section	M	400 ft.		96%			17.5 kt	1	4	20.0					5	
[illegible]erate U-tion [illegible]nt)	M	400 ft.		100%			17.5 kt	1	4	14.6					5	

TABLE 1. - VIBRATION EXCITING FORCES AND MOMENTS

SHIP	MODEL OR PROTOTYPE	PROTOTYPE: Length (between perpendiculars)	PROTOTYPE: Displacement	PROTOTYPE: S.H.P.	PROTOTYPE: R.P.M.	PROTOTYPE: Thrust	PROTOTYPE: Speed	NUMBER OF PROPELLERS	BLADES PER PROPELLER	$\frac{\Delta T}{T}$ % *	$\frac{\Delta Q}{Q}$ % ∅	(HALF p-to-p FLUCTUATION) / MEAN: Vertical Force (Surface)%	(HALF p-to-p FLUCTUATION) / MEAN: Athwartships Force (Surface)%	(HALF p-to-p FLUCTUATION) / MEAN: Couple (About Prop.Axis)%	REFERENCE	REMARKS
Extreme V-Section	M	400 ft.		96%			17.5 kt	1	4	13.5					§	
Tanker Model Series																
Extreme U-Section III	M	205.5 m	39,000 ton d.w.	102%	108	139 ton	16 kt	1	4	22	11				4	
Moderate U-Section (Parent) I	M	205.5 m		100%	110	133 ton	16 kt	1	4	18	16				4	
Extreme V-Section II	M	205.5 m		106%	112	133 ton	16 kt	1	4	20	10				4	
Parent with Hogner Stern IV	M	205.5 m		103%	108	145 ton	16 kt	1	4	16	5				4	
III + Wake Adapted propeller	M	205.5 m		101%			16 kt	1	4	9	2				4	
I + " " "	M	205.5 m		101%			16 kt	1	4	6	3				4	

TABLE 1. - VIBRATION EXCITING FORCES AND MOMENTS

SHIP	MODEL OR PROTOTYPE	PROTOTYPE						NUMBER OF PROPELLERS	BLADES PER PROPELLER	$\frac{\Delta T}{T}$ % *	$\frac{\Delta Q}{Q}$ % ø	(HALF p-to-p FLUCTUATION) / MEAN			REFERENCE	REMARKS
		Length (between Perpendiculars)	Displacement	S.H.P.	R.P.M.	Thrust	Speed					Vertical Force (Surface)%	Athwartships Force (Surface)%	Couple (About Prop.Axis)%		
II + Wake Adapted Propeller	M	205.5 m		103%			16 kt	1	4	6	3				4	
III + Propeller nozzle	M	205.5 m		101%			16 kt	1	4	16	12				4	

* p-to-p Thrust fluctuations as a percentage of mean thrust.

ø p-to-p Torque fluctuations as a percentage of mean torque.

TABLE 2 - BEARING & SURFACE FORCE FLUCTUATIONS ON A SINGLE-SCREW PROTOTYPE SHIP AS DETERMINED FROM A MODEL STUDY. (Adapted from Ref.1).

SHAFT SPEED REV/MIN.	93	93	93
SHIP VELOCITY kt	16.9	20.0	23.0
MEAN SHAFT THRUST lbf	221,000	169,000	112,000
MEAN SHAFT TORQUE lbf-ft	840,000	677,000	497,000
TRANSVERSE FORCE Surface Bearing Total *	 23,800 16,500 28,000	 18,000 16,000 26,000	 21,000 15,000 35,500
TRANSVERSE FORCE (% of total) Surface Bearing	 85 69	 69 61	 59 42
VERTICAL FORCE lbf Surface Bearing *Total	 9,500 5,500 7,500	 4,500 6,000 11,000	 6,500 7,000 14,000
VERTICAL FORCE (% of Total) Su~face Bearing	 126 73	 41 54.5	 46.4 50
COUPLE (ABOUT PROPELLER SHAFT) lbf-ft Surface ϕTotal	 320,000 680,000	 430,000 420,000	 650,000 1,020,000
SURFACE COUPLE (% of Total)	47	102	64
FORCES AS % OF MEAN THRUST Transverse Surface Bearing Total Vertical Surface Bearing Total	 10.8 7.5 12.7 4.3 2.5 3.4	 10.6 9.45 15.4 2.66 3.56 6.5	 18.8 13.4 31.7 5.8 6.25 12.5
TOTAL COUPLE AS % OF THE MEAN TORQUE	38.1	62.1	205

* Total Force = Svrface Force + Bearing Force + Force on Rudder (Vectorial addition)
ϕTotal Couple = Surface Couple + Couple on Rudder (Vectorial addition).

N.B. Values of vibratory forces and couples are for peak-to-peak fluctuations.

TABLE 3. - TYPICAL VIBRATION DATA OBTAINED ON SEVERAL CLASSES OF SHIPS

KIND OF SHIP	LOCATION	SHAFT SPEED, rpm	VERTICAL VIBRATION		LOCATION	SHAFT SPEED, rpm	ATHWARTSHIP VIBRATION	
			FREQUENCY, cpm	DISPLACEMENT AMPLITUDE, in.			FREQUENCY, cpm	DISPLACEMENT AMPLITUDE, in.
Ammunition ship, maritime hull No.MC1575 (four-bladed propeller)	Stern	95 45	95* 105-110‡	0.014 0.013	Stern	40	40*	0.025
Refrigeration ship, maritime hull No.MA36(four-bladed propeller)	Stern	100	400 +	0.018	Stern	37	148+	0.019
Icebreaker (two three-bladed propellers)	Stern	95	285+	0.040				
Destroyer (two three-bladed propellers)					Stern	180 240	540+ 240*	0.007 0.028
Destroyer (two four-bladed propellers)	Stern	220 310 240 60	74 ‡ 310* 960 + 461 ϕ	0.180 0.021 0.004 0.002	Stern	300 123 310	120 ‡ 123* 1,300 +	0.134 0.033 0.003
Aircraft carrier (four five-bladed propellers)	Stern Island	160 153 153 150	160* 765 * 153 750 +	0.008 0.005 0.003 0.002	Stern Island	165 165 115	165* 825 + 575 +	0.010 0.003 0.002
Heavy cruiser (two four-bladed propellers)	Stern	330 280	60 ‡ 1,120*	0.009 0.002	Stern	300 300	90 ‡ 1,200 +	0.017 0.002

