Machinery Induced Vibrations

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The first part of the paper is concerned with main hull vibration. The nature of ship vibration is discussed and a brief review is given of some current methods of predicting critical frequencies and amplitudes for types of vibration caused by unbalanced forces in main machinery. The problem of propeller excited vibration is discussed with particular reference to the factors influencing the forces and hull response characteristics. Some consideration is given to design formulations for propeller-hull clearances.

In the second part consideration is given to the various modes of vibration of propulsion shafting systems, particular attention being given to axial vibration and the estimation of crankshaft longitudinal stiffness. The transverse vibration of oil engines on their seatings is discussed with some observations on the bracing of engines to one another or to the hull. The paper concludes with some examples of suggested tolerable unbalance in engines for typical contemporary ship types.

INTRODUCTION

It can be stated with certainty that vibration in some form will exist in all ships and that it can be recorded by moderately sensitive instruments even when imperceptible to human beings.

In most ships vibration can be felt but is not unpleasant or troublesome, but there are also many in which the vibration exists at levels which give rise to acute personal discomfort and local structural damage, and which interfere with the functioning of ships' instruments. The propulsive system may also be affected in a number of ways.

Vibration may manifest itself in many different forms and the subject generally calls for continued investigation and research of the type which is being carried out by the British Ship Research Association and, indeed, by many other organizations and individuals. The object of such research is to provide naval architects and marine engineers with information which will enable them to design with the reasonable assurance that the level of vibration will not exceed certain acceptable limits.

Rather than attempting to survey the whole field of interest, it has been thought appropriate to confine the contents of this paper to several of the more important problems. The first part deals exclusively with vibration of the main hull structure with special reference to the propeller as the source of vibration. The second part covers several of the more important types of vibration encountered in the propulsion systems of ships.

It would be appropriate to mention that when the authors were invited to present a paper it was suggested that many readers would appreciate some recapitulation of the elements of the subject and an outline of certain current approximate methods of predicting hull and machinery vibration characteristics. This suggestion has been borne in mind in drafting the paper which also contains comment on some of the less well understood aspects of the subject.

PART I

MAIN HULL VIBRATION

The Nature of Ship Vibration

The general characteristics of main hull vibration are considered first. This is followed by a brief description of the manner in which information obtained by measurements on ships may be interpreted to indicate solutions to immediate problems and provide material for future designs.

It will be supposed that records of ship vibration are obtained by means of a simple type of vibrograph in which the vibrations are registered as permanent traces on a chart or film moving at constant speed. Also included on the chart will be a time-base provided by impulses at discrete intervals from a chronometer. The revolutions of the propeller shaft or main engines will also be marked on the chart using a simple electrical circuit in conjunction with a "make and break" contact on the shaft.

If such an instrument is placed, say, at one end of a ship, set to register in any one plane, and the engine revolutions gradually increased from zero to the maximum speed, the resulting record will usually be characterized by a succession of undulations which vary in amplitude and frequency throughout the entire speed range. A simple type of record will normally include several sections in which the amplitudes build up to maximum values and then diminish with increasing revolutions. Fig. 1 illustrates diagrammatically the essential contents of such a record.

If now this record is analysed by measuring the frequency

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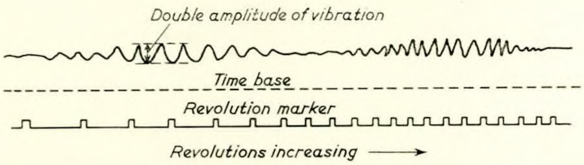


FIG. 1—*Vibrograph record*

of the undulations over short lengths, and plotting these against corresponding amplitudes, the resulting graph will be of the general form indicated in Fig. 2. In this figure the various peaks correspond to conditions known as resonance, and the frequencies occurring at the point of maximum amplitude are known as critical frequencies.

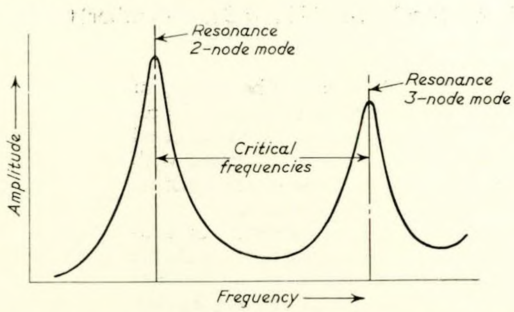


FIG. 2—*Amplitude—Frequency characteristics*

It will now be assumed that the revolutions are kept steady at speeds corresponding to each of the critical frequencies in turn, and that measurements of the vibration amplitudes are made at many points along the hull. If the amplitudes are now plotted at their respective positions on a base corresponding to the ship length, it will be found that the curves joining the spots have characteristic shapes referred to as the mode forms. The general appearance of these curves is indicated in Fig. 3.

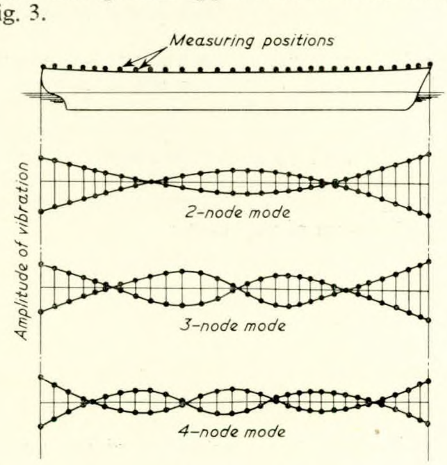


FIG. 3—*Hull mode forms*

The modes are identified by the number of positions of zero deflexion, i.e. two-node, three-node, four-node and so on. The hull may vibrate in these modes in the vertical plane and the horizontal plane. Torsional vibration of the hull can also occur, although the mode forms will be a little different.

The final stage in the analysis of the ship record is to determine the engine or propeller shaft revolutions at which the resonant conditions occur. This will give a direct indication of the source of vibration, the principal causes of which are unbalanced primary or secondary forces in the main machinery or forces from the propeller at a frequency equal to the r.p.m. times the number of blades. The origin of these propeller forces will be referred to in greater detail in a subsequent section. The principal forces acting throughout the speed range, together with the corresponding frequencies are shown diagrammatically in Fig. 4. Wherever the radial force lines intersect the horizontal lines which define the critical hull frequencies, resonant conditions of vibration will be indicated.

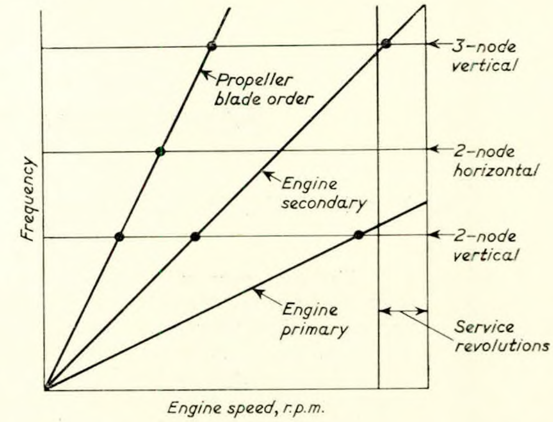


FIG. 4—*Critical frequencies and sources of excitation*

It will be evident that there are many possibilities for the occurrence of resonant conditions. It should be pointed out, however, that a resonant vibration is not necessarily objectionable and only causes concern when it occurs within the service ranges of revolutions and when the amplitudes exceed certain values.

The amplitude-frequency characteristics, amplitude-r.p.m. characteristics together with the vibration profiles, represent all the essential data required for the analysis of any particular case of hull vibration. If vibration exists at such a level that remedial measures are required, there are several courses of action which might be considered. The first of these is to reduce the magnitude of the exciting force or to neutralize it in some convenient way. The second is to alter the frequency of the exciting force such that resonant conditions can be avoided. In some structures it is possible to alter the frequency at which resonance occurs by increasing their stiffnesses. This, however, is not generally considered to be a practical proposition in the case of a ship's hull since a large quantity of additional material is required to produce a relatively small change in the resonant frequency. There are a variety of different ways in which the first two methods may be applied and the ultimate choice will depend upon a number of practical considerations not the least of which is that any change must not incur troubles of a different sort.

Some practical illustrations of the nature of ship vibration are given in Figs. 5 and 6.

Fig. 5 shows four resonance curves measured on a trawler of 185ft. length b.p. These results were obtained by artificially exciting the vessel with a vibration generator. The vibration generator is a conventional research tool and enables hull vibration characteristics to be defined with an accuracy which cannot normally be attained under service conditions. It consists of several masses which are made to rotate about a common shaft. By controlling the eccentricity of the masses and their relative phasing, it is possible to produce alternating forces in any chosen direction. The particular machine used on the trawler is capable of producing a pulsating force of $\pm 3{\cdot}0$ tons at a maximum speed of 600 r.p.m.

Fig. 6 shows the main hull critical frequencies and sources of excitation as measured on a 407-ft. ore-carrier. This ship was propelled by two five-cylinder Diesel engines, there being a gear reduction of 2·25 : 1 between the engines and the propeller. Of the critical frequencies indicated only that of the 3-node vertical mode gave cause for concern and it will be

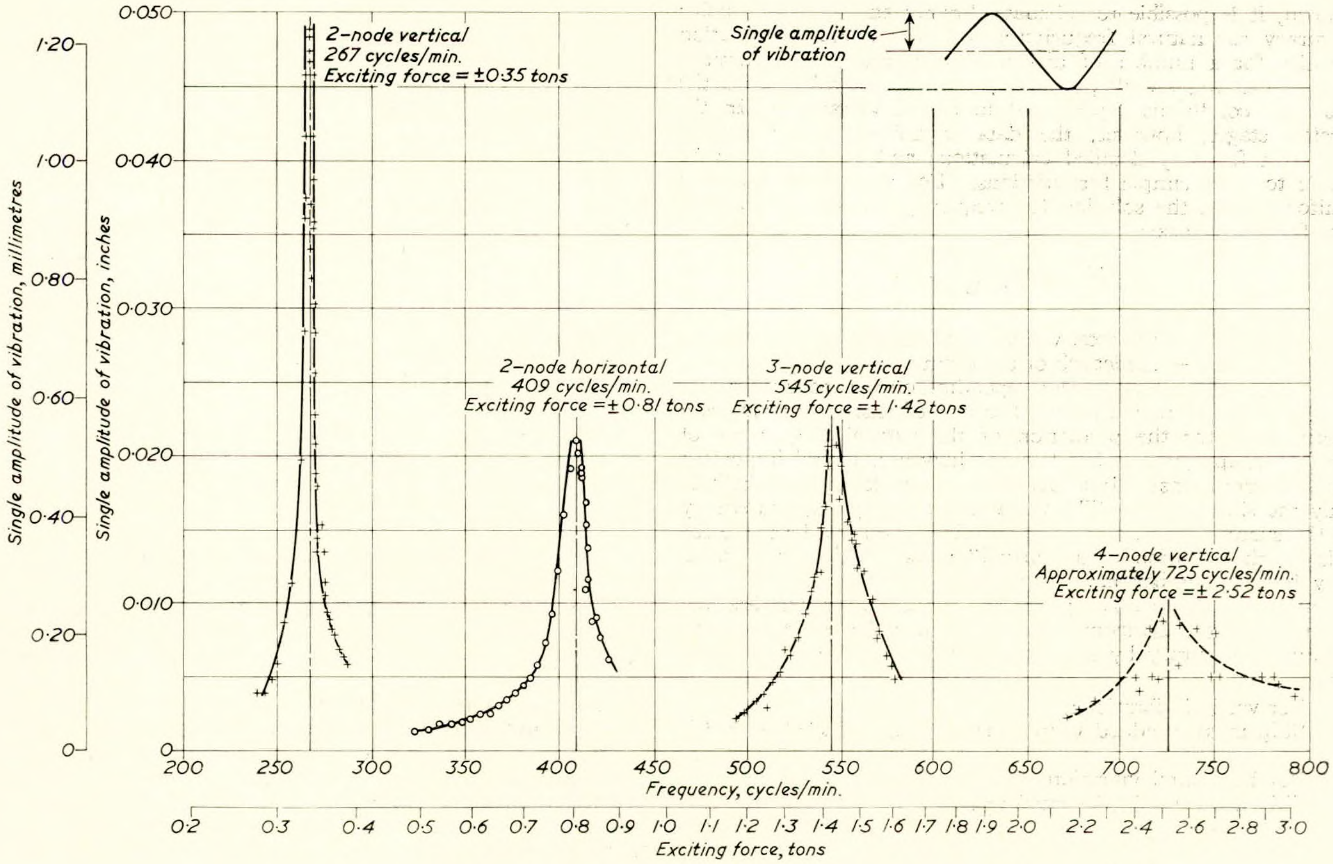


FIG. 5—*Amplitude—Frequency characteristics (185ft. trawler)*

seen that this occurred within the normal range of service revolutions. Although in this instance the amplitudes of the vibration in the hull were not excessive, they gave rise to appreciable fore and aft movements in the upper parts of the bridge structure which was situated aft. In this region the change of slope of the hull girder throughout a vibration cycle is near a maximum value and consequently, a rocking motion in the deckhouse can ensue, the amplitudes increasing with height from the deck. The figure demonstrates generally the propensity of a ship to vibrate in its natural modes.

Generally speaking, engine excited vibration is concerned with the lower modes of hull vibration and the forces at play can be assessed with a fairly high degree of accuracy. With regard to propeller excited vibration, the frequencies involved are higher and the mode forms correspondingly more complex. The exciting forces are also complex in nature and cannot be determined at present with a practical degree of accuracy.

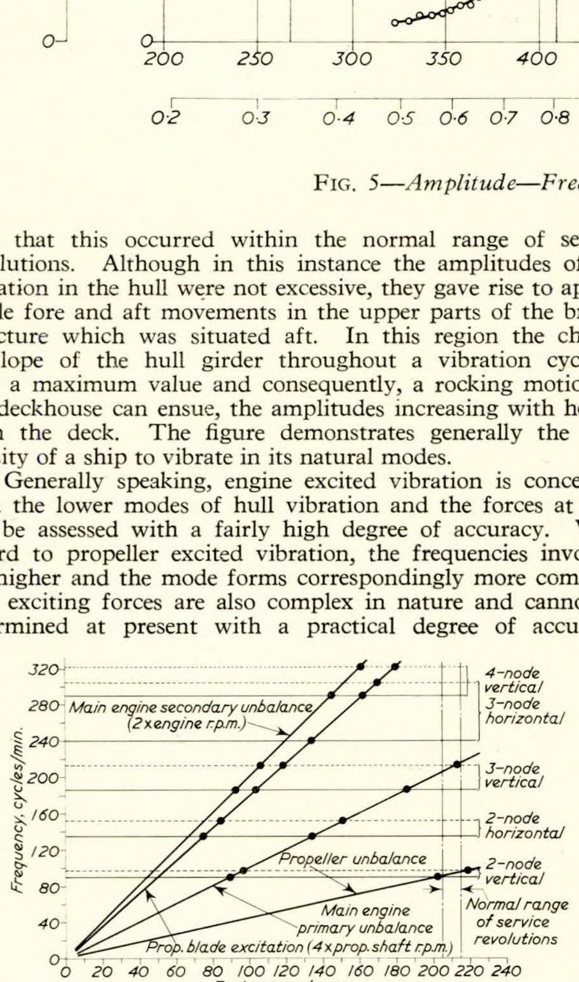


FIG. 6—*Main hull frequencies and sources of excitation*

For these reasons it is appropriate to consider these two types of vibration as separate problems requiring somewhat different treatment on the lines indicated in the following sections.

The Prediction of Ship Vibration—Elementary Modes

In order to avoid troublesome vibration the ship designer is confronted with several separate problems. The first of these is to determine whether resonant conditions are likely to occur near the service speed. If estimates show that such conditions are unlikely to occur, the designer may be reasonably sure that the vibration will be insignificant. If, on the other hand, resonant conditions are indicated it is then necessary to determine the corresponding amplitudes and to consider whether these are acceptable. These three prime considerations i.e. "critical frequencies", "amplitudes" and "acceptability" are dealt with under separate headings below:

a) *Critical Frequencies*

It will have been noted from the previous section that a ship appears to behave essentially as a beam, albeit a rather complicated beam. The mode forms recorded are in fact found to be generally similar to those derived mathematically for the natural oscillations of simple beams having no restraints, a condition commonly referred to as the "free-free" condition.

The basic equation for the vibration of a beam or ship, assuming simple beam theory is:

$$\text{Dynamic load/unit length} = EI\frac{\partial^4 y}{\partial x^4} = \frac{w}{g}\frac{\partial^2 y}{\partial t^2} \quad (1)$$

where I = the section moment of inertia
E = Young's Modulus
w = the distributed weight per unit length
y = the amplitude of vibration at any position x.

This equation yields an infinite number of solutions, and if the precise distributions of weight and section inertia are known, it is possible to calculate directly and with reasonable accuracy the natural frequencies and corresponding vibration profiles for a number of modes of vibration. These natural frequencies are, for all practical purposes, coincident with the resonant conditions experienced in forced vibrations. In the design stages, however, the data available are not usually sufficient for very detailed calculations and recourse must be made to more simple formulations. For the simple case of a uniform beam the solution for frequency can be expressed in the following form:

$$N = C\sqrt{\frac{I}{WL^3}} \qquad (2)$$

where:

W = the total weight of the beam
L = the length of the beam
C = a coefficient depending on the mode form.

It is found that a simple formula of this type provides a useful basis for the prediction of the natural frequencies of ships. In applying such a formula however, several important considerations arise. It is insufficient, for example, to include only the ship masses. The vibration of a ship imparts energy to the surrounding water and due allowance must be made for this by the addition of a "virtual" mass. This virtual mass may be of a similar magnitude to that of the ship itself for vibration in the vertical plane but is usually much less for vibration in the horizontal plane. It may be estimated with reasonable accuracy by approximate formulæ of the following type:

For vertical vibration:
Ship mass + added virtual mass $= \Delta_1 = \Delta(1{\cdot}2 + B/3d)$ (3)

For horizontal vibration
Ship mass + added virtual mass $= \Delta_1 = \Delta + 0{\cdot}016\ Ld^2$ (4)

where:

Δ = the displacement (tons)
B = the beam of the ship (ft.)
d = the mean draught (ft.)
L = the length of the ship (ft.)

It should be mentioned that these formulæ apply only when the ship is in reasonably deep water, i.e. a depth of at least five times the draught. In shallow water the added virtual mass may be very much greater and the hull frequencies correspondingly reduced.

The estimate of the correct section inertia to use often presents a problem, particularly for ships which have extensive deckhouses. The usual practice is to tabulate all the items of continuous longitudinal material together with their areas and levers about an assumed neutral axis (the centre line plane in the case of horizontal vibration). The true neutral axis is determined from the summation of the first area moments and the section inertia from the summation of the second moments corrected to this axis. For corrections which may have to be made for the presence of deckhouses, the reader is referred to paper[1] in the list of references.

For application to ships, therefore, equation (2) may be re-written as follows:

$$\text{Frequency } N = \text{Coefficient} \times \sqrt{\frac{I}{\Delta_1 L^3}} \qquad (5)$$

where:

N = cycles per minute
I = the midship section inertia (in.^{2} ft.^{2})
Δ_1 = the total virtual mass (tons)
L = the length of the ship (ft.)

In a simple formulation of this type the coefficients must embody many separate factors which may vary from ship to ship, e.g. the effects of longitudinal distribution of mass and section inertia. In order to make predictions of frequency with a practical degree of accuracy it is therefore necessary to have recourse to empirical data. Table I gives some typical data obtained from measurements made on ships by the B.S.R.A., together with the principal ship particulars and the relevant coefficients for various modes of vibration. From accumulated data of this type it is usually possible to select the frequency coefficient appropriate to a new design for use in equation (5).

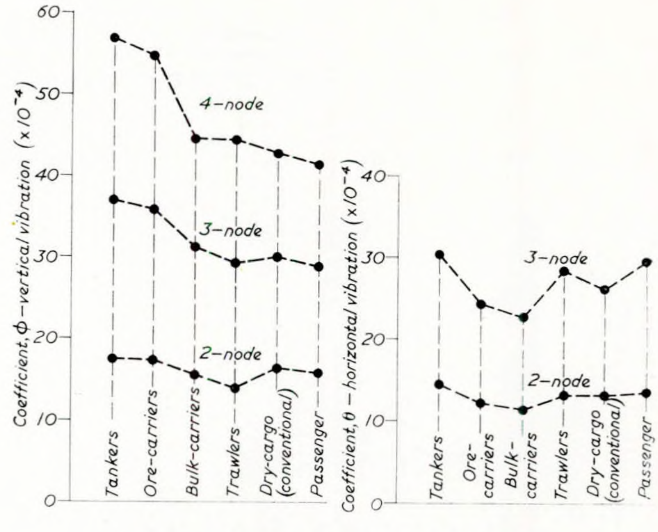


FIG. 7—*Average frequency coefficients*

Average values of the frequency coefficients for each group of ships indicated in Table I are shown plotted in Fig. 7. While the samples upon which the averages are based are generally small and vary appreciably, one or two interesting trends are apparent. For vertical vibration, for instance, the coefficients for tankers and related types increase with number of nodes more rapidly than for types having finer forms, particularly passenger ships. It is also evident that the average values of the coefficients for vertical vibration are generally higher than those for horizontal vibration.

As the number of nodes increases the data available become correspondingly less and in general it would be imprudent to extrapolate the coefficients beyond those indicated in the tables. An exception to this might be when a particular design can be identified very closely with one for which comprehensive data are given.

b) *Amplitudes of Vibration*

The type of vibration considered in this section of the paper is only of practical significance when resonant conditions occur in the regions of the service speeds. It is therefore necessary to have methods of estimating resonant amplitudes and also some criteria for assessing their severities. These aspects have been dealt with at some length in a recent paper[2] to which the reader is referred for more detailed arguments than would be appropriate to give herein.

As a general rule the forms of hull resonance curves correspond reasonably well with those obtained mathematically for simple "mass-spring" systems with fluid dashpot damping.

For the simple "mass-spring" system acted upon by a sinusoidally varying force, the resulting amplitude (y) of the mass (W/g) is given approximately by the following expression:

$$y = \frac{F}{k} \times \frac{1}{\sqrt{\left[1-\left(\frac{n}{N}\right)^2\right]^2 + \left(\frac{1}{Km}\cdot\frac{n}{N}\right)^2}} \qquad (6)$$

where:

F = the maximum value of the applied force
k = the stiffness of the spring
n = the frequency of the applied force
N = the natural frequency of the system
Km = a non-dimensional quantity defining the degree of damping in the system and is referred to as the dynamic magnifier at resonance.

TABLE II.—MODE FUNCTIONS (α)

Fraction of ship length from aft end	Vertical vibration									Horizontal vibration					
	2-node			3-node			4-node			2-node			3-node		
	Upper limit	Mean	Lower limit	Upper limit	Mean	Lower limit	Upper limit	Mean	Lower limit	Upper limit	Mean	Lower limit	Upper limit	Mean	Lower limit
0	1·00	1·00	1·00	1·00	1·00	1·00	1·00	1·00	1·00	1·00	1·00	1·00	1·00	1·00	1·00
0·02	0·936	0·920	0·914	0·935	0·920	0·910	0·897	0·875	0·844	0·945	0·940	0·910	0·925	0·910	0·888
0·04	0·872	0·844	0·825	0·855	0·835	0·800	0·790	0·745	0·695	0·882	0·860	0·825	0·848	0·819	0·775
0·06	0·814	0·770	0·745	0·772	0·738	0·685	0·680	0·604	0·530	0·816	0·780	0·728	0·780	0·734	0·672
0·08	0·748	0·695	0·665	0·682	0·630	0·559	0·564	0·465	0·370	0·750	0·700	0·629	0·710	0·650	0·573
0·10	0·688	0·627	0·585	0·590	0·520	0·434	0·445	0·314	0·208	0·690	0·630	0·545	0·630	0·555	0·456
0·12	0·620	0·545	0·504	0·494	0·415	0·313	0·318	0·178	0·054	0·618	0·546	0·447	0·556	0·464	0·345
0·14	0·558	0·475	0·428	0·410	0·315	0·207	0·207	0·060	−0·082	0·550	0·468	0·355	0·486	0·384	0·245
0·16	0·500	0·404	0·350	0·315	0·208	0·105	0·095	−0·060	−0·202	0·484	0·390	0·265	0·412	0·294	0·140
0·18	0·429	0·330	0·270	0·237	0·115	0·014	−0·010	−0·166	−0·308	0·420	0·315	0·180	0·337	0·206	0·045
0·20	0·365	0·260	0·200	0·154	0·025	−0·080	−0·100	−0·250	−0·375	0·360	0·240	0·100	0·255	0·118	−0·042
0·22	0·305	0·195	0·125	0·070	−0·065	−0·173	−0·186	−0·320	−0·413	0·293	0·165	0·018	0·174	0·037	−0·120
0·24	0·245	0·130	0·060	0	−0·135	−0·242	−0·250	−0·348	−0·429	0·230	0·090	−0·060	0·106	−0·034	−0·192
0·26	0·190	0·070	0	−0·072	−0·200	−0·305	−0·254	−0·347	−0·426	0·170	0·018	−0·132	0·035	−0·095	−0·246
0·28	0·130	0·020	−0·055	−0·130	−0·255	−0·350	−0·208	−0·318	−0·417	0·115	−0·050	−0·200	−0·018	−0·145	−0·296
0·30	0·078	−0·035	−0·115	−0·194	−0·298	−0·382	−0·130	−0·257	−0·386	0·050	−0·122	−0·270	−0·066	−0·190	−0·332
0·32	0·025	−0·082	−0·168	−0·236	−0·317	−0·396	−0·038	−0·185	−0·332	−0·010	−0·185	−0·332	−0·103	−0·216	−0·350
0·34	−0·027	−0·130	−0·218	−0·242	−0·318	−0·396	0·045	−0·108	−0·265	−0·070	−0·248	−0·385	−0·125	−0·234	−0·366
0·36	−0·070	−0·165	−0·260	−0·225	−0·305	−0·383	0·132	0·028	−0·185	−0·122	−0·300	−0·438	−0·132	−0·240	−0·370
0·38	−0·115	−0·208	−0·305	−0·190	−0·270	−0·352	0·210	0·050	−0·114	−0·178	−0·340	−0·485	−0·124	−0·238	−0·368
0·40	−0·150	−0·240	−0·342	−0·150	−0·236	−0·320	0·293	0·134	−0·030	−0·228	−0·380	−0·520	−0·096	−0·222	−0·352
0·42	−0·174	−0·265	−0·370	−0·095	−0·188	−0·276	0·350	0·205	0·048	−0·270	−0·405	−0·545	−0·043	−0·195	−0·327
0·44	−0·195	−0·292	−0·400	−0·035	−0·140	−0·233	0·396	0·272	0·120	−0·300	−0·430	−0·568	0·014	−0·148	−0·293
0·46	−0·205	−0·306	−0·415	0·036	−0·075	−0·176	0·422	0·320	0·190	−0·325	−0·448	−0·584	0·065	−0·097	−0·250
0·48	−0·215	−0·315	−0·425	0·098	−0·010	−0·120	0·428	0·326	0·204	−0·340	−0·456	−0·594	0·120	−0·045	−0·208
0·50	−0·216	−0·318	−0·428	0·154	0·054	−0·064	0·414	0·295	0·145	−0·350	−0·460	−0·600	0·172	0·016	−0·155
0·52	−0·213	−0·315	−0·426	0·210	0·120	0	0·380	0·230	0·067	−0·352	−0·460	−0·600	0·213	0·068	−0·106
0·54	−0·200	−0·304	−0·412	0·260	0·173	0·065	0·332	0·164	−0·005	−0·348	−0·448	−0·592	0·252	0·123	−0·048
0·56	−0·185	−0·290	−0·394	0·304	0·218	0·118	0·267	0·095	−0·072	−0·330	−0·430	−0·580	0·287	0·164	0·008
0·58	−0·160	−0·270	−0·370	0·340	0·258	0·165	0·200	0·024	−0·136	−0·305	−0·410	−0·558	0·310	0·194	0·050
0·60	−0·130	−0·243	−0·342	0·358	0·282	0·198	0·125	−0·038	−0·184	−0·275	−0·384	−0·530	0·328	0·215	0·084
0·62	−0·092	−0·205	−0·305	0·370	0·298	0·215	0·055	−0·100	−0·234	−0·230	−0·345	−0·490	0·338	0·227	0·104
0·64	−0·055	−0·178	−0·265	0·372	0·298	0·220	−0·010	−0·152	−0·276	−0·180	−0·300	−0·444	0·339	0·230	0·112
0·66	−0·012	−0·130	−0·220	0·358	0·284	0·198	−0·066	−0·194	−0·303	−0·120	−0·248	−0·390	0·337	0·226	0·109
0·68	0·040	−0·080	−0·168	0·330	0·248	0·157	−0·115	−0·220	−0·320	−0·063	−0·190	−0·335	0·325	0·208	0·086
0·70	0·088	−0·029	−0·110	0·296	0·208	0·110	−0·134	−0·225	−0·323	0	−0·124	−0·268	0·305	0·170	0·043
0·72	0·137	0·025	−0·055	0·245	0·154	0·050	−0·126	−0·218	−0·316	0·070	−0·060	−0·196	0·276	0·124	−0·018
0·74	0·185	0·076	0·005	0·185	0·085	−0·025	−0·082	−0·194	−0·293	0·132	0·004	−0·134	0·232	0·064	−0·084
0·76	0·245	0·140	0·065	0·123	0·015	−0·096	−0·033	−0·155	−0·262	0·200	0·072	−0·060	0·167	0	−0·155
0·78	0·308	0·207	0·144	0·045	−0·070	−0·178	0·028	−0·105	−0·218	0·265	0·140	0·012	0·094	−0·072	−0·225
0·80	0·360	0·268	0·205	−0·040	−0·155	−0·260	0·098	−0·042	−0·163	0·340	0·220	0·100	0·010	−0·146	−0·296
0·82	0·420	0·335	0·275	−0·130	−0·235	−0·332	0·175	0·036	−0·095	0·410	0·296	0·182	−0·095	−0·230	−0·368
0·84	0·480	0·400	0·345	−0·222	−0·320	−0·408	0·265	0·128	0	0·480	0·375	0·268	−0·193	−0·314	−0·436
0·86	0·542	0·470	0·420	−0·315	−0·400	−0·482	0·334	0·215	0·085	0·545	0·450	0·352	−0·300	−0·406	−0·510
0·88	0·600	0·540	0·495	−0·407	−0·482	−0·550	0·422	0·324	0·200	0·612	0·524	0·440	−0·393	−0·485	−0·572
0·90	0·674	0·622	0·584	−0·500	−0·552	−0·625	0·518	0·426	0·310	0·688	0·610	0·545	−0·490	−0·565	−0·638
0·92	0·735	0·694	0·656	−0·600	−0·645	−0·694	0·617	0·550	0·437	0·742	0·685	0·630	−0·600	−0·655	−0·710
0·94	0·804	0·773	0·740	−0·700	−0·726	−0·770	0·718	0·668	0·585	0·815	0·774	0·725	−0·700	−0·742	−0·779
0·96	0·865	0·845	0·795	−0·810	−0·815	−0·855	0·810	0·776	0·715	0·880	0·850	0·818	−0·805	−0·830	−0·850
0·98	0·930	0·920	0·900	−0·895	−0·900	−0·924	0·912	0·882	0·865	0·945	0·938	0·910	−0·898	−0·905	−0·910
1·00	1·00	1·00	1·00	−1·00	−1·00	−1·00	1·00	1·00	1·00	1·00	1·00	1·00	−1·00	−1·00	−1·00

The first term in this expression, i.e. $\frac{F}{k}$ is the static deflexion of the mass which would be caused by the steady application of the force F. The second term is the general expression for dynamic magnification and it can be seen that it has a maximum value equal to Km when the frequency of the applied force is equal to the natural frequency, i.e.

$$\text{Resonant amplitude } y_r = \frac{F}{k} Km \tag{7}$$

The natural frequency of this system is given by the following expression:

$$N \propto \sqrt{\frac{kg}{W}} \tag{8}$$

Equation (7) may therefore be re-written as follows:

$$y_r \propto \frac{F \quad K_m \quad g}{N^2 W} \tag{9}$$

The dynamic behaviour of each mode form of ship vibration can be expressed in similar terms to that of the simple system. The essential differences are that the spring stiffness is replaced by the flexural rigidity of the hull and the single mass by a distributed mass system. The vibration amplitude will vary with position along the hull according to the mode form. It will also depend upon the position at which the force is applied as well as its magnitude.

Taking these factors into account and allowing for the fact that a distributed system of forces is normally present, the following expression for the resonant amplitude of a ship has been derived:

$$y_x = \frac{35200}{C} \frac{\propto_x K_m}{N^2 \Delta_1} \Sigma F \propto_F \tag{10}$$

where y_x = single amplitude of vibration at any position x (in.)

C = empirically derived coefficient
= 0·094 for two-node vibration
= 0·079 for three-node vibration
= 0·063 for four-node vibration

$\propto_x$ = ordinate of mode form at position x for mode of vibration considered (see Table II)

N = critical frequency of mode considered (cycles/min.)

K_m = dynamic magnifier at frequency N (see Fig. 8)

Δ_1 = total virtual mass (tons)

$\Sigma F \propto_F$ = vectorial summation of the product of the forces and corresponding ordinates of the mode function.

Average values of the $\propto$ functions derived from measurements on a number of ships are given in Table II, for modes of vibration corresponding to those dealt with in the previous section. Also included in the table are figures which indicate limiting values of the function based on existing B.S.R.A. experimental data.

With regard to the values of K_m to be used in conjunction with the above expression, a large number of records have been analysed in an attempt to derive the appropriate values for a wide range of ships. While the scatter of the results is considerable, it is apparent that K_m diminishes generally with frequency for all modes of vibration. It is also evident that dynamic magnifiers for horizontal vibration are appreciably less than those for vertical vibration. As a consequence of this analysis, some tentative relationships for all-welded ships have been derived for practical use and these are indicated in Fig. 8.

c) *Acceptable Limits of Hull Vibration*

In so far as vibration of the main hull is concerned, the most important considerations are usually the psychological and physiological effects on passengers and crew. The resulting stress fluctuations in the hull girder are normally quite small and give little cause for concern even when the vibration is apparently severe. In due course, as ships' instruments and control equipment become more sophisticated it is probable

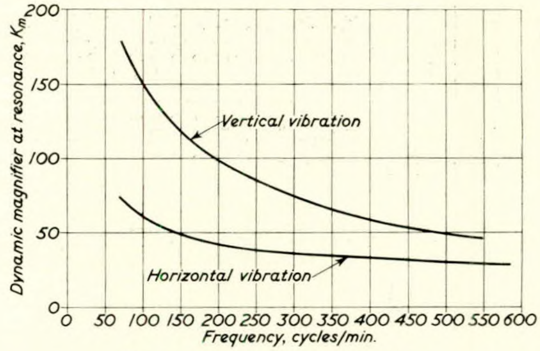


FIG. 8—*Dynamic magnifiers*

that more stringent demands will be made on the ship designer with respect to vibration than are currently thought to be necessary.

There are many opinions as to what constitutes an acceptable level of hull vibration. This is principally because the human reaction to vibration is essentially subjective and human beings vary considerably in their sensibilities to imposed vibration.

As a result of attending tests on many ships and surveying existing literature on the subject, members of the B.S.R.A. staff have suggested[2] some limits of acceptability as defined in the following expressions:

$$\text{Acceptable Amplitude (in.)} \begin{cases} \text{Vertical Vibration} = \frac{1}{n^2}(2n+1000) & (11) \\ \text{Horizontal Vibration} = \frac{1}{n^2}(2n+600) & (12) \end{cases}$$

Propeller Excited Vibration

This type of vibration is perhaps the most important which faces a ship designer at present and at the same time is most complex. While a great deal of research is being applied to the problem it is clear that much remains to be done before the ship designer can be provided with reliable design techniques. The vibratory forces are of two types as dealt with under the following sub-headings a) and b):

a) *Pressure Forces*

Each propeller blade when under load carries a pressure field, negative on the forward face and positive on the after face. As each blade rotates it will give rise to fluctuations of pressure on the adjacent surfaces of the hull and its appendages. These surfaces will therefore be subjected to pulsating forces at a frequency equal to the propeller r.p.m. times the number of blades. Higher harmonics may also be present. Fig. 9 shows for a single-screw form and a twin-screw form the areas subjected to pressure forces. It should be noted that such forces would exist in perfectly uniform flow conditions but that they would be modified in non-uniform flow.

The pressure forces may give rise to vibration of the main hull in the vertical plane, horizontal plane and in torsion. They may also excite vibration of the rudder and the stern frame in single-screw vessels. In multiple screw ships they may excite vibration of the bossings in addition to a variety

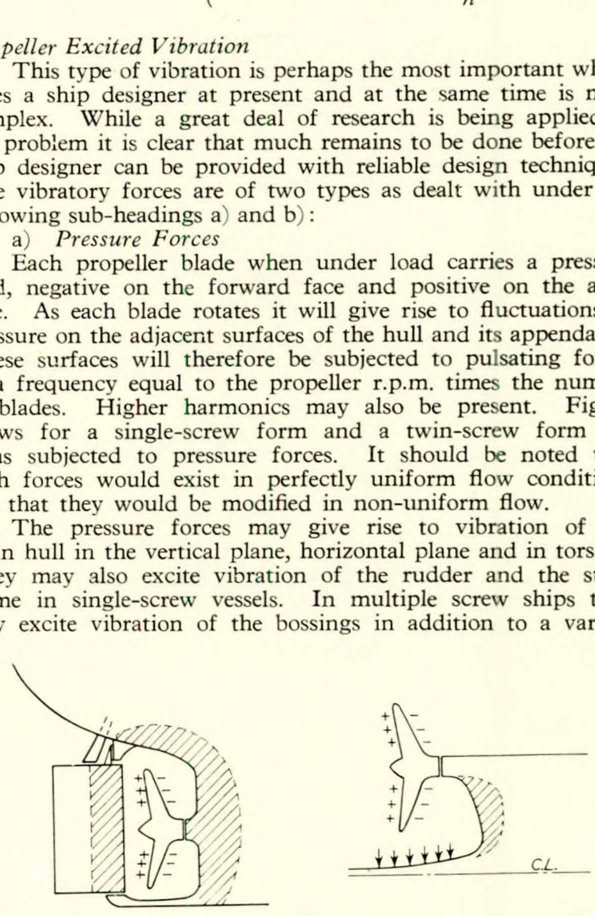


FIG. 9—*Areas subjected to pulsations of pressure from propeller*

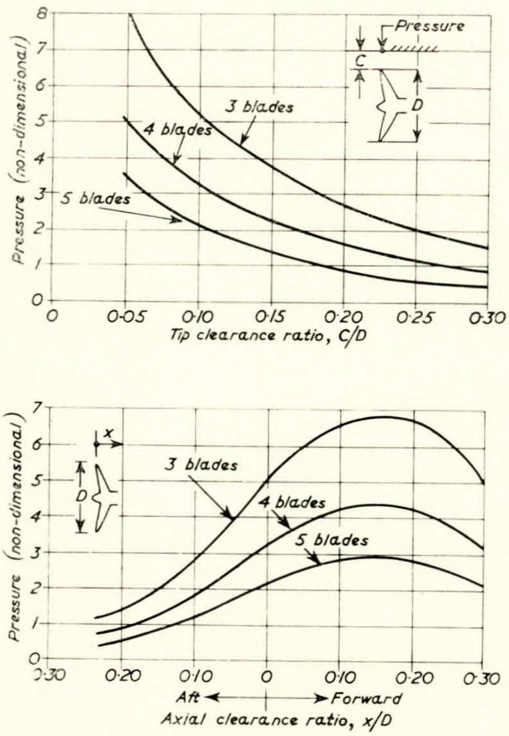


FIG. 10—*Variation of pressure with distance from propeller*

of hull modes and also the rudders if these happen to be situated just abaft the propellers.

The form, magnitude and extent of the pressure field surrounding a propeller will be directly related to propeller design, e.g., the intensity of loading and the geometrical features of the blades such as section shape, pitch distribution and degree of skew. The pressure forces acting on the hull and its appendages will depend upon the respective clearances and the surface contours presented to the propeller.

