PROPELLER VIBRATION TESTS ON A 19,000 TON ORE-CARRIER.

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Synopsis:

During the sea trials of a bulk ore-carrier in July, 1959, a propeller vibration log was made. From this information it has been possible to derive the magnitude of thrust and torque fluctuations over a wide range of ship operating conditions.

Introduction:

Propeller-induced vibratory forces may cause local resonant or forced vibrations in various parts of a ship. With the greatly increased power being transmitted through the propellers of some modern ships more care is needed in the design of hulls in the vicinity of the propeller and the propellers themselves if damaging vibrations are to be avoided.

The Causes of Propeller-induced Vibrations:

The origin of propeller-induced vibratory forces lies in the non-uniform loading which the propeller experiences throughout each revolution. This occurs in part as a result of the variation of inflow velocity to the propeller at any particular radius which arises, principally, from the flow inequalities set up by the surface friction of the hull (fig. 1 & 2). The usually small aperture clearances also cause cyclic interferences to the rotating pressure fields of the blades. What has been measured in these tests is essentially the variation of blade surface loading throughout each revolution of the shaft, modified only by the mechanical dynamic magnifiers of the propeller/shaft system (see appendix II).

For the usual low-speed four-bladed single-screw merchant ship, of block coefficient say 0.75, these peak-to-peak variations of thrust load will be of the order of 15%. This means that a ship operating with 50 tons thrust will experience impulsive load variations of the order of 7 tons. The frequency of these pulsations will be that of blade frequency, say 8 c.p.s.

In the torsional direction these same blade load fluctuations will occur at the blade surfaces. However, because a ship usually operates above its first propeller-shaft torsional critical speed (appendix II), the actual torque pulses transmitted to the bull-gear or output gear, are reduced to less than one third of those corresponding to the actual blade surface forces at these speeds, despite the fact that the water-borne forces remain unaltered. At the torsional critical speeds these propeller-shaft torques may be magnified many times.

Two factors which have an important influence on the magnitude of the pulsating forces and torques are the ship's speed in the water and the propeller r.p.m. With higher ship speeds the velocity gradients of the boundary layer become more intense, hence the flow variations which the propeller encounters during each revolution increase correspondingly for a particular propeller r.p.m. This factor may be expressed as a function of "J", where $J = V_s/nD$. The second factor of r.p.m. also produces an impulsive effect. Here the higher circumferential velocities of the propeller blade elements reduce the accommodation time for the cyclic blade-load changes. Thus it is a function of the angular velocity of the blades.

Many other influences prevailing at the stern of a ship cause fluctuations in the mean propeller blade loading. Examples of these are fluid velocity variations due to seawave orbital velocities as they pass the stern; turbulence velocities in the hull's wake; variation of propeller immersion depth which changes the influence of the free surface on the propeller blade loading; attitude of the vessel in the water which changes the depth of propeller immersion in addition to the changes in the direction of the resultant inflow into the propeller caused by the additional components of the hull's velocity; and the rudder angle with its influence on the propeller's wake.

In addition to the foregoing more obvious influences, there are others as yet imperfectly understood which contribute to the resultant propeller blade load fluctuations. One such important influence is strongly evident in the recording of the torsional fluctuations at resonance (fig. 7). The large variations in amplitude shown in the fully developed resonance are characteristic of hydroelastic phenomena which are being investigated at present by the author.

Method of Data Assessment:

In recording the dynamic propeller loads occurring in an actual ship, it is necessary to keep in mind the complex character of the problem and to endeavour to separate out the main contributing phenomena. Therefore to rationalise the assessment of the vibration information from the log, it was decided that only average peak-to-peak amplitudes would be considered. Waveforms, and variations in peak-to-peak amplitude over several seconds, were ignored. Despite this simplification wide variations in vibration amplitudes were recorded, as is shown by the extent of the shaded areas in figs. 5 and 6. Each point marked on the graphs is an observation. Variations of the order of 100% were observed quite regularly over intervals of several seconds but only the average amplitudes were used for these graphs. Points which would have resulted from rudder angles other than "straight-ahead" were omitted. Some points which occurred during transient conditions, for example the acceleration or deceleration of the ship, have been included to show the influence of "J" values on the vibration phenomena. Of special interest in this connection are the torsional amplitudes at resonance under two different kinds of transient conditions (figs. 6 and 7).

Summary of Results:

To summarize the thrust information under steady conditions, a mean line was drawn through the scattered points (the heavy black line in fig.5) and the probable range of peak-to-peak amplitudes was shaded (area between the dashed lines). To show an approximate relation between mean thrust and thrust fluctuations a series of lines representing various percentages of average mean thrust for the particular propeller r.p.m. were drawn.

The data obtained during runs on the measured mile are specially marked. Large variations are still evident despite the otherwise uniform conditions. The peak-to-peak amplitudes of forces at the thrust block are shown approximately by the vertical ordinates, provided that the compliance of the thrust-block can be neglected. The consequence of thrust block compliance is to allow this fluctuating force to oscillate the whole shaft system, including the bull-gear, in an axial direction with the possibility that gear tooth hammer will occur.

The torsional information was similarly treated. In interpreting this graph the role of the dynamic magnifier of the propeller/shaft system, as explained in appendix II, must be fully appreciated.

The peak-to-peak amplitude of the torsional forces at the bull-gear periphery is shown approximately by the vertical ordinates in fig. 7, again assuming negligible gear-case compliance. The negative torque shown in the fully developed resonance would have almost certainly resulted in loss of tooth contact at the main reduction gear. It should be concluded therefore that gear hammer may occur either at the first torsional critical speed or at the first axial critical speed, which may be within the normal operating speed range of the ship.

The Influence of Blade Numbers on Vibration Intensity.

The vibratory forces which have been measured in these tests are the summations of the forces on each of the four blades of the propeller. Fig. 3 shows the result of adding four hypothetical thrusts from a single blade which have been delayed by 0, 90, 180 and 270 degrees respectively, of a revolution. The two more or less equal thrust peaks 180° apart result from symmetry in the wake about the vertical centreline through the hull (fig. 1). It is evident that as the number of blades increases the resultant vibration amplitudes will tend to decrease in a general way. This trend is quite clear in the case of the second and fourth harmonics (fig. 4) but less obvious in the first and third harmonics. With the odd harmonics, that is the first, third, fifth, etc., the hull symmetry combines with the diametrically opposed blades of an even-bladed propeller to give some reinforcement to the load fluctuations of each blade whereas the even harmonics are unaffected by the symmetry. For odd-bladed propellers, the hull symmetry results in mutual cancellation of the single-blade fluctuations and thus gives a much lower total vibration in the whole propeller. It must be appreciated however that it is a summation of all the individual blade forces that is significant and this does not necessarily mean that local water-borne vibration forces will not be troublesome in the immediate vicinity of the propeller itself. Of course a small advantage is gained from more blades in that the thrust per blade is reduced somewhat, and the magnitude of the force fluctuations of each blade is reduced proportionally. Against this is the possibility of encountering a critical frequency in the ship's structure which, by virtue of its dynamic magnifier may produce large-scale vibration amplitudes which would not have occurred had a propeller with a different number of blades been installed.

In general it may be said that, as the frequency of vibration increases the amplitude which may be tolerated decreases. Therefore, it is essent al that, in contemplating a change of propeller blade number, a survey of the ship's structure for possible troublesome resonancesin the new range of propeller blade frequencies should be undertaken.

Propeller-Excited Hull Vibrations:

Corelation between the instantaneous changes of propeller blade loading and hull vibrations may be seen in fig. 9. This is a simultaneous recording of the propeller's thrust and the output of an accelerometer attached to the after-peak tank bulkhead. Because the accelerometer was not critically damped, its response to the shock loading was a short-duration transient. Therefore, at each transient shown, the hull would have experienced a sharp acceleration. These may have been caused by water-borne forces or by forces transmitted by the propeller-shaft through its bearings.

Conclusions:

(a) In this paper it has been shown how the vibratory forces which travel through the propeller-shaft into the body of a ship have been measured. From these forces the instantaneous propeller-blade surface forces have been found. From which the magnitude of the water-borne forces reaching the hull may be estimated.

(b) The predominant frequency of these vibratory forces is that of fundamental blade frequency. It follows that, so far as propeller-induced ship vibration is concerned, an analysis of the ship structure and machinery over the range of frequencies so generated will ensure satisfactory operating conditions. In some circumstances the second and third harmonic may need consideration especially with three-bladed propellers.

(c) The influence of the boundary layer generated by the ship's forward speed in the water has been shown to be a decisive factor in the severity of the first torsional resonance.

(d) The close relationship between mean propeller thrust and vibration excitation forces at normal "J" values has been demonstrated.

(e) High-intensity torsional forces at the propeller-blade sur aces have been shown to exist despite their isolation by the mass/elastic system represented by the propeller/shaft/bull-gear.