NOTE: Shaft rpm nominal
* shaft frequency
+ blade frequency
‡ hull natural frequency
ϕ double blade frequency

(From Ref.7)

TABLE 5

Comparison of Vibration Magnitudes for 4-bladed and 5-bladed Propellers.

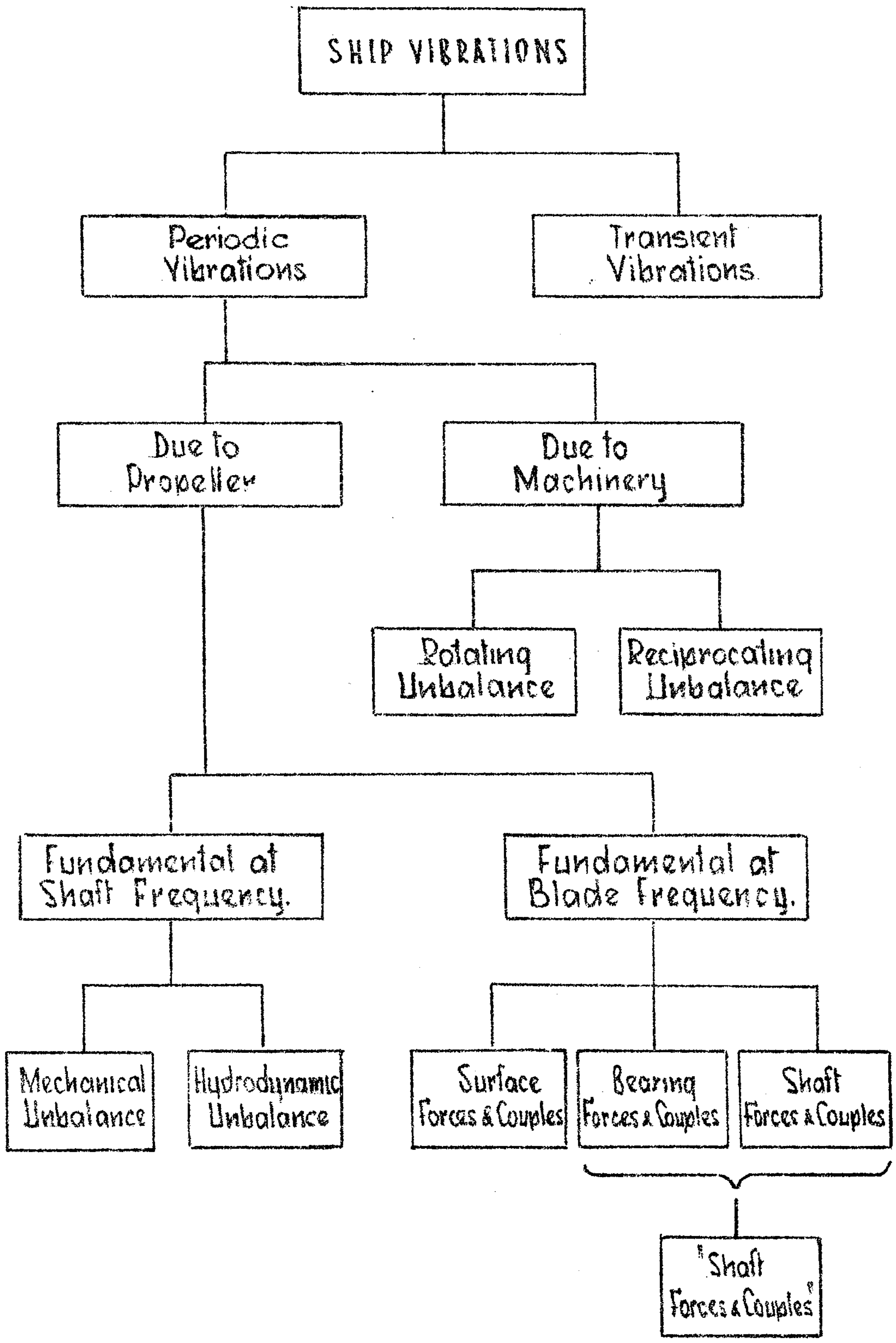
(Adapted from Ref. 9)

TYPE OF VIBRATION		4-BLADED						5-BLADED					
		Single-screw			Multi-screw			Single-screw			Multi-screw		
		L	M	S	L	M	S	L	M	S	L	M	S
HULL	Vertical			4N		4N			5N 10N			5N	
	Athwart-ships			4N			4N			5N 10N			5N
SHAFT	Torsional	4N		8N		4N				5N 10N			5N
	Axial	4N	8N			4N				5N 10N			5N
	Bending			4N			4N	5N		10N		5N	

L = Large M = Medium S = Small

N.B. N is the shaft speed in rev/min. The fundamental and second harmonic frequencies for the four and five bladed propellers are 4N and 8N, 5N and 10N respectively.