At the present time the pressure forces cannot be estimated for a given design with a practical degree of confidence but the general way in which they are affected by various design features is reasonably well understood.

Fig. 10, which is based on work by Ramsay[3], shows how the pressures may be expected to vary with distance from the tip of a propeller in both radial and longitudinal directions and with the number of blades. It will be noted that an increase in the number of blades always produces a marked reduction in the pressure. It is also evident that the pressure forces diminish as the tip clearances increase. At small clearances, for example $C/_D = 0{\cdot}1$, the pressure is very sensitive to small changes in the clearance. As the clearances increase the pressure becomes increasingly less sensitive to small variations in the clearance.

b) *Wake Variation Forces*

These forces arise from the fact that a propeller operates in circumferentially non-uniform conditions of flow created by the presence of the hull. The variations of lift and drag on each blade throughout a revolution produce resultant forces and moments as illustrated in Figs. 11 and 12. The lift or axial forces on the blades as indicated by the letters A, B and C vary continuously throughout a revolution and give rise to net axial variations of thrust and bending moment. The fluctuating bending moment is represented by its components in the vertical centre line plane and the horizontal plane. Similarly the drag or tangential forces on the blades as indicated by the letters D, E and F vary continuously and give rise to resultant fluctuations of lateral force and torque. The lateral variations of force are shown resolved in the vertical and horizontal planes. The predominant frequencies of these fluctuations occur at r.p.m. times the number of blades and at twice that frequency depending on the number of blades and the wake distribution.

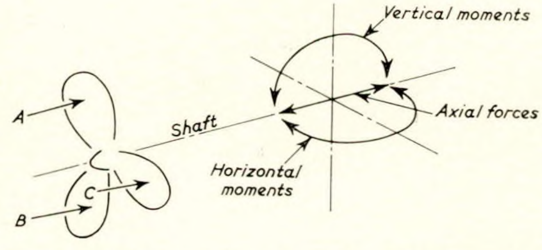


FIG. 11—*Effects of variations of axial loads on blades*

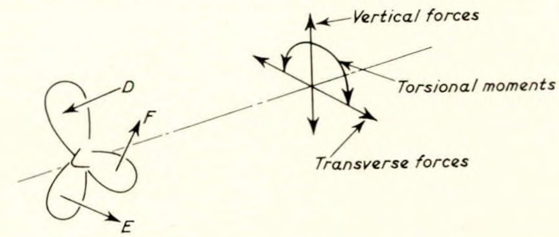


FIG. 12—*Effects of variations of tangential loads on blades*

It will be apparent from the foregoing that the wake forces act on the hull via the propeller, stern bearings, shafting and main propulsion machinery. They may give rise to vibration of the main hull in each of the principal planes as a result of the forces acting normally to the line of shafting. The variation of axial forces is capable of causing vibration of the shafting and propulsion elements but is relatively unimportant as far as the main hull is concerned. Similarly the variations of longitudinal moment and torsional moment are important with respect to machinery vibration but contribute little to hull vibration.

The wake condition is essentially a function of the form of the hull and in particular the stern shape and the problem of minimizing propeller wake forces is largely one of hull and propeller design. It appears that if the wake conditions for a ship can be established it is possible to obtain a reasonable indication of the magnitude of the wake forces for a given propeller by direct calculation. Some success in this direction was achieved by Brehme[4] who, using a very simple technique, performed some calculations for a standard type of propeller operating in a typical wake distribution for a single-screw form. In this method the first step was to average radially the mean inflow velocity which a blade would experience at many positions throughout a revolution. Using these figures in association with standard propeller charts it was possible to build up the picture of the thrust and torque variations experienced by a single blade throughout a revolution. It was then a simple step to complete the picture for a propeller having its full quota of blades. The final stage was to subject the results to harmonic analysis to determine the magnitudes and frequencies of the various components.

Results obtained in this way have been generally substantiated by experimental work carried out subsequently by Van Manen and Wereldsma[5] on tanker forms. More recently there has been a number of papers giving the results of more refined types of calculation together with a variety of results from model experiments. Notable contributions have been by Van Manen and Kemps[6], Stuntz, Pien and Hinterthan[7] and by Kumai[8]. In these investigations attempts have been made to determine the effect of variation of stern shapes and propeller design on the wake forces.

With regard to the influence of stern shape, Fig. 13 shows in outline some forms currently in use in single-screw ships.

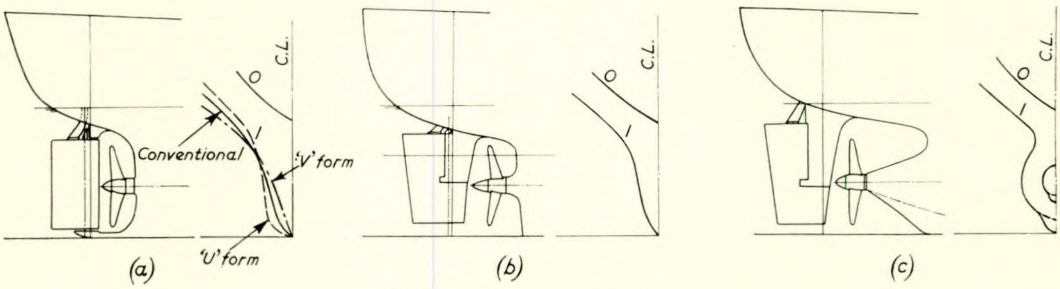


FIG. 13—*Typical stern forms*

Type A represents a conventional closed aperture type of stern in which the sole-piece of the stern-frame ties up with a rudder post carrying the pintles or with the shaft of a semi-balanced type of rudder. The characters of the U and V sections are indicated on the diagram. Type B represents the *Mariner* type of stern. The requirements of the design for these ships which had a speed of 20 knots at a shaft horse-power of about 18,000 included excellent manœuvring characteristics and minimum vibration. In order to maintain the rudder torque within practical limits it was necessary to use a balanced type of rudder and this, coupled with the desire to obtain good propeller clearances, led to the type of design indicated. Type C, known as the Hogner stern or perhaps more generally as the concentric bulb type of stern represents an attempt to obtain more uniform flow conditions into the propeller. While the primary object was almost certainly to improve propulsive performance, the arrangement is clearly a good one from the point of view of vibration.

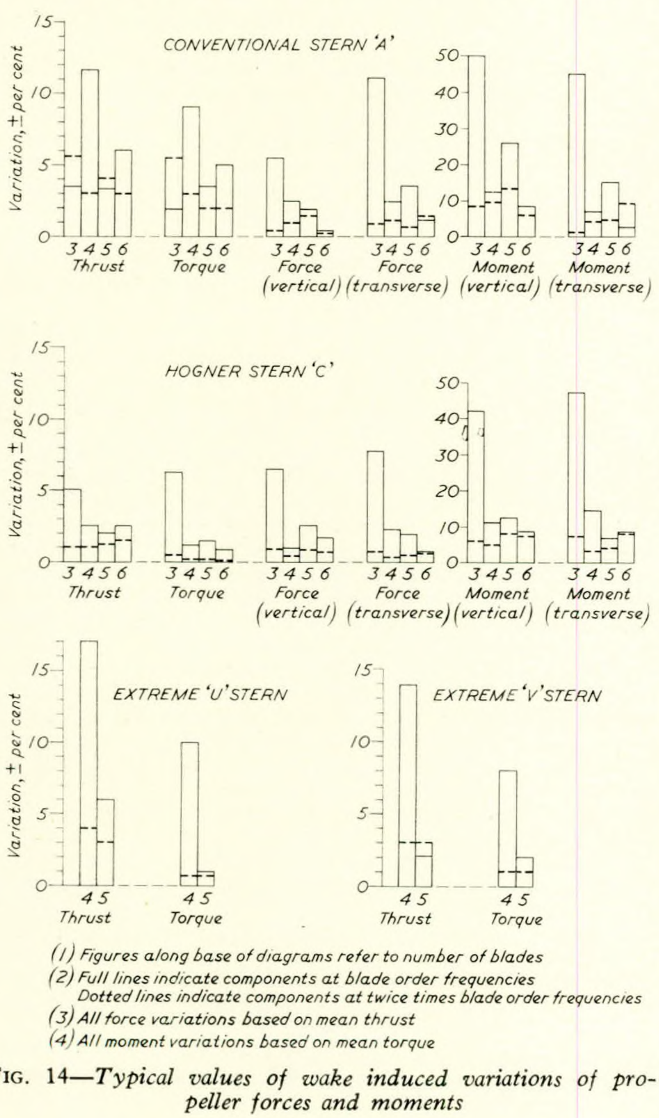


FIG. 14—*Typical values of wake induced variations of propeller forces and moments*

Fig. 14 gives a diagrammatical summary of data collected from various papers referred to in the bibliography. The data include experimental and calculated results for the wake induced forces and moments for several of the types of stern already mentioned and for propellers having different numbers of blades. The ordinates represent the percentage variation of the various forces and moments, referred to the mean thrust and torque respectively. The full lines represent the magnitude of variation of components at a frequency equal to the r.p.m. times the number of blades. The dotted lines indicate the components at twice the blade order frequency and it is interesting to note that these may sometimes predominate.

While it would be imprudent to place too much significance on the magnitudes indicated in Fig. 14 it is thought that the trends are generally correct. With these reservations the following generalizations may be made:

i) Propellers having odd numbers of blades are to be preferred to those having even numbers of blades if the prime considerations are the thrust and torque variations (five blades are preferable to three blades).
ii) Propellers having even numbers of blades are to be preferred to those having odd numbers of blades if the prime considerations are moments in the transverse and longitudinal planes (six blades are preferable to four blades).
iii) The vertical and transverse forces tend to diminish as the number of blades increases.
iv) The vibratory forces and moments may be appreciably smaller for the type of stern which is designed to produce a circumferentially uniform distribution of wake in way of the propeller.

It will be noted that the results for the U-stern are somewhat worse than those for the V-stern and that both are worse than those for the conventional stern. These results are contrary to what might be expected and in this respect it should be mentioned that the evidence is rather scant.

One other matter which should be mentioned in connexion with the wake forces is that the experimental results available suggest that alterations to propeller blade design features such as pitch distribution and skew have a relatively small effect on the wake forces. Such evidence is, however, based essentially on well designed propellers and it is known that if a propeller is initially of poor design appreciable reductions in the wake forces can be achieved by re-design.

c) *Hull Response*

For the lower modes of hull vibration it has been noted that the amplitudes only rise to appreciable values when the frequency of the applied force is close to one of the natural frequencies. Except for certain small vessels, propeller excited vibrations will lie fairly high in the frequency spectrum. In these regions the hull response characteristics are markedly different and it becomes necessary to consider an alternative

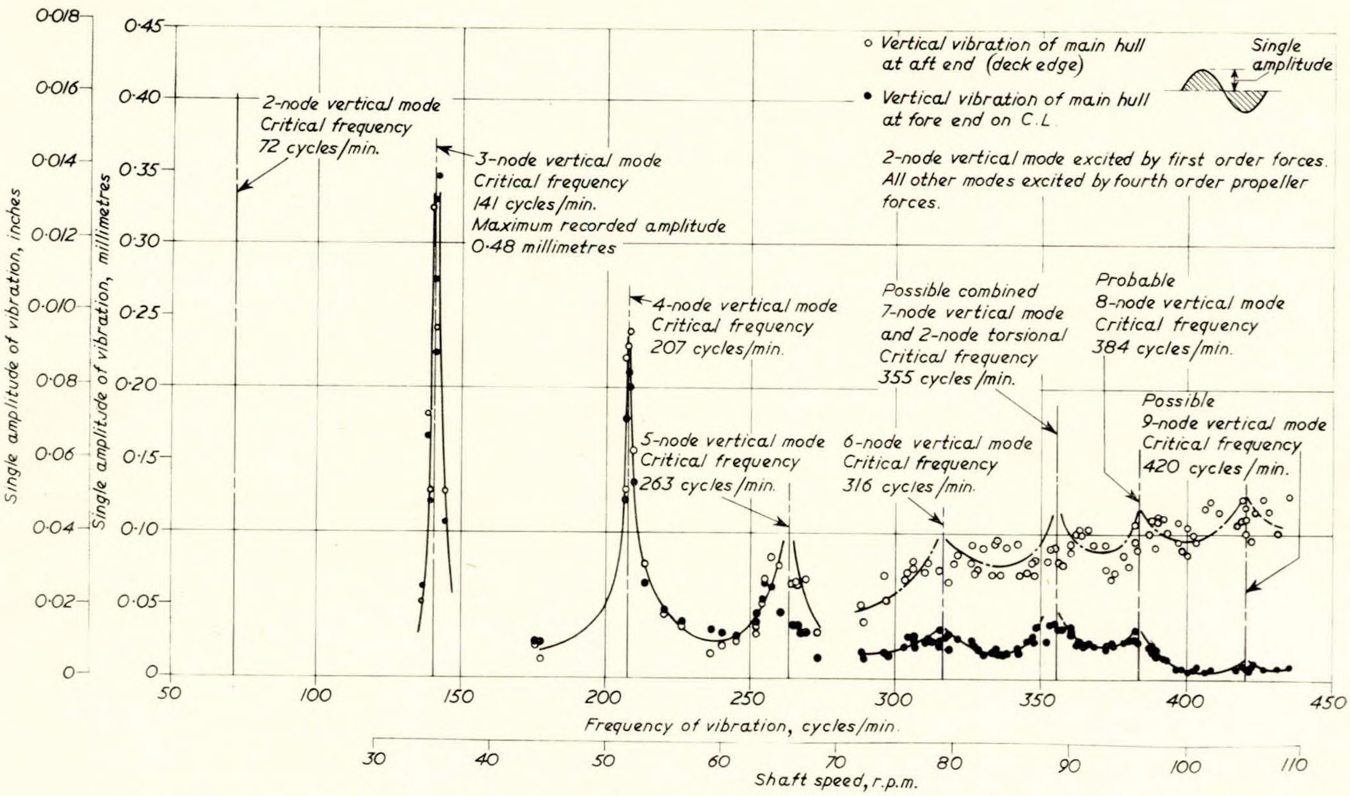


FIG. 15—*Vertical and torsional vibration of main hull*

approach to the problem of "force-amplitude" relationships.

Fig. 15 shows an example of propeller excited vibration on a 520-ft. dry-cargo ship. It will be observed that the sharply tuned resonances of the lower modes do not exist at the higher frequencies. It is also noted that at the after end, in the regions of the propeller forces, the maximum amplitudes first diminish with frequency and then increase continuously. At the forward end of the ship, however, the maximum amplitudes tend to diminish throughout the entire frequency range.

These observed trends can be predicted from theoretical considerations by making several simple assumptions. It is assumed that a ship may vibrate in an infinite number of modes each of which may be represented approximately by an equivalent "mass-spring" system having a single degree of freedom and simple viscous damping. Fig. 16 shows the first few components of the complex system. From this it will be evident that under conditions of forced vibration the amplitude of vibration at any one position will be composed of the vector addition of the residues of the amplitudes of all modes above and below that frequency.

If the exciting force is simple harmonic in form and the maximum value is expressed as $P = Cn^p$, the amplitudes of the component systems at the position of the exciting force at frequency n may be written approximately

$$y_{1n} = \frac{\frac{c}{m_1}\left(\frac{n}{N_1}\right)^2 n^{(p-2)}}{\sqrt{\left[1-\left(\frac{n}{N_1}\right)^2\right]^2+\left(\frac{1}{K_m}\,\frac{n}{N_1}\right)^2}} \qquad (13)$$

$$y_{2n} = \frac{\frac{c}{m_2}\left(\frac{n}{N_2}\right)^2 n^{(p-2)}}{\sqrt{\left[1-\left(\frac{n}{N_2}\right)^2\right]^2+\left(\frac{1}{K_m}\,\frac{n}{N_2}\right)^2}}, \text{ etc.} \qquad (14)$$

In these expressions K_m is the resonant magnification; N_1, N_2, etc. are the natural frequencies of the system; and m_1, m_2, etc. are the equivalent masses associated with the various modes

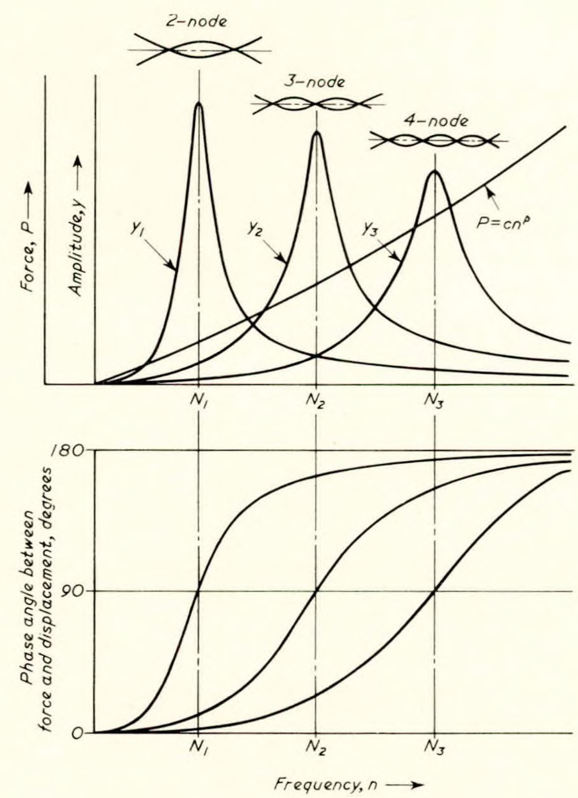


FIG. 16—*Components of vibrating system*

of vibration. It will be assumed for present purposes that the equivalent mass remains constant for all modes of vibration, i.e. that the changes in natural frequency from mode to mode result from changes in the equivalent spring stiffnesses.

The phase angle between the force and the amplitude for each mode is given by:

$$\tan \phi_{1n} = \frac{\frac{1}{K_m}\left(\frac{n}{N_1}\right)}{1-\left(\frac{n}{N_1}\right)^2}, \qquad (15)$$

$$\tan \phi_{2n} = \frac{\frac{1}{K_m}\left(\frac{n}{N_2}\right)}{1-\left(\frac{n}{N_2}\right)^2}, \text{ etc.} \qquad (16)$$

If the total amplitude in way of an exciting force applied at one end of a ship is given by Y and that at the other end of the ship by Z, we may write

$$(Y_n)^2 = (y_{1n}\cos\phi_{1n} + y_{2n}\cos\phi_{2n} + y_{3n}\cos\phi_{3n} + \ldots)^2 + (y_{1n}\sin\phi_{1n} + y_{2n}\sin\phi_{2n} + y_3\sin\phi_{3n} + \ldots)^2 \qquad (17)$$

and

$$(Z_n)^2 = (y_{1n}\cos\phi_{1n} - y_2\cos\phi_{2n} + y_3\cos\phi_{3n} - \ldots)^2 + (y_{1n}\sin\phi_{1n} - y_{2n}\sin\phi_{2n} + y_3\sin\phi_{3n} - \ldots)^2 \qquad (18)$$

In order to obtain numerical solutions it is necessary to make some assumptions with regard to the following:

a) Relationships between the natural frequencies N_1, N_2, N_3, etc.
b) The power (p) of the forcing function.
c) The manner in which K_m varies throughout the frequency range.

A solution for a simple example is illustrated in Fig. 17 which is based on the following assumptions:

$$\frac{N_2}{N_1} = 2, \frac{N_3}{N_1} = 3, \frac{N_4}{N_1} = 4, \text{ etc.}$$

$$p = 2$$

$$K_m \propto \frac{1}{(e-1)} \text{ where } e = \text{number of nodes}$$

It has also been assumed that the value of K_m for the two-node mode of vibration was equal to 60.

Despite the over simplification of the problem it will be observed that the principal features of the calculated results are similar in character to the experimental results given in Fig. 15. It is a relatively simple matter using electronic computers to extend such calculations to determine the vibration profiles at any given frequencies and for any position of the exciting force. Assuming the basic mode forms are those for a uniform "free-free" beam the vibration profiles have been estimated for frequencies corresponding to N_2, N_4, N_6, N_8 and N_{10}, and these are illustrated in Fig. 18. The profiles have been drawn without regard to phase and may be taken to represent the variation of amplitude which would be recorded by a simple vibrograph traversed along the length of a ship.

It will be noted that the true nodal positions gradually cease to exist as the frequencies increase and as indicated previously the amplitudes generally tend to increase towards the after end as compared with those at the forward end.

d) *Some Considerations of "Propeller-hull" Clearances*

Up to the present time the ship designer has relied to a great extent upon simple formulations for propeller-hull clearances. In single-screw ships for instance, empirical formulæ of the type shown in Fig. 19 are commonly used. It will be noted that these formulæ involve only the propeller diameter D. There is no doubt that these were based upon a great deal of experience and represented good practice for many ships built earlier than 1955, perhaps some a good deal earlier. With the ever present trends towards higher speeds and powers however, propeller vibration troubles have increased at a disturbing rate and it is unlikely that the adoption of the recommendations given in 1955 are sufficiently comprehensive to be taken any longer as a general insurance against vibration.

In considering possible modifications to the formulæ for propeller clearances, a useful starting point is perhaps a paper by Ramsay[(3)]. In this paper a formula is given for the pressure fluctuation at a point in the field of a rotating propeller. This formula is as follows:

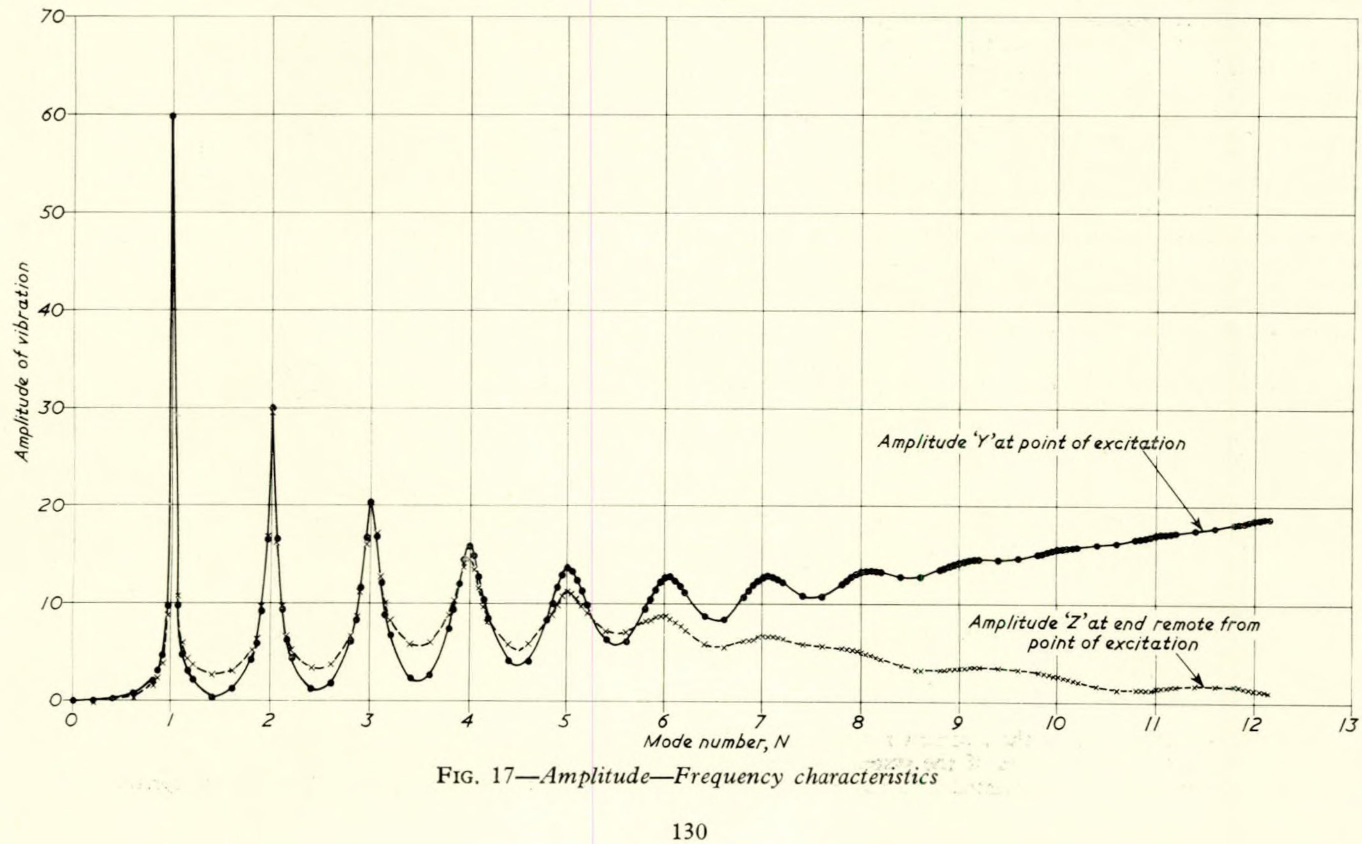


FIG. 17—*Amplitude—Frequency characteristics*

Pressure at point $(x, y) = P_{xy} = \frac{\lambda}{D^2}\left(\frac{D}{Py}\right)^{3/2}\left[\left(\frac{Q}{D}\frac{y}{D}\alpha_z\right)^2 + \left(0{\cdot}7T\frac{x}{D}\beta_z\right)^2\right]^{1/2}$ (19)

where:

x = the co-ordinate measured along the propeller axis

y = the co-ordinate measured radially from the propeller boss

λ = a coefficient depending on the chordwise distribution of pressure over the blade

D = the propeller diameter

P = a function of x/D and y/D

Q = the torque developed by the propeller

T = the thrust developed by the propeller

α_z and β_z = functions of P and the number of blades

The following assumptions will now be made:

i) If a surface is introduced into the vicinity of the propeller, the normal pressure on the surface is given by equation (19).

ii) The distribution of normal pressure on the surfaces in the longitudinal and transverse directions will be independent of the magnitude of the pressure, i.e. the pressure at any one point will, for practical purposes, define the whole field.

Consider now the surface immediately above the propeller at a distance defined by the tip clearance C. The fluctuating pressure on this surface will be one of the causes of vertical vibration in the ship. The maximum pressure on the surface in the plane of the propeller disc can then be deduced from equation (19) thus:

$$p = \lambda\left[\frac{\alpha}{p^{3/2}\left(\frac{1}{2}+\frac{C}{D}\right)^{1/2}}\right]\frac{Q}{D^3} \qquad (20)$$

That part within the brackets can be evaluated from the various relationships given in Ramsay's paper and we may re-write the equation as follows:

$$p = \lambda\,\Phi\,\frac{Q}{D^3} \qquad (21)$$

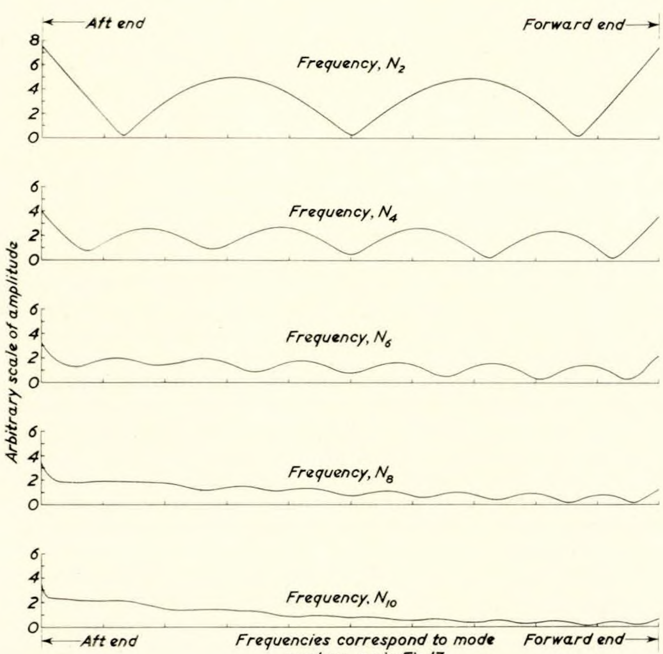
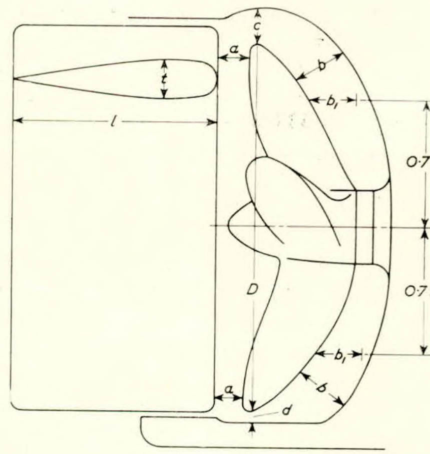


FIG. 18—*Calculated vibration profiles*



Position	Clearance, ft.	
	Bunyan (I.N.A. 1955)	Det Norske Veritas 1955
a	0·08D, or t if t>0·08D	0·08–0·15D
b	0·15D	—
b_1	—	0·13D
c	0·10D	0·08D
d	0·03D	0·03D

$b_1 = 0{\cdot}11(1+\phi)D$ where ϕ (radians) is the half angle of entrance of the waterline at 0·5R above the screw centreline.

For an S.S. cargo vessel $\phi = 9/57{\cdot}3 = 0{\cdot}157$

$b_1 = 0{\cdot}11(1{\cdot}157)D = 0{\cdot}13D$

FIG. 19—*Aperture clearances*

In this equation Φ is a function of C/D and the number of propeller blades and is illustrated in Fig. 20. Also shown in the figure are some lines which suggest practical interpretations of the function for single-screw ships with closed apertures.

If the further assumption is made that the shape of the hull immediately above the propeller is similar for all ships, the total force might be expressed as the product of the pressure and the square of the diameter, with the addition of a constant

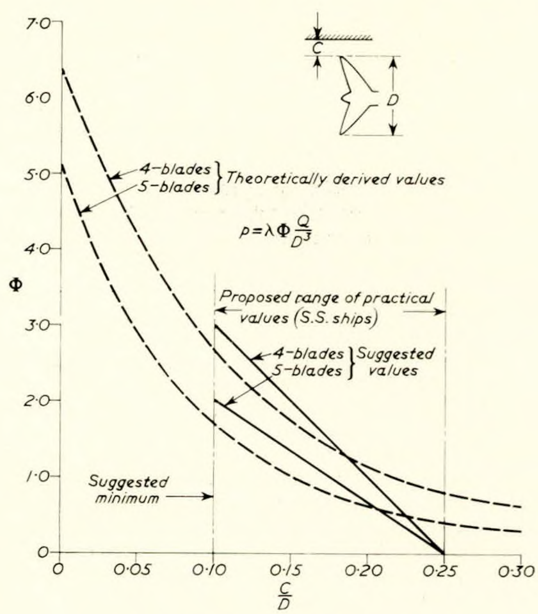


FIG. 20—*Tip clearance functions*

to take care of the integration. In addition, if λ is taken as constant, we may write:

$$\text{Force} = F = K\,\Phi\,\frac{Q}{D} \tag{22}$$

$$\text{now } Q = \frac{\text{s.h.p.} \times 33{,}000}{2\pi\ \text{r.p.m.}}$$

$$\text{hence } F = K_1\,\Phi\,\frac{\text{s.h.p.}}{\text{r.p.m.}\ D} \tag{23}$$

It will now be assumed as a first approximation that the "force-amplitude" relationship for propeller excited vibration can be expressed in a simplified form as follows:

$$y \propto \frac{F}{n\Delta_1} \tag{24}$$

where:

n is the frequency of vibration and is equal to the propeller r.p.m. times the number of blades.

Combining the above expression with equation (23) for force F and equation (11) for acceptable limit for vertical vibration the following expression may be derived for an acceptable value of Φ:

$$\Phi = K_2\,\frac{\text{r.p.m.}\ D}{\text{s.h.p.}}\,\Delta_1\,\frac{(2n + 1{,}000)}{n} \tag{25}$$

In this equation are included some of the parameters which would appear to be significant in defining the clearances at the top of the aperture, considered from the point of view of surface forces only and for vertical vibration.

For an expression of this type to have any practical significance it is necessary to determine the value of K_2. This can hardly be evaluated from first principles and for present purposes it may be permissible to assume that a tip clearance of $C/D = 0{\cdot}1$ has proved adequate in the past for medium sized cargo ships with average power, say 10,000-15,000 d.w.t. at 5,000-6,000 s.h.p. On this basis values of K_2 have been determined from equation (15) and Fig. 20 using the appropriate data for a number of ships. The following average values of K_2 were determined from this analysis:

four bladed propeller $K_2 = 4{\cdot}3 \times 10^{-5}$

five bladed propeller $K_2 = 2{\cdot}9 \times 10^{-5}$

Reference so far has been made only to the tip clearance C but it may be assumed that all the other clearances may be governed by similar considerations with the exception of the possible effects of blades entering the boundary layer. As a further stage therefore towards a practical design solution it might be appropriate to adopt a basic set of clearances such as those proposed by Bunyan[9] and to modify those in the light of an expression such as that given in equation (25). The principle might be adopted that if it appears appropriate to modify the clearance ratio C/D by a certain percentage then all the remaining clearances be altered similarly.

There are various views on the effects of the propeller tips entering the boundary layer, but little definite information. It is doubtful whether the thickness of the boundary layer in way of the propeller in single-screw ships can be defined accurately or indeed whether it exists in the commonly accepted form. However a review of various published works indicates that the thickness of the boundary layer might be related to ship length in the general manner shown in Fig. 21. In the absence of more adequate data it might be prudent to ensure that the tip clearance is not less than that indicated in Fig. 21.

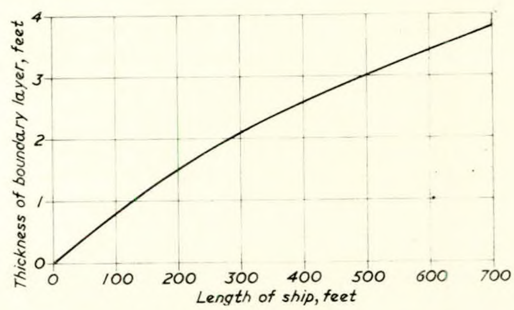


FIG. 21—*Thickness of boundary layer in way of propeller*

As mentioned previously, the variable wake conditions give rise to pulsating forces which act upon the ship through the shaft bearings. It appears probable that these wake forces would not be greatly influenced by small changes in the aperture clearances.

It might be mentioned in conclusion that the clear water types of stern have much in their favour since the designer is presented with an opportunity of providing generous all-round clearances without the difficulties associated with closed apertures.

PART II

VIBRATION OF PROPULSION SYSTEM

Torsional Vibration of Shafting Systems

A large number of the vibration problems encountered in ships' machinery are concerned with the shafting; they can profitably be divided into two fundamental types, namely, torsional and rectilinear and it is a reasonable generalization to observe that the adverse effects of excessive torsional vibration are confined to the shafting system whilst excessive rectilinear vibration is just as likely to produce damage to the surrounding structure as to the shafting system itself.

Torsional vibration is a twisting motion and in shafting it takes the form of an oscillation superimposed on the steady rotational motion of the shaft. Every shafting system has a variety of modes of torsional vibration and the natural frequencies of these can be calculated accurately nowadays in the design stage. Some people take a lot of the computation labour out of the process by using an electronic computer. Account must be taken of the whole shafting system as a unit so that, for instance, in a direct-coupled oil engine installation the crankshaft, line shafting, tailshaft and propeller must be treated as one unit and modification of any of the component parts means that the torsional characteristics of the shaft system must be recalculated.

In addition to the in-line shafting systems such as the direct-coupled oil engine, there are also, of course, what are known as branched systems, where two or more shafts operating in parallel are connected to line shafting by means of gears. Two such applications are where two oil engines are geared to drive a common propeller and, of course, the conventional two or three cylinder steam turbine set. Problems of this type can also be solved although the solution is more complex. Among the aspects which have to be considered is the reduction in natural frequency which arises from backlash in gearing and prediction tends, therefore, to be somewhat less reliable than for in-line systems.

Having noted that procedures exist for the examination of the torsional vibration characteristics of shafting systems, it is time to turn to consideration of the harmonic disturbances that may enter such a system and possibly excite it into large amplitudes. The two sources of disturbance are the engine and the propeller and in considering each of these it is important to keep in mind that each harmonic of each disturbance tends independently to excite the shafting system at the frequency of that harmonic. Each exciting frequency can be expressed as the product of the speed of revolution of the shaft and the number of impulses which occur in each revolution. The number of excitations during each revolution is known as the order number.

In an oil engine there is a torsional disturbance represented by the variations in torque arising from:

a) the gas forces and
b) the inertia forces.

As the pressures in the cylinders vary the net gas forces that are transmitted along the connecting rods and appear at the crankpins are vibratory. At the same time, due to the acceleration and deceleration of the reciprocating parts, inertia forces are developed and these are also transmitted along the connecting rods and appear also at the crankpins as vibratory forces. The inertia forces are made up of two terms, the primary inertia forces which act at the same frequency as the revolution speed of the engine, and the secondary inertia forces which act at twice the revolution frequency of the engine. These secondary inertia forces are due to the angularity of the connecting rods which prevents the pistons moving with simple harmonic motion. If the connecting rods were infinitely long, there would be no secondary inertia forces.

The values of torque input to the shaft are plotted in curve form as shaft torque, or crank effort, against the crank angle and the resulting curve, which is known as a crank effort diagram, is subjected to harmonic analysis, thereby giving the magnitude of each of the harmonic frequencies capable of exciting the shafting system to resonance.

For a four-stroke cycle engine the order numbers to which attention must be paid are those at one-half the number of cylinders and multiples thereof. These are called the major orders. For a two-stroke cycle engine the major orders are those which are integer multiples of the number of cylinders. These concepts require care in application, however, firstly because the presence of scavenge pumps may affect the inertia forces and, secondly, because the effect of a torque applied at a point far from a node will have an effect greater than one close to a node; in certain cases the minor orders cannot, therefore, be disregarded.

The analysis of the torsional characteristics of a shafting system must go farther than the computation of the natural frequencies of the various modes of torsional vibration. From the torsional stiffnesses of the various elements of the shaft the amount of deflexion of each of these elements can be assessed. Since the stiffness of a crankshaft is not uniform along its length, for instance, its rate of deflexion will not be uniform along its length either. The total deflexion at any point will be the sum of the deflexions of all the elements between that point and a node; the maximum values of deflexion occur at the anti-nodes. The deflexions at all anti-nodes need not be of the same magnitude. The relative total deflexion of each point of the shaft can be plotted to a base of shaft length giving what is known as a torsional profile for the particular mode, e.g. one-node, two-node, etc.

Once the profile has been computed it is only necessary therefore to measure the actual deflexion at any one point to determine the deflexion at all points in the shaft length. The most convenient measuring point on an oil engine installation is generally the forward end of the crankshaft. There are advantages in using this point since it is an anti-node for all modes of vibration and the deflexions are therefore large; a seismic type of instrument is used in this case. Sometimes, however, it is found that shaft speed is variable, as in the case of an engine with a small number of cylinders and a light flywheel making interpretation of seismic records rather difficult, and in such a case the use of strain gauges on the line shafting is more appropriate. Calibration of such strain gauges output is necessary, however, if an accuracy better than about ± 10 per cent is to be guaranteed. In many cases calibration by a torsionmeter can be arranged and an accuracy of ± 2 per cent may then be anticipated. B.S.R.A. has also used a modified form of Siemens-Ford torsionmeter to give good torsional records.

The deflexion between any two points on a shaft can be used in conjunction with the geometry of the shaft between these points and the modulus of the material to determine the stresses in the shaft. Using the torsional profile the stresses at all points in the shaft can therefore be computed for unit deflexion of the forward anti-node, that is, the forward end of the crankshaft. This enables a curve of deflexion (at the forward end) against shaft speed to be expressed as a curve of torsional vibration shaft stress (at a critical point) against shaft speed simply by computing a suitable vertical scale.

