Design Aspects of Marine Propulsion Shafting Systems

I. K. MOTT, B.Mech.E. (Member)* and R. FLEETING, B.Sc., A.F.I.M.A.†

Transmission problems arising during the last two decades have led to a growing awareness of the troublesome features of single and multi-screw shaft systems. The basic problem is to design a system to transmit torque from the main engine output flange to the propeller and to transfer thrust generated at the propeller to the main thrust block. Additional requirements are that the system should not develop harmful torsional, axial or whirling vibrations yet retain sufficient transverse flexibility to accept the most extreme hull movements without overloading any one bearing. The paper examines each of these requirements in detail and correlates them, using examples mostly drawn from Pametrada experience, but including the experience of others where necessary.

A survey of the shafting systems of merchant and naval vessels ranging from fast, small twin-screw ferries to a large multi-screw aircraft carrier and passenger liners is included. Several complete vibration design studies are given and rule-of-thumb guidance to accelerate preliminary shaft design is derived. Service experience is given in the discussion of several recent case histories and two Algol computer programmes to determine the axial and/or torsional characteristics of a turbine geared-shaft system are described in an Appendix.

INTRODUCTION

Transmission problems encountered over the last two decades have highlighted some of the more troublesome features of the propulsion systems of single and multi-screw ships. For single-screw ships, these have included cracking of tailshafts, particularly at the propeller keyway. In extreme cases, complete fracture and subsequent loss of a propeller eventuated, but present day survey methods can detect cracking at a much earlier stage and tailshaft replacement is then required. Rapid weardown of wood, rubber and plastics-lined stern tubes is also prevalent in this group and severe patterned wear of the bronze shaft liners due to a cavitation-erosion mechanism is also common. Torsional vibration problems can arise but, even on aft-end installation single-screw ships, these are not normally serious when engine excitation is eliminated as with geared turbine propulsion. Tunnel-bearing failures and main-engine damage have arisen from permanent hull deformation incurred during grounding and is a hazard to be borne in mind when single-screw ships use massive trim aft to improve propeller immersion when travelling in ballast.

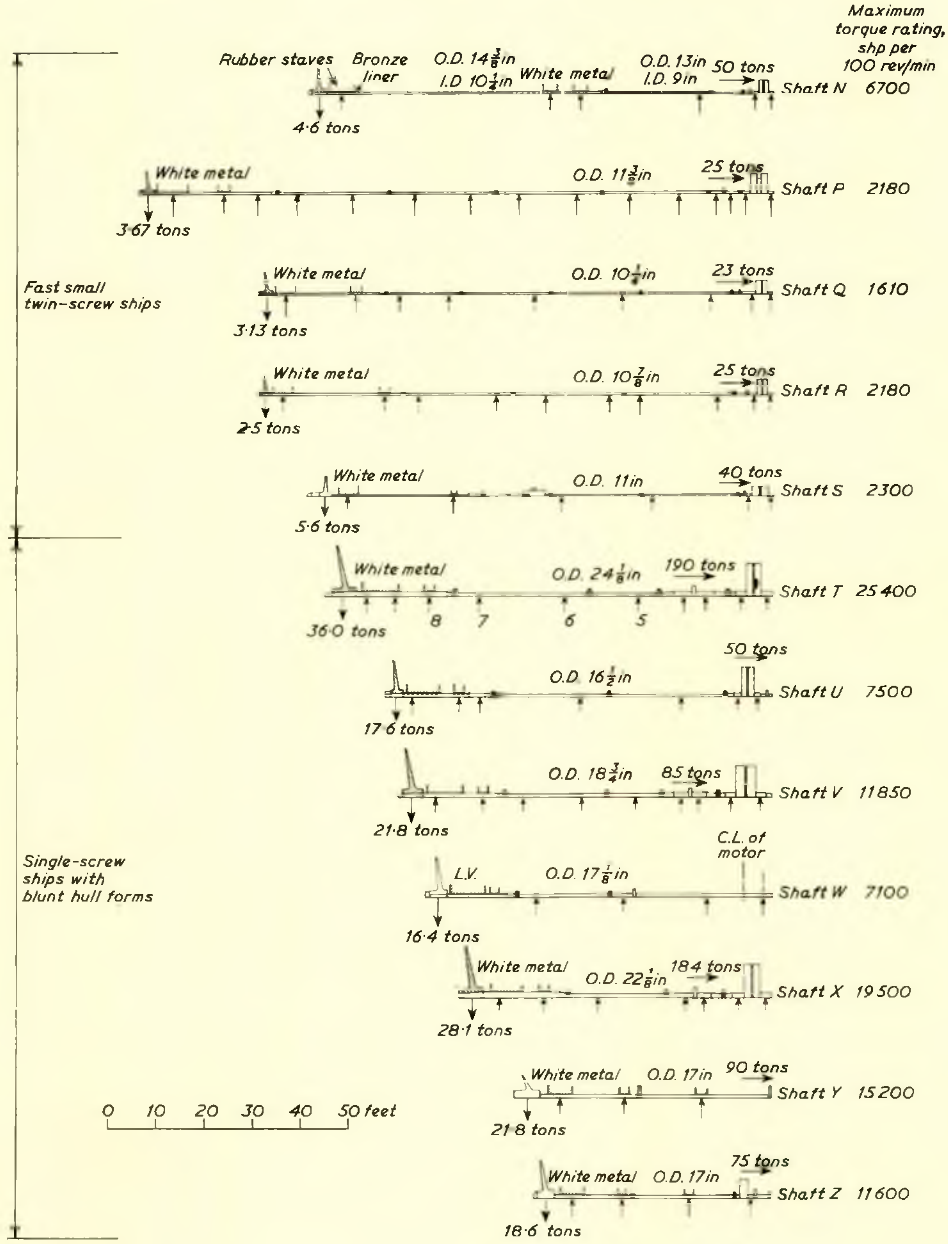
Multi-screw ships encounter the foregoing problems to a significantly lesser degree, but also suffer from troubles more peculiar to themselves, including bent propeller blades and tailshafts as a consequence of wing-propellers striking a dock-wall or pack ice. Larger shafting lengths are generally required and, combined with higher propeller speeds, give rise to resonant axial vibrations in the main shafting within the operating speed range. When bossings are used to support the outboard shafting, cracks in the bossing frames and plating occur with surprising frequency.