FIG. 1 CLASSIFICATION of SHIP VIBRATIONS.



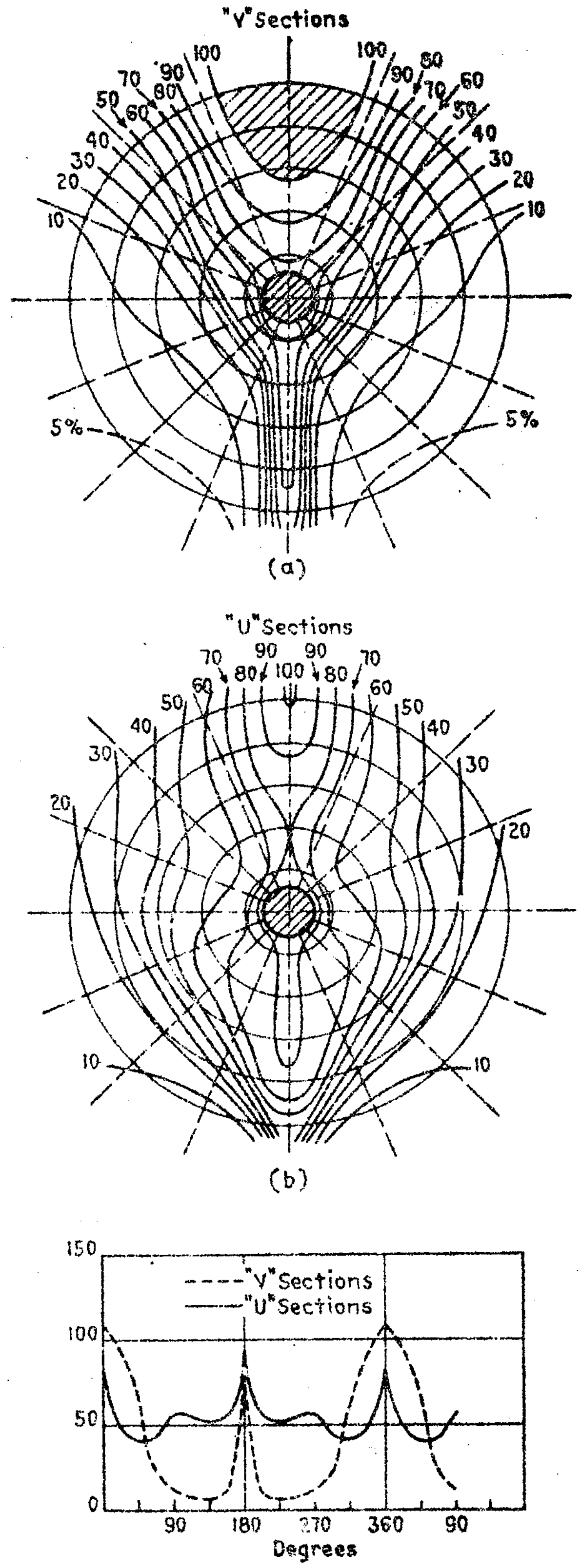


Fig. 2. Wake variation over the propeller disk, for V- and U-shaped afterbodies. The numerical values give the local velocities as a percentage of the ship speed. (c) gives the wake variation at the 2/3 radius.

(Adapted from Ref. 6).

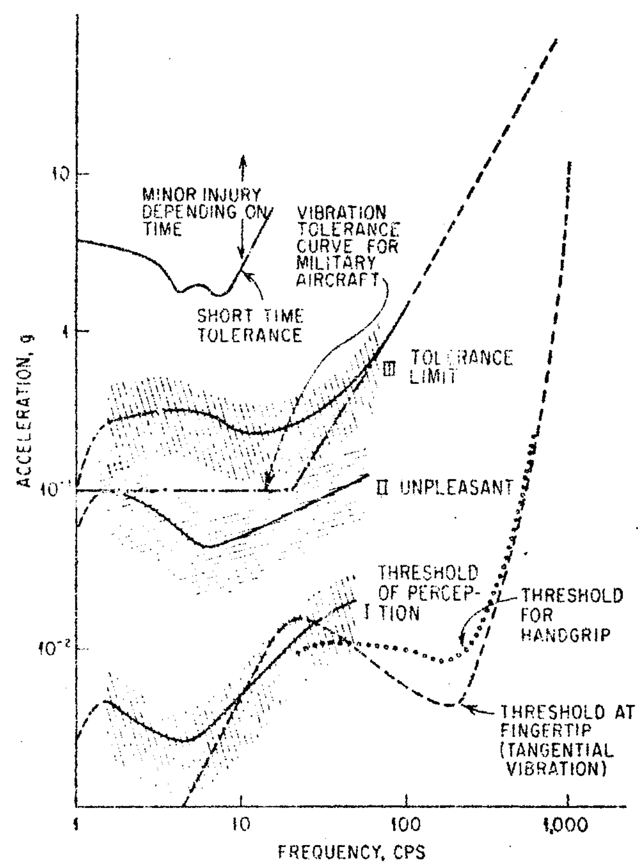
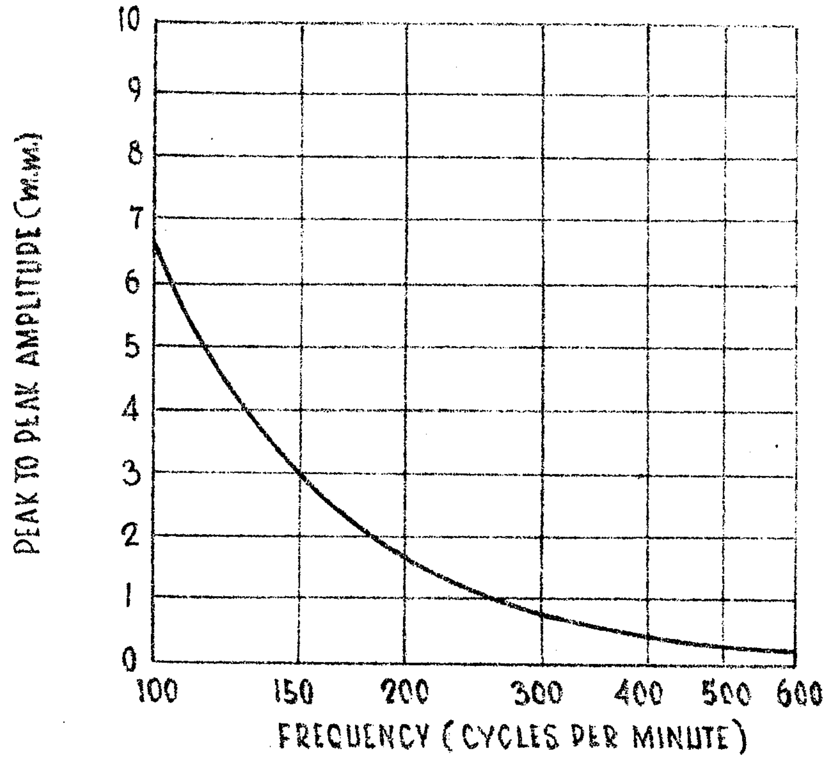