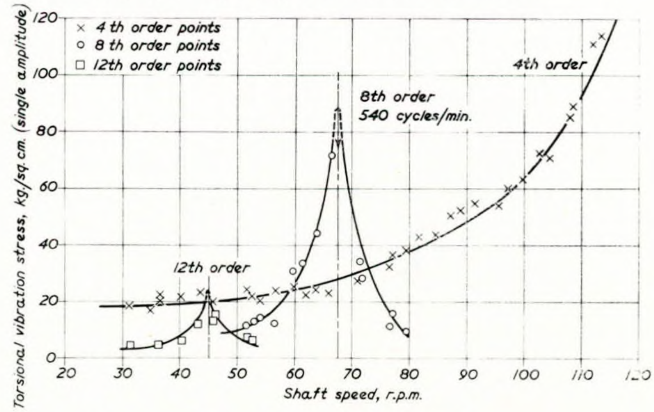


FIG. 22—*Graph of torsional vibration stresses against shaft speed*

An example of such a graph of torsional vibration stresses against shaft speed is shown in Fig. 22. This analysis is based on records taken by wire resistance strain gauges on the 20-in. diameter intermediate shaft of an aft-engined installation incorporating a four-cylinder two-stroke opposed piston engine and a four-bladed propeller. The gauges were calibrated by simultaneous readings on a Siemens-Ford torsionmeter.

As the shaft speed rises towards the critical speed of the particular mode of vibration, the stress will rise steeply, reaching a maximum at the critical speed and then falling again. Between certain speeds the stress may exceed an acceptable level and the range between these speeds is then designated a "barred range" in which continuous running is not permissible.

Although it was observed earlier that the adverse effects of excessive torsional vibration are generally confined to the shafting system, mention must be made of the damage to gears and chain drives which may result if a drive to engine auxiliaries is taken from a position having excessive torsional deflexion. Such an instance could be a fuel pump drive, and pitted and even fractured teeth of a gear wheel could result. A number of cases of damaged drives have been encountered which have indicated that this point must be watched. An apparently innocuous change of intermediate shaft diameter, for example, may so shift the node of the system as to give excessive amplitudes at a gear on the crankshaft.

Harmonic impulses are received by a steam or gas turbine shaft as the blades pass a nozzle, the order number being the number of blades of the disc.

It was pointed out earlier that the exciting force may also come from the propeller; the order numbers are the number of blades on the propeller and multiples thereof.

This brings us back to the forces generated at the propeller to which reference has already been made in considering main hull vibration. The torque fluctuations which act as the disturbing forces in the case of torsional vibration are but one of the three forms of disturbance applied at the propeller to the shafting system. The other two vibratory disturbances give rise to shafting problems which are both of the second general type of vibration, to which earlier reference was made, namely, rectilinear.

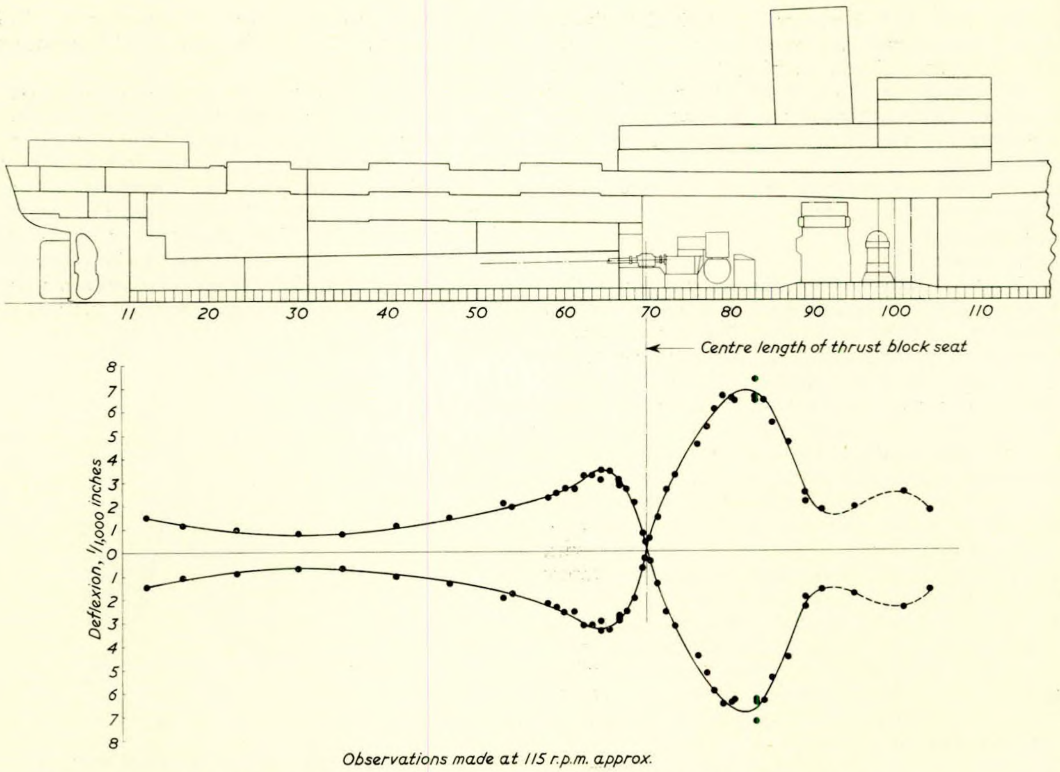


FIG. 23—*Profile of vertical vibration along centre line on tank top*

Axial Vibration of Shafting Systems

The first of these types of rectilinear vibration, axial vibration, has been increasing in importance of late. It consists of axial compression and extension of the line shafting together with the accompanying motion of the thrust block, thrust seating and associated bottom structure. In the case of a turbine installation, it also involves motion of the main gear wheel and possibly the pinions, whilst in a direct-coupled oil engine installation it also involves the crankshaft. Until a few years ago, axial vibration in turbine installations was only encountered on multi-screw naval vessels, but it has now been found in single and twin-screw merchant ships, and is likely to become of increasing importance with increases in the power transmitted by a single shaft and the rise in shaft speeds. The most severe cases give rise to hammering of the thrust collar between the ahead and astern pads of the thrust block of a turbine installation and possibly also to gear damage; in oil engines with integral thrust blocks there may be signs of distress at the after end of the casing such as cracks in the casing, the failure of bolted connexions and loss of lubricating oil; in the course of the past few years a number of instances of fractured crankshafts have occurred which can be attributed to axial vibration. Fig. 23 shows a profile of vertical vibration along the centre line on the tank top of a single-screw cargo liner and illustrates the extent to which the bottom structure of a ship may vibrate when excited by the fore and aft rocking movement of the thrust block and seating, which is illustrated in Fig. 24. The nature of the bottom structure vibration is

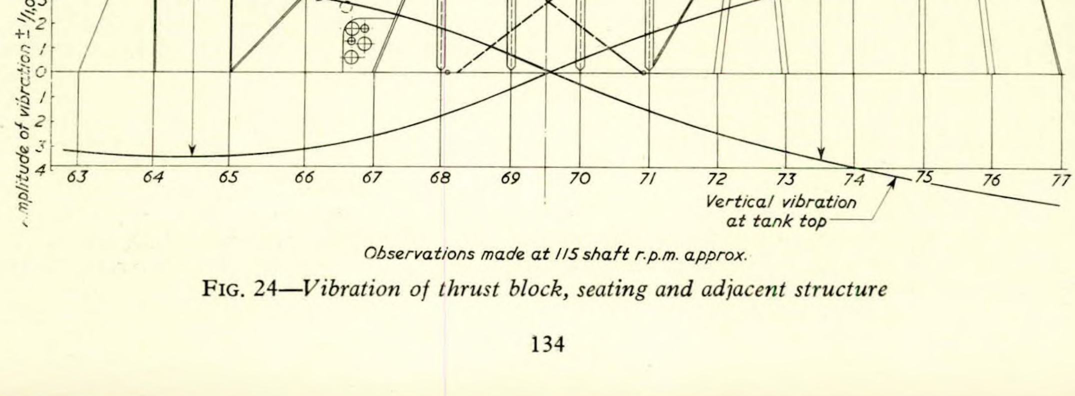


FIG. 24—*Vibration of thrust block, seating and adjacent structure*

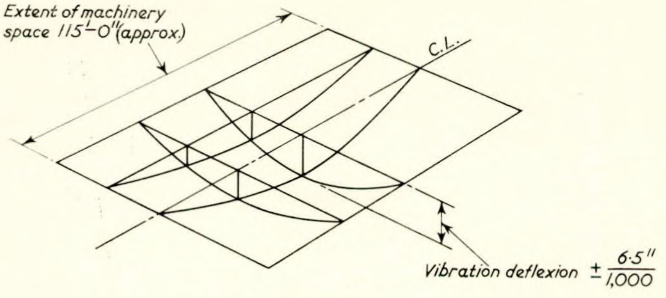


FIG. 25—*Vibration of bottom structure in way of machinery space*

is the magnification factor at resonance and V_P is the thrust variation at the propeller.

Assuming that V_P is a constant fraction of the mean thrust and that the mean thrust varies as the square of shaft speed, equation (26) can be rewritten

$$V_{TB} = \frac{1}{\sqrt{\{(1-x^2)^2 + x^2\delta^2\}}} \cdot x^2 V_{PC} \qquad (27)$$

where V_{PC} is the thrust variation at the propeller at the axial critical speed.

Using a value of ± 1 ton for V_{PC}, curves can then be constructed of V_{TB} for a range of values of δ, which is a function of the damping of the axial system.

Experience with a sizable number of ships has shown

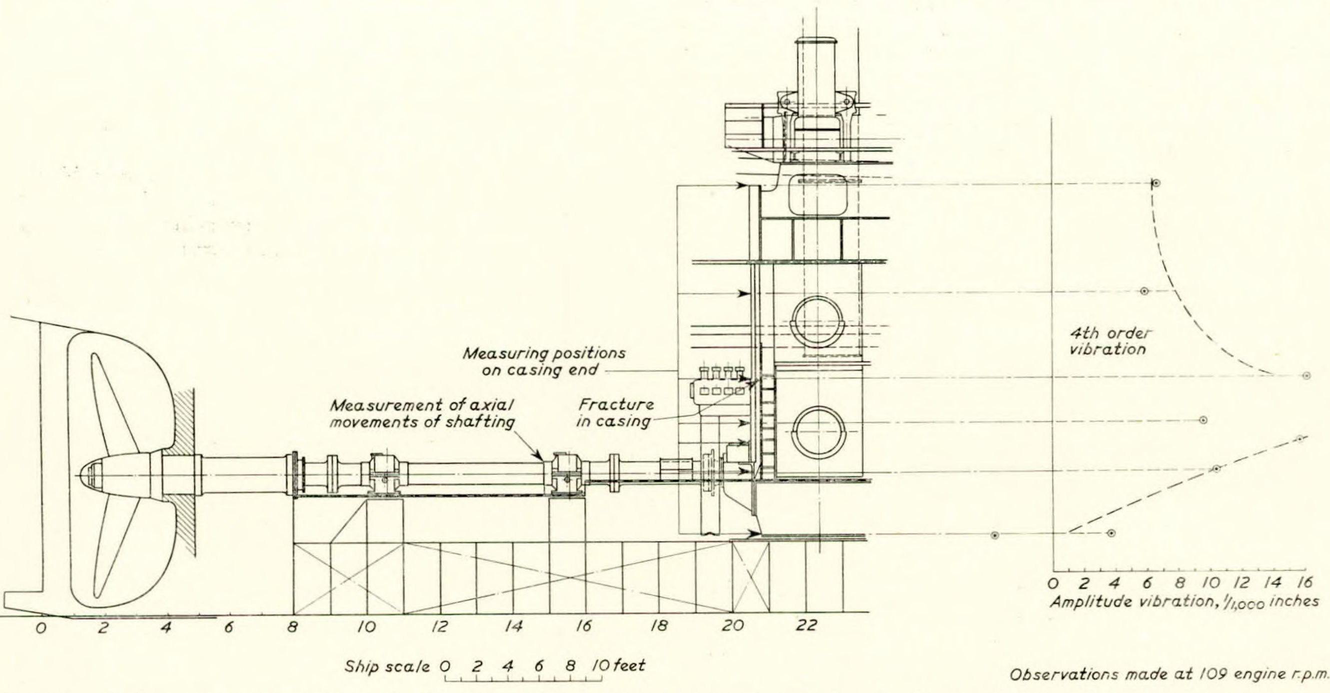


FIG. 26—*Main engine and shafting showing measuring positions and 4th order longitudinal vibration on engine casing end*

shown diagrammatically in the three-dimensional representation in Fig. 25. An example of an integral thrust block and engine casing problem is shown in Fig. 26 which shows the main engine and shafting of a single-screw motor ship with an after end installation; the fourth order longitudinal vibration of the engine casing end will be seen.

As in the case of torsional vibrations, the significance of axial vibration criticals within the running range depends upon the magnitude of the exciting forces. These forces have been discussed earlier in the paper and reference has been made to the work of Brehme[4] in calculating their magnitude; however, such analysis requires that the wake conditions be established and, in spite of the experimental work of Van Manen and Wereldsma[5] on model tanker forms, knowledge of actual ship conditions in this field is still very scanty. B.S.R.A. has collected data of thrust variations from a number of ships; the earlier ones were mostly twin-screw but recently a number of single-screw vessels has been included. The data are not yet, however, sufficient for presentation in systematic form.

It can be shown that the thrust variation at the thrust block at any shaft speed N is given by

$$V_{TB} = \frac{1}{\sqrt{\{(1-x^2)^2 + x^2\delta^2\}}} \cdot V_P \qquad (26)$$

where x is the ratio of shaft speed N to axial critical speed, $\frac{1}{\delta}$

that the damping of the axial vibration system lies within the range, expressed by the log decrement, 0·20 to 0·52 (log decrement $= \pi\delta$). Fig. 27 shows the extreme values of thrust variation at the thrust block, V_{TB} (for ± 1 ton thrust variation at propeller at critical speed) which are associated with these

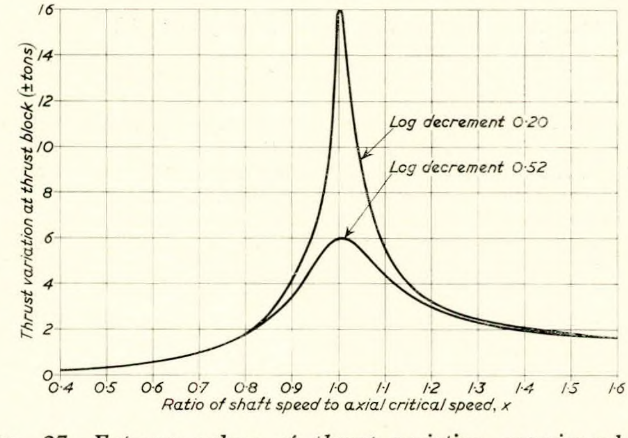


FIG. 27—*Extreme values of thrust variation experienced at thrust block when thrust variation at propeller at axial critical speed is $\pm$ 1 ton*

limits of damping; all records of thrust variation have a form which lies within the area bounded by these two curves.

It will be seen from Fig. 27 that V_{TB} is effectively independent of damping up to a value of $x = 0{\cdot}8$ for the range of damping experienced. In fact the difference between the two curves in Fig. 27 is less than 0·6 per cent at $x = 0{\cdot}8$. Provided, therefore, that one can determine the axial critical frequency from one of the higher orders, such as the eighth order excitation of a four bladed propeller, it is possible to determine the magnitude of the blade frequency excitation even though the blade frequency axial critical is not in the running range and the damping characteristics of the particular system have not been determined.

There are a number of ways in which V_{TB} may be determined of which the most accurate is the use of load cells in the thrust block. Use has also been made of a pressure pick-up in the hydraulic system of a Michell thrust meter; this is a cheaper arrangement for which facilities can be more readily obtained but it is still being examined critically to see whether damping in the hydraulic system can have a significant effect on the readings. Deflexion measurements of the thrust block can be used in conjunction with stiffness data as a less direct approach. Fig. 28 shows vibration amplitudes

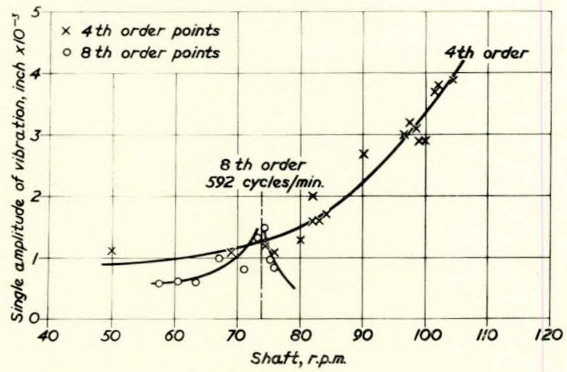


FIG. 28—*Longitudinal vibration measured on thrust block of 40,000 ton tanker*

measured on the thrust block of a 40,000 ton tanker. These have been analysed by the foregoing method and produced values for the four bladed propeller:

fourth order thrust variation at propeller ±4·4 per cent of mean thrust

eighth order thrust variation at propeller ±1·0 per cent of mean thrust

Similar readings were taken on another tanker of about the same size and power but with a five bladed propeller and analysis showed:

fifth order thrust variation at propeller ±2·7 per cent of mean thrust

tenth order thrust variation at propeller ±0·8 per cent of mean thrust

Fig. 29(a) shows the thrust variation at the thrust block of the first of these tankers, with the four bladed propeller, derived from the vibration amplitude readings. In this figure has also been shown the effect of combining the stern and propeller characteristics of the second (five bladed) tanker with the propulsion system (excepting propeller) of the first tanker. The mean thrust at 118·5 r.p.m. was 167·5 tons. It will be seen that the five bladed arrangement is a less happy one than the original four blade arrangement as far as axial vibration is concerned. Fig. 29(b) gives the corresponding data for the thrust variation at the propeller. Clearly, much more data are needed and the matter is being very actively pursued.

Although the source of excitation of axial vibration has usually in the past been the thrust variations at the propeller, the possibility of engine excitation is also a very real one. It has been shown by Kern[10] by running an engine on the

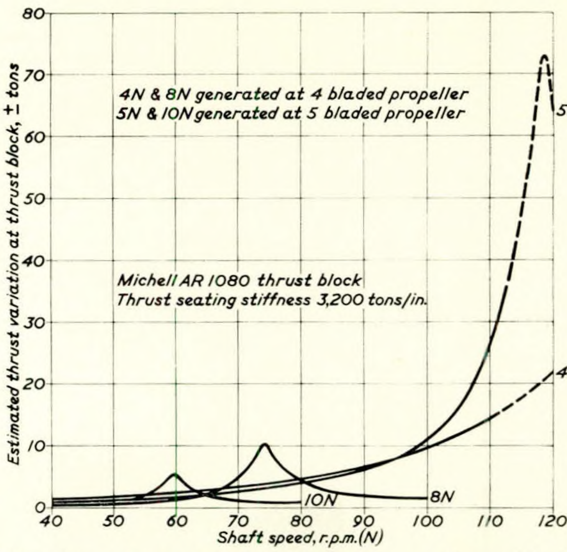


FIG. 29a—*Estimated thrust variation at thrust blocks of two 40,000 ton tankers with identical propulsion systems but 4 and 5 bladed propellers respectively (stern forms also vary)*

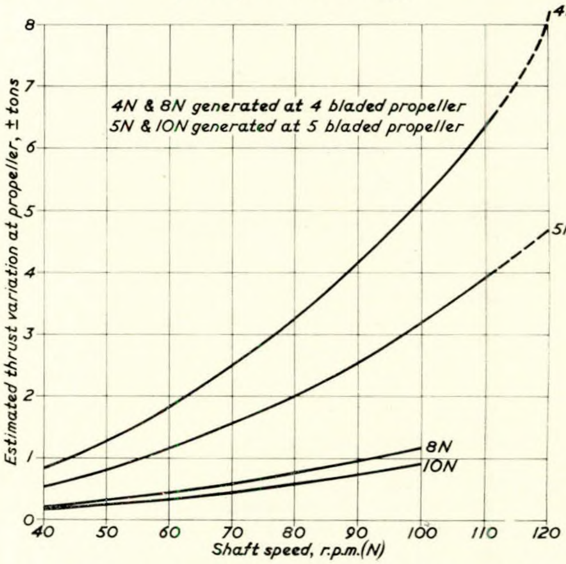


FIG. 29b—*Estimated thrust variation at propellers of two 40,000 ton tankers with 4 and 5 blades respectively (stern forms also vary)*

test bed with a thrust applied hydraulically, equal to the normal propeller thrust, that an axial critical can be produced by the engine forces alone. Whether this is solely due to the piston forces acting on the crankpins causing "nipping" and "splaying" of the crank webs, and consequent axial movement of the journals and crankshaft generally, is something which has not been clearly determined. The harmonic axial exciting forces which can be produced in this way are not large, but they may well be sufficient to produce sizable amplitudes of axial vibration at resonance. Determination of the mechanism of excitation has often been obscured by the possible presence of torsional critical frequencies coincident with the axial critical frequencies but evidence is accumulating that such coincidence of frequency is not a prerequisite for the occurrence of axial vibration. At the same time, whilst it can be shown from purely mechanical considerations that a cyclic fluctuation of torque will produce an axial fluctuation of strain at a frequency twice that of the torque, the effect is again not large; again, however, this may be sufficient excitation.

Whilst the problem of coupling between torsional and axial vibration is one which at present does not appear to be fully understood, there is general agreement that axial vibration is particularly intense when the natural frequency of axial and torsional vibration is the same.

It may well be that in cases of coupling between torsional and axial vibrations the propeller plays an important role. When a torsional oscillation of the shafting system is excited either by the engine or by wake variations, the propeller is at an anti-node and there is automatically a variation of torque at the propeller of considerable magnitude. This gives rise in turn to a variation of thrust of the same frequency, which may be very pronounced, and if the natural axial frequency of the shafting system is coincident, axial vibration of considerable severity may result. Ships with machinery aft are the most prone to give conditions where the axial and one-node torsional frequencies coincide.

In instances where axial vibration is a possibility, calculations can be made. There are a number of approximate calculation methods available; more detailed, and possibly more laborious, calculations can also be made[11] either by a method analogous to the Holzer method for torsional calculations, or else by a method of successive approximation analogous to that used for hull frequency calculations. Mechanical engineers who are well versed in electro-mechanical analogies may find it profitable to translate the problem into an electrical analogy and to use that as the starting point in the production of a computer programme. With such a computer programme available, estimates could rapidly be made of the effect of varying the physical data.

There are still gaps in our knowledge of the physical values to be inserted in axial critical calculations. However, these are receiving attention and one recent step forward has been a programme of tests made by B.S.R.A. to determine the stiffnesses of thrust blocks. These tests were done with dry non-rotating thrust blocks, whereas under running conditions an oil film is developed between the thrust collar and the pads and the pads tilt relative to the collar face. The oil film thickness is generally of the order of 0·003 to 0·004in. and should vary as the inverse of the square root of the variation of axial thrust. This will reduce the stiffness of the thrust block, but the magnitude of the reduction is not known. The behaviour of oil films under such conditions is doubtful; the results of attempts to evaluate oil film behaviour analytically cannot be considered very reliable. Some experimental work in this direction appears to be called for. Now B.S.R.A. is engaged in obtaining information on the stiffnesses of thrust seatings and associated bottom structures.

Calculations of axial critical frequencies of oil engines can be made using the Holzer method, the usual practice being to simplify the crankshaft into a system of masses concentrated at the centre lines of each of the cylinders and connected by springs whose stiffnesses are those of the portion of the shaft between each of the assumed mass positions. Thus a six-cylinder engine system would be divided up as in Fig. 30. In considering crankshaft masses, it appears correct to consider only the crankshaft itself; in particular, it seems to be reasonable to assume that the connecting rods take no part in the axial vibration and no allowances should therefore be made for connecting rod masses.

Formulæ for the axial stiffness of crankshafts have been suggested by Dorey[12] and by Draminsky and Warning[13], but experience has not shown them to have the required accuracy. Dorey introduced a factor which he justified by the presence of shrink fits and which he called a penetration factor; this factor does not appear to have either theoretical or practical justification and seems only to compensate for the ignoring of journal bending. Dorey's analysis had the merit, however, of making allowance for the angles between the crank under consideration and adjacent cranks, which appears to be ignored in the analysis of Draminsky and Warning. In recent years direct measurements of deformation under known loads have been made on a number of engine types, some of the results having been published[10, 14, 15]. In the light of these data and the results of observations of axial critical frequencies on a number of ships, it has been considered desirable to examine again the theoretical spring constants of various crankshafts and to relate these to the observed values.

For a single throw of the type shown in Fig. 31, it can be shown from simple beam theory that:

$$\frac{1}{k} = \frac{C^2}{2E}\left[\frac{b_p}{I_p}(1 - \cos\theta) + \frac{2b_j}{I_j}(1 + \cos\theta) + \frac{C}{I_w}(\tfrac{1}{3} - \cos\theta)\right] \tag{28}$$

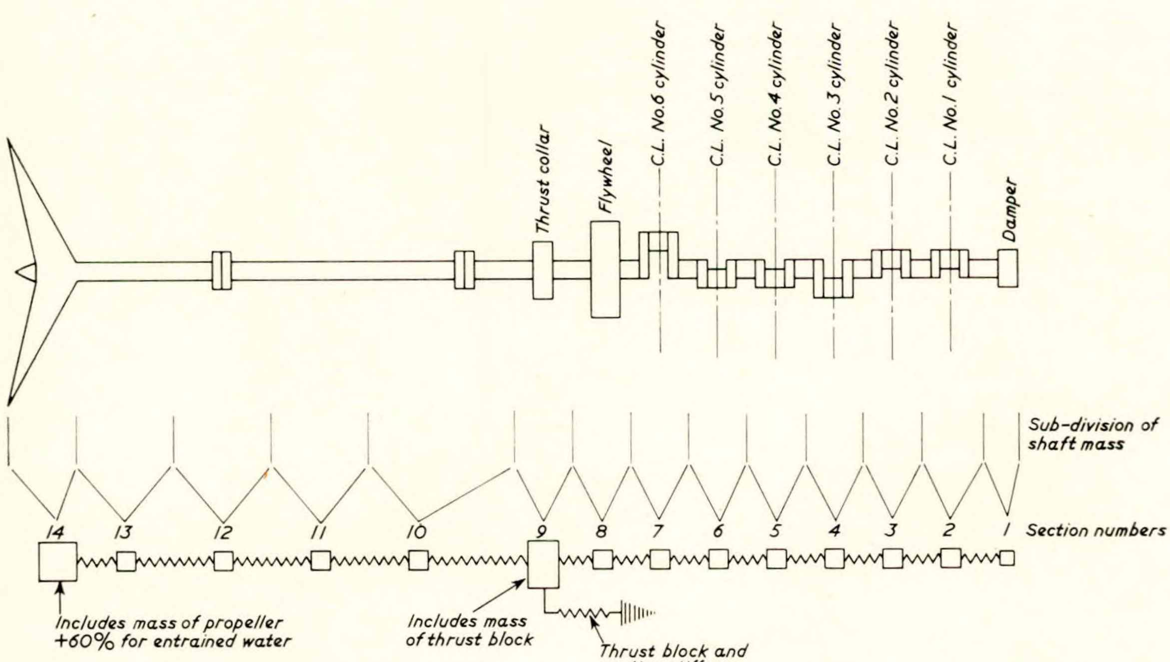


FIG. 30—*Subdivision of mass and allocation of stiffness for a 6-cylinder engine system*

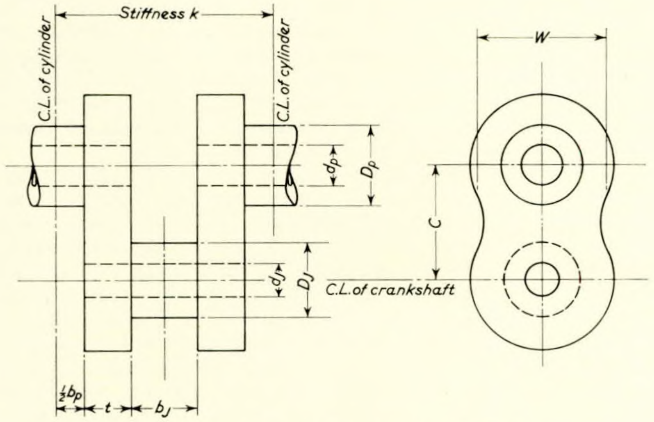


FIG. 31—*Axial stiffness of crankshaft*

where k = stiffness of centre of one crankpin to centre of adjacent crankpin

C = radius of crank throw

b_p = width of crankpin

I_p = moment of inertia of crankpin = $\frac{\pi}{64}(D^4_p - d^4_p)$ where D_p and d_p are outer and inner diameters of crankpin

b_j = width of journal

I_j = moment of inertia of journal = $\frac{\pi}{64}(D^4_j - d^4_j)$ where D_j and d_j are outer and inner diameters of journal

I_W = moment of inertia of web = $\frac{1}{12}wt^3$ where w = mean width of web between centre of pin and journal and t = thickness of web

θ = angle between crankpins

The first term within the square brackets relates to bending of the crankpins and varies from zero when the cranks are in line with one another to a maximum when the angle between cranks is 180 degrees. The second term relates to the journals and varies from zero when the cranks are at 180 degrees to one another to a maximum when the cranks are in line. The third term covers bending of the webs. No account has been taken of restraint in the form of couples at the journal bearings; with the combinations of short journal bearings and the clearances in general use this has been considered valid and the good correlation obtained between the spring constants computed from the above equation and observed values seems to indicate that journal bearing restraint is not a significant factor in crankshaft axial vibration.

A similar treatment of a crankshaft for an opposed piston engine, as shown in Fig. 32, gives:

$$\frac{1}{k} = \frac{C^2_{CP}}{2E}\left[\frac{b_{CP}}{I_{CP}}\left\{2 - R_1(1+\cos\theta)\right\} + \frac{C_{CP}}{I_{CW}}\left\{\frac{4}{3} + R_1(1-\cos\theta) - 2R^2_1(1+\cos\theta)\right\}\right] + \frac{C^2_{SP}}{2E}\left[\frac{b_{SP}}{I_{SP}}(1-\cos\theta) + \frac{2b_j}{I_j}(1+\cos\theta) - \frac{C_{SP}}{I_{SW}}\left\{\tfrac{1}{3}(1+7R_2) + \cos\theta\,(1+R_2)\right\}\right] \quad (29)$$

where k = stiffness

C_{CP} = radius of centre crank throw

b_{CP} = width of centre crankpin

I_{CP} = moment of inertia of centre crankpin = $\frac{\pi}{64}(D^4_{CP} - d^4_{CP})$ where D_{CP} and d_{CP} are outer and inner diameters of centre crankpin

R_1 = $\frac{C_{SP}}{C_{CP}}$

θ = angle between cranks

I_{CW} = moment of inertia of centre web = $\frac{1}{12}w_{CW}t^3_{CW}$ where w_{CW} = mean width of centre web between pins and t_{CW} = thickness of centre web

C_{SP} = radius of side crank throw

b_{SP} = width of side crankpin

I_{SP} = moment of inertia of side crankpin = $\frac{\pi}{64}(D^4_{SP} - d^4_{SP})$ where D_{SP} and d_{SP} are outer and inner diameters of side crankpin

b_j = width of journal

I_j = moment of inertia of journal = $\frac{\pi}{64}(D^4_j - d^4_j)$ where D_j and d_j are outer and inner diameters of journal

I_{SW} = moment of inertia of side web = $\frac{1}{12}w_{SW}t^3_{SW}$ where w_{SW} = mean width of side web between pin and journal and t_{SW} = thickness of side web

R_2 = $\frac{I_{SW}}{I_{CW}}$

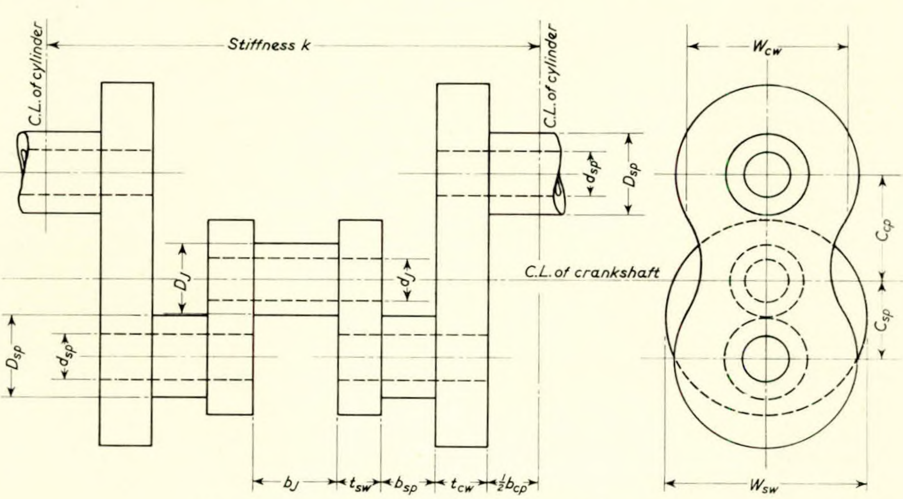


FIG. 32—*Axial stiffness of opposed piston crankshaft*

Correlation of computed stiffnesses with observed values has again been good. The authors hope that designers will comment on the validity of these relationships in their experience.

Information on the stiffnesses of integral thrust blocks is scanty and may well require to be the subject of a systematic investigation.

Remedial measures fall into two groups, modification of the exciting forces or modification of the response of the shafting system. Excitation forces can be changed either by changing the running speed or the number of blades on the propeller and, if planning a new installation, by altering the number of cylinders in an oil engine. The natural frequency of the shafting system of a turbine installation can be altered by altering the position of the thrust block; for instance, the frequency can be raised by positioning the thrust block further aft, anywhere up to the mid-length of the line shafting, rather than in the conventional position. Another possibility is to fit a hydraulic detuner(16) at the thrust block; this involves an electrically-driven pump, requires maintenance and its service reliability has yet to be proved in the merchant service. In an oil engine installation where undue axial vibration of the crankshaft is a problem, the fitting of a hydraulic damper to the forward end of the crankshaft may be a solution; this consists of a piston on the end of the crankshaft reciprocating in a cylinder attached to the engine casing which has two chambers, one on each side of the piston, which are filled with engine lubricating oil; flow between the chambers, and therefore the amount of damping is regulated by the amount of the clearance of the piston in the cylinder. It is noted that at least one Continental engine builder appears to consider such a device necessary for engines with a large number of cylinders but once again, service experience is required. An attractive possibility is a combined torsional and axial damper.

Transverse Vibration of Shafting Systems

As yet, relatively little is known about the second form of rectilinear vibration, namely transverse vibration of line and tail shafting. Because the phenomenon of shaft whirl is usually a high speed one, it should not be overlooked that a propeller shaft system subjected to considerable fluctuating bending forces may well have significant amplitudes of forced vibration and may even reach a resonant condition at relatively low speeds. The source of excitation is again as indicated in Part I, the movement of the propeller through a varying wake, the axial loads on the blades varying continuously throughout a revolution and giving rise to fluctuations of bending moment. The bending moment is most conveniently expressed in terms of its components in the vertical and horizontal planes of the shaft axis, and in single-screw ships the component in the vertical plane is generally greater than in the horizontal plane. Full scale data on the magnitude of these moments are rather scanty and the results of model tests are better regarded as indicating a trend rather than providing working data. Tests reported by Panagopulos and Nickerson(17) on the *Chryssi* provide useful data on the stresses induced by these moments although the analysis is not sound, the apparent observations of whirl and counterwhirl being in fact manifestations of an instrumentation phenomenon. However, it is a fair generalization that propellers having even numbers of blades give smaller moments than those having odd numbers, with six blades preferable to four.

Perhaps the most serious effect of large bending moments at the propeller is to be found in fatigue failure of tailshafts, mostly at the forward end of the propeller boss, in many cases in way of the forward end of the keyway. This problem has been studied by B.S.R.A. in collaboration with Lloyd's and the influence of bending stresses is set out in a paper by Batten and Couchman(18).

Among the factors which influence the critical speed of shafts is bearing length. The effect of the stern tube bearing is difficult to estimate but the longer its effective length, the higher the natural frequency of the shaft. As wear-down occurs, the critical speed is likely to come down and a ship

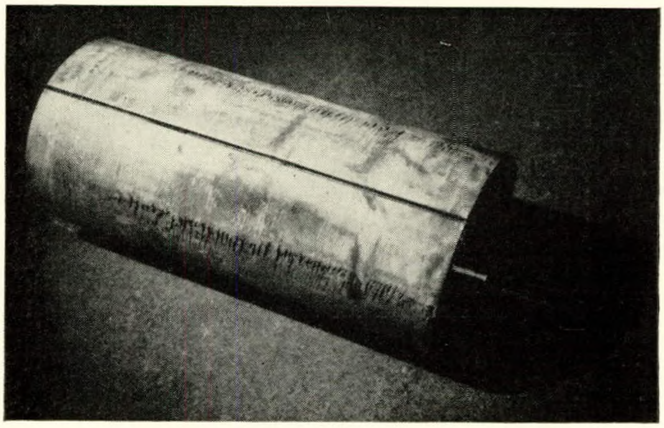


FIG. 33

which is free from transverse shaft vibration trouble when new may well run into it after a year or two in service. A variety of troubles with bearings may result from transverse vibration and it is almost certainly the cause of the deep erosion of tailshaft liners in patches which has been noted rather frequently in recent years. An example of this is shown in Fig. 33 where eroded patches can be seen each of which is in line with the gap between the roots of the propeller blades. Fig. 34 shows the erosion grooves and pits in more detail.

Transverse Vibration of Engines

A form of machinery vibration which is encountered fairly frequently is transverse vibration of oil engines. Although described in this way, it is more accurately a combination of rectilinear and torsional modes and, in analysis of the phenomenon, the engine, its seatings and the adjacent bottom structure should be treated as a body free to vibrate in a composite manner made up of these modes; it can for this purpose be considered as constrained against vertical motion. Analysis is complex and it is not yet possible to calculate critical frequencies, though empirical methods have been used by some designers which give reasonable results when applied to engines of one design fitted to structures of which the general design and scantlings are similar. Several organizations are looking into the problem, B.S.R.A. included, and in some cases electronic computers are reported to be in use. It is doubtful, however, whether this has yet reached the point of being a computational problem, as it is doubtful whether the appropri-

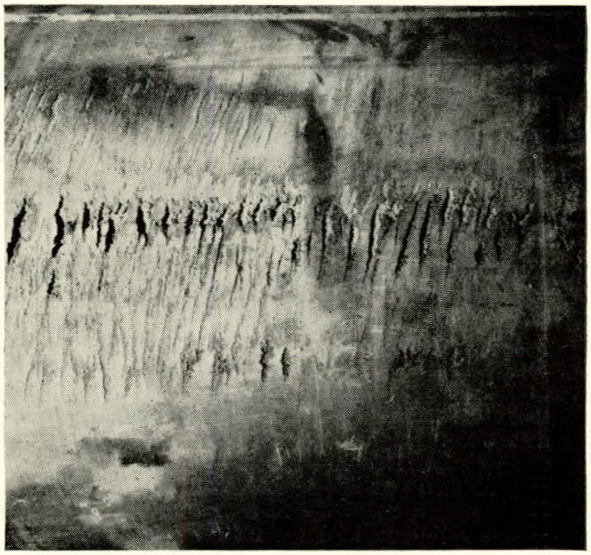


FIG. 34

ate mathematical treatments have been produced and, even if they have, knowledge of the physical data to be inserted is unlikely to be available. Among the more important unknowns is the stiffness of the horizontal bolted joints to be found in most engine casings; it may well prove that an engine can be treated for this form of vibration as a number of rigid bodies bolted on top of one another, and there may not even be a significant error in neglecting the elasticity of the bolted joints and treating the engine as a rigid body.

The forces causing transverse engine vibrations are the thrusts on the crossheads of the engine (or, of course, on the cylinder walls in the case of a trunk piston engine) and the reactive horizontal forces at the main bearings which together form couples that are equal and opposite to the torque delivered by the engine. As long as the engine delivers power or torque, these reactive couples must be present. Since the output of each cylinder varies continually with the pressure in the cylinder and the crank position, the reactive torque on the engine frame due to each cylinder will be vibratory. Vector summation of these individual reactive torques will give a diagram of the engine torque reaction over a complete revolution. The magnitude and form of the torque reaction variation will depend on such things as number of cylinders, firing order and the cylinder pressure diagram.