APPENDIX I.

University of Adelaide Research on Ship Propulsion.

The Department of Mechanical Engineering under the direction of Professor Davis has developed several research facilities for model tests on hulls and propellers. Total hull resistance is measured in a 100 ft. long rectangular tank using electrical resistance strain gauges in the force-detection elements. Skin friction is measured ina low-velocity wind tunnel using the wake-velocity defect method. This work is directed towards obtaining better prototype predictions from 3 ft. models.

Propeller vibrations are being studied in an open-circuit water tunnel using strain gauges for force measurement with transistorized d.c. preamplifiers on the rotating system (fig.13). The propeller dynamometer for this tunnel was designed by the author (M.E. thesis) and forms the basis for the full-scale tests reported in this paper. Further developments in propeller dynamometry for full-scale ships is in progress.

APPENDIX II.

The Measurement of Propeller Blade-Loading by Tailshaft Stresses.

The ability to measure fluid forces acting on the propeller blade surfaces depends on the following argument. Any static force acting on a mass-elastic system will load all the elements of the elastic system equally. For example, the constant thrust and torque loads on the propeller blades will cause constant axial and torsional stresses in the tailshaft. If the forces are now variable with time, such as the vibratory forces in these tests, the stresses in the elastic system will be modified by the inertia forces of the masses in the system. Depending on the frequency of the force fluctuations so will their magnitude vary in strength and phase. One relationship expressing these variations is:

$$P = \frac{P_o\left[1 + (2 \times \frac{c}{c_c} \times \frac{\omega}{\omega_n})^2\right]^{\frac{1}{2}}}{\left[(1 - \frac{\omega^2}{\omega_n^2})^2 + (2 \times \frac{c}{c_c} \times \frac{\omega}{\omega_n})^2\right]^{\frac{1}{2}}} \quad \ldots\ldots (1)$$

Where P is the force transmitted throughout the elastic system,
P_o and ω the magnitude and frequency of the fluid forces applied to the propeller blade surfaces,
ω_n is the natural frequency of the system,
c and c_c the actual and "critical" damping for the system.

A simple first order mode of vibration with the bull-gear as an infinite mass is assumed in both the torsional and axial directions. In an actual ship the flexibility of the attachment of the thrust-block to the hull in combination with the total mass of the propeller/shaft system, introduces another critical speed at a frequency which is much lower than the axial ω indicated by the foregoing argument. In the case of the ship used in these experiments it occurred at about 111 r.p.m. or 444 c.p.m. (operating speed range 60 to 118 r.p.m.) which could explain why the peak-to-peak amplitudes increase faster than the average mean thrust as this critical-speed is approached. Fortunately it was a low-intensity resonance which means that the assumptions which follow as substantially correct.

If $\omega \ll \omega_n$, as in the case of this ship's thrust forces, the relationship reduces to $P \approx P_o$. It is therefore assumed that the thrust vibration record shows essentially the axial components of the blade-surface forces. This is seen to be a reasonable assumption if the relationship given by den Hartog (ref. 2):

$$\frac{\omega \text{ axial}}{\omega \text{ torsional}} \approx 1.6 \; \frac{\text{Propeller Diam.}}{\text{Shaft Diam.}}$$

is applied to the known torsional natural frequency of 205 c.p.m. for the ship. That is:

$$\omega \text{ axial} \approx 1.6 \times 205 \times \frac{17.5}{1.3} \approx 4,400 \text{ c.p.m.}$$

Since blade frequencies were always less than 450 c.p.m. quite accurate detail up to their fourth harmonic could be expected from measurements of the tailshaft stresses.

It has been shown (ref. 6) that any vibratory force present in the axial direction will have a corresponding force in the torsional direction by virtue of the division of the blade-surface forces into components of torque and thrust. The ratio of their magnitudes will be approximately the same as the ratio of the mean torque and thrust, subject however to the radial distribution of the wake inequalities (see figs. 1, 2 and 11).

On this premise it might be expected that the recordings of shaft torsional stresses would show a similarity in waveform to the axial stresses referred to above. This is not so as is shown in the reproduction of a simultaneous recording of torsional and axial stresses (fig. 10). The much smoother waveform of the torsional stresses may be explained as follows. In equation (1) it is shown that, if ω_n is only half of the blade frequency of 400 c.p.m., the magnitude of the resultant shaft stresses will be reduced by approximately a 3:1 ratio. For the second harmonic the reduction is much greater, and so on. Thus, in this case the higher harmonic content in the torsional stress waveform will be much smaller than in the blade-surface force waveform, which is shown approximately by the axial stresses. Despite the obvious differences these two waveforms contain essentially the same information in regard to blade-surface forces, but in a less tractable form in the torsional case. The phase-shift of 180° occurs as the frequency being considered passes through resonance. Compare fig. 10 with the corresponding records for a 3-bladed propeller attached to the Adelaide University's Water Tunnel Dynamometer (fig. 11). In this case the existence of almost identical blade-surface forces in the torque and thrust directions is clearly shown. In this dynamometer the system natural-frequencies are well above blade frequency hence the actual blade-surface forces are recorded almost without change. The small differences in the torque and thrust waveforms which may be seen are due largely to the residual responses of the recording system to these dynamometer natural frequencies.

A comparison of figs. 5 and 6 will show the way in which the torsional peak-to-peak amplitudes increase as resonance between blade frequency fundamental and torsional natural-frequency is approached, and the way in which attenuation of the torsional stresses proceeds with increasing r.p.m. These effects are best seen by referring to the lines indicating percentages of the average values of mean torque and thrust for the particular propeller r.p.m.

APPENDIX III.

Equipment for measuring the dynamic stresses in the tailshaft.

Electrical resistance strain-gauges in combination with transistorized d.c. pre-amplifiers attached to the rotating shaft were used to detect the instantaneous stress in the tailshaft. The amplifiers and gauges were identical to those which have been used successfully in the propeller dynamometer installed in the University of Adelaide's water tunnel.

The principal barrier to the use of strain gauges for low-value stress measurements on a rotating shaft is the ambient electrical noise-level of the signal-collecting slip-rings. Noise levels of the order of 0.05mV can be expected under the most favourable circumstances. It was anticipated that the strain-gauge signal voltages would range from 0.002mV. Under these circumstances most of the vibration information would have been lost in the background noise. It was therefore necessary to install pre-amplifiers on the tailshaft to lift the signal level to not less than 0.5mV, that is a gain of 250 times. For this purpose two special pre-amplifiers were constructed which had low sensitivity to shock and stray magnetic fields. The latter property was necessary for the avoidance of induced voltages which may have resulted from rotation of these amplifiers on the shaft in the vicinity of large steel masses.

The stresses measured in the shaft were the surface stresses. This was achieved by placing the strain-gauges in intimate contact with the shaft surface with a suitable cement. For the thrust or compression stresses, the gauge wires were placed parallel to the axis and at right-angles to it. With a suitable electrical network, the axial compression stress and the corresponding transverse tension stress expressed by Poisson's ratio were made additive in the output signal. For the shaft torsion stresses, the gauges wires were placed on lines making angles of 45° and 135° to the shaft axis, that is along the lines of the principal stresses. The four gauges used in each circuit were symmetrically distributed around the circumference of the shaft so as to avoid the detection of shaft bending stresses.

APPENDIX IV.

Siemens' Torsionmeter S.H.P. compared with the Vibration Strain-gauge Meter Readings.

To assess the accuracy of mean values of torque obtained during the vibration tests the two independently derived values for shaft-horsepower were compared:

	TIME	R.P.M.	S.H.P. (gauges)	S.H.P. (Siemens)	% Difference
22/7/59	9:35 am.	61	1080	1083	0
	10:39	60.1	1060	1105	-4
	11:26	79.7	2400	2460	-2½
	12:04 pm.	79.9	2430	2530	-4
	12:38	99.9	4790	4900	-2½
	1:14	99.5	4850	4970	-2½
	1:43	110	6560	6650	-1½
	2:13	110	6460	6700	-4
				av.	-2½%
23/7/59	9:32 am.	100.3	5040	4840	+3
	10:05	99	4890	4820	+1½
	11:40	103	5460	5450	0
	12:28 pm.	103.5	5380	5450	-1½
				av.	+¾%

APPENDIX V.

Hydrostatic Particulars of the 19,000 ton Ore-carrier During the Trials.

80% Draught

S.S. 550' BP	Mld. Drt. Ft.	24.00
x 70' B.Mld.	Displt. T.S.W.	20278
	Skin Sq. ft.	51463
LCB % from ⊠		1.66F
(s)		6.46
(m)		6.15
$\frac{1}{2}e^{o}$		30.0
C_b		0.76
C_p		0.78
C_m		0.98
C_w		0.84
CI_t		0.75

Particulars of the Propeller.