Fig. 3. Vibration tolerance criteria. Peak accelerations at which subjects perceive vibrations (1); find it unpleasant (11); or refuse to tolerate it further (111). The shaded areas are one standard deviation on either side of the mean. These curves are for subjects without any protection, exposure time 5 to 20 min. The short-time tolerance curve is for subjects with standard Air Force lap belt and shoulder harness. exposure time approximately 1 min. The WADD "Vibration Tolerance Curve for Military Aircraft" is used for long-time exposure in military aircraft.

(Adapted from Ref. 7).



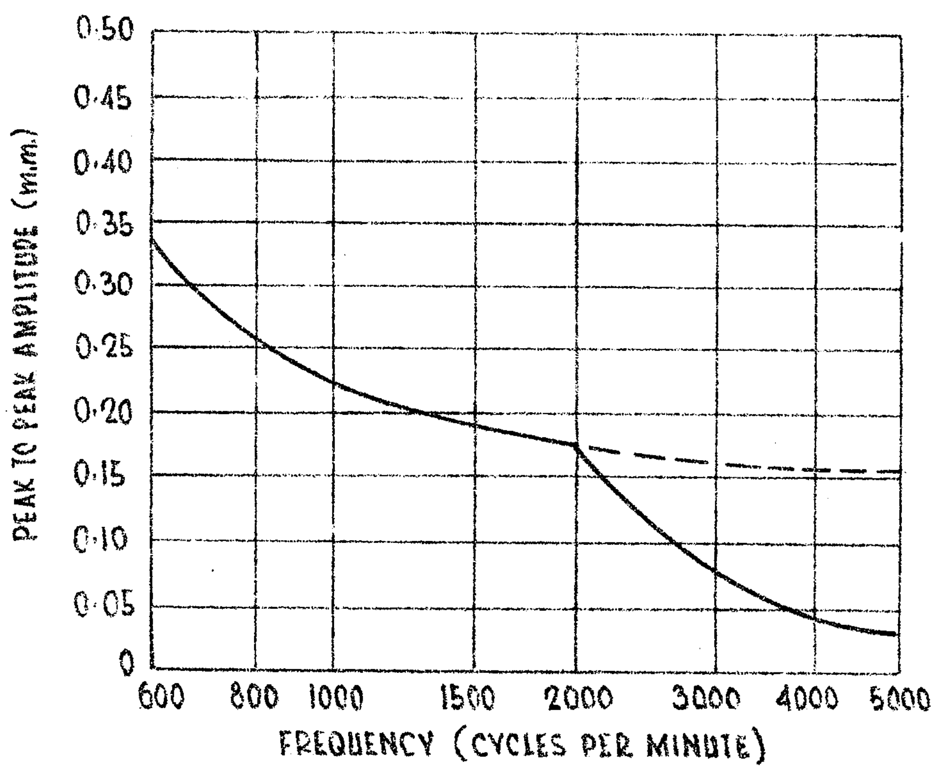


FIG. 4 PERMISSABLE VIBRATION LEVEL ON SHIPS

(Adapted from Ref. 8.)