There are two important types of transverse vibration, one known as "parallel", in which all the cylinders are moved horizontally in the same phase, and the other known as "skew", in which the ends of the engine move horizontally in opposite directions (that is, out of phase) with a node located somewhere about the middle of the engine. Fig. 35 shows the

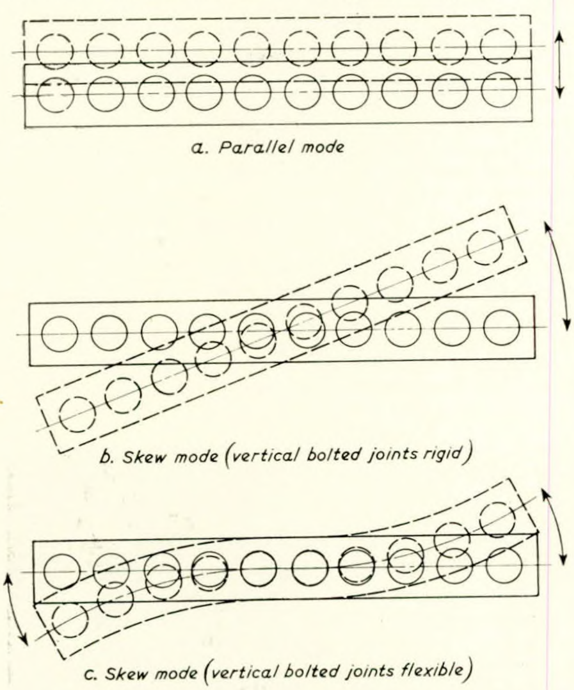


FIG. 35—*Forms of transverse engine vibration*

motions of the cylinder tops in the two forms. The resonant frequency of the skew type vibration is considerably higher than that of the parallel type and, up to now, only the latter has been significant. However, as engines become longer, the skew type resonance will decrease to a frequency where it may become troublesome, and for 10 and 12-cylinder engines attention may have to be paid to it.

In Fig. 35 two possible variations of the skew mode are shown. It has been usual up to now to consider the bolted joints between cylinders in large oil engines as rigid and to neglect longitudinal deflexion of the engine frame when considering this mode, but there is some evidence that the first of these assumptions is particularly invalid. Attention is now being given to the significance of this point both by analytical treatment and by measurements on engines. One hopes that this will not add yet another complication to the work of the engine designer but it may be that for engines with a large number of cylinders, the selection of firing order may have to take into account this additional factor.

Transverse engine vibration problems are probably most common in smaller ships particularly those with relatively light scantlings, and until more is known about frequency prediction, it is a reasonable precaution to make provision in such ships for the installation of tie-bars between an engine and the ship side. Such tie-bars can raise the natural frequency of the transverse vibration of the engine very considerably, and so get one well away from any exciting frequency.

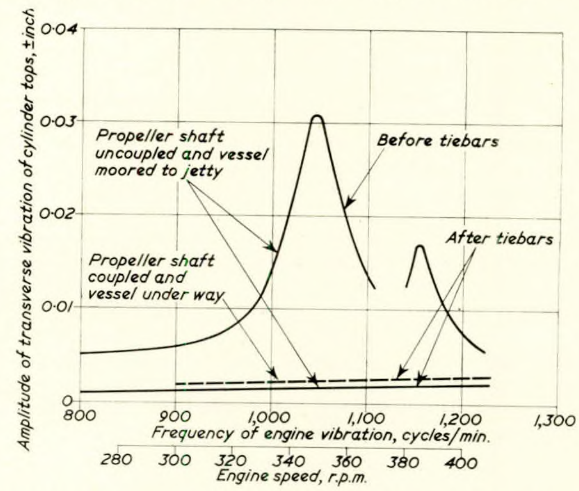


FIG. 36—*Transverse vibration of 3-cylinder engine in a tug*

Fig. 36 shows measurements of transverse engine vibration on a motor tug before and after fitting tie-bars. The upper curves of this figure show amplitudes of transverse vibration measured at the top of the three-cylinder engine with the propeller shaft uncoupled. Insufficient information is available to comment confidently on the two peaks but the one at 1,040 cycles/min. may well be the parallel mode and the other at 1,150 cycles/min. the skew mode. Angle bars were then secured transversely across the fore and after ends of the engine, and anchored securely to the decks and engine casing. The lower line shows the amplitudes then observed whilst the dotted line shows the service condition, with vibration reduced to an insignificant level.

As a variation on connexion to the ship side, there may be cases where connexion to the deckhead is more appropriate. Although the loads on these struts are not normally high, it is essential to provide good foundations at their ends. Rigidity is the aim and the anchor points should therefore be designed for maximum rigidity. When selecting the point of attachment to the engine, it is worth bearing in mind that the forces with which these struts are intended to deal are applied to the engine crossheads. Anxiety is sometimes expressed about what may result from collision, but properly designed shear pins in the tie-bars should obviate impact forces on the side of the engine.

Transverse engine vibrations, since they involve the engine seatings and major portions of the bottom structure of the ship can well have coupling effects with the main hull. Not only can transverse engine vibrations induce hull horizontal or torsional modes of vibration if the natural frequencies of any of these happen to be near to the engine transverse frequency, but the reverse process may also take place and an engine transverse critical may be excited by a hull critical which may itself be excited by propeller forces. Fig. 37 gives records on another motor tug, this time with a four-cylinder oil engine, service

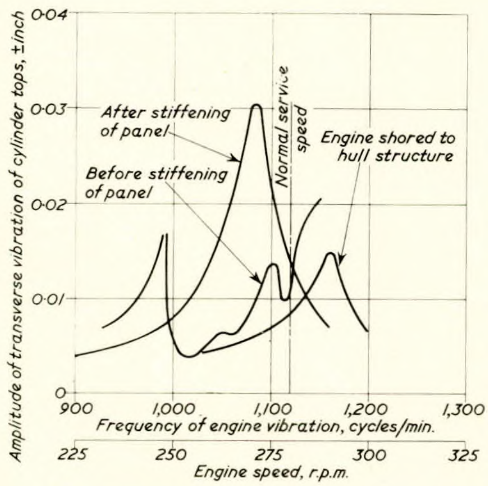


FIG. 37—*Transverse vibration of 4-cylinder engine in a tug*

speed on Fig. 37 corresponding with 1,120 cycles/min. The problem was vibration of a panel in the after cabin, the amplitudes having a peak just below 1,000 cycles/min. The engine transverse vibrations at this stage were also quite apparent but did not give rise to concern; an important point, however, was the coincidence of the resonance of the transverse engine vibration and the vibration of the cabin panel. The panel was stiffened up and the vibration of the panel reduced to a negligible level, but in the process the transverse engine vibration had become significant with a resonant peak at about 1,070 cycles/min. The vibration of the engine and its seating had involved forced hull vibration and local vibration of the cabin structure. Alteration of any one of these components will interact with the others. Temporary shoring between the top of the engine entablature and the decks reduced the transverse vibration of the engine to an acceptable level as shown by the right-hand peak in Fig. 37.

In ships with twin oil engine installations, a problem which often occurs is tuning fork vibration where the two engines vibrate horizontally like the two prongs of a tuning fork, that is, the motions of the two engines are out of phase by 180 degrees; the normal solution to the problem is the tying together of the two engine tops. Temporary bracing between the engine entablatures of two five-cylinder oil engines geared to the single screwshaft of a 407ft. ore carrier produced the marked reduction of transverse engine vibration amplitude

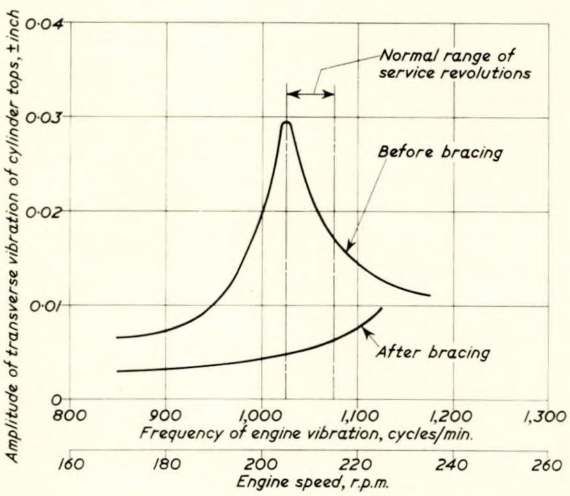


FIG. 38—*Transverse vibration of 5-cylinder engine in an ore carrier*

throughout the service revolutions which can be seen in Fig. 38. The temporary bracing members consisted of five $8\frac{1}{2}$-in. × $2\frac{3}{4}$-in. section timber planks wedged between the engine entablatures; they were necessarily in compression and careful observations were made to determine the linear displacement between the engines caused by the compressive forces. At the level of the cylinder tops the effect was to force the engines apart by 0·026in. a value quite small compared with the relative movement due to unrestrained vibration of 0·118in.

Tuning fork vibration is an intermittent phenomenon. As the relative crankshaft positions of the two engines change by $\frac{360}{n}$ degrees (n being the number of cylinders per engine) the torque reaction forces change phase 360 degrees and the corresponding vibration amplitudes vary from a maximum to zero and back to a maximum. In the case of six-cylinder engines, for example, a speed difference between engines of 0·01 of a revolution per minute will change the amplitude of tuning fork vibration from zero to a maximum in 8·3 minutes. This kind of rate of change allows amplitudes to be built up and can be most annoying. It has been suggested that some device might be produced to control the relative positions of the crankshafts of the two engines so that the torque reaction forces were in phase but this does not appear a practical proposition.

Effect of Oil Engine Unbalance

Oil engine builders normally balance their engines on the assumption that they are resting on a uniform horizontal platform. In this case the point of application of the unbalance forces or couples is not relevant. In a ship, however, the unbalance of each cylinder line has a different effect depending on the relationship between the point of application of the unbalance force and the vibration profile of the ship. Put as simply as possible, the effect of the unbalance of each cylinder is proportional to the ordinate of the vibration deflexion profile at the line of the cylinder. Deflexion profiles have already been discussed in Part I; examples of these are shown in Fig. 39. It is no longer sufficient to make a vector summation of the forces at each cylinder to determine the resultant force, but instead the unbalance force at each cylinder is multiplied by the ordinate of the vibration profile at the point of application taking due account of the ordinate signs and a vector summation is made of these products. It will readily be seen that a totally different resultant is obtained when the engine straddles a node than when it is centred at

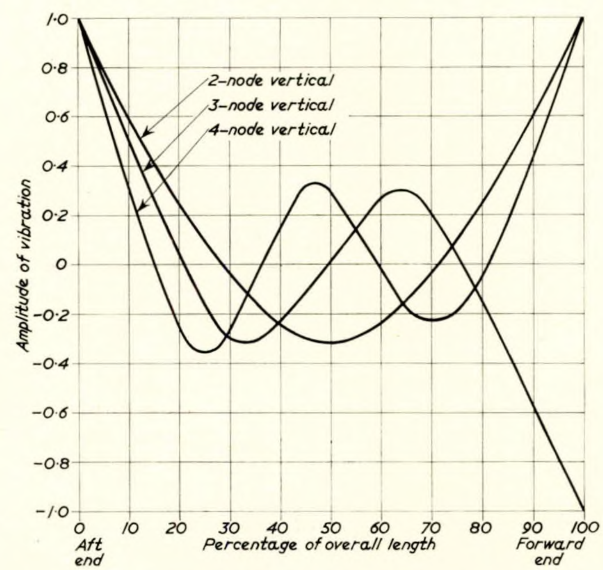


FIG. 39—*Amplitude profiles of vertical modes of hull vibration*

an anti-node. For convenience the ordinate at the end of the ship is taken as unity, so that the vector sum obtained is in fact the value of the force, which, if applied at the end of the ship, would give the same amplitude or acceleration at any point of the ship as would be produced by the engine in its actual position. The subject has been treated very fully in a recent paper[(2)].

It is not proposed to consider here the fundamentals of engine balancing but rather to make observations on a few points.

By increasing the magnitude of the balance weights used to counteract centrifugal forces, some or all of the primary inertia force of each cylinder line may be balanced out. The difficulty with this procedure is that a new horizontal unbalance is introduced, but it is usual to make a compromise in order that the maximum horizontal and vertical loads on the bearings will be approximately equal. Although the inertia forces cannot be eliminated in a single-cylinder engine it may be possible to cancel some or all of them in multi-cylinder engines by the proper positioning of the cranks of the various cylinders. Here, of course, due note must be taken of the deflexion profile of the ship in way of the engine. By using the method outlined earlier of taking the product of force and ordinate of deflexion profile, there is no need to consider separately the couples produced by the inertia forces acting axially along the crankshaft and the whole analysis can be made on the basis of the forces at each cylinder line.

Since the frequency of the secondary inertia forces is twice that of the engine, it is generally more difficult to reduce their unbalance although improvements sometimes result from a change in the firing order.

Engine manufacturers are inclined to use as the criterion of balancing the magnitude of the unbalance forces and couples irrespective of whether they are primary or secondary. In some cases one has a choice whether the residual forces or couples of significant magnitude are primary or secondary as, for instance, in a five-cylinder engine where the two alternative firing orders commonly in use give opposite results. It is suggested that it is sounder policy to select primary unbalance conditions for main propulsion machinery rather than secondary. There are two reasons for this. The first reason is that secondary unbalance, being at twice the shaft frequency is liable to excite higher modes of hull vibration than would be excited by primary unbalance; in the higher modes the deflexion profiles are less clearly predictable and the likelihood of adverse positioning of the engine is much increased; furthermore, draught changes, which alter the deflexion profile, will accentuate this problem, in the higher modes. The second

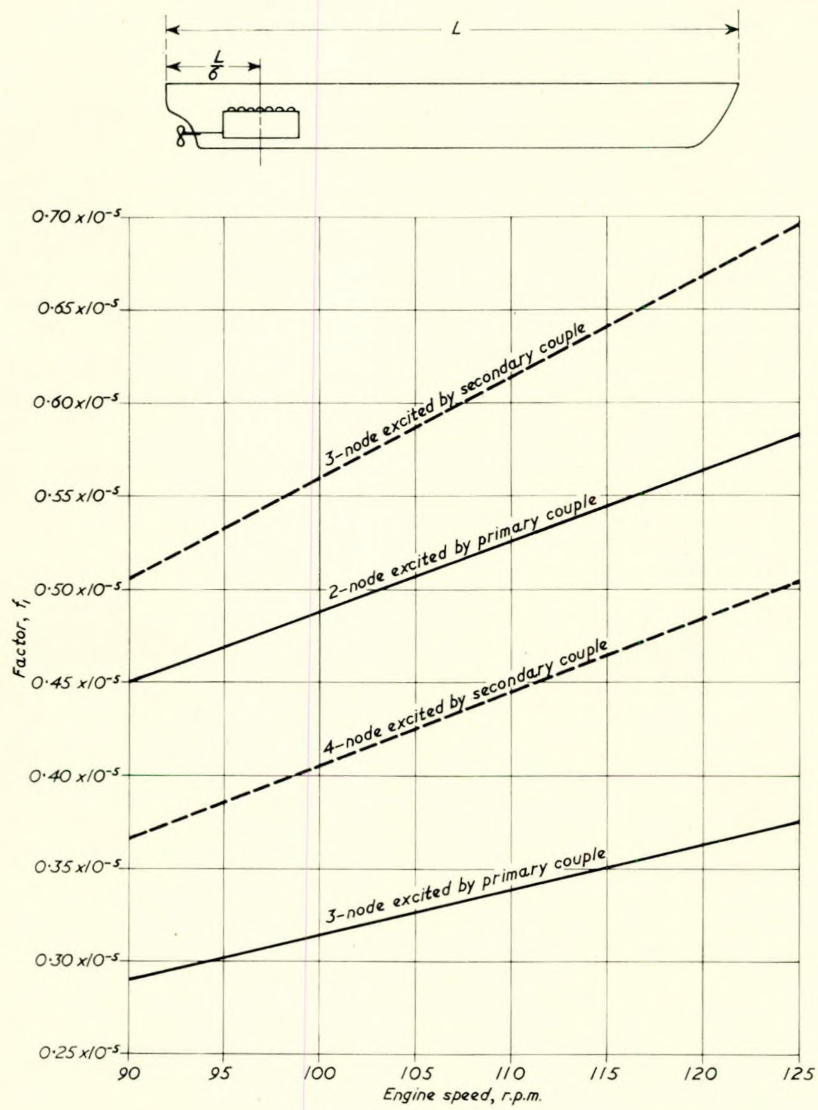


FIG. 40—*Tanker or bulk carrier with engine at 1/6 × (length of ship) from aft end*

reason is that, if trouble is experienced on installation, it is much easier to arrange a balancing force at shaft speed than at twice shaft speed.

The tolerable level of unbalance is obviously a function of the ship dimensions and the position of the engine but two typical examples may be of interest. Nowadays, with large, slow speed oil engines generally turbocharged, it is unusual to encounter resultant out-of-balance forces in engines, the unbalance being usually in the form of a couple. The following examples refer, therefore, to the maximum values of unbalanced couples which may be tolerated in certain circumstances. From the treatment of Johnson, Ayling and Couchman[(2)] it can be shown that the tolerable unbalance couple M (tons ft.) can be expressed

$$M \not> \frac{L}{i_m}(2N_c + 1{,}000)\ (C_1/35{,}200)\ (\Delta_1/K_m)$$

where L = length of ship (ft.)
i_m = slope factor, depending on mode of vibration and position of engine
N_c = critical frequency for mode of vibration concerned (cycles/min.)
C_1 = factor depending on mode of vibration
Δ_1 = total virtual mass (tons)
K_m = dynamic magnifier at resonant frequency N_e

This expression may be written

$$M \not> f_1\, L\, \Delta_1$$

or $M \not> f_1\, L\, \Delta\, (1{\cdot}2 + B/3d)$ for vertical vibration

where f_1 = a factor depending on position of engine, mode and resonant frequency of vibration, and assumes least favourable deflexion profile is experienced.
Δ = displacement (tons)
B = beam moulded (ft.)
d = mean draught (ft.)

Consider now a common type of ship at the present time, namely, a tanker or bulk carrier where the centre of the engine is at a position $1/6 \times$ (length of ship) from the aft end, the ship being of all-welded construction. Fig. 40 shows the appropriate values of f_1 to a base of engine r.p.m. for vertical hull vibration of the two, three and four-node modes. This chart is used by selecting the engine speed at which the critical frequency of a particular mode is predicted. For example a

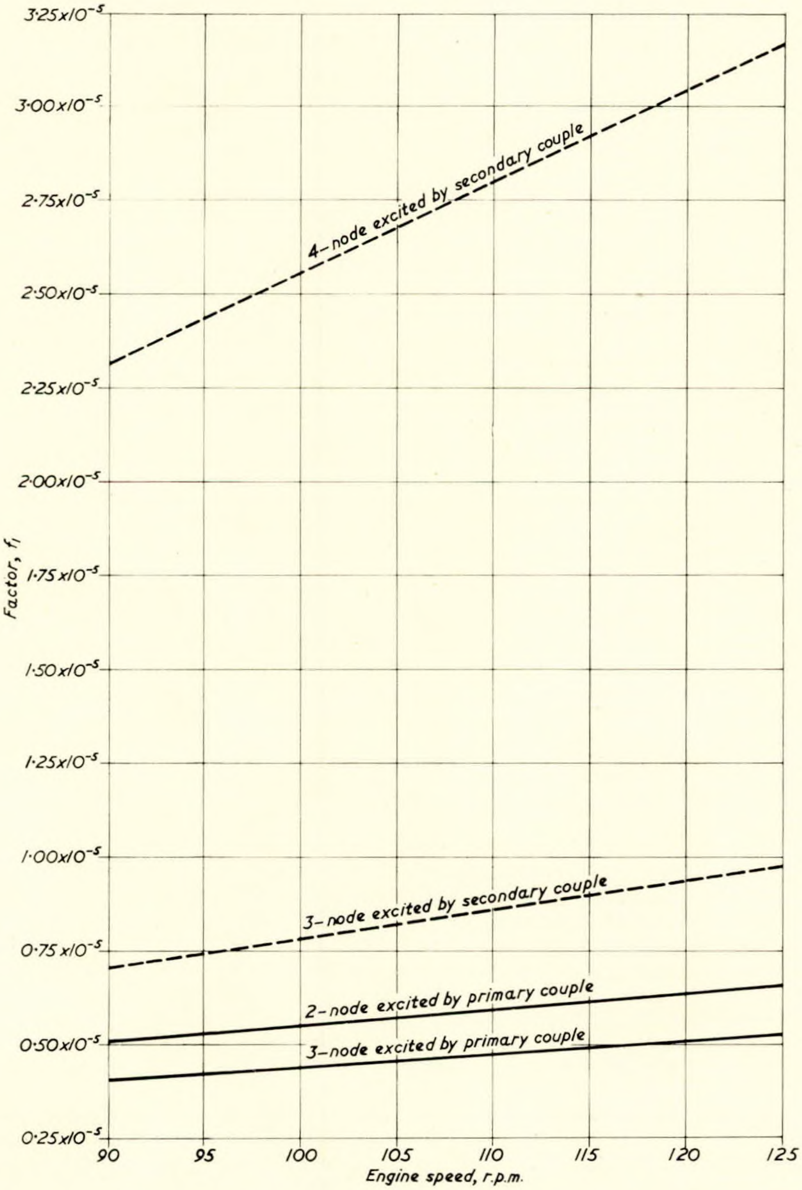


FIG. 41—*Dry-cargo ship with engine at 1/4 × (length of ship) from aft end*

tanker 525ft. long, 69ft. beam, 29ft. 6in. draught and 22,300 tons displacement might well have a four-node vertical critical frequency of 238 cycles/min. which would be excited by a secondary couple at an engine speed of 119 r.p.m. The corresponding value of f_1 would be $0{\cdot}480 \times 10^{-5}$ and the maximum tolerable secondary couple would then be:

$$0{\cdot}480 \times 10^{-5} \times 525 \times 22{,}300 \left(1{\cdot}2 + \frac{69}{3 \times 29{\cdot}5}\right)$$

$= 110$ ton ft. at 119 r.p.m.

Similarly, a tanker 725ft. long, 97ft. beam, 41ft. draught and 65,500 tons displacement might well have a three-node vertical critical frequency of 108 cycles/min. which would be excited by a primary couple at an engine speed of 108 r.p.m. The corresponding value of f_1 would be $0{\cdot}333 \times 10^{-5}$ and the maximum tolerable primary couple would then be:

$$0{\cdot}333 \times 10^{-5} \times 725 \times 65{,}500 \left(1{\cdot}2 + \frac{97}{3 \times 41}\right)$$

$= 315$ ton ft. at 108 r.p.m.

Another contemporary ship type is the dry-cargo ship with the centre of the engine at a position $\frac{1}{4} \times$ (length of ship) from the aft end, the ship again being of all-welded construction, and Fig. 41 shows the appropriate values of f_1. A vessel of this type 500ft. long, 62½ft. beam, 14½ft. draught and 7,500 tons displacement in ballast condition might well have a four-node vertical critical frequency of 224 cycles/min. which would be excited by a secondary couple at an engine speed of 112 r.p.m. The corresponding value of f_1 would be $2{\cdot}88 \times 10^{-5}$ and the maximum tolerable secondary couple would then be 285 ton ft.

Consider further the case of the foregoing ship with a four-node vertical critical at 112 r.p.m. and examine the effect of moving the engine position. Fig. 42 shows how the

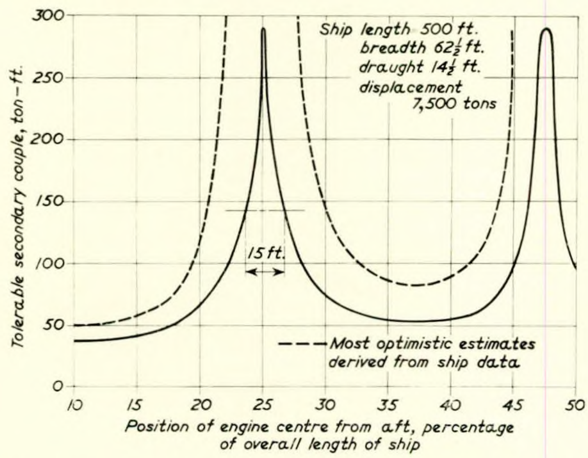


FIG. 42—*Effect of engine position on tolerable secondary couple in a 500ft. dry-cargo ship (ballast condition) having a 4-node vertical critical near service speed*

tolerable secondary couple would vary as the engine centre moves over the range $L/10$ to $L/2$ from the aft end. The effect of moving the engine a few feet forward or aft from the $L/4$ position can be seen; moving the engine some 7½ft., in either direction will halve the tolerable couple.

The foregoing estimates have all been derived from the least optimistic use of the data regarding deflexion profiles, which is considered to be a reasonable approach when considering the effects of unbalance in a general way. However, data from similar ships may be considered in more detail when examining a specific installation proposal and may well give more optimistic results. The dotted lines on Fig. 42 show the limits of more optimistic forecasts which might be made.

ACKNOWLEDGEMENTS

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Discussion

MR. P. JACKSON, M.Sc. (Member of Council) said that the paper was an excellent one, well written and easy to understand, without too much mathematics; he considered it one of the best papers which had come from B.S.R.A., incorporating, as it did, a great deal of data from many ships.

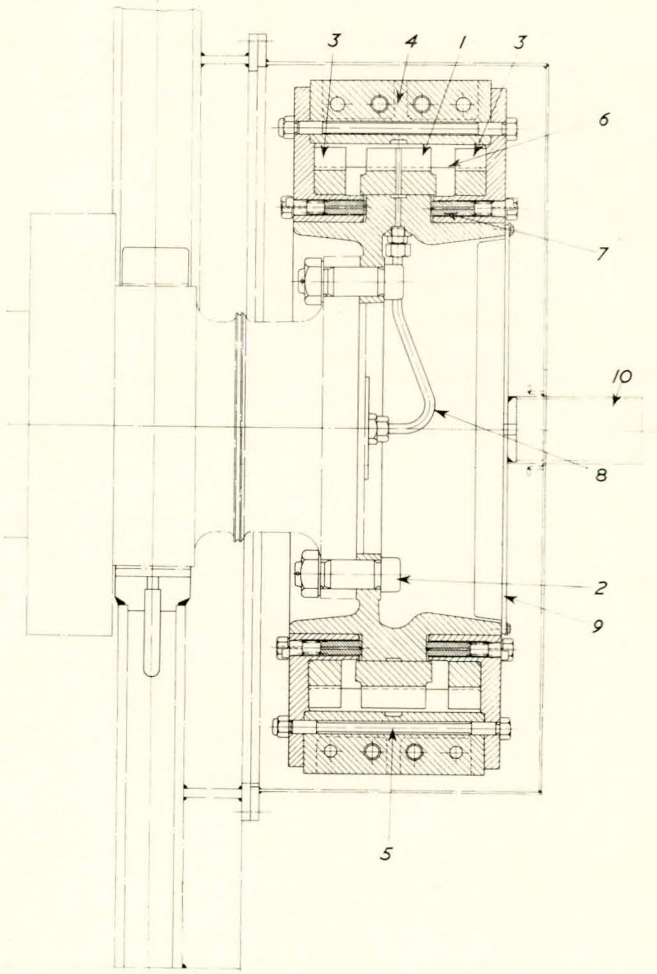
However, the authors would not expect him to agree with everything.

Dr. Johnson had said that the paper was in two parts, dealing with the hull and with the engine forces. Well, why not? The ship was the bell and the engine was the clanger. The first part of the paper dealt with ship frequencies and amplitudes and many examples were quoted from which much data had been gathered. It was not possible to calculate complex structures like a ship, with its varying sections and loadings, from first principles; one had to assume a form of a symmetrical nature and then make corrections such as had been done in the paper by the substitution of a virtual mass.

One omission was that the paper did not give the differences in the natural frequencies of the ships between the loaded and the unloaded condition, which could be as much as 5 to 7 per cent, but it did deal with a point which he had not appreciated before, namely, the effect of the depth of water and also the difference of the effect of the surrounding water on the frequencies of the horizontal and the vertical modes of vibration. Much the same thing happened with stationary engines. He had known cases where there was no vibration around an engine but half a mile away pots were being shaken off the shelves of houses.

The forces which caused these vibrations arose from the engine, the propeller, or the "wake" and sometimes from waves or wind. The first part of the paper dealt with the effect of the propeller at considerable length and justifiably so. More and more vibration problems were caused by the propeller, particularly when running astern. Fortunately it was not necessary to run astern very much, but vibration caused by the propeller was often the limitation to the speed that could be achieved astern. The early engines produced by his company would give only about half power astern and there was some criticism with regard to this limitation although all turbine ships could do no more than half speed astern. With the new injection system full power could be given when running astern, but no ship had yet been engined which could take the vibration arising from the propeller when running only 85 per cent speed astern. Vibrations arising from the propeller could be serious, but he had not yet met a case of vibration which had been attributed to "wake".

The formula for acceptable amplitudes was a very simple one. He recalled making a contribution to the discussion on a paper* to the Diesel Engine Users Association in 1931 in which he gave a curve of acceptable amplitudes based on practical cases which he had encountered. The human frame seemed to be very susceptible to vibration frequencies of around 600 cycles/min. At much lower frequencies vibration could be pleasant, but at much higher it could be painful. There were vibration machines in some Continental parks where for the equivalent of sixpence it was possible to stand on them and have your feet vibrated which relieved tiredness. This was a vibration of about 0·002in. at 1,500 or 3,000 cycles/min, he was not sure which. The authors' formula gave about 6 thou.



1) Fixed wheel toothed rim
2) Coupling bolts
3) Floating wheel toothed rims
4) Cast iron outer rim
5) Floating wheel bolts
6) Spring leaves
7) Axial location plungers
8) Oil supply pipe
9) Beam for pulley support
10) Pulley for torsiograph recordings

FIG. 43—*Arrangement of detuner*

* Jackson, P. November 1931. Contribution to discussion on "The Elimination of Vibration" by R. B. Grey. Diesel Engine Users Association, Vol. S102-109.

as the permissible vibration at 600 cycles/min. and he fully agreed with this with regard to individuals working and going about their daily duties, but half that amplitude would prevent a man going to sleep at night.

With regard to Part II of the paper, torsional vibration problems were now well known and all engine manufacturers could calculate their frequencies with a high degree of accuracy. The calculation of stresses was rather more complex and now with turbocharged engines increasing in power there would have to be more research into the higher harmonics of the tangential effort. He did not know of any reliable coefficients above about 140lb./sq. in. m.i.p., whereas many engines were now being rated higher than this and harmonic orders up to the 12th would have to be taken into account.

He had not found axial vibration to be as serious as Mr. McClimont had outlined. He had not met a case of crankshaft breakage due to axial vibration, but it was a contributory factor in the case of the 750 mm. bore engine some years ago. In addition there was torsional vibration and misalignment, together with the fact that the design of the crankshaft gave a concentrated stress at one particular point. Axial vibration itself was relatively sharply tuned. It was also very easy to damp.

Mr. McClimont had suggested a combined torsional and axial damper. Mr. Jackson said that his company actually had a combined torsional detuner and axial damper, the idea being to damp the axial vibration by a hydraulic restriction between the fixed mass and the floating mass, oil being squeezed through small holes in the location plugs (7), shown on Fig. 43. The amplitude due to axial vibration caused by the propeller could have an amplitude of 3 mm. on the forward end of the engine. Axial vibration frequencies could now be calculated with some degree of accuracy and he believed that their Dr. Orbeck had a computer programme for axial vibration frequencies and stresses.

With regard to the case illustrated in Fig. 29b where a four bladed propeller was shown as being better than a five bladed propeller, this was due to the 5th order critical being near to the running speed, whereas the 4th was removed. He believed that in a discussion* at the Institute in 1960 Ir. A. Meijer of Wilton Fijenoord had shown an example of a reverse case where a five bladed propeller was better than one with four blades.

He agreed with the authors that axials were either actuated by the propeller or by a torsional which was very close to the critical frequency of the axial. He could not admit that Fig. 32 represented a typical opposed piston engine crankshaft and he therefore presented Fig. 44 showing the corresponding section of the new J-engine crankshaft which was much more rigid.

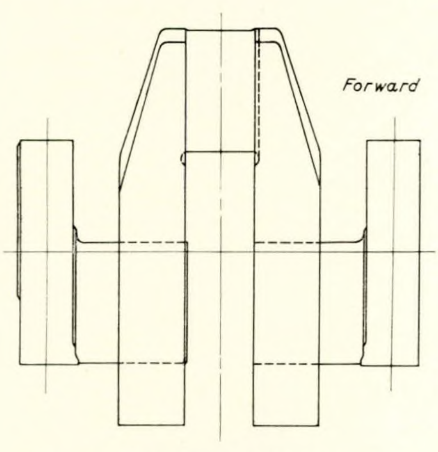


FIG. 44—*One cylinder section of J engine crankshaft*

* Reference (14) of the paper.

Transverse vibrations were very rare and were so easy to remedy that little account was taken of them. Generally one fitted tie-bolts, or alternatively, practically all engines had the upper grating fixed either to the engine and loose at the bulkhead or *vice versa* and once or twice he had put bolts into both sides to act as a restraint, though it was very rare that such a case was encountered.

The authors mentioned synchronizing twin engines. This had been done. Mr. Charles Day once synchronized two four-cylinder engines at Harrods store—by means of a differential mechanism on the governors. He kept them running so that the secondary forces of the two engines were opposed and so balanced each other.

As Mr. McClimont had said, unbalanced forces were not common these days and he further remarked that it would be better, on five-cylinder engines, to have a firing order that would give good secondary balance but not such a good primary balance. Mr. Jackson would have agreed with Mr. McClimont if that were the only consideration, but one had to consider the turbocharger arrangements and the firing order that gave good primary balance but bad secondary balance had been considered the best arrangement for the turbochargers some two or three years ago. Brown Boveri could now offer to turbocharge the other firing order which gave really good secondary balance on a five-cylinder engine.

When his company were considering their five-cylinder P-type engine some two years ago, he found that the out-of-balance secondary couple with a firing order 1-5-3-2-4 was about 330 tons-ft. and he had wondered whether it would cause vibration. He knew that the company had had vibration trouble arising from secondary couples in the days of Mr. Keller in 1926, but he found that the couple causing it was well over 1,000 tons-ft. He had consulted one or two other people and all had come to the conclusion that at 330 tons-ft. they were safe. However, it was not long before he heard of two seven-cylinder engines in Holland where the crankshafts were being changed and he was able to obtain from the designer some data from which he came to the conclusion that the trouble was due to a couple of only 500 tons-ft. Then, very shortly after that there was news of two ships on the Tyne with vibrations from couples of about 230 tons-ft. and he had thought that he was really in trouble. Fortunately it turned out not to be so. The actual 4-node frequency of that ship was 208 cycles/min. light and 194 when loaded, also the vibration at 97 r.p.m. when loaded could be noticed, but it was so sharply tuned that one got through it very quickly and it had never caused any complaint.

The engine designer had to make engines for all classes of ships under all circumstances. He was very interested in the authors' method but it was not really applicable to the first design of an engine. The couples had to be made as small as possible and in these days he, Mr. Jackson, aimed at couples of under 200 tons-ft.—his company's 5-cylinder engine with a firing order of 1-3-5-4-2 complied with this as did all their other engines.

With regard to the curing of these vibrations, Lloyd's on one occasion had cured a vibration from a primary couple by putting a revolving weight on the intermediate shaft to create an equal and opposite amplitude to that which was causing the trouble. Up to that date he had always taken the view that one could not cure the vibration from a couple by a force, but Lloyd's had done this. It was fortunate that the horizontal force created by the revolving weight had not caused another vibration.

He was very interested in the method which Mr. McClimont described on the last page of the paper and in Fig. 42, for determining that an engine installation would not cause vibration. He hoped that this method would be widely adopted since his company's engine, being the best balanced engine would show up well, since it had smaller unbalanced couples than any single piston engine of equivalent type.

There were so many modes of vibration that he sometimes wondered how they succeeded in building ships free from vibrations, but they did. He again thanked Dr. Johnson and Mr. McClimont for an excellent paper.

MR. T. W. BUNYAN, B.Sc. (Member) underlined Mr. Jackson's remarks about the quality of the paper. There were, however, a few points that he wished to raise.

Fig. 14 on page 128 was a most interesting collection of valuable data on various types of propeller and stern arrangements. Would the authors confirm that the values given were in fact for the same wake variation at the propeller disc for each stern arrangement, or that an attempt had been made to correct this? If this were so the value of that diagram was very considerable. Indeed, Fig. 14 put up a very good case for the six bladed propeller in relation to hull and tailshaft vibration. The performance of the six bladed propeller was probably, if anything, slightly better than the four or five. He understood from the "bush telegraph" that the mammoth tankers were raising the tailshaft problem again, so anything that would reduce transverse bending stresses was of considerable interest and value. The six bladed propeller was certainly one of these, and, indeed, some had been fitted. The oil gland bearing was another. He was sorry that the authors did not mention this in the paper. He mentioned it now for what he considered to be three good reasons. The first was that the vibration characteristics of a tailshaft running in an oil lubricated stern bearing were known at the time of the trials and would not alter in service because the rate of wear down was insignificant. It had happened in so many cases that, due to wear down and bell mouthing of the lignum vitæ bearing, a vicious tailshaft vibration had developed in a matter of two to three months' service. The authors referred to this problem on page 139 of the paper.

The second point was that the inherent damping of the oil lubricated bearing—which had clearances of $\frac{1}{1{,}000}$ in./1in. diameter of the shaft—ensured a very smooth running tailshaft.

The final point in this connexion was that due to the absence of any significant wear down there was little danger of straining coupling bolts, fretting up coupling faces and hot plummer block bearings which often occurred as a result of wear down of lignum vitæ bearings in large tankers which had stiff and relatively short shafting.

The slope boring of the stern tube was good practice whether oil lubricated or water lubricated bearings were used as this process ensured a much more favourable load distribution along the bearing.

On the matter of hull vibration generally, it appeared that it was the propeller excited modes of vibration that were likely to be a nuisance in ships over 400ft. in length. The bad old days when structural damage, particularly in the region of the after peak of tankers, due to propeller excited vibration, took place, were a thing of the past. At one time, due to tight aperture clearances and indifferent propeller design, these forces were considerable and he personally had been associated with problems where they had finished up with 70 tons of material going into the tanker to stiffen up the structure.

The authors would, he thought, confirm that the vibration amplitudes given as acceptable levels in their formulæ 11 and 12 on page 126 were in fact seldom exceeded in modern tonnage.

Fig. 15 was a typical case in point. The propeller excited modes were acceptable, but were this ship driven by, say, a nine-cylinder Diesel engine running at 105 r.p.m., the 4-node vertical mode would be excited at the running speed. He had limited recent experience of hull vibration excited by engine forces or couples in large tankers, in which the large bore engines were now becoming a popular drive, but such first hand experience as he had, indicated that the vibration excited by these forces was no problem. What was the authors' experience in this field?

Mr. Jackson had mentioned an engine balancing problem in which a strong vertical hull vibration was excited in the running speed range and had been completely cured by Mr. Bunyan, who fitted a balance weight on the intermediate shafting. Mr. Jackson inferred that it might have been fortuitous that a transverse mode of hull vibration was not excited by the balance weight. Mr. Bunyan assured him that this possibility was, of course, considered and hull vibration measurements were first made to confirm that this possibility would not occur in the running speed.

MR. E. J. NESTORIDES said that of the very wide field of problems covered by the authors he would attempt in his contribution to spotlight some of the aspects which in his view required further research.

Part I of the paper dealt with hull and propeller vibrations and, although he did not propose to enter into their discussion, he noted that Fig. 6, page 123, referred to engine unbalance and that primary and secondary forces and couples excited hull 2- and 3-node horizontal vibrations and 2-, 3- and 4-node vertical vibrations.

It would be interesting to know for various engines what the engine unbalance forces and couples were, in tons and tons-feet, in comparison with the maximum transmitted torque in tons-feet.