Although the main shafting of a ship is the vital component transmitting power from main engine to propeller, most of its obvious features are imposed by factors not necessarily concerning the propulsion system. For example, location of the engine room is largely a function of hull trim for fast cargo and passenger vessels and of hull bending moments for bulk carriers and oil tankers. Consequently, the detailed design of the shaft system tends to occur fairly late in the development of a new ship, by which time, other factors, including the number of screws, type of main engine, type of propeller (solid, controllable-pitch, built-up, etc.) and shp-rev/min have been fixed. Some of the variables and non-variables (i.e., those parts of the system fixed by considerations other than design of the shaft system) and their influence, if any, on the bending and vibration of the shafting are listed in Table I. The extent to which these variables and non-variables affect the bending and vibration characteristics of the shaft is referred to later in the paper.

Shafting performance is also influenced in various ways by hull form; a fine form giving poor accessibility to the last few inboard bearings, a full form accentuating the wake effect which reduces the mean approach velocity and, therefore, the propeller pitch, but which increases the propeller weight and the excitation arising at the propeller.

The basic problem then is to design a system for transmitting torque, extending from the main engine output flange to the propeller and also capable of carrying propeller-generated thrust to the main thrust block. Further additional requirements are that the system should not develop harmful torsional or axial vibrations between the two ends, should be adequately supported so that bearings will not become overloaded nor will whirling occur at any likely service condition, whilst at the same time retaining sufficient transverse flexibility to accept the most extreme hull movements. The solutions to these requirements often conflict and a degree of compromise is necessary. This present paper has been compiled with the purpose of examining each of these requirements in detail and then correlating them using, where possible, examples drawn from Pametrada experience gained in the last 22 years, and from the experience of others where necessary. From such experience, good rule-of-thumb guidance can be drawn so as to accelerate the preliminary ship layout whilst reducing the

*Senior Projects Engineer, Pametrada.
†Senior Lecturer in Mathematics, Rutherford College of Technology.



L.V. = Lignum Vitae

FIG. 2—*Comparison of shafting arrangements for merchant and naval vessels*

wharf cranes, this pointing to equal numbers working at forward and aft holds and therefore placing the main engines at about midship, or position 4.

Channel steamers (shafts Q, R, S) have their machinery spaces located amidships mainly for stability considerations. Because of draught limitations on most of the harbours used, small high speed propellers are also required. Consequently, small diameter shafting is used to handle the low torque and, in turn, causes this class of ship to have a very slender shaft system, even though the absolute length is not great. Shaft S represents an interesting example in propulsion of channel steamers whereby a medium-speed Diesel is geared to a controllable-pitch propeller. The marked increase in propeller weight is evident when compared with shafts Q and R and its effect on the vibrational characteristics of the shafting system requires careful consideration as will be discussed later.

Dry-cargo ships of the older type (shaft A) invariably had the machinery spaces located amidships, resulting in long shafting systems. In recent years there has developed a trend to locate machinery further aft, at about positions 3 or 2, and some installations have been placed in the extreme aft position to leave the wider hull sections for cargo. The problem then is to minimize the weight of the whole propulsion and auxiliary machinery installations so that acceptable hull bending moments are attained. Such a location does, of course, contribute to weight reduction by eliminating a considerable weight of shafting and the attendant tunnel bearings.