Diameter in feet	17.5
Mean face pitch	13.4
Face pitch ratio	0.76
No. of blades	4
Developed area (sq.ft.)	120
Blade area ratio	0.50
Rake (inches)	18.5
Skewback (inches)	19

APPENDIX VI.

Notes on Figure 12.

This analysis was made so as to isolate the major influences which generated the tailshaft stress waveforms in a diesel tug during its acceptance trials. The tug was operating under free-running conditions. The recorded signals of the figure were taken from a simultaneous record of tailshaft thrust and torque stresses.

Two special characteristics appear in these waverorms. One is the engine firing pulsations in torque and the other, the almost continuous and conspicuous excitation of the torsional natural frequency.

As in the case of the 19,000 ton ore-carrier (fig.10) the propeller torque waveform is greatly modified by the dynamic magnifier of the propeller/shaft/bull-gear system.

Considering the engine firing pulses. These were transmitted from the engine flywheel through a 3:1 reduction gear-box to the tailshaft and thence to the propeller. Because of their high frequency and the mass/elastic relationship of the propeller to the tailshaft, the actual motion of the propeller in response to these forces was negligible. This is shown by the thrust recorded signal, which reveals no significant amplitudes at engine firing frequency. The pulses were in fact almost completely absorbed in the elastic deformation of the tailshaft itself. Since the propeller performance, only, is of special interest here, these pulsations should be disregarded. This has been done with the continuous narrow line of the lower analysis. It must be stressed, however, that if resonance between engine pulsations and any torsional natural frequency of the propeller occurs, the dynamic magnifier of the exciting forces from the engine could result in undesirable vibration amplitudes at the propeller.

Consider now the excitation of the torsional natural frequency. By drawing a mean curve through the resultant waveform so as to remove the blade frequency components, oscillations at the torsional natural frequency are revealed (dotted line). The waxing and waning of the peak-to-peak amplitudes at this frequency bear a close resemblance to the effect observed in fig. 7, which has been referred to previously. Excitation of the torsional natural frequency implies that cyclic blade load changes occurring at the propeller contain components of this frequency. Since there is no obvious cause for this it is concluded, that the intensity of the blade load changes as each blade passes the stern post is such, that a series of short-duration transients is generated which contain components at the torsional natural frequency, in significant amounts. Thus excitation occurs.

The higher than usual vibration of a free-running tug may be explained. It has been shown elsewhere in this paper that the propeller vibration excitation of a given ship increases with increasing "J", where $J = v_a/nD$. With a tug the "J" value for the free-running condition would be much higher than for an ordinary ship, hence a higher vibration excitation exists. Secondly, the deep-skeg stern of this vessel would have narrowed the boundary-layer velocity gradients about the vertical centre-line of hull, and so increasing the rapidity of blade load changes.

In the thrust record the removal of all upper frequencies, as shown by the thin line, reveals the regular character of the blade frequency pulsations. These are superposed on to a torsional natural frequency pulsation which corresponds to this component in the torque record. The upper frequencies which have been removed comprise mainly the second and third harmonics of blade frequency together with a series of random transients of much higher frequency.

The heavy black lines in the diagram represent the mean propeller loading, when all the components discussed above are removed.

APPENDIX VII.

Notes on fig. 6. (Torsional Vibration Intensity vs Propeller r.p.m.).

Considering curve (1): Commencing at 40 r.p.m. the propeller was rapidly accelerated to 83 r.p.m. over a period of 256 seconds. During this time the ship was almost at rest. Local circulations of water about the stern would have occurred, so causing small-scale non-uniformities in the inflow velocity into the propeller. No normal boundary layer would have existed. There would have been interference with the pressure field of the propeller by the hull and appendages. These combined effects would have resulted in forcing frequency amplitudes considerably below the normal values for these shaft speeds. A comparison with the vibration amplitudes for the measured mile conditions (heavy solid line), whose dynamic magnifiers for the various speeds are almost identical, shows this to be true.

A Comparison of the above conditions and those recorded in fig. 7:

The maximum peak-to-peak vibration intensity at the torsional critical speed for a rapid acceleration of the shaft with the ship at rest was approximately 7.4 tons-feet whereas, for conditions such as those which arose due to the emergency closure of the main steam valve, a maximum value of 57 tons-feet was recorded. This is an increase due to the ship's forward velocity in the water of 7.7 times.

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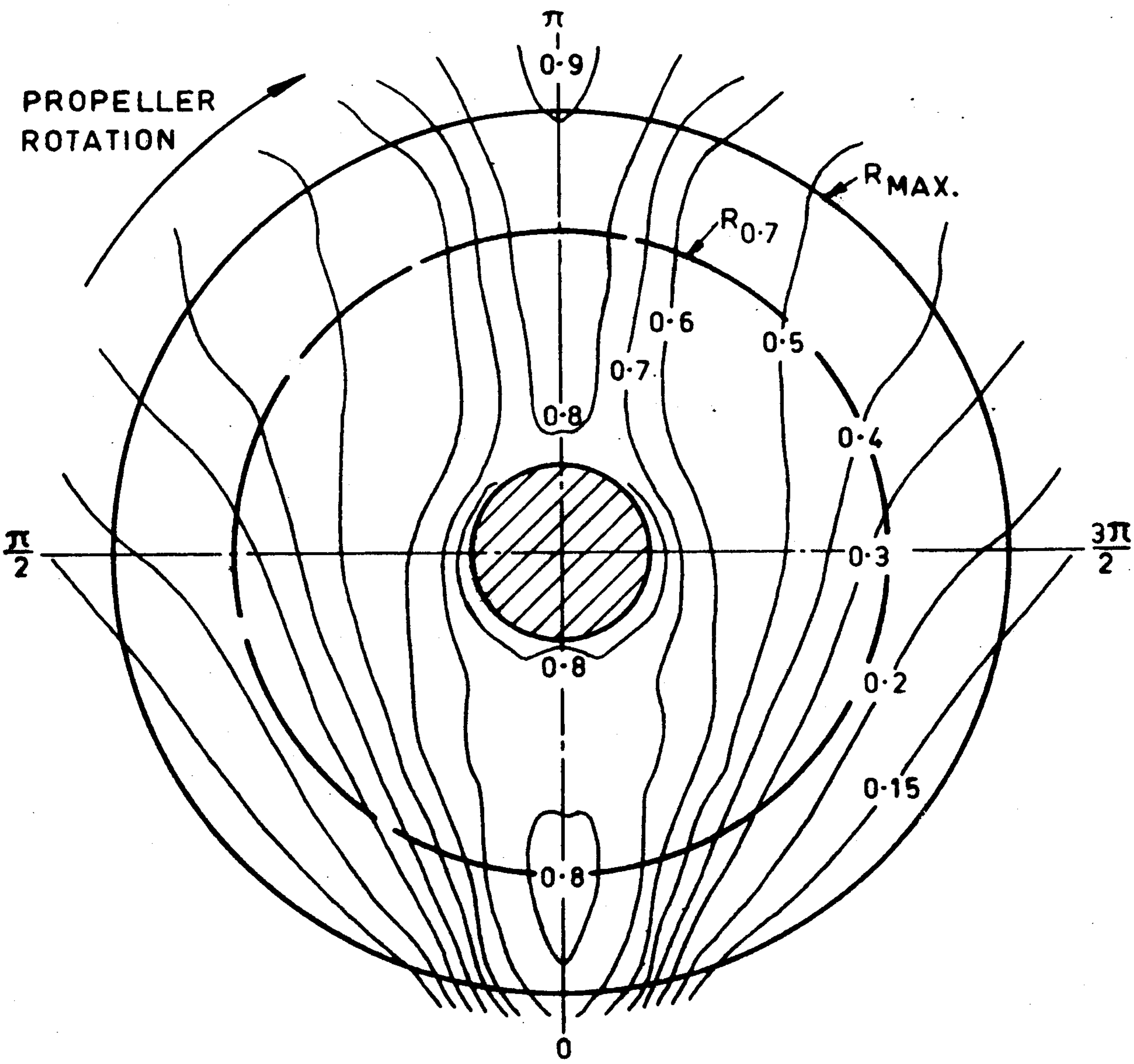
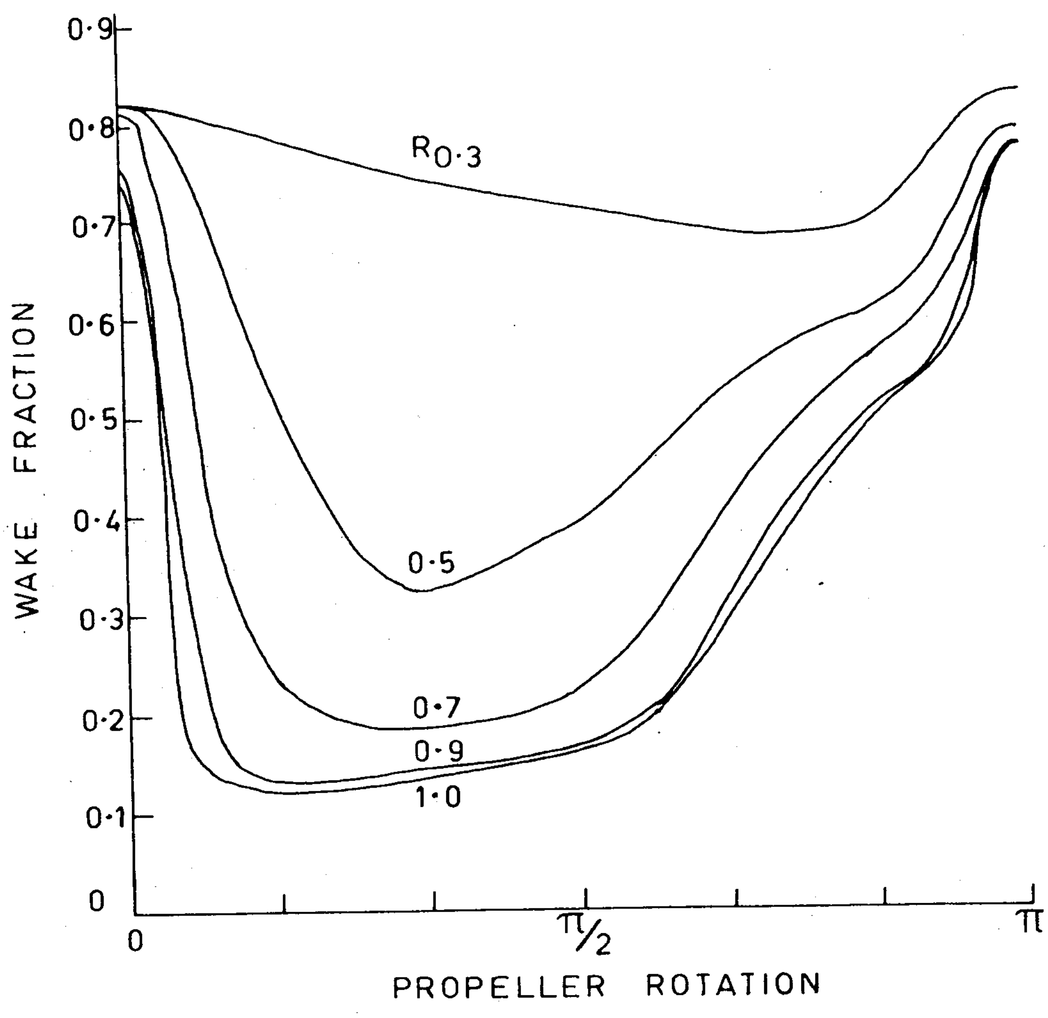
7.	Bunyan	Practical Approach to Some Vibration and Machinery Problems in Ships. INA Trans. v.97, n.4 (1955) p.J41.