These excitations were transmitted to the hull via the supporting arrangements of the engine; hence the height of the attachment positions and the transmissibility of the supports were factors to be considered. He wondered whether, among the various types of supports used in marine practice, there were some which gave reduced transmissibility of unbalance forces and couples. It was realized that a complete answer to this would require vibration velocity measurements both on the engine and on the hull to determine attenuation due to the supports, but it might indicate which engine support designs were most suitable. The possibility of incorporating some form of damping in the engine supports might also be something to consider, particularly for vertical vibrations, for which the corresponding dynamic magnifiers (see Fig. 8, page 126) were between 175 and 45 and were therefore extremely high in comparison with those encountered in stationary installations.

Part II dealt with torsional, axial and bending vibrations of the propulsion system. Regarding strain gauges, he could not see justification for ± 10 per cent accuracy, indicated on page 133, without mention of the corresponding level of measured stress. For instance, it could be shown that good readings were obtainable down to, say, ± 500lb./sq. in. in field work (or half this value in laboratory work). He believed that this should be sufficient accuracy for vibratory stresses. The use of strain gauges for transmitted torque measurements was, however, another problem and it was quite conceivable that in such cases specially designed torsionmeters with their own flexible shafts might be more effective. For vibratory stresses, however, he would be inclined to rely on amplitude-modulated strain gauge readings just as much as on readings of other types of instruments or pick-ups. All required calibration.

He fully agreed with the authors that careful consideration should be given to the position and vibratory behaviour of gear drives. Branched system calculations were tedious, but made it possible to take account of all relevant auxiliary drives and to determine whether or not a nodal position in the simplified straight system was also a node in the complete system. Such calculations, which might save considerable subsequent trouble, could be speeded up with the aid of electronic computer programmes.

Axial vibrations of crankshafts were a problem which was also acquiring some importance in stationary engines, although it was not yet established whether in such engines the vibrations were purely axial or coupled to some extent with flywheel flap and bending or torsional vibrations of the shaft system. It would be most valuable to have results of static axial stiffness tests for thrust blocks. Reliable values for thrust block stiffness might also help to clarify the question about the omission of the connecting rod masses in frequency calculations. It seemed that good values of natural frequency were obtained by omitting these masses for engines with self-aligning bearings and by including these masses in engines with non-self-aligning bearings. The analysis of Draminsky and Warning* was published some time ago; he believed that it contained some misprints, which made it difficult to check its derivation. He would be glad to have a full derivation of this formula, if it was available.

The inclusion of crank angle in formulæ for axial stiffness appeared very logical. It would be interesting if the authors could add some typical examples of calculated values and values obtained from static stiffness tests of crankshafts.

The difficulties with transverse crankshaft vibrations were not only in the calculations but, again, in the basic assumptions to be made. Did one calculate the single-span mode for a sub-system between two successive bearings, or for the entire crankshaft, assuming sufficient intermediate-bearing clearances, or for both cases? Did one take the mean of the in-plane bending stiffness of the crank throw and the stiffness at right angles to this plane, for simplicity? What was the effect of the rigidity of bearing supports on these types of crankshaft deflexions? The authors' views on these aspects would also be appreciated.

Increased attention was being given to vibrations of the entire engine. It was well known that it could have three rotational motions (yaw, roll and pitch) and three linear motions (shimmy, vertical and horizontal bounce). Exploratory tests with vibrometers indicated that the centre of yaw was not at the centre of symmetry of the engine (as indicated in Fig. 35, page 140, but somewhat further back, towards the flywheel. To explore the entire surface of a medium size engine was a considerable undertaking, but if the vibrometer indications at various levels and positions along, across and on the top surface of the engine were harmonically analysed, for instance with an electronic frequency analyser, they should indicate not only the modes and frequencies of the vibrations, but also the effects of load and speed. In this manner it should also be possible to determine whether primary or secondary unbalance was the important factor. He felt that it was not possible at present to legislate on this point, since much depended on the number of cylinders, the in-line or Vee-type arrangements, and the number of connexions to the engine supports, which should preferably be as numerous as possible. Such detailed explorations of engine crankcases could furnish valuable indications for further design-stage improvements of engine crankcase stiffness and might counteract the previous tendency to consider solely the crankshaft balance weights, without detailed consideration of the overall vibration conditions of the engine and other possible solutions.

The authors were to be complimented on their paper, which was an eminently readable survey of various aspects of engine and structural vibrations.

Mr. H. Lackenby, M.Sc. said that with the increasing speed and power of ships over the years vibration continued to be a matter of concern to both shipbuilders and shipowners. With these changes, which included developments in ship design as well, experience showed that new problems were continually cropping up which kept the Vibration Section at B.S.R.A. in a high state of activity. For these reasons the paper, which gave a review of the present position of the subject, should be of considerable interest to those who became involved in these problems.

Having made this general comment there were a few points of detail that he wished to raise. As explained in the paper, for the first few modes of main hull vibration one had either to avoid resonance or, if it could not be avoided, find out whether the amplitude would be acceptable. If it was not acceptable then one had to do something about reducing the cause of excitation. Unfortunately, in many of the cases B.S.R.A. were involved in it was often too late to carry out the best cure as the die had already been cast, so to speak. This underlined the prudence, of course, of looking into these matters in the early stages of design before things had become too set, as had already been mentioned in the discussion.

* Reference (13) of the paper.

In connexion with avoiding resonance he thought it should be clearly understood that for a ship of given dimensions, scantlings and design there was nothing much which could be done about altering the natural frequencies of the hull; this was part of the personality of the ship and had to be lived with. Having said that, however, it was of interest to mention that a case arose recently where a builder had at least looked into the possibility of altering the natural frequency of a ship already built rather than doing something to the engine. It was a medium sized bulk carrier and it was pertinent to mention that it would have meant welding hundreds of tons of steel plate on to the deck of the ship to produce the required, rather modest, change in inertia of the ship section. There were some doubts, however, as to just how effective this material would be, bearing in mind the limited space available to fit it and, in the event, it was decided not to go through with it. He did not blame them either. From the research point of view, however, he was sure that it would have been a most interesting exercise. This did at least serve to underline the importance of the problem and the drastic steps that builders were sometimes prepared to consider to overcome a difficulty in connexion with vibration.

Returning to the question of avoiding resonance, Mr. Lackenby said that in Fig. 5, on page 123 of the paper, the first few modes were quite sharply tuned and it was not necessary to go very far off to obtain a considerable reduction in amplitude, as Dr. Johnson had already mentioned. The sharpness fell off, however, as one went up the scale and the extent of the detuning would have to be greater as a consequence in these instances. In this connexion, he wondered if the authors would care to comment on whether any general guidance could be given on how much it was desirable to keep off, or detune, the forcing frequency to avoid the worst effects, bearing in mind that the tuning sharpness varied up the scale.

Fig. 7 on page 124 was interesting, showing typical frequency coefficients for a number of different ship types. As the spots had been joined by lines one might be forgiven for assuming that they were plotted on some sort of base scale, but he understood that, strictly speaking, this was not so. There was a suggestion, however, that the hull forms, proceeding from left to right, were becoming finer. A possible exception was the trawler about half way along. What he was getting at here was that it would be extremely useful if these coefficients could be sorted out and plotted on a genuine base scale. As change of ship type was involved, in going from one spot to the other, it might be possible to do something about this in terms of certain hull parameters, perhaps proportions of the hull, fineness and so on.

Coming to the vexed question of propeller excited vibration, Fig. 10 on page 127, taken from Ramsay's work, showed among other things how the pressure on the hull varied with the number of propeller blades. He presumed that in varying the number of blades the total blade area was kept constant. He did not think any reference was made to this in the paper. Perhaps the authors would care to comment on it.

With regard to the tentative suggestions for propeller hull clearances discussed on page 131, it had been usual in the past to express these clearances in terms of propeller diameters as, indeed, was shown in Fig. 19 on page 131. He had never been very happy about this practice, however, as one could infer that if the diameter was reduced the clearance could be reduced also, but he was sure this did not necessarily follow. If the smaller propeller was to do the same job, i.e. provide the same thrust, then one would expect the pressure forces to be of the same order and on that account the clearance should be the same. In fact, he could imagine that on a smaller propeller it might not be practical to fit the same blade area as on a larger one, in which case the pressure forces might then be greater, requiring an even larger actual clearance. For that reason he was very glad to see that the approach put forward in the paper did take propeller loading into account, which was certainly a step in the right direction.

There was much more that required to be known about propeller excited vibration and it was appropriate to mention that at this very moment B.S.R.A. had a comprehensive investigation under way on the new oceanographic research vessel *Discovery* which had just been delivered. Careful measurements were being taken of propeller blade forces by means of pressure gauges let into the hull round the stern aperture, of stresses in the tailshaft (from which it was hoped to get the bending moment on the tailshaft) and also careful note was being taken of transient thrust and torques on the propeller shaft. It was also intended to vibrate the ship by means of an exciter and by comparing the exciter deflexions with the deflexions measured by the propeller excitation, it was hoped to obtain a measure of the actual propeller forces. These measurements were being taken with both a four bladed propeller and a five bladed propeller.

The final point he wished to make concerned Fig. 21, which showed the thickness of the boundary layer plotted in terms of ship length in feet. It was stated in the paper that it might be prudent to ensure that the propeller tip clearance was not less than the boundary layer width given in this figure. The curve shown corresponded closely to that given for speeds of about 10 to 15 knots by a formula for plane surfaces, based on velocity distribution in the boundary layer according to the well known "seventh power law". This formula only held however for Reynolds numbers (based on ship length) up to 10^7, which were not up into the ship range and in fact were barely out of the usual model range.

Mr. Lackenby said that in his Thomas Lowe Gray Lecture* a year previously he had developed and put forward another formula for frictional belt thickness which could be reasonably extrapolated up to the highest ship Reynolds numbers and it might be a better one to use in this connexion. It gave somewhat thicker frictional belt widths than those shown in Fig. 21, e.g. at speeds of about 15 knots, instead of 3ft. for a 500ft. ship, it was of the order of 4ft. He would mention the formula and give further details in a written contribution.

In conclusion he congratulated the authors on this very readable and excellent account of their work at B.S.R.A., which he was sure would be found useful by people who had to deal with ship vibration.

MR. S. ARCHER, M.Sc. (Member) said that this paper contained much valuable data and useful guidance to naval architects and marine engineers in waging the never-ending war against vibration on shipboard.

Some valuable experimental work on added virtual mass had recently been published by Burrill, Robson and Townin†. Could the authors state to what extent the results of that research, carried out mainly on prismatic bodies, modified the values derived from the use of the semi-empirical formulæ in the paper and were such deviations considered significant?

Tables I and II were extremely valuable for reference purposes and represented the fruits of a vast amount of experience in the field of ship vibration research.

Equations (11) and (12) for acceptable amplitudes were interesting. They were hybrid in form, containing a presumably judicious blend of velocity and acceleration functions. In an attempt to see how the vertical amplitude formula lined up with experimental work in other fields based on human susceptibility, he had plotted equation (11) in Figs. 45 and 46, as amplitude versus frequency and amplitude versus acceleration, respectively. In both cases the acceptable limits recommended appeared to lie between the Reiher and Meister thresholds of "Disagreeable (injurious after a long time)" and "Dangerous (injurious after a short time)" and were beyond the

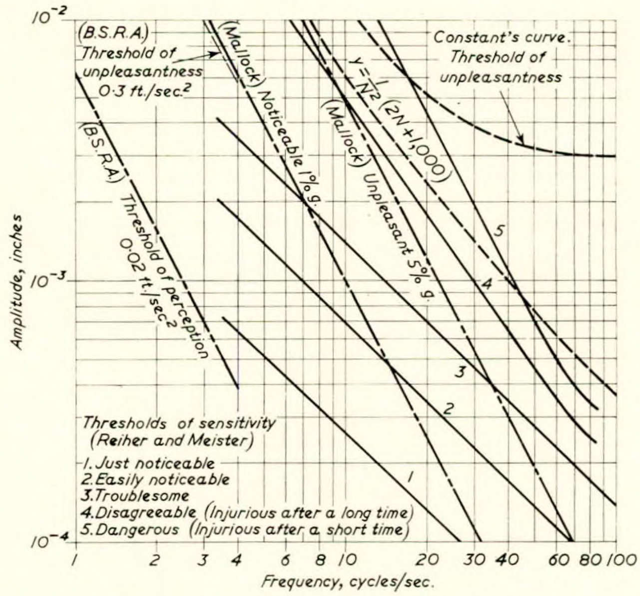


FIG. 45—*Equation (11) plotted as amplitude versus frequency*

Malloch threshold of "Unpleasantness" at 5 per cent *g* acceleration. A plot against velocity gave much the same results. On this score he would suggest that the authors' formula for acceptable vertical amplitudes was by no means restrictive, but from a number of known cases of structural damage consequent upon vibration, it would seem to impose adequate safeguards against such destructive effects. It was, however, a moot point as to whether the limits were tight enough to ensure reasonable human comfort. It was assumed that the formulæ applied strictly to measurements at the after anti-node, or were they intended to be applicable anywhere in the ship, in crew accommodation, on navigating bridge, etc? The problem of crew, and especially passenger, comfort was indeed difficult to bring to rule. For one thing, the human body did not exhibit the same degree of tolerance to acceleration at all frequencies

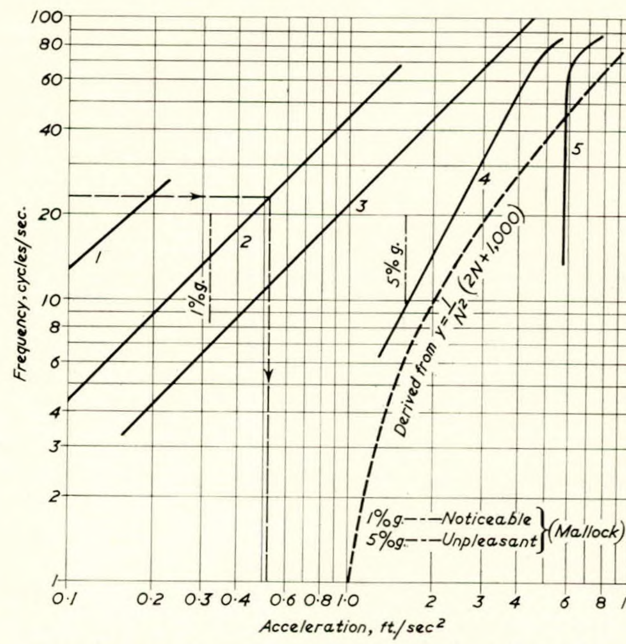


FIG. 46—*Equation (11) plotted as amplitude versus acceleration*

* Lackenby, H. 1962. The Thirty-fourth Thomas Lowe Gray Lecture—"The Resistance of Ships with Special Reference to Skin Friction and Hull Surface Condition". Proc. I.Mech.E., Vol. 176, p. 981.

† Burrill, L. C., Robson, W. and Townin, R. L. 1962. " Ship Vibration: Entrained Water Experiments". Trans. R.I.N.A., Vol. 104.

and in this respect resembled the behaviour of the human ear, where the assessment of equal loudness varied in a somewhat complex manner with frequency and was by no means constant. What little evidence there was suggested that, at least under fairly low frequency vertical vibration, the body could tolerate increasing acceleration as the frequency rose (see Figs. 45 and 46). One factor, perhaps not generally appreciated, was that under vertical "whole body" vibration, human beings exhibited resonance within the range of 200 to 300 cycles/min. ($3\frac{1}{2}$ to 5 c/s)*. This, of course, came well within the range of propeller blade frequency excitation and might partly account for the reduced human tolerance of vertical vibration over this frequency band and increasing tolerance above it. It was clear that there was need for much further experimental work in this field and it would be interesting to have the authors' views on the subject.

Fig. 14 giving typical values of wake-induced variations of propeller forces and moments painted a somewhat alarming picture, especially with regard to the three bladed propeller which not even the Hogner form, according to the authors, appreciably ameliorated. This was difficult to accept and, in general, there was a good deal of evidence to suggest that the results of vortex theory calculations and small scale model tank tests gave pessimistic predictions which were not generally substantiated in the actual ship. In a good example reported by Nitzki (1959)† the measured full scale torque and thrust variations with a modified Hogner type stern were only 60 per cent and 80 per cent, respectively, of the model results at full power. The authors' quoted full scale results from the 40,000 ton tanker (Fig. 28) would also tend to support this view, unless the ship had something in the nature of a Hogner stern or equivalent. Nevertheless, model tests were valuable for indicating trends and for comparative purposes, e.g. assessing the *relative* merits of different shapes of stern frame, after body form, aperture clearances, etc.

In Part II of the paper the authors discussed the question of the most suitable location for a torsiograph, suggesting that the free end of the crankshaft was to be favoured. In this respect, for Diesel installations, he would agree only so far as 2-node or crankshaft criticals were concerned where damping forces were small. For the more heavily damped modes involving large propeller swings, however, serious errors of interpretation, up to +20 per cent or more, could arise when using the normal undamped Holzer natural frequency table and in such cases, where only a single Geiger was available, it was far better to record from a position just forward of the screwshaft coupling as far aft as practicable, in order to minimize the effect of out-of-phase propeller amplitudes due to hydrodynamic damping‡§. In the case of steam reciprocating installations, however, of low engine inertia relative to that of the propeller, the converse applied and recording at the forward end was preferable.

The use of the term "barred range" was now unfortunately more or less established by common usage, but the writer much preferred the term "restricted range" as implying a more realistic régime. Some owners' reactions were perhaps inclined to be a little less obstinately unsympathetic to builders when the more diplomatic term was used!

The authors' mention of the subsidiary damaging effects of torsional vibration upon such parts as fuel pump drives and timing gears, etc. was very much to the point and was, in fact, one of the reasons why Lloyd's strongly recommended that over the 10 per cent speed range just below maximum service revolutions not more than 50 per cent of the f_c value for continuous operation should be allowed.

In response to the authors' request for comments on the validity of their formulæ for crankshaft axial stiffness, equation (29) had been evaluated for the case of the 6-cylinder, 750 mm. bore × 2,500 mm. combined stroke opposed piston engine crankshaft which formed the subject of the recent paper to this Institute by Atkinson and Jackson‖.

For the centre cranks, i.e. centre line of No. 3 main throw to centre line of No. 4 throw, and assuming a "solid" centre coupling, also that θ, the angle between cranks, is taken as 180 deg. for that section, the stiffness works out at $k = 2{,}020$ tons/in., the general formula being reduced to,

$$k = \frac{10^6}{78 - 415 \cos \theta} \text{ tons/in.}$$

For the sections, centre line No. 1 to centre line No. 2, No. 2 to No. 3, also No. 4 to No. 5 and No. 5 to No. 6, for all of which $\theta = 120$ deg., the stiffness obtained is $k = 3{,}490$ tons/in.

For the end sections, presumably the stiffness would be taken as twice the value obtained for the case of $\theta = 0$ deg., which in that event would amount to $k = 5{,}930$ tons/in.

For the case of a half-shaft without flexible coupling, as tested by Wilton Fijenoord,¶ the respective overall stiffnesses obtained could be compared as follows:

Authors

$$\frac{1}{k} = \frac{1}{5{,}930} + \frac{2}{3{,}490} + \frac{1}{2 \times 2{,}020}$$

or $k = 1{,}012$ tons/in.

Doxford

$$\frac{1}{k} = \frac{3}{2{,}820}$$

or $k = 940$ tons/in.**

Wilton

$$\frac{1}{k} = \frac{3}{2{,}120}$$

or $k = 707$ tons/in. (as derived by S. Archer)¶

or $k = 250 \times 2{\cdot}54$

$= 636$ tons/in. (as reported by Ir. A. Meijer)¶

For the reasons indicated by Mr. Archer in discussing the authors' reference (14), the Doxford figure of 940 tons/in. must be regarded as suspect and probably about 40 per cent high. Accordingly, the authors' comments would be appreciated, both as to the overall numerical results derived and as to whether the formula has been correctly interpreted, with special reference to the end half-cranks.

Formula (30) on page 143, with associated Figs. 40 and 41, should prove a most valuable guide to designers for determining approximate limits of unbalanced external couples, especially in vessels for which no previous type experience was available.

MR. A. A. J. COUCHMAN, B.Sc. (Associate Member) said that in a number of places in the paper attention had been drawn to analytical work and model tank results which indicated the magnitudes of the alternating thrust forces emanating from the propeller. Some of this work had suggested that for five bladed propellers the principal forces were those occurring at twice blade frequency, while those at blade frequency were small enough to be considered insignificant. The conception had, therefore, arisen that if one fitted a five bladed propeller to a single-screw ship the system would not be troubled by axial vibration. However, a glance at Fig. 14 and the values quoted on page 136 showed the 5th order thrust variation at the propeller to be far from insignificant, and Fig. 29a indicated that if a five bladed propeller had been fitted to the 40,000-ton tanker discussed, the blade order axial resonant frequency

* Schmitz, M.A. and Boettcher, C.A. "Some Physiological Effects of Low-frequency, High Amplitude Vibration". A.S.M.E. Publication No. 60-PROD-17.

† Nitzki, A. 1959. "Resistance and Propulsion of High Powered Single-screw Vessels", European Shipbuilding No. 3.

‡ Archer, S. "Screwshaft Casualties—The Influence of Torsional Vibration and Propeller Immersion". Trans. R.I.N.A., 1949, Vol. 91, p. J56 *et sea;* also Trans. I.Mar.E., 1950, Vol. 62, p. 43 *et seq.*

§ Archer, S. "Contribution to Improved Accuracy in the Calculation and Measurement of Torsional Vibration Stresses in Marine Propeller Shafting". Proc.I.Mech.E., 1951, Vol. 164, pp. 351-366.

‖ Reference (14) of the paper.

¶ Discussion of reference (14) of the paper.

** Appendix 2 of reference (14) of the paper.

would have been placed in a most unfortunate position just above the service revolutions. In fairness, however, it should be pointed out that the thrust block seating fitted in this vessel was a particularly soft structure. Generally one would expect the axial resonant frequency for a tanker shaft system, with turbine machinery, to be in the region of 800 cycles/min., thus placing the 5th order resonant condition at 160 shaft r.p.m.

Figs. 23 and 25 indicated a case of extreme flexibility in the double bottom structure. It was not always appreciated that with increasing double bottom flexibility the equivalent mass associated with the vibration increased rapidly and the stiffness of the seating structure fell; the trend of each of these variables contributed individually to a lowering of the axial resonant frequency of the shafting system.

A great deal of experimental work had been done in recent years to evaluate the axial stiffness of Diesel engine crankshafts. It was his opinion that stiffness values derived either by splaying the webs of a single crank throw with a known force, or by hanging the crankshaft vertically and measuring the compression at the bottom pair of webs, had questionable value. It would appear that realistic static stiffness results could only be obtained when the complete crankshaft was tested horizontally.

The crankshaft stiffness formulæ contained in the paper were undoubtedly a valuable contribution and it would be useful to know to which engine crankshafts they had been applied and the degree of success achieved.

It was extremely interesting to see that the authors considered that journal bearing restraint was not a significant factor in crankshaft axial vibration. This might be a reasonable assumption in the case of certain crankshafts, up to, say, six cylinders, but it had to be remembered that, under dynamic conditions, bearing clearances were reduced by oil film thicknesses and, in the case of, say, a 10 or 12-cylinder installation, the bending of pins and splaying of the webs would be considerably increased. In these circumstances would the authors consider bearing restraint insignificant and, if not, could they suggest how the basic formulæ might be modified to take this into account?

The problem of transverse engine vibration, in his view, was not simply one of considering only the engine and engine seating. Due to the nature of the problem the hull vibration characteristics and engine room double bottom flexibility were important aspects which were pertinent. Hull vibration modes, particularly horizontal and torsional, were just as likely to produce transverse engine vibration as the engine forces themselves and one might draw some support for this view from the fact that in some cases tie-bars had proved ineffective.

It was true to say that for all the transverse engine investigations carried out by B.S.R.A. a skew mode of vibration had never been definitely identified, which tended to suggest that the engine bedplate and entablature stiffness was adequate and that improvements, if necessary, could be best achieved by stiffening the double bottom structure. Perhaps the authors would care to comment on the possible connexion between hull and transverse engine vibration.

The authors had stressed the importance of maximum rigidity when fitting tie-bars. Unless this was adhered to, not only at the anchor points but also in the transverse planes of the bars, it was possible to create an ineffective system which could possibly worsen the severity of vibration.

It was agreed that where thrust and torque variations were prime considerations, propellers with odd numbers of blades were in general to be preferred, but there were possible exceptions; in particular there appeared to be reasonable justification for saying that the large alternating bending moments were at least a contributory cause of the high percentage of tailshaft failures occurring in tankers, which appeared to be fitted with a predominance of five bladed propellers. The experimental evidence available suggested that in the fully loaded condition the hydrostatic bending moment tended to relieve the weight of the shaft and propeller, while in the ballast or light conditions it would partly or completely augment this weight. Thus for tankers, which spent considerable periods of time in ballast, more rapid wear of the stern tube woods was likely, thus increasing the possibility of transverse vibration of the shafting system. The result was to further magnify shaft stresses, which were already increased due to reduction in ship draught, to limits sufficient to produce failure of the tailshaft.

In Fig. 15, for the experimental points obtained at the aft end of the vessel, at the higher frequencies a curve drawn almost anti-phase to that shown appeared to be indicated. This would move the 6-node vertical to a lower frequency and the possible 9-node would perhaps be a 10-node. Using the 2-node vertical as the basic frequency and applying the theoretical formulæ $\frac{N_2}{N_1} = 2$, $\frac{N_3}{N_1} = 3$, etc., the ratio of the experimental to theoretical frequencies for the 3, 4 and 5-node vertical modes was 0·98, 0·96 and 0·91, which suggested that the next ratio would be approximately 0·85 and thus experimental frequency 302 cycles/min. which tended to confirm that the 6-node and subsequent nodes should be lower. Would the authors care to comment on this?

Mr. A. R. Hinson (Associate Member) said he particularly liked the way in which the authors had demonstrated, in both sections of the paper, how the academic approach could be used to provide practical solutions to vibration problems. It seemed to him that this was a theme of the paper and being so, the fundamental equation from which equation (6) was derived could have been included to advantage:

$$\text{Spring force} + \text{damping force} + \text{inertia force} + \text{exciting force} = 0$$

$$\text{or} \quad kx + c\dot{x} + m\ddot{x} + F\sin\omega t = 0$$

where the symbols had their usual meanings.

The above equation was relevant since it indicated that when trying to eliminate vibration, there were only four types of force which could be modified and if these four were considered, all types were considered.

On the question of acceptable vibration, as stated in the paper, the most important aspects were usually the psychological and physiological effects on passengers and crew. They would complain long before a ship sustained damage. It could be argued that acceleration and not amplitude was a more suitable criterion since the body felt the force generated by acceleration. Be that as it may, since the authors had chosen to define acceptable limits in terms of amplitude for a given frequency, it was an easy matter to convert.

The formula for acceptable vertical vibration gave accelerations which varied from 1ft./sec.^{2} at 50 cycles/min. to about 2·5ft./sec.^{2} at 800 cycles/min. At 400 cycles/min., the blade frequency of a four bladed propeller turning at 100 r.p.m., the acceptable acceleration was 1·6ft./sec.^{2}

He understood that these limits applied to vibrations measured at the largest anti-node and would not be accepted within the body of the vessel. Acceleration was proportional to amplitude for any given frequency and when the above values were corrected by multiplying by the amidships mode functions given in Table II, the following values were obtained for the 2 and 4-node modes:—0·4ft./sec.^{2} at 50 cycles/min.; 1·0ft./sec.^{2} at 800 cycles/min.

He suggested that those values were high for living quarters where the human body was at rest, i.e. restaurants, bedrooms and the navigating bridge. It should be possible to reduce them by half. They would, however, be acceptable for the open deck and other places where the body was active. Attempts should be made to reduce any vibration greater than that refined as "Easily Noticeable" by Meister's curves.

With regard to propeller excitation, Fig. 10 would be true only if applied to propellers having similar blade shape, pitch reduction and skewback. It was true that experimental results, particularly those of reference (5), indicated that these factors were relatively unimportant, but in practice the converse had been proved. It was generally beneficial if skewback, measured at the propeller circumference, was not less than 8 per cent of the diameter, with pitch reduction from 0·7R to R at least 5 per cent of the maximum.

Mr. Hinson agreed that it was difficult, if not impossible to calculate main engine transverse vibration criticals. He had measured transverse vibrations on sister ships and found the natural frequencies to be 12 per cent apart. A calculation with a probable error greater than 10 per cent was of academic interest only; it could not be accepted as a basis on which to make a decision. The vibrations were suppressed by staying the engines to the hull, a common enough device, but one which should not be employed lightly. On one occasion where the builders had stayed the engine to the hull, it was not until the stays were removed that both hull and engine ceased to vibrate.

The author's opinion on the acceptable amplitude for transverse vibration of an oil engine would be welcome. As a rough guide what did they think of 0·5 thousandths of an inch per foot of engine height above the tank top?

MR. A. SILVERLEAF said that he was delighted to be invited by the Institute to contribute to the discussion on the paper, which was an admirable survey of the present situation in regard to many important problems in ship vibration. It was an intriguing, interesting and infuriating subject to those who had sometimes to deal with it in more than an academic fashion. The paper contained a great deal worth commenting upon and, as the authors knew, he largely agreed with the great majority of the views they had expressed, so his comments would now be confined to one or two topics which he found either puzzling as they expressed them or, alternatively, where he was not quite in sympathy with their approach.

First of all, in the discussion of methods of estimating the critical frequencies of hull vibration, would the authors express an opinion as to what they meant on page 124 by reasonable accuracy which could be obtained in using approximate formulæ. He would have welcomed in Fig. 7 some indication not of their estimated mean values but of the sort of scatter that occurred in these coefficients, which in fact could be derived largely from Table I if one were not too lazy. It was his feeling that the authors did not expect accuracies of much more than something like ± 4 per cent or so either in vertical or horizontal vibration frequency estimates. If this was so he was a little surprised that in equation (5) and in Fig. 7, which went with it, the authors had not gone the whole hog and given some indication of simple formulæ dependent largely on mean ship dimensions which could have been used to estimate section moment of inertia, for, as he understood it, strictly speaking, this was what Fig. 7 was, some sort of indication of the way in which moments of inertia varied with ship types. Like Mr. Lackenby, he found it more confusing than helpful, incidentally, in Fig. 7 to see the points joined by lines which had no validity. One could very nearly have put them in any order unless the authors had some logical suggestion or reason for putting them in their order; if so, he would be delighted to know of it.

Fig. 8 puzzled him not because of anything the authors had done in presenting it but because of a basic dilemma. Dynamic magnifiers or damping factors for ships were almost entirely a function of structural damping. Hydrodynamic damping was a very minor factor and he found it a little difficult, therefore, to understand why at low frequencies there should be such a great difference in the damping and energy absorption characteristics of the structure between vertical and horizontal modes of vibration and so little difference at higher frequencies. The authors' comments on this point would be appreciated.

Turning to a point discussed by many previous speakers, Mr. Silverleaf said that in dealing with propeller excited vibration the authors dealt in both parts of the paper with the pressure forces. It was disappointing to find that they still quoted solely Ramsay's work which, valuable as it was at the time it was done, had been almost entirely outmoded by subsequent work. First of all, the measurements (which were shown in a rather useless form in Fig. 10 because the non-dimensional coefficient was given no physical significance) did not agree with subsequent measurements made elsewhere, in particular with measurements made at the National Physical Laboratory, which agreed well with a very much better theory than that which Ramsay used and which the authors quoted in the paper on pages 130 and 131. The much better theory was, of course, that due to Breslin and in a recent paper given in Washington, in August 1962, he showed that his much more comprehensive theory agreed excellently with these rather better measurements. The inadequacy of the Hubbard and Reiger work which Ramsay quoted and the authors reproduced could be seen very quickly by looking at equation (21). From this it appeared that at zero torque there would be zero pressure but this was incorrect. What Hubbard and Reiger omitted to take into account, but Breslin appreciated, was that there was a separate and quite independent blade thickness effect, since a rotating propeller blade of finite thickness, even though it was delivering no thrust or absorbing no torque, would create a fluctuating free-space pressure field. The magnitude of this was about equal to the magnitude of the pressure field due to the thrust and torque forces and he therefore suggested that since this paper was an excellent guide to the person interested in following the broad story of ship vibration research, this particular section was today not sufficient.

He also wished to add a cautionary note about the next section, dealing with wake forces, which both parts of the paper discussed. Throughout the paper and throughout the discussion it was taken for granted that the vibration frequencies excited by propellers were always blade frequencies or integral multiples thereof. Experience did not always confirm this and since, as engineers, they were interested in what really happened on ships, it might be very misleading and dangerous to base the whole approach on something which was not shown to be true in practice. First of all, N.P.L., working in collaboration with B.S.R.A., had been extremely anxious to obtain more data to supplement what the authors rightly called the scanty data available on unsteady propeller forces and some very surprising and perturbing things had been found. One of these was that without question very often a shaft order force was generated by a propeller. The second was that in certain cases not only was a first order force generated but also forces of one less and one more than the blade frequency. Those who were interested in helicopter design would know that analogous phenomena had been encountered tackling that particular vibration problem.

Next, he wished to sound a cautionary note about taking too much for granted in Fig. 14 and all that was built around it. As Mr. Archer had pointed out, much of this was based on extremely inadequate quasi-steady state theory calculations; when they were the only things that could be done they were worth considering, but progress had been made since then. Just as there was at the present moment some doubt and confusion as to the true value of much that had been measured on shipboard, there was equally great doubt among those doing model experiments as to the validity of much that had been published so far. In fact, there was open agreement that they did not believe all they had published and this was exemplified in the most practical and useful way possible by the fact that the Propulsion and Cavitation Committees of the International Towing Tank Conference had agreed to carry out a comparative series of experiments in many model establishments throughout the world to see whether, for an identical situation and problem, they got anything like the same answers. All those who were members of those Committees were awaiting the results with the greatest of interest and trepidation.

In passing he wished to make one comment on something said earlier. Experience at N.P.L., both on the ship and in the laboratory, in attempting to measure unsteady propeller forces, had convinced them that sensitive strain gauge techniques were not only the most reliable and sensible way of measuring such fluctuating quantities but also that for mean value measurements they were at least as accurate as and much simpler than any other ways of measuring thrust or torque.

Referring to the work shown in Figs. 16, 17 and 18, Mr.

Silverleaf said that he considered this one of the most significant advances in recent years in their understanding of many of the measurements which had been made on shipboard and, like so many other advances, many of them implicit in this paper, he thought it was worth pointing out that the credit for this went to the authors and their colleagues at B.S.R.A.

Dealing with another general point, which Mr. Jackson had hinted at in his opening remarks, the whole of this paper and much of the work on vibration problems (not all, of course) implicitly assumed that vibration was a regular phenomenon. All who had experienced it on shipboard knew that this was generally far from the truth and that in fact one of the most infuriating aspects of it was its irregularity; it came and went and eventually it wore one out as much as water dripping on a stone. Mr. Jackson had hinted at one of the possible ways in which such unsteadiness might occur. He said that it was not uncommon in propeller shafting systems for axial and torsional criticals to be not too dissimilar. In an effort to understand some of the perplexing results obtained on shipboard he had had many discussions with friends in shipyards and with shipowners. As a result of one particular discussion he was now able to demonstrate a model devised by Mr. Parker of Harland and Wolff. This was a simple system with a heavy mass suspended on a spring and having its torsional or rotational, natural frequency and its axial or vertical natural frequency very close to one another. The vertical axial movement could be observed very readily, and some indication could be seen from the spigot of the torsional amplitude. When he started it with an axial displacement a typical example of energy interchange between two modes of vibration could be observed; the maximum amplitude of torsional oscillation coincided with the minimum of axial and *vice versa*. A propulsion shafting system was not quite this but it had many similarities. These were the sort of phenomena not discussed in the paper which might become increasingly important in the study of ship vibration in the next few years and just as this paper indicated that considerable progress had been made in the last five or ten years, it could only be hoped that the authors would be able to present an equally vivid and exciting story in a few years time.

Correspondence

Mr. G. A. Bourceau (Member) in a written contribution, said that the understanding of mechanical vibrations in ships still presented many imponderables to the investigator and it was with sincerity that he congratulated the authors for this valuable paper which contained interesting experimental data and theoretical analysis.

On page 128 he was surprised to observe that the authors considered the wake forces to be only slightly influenced by propeller blade skew. This was at variance with experience by Bureau Veritas and he would recommend to the authors his paper in the 1961 Transactions of the Association Technique Maritime et Aeronautique, where the effect of propeller blade skew was considered. This paper gave some quite remarkable results of reductions in shafting vibration achieved by increasing propeller blade skew, these shafting vibrations having been induced by wake forces. He would like the authors' further opinions on this matter and would be pleased to give them the benefit of his society's experiments on the effect of propeller blade skew. Mr. Bourceau found himself in full agreement with the authors' views on hull-propeller clearances, also with their remarks on the variation of shaft bearing reactions, this being largely a question of adequate alignment of the shafting. Recent experience in Bureau Veritas had indicated that by imposing adequate reactions at shaft bearings it was possible not only to reduce vertical vibration in shafting, but also to improve the running of the main propulsive machinery.

The question of adequate alignment of shafting was closely linked with the axial vibration of the shafting and it was to be emphasized that optimum alignment was dependent upon the magnitude of the bending moment on the thrust block collar. This problem had been dealt with in a recent paper published by Nouveautes Techniques Maritimes in 1962, in which the position of the thrust block relative to the main propulsion unit was shown to be of importance. Bureau Veritas experience, practical and theoretical, indicated that the thrust block should be located as far as permissible from the main engines and it was desirable that at least one intermediate bearing should be fitted. The authors opinions on this would be appreciated.

The combination of increased power and the increased hull flexibility of the modern ship had magnified the influence which shafting vibration exerted on hull vibration and at present the society's mechanical research section were completing an investigation on this subject.

With regard to the transverse vibration of engines, it had been found that the consecutive curved modes were not only influenced by crosshead and trunk piston transverse forces, but were also strongly affected by the firing order of the engine. At present an investigation was being held by the Bureau Veritas mechanical research section on the general design of the Diesel engine beam form.

The authors had shown considerable courage in stating their views on the limits of engine unbalance and in particular their formula for the tolerable unbalance couple was of interest to Bureau Veritas. The society had experience of new ships having machinery aft where the unbalance engine couple coincided with a hull node resulting in the dynamic response of the hull and consequent hull vibration. It would be most interesting to know the views of British engine builders concerning this formula for tolerable unbalance couple, shown on page 143.

Finally the authors were to be thanked for a paper which had excited attention not only in Britain but also on the Continent.

Mr. R. Anscomb, O.B.E. (Member) wrote that in the introduction the authors had stated that their survey of vibrations was confined to the more important problems and that the paper did not attempt to cover the whole field of interest. There were however, additional aspects of hull vibration which were of importance to the designer, if he was to evolve a satisfactory and successful ship.

Under the heading *Acceptable Limits of Hull Vibration*, the authors stated that the most important considerations were usually the psychological effects on passengers and crew. Whilst this was undoubtedly true, there was always present the added possibility of stress fluctuations—acceptable enough in themselves—which might give rise to fatigue failure of certain parts of the hull structure, especially in way of the stern frame and rudder mountings. The clear water type of stern could do much to alleviate this particular problem and indeed was becoming more popular in current designs, but the more sophisticated designs of stern as shown in Fig. 13(c) of the paper, albeit good from the point of view of vibration and flow characteristics to the propeller, would seem to be an expensive type of structure. With reference to twin-screw passenger ships, it would appear that conventional bossings were liable to flap with the associated risk of fatigue and failure. For many years the "A" bracket had given very satisfactory service under stringent operating conditions in naval ships. It afforded a more rigid support of the propeller and shafting, as well as having a good propeller flow characteristics.

During the preliminary stages of a design, it was possible to estimate the natural frequency and vibration profiles of a given hull form with a reasonable degree of accuracy and to take due account of a number of possible load distributions. In service the ship might be so loaded that the natural frequency of the hull lay at or near the propeller frequency, giving rise to undesirable vibrations. The alternative courses of action then available were either to alter the propeller revolutions, or to attempt to redistribute the load, by shifting internal liquids or flooding empty tanks. The authors opinion on the feasibility and effect of these courses of action would be appreciated.