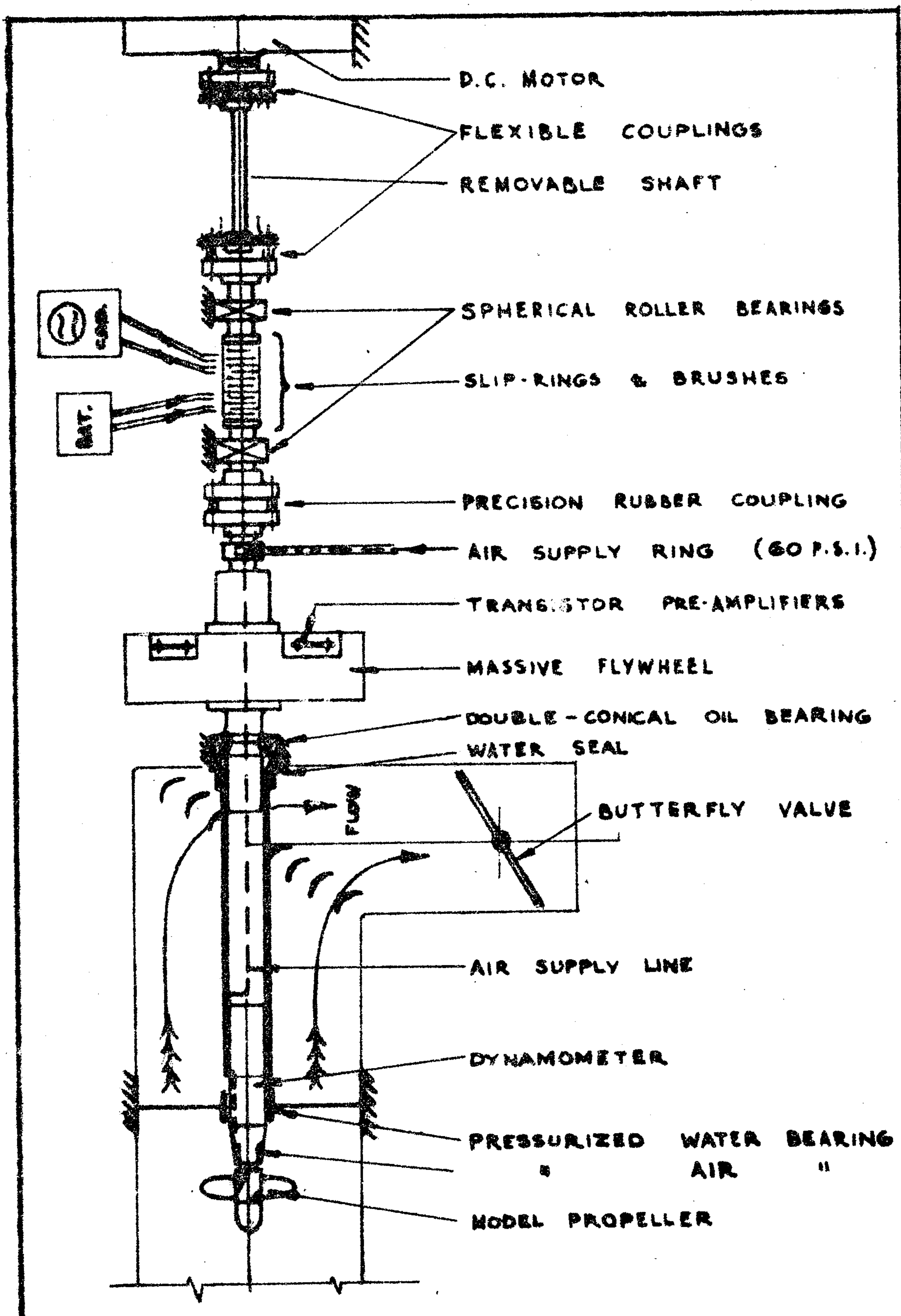


FIG. 5 GENERAL ARRANGEMENT OF DYNAMOMETER.