Oil tankers and other bulk carriers are essentially aft-end installations (shafts T, U, V, W, X, Y, Z) although a surprising length of shafting is necessary in the case of fleet tankers (shaft B) where the need to replenish naval vessels at speed dictates a fine hull form causing the engine room to extend much further forward in order to give the necessary buoyancy space in this area. This vessel is therefore more typical of cargo liners than of tankers. The merchant tanker and bulk carrier are distinctive for their very short shafting

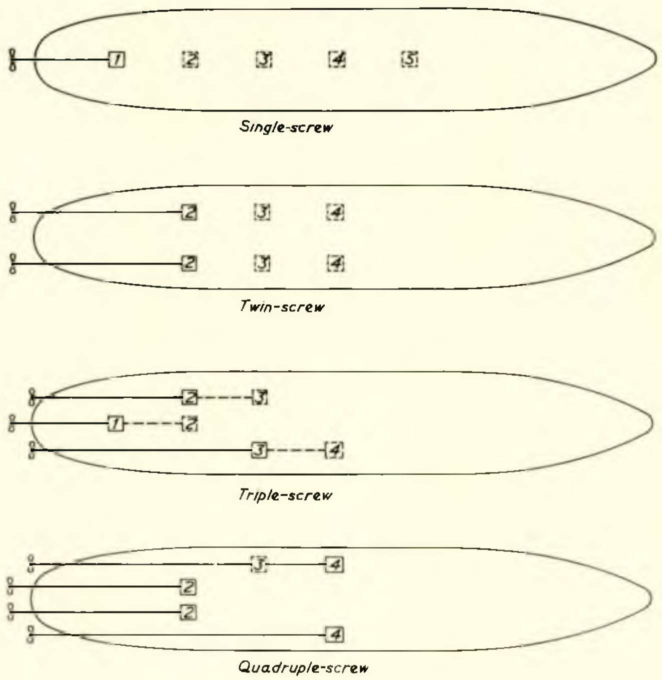


FIG. 3—*Location of main engines*

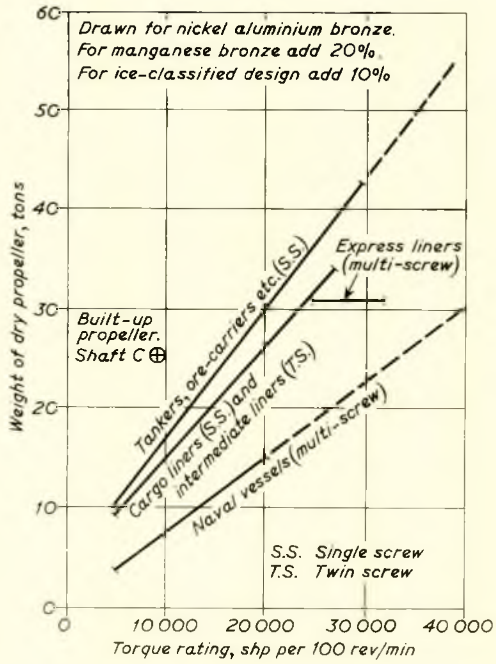


FIG. 5—*Dry weight of propellers*

systems, which in many instances comprises only a short tailshaft, one length of intermediate shafting, and a thrust shaft. This arrangement has been more generally used in Diesel-powered ships where the great length of the main engine must be compensated for by placing it as far aft as possible, and is also evident in T.2 tanker installations (shaft W). Recently this arrangement has been widely adopted in turbine-powered tankers.

Intermediate shafting diameter is used as the basis for fixing the diameter of all other shaft lengths in the system and it is, logically, based upon torque rating which is conveniently expressed in shp per 100 rev/min. In this regard it is compatible with the method of rating the gearbox and also the propeller, which in reality is simply a form of torque dynamometer subject to cavitation and wake influences as special hazards. The merchant ships referred to in Figs. 1 and 2 all have shafting sizes determined in accordance with Lloyd's Register of Shipping Rules where $d = C \sqrt[3]{shp/N}$, but they were designed over a period of 30 years in which time the constant C has been reduced when trouble-free experience has justified this action. Fig. 4 shows the relationships over this period.

Naval shafting, naturally, is designed to higher stress levels consistent with both the low utilization of maximum power

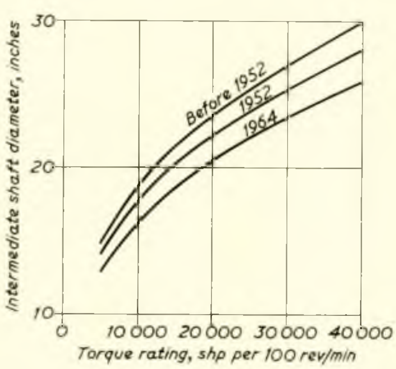


FIG. 4—*Changes in intermediate shaft diameter*

and the use of high-tensile materials. In addition advantage is taken of hollow shafting to reduce weight which, in turn, confers further advantages in requiring a larger bearing, having greater load-carrying capacity and permitting fewer bearings to be used while the whirling characteristics still retain an overall improvement. Hollow shafting has found little application in merchant work because of the substantial increase in cost, and its use in ship F is exceptional. While its use for express liners is attractive, in that weight reduction of one ton reduces the annual fuel cost by about £8, proposals for recent liners have been rejected on the score that transverse stability demands that this weight be restored as bottom ballast so that no overall advantage is gained. In the case of ship E the use of hollow shafting would have reduced the total shaft weight by 100 tons.