FIG. 1



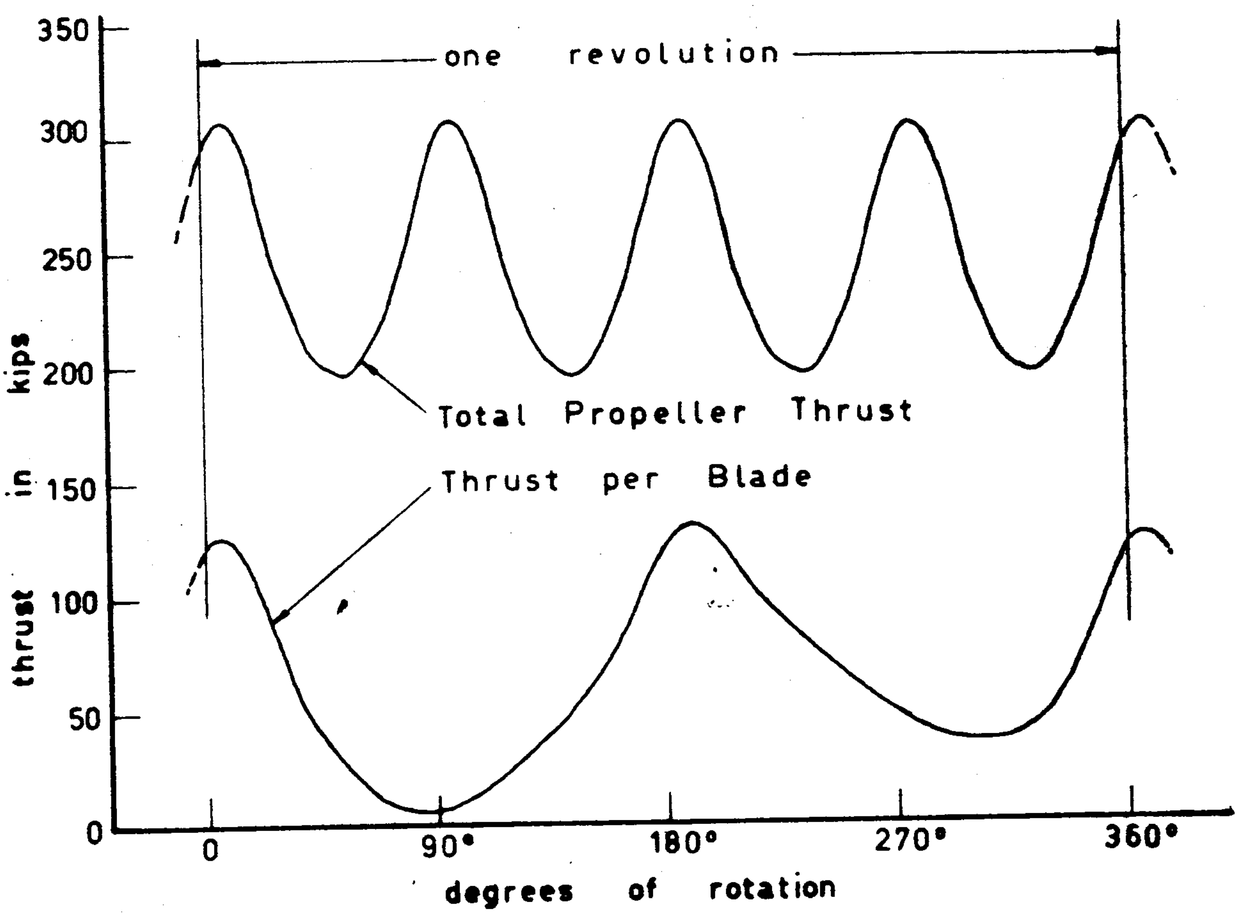
VARIATIONS OF WAKE FRACTION FOR A SINGLE-SCREW MERCHANT SHIP [FROM MODEL DATA WITHOUT PROPELLER]

FIG. 2



VARIATIONS OF IN-FLOW VELOCITY EXPERIENCED BY THE PROPELLER-BLADE ELEMENTS AT DIFFERENT RADII.

FIG. 3



THEORETICAL THRUST FLUCTUATIONS FOR A FOUR-BLADED PROPELLER. [based on ref. 3]