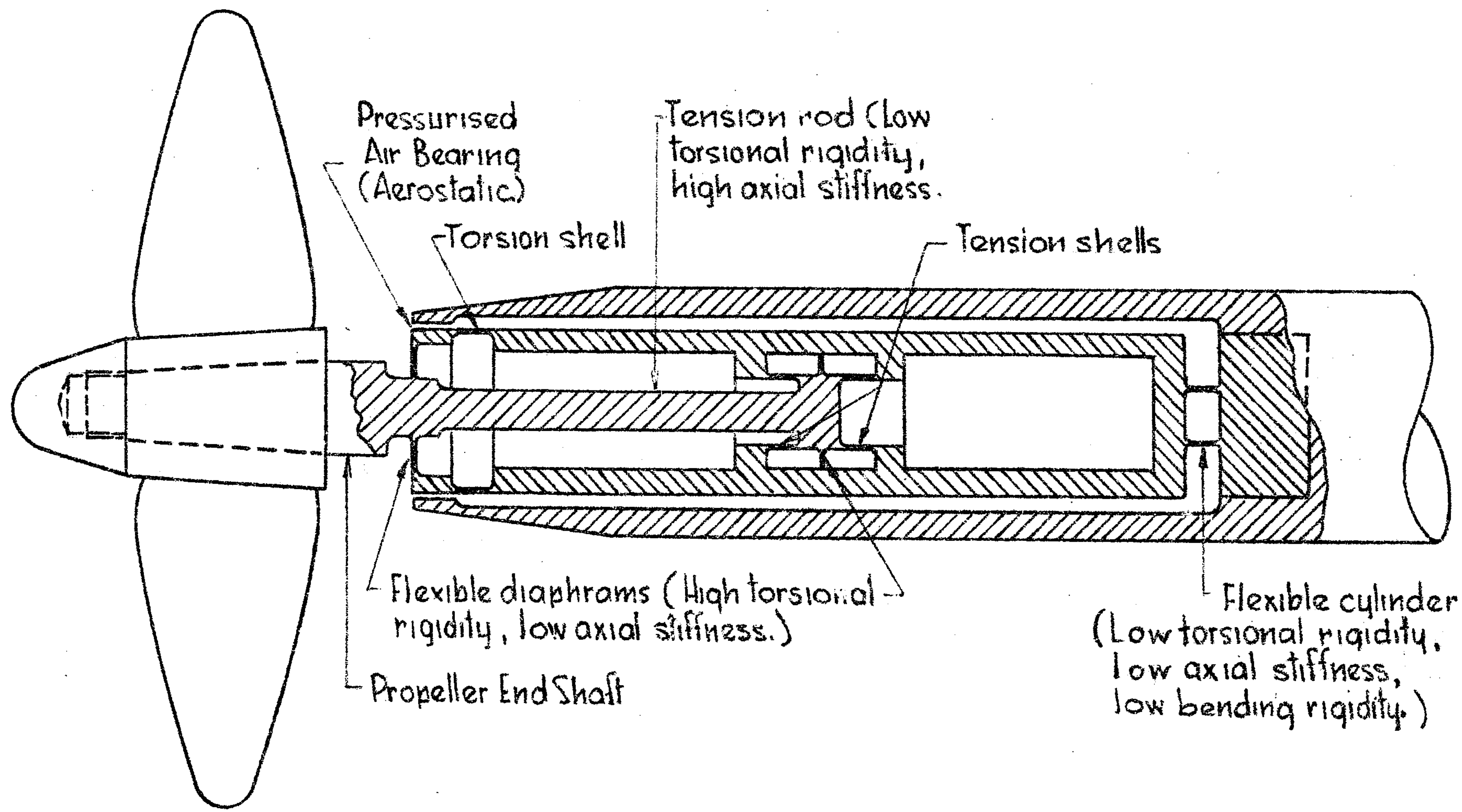


FIG. 6 CONSTRUCTION OF PROPELLER DYNAMOMETER (Diagramatic.)