Much of the detail design of a passenger ship these days lay outside the field of the pure naval architect. With sufficient data to hand he could ensure the avoidance of main hull vibration over the normal range of working speeds, but the problem of local panel vibration due to propeller-hull interaction and to auxiliary machinery was perhaps not so easily resolved. There was a growing tendency towards larger public rooms, extending from ship side to ship side, with the inevitable large spans of unsupported deck. The careful positioning of these large rooms within the ship was of paramount importance and it was hoped that those responsible for the design and décor of these rooms would become more aware of the susceptibility to vibration of large unsupported areas of deck. It was felt that a suitable design based upon a pillaring system need be no less attractive aesthetically than a wide open space. The same argument might equally well be levelled at the design of furniture and fittings. Well supported chairs and tables need not necessarily detract from a pleasing design, although their design could have a marked effect on their susceptibility to vibration. Admittedly these aspects of vibration were not vitally important and contributed more to nuisance value than to anything more serious. Nevertheless they were a contributory factor that affected passenger comfort and, therefore, deserved active consideration, if modern designs of passenger vessels were to compete effectively with aircraft.

Mr. A. Muirhead (Member) wrote that he had been present on several occasions when the authors had been taking vibration records and they had always impressed him as being a "balanced" combination of practical engineer and scientist. The paper now bore this out. On page 126, equations (11) and (12) gave the limits of the acceptable amplitude; Mr. Muirhead wished to know how these compared with the earlier limit based on the accompanying acceleration being limited to 1ft./sec.^{2} He assumed that the acceptable amplitude was taken at the after end of the vessel.

The graph given for the boundary layer thickness in Fig. 21 differed considerably from the formula given by Van Lammeren in his book, *Resistance, Propulsion and Steering of Ships;* an explanation of this would be welcome.

The last figure in the paper (Fig. 42) most effectively underlined the absolute necessity for close co-operation between engine builder and ship designer and Mr. Muirhead added that the late Admiral Taylor's saying about propulsion was applicable to vibration as well, "The time for pessimism is when the powering is being done, not when the trial is being run".

Mr. H. J. Adams, in a written contribution, observed that this was a very welcome paper in which the authors had covered a wide range of vibration problems in a manner which enabled both the naval architect and the marine engineer to understand each other's problems. The present paper used in conjunction with the first paper in the list of references would be a most useful tool in the hands of ship designers, shipbuilders, and engine builders.

The first section of the paper contained references to prediction of main hull frequencies. These were required at a very early stage of the design. The usual formula used was, as indicated, of the form $C\sqrt{\dfrac{I}{\Delta_1 L^3}}$ and this had frequently been expressed as a curve of frequency on a base $\sqrt{\dfrac{I}{\Delta_1 L^3}}$. It was unlikely that any single curve would be successful for all types of ship and, as the authors proposed, the use of the formula given with C taken from a similar ship was most likely to give a good result. Fig. 7 confirmed this as C was shown to vary with ship type. One reason for the difference in coefficient for different ship types might be that while the calculated I value for two ships might be the same, the effective value might be different if the types were different, e.g. in an ordinary dry cargo ship the inertia of the deck would be less fully developed than in a tanker with two longitudinal bulkheads. Fig. 7 confirmed this as the frequency for tankers was greater than that for dry cargo ships, i.e. the ships were stiffer. Ore carriers with twin bulkheads were likely to be similar to tankers and bulk cargo ships with overhanging topside tanks would have lower frequencies. Fig. 7 also confirmed this and it might be a field of useful study. Vertical vibrations were referred to.

The authors also pointed out that the use of Δ_1, in the formula just given, ignored the distribution of mass. This suggested that the use of a C value derived from a ballast condition would not be suitable for predicting a loaded condition. A further suggestion was that not only the distribution of mass, but the type of mass might have an effect, i.e. liquid cargoes or ballast might have a different effect from iron ore or grain. This also might bear study.

It was important to be able to decide at an early stage whether certain values of unbalanced forces or couples might be tolerated; inability to do so might result in the rejection of a suitable engine due to uncertainty. Also the facility to predict the acceptable values might enable the engine builder to choose the firing order which produced the correct proportion between forces and couples as required. It was rarely possible to move the engine position by an appreciable amount or appreciably to alter the natural frequency of the hull.

Particularly welcome therefore was the formula on page 143 which with the assistance of Figs. 40 and 41 enabled an estimate to be made of the tolerable couples. To enable full value to be obtained cross-curves for similar data to that on Figs. 40 and 41 on a base of position of engine centre were necessary. Data were scanty and Mr. Adams felt sure the authors would support a plea to owners and builders to give every opportunity for the data to be supplemented. This would enable the authors to supply fuller information in due course.

The values given appeared to be conservative. Using Fig. 42 to extrapolate from Fig. 41 for an engine position about 15 per cent from aft the tolerable secondary couple for 4-node vibration in a particular ship measured, would be given as about 100 tons-ft. at the engine speed corresponding to resonance. In actual fact there was a couple of about 230 tons-ft. and little appreciable vibration was evident in the main hull although there was some fore and aft vibration in the wheelhouse, the ship having all accommodation and machinery aft. This effect was mentioned on page 123 of the paper.

Mr. J. L. Millar, B.Sc., wrote that the paper made interesting reading, as it contained much useful information which should be of value to those involved in ship design.

In Part I of the paper, the likelihood of torsional vibration of the hull occurring was mentioned very briefly. There appeared to be very little data or information available concerning the seriousness of this mode of vibration. If this was so, would the authors be willing to explain why such scarcity existed? Would they agree that when investigating the natural frequencies of cargo vessels, especially with machinery aft and having wide continuous cargo hatches, the torsional modes should be looked into?

With regard to the propeller shaft, what was the influence of the vibrations on propeller efficiency? Depending on the severity of the vibration, considerable variation in the rotational speed of the propeller could ensue and it would be interesting to know if the authors had any data relating to this loss in efficiency.

In Fig. 23 a profile was given of the vertical vibration along the centre line on the tank top. Could the authors please explain why the profile did not tend to read zero at the stern tube? Was it found in practice that there was no significant stiffening effect afforded by the shaft stools?

Would the authors agree that in general few cases of hull vibration were due to unbalanced forces or couples in the engine, whilst more were brought about by flexibility in the engine framing or seats and the majority was caused by forces set up by the propeller passing the hull?

Mr. K. V. Taylor, B.Sc.(Eng.) (Associate Member) wished to say at the outset that he thought that the authors were to be congratulated on the manner in which the paper was written and, though some of the contents were only too familiar to the specialist, it was to be hoped that it would be widely read and digested by both the naval architect and marine engineering fraternities.

Early in the paper a statement had been made which required some explanation and clarification. In the text it was stated that the vibration exciter used in the tests had only a maximum speed of 600 cycles/min. but the vibration data for a 185-ft. trawler, shown in Fig. 5, indicated that the frequency range covered by the measurements was up to 800 cycles/min. In actual fact the exciter was originally capable of the greater speed, but the fitting of oil seals subsequent to this particular test had restricted the speed to the lower figure. A report* dealing specifically with tests on trawlers had recently been published and the series of measurements carried out presented a unique opportunity to obtain vibration data on a group of ships of similar design but ranging in size.

It would be of interest to know how the authors came to choose the particular sequence of types of ships shown in Fig. 7. It would appear from a first glance that it was selected in order that the Φ values for the 4-node vertical mode were continually decreasing, a tendency not necessarily borne out by the other four sets of curves. Actually, the conclusions to be drawn from these curves would be illustrated more clearly if plotted in the form given in a previous paper by Dr. Johnson† in which mode frequency/2-node vertical mode frequency was plotted against node number for specific types of ships. In that paper were shown two curves based on results for tankers and passenger and cargo ships. The former group was linear with increasing mode number whereas the latter was less steep and showed a tendency to concave downward.

With the emphasis at the present towards the solution of problems associated with propeller excited vibration he wished to have the authors' views on vibration measurements resulting from unsteady propeller forces as distinct from machinery where the exciting force or couple was constant for a given frequency. Subject to reasonably accurate analysis of the data he felt that a truer and more accurate picture of the vibration characteristics would be obtained from plotting a curve of "mean" spots rather than a few selected maximum values. It might well be necessary when comparing the exciting quality of two different propellers, say a four and a five bladed propeller to adopt a technique which would make use of a far greater part of the measured data. Analysis would take somewhat longer but it could be carried out without the need for personal selection. For an example, a technique of this type might have improved on the scatter of spots shown for the aft end in Fig. 15.

In the second part of the paper the problem of a twin oil engine installation was discussed and reference was made to vibration which was likened to that resulting from the two prongs of a tuning fork. While the analogy was useful to describe the motion taking place there was, however, one important difference. A tuning fork, once set in motion, resulted in the two prongs vibrating symmetrically about a plane normal to the direction of motion and containing the handle of the fork. The prongs vibrated in opposition, the motions being equal and anti-phase to each other. With two engines, however, each was excited independently both in respect of the direction of the exciting torque and running speed. This could result in the engines vibrating in both the anti-phase mode (tuning fork vibration) and an in-phase mode. Should resonant conditions be experienced with vibration of the former type, the tying of the two engine entablatures together would readily reduce movements of both engines as indicated in Fig. 38 of the paper. On the other hand should the phase of the movements of the two engines be similar, no restraining action occurred and the tie bar no longer served a useful purpose.

The problem was further complicated by the part played by the double bottom structure on which the engines were seated and which also formed part of the vibrating system. Normally, however, in-phase and anti-phase modes of this vibration did not occur at the same frequencies. Generally for the anti-phase mode there was no appreciable drain of vibration energy into the main hull, apart from that to the structure in the immediate vicinity of the engines and seatings. With the in-phase mode of the engines the dynamics of the motion was such that the hull might be subject to considerable torsional and horizontal excitation. For this latter mode of vibration staying each engine separately to the ship's side frames would be a possible solution subject to the precautions discussed in the paper.

The problem of precise speed control between two or more shafts was a thorny one both with oil engines and steam turbine installations. During several vibration trials conducted by B.S.R.A. synchronism had been achieved for relatively short periods of a few minutes by manual control, the difference in the two shaft speeds being indicated by means of a simple electric indicator activated by impulses from a "make and break" device on each shaft. There would appear to be a need for automatic speed control between shafts and since various techniques for this were in common usage in the aircraft industry, it might be that they could provide a solution applicable for marine use.

Ing. A. Guglielmotti wrote that he had very much appreciated the paper presented by Dr. Johnson and Mr. McClimont, which summarized very clearly the main problems of vibration concerning both hull and propulsion machinery. However, he wished to contribute some remarks and observations to the discussion.

On page 141, the authors assessed as being of little practical value, the employment of some device suitable for controlling, during operation, the phasing of the engines in a ship installed with two or more engines. His company had constructed, in co-operation with a manufacturer of electric equipment, a device which permitted effective synchronization of two or more engines. This solution was adopted with the intention of minimizing the hull vibrations, induced by the existing inertia action of the engines, in certain motor ships.

In order to ascertain its efficiency, the device was initially fitted in an Italian-owned vessel, installed with two propellers driven by four engines, on which several tests were carried out and records made.

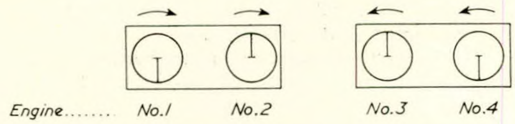
The results obtained were very satisfactory, so much so that the installation of the device was extended to all the motor ships belonging to the same owner. The installations, after many years of service, had proved to be capable of meeting the somewhat difficult operational requirements on board these vessels.

In Fig. 47, the results of the vibration records, obtained with different phasing of the engine crankshafts in one of these ships, were shown.

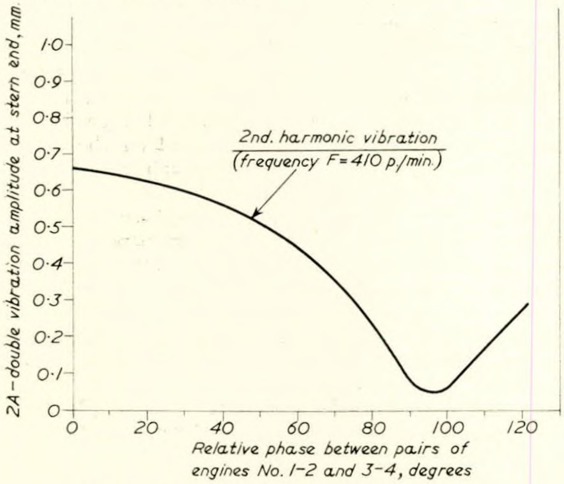
On page 126, the authors gave equation (10), which permitted the calculation of the hull vibration amplitudes excited,

* Taylor, K. V. and Joy, A. W. 1962. "Hull-vibration Tests on Three Modern Trawlers". The Shipbuilder and Marine Engine-builder, Vol. 69, No. 661, p. 607.

† Reference (1) of the paper.



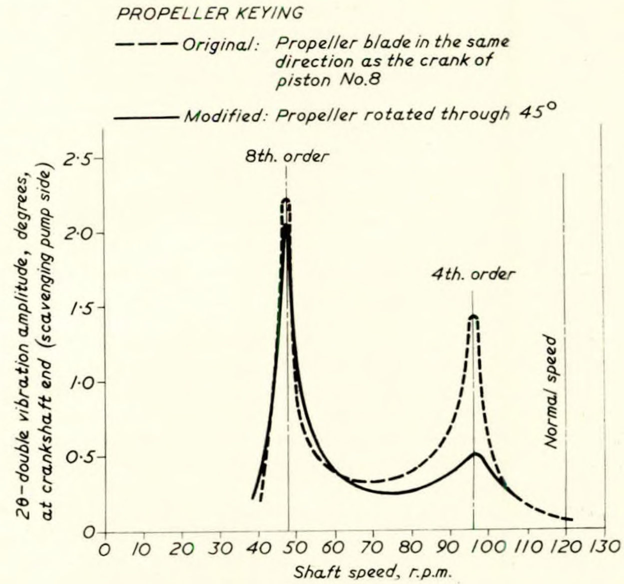
Initial positions of the No. 1 cranks in the four engines (as seen from flywheel side):
Note: The records have been made, keeping the phase displacement of the crankshafts of the engine pairs 1-2 and 3-4 at 180 deg.



Characteristics of the propulsion plant:
Four Fiat engines type B487T on two shaftlines.
Characteristics of the engines:

Number of cylinders 6 (2 stroke—single acting)	
Cylinder bore	480 mm.
Piston stroke	640 mm.
Rotating speed	205 r.p.m.

FIG. 47—*Vertical vibrations of m.s.* Torres—*Records with synchronization set*



Propulsion plant characteristics:
Fiat engine type 758D, Four bladed Lips propeller.
Engine characteristics:

Number of cylinders 8 (2 stroke—double acting)	
Cylinder bore	750 mm.
Piston stroke	1,320 mm.
Rotating speed	120 r.p.m.

Frequency recorded: F = 384 p/min.

FIG. 48—*Shaftline torsional vibrations of m.t.* Dogaressa *for two different propeller keyings*

for instance, by the free inertia action of the engine. This equation should evidently be applied at prevision stage, by utilizing one of the elastic lines shown in Table II on page 125.

According to his experience, there was always some difficulty in calculating the hull vibration amplitudes with sufficient accuracy, due to the few facts available concerning the shape of the elastic line and the damping coefficients. Generally, sufficiently accurate results could be obtained by comparison with similar ships which had already been built.

In the authors' experience, what degree of approximation could be reached with equation (10) in the evaluation of hull vibration amplitudes?

Torsional vibration nowadays constituted a well recognized problem which was dealt with both at the theoretical and experimental stages and the preventive evaluation of the stresses, induced in the propulsion machinery, attained a considerable accuracy. Both the propeller and the engine were able to excite torsional vibrations, the one independently of the other. In this connexion, he recalled a singular case which occurred in a set of propulsion machinery built by his company some years previously.

During the compilation of the torsional vibration records, the 4th harmonic vibration was appreciably greater than forecast and the suspicion arose that both the engine and the four bladed propeller contributed to this vibration excitation. In order to check this hypothesis, the propeller was rotated 45 deg. with respect to the engine, so as to obtain a 180 deg. phase displacement of the 4th harmonic excitation produced by the propeller.

The torsional vibration records made afterwards proved that the amount of stress due to the 4th harmonic was considerably reduced, as shown in Fig. 48.

When, therefore, a phenomenon similar to that just mentioned was reasonably expected to occur, it was advisable to build a shaftline section with the end flanges having different numbers of coupling holes, e.g. eight and nine; in this way it was possible to find out by trial the most suitable keying between propeller and Diesel engine (the suggested arrangements of holes allowed successive rotations of 5 by 5 deg.) for the purpose of counterphasing the vibrations produced by propeller and engine.

On page 133, the authors made a difference between fundamental and secondary harmonics which could excite torsional vibrations. Of course such a distinction had a limited value because, in consequence of the shape of the torsional vibration elastic line, the secondary harmonics were able to excite vibrations of a remarkable amount. For instance, in a four stroke, eight-cylinder engine the harmonics of 3·5, 4·5, 5·5 and 6·5 orders might cause considerable vibration amplitudes in resonance with the 2-node vibration.

Alternatively and always with regard to the shape of the elastic line, especially in the second vibration mode, the main harmonics could give rise to somewhat limited stresses.

As far as vibration records were concerned, it had to be pointed out that, adhering to the principle expressed by the authors that the more convenient point for recording the amplitudes was at the crankshaft forward end, the measurement of the torque by means of electric strain gauges placed on the shaftline did not furnish very satisfactory results when it was required to measure the stresses due to the second vibration mode.

The axial vibrations of propulsion machinery had latterly acquired some importance, as some trouble, due to such vibrations, had occurred in propulsion units, mostly those installed with Diesel engines. In his opinion, the difference between torsional and axial vibrations was only a rough distinction, convenient for calculation. A propulsion unit was indeed subjected to some forces and periodic moments which caused prevalently axial or torsional vibrations.

Accordingly, it should always be necessary to carry out a combined verification of axial and torsional vibration, but in most cases it was sufficient, for simplicity of calculation, to deal with the two arguments separately. In any case the causes of the axial vibrations were clearly identified and consisted of the motion which the piston transmitted to the crankshaft, causing axial movements in the cranks and also of the axial forces produced by the propeller.

Naturally, when an axial frequency coincided with or was twice the torsional frequency, some dangerous instances of contemporary large amplitudes of axial and torsional vibration might occur.

A reliable step for eliminating axial vibration was to fit a viscous friction damper, as mentioned on page 139 of the paper. On the engines built by his company, to which he believed the authors were referring when discussing the axial vibration damper, the latter was only installed when necessary, after the calculation, to establish whether the axial vibration main harmonic occurred within the normal range of engine operation, had been made.

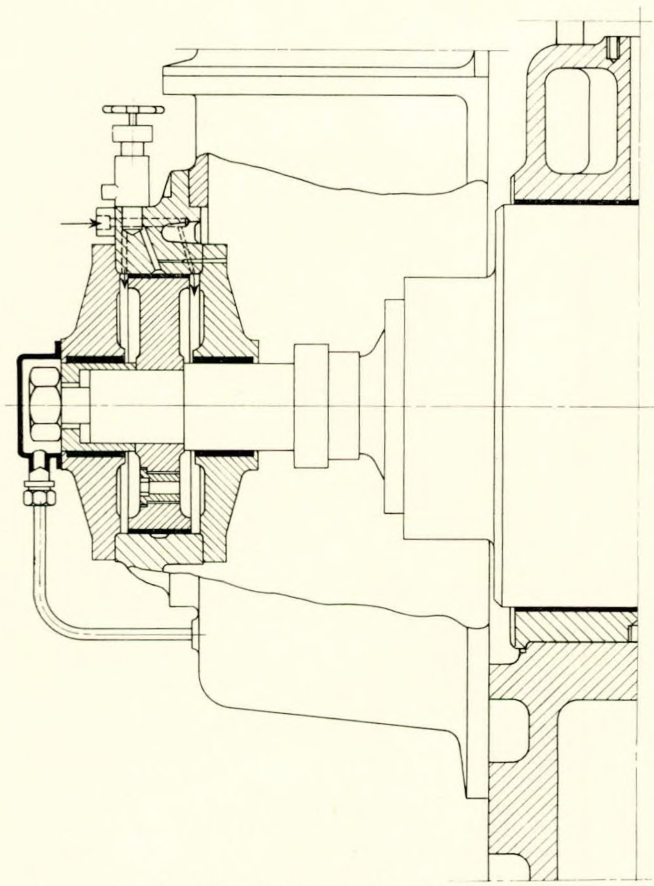


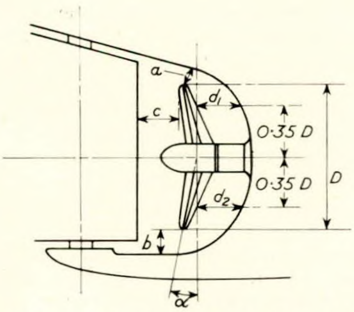
FIG. 49—*Axial vibration damper*

The damper manufactured by his company and shown in Fig. 49 was of a particularly simple type; it needed practically no maintenance and retained its damping characteristics, even after several years' service.

Fig. 30, on page 137, showed a scheme of the subdivision of the mass in a propulsion unit as far as axial vibration was concerned and gave a figure for the water mass, which was to be considered as part of the propeller in the calculation of axial vibrations, as equal to a 60 per cent increase of the propeller mass. Based on his own experience, he considered it necessary to allow about 100 per cent increase of the propeller mass to obtain a satisfactory agreement with the frequencies and the elastic lines recorded.

Transverse vibrations of Diesel engine frames generally did not give rise to particularly difficult problems to be solved. Usually such vibrations were not likely to occur and the corresponding amplitudes were within restricted limits. Naturally, this implied, on the part of the engine builder, a careful study of the firing order which, besides the recognized balancing exigencies, load on main bearings, etc., could prevent the occurrence of transverse vibration of the frame. He agreed with the authors about the difficulty of foreseeing, at the design stage, the frequencies and amplitudes of this type of vibration; however, engine builders were generally able to evaluate quite accurately the vibration frequencies of the engine frames for their own engines and to take the proper steps to prevent them from occurring in a normal operational range of the engine.

As suggested by the authors, this vibration could be eliminated by fastening the engine to the hull structure with the help of tie-bars. His company had met with success in testing a type of tie-bar divided into two parts, the parts being connected through a clutch coupling. Therefore, in addition to the modification of the proper vibration frequencies of the frame,

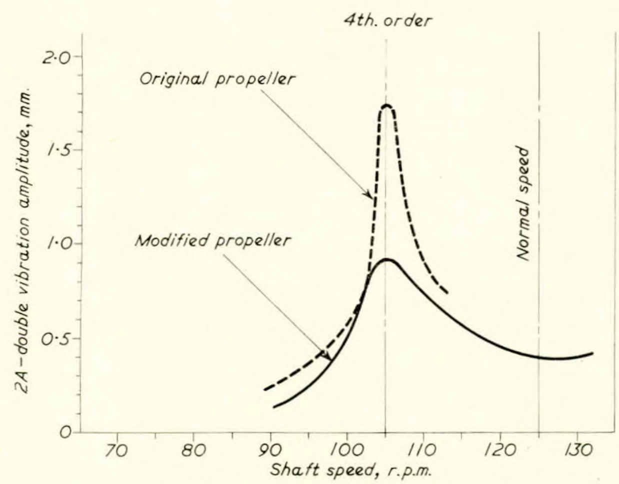


Distance of four bladed propeller from the hull:

Original propeller (diameter = 5,100 mm.)		Modified propeller (diameter = 5,005 mm.)	
a	= 380 mm.	a	= 475 mm.
b	= 133 mm.	b	= 155 mm.
c	= 775 mm.	c	= 475 mm.
d_1	= 775 mm.	d_1	= 1,015 mm.
d_2	= 575 mm.	d_2	= 820 mm.
α	= 13·75 deg.	α	= 15 deg.

Position of recordings: on the head of cylinder No. 6 (counting from flywheel)

Frequency recorded: F = 420 p/min.



Characteristics of Fiat engine type C756S:

Number of cylinder 6 (2 stroke—single acting)

Cylinder bore	750 mm.
Piston stroke	1,320 mm.
Rotating speed	125 r.p.m.

FIG. 50—*Transverse frame vibrations of m.s.* Ordu *main engine with two different propellers*

a damping effect was introduced into the system, limiting the maximum vibration amplitudes.

As confirmed by the authors, it had already been ascertained that the transverse vibration of an engine frame could also be caused by the propeller. In one particular case he had noticed that the engine was subjected to 4th harmonic vibration due to the four bladed propeller, installed inside a well of very limited dimensions. The replacement of the propeller by one more suitable, having a smaller diameter, practically eliminated the trouble.

Fig. 50, besides showing the clearances between the two propellers tested and the well, also indicated the transverse vibrations of the engine, before and after the replacement of the four bladed propeller.

On page 141, the authors stated their doubts about the efficiency of balancing a Diesel propelling engine as completely as possible, with respect to the hull vibration. In fact, the amount of hull vibration was not only to be attributed to the magnitude of the free forces, but also to their distribution along the elastic line of oscillation. The hull vibration was, therefore, proportional to the vectorial sum of the force product for the ordinates of the elastic line, in agreement with which such forces were applied. However, it should be noted that in engines not provided with a scavenge pump at the free end, where a moment due only to the inertia forces could be present as a free action, the aforementioned vectorial sum was nil when the moment of inertia forces was also nil, provided that the elastic line along the engine was practically a straight line.

Bearing in mind that the length of the engine was very short compared with the full length of the hull, the foregoing condition was nearly always verified to a good degree of approximation, unless the engine were in a particularly flexible area of the hull or in an anti-node where the bend of the elastic line was especially high. Therefore, he considered the policy of Diesel engine builders, to reduce the free inertia moments in the engine as much as possible, at design stage, to be justified.

MR. J. W. ECKHARD, in a written contribution, pointed out that this paper put into perspective many of the problems of vibration in ships and that the recapitulation of the elements of the subject at the beginning of the paper would encourage readers to continue to the end of the paper.

He believed that there was an inference on page 128 of the paper relative to types of sterns, which might imply that, for high powers on a single screw, the so-called conventional stern, of the U or V section could not be accepted from the vibration point of view and therefore, the "Alfred Holt Clearwater" (or *Mariner*) stern or the concentric bulb stern was essential.

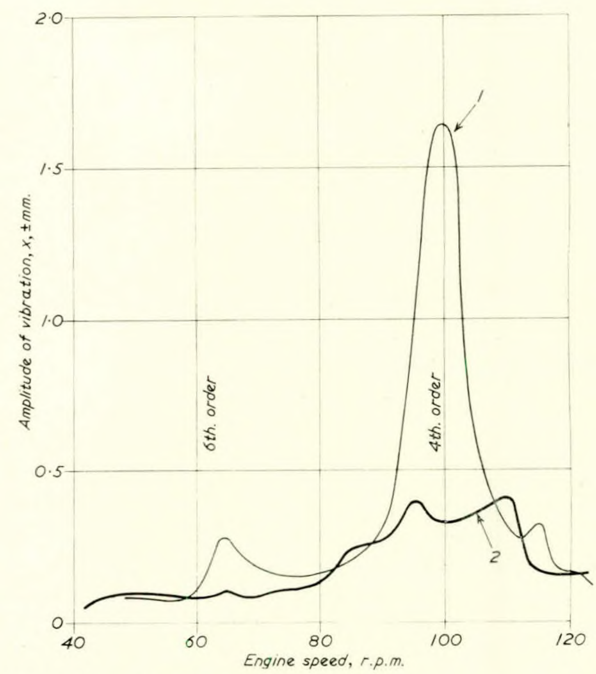
What did the authors mean by a conventional stern?

Well designed sterns, with Simplex type rudders were present on single-screw vessels of higher powers than quoted in the paper and it was well known that where propeller clearances had been provided well in excess of those recommended by the classification societies such sterns were free from propeller induced vibrations.

Thus one could obtain this freedom, without being fashionable and extravagant and without resorting to the expensive forms of stern shown in Fig. 13(b) and (c).

MR. A. KLEINER wrote that he wished to congratulate Dr. Johnson and Mr. McClimont for their paper, dealing with a very topical problem. This paper also contained the essential points concerning engine vibrations, in which naval architects were interested. For an engine builder it represented an excellent resumé of the latest knowledge of ship vibrations. As engine builders, his company was very glad to learn that during the last few years hull vibrations, which were excited by the propeller, had been investigated very thoroughly, particularly by B.S.R.A. Not so long ago, the main or auxiliary engines were thought to be responsible for all hull vibrations and the engine builder was asked to eliminate them. He welcomed this opportunity of adding, from his company's experience, a few remarks about this problem.

It was more or less impossible to calculate exactly the natural frequency of the transverse vibration of the engine. Fortunately, such a calculation was not at all necessary, at least not for engines of normal height. However, this might be different for opposed piston engines. The tie-bars mentioned by the authors represented a cheap and effective detuner for such vibrations. Those tie-bars shored the engine, at the level of the cylinders, against the decks, increasing the natural frequency of the transverse vibrations, so that an exact calculation was unnecessary. Therefore, it was generally recommended that provision should be made for these tie-bars to be fitted on all ships equipped with large engines. If such provision was made from the very beginning, the fitting of these tie-bars was not expensive. The engine builder did not know in advance whether there would be disturbing torques or forces due to the propeller, which might be in resonance with the transverse vibrations of the engine.



1) Engine originally not stayed
2) Engine definitely stayed
The vibrations of the 4th order are excited by the propeller
The vibrations of the 6th order are excited by the crossheads

FIG. 51—*Horizontal athwartships vibration on the cover of a six-cylinder two stroke marine engine*

Fig. 51 showed a typical case of a transverse vibration in a six-cylinder two stroke engine, many of which had been built before and had never required tie-bars. A very small parallel transverse vibration, excited by the forces from the crossheads, of the 6th order at 64 r.p.m. had a maximum amplitude of only $\pm 0 \cdot 3$ mm. in the resonance. It was obvious that this did not cause any disturbance at all and that no damper was necessary. The extraordinary feature in this case was, however, that a very pronounced parallel transverse vibration of the 4th order, resulting in a maximum amplitude of $\pm 1 \cdot 65$ mm. at 98 r.p.m. was experienced. Obviously, this vibration had not been excited by the crosshead forces. The excitation of the 4th order was either caused by the torque variations or by the forces coming from the four bladed propeller. Tie-bars fitted at the level of the cylinders were able to cure this phenomenon completely. As there was no provision made on board this vessel for the fitting of such bars, the installation of the latter was rather costly. In such a case, there would always be disagreeable discussions as to who had to pay for the additional cost, engine or shipbuilder? For this reason, his company generally recommended that provision for such tie-bars be made right from the beginning—even if they were not expected to be necessary.

The forces to be carried by these tie-bars were only a few tons and it was advisable, not to make them so strong, that they could not break, should the forces acting from the hull to the engine become excessive, as, for instance in the case of a collision. This arrangement did not require the fitting of shear bolts.

Axial vibrations of the shafting were becoming more and more important in motor vessels with the increasing size of Diesel engines and, therefore, increasing length of crankshafts.

The calculation of the natural frequency of axial vibrations was very difficult, since there were three vital unknown facts:

1) It is not known which part of the connecting rod masses takes part in the axial vibrations.
2) It is not known how much the axial elasticity of a crank throw is influenced by the clamping effect of the main bearings.
3) The effective elasticity of the thrust bearing is not known.

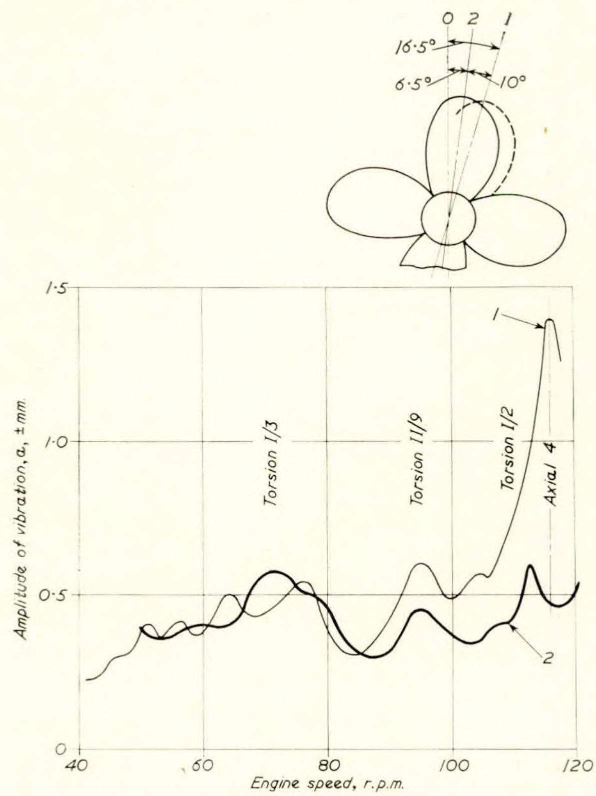
In the case of the calculation of an observed axial vibration, there was an opportunity of neglecting two of these factors and to dimension the remaining third so that the calculated natural frequencies conformed with the measured values. It might be assumed, for instance, that the mass of the connecting rods or of the moving parts had no influence on the axial vibration and also that the clamping effect of the bearings was not effective. However, it was not certain how far this assumption was correct. Observations on stationary plant, where the engine drove a heavy generator and where the elasticity of the thrust bearing had a quite different effect than on ship installations, permitted the presumption that at least the rotating part of the connecting rod masses took some part in axial vibration.

However, the axial elasticity differed if the crankshaft were properly bedded in the bedplate or if this shaft could be considered as free deformable. There was certainly some influence from the main bearings.

All the described observations did not of course, simplify a pre-calculation of the natural frequency of an axial vibration.

Axial vibration of the crankshaft and of the line shaft could be excited by the engine itself and also by the propeller. In cases where the excitation of the same frequency came from both sources, there was a superimposed forced axial vibration. In such a case, the possibility existed of phasing the excitation of propeller and engine, so that the exciting forces were counteracting each other, reducing the resulting forces to a minimum.

Some years ago his company had a very interesting case, in a British cargo liner, which was cured quite easily, the vessel now operating to the full satisfaction of the owner. At the forward end of the crankshaft of the 12-cylinder engine, an unexpected axial amplitude of the 4th order was identified at full load speed, having a value of $\pm 1\cdot 4$ mm. The position of the four bladed propeller with respect to the crankshaft was altered by only 10 deg., almost completely suppressing the excitation of the disturbing axial resonance (see Fig. 52).



0) Piston No. 1 at top dead centre
1) First position of the propeller
2) Definite position of the propeller

FIG 52 —*Axial vibrations at the fore end of the crankshaft of a 12-cylinder marine engine*

Referring to the problem of balancing the engine the authors recommended—as opposed to the usual practice—choosing a crank sequence, balancing as much as possible the couple of the inertia forces of the 2nd order. The balancing of the couple of the 1st order had to be considered last. Theoretically, the authors were absolutely right, as the unbalanced couple of the 1st order could be compensated by counterweights on the crankshafts. However, it was rather difficult to find enough room for the fitting of such large counterweights, even when bolting weights as large as possible on each web of each crank, which required the total available space of the bedplate. These balance weights would certainly have a very pronounced compensating effect, but the usual accessibility to the main bearings would be sacrificed by such a design. His company considered this a nearly unacceptable disadvantage for the maintenance and service of an engine.

He further wrote: "Hull vibrations which are excited by an unbalanced couple of the 2nd order in a slow running engine (below 200 r.p.m.) have, according to experience, a very sharp resonance. They have therefore a very large magnification factor. Consequently, they can only be noted in a very small speed range of the engine. In most cases it is sufficient to increase or decrease the engine revolutions by 2-3 per cent to be outside the disturbing vibrations.

Furthermore, such vibrations are more pronounced on a light ship than on a loaded one. If vibrations of 2nd order should appear on a light ship, they can be eliminated by a small alteration of the engine speed without any principal modification on an otherwise approved engine type—a modification which may represent disadvantages in other respects.

In particular cases, where a naval architect asks for good balancing of both the couples of 1st and 2nd order, it is advisable to choose an engine type which complies better with this request—e.g. two stroke-engines with more than 6 cylinders."

MR. J. BURTON DAVIES, B.Sc. (Member) wrote that there were many points in this most useful paper which were worthy of comment, but special mention might be made of the authors' summary of previous work on propeller excited vibration and their presentation of some new ideas on the subject. The sections on wake forces and vibration of shafting systems raised again the question of the choice of the number of blades and while, clearly, much more knowledge was required before the optimum selection for a given hull and machinery combination could be made with complete confidence, perhaps the authors would agree that the six bladed propeller had not had the consideration it merited.

With typical propeller-excited vibration as shown in Fig. 15, the problem might well be one of ensuring that the propeller forces were kept to an acceptable level rather than predicting critical frequencies and the authors' approach to force-amplitude relationships certainly appeared to capture the trends of actual measurements. The method should prove extremely useful when propeller forces could be predicted with greater accuracy than was now possible.

He had felt, for some time, that the basing of propeller tip clearance solely on propeller diameter was hardly adequate and the proposed Φ value to include parameters other than diameter would be examined with interest.

It would appear that different definitions of the length L had been used in the two parts of the paper. In Fig. 40 L was the length overall and while L in formulæ (4) and (5) was not specifically defined, it was noted it was the length between perpendiculars that was used in Table I. Perhaps the authors would clarify this. The writer would put in a plea for length b.p. as being more convenient even if possibly not quite so accurate.

MR. C. W. IRVIN wrote that the problem of hull vibration and vibration associated with main machinery was one which assumed greater importance with increasing powers. This paper should prove very valuable to design offices by enabling the hull frequencies and amplitudes to be calculated fairly readily at an early stage in the design. It was particularly valuable to have formulæ for acceptable amplitudes of vibration.

The section on propeller-excited vibrations brought to light some interesting facets of the problem and should help to decide the question of how many blades the propeller should have.

From an examination of Fig. 14 and taking the conventional stern "A", there would appear to be no advantage in departing from the normal four bladed propeller except insofar as thrust and torque were concerned, the effect of which was more likely to be felt on the machinery rather than on the hull.

Again, referring to Fig. 15 it seemed to be impossible to avoid resonance occurring at one or more of the higher modes of vibration whatever the choice of the number of blades, but in this region there were no well defined peaks and the amplitudes of vibration were comparatively small.

So far as hull vibration was concerned the aperture clearances would appear to be the important criteria and the authors' approach to this problem should help in the selection of suitable clearances.

By using equation (25) and Fig. 20 the minimum acceptable tip clearance could readily be obtained and the remaining clearances could be derived from Fig. 19 on a *pro rata* basis. By adopting this technique it should be possible to ensure that no excessive propeller-excited hull vibration would occur irrespective of whether a four or five bladed propeller was adopted.

Referring to Part II of the paper, which related to the vibration of the propulsion system, the torsional vibration problem was now well under control, the calculation of frequencies and stresses being a fairly routine procedure and submitted to Lloyd's for approval.

The characteristics of axial and transverse vibration of the shafting were not so well understood. One of the difficulties so far as axial vibration calculations were concerned was the determination of the stiffness of the thrust block and it was interesting to note that B.S.R.A. was now carrying out systematic tests on thrust blocks and thrust block seatings.

Another difficulty insofar as Diesel machinery was concerned was the calculation of the crankshaft stiffness and in this connexion the formulæ given by the authors should be very valuable.

The accurate prediction of screwshaft transverse vibration frequency would also appear at present to be somewhat problematical. One difficulty was that as wear-down of the stern tube bearing occurred the critical speed was likely to come down. This seemed to be an argument in favour of adopting white metal bearings.

The transverse vibration of engines was even more unpredictable. It was thought that models could be used to help in the solution of this problem. One normally assumed that this type of vibration was engine excited, with a critical speed depending on the natural frequency and the number of cylinders. This presumably would be the case if the mode of vibration were always as in (a) of Fig. 35. If, however, the mode of vibration was not parallel then other order numbers could presumably become significant. The shafting must surely have an influence on the mode of vibration and perhaps the authors could comment on this aspect.