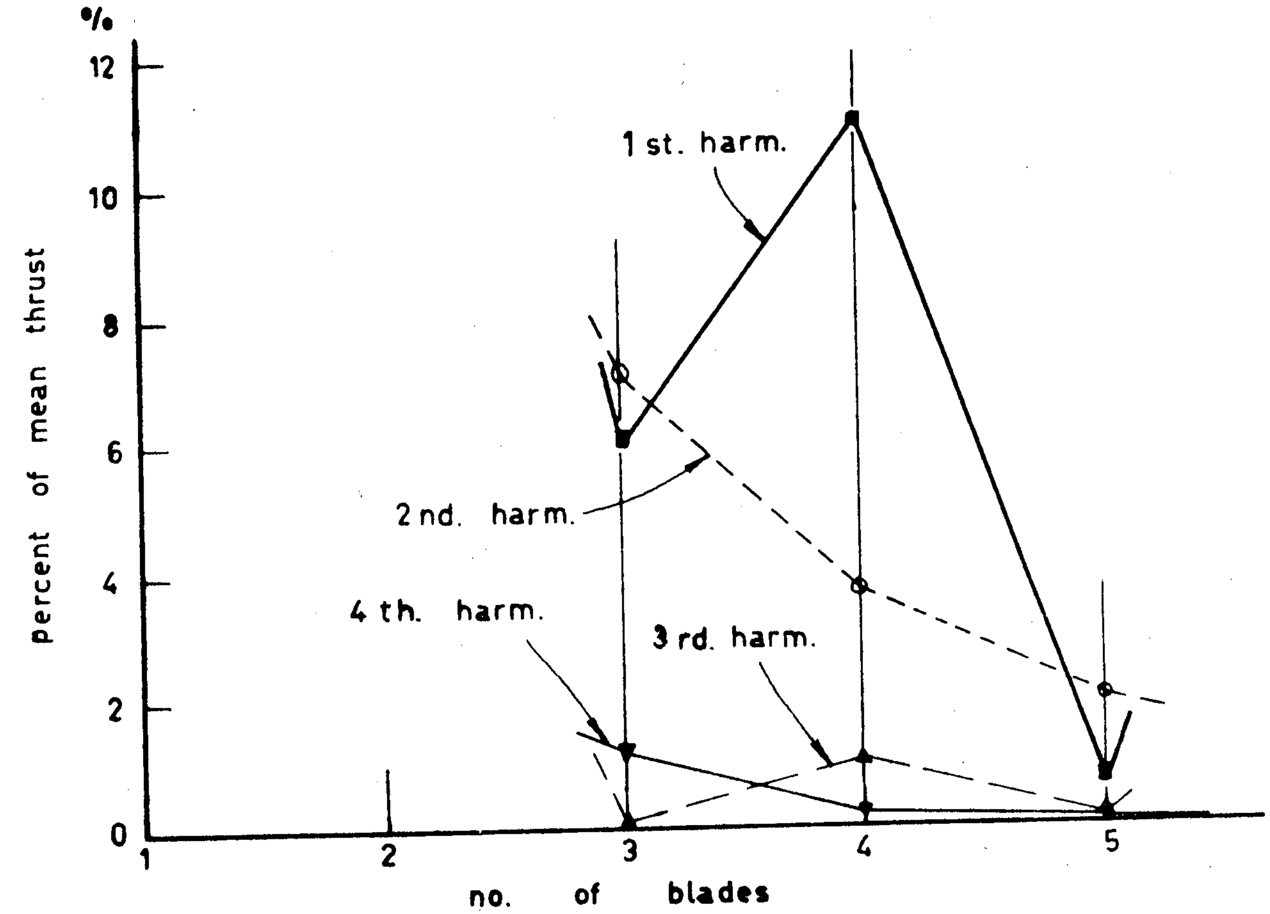


FIG. 4 THEORETICAL VALUES OF VIBRATION (PERCENTAGES OF MEAN THRUST) (based on ref. 1)

FIG. 5

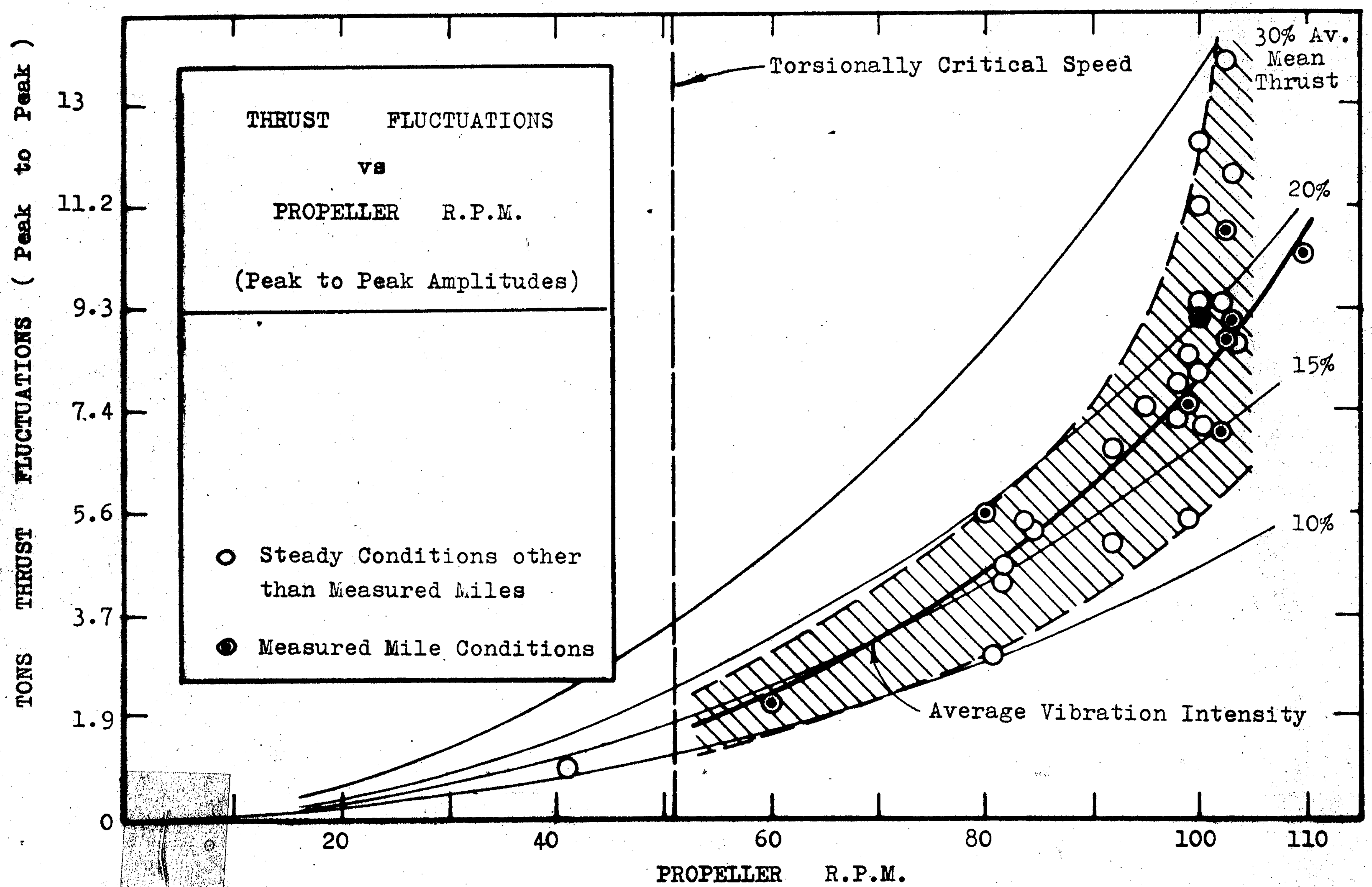
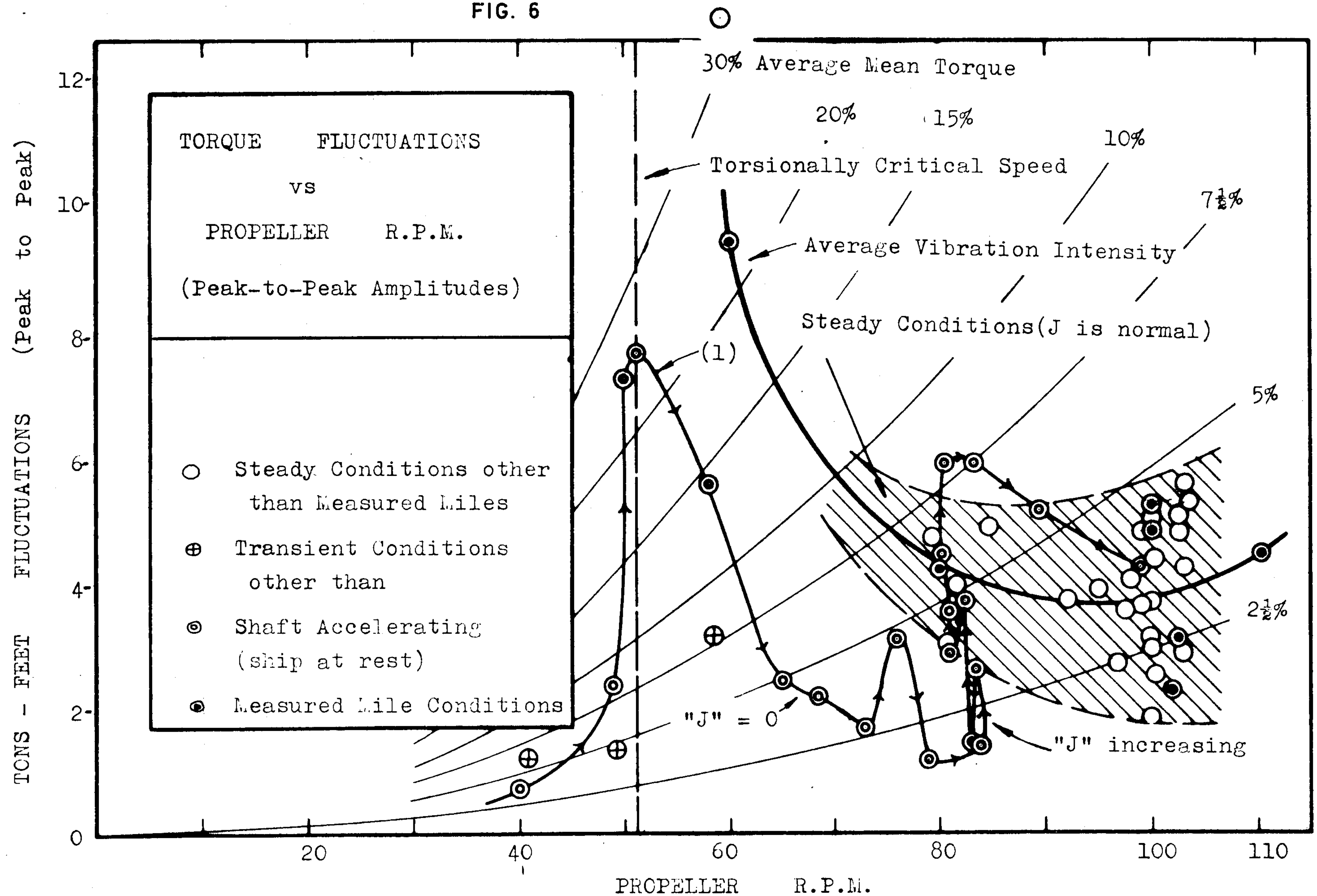
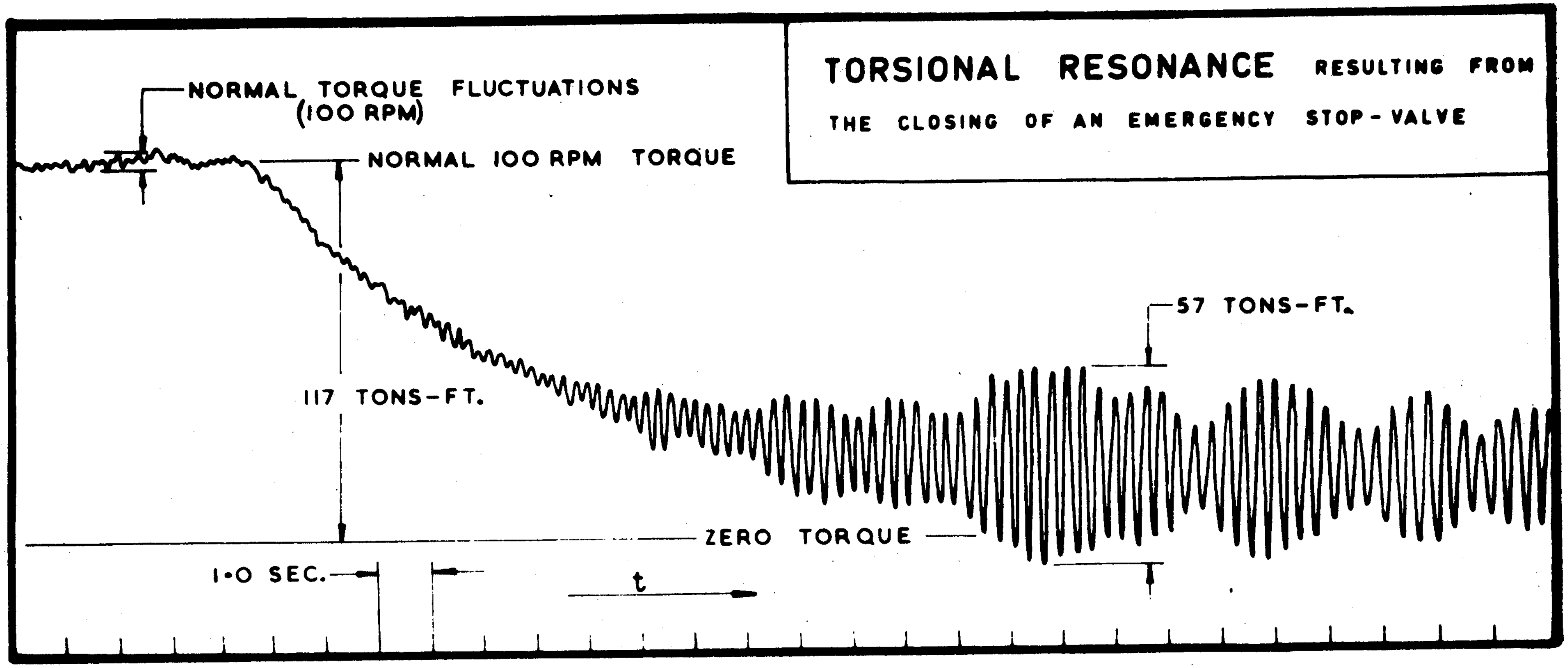