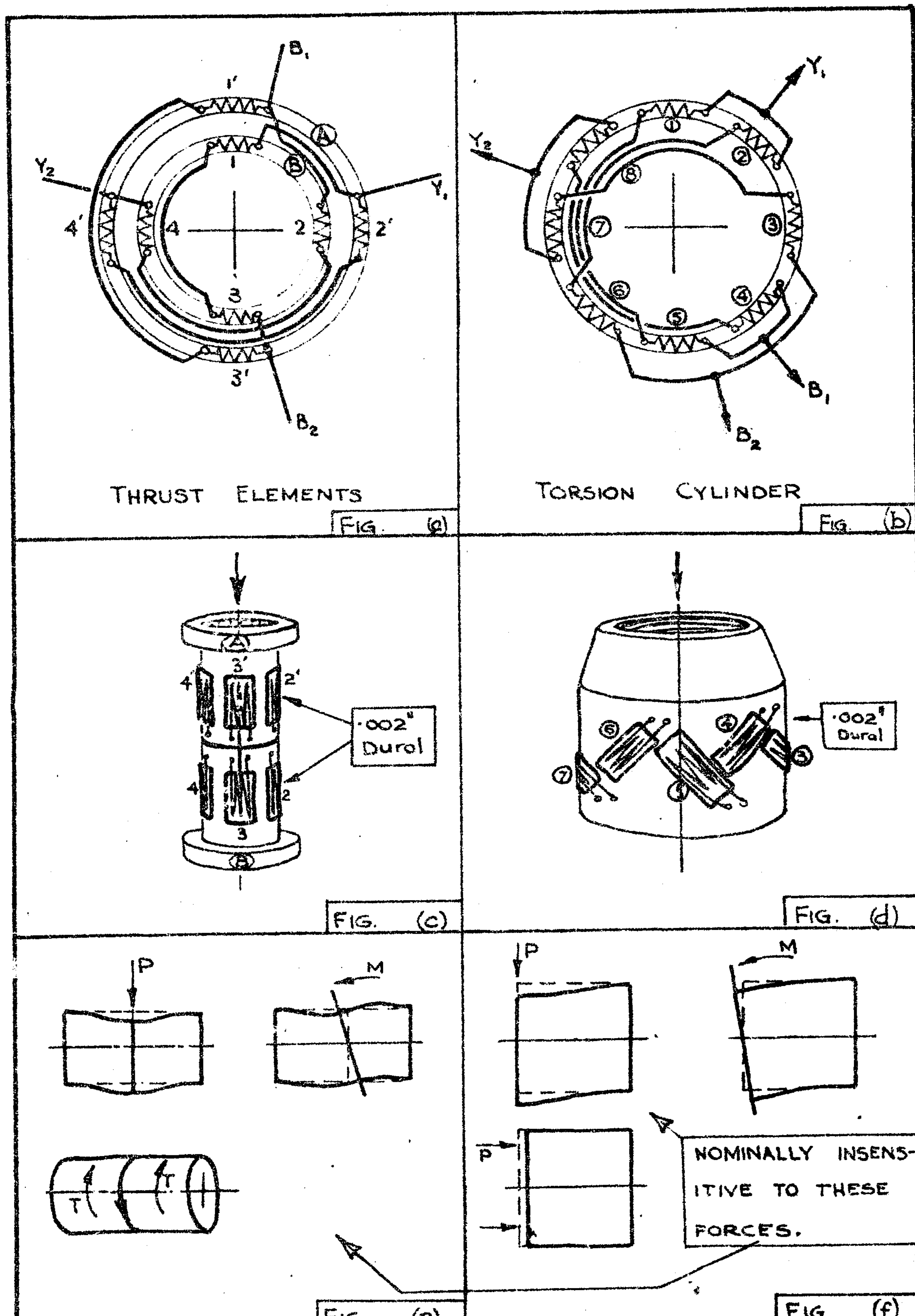
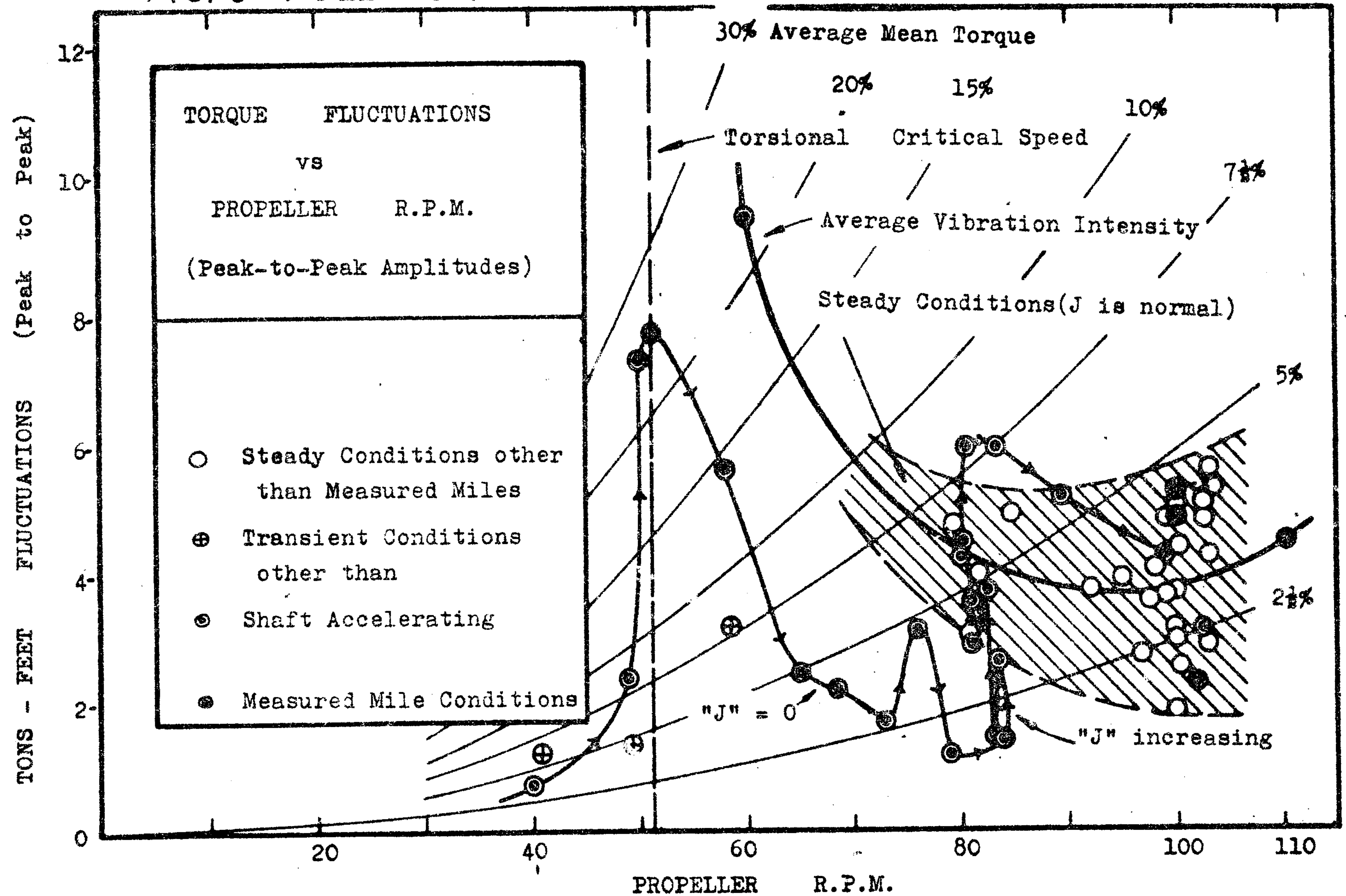
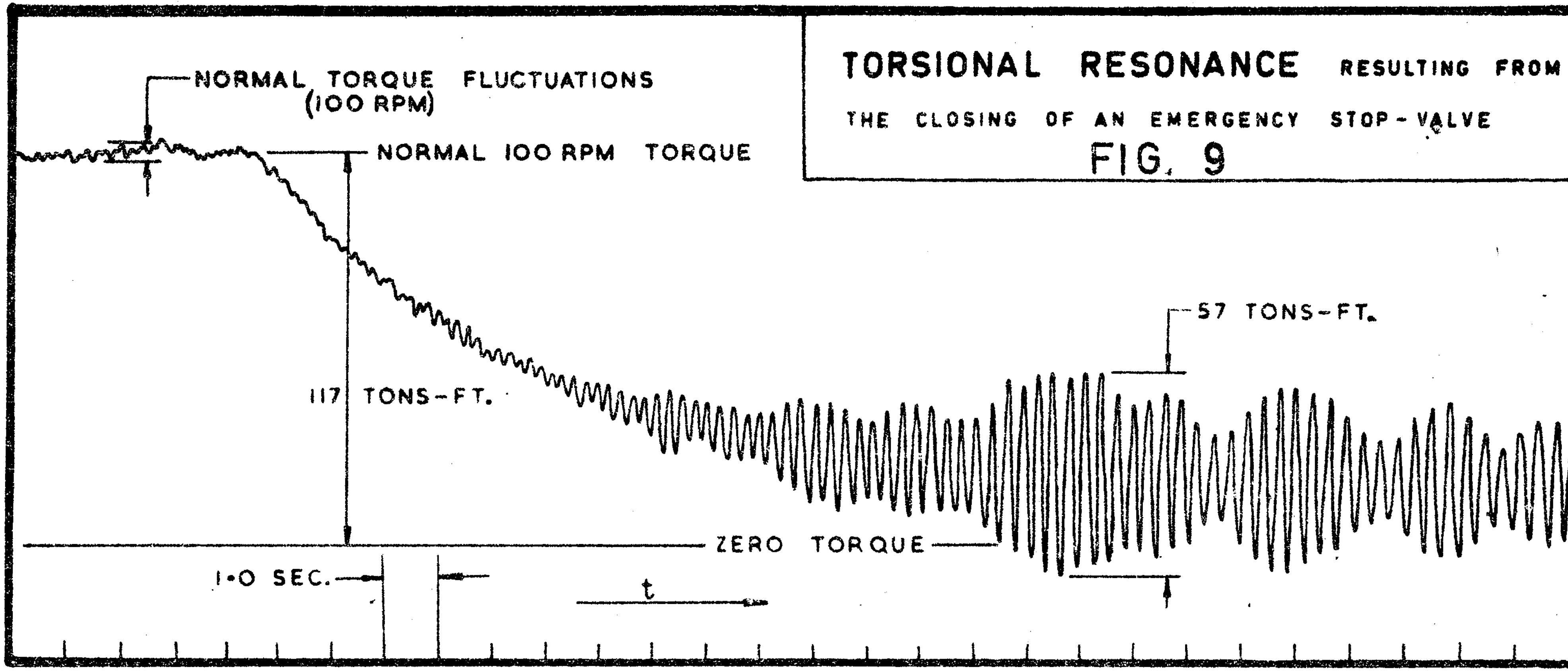