High tensile shafting does, itself, offer reductions in weight, but here again extra cost is incurred if alloy steel is specified as in naval practice. Shafts Y and Z utilize 40 tons/in^2 plain carbon steel giving a significant reduction in weight at no extra cost in the forging rate, so providing an overall reduction in cost also. A further gain is that torsional and transverse flexibility are increased substantially for these short-shaft installations.

Other aspects of the shafting systems as shown in Figs. 1 and 2 will be referred to later.

2. PROPELLER AS SOURCE OF EXCITATION

The propeller is one of the main sources of vibration associated with marine engine-shaft systems. To eliminate propeller excitation would require one of two conditions:

1) radially uniform flow past the propeller, or
2) a propeller with an infinite number of blades.

Since neither of these conditions obtains in practice the designer has to accept periodic variations in torque, thrust and bending moment in the shaft system.

Many examples of operational difficulties arising from propeller-induced vibration of the main shafting have been described in the literature. Two classic cases of axial vibration have been included in Fig. 1 (shafts C and L). Shaft C was described by Johnson and McClimont[14] and concerns a single-screw dry-cargo ship which was originally characterized by a particularly heavy propeller (built-up design),

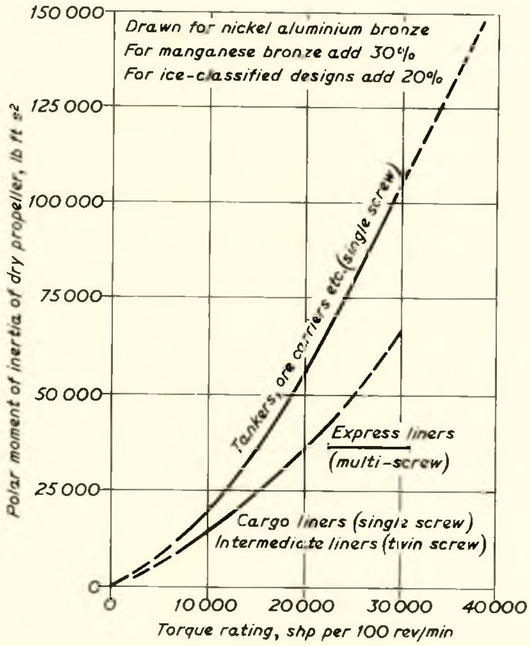


FIG. 6—*Polar moment of inertia for dry propellers*

long shafting system, very flexible main thrust block and seating, and very small aperture clearances around the propeller blades. Shaft L is the centre shaft of a triple-screw aircraft carrier and the investigation in this instance was fully presented by Rigby[24] and Goodwin[11].

Where geared-turbine propulsion is used, propeller-induced torsional vibrations are seldom troublesome provided the system is analysed early in the design stages. Extensive investigations conducted in the early post-war years (for example, Yates[26]) established the levels of excitation and damping occurring in propulsion systems then current. Since then, propeller clearances have been greatly increased to minimize excitation and, in recent years, adoption of the various "clear-water" sterns has also been beneficial in this regard. At this time, only short-shaft systems are likely to present torsional difficulties, but with the current tendency towards shorter engine rooms for bigger tankers, there is still clearly a need for vigilance in this particular aspect. An example of a recent investigation (shaft T, Fig. 2) is included in a later section of this paper.

Periodic variations in bending moment at the tailshaft outer end also received intensive investigation in the early post-war period, mainly because of the high frequency of rejection of tailshafts of Liberty and Victory ships and T.2 tankers which in extreme cases also led to loss of the propeller (Archer[28] and Gatewood[29]). In general, these failures were due to a combination of cyclic stresses arising from transverse bending influences and those due to torque fluctuations originating at the propeller or, in the case of the Liberty ships, from the steam-reciprocating main engine.

The importance of the propeller as a vibration exciter is clearly established and is further evident by the increase, over the last ten years, in the number of investigators working on experimental and theoretical research in this field. The problem has arisen largely from the increasing power of single-screw ships and the reduction in hull scantlings. This has led in general to the fitting of propellers with increasing numbers of blades and several six and seven-bladed propellers are now in service.

The *modus operandi* of a propeller as a vibration exciter is lucidly described by Archer[3] Johnson and McClimont[14] and others. Methods of calculating the propeller wake forces from a knowledge of the wake pattern together with results obtained from model experiments have also been published in particular by Brehme[6], Van Manen and Kamps[15] and Kumai[18].

It is not the purpose of this paper to discuss the propeller in detail, but, since it is a main cause of shaft vibrations and the choice of the number of propeller blades influences directly the vibration behaviour of the shaft system, it is necessary to discuss briefly the main effects of the propeller and the factors which influence the choice of the number of propeller blades. Other aspects of the propeller which influence the vibration of the shaft system, such as weight and polar moment of inertia, are referred to later.