THRUST FLUCTUATIONS
vs
PROPELLER R.P.M.
(Peak to Peak Amplitudes)
Steady Conditions other than Measured Miles
Measured Mile Conditions
Torsionally Critical Speed
30% Av. Mean Thrust
20%
15%
10%
Average Vibration Intensity
TONS THRUST FLUCTUATIONS (Peak to Peak)
13
11.2
9.3
7.4
5.6
3.7
1.9
0
20
40
60
80
100
110
PROPELLER R.P.M.

FIG. 6



TORQUE FLUCTUATIONS
vs
PROPELLER R.P.M.
(Peak-to-Peak Amplitudes)
Steady Conditions other than Measured Miles
Transient Conditions other than
Shaft Accelerating (ship at rest)
Measured Mile Conditions
30% Average Mean Torque
20%
15%
10%
7½%
5%
2½%
Torsionally Critical Speed
Average Vibration Intensity
Steady Conditions(J is normal)
(1)
"J" = 0
"J" increasing
TONS - FEET FLUCTUATIONS (Peak to Peak)
PROPELLER R.P.M.
0
2
4
6
8
10
12
20
40
60
80
100
110



This d.c. recording illustrates the consequence of allowing the shaft speed to fall suddenly to a value close to a major critical speed. The ship's forward velocity would have created a Boundary Layer appropriate to 100rpm which would have resulted in forcing frequency amplitudes such as to cause "normal torque fluctuations" as shown. Lowering of the blade frequency and hence the blade velocity should have reduced the magnitude of the forcing pulsations however the greatly increased value of the dynamic magnifier for the system at these frequencies resulted in torque fluctuations of much greater magnitude.

(see appendix II)