It was also interesting to note that a transverse engine vibration might well be excited by a propeller-excited hull vibration, so that, irrespective of the number of cylinders, the possibility of a critical speed at propeller blade frequency should not be overlooked.

In regard to the effect of oil engine unbalance, the authors' plea for engines to be designed with primary rather than secondary unbalance was very interesting and deserving of further study.

Incidentally, it would appear that more consideration should be given to the effect of the position of the engine in the ship. Indeed the formula for vibration amplitude, equation (10), should be the criterion rather than the accepted figures for out of balance forces and couples. It appeared probable that, by judicious placing of an out of balance force at a suitable anti-node of the hull, profile hull vibration could be effectively damped, even though there was a resultant out of balance force. Conversely, it would be possible for a badly placed engine with no out of balance forces or couples in the accepted sense to excite hull vibration.

In the section on oil engine axial vibrations mention was made of the connexion between torsional and axial vibration. Mr. Irvin had observed that an apparent axial vibration occurred in way of the torsional critical speed but had always assumed that this was not a true axial vibration, but purely due to the excessive twist occurring in the crankshaft. It was normally possible also to detect a pronounced engine excited axial vibration with an order number equal to the number of cylinders and a frequency equal to that of the propeller excited axial vibration.

MR. H. LACKENBY, M.Sc., in a further written contribution amplified remarks made in the concluding paragraphs of his verbal contribution concerning the thickness of the frictional belt shown in Fig. 21 and the statement made in the paper, that it might be prudent to ensure that the tip clearance was not less than that indicated in that figure—that was the boundary layer

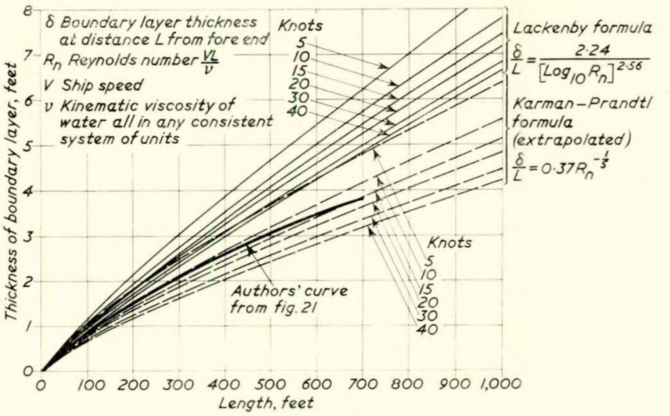


FIG. 53—*Thickness of boundary layer in way of propeller*

thickness. In this connexion Fig. 53 showed for ship lengths up to 1,000 feet boundary layer thicknesses according to the following:

i) the Karman-Prandtl formula for plane surfaces;

ii) the writer's formula developed in the 34th Thomas Lowe Gray Lecture*.

* Lackenby, H. 1962. The Thirty-fourth Thomas Lowe Gray Lecture—"The Resistance of Ships with Special Reference to Skin Friction and Hull Surface Condition". Proc. I.Mech.E., Vol. 176, p. 981.

The Karman-Prandtl formula was based on velocity distribution in the boundary layer according to the well known "seventh power law" and was only intended to be valid up to Reynolds numbers of about 10^7. This limit was reached at a length of 15ft. at 5 knots and 2½ft. at 40 knots which meant that for all practical purposes almost the whole extent of the Karman-Prandtl curves shown in Fig. 5 was in the extrapolated region and open to some doubt. The writer's formula should be applicable to the highest ship Reynolds numbers. It would be seen that it gave somewhat greater boundary layer thicknesses than the extrapolated Karman-Prandtl formula.

It would be noted that these formulæ indicated a significant speed effect—the boundary layer thickness diminishing with increasing speed at a given length.

The frictional belt thickness given in Fig. 21 was also shown in Fig. 53 where it would be seen that it corresponded closely to the Karman-Prandtl extrapolated values for speeds of about 10 to 15 knots.

The point made here was that if the Karman-Prandtl formula had any influence in determining the curve of boundary layer thickness shown in Fig. 21 then one would have to bear in mind the limitations and other matters referred to above. At the same time it had to be remembered that the "run out" of the velocity distribution in the boundary layer was very gradual indeed in the region of its full thickness and as far as hull-tip clearance was concerned there was little doubt that a significant reduction on the full width could probably be considered without adverse effect on hull and propeller. In this connexion it had been suggested by Van Lammeren† that a hull tip clearance of 80 per cent of the full boundary layer width would be adequate.

It might be reasonable in the circumstances to apply such a percentage to the writer's curves shown in Fig. 53 which would give the following formula for tip clearance:

$$\text{tip clearance} = \frac{1 \cdot 79 . L}{[\log_{10} R_N]^{2.56}}$$

where R_N = Reynolds number $= \dfrac{VL}{\nu}$ in a consistent system of units.

Taking ν for salt water at 59 deg. F. $= 1 \cdot 278 \times 10^{-5}\text{ft.}^2/\text{sec.}$, V_K = speed in knots and L = length in feet this reduces to:

$$\text{tip clearance in feet} = \frac{1 \cdot 79 . L}{[5 \cdot 1211 + \log_{10} V_K . L]^{2.56}}$$

As it happened clearances calculated by this formula agreed closely with the authors' curve for speeds of 10 to 15 knots and lengths up to about 300 feet. For longer lengths, however, the speed effect was more significant, and the authors' curve would generally underestimate the clearance except at very high speeds.

Again one had to bear in mind that the formulæ discussed above applied strictly to smooth plane surfaces rather than to the hull form in way of the propeller. Nevertheless, in the absence of more adequate data it would appear not unreasonable to relate hull-tip clearances to such boundary layer thicknesses.

† Van Lammeren, W.P.A., Troost, L. and Koning, J. G. 1948. "Resistance, Propulsion and Steering of Ships", p. 279.

Authors' Reply

In replying to the discussion the authors said in answer to Mr. Jackson that while the paper did not give the differences between hull frequencies in the loaded and light conditions the displacements relating to the measured frequencies were given in Table I. From these results it was a simple matter to obtain an indication of the effect of changing displacement by using equation (5) in association with the relevant coefficient, the total virtual mass being modified accordingly using equations (3) and (4). With regard to the influence of depth of water it should be mentioned that while the effect on vertical modes of vibration could be considerable, the horizontal modes of vibration were relatively insensitive to this effect.

Mr. Jackson made the interesting observation that while vibration from the propeller could be serious he had not yet encountered any which he could attribute to " wake ". In instances of the propeller-excited vibration it was not generally possible to determine the separate contributions made by pressure forces and wake forces. The authors held the view, however, that for average hull forms the wake forces formed a substantial proportion of the total force.

With regard to the suggested limits of acceptable vibration, the authors should have made it clear that those were really intended to apply to the extreme aft end of a ship. This meant that at propeller-excited frequencies the amplitudes throughout the main body of a ship would be somewhat less than that at the extremities and it was hoped that this explanation would meet Mr. Jackson's objections.

The authors concurred on need for more research into higher harmonics of the tangential effort as engine ratings were raised. This called for sophisticated indicating techniques and a refinement of existing data on higher harmonics using modern indicating techniques would also be valuable. The use of a computer would speed the harmonic analysis of records.

The use of a computer programme for the determination of axial vibration frequencies was to be commended but it might be pertinent to observe that a computer was a " glorified slide-rule " and that the accuracy of the results obtained by its use was only as good as the accuracy of the physical data with which it was supplied and the method of analysis employed.

The authors thanked Mr. Jackson for underlining that Fig. 29b was an example and not a general criterion. There were advantages and disadvantages associated with any number of propeller blades; the important thing was to weigh these up for each application and make the appropriate selection.

Fig. 32 was intended to be diagrammatic and the authors had tried to avoid making it look like any contemporary engine. They appeared to have carried this approach too far for Mr. Jackson's liking. They were grateful for details of the Doxford J-engine crankshaft though they believed that even this did not represent Mr. Jackson's view of the ultimate in crankshaft rigidity.

The authors were intrigued to find that engineers more intimately concerned than themselves with the day to day running of installations looked so favourably at the synchronizing of twin engines. They had always been hesitant to put forward this proposal for the solution of vibration problems as being one which would be regarded as the approach of " academic men " who did not give due consideration to practical problems.

Mr. Jackson's observations on the selection of firing order were fully endorsed; the authors were now sorry that in the interests of brevity in an already long paper they deleted their observations on the other considerations which arose on the selection of firing order. They wished also to take this opportunity to stress that although they had mentioned a 5-cylinder engine as an example in the paper, too much should not be read into this. It was possible to have good balance or bad balance in an engine of any number of cylinders; admittedly it was easier to achieve good balance with an engine with a certain number of cylinders but it should never be assumed that the number of cylinders was a direct indication of the balancing characteristics of an engine. A poorly balanced 6-cylinder engine might well be less acceptable than a well balanced 5-cylinder one.

The target of 200 tons-ft. for the maximum couple appeared a very reasonable design basis for large marine engines for general application.

In reply to Mr. Bunyan as the authors mentioned in the paper, the data given in Fig. 14 was collected from various papers and as a consequence it would not be correct to say that the wake conditions were precisely the same for each stern arrangement. Care was taken, however, to ensure in so far as it was possible that the results selected were related to similar forms, i.e. tanker types and in this sense the authors felt that the comparisons were realistic.

Unfortunately, the authors could not confirm from their experience that the vibration amplitudes as given by equation (11) were seldom excited in modern tonnage. Much of the work of the vibration section at B.S.R.A. was taken up in investigating excessive vibrations from the propeller and also from other sources such as secondary unbalance in main machinery.

The authors' comments on stern tube bearings had been restrained in view of the paper to be read a month later by Batten and Couchman[18]. Experience to date indicated that oil-lubricated stern bearings did offer advantages in the avoidance of tailshaft vibration phenomena. Slope boring correctly applied, was fundamentally sound for all forms of tailshaft bearing and again experience indicated that it could give in practice the expected improvement.

Mr. Bunyan, referring to Fig. 15, had asked about the acceptability of the 4-node vertical mode of vibration if excited by the secondary unbalance of a nine-cylinder Diesel engine at 105 r.p.m. The tolerable secondary couple with the engine at a position 25 per cent from the aft end would be 566 tons-ft. and most modern nine-cylinder engines would be expected to be balanced to within this limit. If the engine were more nearly amidships (the worst point being at a position about 38 per cent from the aft end), the tolerable secondary couple by most pessimistic estimates would fall to about 105 tons-ft. and by most optimistic estimates about 160 tons-ft. This degree of balancing could be achieved on a nine-cylinder engine but it was doubtful whether it would be available as a standard production job although the remarks of Mr. Jackson should be noted in this context. The occurrence of engine-excited vibration could not, therefore, be discounted.

Experience in this country with really large tankers propelled by large bore oil engines was hardly sufficient to justify comment on whether vibration was a problem. One could, however, observe that with proper care it need never become a problem.

Regarding Mr. Nestorides' comments the authors doubted whether the tabulation for various engines of the unbalance forces and couples would be informative; whilst they had many data they would hesitate to present these as an up to date picture which did due justice to the many manufacturers and designs concerned.

The study of the response and transmissibility of engine seatings was not being neglected. However, in the case of large direct-coupled engines, problems of alignment and of engine movement in heavy seas would appear to limit the scope for the redesign of engine supports. Nevertheless, it would be unwise to be over-conservative in ones approach to this matter.

Mr. Nestorides was surprisingly pessimistic about the accuracy of strain gauges and the authors would refer him to the remarks of Mr. Silverleaf. The authors had certainly been at fault in quoting accuracies without mention of the corresponding level of measured stress. They had a stress level of about $\pm$3,000lb./sq. in. in mind when quoting accuracies of $\pm$ 10 per cent uncalibrated and $\pm$ 2 per cent calibrated.

The authors did not think that sufficient was known about transverse crankshaft vibrations to enable assumptions to be made other than speculatively about the important parameters. The current development of engines of higher output per cylinder and, perhaps more important, with more cylinders on one crankshaft called for an intensive study of this whole field.

Mr. Nestorides had drawn attention to the magnitude of the undertaking involved in the study of the entire engine vibration of a medium size engine. The authors could confirm that the task was certainly no less onerous for a large slow speed marine oil engine. B.S.R.A. had taken many measurements but these had served principally to indicate the complexity of the problem.

Mr. Lackenby asked whether guidance could be given on the extent to which it was desirable to avoid resonant vibration in order to avoid the worst conditions. It would be difficult to give guidance in a few words without running into certain ambiguities. The adjustment in frequency which might be necessary to reduce a serious vibration to an acceptable level would, in the first instance, depend upon how far the resonant peaks intruded beyond the boundaries of acceptability. It would also depend, as Mr. Lackenby indicated on the degree of damping associated with the particular mode of vibration under consideration. Further points to be borne in mind were that the resonant frequency would alter according to the loading condition and that in a normal seaway steady state conditions did not exist, the revolutions varying to some extent with ship motions. Some idea of the reduction of amplitude to be gained by altering the forcing frequency from the resonant frequency was given in Fig. 25 of Reference 2.

Mr. Lackenby's comments regarding Fig. 7 were appreciated. This particular form of presentation was adopted in preference to a tabulated statement since certain trends could be more readily perceived. The reasons for the trends illustrated were not fully understood at the present time but attempts would be made to rationalize the data in the manner suggested by Mr. Lackenby.

With regard to Fig. 10, the results for propellers having three and four blades were derived from models in which the propellers had different blade area ratios and different pitches. Within the practical range chosen, however, it appeared from Ramsay's experiments that the pressure coefficient was relatively insensitive to differences in these parameters. The results for the 5-bladed propeller were in fact computed on the basis of the other results.

The authors were greatly obliged to Mr. Lackenby for clarifying the situation with regard to the boundary layer thickness. The curve shown in Fig. 21 was based on published data and a certain amount of experimental evidence but had certainly not taken full account of the Reynolds number effect. Fortunately, the graph in the paper gave roughly the right answer but not necessarily for the right reasons. Mr. Lackenby's alternative formulation was clearly to be preferred particularly as it took acount of the speed effect which became significant at ship length above 300 feet.

It would probably be correct to say that the valuable experimental investigations to which Mr. Archer referred had broadly confirmed the methods for calculating entrained mass which had been in use for some years. The semi-empirical formulae had been based on these well-established methods and such deviation as would be indicated by the more recent experiments were likely to be of a small order.

The authors were grateful to Mr. Archer for his detailed examination of equations (11) and (12) and were pleased to think that he did not regard them as restrictive. They were as he suggested intended to apply to the aftermost anti-node. This definition was chosen for simplicity as to identify an acceptable level for every location in a ship adds considerable complications, as he would appreciate from Figs. 15 to 18. The point which he made, however, was perfectly valid and further research must be carried out in order that the concept of acceptable limits could be taken to its logical conclusion. One of the important steps in such research was to obtain a more complete understanding of the response characteristics of hull girders at propeller-excited frequencies and some indications of the way in which this was being tackled by B.S.R.A. were outlined in the paper.

Mr. Archer's comments on Fig. 14 were very interesting. The authors had stated in the paper that it would be imprudent to place too much significance on the amplitudes given in this figure, but that the trends might be substantially correct and it was encouraging that Mr. Archer also appeared to take this view.

His observations concerning the application of equation (29) to a 6-cylinder 750 mm. bore $\times$ 2,500 mm. combined stroke opposed piston engine crankshaft were very welcome. These indicated that the equation was not as generally applicable as had been thought; the reduction of the general formula to the expression $k = \dfrac{10^6}{78 - 415\cos\theta}$ tons/in. in this particular case showed that anomalous results could be obtained from the equation as stated in the paper since this particular expression would indicate that when the angle between cranks was 79 deg. the crankshaft would be infinitely stiff. Accordingly the derivation of expressions (28) and (29) had been re-examined and it had been concluded that $\cos\theta/2$ was a more appropriate term in which to express the effect of the angles between cranks. At the same time it had been considered that the influence of adjacent cranks had been overestimated and this had also been modified. There was considerable diffidence about putting forward the revised equations at this stage; however, the modified expression for the single throw of the type in Fig. 31 is:

$$\frac{1}{k} = \frac{C^2}{6E}\left[\frac{b_p}{I_p}(6 - \cos\theta/2) + \frac{b_j}{I_j}\cos\frac{\theta}{2} + \frac{C}{I_w}(4 - \cos\theta/2)\right]$$

and the corresponding expression for the opposed piston engine crankshaft as shown in Fig. 32 is:

$$\frac{1}{k} = \frac{C_{cp}^{\,2}}{6E}\left[\frac{b_{cp}}{I_{cp}}(6 - \cos\theta/2) + \frac{R_1 b_j}{I_j}\cos\theta/2 + \frac{C_{cp}}{I_{cw}}\left\{(4 + 2R_1) - (1 + 2R_1)\cos\theta/2\right\}\right]$$
$$+ \frac{C_{sp}^{\,2}}{6E}\left[\frac{2b_{sp}}{I_{sp}}\left(6 - \frac{\cos\theta/2}{R_1}\right) + \frac{b_j}{R_1 I_j}\cos\theta/2 + \frac{C_{sp}}{I_{sw}}\left\{(4 + 2R_2) - \frac{(1 + R_2)}{R_1}\cos\theta/2\right\}\right]$$

The use of this expression gave a value of 684 tons/in. for the half-shaft without flexible coupling in the case of the 6-cylinder, 750 mm. bore $\times$ 2,500 mm. combined stroke opposed piston engine crankshaft which formed the subject of reference

14 of the paper. The same expression had been used to derive a stiffness of 217 tons/in. for the complete shaft of a 4-cylinder, 1,600 mm. bore × 2,320 mm. combined stroke opposed piston engine with flexible coupling and the critical frequency estimates derived therefrom were within 4 per cent of the observed values in an extensive investigation. The same expression had been used to derive the stiffnesses of a single crank throw of the 670 mm. bore × 2,100 mm. combined stroke P type Doxford giving values of 3,310 tons/in. and 2,980 tons/in. for throws at 90 deg. and 120 deg. respectively and these values accorded well with the critical frequencies which appeared to apply to engines of this type.

In reply to Mr. Couchman, the determination of the part played by journal bearing restraint in the axial deflexion of a dynamically loaded crankshaft would be exceedingly difficult to determine experimentally. Much more data would be needed, particularly from ten and twelve cylinder engines before there was a reasonable prospect of noting whether journal restraint was a factor to be taken into account of these longer engines. Should it prove to be significant the authors would hazard that allowance for it would have to be made by empirical modification of any expression for crankshaft stiffness.

In the transverse vibration of engines, the engine and its seating certainly could not be considered in isolation from the hull and the authors had mentioned this briefly in connexion with the phenomena observed in the tug to which Fig. 37 of the paper referred. The principal lesson here might be that the use of tie-bars might not always be a solution to transverse vibration problems.

Mr. Hinson's observations on acceptable limits were most welcome and tended to underline some of the comments made by Mr. Archer. The particular figures quoted by Mr. Hinson of 0·4ft./sec.^{2} at 50 c.p.m. and 1·0ft./sec.^{2} at 800 cycles/min. would not seem to the authors in the light of their experience to be unduly high in living quarters in ships. They were inclined to the view that to adopt the Meister curves at the level suggested might be unduly restrictive at the present stage of development.

Fig. 10 was intended only to indicate the trends to be expected regarding the variation of pressure forces with axial and tip clearances and for varying numbers of blades and the authors had commented in reply to Mr. Lackenby on the effect of design parameters on these results. Reference 5 dealt specifically with the effects of wake forces and Mr. Hinson's observations presumably related to the combined effect of the wake and pressure forces. It seemed probable that the pressure forces were the least likely to be affected by variations in the propeller design parameters, these being largely a function of the intensity of loading.

Mr. Hinson's remarks on the determination of transverse engine vibration criticals and the use of tie-bars underlined very usefully the views of the authors on these matters. It was doubtful, however, whether amplitude was the correct criterion of acceptability. The authors were not yet clear exactly what was unacceptable about an engine which vibrated transversely; perhaps damage to instruments mounted on the engine might be the most significant consideration in which case acceleration would be the most useful criterion. Mr. Hinson's suggested guide to acceptability would give an amplitude of 0·010in. at a position 20ft. above the tank top; from instrument considerations this appeared conservative and a value of up to three times this would not appear unreasonable.

Mr. Silverleaf's comments were as usual both stimulating and provocative and the authors must be excused for not taking up in great detail every point which he raised.

They believed that Mr. Silverleaf had attempted to read more into Fig. 7 than was intended. As explained earlier this was essentially a visual presentation of some results which was thought to show some trends more clearly than if they had been tabulated. The data indicated that the coefficients tended to diminish for ships having finer forms and they were arranged simply to bring out this effect. Other observations on these results were given on page 124.

It was not possible to be specific on the question of accuracy with which hull critical frequencies could be calculated in the design stage. If measured frequency data were available for a similar ship, it was possible to make very accurate estimates making due allowances for any differences in section inertia, total virtual mass and length. The accuracy would depend entirely on how closely a new design could be identified with that of the ship for which measured data were available. An alternative was to make use of graphical presentations of accumulated full scale results or approximate formulae such as those given in Reference 1, in this case the accuracies might be expected to be of the order of 5 per cent for two-node vibrations, 10 per cent for three-node vibrations and 15 per cent for four-node vibrations.

The figures quoted in no way invalidated the necessity for calculating the section inertia from the structural drawings. Attempts to use simple formulae based on ship dimensions had not proved very successful and could often lead to gross errors. In this connexion the variation of the coefficients in Fig. 7 for different ships might not be related in any way to section inertia but perhaps more probably to the distribution to shear resisting material, ship masses and entrained water.

With regard to the curves given in Fig. 8, the authors were of the view that the greater degree of damping in horizontal vibration was related to the additional energy absorbed in torsional movements which were always induced by horizontal vibration. This was a consequence of the absence of a horizontal plane of symmetry for the mass-inertia forces.

On the subject of propeller-excited forces, Mr. Silverleaf, made a number of criticisms which, however, he had not substantiated with any results. The authors were well aware of the recent publications to which he referred, but admitted to some difficulties in reducing those to the simple engineering terms which had been achieved in the paper using earlier theories. It was hoped that Mr. Silverleaf would take an early opportunity to substantiate his comments by publishing his results together with the comparisons to which he referred.

In referring to Fig. 10 which was based on carefully conducted experiments, the authors were careful to state that this was only indicative of trends. Mr. Silverleaf's comments would have been more constructive if he had stated the extent to which his own observations differed from those of Ramsay.

With regard to Fig. 14, it was not correct to say that much of this was based on " extremely inadequate quasi-state theory calculation ". In fact the bulk of the data was based on experimental results. The authors had clearly expressed their doubts concerning its quantitative value as applied to ships, but believed that the trends were realistic. Perhaps this was all that could really be expected from model tests of this type, but the value of such results was nevertheless considerable in helping our understanding of the general nature of the problem.

Mr. Silverleaf's well known experiments on the exchange of energy between linear and torsional movements helped to make an interesting point. This experiment, however, would have made more impact if he could have demonstrated the behaviour of the system under more realistic conditions, that is, under the influence of a pulsating force.

The authors were, of course, aware of the results obtained which suggested the presence of propeller-generated forces of one less and one more than the blade frequency. These data were in the form of records of the stress variations at points in the shaft and had been derived by the use of strain gauges. To derive information on the forces at play, careful interpretation of those data was necessary; certain assumptions must also be made about the validity of the strain gauging techniques. The authors suggested that the generation by the propeller of forces at these unexpected frequencies could not yet be accepted beyond question.

The authors felt that they could not do better on the matter of the accuracy of strain gauge measurement of torque than refer to the comments of Mr. Nestorides. The determination of the real accuracy of any measurement technique was always difficult and few subjects were more contentious. So much depended on the standpoint from which one looked at a particular case, whether or not one wanted to believe the results. B.S.R.A. had a programme at the moment in which it was trying to assess this matter as disinterestedly as possible.

Mr. Bourceau's comments on the effect of skew back on wake force were most interesting and his 1961 A.T.M.A. paper which they had unfortunately overlooked would be studied with interest. The authors had no strong opinions on the subject on which there was little available evidence in published literature. The view expressed in the paper was based largely on model work described in reference 5 in which propeller design parameters were varied systematically. The caveat to the appropriate paragraph on page 14 was based on some full-scale tests with which the authors were concerned.

In an earlier paper, which was reference 11 of the present paper, the authors drew attention to the potential advantage in respect of axial vibration of the shafting which derived from siting of the thrust block as far aft as permissible from the main engines, up to one half the length of the shafting. Whilst they had no direct experience of the matter, they would anticipate that this could also be beneficial to the alignment of the shafting at the point where the shafting links with the main engine, whether this involved a main gearwheel or a crankshaft coupling. They had evidence of much heavier loading on the lower half of a thrust collar than on the top half, indicating a substantial bending moment on the thrust collar. The resultant deflexion of the shafting would be less severe at the aft end of the engine when the thrust block was as remote as possible; the insertion of a bearing between the thrust block and the engine would also assist in ameliorating the effect.

Mr. Bourceau would no doubt have noted the favourable comments of Mr. Jackson and Mr. Archer on the applicability of formula on page 143 and these would no doubt go a long way to answering his question about the views of British engine builders.

The authors were grateful to Mr. Anscomb for drawing attention to a number of vibration problems not covered in the paper and agreed with all he had to say on these matters.

In the course of their investigations the authors had been concerned with a number of instances in which resonant conditions had been avoided by altering the revolutions and by re-distributing ballast. The feasibility of such measures depended however, on the particular type of ship and machinery concerned. To give some idea of the extent to which it was possible to alter a critical frequency by the re-distribution of ballast it might be of interest to recall the case of an ore-carrier with which the authors were concerned. This ship experienced a severe three-node vertical vibration in the ballast condition at 210 engine r.p.m. By re-distributing the ballast so as to give a greater concentration of mass at the anti-nodes it was possible to reduce the critical frequency to correspond to about 190 engine r.p.m. It might be necessary in such cases however to determine whether such a re-distribution was acceptable not only from the point of view of operating requirements but also from strength considerations.

On the subject of acceptable limits Mr. Muirhead would find a fairly adequate discussion in reference 2 of the paper. Briefly the new limits suggested in the paper were based on more recent experience and evidence which indicated that the tolerable level of acceleration rose with frequency within the range of ship problems. The value of 1·0ft./sec. was still roughly applicable for two-node vertical modes of vibration in average sized ships. The acceptable vertical acceleration was taken to be about 2·0ft./sec. at about 600 cycles/min. and the formula would give this value. Horizontal vibrations were known to be more troublesome than vertical vibrations and the acceptable limits had been adjusted by an amount which was based on their experience.

With regard to the boundary layer thickness Mr. Muirhead was referred to Mr. Lackenbys' contribution to the discussions and the authors' reply.

The authors welcomed Mr. Adams' views on the possible reasons for the variations of the coefficient *C* as indicated in Fig. 7. The authors also took the view that some of the variations might be explained on the basis of structural differences but not necessarily as a result of different effective section inertias. Hull frequencies were determined not only by the characteristics in pure bending but also by the degree of shear deflexion which in some ships might form an appreciable proportion of the total deflexion according to the mode of vibration. Differences between estimates of frequency for ordinary types of dry cargo ships and tankers could be partly explained on the basis of shear deflexion. The shear material in tankers was such that in vibration the shear deflexion as a proportion of the total was usually less than in ordinary dry cargo ships and as a result the tendency had been to underestimate the frequencies of tankers. There were however many other considerations which arose in attempting to explain these variations and the authors agreed with Mr. Adams that studies must be pursued in order to clarify the situation.

To have included cross-curves on a base of position of engine centre for a range of examples would have extended the paper beyond tolerable dimensions and only the example in Fig. 42 was therefore included. However, all the data necessary to develop similar cross-curves were given in reference 2 of the paper.

The authors were aware of the ship to which Mr. Adams had referred and were pleased that he had mentioned it as this raised an important point concerning the application of the method of amplitude prediction set out in the paper. It would be seen in Fig. 15 that the lower modes of vibration, which were those excited by engine unbalance had sharply tuned resonances. Consider the four-node vertical resonance in Fig. 15. The acceptable amplitude at the frequency of this resonance, 207 cycles/min. was 0·033in. The maximum amplitude recorded was 0·096in. but the amplitude fell to the acceptable level of 0·033in. at 205·5 and 212 cycles/min. which corresponded to shaft speeds of 51·4 and 53 r.p.m. respectively. This meant that outside the narrow range of 1·6 r.p.m. around about 52 r.p.m. the four-node vertical mode of hull vibration would not be excited to unacceptable amplitudes. In normal acceleration of the ship one would pass through this speed range in less than a minute; there would not be time for the vibration to build up to significant amplitudes and even if it did it would probably not even be noticed in the short duration at such levels. Only if there were prolonged running at 51½ to 52½ r.p.m. would this four-node mode of hull vibration have any practical significance. The same argument applied to the ship to which Mr. Adams had referred. There the four-node critical frequency was at 97·5 r.p.m. on a ship having a service speed of 115 r.p.m. Observers had reported that in fact vibration could be noted when increasing speed through 97·5 r.p.m. but that this was not an objectionable feature. It should be stressed that the restrictive values of tolerable couples derived from the calculations in the paper need only be applied where prolonged running at a critical speed could not be avoided. If in practice a small "barred" speed range of some 2 to 3 r.p.m. could be accepted, very much larger unbalance couples could be tolerated quite happily.

In reply to Mr. Millar it was true that little reliable data had been published on torsional hull vibrations. As a consequence the problem had not received the same attention as the more common vertical and horizontal modes of vibration. Various approximate methods for estimating hull torsionals had been published and some appropriate references were given here.* Whether any of these methods had been developed to the point of reliability for design use was open to debate. The calculation of the torsional rigidity of hulls, particularly those with larger openings presented special problems. Additional difficulties arose due to the distribution of mass about the

* Taylor, J. L. 1927-28. "Ship Vibration Periods". Trans. N.E.C. Inst. Eng. and Shipb., Vol. 44, p. 143.

Horn, F. 1925. "Horizontal—und Torsions—Schiffschwingungen auf Frachtschiffen". Werft Reederei Hofen, 6, p. 577.

Kumai, T. 1955. "Estimation of Natural Frequencies of Torsional Vibration of Ships". Reports of Research Institute for Applied Mechanics, Kyushu University, Vol. 4, No. 13, July.

torsional axis and the influence of entrained masses for torsional oscillation.

Torsional modes of vibration could nevertheless be serious and the authors had had experience of some which were excited at blade order frequencies. A case in point concerned a 420ft. dry cargo ship with a 4-bladed propeller which experienced a severe two-node torsional at 400 cycles/min. (100 shaft r.p.m.). The two-node vertical for this ship occurred at 93 cycles/min. and the three-node vertical at 175 cycles/min.

This torsional mode gave rise to a disturbing degree of vibration in various parts of the ship, particularly in the midships deckhouse. It was considered that the vibration would be much reduced if the aperture clearances had been larger.

The authors had no first-hand knowledge regarding any loss of propulsive efficiency arising from shafting vibrations. Nevertheless, judging from the normal variations of thrust and torque on each blade consequent upon normal wake variations it appeared probable that the additional effect of vibration on propulsive efficiency was likely to be small.

The observations recorded in Fig. 23 were taken some time ago during an *ad hoc* investigation using instrumentation which did not permit of the determination of the relative phasing of readings and there must be some doubt about the relative phases of the readings between frames 11 and 30 on one hand and those between frames 30 and 60 on the other. The frequency of these tank top vibrations was that of the propeller blade passage and the propeller forces were known to be large on this ship. There was probably a considerable element of local propeller-excited vibration of the aft end of the tank top present in these readings. It was not easy to visualize an arrangement whereby the stiffening effect of the shaft stools could be determined experimentally.

The authors would be loth to concur in any generalization concerning the causes of hull vibration although they would repeat the observations in the paper that, generally speaking, engine-excited vibration was concerned with the lower modes of hull vibration, whilst with propeller-excited vibration the frequencies involved were higher and the mode forms more complex; the latter source of excitation was perhaps the more important but this observation was based on the complexity of the matter rather than on an assessment of the frequency with which it occurred. As to the flexibility in the engine framing or seats, there were believed to be instances where the engine forces had been, in effect, magnified considerably due to local vibration of the engine masses and seating structure, though there was little conclusive evidence on this point.

The authors thanked Mr. Taylor for drawing their attention to the anomaly regarding the description of the vibration exciter on page 122 and the results given in Fig. 5. In fact, the machine was originally designed to a maximum rating of $\pm 3 \cdot 0$ tons at 600 r.p.m., but in practice had been used at higher speeds.

With regard to Fig. 7, Mr. Taylor was referred to the earlier description. The authors were intrigued that this plotting should have attracted so much attention and perhaps an additional plot of the type to which Mr. Taylor referred would have helped the reader. Such a plot could, of course, be produced very simply from the data given in Fig. 7.

On the question of presentation of vibration data resulting from unsteady propeller forces, the authors considered that it would be unwise to attempt to lay down hard and fast rules. The general conditions under which the observations were made, the recording equipment available and the methods of analysis used, all had to be considered. Maximum values were, of course, of intrinsic interest, but for purely comparative purposes the averaging processes suggested by Mr. Taylor might well have some merit.

The occurrence of two different modes of vibration of twin oil engine installations had been recognized, one an anti-phase mode and the other an in-phase mode and it was to the former that the term "tuning fork vibration" had been applied. Mr. Taylor rightly pointed out that these two modes normally did not occur at the same frequency. The difference between the two frequencies was usually wide since the double bottom structure played a large part in the vibrating system in both cases and the part it played was very different in the two cases. This frequency difference was normally great enough to prevent both modes occurring within the service range of speed. The in-phase mode had the higher frequency and it was the anti-phase mode at the lower frequency which occasionally obtruded into the running range. Trouble with the in-phase mode was more likely to occur as a forced vibration excited by hull horizontal or torsional resonance as in the case of single engine installations and, as in the single engine case, the staying of the engines to the ship's side frames was then of doubtful value.

Other contributors had commented on the precise speed control of shafts and the authors could not usefully add to their constructive observations.

The authors were grateful for the information given by Mr. Guglielmotti on the successful phasing of the engines on Italian ships. The reduction in hull vibration on the m.s. *Torres,* shown in Fig. 47 was most impressive and, taken with the observations of other contributors, indicated that such arrangements were of greater practical value than had been thought.

The use of equation (10) had been dealt with much more fully in reference 2 of the paper; included in that reference were the limits within which data had fallen and the authors would refer Mr. Guglielmotti to this reference for observations on the degree of approximation which might be achieved in prediction.

The authors concurred with the recommendation regarding counterphasing the excitations of engine and propeller to reduce torsional vibration and suggested that the same procedure might be beneficial also for axial vibration of shafting.

It was doubtful whether the amount of entrained water assumed to take part in axial vibration was really important. In a particular case of axial vibration examined some time ago it was found that an error of 20 per cent in the added virtual mass at the propeller produced an error in frequency estimation of 1·3 per cent while an error of 50 per cent in virtual mass gave rise to a frequency error of 4·2 per cent.

The use of a clutch coupling in tie-bars added to prevent engine transverse vibration was a neat solution; one would, however, have to be very sure that it would not seize up in service before one would be prepared to dispense with a shear pin.

Mr. Guglielmotti's experience of transverse engine vibration caused by propeller forces was in accord with that of the authors and was complementary to their remarks in reply to Mr. Taylor on the use of tie-bars.

Mr. Eckhard had set out a case for the more traditional type of stern arrangement, provided that adequate propeller clearances could be adopted. It was appreciated that the special types of stern arrangement might be more costly and the importance of this aspect in the tendering stage could not be overlooked. The authors, however, were interested only in recording as objectively as possible, the technical facts available at the present time. If pressed on these facts they were bound on balance to draw the tentative conclusion that for high powered ships the data favoured the clear water concentric bulb type of stern. Not only from the point of view of vibration, but also from aspects of propulsive efficiency. The fact that there were some ships in service with traditional sterns and high powers without suffering vibration troubles would not seem to prove the case one way or the other. It seemed to the authors that shipowners themselves were taking an added interest in these developments and would be prepared to meet any extra costs if they could be assured of the technical advantages which might accrue. It might also be mentioned that if the overall dimensions of the closed type of aperture became unduly large without great care given to suitable increases in scantlings, there was a considerable risk of running into local vibrations of the rudder and stern frame assembly.

In defining the conventional type of stern, Mr. Eckhard was referred to the description given at the top of page 128.

Mr. Kleiner's remarks on transverse engine vibrations and the provision of facilities for the fitting of tie-bars accorded very well with the views of the authors, though the omission of shear bolts was not considered advisable.

The observations concerning the factors which required further study in the development of predictions of crankshaft and line shafting axial vibrations were most valuable. This was an important field of study and the more work done on it the better for everyone. The attention being given to the matter by Sulzer which was indicated here by Mr. Kleiner was welcome and together with the work being done by Doxford and Fiat to which reference had been made by Mr. Jackson and Mr. Guglielmotti respectively, should ensure that axial vibration should not be a problem which would hinder the development of very large Diesel engines. Reference had been made in dealing with the points raised by Mr. Guglielmotti to the potential advantage of selecting the phasing of the excitation of propeller and engine and the authors were pleased to have the supporting remarks of Mr. Kleiner exemplified by Fig. 52.

The wide experience of Mr. Kleiner's company in the balancing of engines made his remarks on the practical aspects particularly valuable; there was of course, room for difference concerning the best solutions for these problems. What was most important was that both naval architects and marine engineers should appreciate the nature of the problems and collaborate to overcome them.

The authors agreed with Mr. Burton Davies that the six bladed propeller did not appear to have received the attention which it merited. It seemed that until recently there were doubts concerning the propulsive efficiency which could be obtained with six blades as against smaller numbers and also on the score of strength. These doubts now appeared to have been largely unfounded and it was probable that six-bladed propellers would be adopted on an increasing scale in new ships.

Apologies were due regarding the alternative definitions of the ship length L used in Parts I and II of the paper. In using approximate formulæ of the types given in equations (4) and (5) the authors had adopted L as representing the length between perpendiculars and the subsequent use of a different definition was an oversight. It was fortunate that the alternative use of L for estimating vibration amplitudes was only likely to incur small errors in relation to other approximations which had to be made.

In reply to Mr. Irvin it was not thought that the shafting had a significant influence on the mode of transverse vibration of engines but in the present state of knowledge on this phenomenon, the factors of significance could not be stated with confidence. This lack of knowledge of the significant factors, would also appear to make the use of models of doubtful value at present.

The derivation of the tolerable unbalance couples on page 143 of the paper stemmed directly from equation (10). A couple at the centre of the engine might be replaced by two equal and opposite forces F separated by the length of the engine δx, the ordinates of the deflexion profile at the ends of the engine being α_1 and α_2 referred to unit deflexion at the end of the ship.

Then $\Sigma \overline{F\alpha_F} = F(\alpha_1 - \alpha_2)$

which can be written $\Sigma \overline{F\alpha_F} = \dfrac{F\delta x}{L}\left\{\dfrac{(\alpha_1 - \alpha_2)}{\delta x / L}\right\}$

where L = overall length of ship

Now $\dfrac{(\alpha_1 - \alpha_2)}{\delta x / L} = i_m$ = mean slope of the deflexion profile over the length of the engine δx and we can write

$$\Sigma \overline{F\alpha_F} = \frac{F\delta x}{L} \times i_m$$

$F\delta x$ is equivalent to a couple M

and therefore $\Sigma \overline{F\alpha_F} = Mi_m/L$

which allows equation (10) to be re-written

$$y_x = \frac{35200}{C} \quad \frac{\alpha_x Km}{N^2\Delta_1} \quad Mi_m/L$$

so that for the end of the ship where y_e is the amplitude of vibration and $\alpha_x = 1$,

$$y_e = \frac{35200}{C} \quad \frac{K_m}{N^2\Delta_1} \quad \frac{Mi_m}{L}$$

For vertical vibration the acceptable amplitude at resonant frequency N_c, y_e

$$\not> \frac{1}{N^2} \quad (2N_c + 1000)$$

from which the acceptable couple

$$M \not> \frac{L}{i_m} \quad (2N_c + 1000) \quad (C/35{,}200) \quad (\Delta_1/K_m)$$

Correspondence on "The Design and Development of Two-drum Marine Boilers"

A further contribution to the discussion on the paper by Mr. E. G. Hutchings, B.Sc. (Member), published in the February 1963 issue of the TRANSACTIONS *(Volume 75, page 37), has been received and is given below, together with a reply by the author.*

MR. C. A. MASON (Associate) wrote that Mr. Hutchings was to be congratulated on his interesting presentation. His conclusion No. 3 that "A general advance in steam temperature much above 950 deg. F. with low grade fuels is unlikely unless some economic method is found to prevent attack of the superheater tubes by the corrosive elements of the fuel" prompted comment.