Vibrations are excited by the propeller in two ways, namely by pressure forces and wake forces, the former can excite vibrations of the hull, rudder and stern frame and, in the case of multiple-screw ships, also of the bossings, the latter can excite both hull and shafting vibrations.

The pressure forces arise from the pressure field associated with each propeller blade acting on the adjacent surfaces of the hull and appendages. The frequency of these pressure forces is propeller blade frequency, i.e., the number of propeller blades multiplied by the shaft revolutions and is of the same order as the natural frequencies of the hull when considered as a free-free beam. As yet, theoretical studies have not produced practical predictions of the magnitude of these pressure forces, but both theory and model tests indicate that the intensity of the pressure fluctuations decreases as the number of propeller blades increases.

The wake forces are the variations of axial and tangential forces on each propeller blade produced during each revolution as the propeller rotates in a non-uniform flow. The variation in axial force on each blade produces a periodic axial force and bending moment in the shafting, whilst the variation in tangential force on each blade gives rise to a periodic torque and lateral force in the shafting. These forces and moments

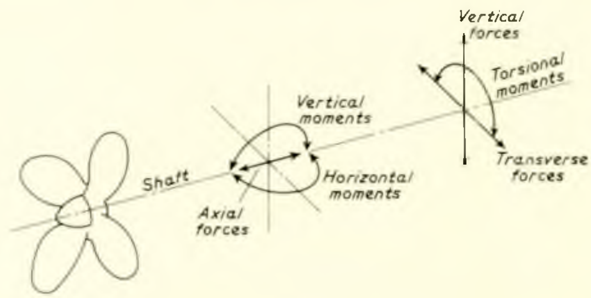


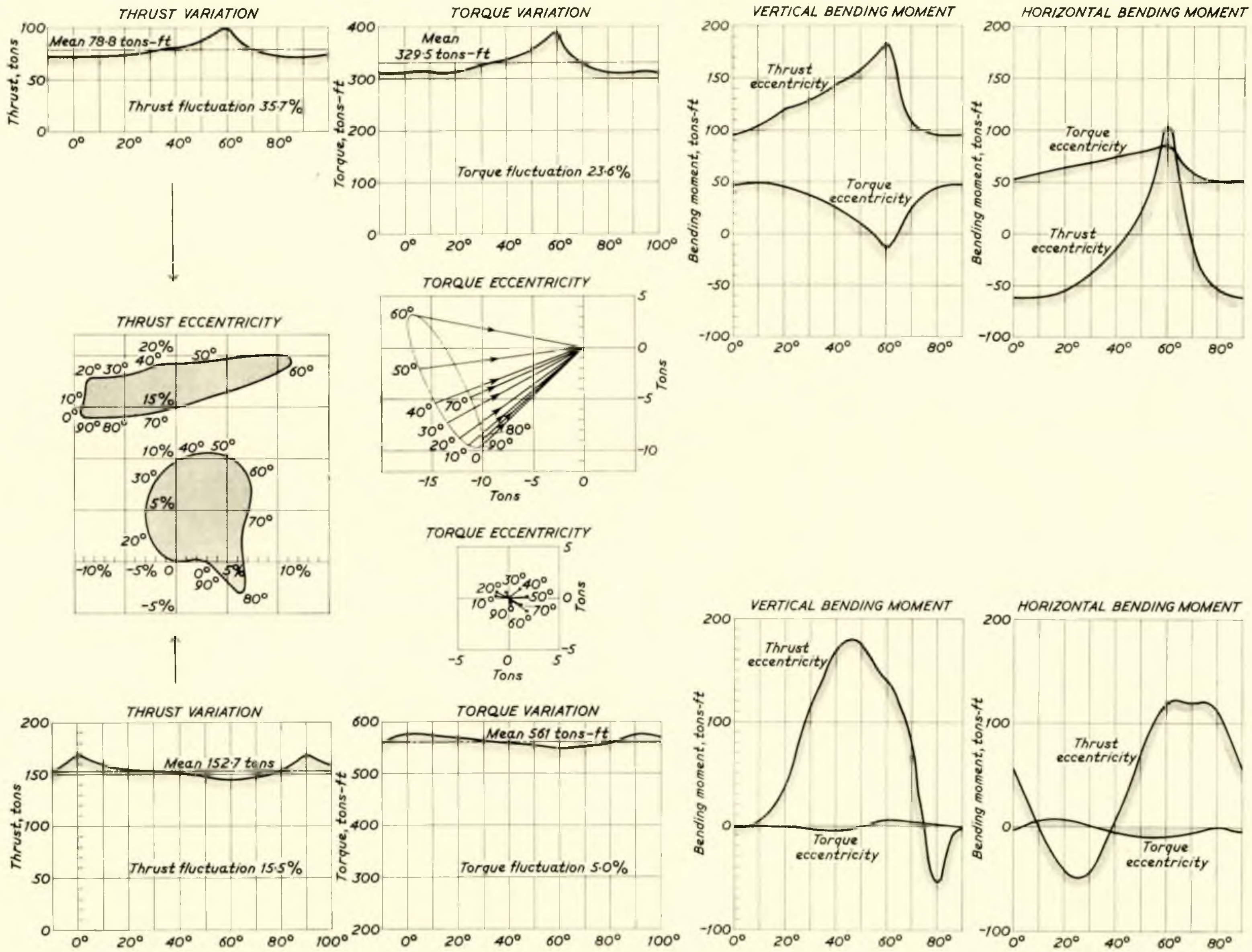
FIG. 7—*Forces and moments due to variations in thrust and torque at the propeller*