FIG. 7

FIG. 8

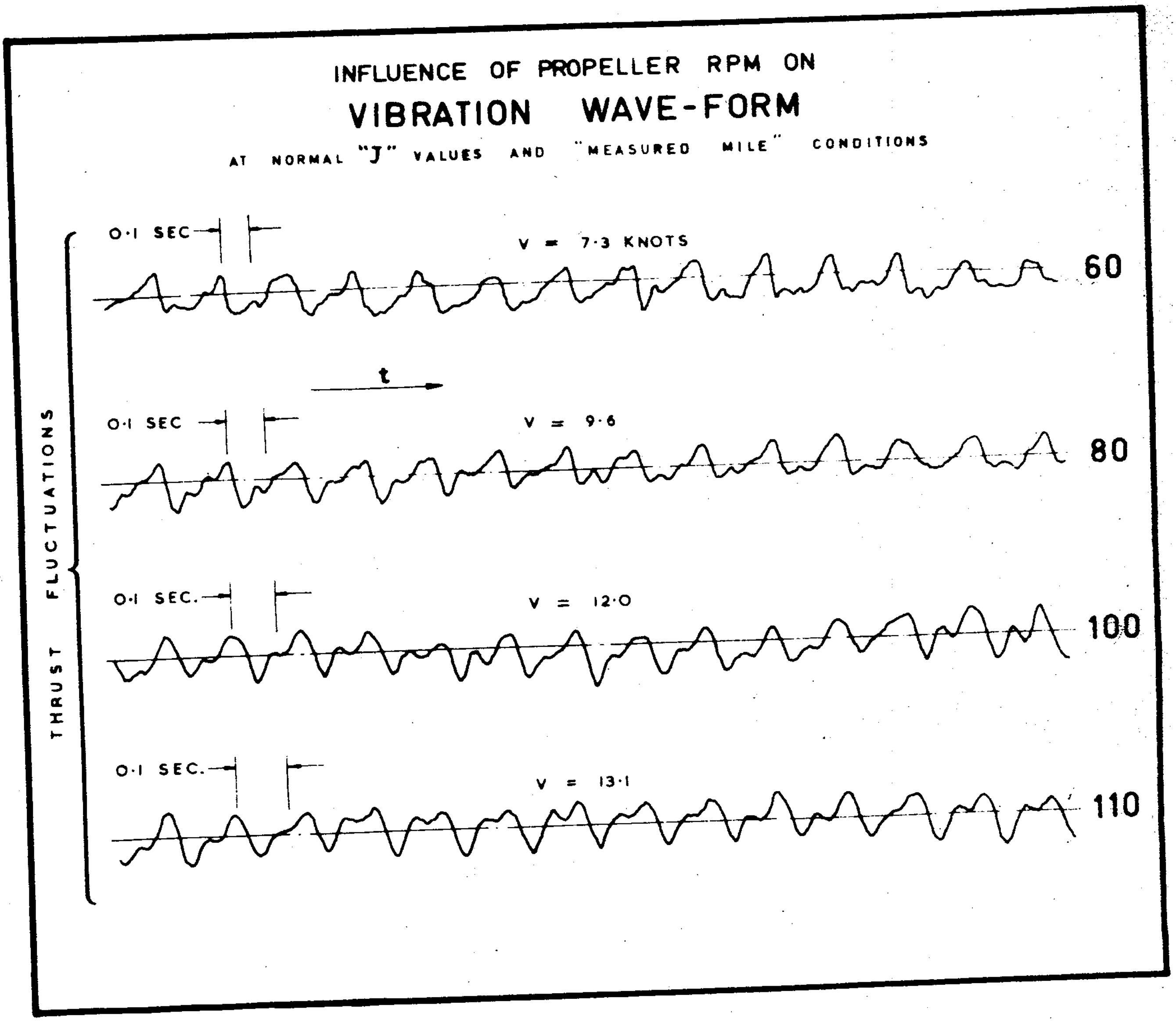
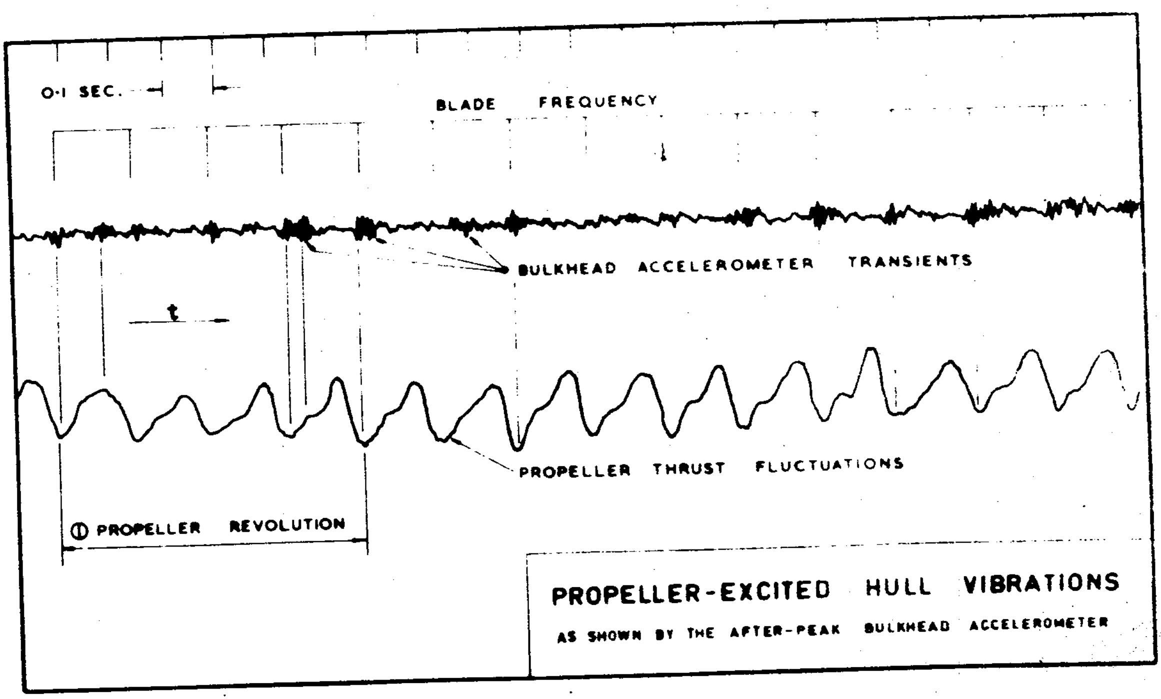

INFLUENCE OF PROPELLER RPM ON
VIBRATION WAVE-FORM
AT NORMAL "J" VALUES AND "MEASURED MILE" CONDITIONS
THRUST FLUCTUATIONS
0·1 SEC
V = 7·3 KNOTS
60
t
0·1 SEC
V = 9·6
80
0·1 SEC.
V = 12·0
100
0·1 SEC.
V = 13·1
110

FIG. 9


0·1 SEC.
BLADE FREQUENCY
BULKHEAD ACCELEROMETER TRANSIENTS
t
PROPELLER THRUST FLUCTUATIONS
① PROPELLER REVOLUTION
PROPELLER-EXCITED HULL VIBRATIONS
AS SHOWN BY THE AFTER-PEAK BULKHEAD ACCELEROMETER

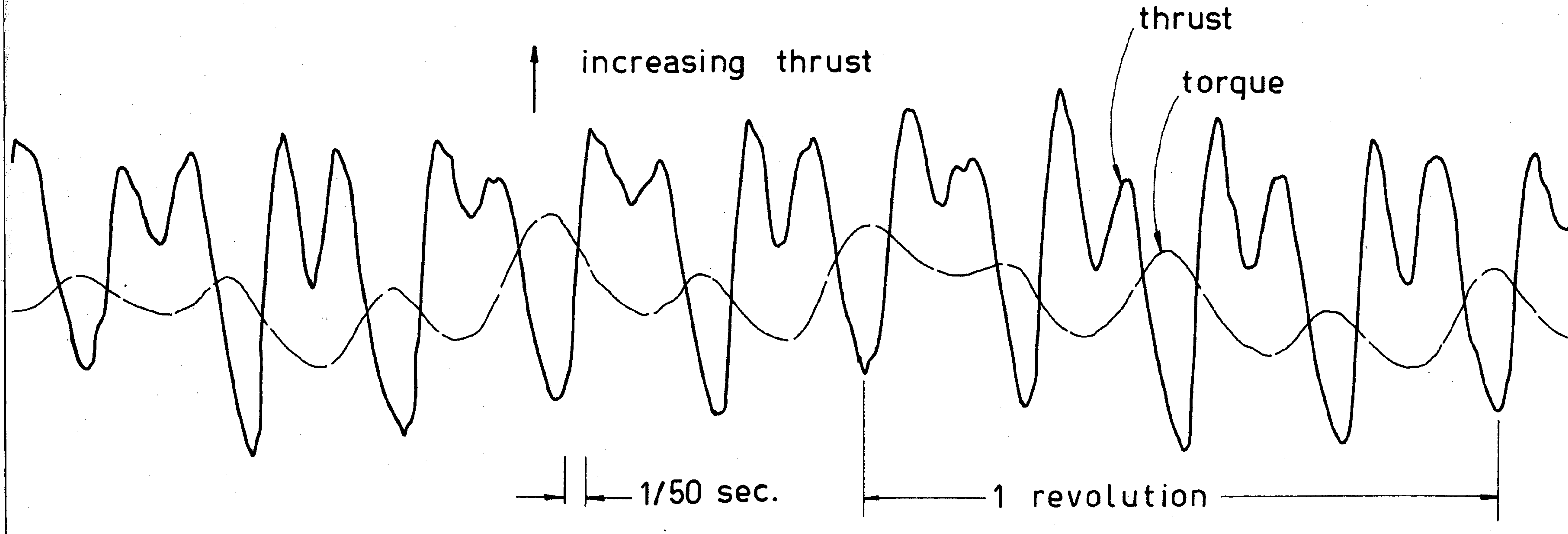
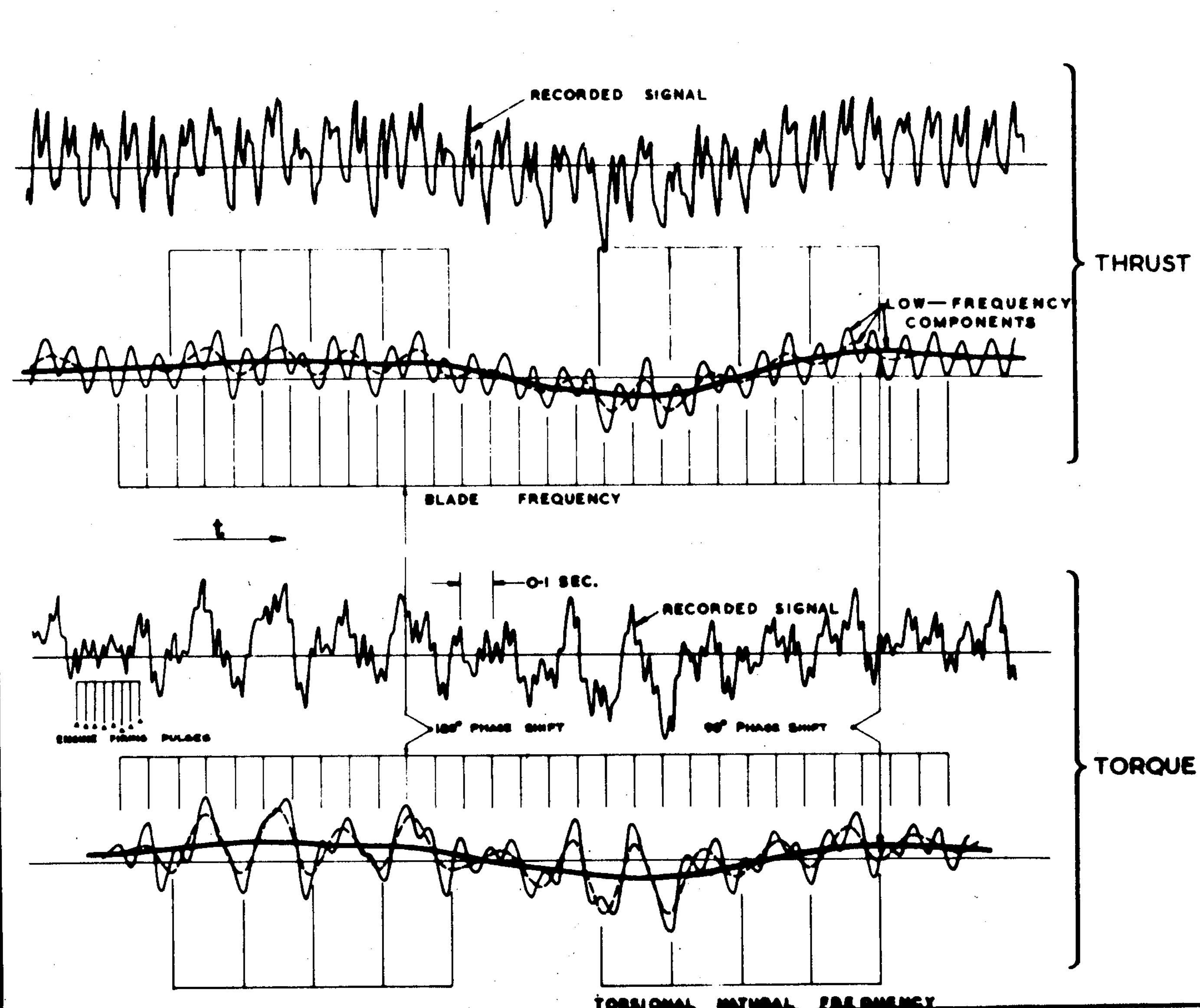