The automatic application of additives through retractable soot blowers had been an important and valuable development. Experimental work started eight years ago and all reports indicated that the final system was giving complete satisfaction on approximately fifty boilers. Although experience had been gained largely on land based installations, there had been marine application as well.

Essentially, the soot blowers were first operated with normal blowing pressure. The blowing pressure was then reduced and the blowers operated again, introducing the additive slurry (magnesium and calcium oxides) with the blowing medium.

"Additive slurry spraying" was used first on a clean boiler and, depending on the fuel and boiler design conditions, frequency of application might be from twice a day to once every two days.

A typical analysis of oil fired in many central power stations and industrial boilers with a steam temperature of 1,050 deg. F. and metal temperatures as high as 1,225 deg. F. was vanadium 500 p.p.m. as V_2O_5, sodium 140 p.p.m. and sulphur 3 per cent. The deposits on superheater and reheater tubes were tenacious and semi-molten up to approximately 1in. thick.

It would appear that in the past it was the general practice in the United States of America to omit even long travel soot blowers in positions on high steam temperature oil fired units, having gas temperatures above 1,500 deg. F. In these circumstances, the boiler availability was approximately three months, with severe corrosion limiting life of superheaters to four years. Following development of the "additive slurry spraying" technique, the boilers were operated for 14 months using slurry once per day. A summary of all the reports was as follows:

A high concentration of additive on the tubes resulted in deposits free from low melting temperature compounds throughout the full thickness, due to migration of additive outwards from the tube surface. The ash particles had little adhesive strength to the tubes and to each other. Much of the deposit fell off due to gas velocity and, consequently, was easily removable by soot blowers.

Reports on a large number of units claimed that there had been no evidence of high temperature corrosion in a five year period. The rate of failure of superheater supports and spacers was considerably reduced. When the supports were repositioned to permit fuller coverage by additive, the life was appreciably extended.

The initial thermal shock on the stainless steel tubes resulting from the magnesium oxide and water slurry was eliminated by control of slurry flow rates, nozzle design, and spray particle size.

SO_3 and dewpoint were appreciably lowered for a period up to four hours after "additive slurry spraying" and the effect was still noticable at 12 hours.

"Additive slurry spraying" soot blowers were installed on the s.s. *E. J. McClanahan* burning fuel containing 1,900 p.p.m. as V_2O_5 and 5·4 sulphur, and maintained clean superheaters.

The economics of "additive slurry spraying" through soot blowers on large central station boilers might be made by comparison between the quantities used and resultant effects. Power Stations had reported that 200lb. MgO + CaO at cost of approximately £3 were used per boiler per day through soot blowers compared with 3,800lb. $Mg(OH)_2$ at £42 per boiler per day injected in burner zone. Furthermore, the latter provided insufficient concentration to completely arrest corrosion, although it decreased deposit formation.

It was believed that many boilers in the United States had been designed for comparatively high ratings to reduce size and cost but with "additive slurry spraying" as a safeguard against corrosion and deposition.

The author's comments on long travel (rack) soot blowers were most apt and as in his view low grade fuels seemed to be the rule rather than the exception, it was a point of some regret that boiler designers omitted, generally, to cater for the application of this essential mechanism.

In view of additive application being refined to the degree where it was possible to operate boilers with high metal temperatures in the corrosive range even with low grade fuels, it would appear that with relatively minor modifications to the author's later boiler designs, higher steam temperatures would be practicable with no fouling or corrosion—even with smaller superheaters exposed to direct radiation.

Mr. Hutchings thanked Mr. Mason for his most interesting contribution on the automatic application of additives through retractable soot blowers. Although, of course, the author had knowledge of this development, he had no direct practical experience and it was extremely interesting to hear of the apparent high degree of success obtained by "slurry spraying". If further experience confirmed this apparent success, then this might well pave the way for higher steam temperatures. It was unfortunate, however, that Mr. Mason had given no indication of how this slurry would be stored and prepared in a ship. This, of course, might not prove a serious problem, but it had to be borne in mind, particularly for vessels spending a considerable time away from their home port.

Annual Dinner

The Sixtieth Annual Dinner of the Institute was held at Grosvenor House, Park Lane, London, W.1, on Friday, 8th March 1963 and was attended by 1,478 members and guests,

The Senior Vice-President, Mr. James Calderwood, M.Sc., was in the Chair.

The official guests included: His Excellency Monsieur Armin Daeniker, The Swiss Ambassador; His Excellency Monsieur Jacques de Thier, The Belgian Ambassador; His Excellency The Honourable Sir Thomas Macdonald, K.C.M.G., The High Commissioner for New Zealand; His Excellency Mr. M. C. Chagla, The High Commissioner for India; His Excellency Senhor Jose Cochrane de Alencar, The Brazilian Ambassador; His Excellency Monsieur Michel Melas, The Greek Ambassador; R. C. S. Koelmeyer, Esq., The Acting High Commissioner for Ceylon; The Right Honourable The Viscount Simon, C.M.G., President, The Royal Institution of Naval Architects, and Past President; Sir Gilmour Jenkins, K.C.B., K.B.E., M.C., Past President; Vice-Admiral M. Le Fanu, C.B., D.S.C., Third Sea Lord and Controller of the Navy; Sir Harry Melville, K.C.B., D.Sc., F.R.S., Secretary, The Department of Scientific and Industrial Research; Vice-Admiral Sir Frank Mason, K.C.B., Chairman of Council; Sir Victor Shepheard, K.C.B., Director of Research, British Ship Research Association; Sir Gordon Sutherland, Sc.D., LL.D., F.R.S., Director, The National Physical Laboratory; Captain Quintilio Rivera, Naval Attaché, representing His Excellency The Chilean Ambassador; Sir Kenneth Pelly, M.C., Chairman, Lloyd's Register of Shipping; Dr. T. W. F. Brown, C.B.E., S.M., Director of Marine Engineering Research, British Ship Research Association; Captain G. E. Barnard, Deputy Master, Trinity House; Captain L. W. L. Argles, C.B.E., D.S.C., R.N., Captain Superintendent, H.M.S. *Worcester*; T. F. Bird, Esq.,

At the reception before the Annual Dinner. From left to right: Mr. R. Cook, M.Sc. (Member of Council), Sir Victor Shepheard, K.C.B., Director of Research, British Ship Research Association, Vice-Admiral M. Le Fanu, C.B., D.S.C., Third Sea Lord and Controller of the Navy, and Mr. J. H. Pitchford, M.A., President of the Institution of Mechanical Engineers

Annual Dinner, 1963

C.B., Under-Secretary (Marine), Ministry of Transport; H. A. J. Silley, Esq., C.B.E., Past President; The Reverend Maurice Dean, B.A., R.N.V.R., The Rector, St. Olave's, Hart Street, London, E.C.3; R. G. Grout, Esq., President, The Chamber of Shipping of the United Kingdom; J. N. S. Ridgers, Esq., Chairman, The Corporation of Lloyd's; Captain J. D. Elvish, C.B.E., Master, The Honourable Company of Master Mariners; C. W. Warwick, Esq., Chairman, The Baltic Exchange; C. C. Pounder, Esq., Past President; R. B. Shepheard, Esq., C.B.E., B.Sc., Director, The Shipbuilding Conference; A. W. Wood, Esq., Assistant Secretary, Ministry of Transport; Stewart Hogg, Esq., O.B.E., Chairman, Social Events Committee; Captain T. W. Murphy, U.S.N., United States Assistant Naval Attaché; J. H. Pitchford, Esq., M.A., President, The Institution of Mechanical Engineers; Captain J. D. Mody, I.N.; Captain S. F. Mercer, R.N.Z.N.; D. S. Tennant, Esq., C.B.E., General Secretary, The Merchant Navy and Airline Officers Association; Iain M. Stewart, Esq., B.Sc., President, The Institution of Engineers and Shipbuilders in Scotland; G. R. H. Towers, Esq., J.P., President, The North East Coast Institution of Engineers and Shipbuilders; Commander E. H. W. Platt, M.B.E., R.N., Denny Gold Medallist 1962; G. Strachan, Esq., B.Sc., Denny Gold Medallist 1962; S. E. Tomkins, Esq., O.B.E., Secretary, The Salvage Association; J. S. Tritton, Esq., President, The Diesel Engineers and Users Association; N. E. Thompson, Esq., President, The Society of Consulting Marine Engineers and Ship Surveyors; C. M. Brain, Esq., President, The Institute of Refrigeration; K. H. Platt, Esq., M.B.E., B.Sc., Secretary, The Institution of Mechanical Engineers; R. W. Reynolds-Davies, Esq., O.B.E., B.Sc., Secretary, The Institute of Fuel; A. Dearden, Esq., Institute Silver Medalist 1962; Commander R. F. A. Whately, M.A., R.N., Secretary, The North East Coast Institution of Engineers and Shipbuilders; J. D. C. Stone, Esq., F.C.A.; Victor Wilkins, Esq., F.R.I.B.A.; Ronald Ward, Esq., F.R.I.B.A.

The Loyal Toasts, proposed by the CHAIRMAN, having been honoured, HIS EXCELLENCY MONSIEUR JACQUES DE THIER (The Belgian Ambassador) proposed the toast of "The Royal and Merchant Navies of the British Commonwealth".

He said: I am indeed honoured to have been invited to your Annual Dinner and asked to propose the toast of "The Royal and Merchant Navies of the British Commonwealth". I should have been more than flattered and pleased just to propose a toast to the Royal Navy or the Merchant Navy or to the Navies of the Commonwealth, but to have to toast the three together seems to me a formidable task.

Being an old diplomat I had the natural reaction of trying to find some help and encouragement by looking up the precedents. I was kindly given an issue of your TRANSACTIONS which contains the texts of the speeches made at the last Annual Dinner, and I discovered that this toast was proposed last year by my distinguished colleague, the Ambassador of Switzerland. (*Applause.*) I hope he will not mind if I say that his precedent gave me no encouragement at all! (*Laughter.*) His speech was delightful and witty and set such a high standard that it seems very hard for his successor at this table to match his eloquence.

However, having given some thought to the matter it soon appeared to me that for a toast few subjects could be more appealing, more fascinating, than the Royal and Merchant Navies of the British Commonwealth. To reflect upon them, as we follow our daily routine under the dark skies, is like feeling the fresh air from the high seas, the sunshine of the tropics, and the excitement of adventure, hardship, power and glory. To stimulate such reflections I know of no place more interesting, more inspiring, than the National Maritime Museum at Greenwich.

While the Royal Navy, with its highly technical developments today, opens up wide horizons to scientific minds, the Merchant Marine offers great opportunities to those who want to make fortunes—and are not afraid of losing them! (*Laughter.*) Together they have filled illustrious pages of history and provide numerous examples of determination, courage and sacrifice.

Speaking about the Royal and Merchant Navies it would be natural to extol their glory and to remember that Britain is an island, but these days national pride and even the insular character of this country have become very delicate subjects, and I hope you will excuse me if I do not dwell upon them!

I am sure however, that nobody will object if I say that without the Royal and Merchant Navies there would have been no Empire, there would be no Commonwealth. Without the Royal Navy the world would not have enjoyed safety and peace on the high seas for many years and we should not have won the two world wars. Without the Merchant Navy we should not have had the benefit of so many exchanges of goods and reached our high level of economic prosperity. (*Applause.*)

Although Belgium possesses one of the most important ports in the world, Antwerp, for years she had no Navy. Why should we have tried to compete with the Royal Navy, who were doing an excellent job guarding the Channel, the North Sea, the Atlantic, with ships much more powerful than any we could have afforded? Practical as they usually are, the Belgians were less interested in prestige than in not wasting money. (*Laughter.*) The problem of disarmament might be easier to solve if more people throughout the world could come to share this same view. (*Applause.*) It certainly never occurred to us that we would have needed protection against British men-of-war. Britain is the only one of our neighbours with whom we have never been at war, and the only British soldiers who ever crossed our borders came as liberators. (*Applause.*)

Since the Second World War, however, we have a Navy—a modest one, no doubt, but one which has the distinction of having been born in Britain. Our sailors still bear the stamp of this origin. Like their British companions, they wear the black bow of mourning for Admiral Lord Nelson. As we gave you the name of one of our small villages, Waterloo, for one London station, perhaps we may consider that the courtesy has been returned. (*Laughter and applause.*)

After Hitler's invasion in 1940 a number of small Belgian boats of various types succeeded in reaching British harbours and took part in naval operations, being assembled in 1941 to form in the British Navy a new Belgian section. Those vessels flew the White Ensign next to the Belgian flag. Later this small fleet, with several minesweepers, took part in the Atlantic convoys and other naval operations. Belgian officers serving in these ships had been trained by the Royal Navy. After the war the Belgian section of the Royal Navy became the Belgian Naval Force; made up mainly of minesweepers, they contribute today under the NATO Command with ships of British and other allied countries to the protection of the Channel. The Royal Navy, however, continued to give us valuable assistance in the training of our officers and petty officers. Although young and small, our Navy enjoys already a high prestige in Belgium, and one of its officers is Prince Albert, the brother of King Baudouin. (*Applause.*)

Some Belgian towns are proud to have their names included in the annals of the Royal Navy—for instance, Zeebrugge, where during the First World War they succeeded in scuttling some of their ships in order to block the entrance to the harbour, thus preventing the enemy submarines from sailing, and Antwerp, which at the end of the Second World War became a base from which the British and other Allied Forces on the Continent could be supplied. Being the only harbour which they could use at that time, Antwerp played an essential part in the success of their final offensive.

As my toast is also addressed to the Navies of the Commonwealth I think it would be appropriate for me to mention that two of the minesweepers of the Belgian Navy were given to us by the Government of Canada. (*Applause.*) They were given to us in accordance with the NATO Mutual Assistance pacts. I had the good fortune to be Ambassador to Canada when one of them was handed over to my Government at Esquimalt in British Columbia, and it was a delightful and impressive occasion. The Minister of National Defence of Canada took his Belgian colleague, myself, and a few other persons in his 'plane from Ottawa to Vancouver and from there to the lovely residence of the Admiral Commanding the Base as Esquimalt. The gardens slope down to the sea and one can admire one of the most beautiful views of the Bay of Vancouver and the Rocky Mountains. The ceremony of handing over the minesweeper took place in beautiful sunshine and we could not help being moved when we saw the White Ensign slowly being lowered while the Belgian flag was hoisted to the top of the mast. It was a splendid display of friendly assistance between NATO countries, and this visit to Vancouver remains one of the most delightful memories of my stay in Canada.

In conclusion, may I say that I am very thankful for your generous hospitality. It has been a great pleasure for me to attend this Dinner and to give this toast. It is also a great privilege to have this opportunity to pay tribute and express my gratitude on behalf of Belgium to the Royal and Merchant Navies of the British Commonwealth. (*Applause.*)

Vice-Admiral M. LE FANU, C.B., D.S.C. (Third Sea Lord and Controller of the Navy) responded and proposed the toast of "The Institute of Marine Engineers".

He said: When I said where I was going they told me, "You had better be good—they're a rough old lot." (*Laughter.*) Well, a high standard was set by His Excellency the Belgian Ambassador, and I am very honoured to be appearing in the same programme.

This country has many reasons to be grateful to the Belgian people, not least because of the good humoured toleration they show us when we gallop through and around their country on our summer holidays. Not only are they very kind and very charming but also they are very clever and very industrious, and they know quite a lot about marine engineering (*Hear, hear*), and particularly about nuclear reactors for same (*Applause.*) Obviously I must not say another word! (*Laughter.*) Particularly a word like Vulcain. If I do I will shortly be run in by the First Lord or Richard Dimbleby. (*Laughter.*)

Thank you, Your Excellency, for everything you have said about the Royal Navy and the Merchant Navies of the Commonwealth—most of whom seem to be here this evening. (*Laughter*). I am uncertain of my qualification for speaking on behalf of this mob. The Chairman of Council fixed me with his beady eye, and, as you probably know, when that happens, brother, you stay fixed! (*Laughter.*) Even to the extent of having to propose a toast as well as to reply to one. I think my union is going to object to this practice. (*Laughter.*)

As a member of the Board of Admiralty—who, by the way, should not be confused with the Temperance Seven . . . (*Laughter and applause*) . . . I do have a union. It is known as BOAMAS—Board of Admiralty Mutual Admiration Society. (*Laughter.*) "BOAMAS Washes Whiter·" (*Laughter.*) I am working this joke absolutely as hard as I can on account of I read in the paper that before there wasn't going to be a Board of Admiralty· (*Laughter.*)

When I joined this union and accepted the job of Third Sea Lord I didn't quite realize what was entailed in the way of after-dinner speeches. When I say "accepted" it wasn't quite like that. The First Sea Lord said to me, "We're due for a new Controller. Get fell in, Ginger!" (*Laughter.*) "You've been loafing around Hong Kong a lot too long. Very dodgy· Don't answer back—and what do you mean 'punishment draft' ?" (*Laughter.*) What he didn't tell me was that this draft carried with it the secondary qualification of after-dinner speaker, so now I'm an A.D.S. (Third Class). (*Laughter.*) This all happened in the early summer of last year, and there was not time to take a course in A.D.S. because everything in the Navy stops then and we all do what is called "showing the flag", or else push rather old guns round Earls Court. (*Laughter.*) However, we got the show on the road in the Autumn under the title "Let's Twist with the Oldies", and had a very good series of one-night stands in the Provinces. "Are Sea Lords Human?" and "Artificial Pearls Before Real Swine" were two of the numbers. (*Laughter.*) This season we have achieved West End booking, and this is it, men! (*Laughter.*)

During the so-called Christmas day-off in Whitehall, when we were all under stoppage of leave or required for duty on board—that is a very subtle joke, actually—the peasants in the outer office put together a new libretto, and this season we have "This Is Your Life" or "Any Old Iron" . . . (*Laugh*ter) . . . the title "An Evening of British Rubbish" having already been pre-emptied. (*Laughter.*) So "Let's Twist again"!

I am told that if I had taken an A.D.S. course the instructors would have taught me that any proper speech should have a noble thought or stirring message running through it like a golden thread. First you have your introduction, during which a slightly humorous note may be struck, and then you come to your golden thread which is dead serious, mate. I am sure that such an intellectual shower as we have on the top table will appreciate what I am saying (*Laughter.*) You now have a firm grasp of the thread, Fred. (*Laughter.*) Here it is.

I must admit to a life long interest in the Royal and Merchant Navies of the British Commonwealth, and you can't be in any sort of Navy for long without running slap bang into marine engineers—and everybody knows that all collisions are entirely due to marine engineers! (*Laughter and applause*).

"When in danger or in fear
Always blame the engineer."

(*Laughter.*)

I think I can say without fear of contradiction—and, somehow, the more senior I get the less I fear contradiction . . . (Laughter) . . .—that the wellbeing of the Commonwealth and of the Royal Navies still depends on Healthy Merchant Navies and shipbuilding industries. That lot may sound like Rule 1 Line 1 stuff to you but in the cold world outside no one takes anything for granted any more. We have to keep on proving it, and one way of proving it is to show the world we are ahead of it: progressive, efficient, economical—the lot. To do this we have to help each other take advantage of each other's know-how and experience, and in this the Institute of Marine Engineers plays a very important role. Long may it prosper. (*Applause.*) If I or any of my colleagues in the Admiralty are falling down on their job somebody here this evening will let us know all about it. Deep calling to deep! The Admiralty's simple organizational structure . . . (*Laughter*) . . . is well known and is not dissimilar to yours, with your Chairman and Vice-Chairman, your 23 Councillors and 9 co-opted Councillors, your 47 Vice-Presidents of one sort or another, all your various committees, and the vast output of papers, written for you almost entirely by Naval Officers during the hours when they ought to be working for me. (*Laughter and applause.*)

There is no prospect of my joining you, you will be relieved to hear. I have studied the rules of your examinations and, candidly, men, I don't measure up. For example, it says that it is expected that candidates will have had experience in the design and use of internal combustion engines, and therefore (the "therefore" is good) be familiar with the construction and operation of much of the apparatus mentioned in the syllabus. That is good, but it is not me. I have only one of the qualifications for Associate Membership: I have attained the age of 26 years. (*Laughter.*)

There is one thing about marine engineering. On the 16th December last, three months ago, I was quietly contemplating that, despite the many difficulties we have in the hardware world, the Navy, my Navy, your Navy, was in pretty good nick. The *Ashanti* and the *Devonshire* have had some troubles with their gas turbines but they are coming good on the whole. *Dreadnought* has had her first run to sea, the carrier design is coming along very well. We were, I thought, slowly getting to grips with the problems of the Fleet's reliability and availability, and, I thought, whatever new problems come our way, and they whistle past my ear'ole every hour on the hour, we have a pretty good Board of Admiralty to deal with them. BOAMAS! (*Laughter.*) Just when I was indulging in these smug reflections the telephone rang and a voice said, "Le Fanu? Ministry of Defence here. The Minister wants you to go to the Bahamas tomorrow. Happy Christmas." What I said back was, " Me no savvy, master. Me Lee Fan U, me happy go Bahamas long Chinese New Year. Not Kismis. My missy no likee." (*Laughter.*) You can say that again! Well, I went to the Bahamas with the Minister, and he came back with Polaris.

You men ought to know that Polaris is the toughest job our Navy has ever tackled in peace on account of the Polaris project consists of scarfing a highly complex and sophisticated American weapon system into a highly sophisticated and complex nuclear submarine design—not yet proved at sea. Furthermore, the resultant submarine/weapon system has the unique feature of having to stay at sea for very long periods, 100 per cent reliable and 100 per cent available, and this programme has got to be run through on the toughest possible sched. We are going to require from our own people, in plain clothes and in uniform, and from our contractors, extraordinary efforts of mind and will if we are to honour, as we must honour, the Navy's pledge and the Government's pledge. This is a big, big deal, and the biggest part of the deal—in which the marine engineers are going to help us—is on the reliability front. We have been edging forward, but we have got a long way to go, and in a few years my successor—I hope it is my successor and not me—will have to put his hand on his heart and say to his colleagues, to the Minister of Defence and the Prime Minister and others, that these ships are going to go to sea for all those days and do anything that is required of them at a moment's notice, and that there will be no temperamental pumps playing up and no double-acting valves on the blink. We have never had to do this before. We have to set new standards of design and workmanship, and I have no doubt that you lot will solve them with time to spare. We have a very tough problem. We are going to enjoy it enormously and enjoy using every bit of brains and endurance we have, and every bit of the press-on spirit that we have got.

My last word is one message that my masters give me from time to time. It is a message of three words, deeply moving. It is a message from which they derive hope and encouragement. "Belt up, Mike." (*Laughter and applause.*) So I will belt up, and for the benefit of anybody who missed any part of this programme, there will be a repeat next week at the Annual Dinner of the Admiralty Electrical Engineers. (*Laughter.*)

Thank you for your kind hospitality and for your kind attention. Thank you, Your Excellency, for your speech. And now I will give you the toast of "The Institute of Marine Engineers'.' (*Prolonged applause.*)

The CHAIRMAN (The Senior Vice-President, James Calderwood, Esq., M.Sc.), in response, said:

I am afraid that after the two speeches to which you have just listened I must come as a terrible anti-climax. In the first place, I am sure that most of you hoped that our President would be here this evening to answer the toast. As you know, he is on national business on the far side of the world.

There is one respect in which I think I can follow the other two speakers, and that is brevity. They have both followed the rules closely on that side. Before the war, at a meeting I think of the petroleum industry's club, a very famous speaker of those days gave some advice to speakers on these occasions. He said, "If after five minutes you don't strike oil, stop boring." (*Laughter.*)

I cannot guarantee to sit down after five minutes but I will be as short as I can. The reply to this toast is expected to be a review of the year's working of the Institute and I asked our Secretary for some headings. I have them all written down here, five of his and one of my own.

During the year the Institute has had in mind the expansion of an old activity. Some of you may remember that our Past President, Lord Simon (whom I am very glad to see here tonight), mentioned at our Dinner that we hoped that some of the funds surplus from the building would be allocated to start an education programme. That education programme had been proposed by a well-known and very esteemed member of Council. Unfortunately, it proved rather too great for any of the funds available, but your Council have considered it and set up a special committee, and we do hope that we shall be able to introduce something that will help some of the younger engineers to go to higher education than they would otherwise have been able to achieve. I cannot say anything more at the moment. I might say, however, that the progress of this Committee was very much encouraged by the Presidential Address last autumn. There is one other thing arising from it, and that is that when we talk about education I hope it will be education and not what I call "inducation", that is to say, pumping in a lot of specialized knowledge. Education involves broadening a person and not narrowing him in grade specialization.

A short time ago one of our Members of Council suggested that we should have a series of lectures and talks on some of these new words that are coming into the language—cybernetics, ergonomics, and so on; I cannot remember the others for the moment. Curiously enough, I thought I knew what cybernetics meant, but last week, in an article in the "New Scientist", there was a reference to Cybernetics in Ladies' Fashions. I could not see any connexion with ladies' fashions. I thought it was the science of communication. I still cannot understand it. Anyhow, it is just possible that we may have some lectures to try and teach some of us older people—not the youngsters—the meaning of some of this new language coming in nowadays. Frankly, I thought I understood some of it, but I am beginning to think I do not know a thing about it.

I must now refer to the local section activities. Admiral Mason and I have during this winter visited between us, one or other, every local section, and we have found great enthusiasm wherever we went. In addition, I can mention that three more local sections have been formed during the last few months: St. John's, Newfoundland; Perth, Australia; also South-East England. (*Applause.*)

I think the Engineering Institutions Joint Council ought to be mentioned. The Institute has joined up with this organization during the year. It is something that most people would agree has been wanted for a long time—closer association between all the engineering societies. It has got off to a remarkably good start and I hope it will be very successful. (*Applause.*)

The highlight of the season was the International Conference last May. I have been to several international conferences of one type and another, and the one we held was different from all the others in one respect, and a most important one. At ours the nations mixed up together. At some conferences, outside of the technical meetings, on the social occasions, the French get together and the Germans and the others get together in their own separate groups, but at our Conference they mixed, and I have had many letters of appreciation from delegates, from Japan, Belgium, Germany, Sweden, New Zealand, Canada, all expressing appreciation of the way in which the people from various parts of the world got together. Most of the Commonwealth countries and a large number of European countries were represented, and it was a conference at which delegates mixed freely not only on the technical side but also on the social occasions. (*Applause.*)

I think I should say at this point how very grateful we were to the former Lord Mayor, Sir Frederick Hoare, Bt., for all his help, at the opening of the Conference, at the official luncheon, and so on. These functions helped greatly towards the getting together. (*Applause.*)

Finally, I should like to refer to the staff of this Institute. I wonder how many of you realize how much extra work for the staff is involved in a gathering like this, in particular for the Secretary and for Mrs. Sullivan. (*Applause.*) Apart from this special occasion, they have all, without exception, from the Secretary and Mr. Franklin right down to the juniors, given those of us on the Council the utmost help whenever they could, and I thank them and we all thank them very sincerely for their work throughout the year, and for the real energy and interest they put into the Institute. The Institute depends on the staff for its enthusiasm and we have a most enthusiastic staff, I can assure you.

There is no toast to the guests tonight but I must welcome Their Excellencies from the Commonwealth and other nations, the Presidents and Secretaries of many of our associate institutions, and also all the guests of all our members. Thank you. (*Applause.*)

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting held at The Memorial Building on Tuesday, 8th January 1963

An Ordinary Meeting was held by the Institute on Tuesday, 8th January 1963 at 5.30 p.m., when a paper entitled "Machinery Induced Vibrations" by Dr. A. J. Johnson, B.Ss., A.C.G.I. and W. McClimont, B.Sc. (Member) was presented by the authors and discussed. Vice-Admiral Sir Frank Mason, K.C.B. (Chairman of Council) was in the Chair and sixty-two members and guests were present.

Eight speakers took part in the discussion which followed.

The Chairman proposed a vote of thanks to the authors which was greeted by prolonged acclamation.

The meeting ended at 8.0 p.m.

Section Meetings

Northern Ireland Panel

A senior meeting was held on Friday, 15th March 1963, at the College of Technology, Belfast, at 7.0 p.m.

Chairman of the Panel, Mr. D. H. Alexander, O.B.E., Wh.Sc. (Local Vice-President) presided at the meeting which was attended by one hundred and ten members and visitors.

The speaker at the meeting was Mr. R. Munton, B.Sc. (Member of Council), who presented a paper entitled "Automation" . Many contributed to the lively discussion which followed the lecture.

Mr. Campbell Brown, Director of Sirocco Engineering Works, proposed a vote of thanks to the speaker, which was seconded by Mr. C. C. Pounder (Past President of the Institute).

The meeting closed at 9.30 p.m.

Scottish

A joint meeting with the Aberdeen Mechanical Society was held at Robert Gordon's College, Aberdeen, on Friday, 22nd March 1963, at 7.30 p.m.

The meeting was presided over by Mr. W. L. Symon, President of the Aberdeen Mechanical Society, and Mr. R. Beattie (Chairman of the Section) and Mr. A. W. Clark attended on behalf of the Scottish Section.

Mr. Symon introduced the speaker, Mr. John Wright, of Hall, Russell and Co. Ltd., who then read his paper entitled "Recent Developments in Trawler Design" which was illustrated by slides.

This paper caused great interest as was evidenced by the lengthy discussion that followed.

Mr. Beattie aptly proposed a vote of thanks to Mr. Wright, which was carried with loud applause.

The attendance was seventy-five.

West of England

General Meeting

A general meeting of the Section was held on Monday, 11th March 1963, in the Small Engineering Lecture Theatre, Queen's Building, University of Bristol, at 7.30 p.m. Captain R. G. Raper, R.N. (Chairman of the Section) presided at the meeting and among those present was Mr. D. W. Gelling (Local Vice-President, Bristol).

A paper entitled "The Application of Planned Maintenance to Steam Turbine Tankers" by J. Scott, D.S.C. (Member) and H. Vickerstaff (Member), was presented by the authors and proved to be a most interesting and instructive paper which stimulated much discussion.

The paper was in the main a study of the loss of earning power due to various reasons, of two classes of ocean-going oil tankers and the planned maintenance necessary to minimize these financial losses. The authors gave a well informed and lucid account of their ideas in this connexion and a discussion was started in which twelve members of the audience took part.

A point was made by one contributor that in view of this highly efficient method of self-maintenance, dockyards would no longer be required except for dealing with major repairs and the handling of the heavier equipment and machinery. The authors did not feel that this was wholly true, although the idea was to cut down time spent in dockyards by doing this survey and maintenance work afloat as far as practicable, in a progressive four-yearly cycle, so helping to keep the vessels at sea for longer periods.

A vote of thanks to Mr. Scott and Mr. Vickerstaff, was proposed by the Chairman and the meeting ended at approximately 9.30 p.m.

Junior Meeting

A junior meeting was held on Wednesday, 3rd April 1963 at the City of Bath Technical College New Lecture Theatre, at 7.30 p.m., when a paper entitled "Launching of Ships" by R. S. Hogg (Member) was presented.

Captain R. G. Raper, R.N. (Chairman of the Section) presided at the meeting which was attended by both junior and senior members.

The lecture was in the main, intended for the junior members, but proved of immense interest to everyone present. Mr. Hogg dealt with all aspects of the launching of vessels starting with the older methods, one point of interest being that launching by building in dry docks will eventually replace the slipway, especially for larger vessels.

It is worth noting here, that the lecture theatre in which the paper was read, has just been completed and must be considered one of the finest in the West of England.

The Chairman, after giving a vote of thanks to Mr. Hogg for the very interesting lecture he had given and for the very efficient way in which he had dealt with the questions that followed, went on to say how indebted he was to the Principal of the Bath Technical College, for allowing the Section to use the lecture theatre which up to this time had hardly been used.

The meeting ended at 9.15 p.m.

Election of Members

Elected on the 8th April 1963

MEMBERS

James William Anchant
William James Baxter
Howard C. Blanding
Ronald Richard Bolton
Ernest Edwin Bustard
William Floyd Callender
Andrew Anderson Crawford

John Stirling Dickson Douglas
James Ivan Dow
Ernest Charles Henry Featherstone, Cdr., R.N.
Leonard Ernest Fishburn
Axel Gunnar Hellstrom
James Elmer Jackson
John Kirkwood
David Marshall
John J. O'Loughlin
Kenneth Pennington
Pietro Rizzi
James A. Stasek
Thomas Stead, Lt. Cdr., R.N.
Marshall E. Turnbaugh

ASSOCIATE MEMBERS

Robert K. Allan
Peter Arnold
Peter Edward Bailey
David John Bennett
Roger Brown
Alban Michael Bruhns
John Terence Carr
Charles Andrew Clyde
David George Craig
Bartholomew Donovan
Derek Easman
Sydney Fewlass
Gordon Henry Hamilton
Robert Charles Finch Hill, B.Sc.(Eng.) (London), Lieut., R.N.
John Edward Howey
James Ringland MacDonald
Angus McKinnon
Ronald Henry Parsons
Robert Arthur Platt
Franf Douglas Randall
John Clifford Sealey
Alan Thewlis
Timothy Edward Thornycroft, M.A.
David George Wixon, B.Sc., Lieut., R.N.
Kai Yi Woo
David Charles Woodward, B.Sc.(Eng.) (London)

ASSOCIATES

Robert Nicholson Adair
James Anderson
John Paterson Brydon
Henry Augustus Chung
Joseph Lister Cochrane
William Sidney Adams Ham
Jean Lebas
Ian Douglas Lomas
Gordon Maxwell Pinkard

GRADUATES

Vijai Kumar Arora, Sub. Lt., I.N.
John Kenneth Bryant
Leonard George Dalmon
John Gordon Ingram
John Kidd
Jag Bhushan Malhotra, Sub. Lt., I.N.
Gordon Robson
Geoffrey Maundrell Stephens
Roy Semple Steven
Norman Liddell Tulip
John Hugh Williams

STUDENTS

Michael Kingston Bowers
Gene Lionel Clements
Raymond Barry Coombs
Lee Frederick Curtis
Leslie Douglas
Alexander William Fisher
Keith Graham Gibb
Joseph Hyde
David Anthony Langridge
Peter Malcolm Lawford
Anthony David McConnell
Robert Anthony Robinson
Alexander Gordon Thomson
Jacques Therrien
Robert Tinkler
Kenneth Charles Williamson

PROBATIONER STUDENTS

Michael Robin Philip Davies
Michael Paulusz
Anthony Walter Wooff

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER

Henry James Ellis
Kenneth Norman Lane
Alexander Tolmie Mitchell
Charles Shepherd
Alan Christopher Wyndham Wilson, Captain, R.N.

TRANSFERRED FROM ASSOCIATE TO MEMBER

John Whittle

TRANSFERRED FROM ASSOCIATE TO ASSOCIATE MEMBER

Derek Raymond Perkins
Dennis John Capel

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER

Michael Alport
Donald Lloyd Backler
Peter Burrows
Robert Alexander Davie
Malcolm Evison
Gordon Fisher
Karl Jacobson G. A. Fraser
John Peverley Lamb
Setlur Ranganna Parthasarathy
Patrick Andrew Sparrow
David Graham Walters, B.Sc.(Eng.), London

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER

George Keith Sumner
Gerry Vagliano, B.Sc.

TRANSFERRED FROM STUDENT TO GRADUATE

David Rodney Patterson, B.Sc.

TRANSFERRED FROM PROBATIONER STUDENT TO ASSOCIATE MEMBER

Derek Charles May

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE

John Christopher George Halliday
Malcolm Robert Marjoribanks Kean
David John Roderick

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT

Joseph Butler
Christopher Stephen Hesketh
David Robert Ireland
Thomas Henry Lynchy
Peter Francis Robinson
Geoffrey Malcolm Watson

OBITUARY

SIR JOHN RITCHIE RICHMOND, K.B.E., LL.D., J.P. (Member 1890) died on 25th February 1963, at his home in Kirkoswald, Ayrshire, aged 94 years.

A half-brother of the first Viscount Weir, with whom he became closely associated in the development of G. and J. Weir Ltd., the Glasgow engineering company, Sir John was born in Kilmarnock. He was educated at the High School of Glasgow, of which he later became a governor, and at Glasgow University, of which he became an honourary Doctor of Laws.

With acknowledgements to The Scottish Field

He began his association with Weir's as an apprentice in 1889 and, when he retired as senior deputy chairman in 1954, had been a director of the company for 59 years. An efficiency expert, he impressed his ideas on both the works department and commercial organization of the firm during a period of steady expansion. He played an important part in organizing G. and J. Weir's vast munitions output during the First World War, a service which was recognized by the award of the C.B.E. in 1918. Further recognition of his many professional and public services came, in the Birthday Honours List of 1939, when he received a knighthood.

During his career, Sir John had served on the advisory committee of the Board of Trade and on various Government Committees concerned with housing and the preservation of natural amenities. In 1935 he was chairman of the Scottish Architectural Advisory Committee and for many years was chairman of the Pollok Unionist Association.

Sir John was elected a Member of the Institute on 6th September 1906 and served for several years as a Vice-President. He was connected with a wide range of technical societies and other organizations. In 1903 and again in 1917, he was president of the North-West Engineering Employers' Association and was vice-chairman of the management committee of the Engineering Employers' Federation in London, for several years.

A discriminating art collector, he had travelled extensively in Europe and America and his private collection of pictures included a good selection of modern works by British and foreign painters. He was a trustee of the National Gallery of Scotland, chairman of the Royal Glasgow Institute of Fine Arts, chairman and later honorary president of the Glasgow School of Art and a senior member of the Glasgow Art Club. As patron of the old Glasgow Repertory Theatre he revealed his interest in drama.

Twice married, Sir John was predeceased by both his wives. There is no family.

MAJOR JOHN HENRY CULLINANE (Associate 7462) died on 1st February 1963, after a long illness. He had been an Associate of this Institute since 8th January 1934 and was also a Member of the Institute of Petroleum Technologists.

Born in Lancashire on 13th August 1889, he received his technical training as an apprentice with Messrs. Mills and Co. of Radcliffe. This apprenticeship was the prelude to a long career in the oil production industry, much of which was spent in the Middle East. From 1914 to 1919 he was in charge of Anglo-Persian Oil Refineries in Abadan. In the latter year he transferred to Egypt, where he became Installation Manager for the Shell Petroleum Oil Co. and was responsible for the company's installations in the Suez Canal area.

During the Second World War he rendered very valuable service to the Royal Navy and the Royal Air Force and was untiring in his exertions to maintain the impetus of the war effort.

After the war, he continued in his employment with Shell in Egypt, until 1946, when he retired. Returning to the United Kingdom with his wife, Mary, who survives him, he went first to Jersey, in the Channel Islands, then for a time to Ireland and finally settled in his native Lancashire, where he was residing at the time of his death.

WILLIAM ARNILL ROXBURGH DOUGLAS (Member 4589) was born on 21st December 1880. On completion of his apprenticeship, served from 1896 to 1901 with the Glasgow and South Western Railway at Kilmarnock, he went to sea as a junior engineer, first with the Red Star Line and then with the Shire Line. In later years, after obtaining his First Class Board of Trade Certificate, he served as chief Engineer with the Auchan Line, the Lanarkshire Steam Ship Co. Ltd. and the Clyde Steam Ship Co. Ltd.

He was elected a Member of the Institute on 31st October 1921, at which time he was superintendent engineer to the Illawarra and S.C. Co. in Sydney, New South Wales. In 1939, he became a consulting engineer, with offices in Sydney.

Mr. Douglas, who died on 4th February 1963, leaves a widow.