are shown diagrammatically in Fig. 7. The principal frequency of these wake forces and moments is again propeller blade frequency, but the harmonics are also present and, for propellers with an odd number of blades, the first even order, i.e., twice blade frequency component can be of magnitude similar to the fundamental.

Fig. 8 shows the results of the variable wake analysis carried out for a twin-screw fast cargo liner (port shaft only) fitted with bossings and a fast single-screw tanker (shaft B, Fig. 1) having a fully-framed aperture, both ships having four-bladed propellers fitted. The diagrams show by columns the thrust variation, torque variation and vertical and horizontal bending moments with the top row of diagrams relating to the twin-screw vessel and the bottom row to the single-screw tanker. The thrust eccentricity and resultant torque vector diagrams for both ships are also included.

For calculation purposes each blade has been replaced by a radial lifting line or vortex, so that neither the influence of skewback nor the averaging effect of a finite chordal blade width has been taken into account. Also, wake surveys have been used which are representative of the respective hull forms.

Cyclic variations of thrust and torque occur at blade frequency, i.e., at four times propeller frequency in this instance,



Top row twin-screw cargo liner (bossings)
Bottom row single-screw tanker (fully framed aperture)

FIG. 8—*Variable wake analysis*

and only one complete cycle has been shown in Fig. 8. The datum for angular position in both cases was the upper vertical axis of the propeller position. Fluctuations are in the form of a complex wave form which can be separated into a fundamental wave form with harmonics. For the present four-bladed propellers the second harmonic (i.e., twice blade frequency) is small in comparison to the fundamental, but this would not be true if the number of propeller blades was odd (see Table II [b]).

Fluctuations in total thrust as calculated are higher for the twin-screw ship and, of greater importance in predicting behaviour of the tailshaft in its bearings, the thrust eccentricity is also very much greater than for the single-screw ship. Comparison of the locii of thrust vector positions shows that there is little change in the vertical position for the twin-screw vessel. Thrust eccentricity, therefore, tends to lift the tailshaft out of the aft stern-tube bearing bottom half. There is a reversal of sign in the horizontal component of thrust eccentricity suggesting that the tailshaft will oscillate horizontally at blade frequency. Both tendencies are greatly reduced with the single-screw ship. Bending moments have been calculated using the corresponding instantaneous thrust and eccentricity values, and have been plotted as the horizontal and vertical components.

Total torque fluctuations are also calculated to be higher for the twin-screw example and, furthermore, give very much greater eccentricity effects than for the single-screw ship. Torque eccentricity arises because of an unequal distribution of torque absorption between the four blades, giving rise to a transverse force acting in conjunction with the torque couple. The magnitude and direction of this transverse force is plotted as torque eccentricity, and this, in turn, causes a bending moment to be applied to the tailshaft since the force is in the propeller plane and there will be a moment arm present when any other transverse section of the tailshaft is considered. For present purposes, the effective point of support in the aft stern-tube bush has been used and the axial displacement or moment arm has been assumed to be 60in for both examples. Using the instantaneous values of torque eccentricity, the horizontal and vertical components of bending moment have then been calculated and plotted.

On comparing the bending moments arising from these effects, it is clear that thrust and torque eccentricity are of equal importance for the twin-screw ship whereas for the single-screw ship, torque eccentricity is a negligible influence. Furthermore, the order of bending moments is comparable for both ships, whereas for the twin-screw ship, the torque rating is only 15 000 shp per 100 rev/min, so requiring a small tail-shaft diameter under present classification rules; the lighter propeller (17·3 tons) helps from the point of view of shaft stresses, but also produces a smaller static loading in the stern tube on which to superimpose these dynamic effects.

The value of the calculations made to produce the results shown in Fig. 8 is very much dependent upon the actual wake distribution occurring across the propeller disc and, disappointingly, the number of distributions actually measured during model testing is still too small for any particular class of hull form to enable confident predictions to be made for a new design. It is not suggested that the results shown in Fig. 8 are typical, nor have they as yet been confirmed by measurements on the actual ships. They have been included to illustrate the forces and moments arising at the propeller and their effect, and also to show that, given a wake profile, the propeller designer is now able to derive much useful data relevant to the performance of the main propulsion system.