FIG. 10

A VIBRATION LOG OF A 4-BLADED PROPELLER INSTALLED IN A 19,000 T. ORE-CARRIER SHOWING THRUST AND TORQUE VARIATIONS THROUGHOUT EACH REVOLUTION [a random sample]



A PRELIMINARY ANALYSIS OF TORSIONAL AND AXIAL TAILSHAFT STRESS VIBRATIONS IN A 1,100 S.H.P. DIESEL TUG FITTED WITH A REDUCTION GEARBOX SHOWING THE PRINCIPAL VIBRATION COMPONENTS.

FIG. 12

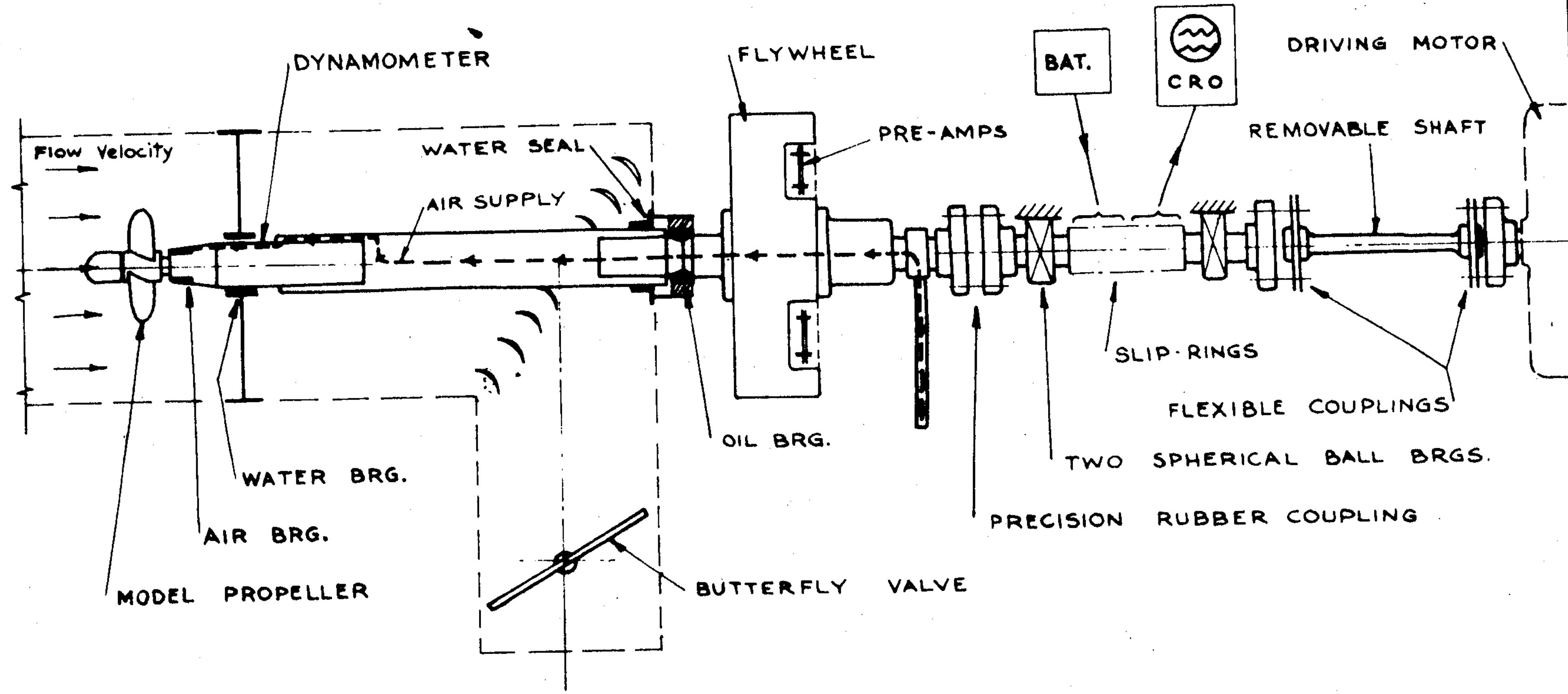


FIG. 13 SCHEMATIC LAY-OUT OF THE PROPELLER DYNAMOMETER

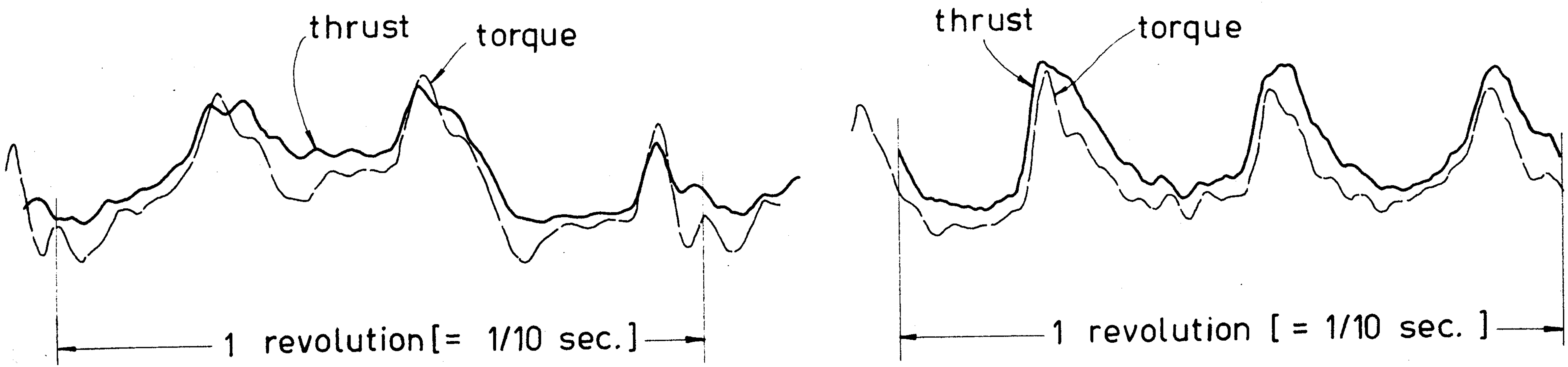


FIG. 11

VIBRATION RECORDINGS OF A 3-BLADED MODEL PROPELLER SHOWING FLUCTUATIONS IN BLADE - SURFACE LOADING EXPRESSED AS FUNCTIONS OF THRUST AND TORQUE (from the Adelaide University's Water Tunnel Dynamometer)