FIG. 7 THRUST AND TORSION CELLS

FIG. 8 FULL SCALE TORQUE o FLUCTUATIONS.





TORSIONAL RESONANCE RESULTING FROM THE CLOSING OF AN EMERGENCY STOP-VALVE

FIG. 9

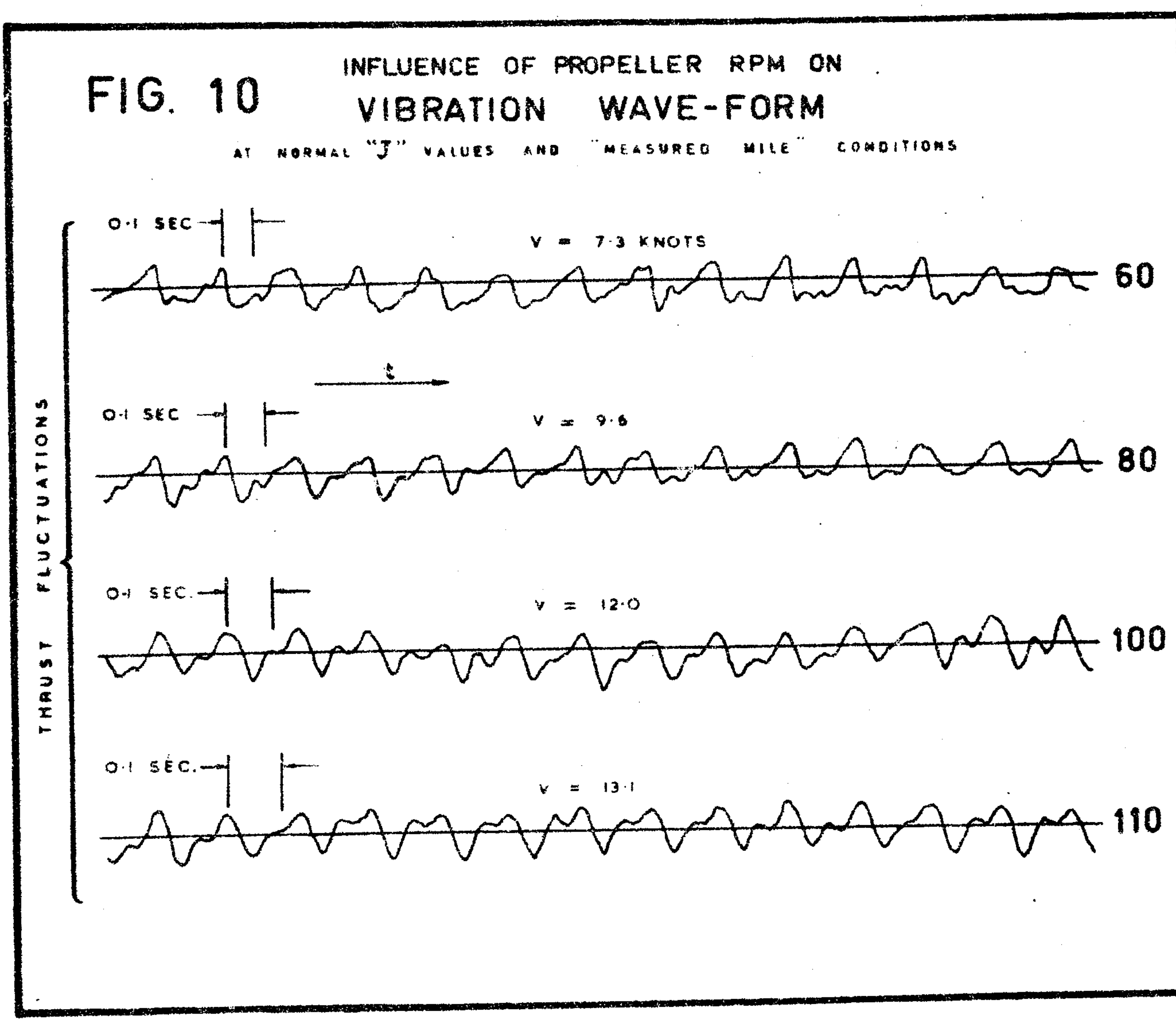


FIG. 10
INFLUENCE OF PROPELLER RPM ON
VIBRATION WAVE-FORM
AT NORMAL "J" VALUES AND "MEASURED MILE" CONDITIONS
THRUST FLUCTUATIONS
O·I SEC
V = 7·3 KNOTS
60
t
O·I SEC
V = 9·6
80
O·I SEC.
V = 12·O
100
O·I SEC.
V = 13·1
110