Theoretical and model-test values for the variation of the thrust and torque fluctuations (expressed as percentages of the mean value) with the number of propeller blades are given in Table II. The figures given in Table II(a) are taken from reference (6) and are values calculated from an assumed wake distribution. Table II(b) is taken from reference (30) and gives figures based on model-test results. More detailed figures covering the effect of hull form, type of propeller, stern arrangement, draught, etc. are given in references (30) and (14).

In general, for propellers with up to six blades, the fluctuations in thrust and torque are greater for an even number of blades than for an odd number of blades, but in both cases the fluctuations decrease as the number of blades increases. The lateral forces and moments also decrease in magnitude as the number of blades increases, but with a tendency towards lower values for propellers with an even number of blades.

Thus the choice of propeller indicated by the foregoing considerations is one with the maximum number of blades, odd if axial or torsional effects are to be minimized, even if bending moments are important. There are, however, other factors which have to be taken into account.

The design of propellers with up to six blades raises no serious problems. Propellers designed for the same ship and having the same diameter and total developed area would have the same thickness and hence the same steady stress. Also experience shows that there is less cavitation risk as the number of blades is increased. In general, however, there is a reduction in efficiency for geometrically similar propellers as the number of blades is increased. This drop in efficiency is small however, and where the propeller diameter is restricted to less than the optimum, increasing the number of blades may even improve the efficiency[21]. Finally, the cost of a propeller increases with the number of blades, but this again is small, the increase in cost up to six blades being approximately two per cent per blade.

The major consideration, therefore in determining the number of propeller blades is vibration, either of the hull and appendages, or of the shafting and machinery. Apart from the effect on the magnitude of the exciting forces, the number of propeller blades also determines the main-shaft speed at which resonance occurs. Thus resonance with a five-bladed propeller will be excited at 80 per cent of the main-shaft speed at which it is excited by a four-bladed propeller. By this means, it is sometimes possible to shift a resonance out of the speed range or to move it down the speed range to a speed at which it is no longer dangerous.

TABLE II—PROPELLER THRUST AND TORQUE FLUCTUATIONS

(a) *Calculated Values for Single-screw Ship of Average Fineness*

	Number of propeller blades					
	Three		Four		Five	
Frequency	$3n$	$6n$	$4n$	$8n$	$5n$	$10n$
Thrust variation, per cent	6·4	7·3	11·3	3·7	0·6	2·0
Torque variation, per cent	4·2	4·3	7·6	2·6	0·9	1·5

(b) *Measured Values on Model of 32 000-dwt Single-screw Tanker with Conventional and "Mariner" Stern Arrangements*

	Number of propeller blades					
	Four		Five		Six	
Frequency	$4n$	$8n$	$5n$	$10n$	$6n$	$12n$
Thrust variation, per cent:						
Conventional stern	13·0	3·8	2·4	2·8	9·0	2·6
"Mariner" stern	9·7	2·7	3·1	2·0		
Torque variation, per cent						
Conventional stern	7·5	2·1	1·1	1·4	4·3	0·8
"Mariner" stern	5·3	1·1	2·0	0·7		

3. DESIGN ASPECTS OF AXIAL AND TORSIONAL VIBRATIONS

Calculation of Resonant Frequencies

Several well established methods[16, 22] exist for determining the axial and torsional frequencies of a geared-shaft system.

The method used by the authors is the conventional electrical analogy method in which current is taken to be the analogue of velocity. The electrical analogues of mass (M) or inertia (J), flexibility (F) and damping (Q) are then respectively inductance (L), capacitance (C) and resistance (R). This method was described in detail by Yates[26]. The frequencies obtained using the electrical analogy method are the same as those obtained using the more widely known Holzer method, but, by adopting the analogy approach, it was possible to use one general network computer programme to determine the axial or torsional resonant frequencies of a mechanical system. The computer programme is given in Appendix I. The real advantage of the electrical analogy method, however, arises when vibration amplitudes and thrusts or torques are required and damping is introduced into the system. This advantage is discussed by Yates[26] and will be referred to later.

The main difficulties in calculating the resonant frequencies of a shaft system, however, are not in the numerical solution of the problem, but in assessing the principal elements of the vibrating system, particularly in the case of axial vibrations. The torsional system clearly consists of all rotating parts and is well defined, but when axial vibrations are considered, the system also includes non-rotating parts of the machinery and their foundations, including the hull structure itself. Thus, the most difficult assessment in the axial system is the stiffness between the thrust collar and "earth", i.e., that part of the hull structure which is stationary relative to the vibrating system. This problem has been tackled in two stages. A series of tests were first carried out by Pametrada in conjunction with the British Ship Research Association to determine the stiffness (K_b) of a range of standard thrust blocks. B.S.R.A. then determined the combined thrust block and seating stiffness (K_{bs}) for a number of ships by substituting the measured axial frequency into the Holzer table and, hence, determined the thrust block seating stiffness (K_s) from the formula:

$$\frac{1}{K_{bs}} = \frac{1}{K_b} + \frac{1}{K_s}.$$

From the thrust block seating stiffnesses determined in this way B.S.R.A. has been able to derive empirical formulæ for estimating the seating stiffness at the design stage.

The allowance for water entrained at the propeller is another part of the vibrating system which is not clearly defined. The allowance is conveniently expressed as a percentage of the weight or inertia of the actual propeller, but this takes no account of the blade shape, propeller material or size of the boss. Thus for two propellers of different materials, but which are otherwise identical, the entrained water allowance is different. The discrepancy is even greater when considering controllable-pitch propellers for which the weight is approximately twice that of a conventional propeller of the same developed area, the additional weight being mainly in the hub and, hence, not adding to the entrainment.

From measurements of axial vibrations of multi-screw naval vessels including shaft L Fig. 1 Rigby[24] deduced an allowance for entrained water of 0·0481 tons/ft^2 of developed blade surface area. More elaborate formulæ based on the propeller geometry have been proposed by Kane[16] and Lewis[4] for both the axial entrained mass and entrained inertia with the following relationship between the two:

Inertia entrainment = 0·02 × entrained mass × (pitch)^2.

This relationship was confirmed by Burrill[7] from measurements made on model propellers, but with a mean value of 0·024 for the constant.

For most preliminary design calculations, however, for shafts fitted with conventional propellers, allowances of 25-30 per cent of propeller inertia for inertia entrainment and 50-60 per cent of propeller weight for entrained weight are sufficiently accurate.

The behaviour of the flexible couplings in service introduces yet another uncertainty into the calculation of axial frequencies. For ships with the thrust block integral with the gearcase, relatively large masses become involved and the effectiveness of the flexible coupling is less important. With a separate thrust block down the tunnel, however, the frequency of the system is more sensitive to the effective mass of the gears, etc. and, hence, to the sliding or sticking of the flexible couplings. Couchman[9] in his measurements on five multi-screw passenger liners including shafts E, F and H see Fig. 1, obtained the best agreement between calculated stiffness values from the Holzer tables and those derived from the load/thrust block deflexion measurements by assuming the couplings functioned as intended and isolated the turbines and primary gears from the system.

Calculation of Vibration Amplitude Thrust or Torque

When it is not possible to design the axial or torsional criticals out of the service speed range, then it becomes necessary to estimate the vibration amplitudes and thrust or torque variations in the system. A second computer programme which will do this is also included in Appendix I.

Two further uncertainties, however, enter into the calculation, the magnitude of the exciting force and the damping present in the system.

The factors which effect the magnitude of the exciting force, when the source is the propeller, have already been discussed. Since the response of the vibrating system is directly

TABLE III—AXIAL VIBRATIONS SHAFT E

Variation of thrust block amplitude at resonance with damping factor, q

	Weight, tons	Damping factor, q							
Propeller	45	10	10	20	10	10	20	20	20
Shafting	197	10	10	10	10	20	20	20	20
Thrust block	16	10	20	10	10	10	10	20	20
Gearing	79	10	10	10	20	10	10	10	20
Weight ratio		0	0·05	0·13	0·23	0·58	0·72	0·77	1·0
Response ratio		1·0	1·01	1·14	1·06	1·45	1·77	1·79	2·0

$$\text{Weight ratio} = \frac{\text{Weight of system for which } q = 20}{\text{Total weight of system}}$$

$$\text{Response ratio} = \frac{\text{Amplitude at resonance}}{\text{Amplitude at resonance for } q = 10}$$

proportional to the exciting force, it is usual to assume the largest expected thrust or torque variation at the propeller in the design calculation.

There are many sources of damping in a shaft system, e.g. propeller, journal bearings, flexible couplings, turbine blades, etc.

Damping at the propeller has been discussed by many authors including Lewis (Seward 1944) and Archer[2] and can be estimated with sufficient accuracy for practical purposes. Other sources of damping in the system, however, are more difficult to estimate and the method used by the authors is to associate with each mass or inertia in the system a damping factor, q, corresponding to viscous damping. In the equivalent electrical network this is done by associating a resistance, R, given by

$$R = \frac{pL}{q}$$

where p is the circular frequency, with each inductance, L, in the circuit. This is the same method as proposed by Yates[26] and was chosen by him because q defined in this way represents, in many cases, the ratio of the amplitude at resonance to the amplitude away from resonance when a constant exciting force is assumed.

If the same damping or q value is used for each mass or inertia in the system, then the response at resonance is directly proportional to the damping factor q. This can be seen in Fig. 9 which shows the axial resonance curves for shaft E with assumed damping factors of 10 and 20.

Changing q for individual parts of the system, however, does not produce a simple change in response proportional to the change in q. This can be seen in Table III which shows the change in response at resonance for various changes in q, expressed as a percentage of the response at resonance when q = 10 throughout the system. It can be seen from the table that the damping assumed for the propeller or propeller shaft has proportionately more effect on the response at resonance than the values assumed for the thrust block mass or gearing.

Even after the exciting force and damping have been estimated and the vibration amplitude and vibratory thrust or torque calculated, there is still the problem of acceptability.

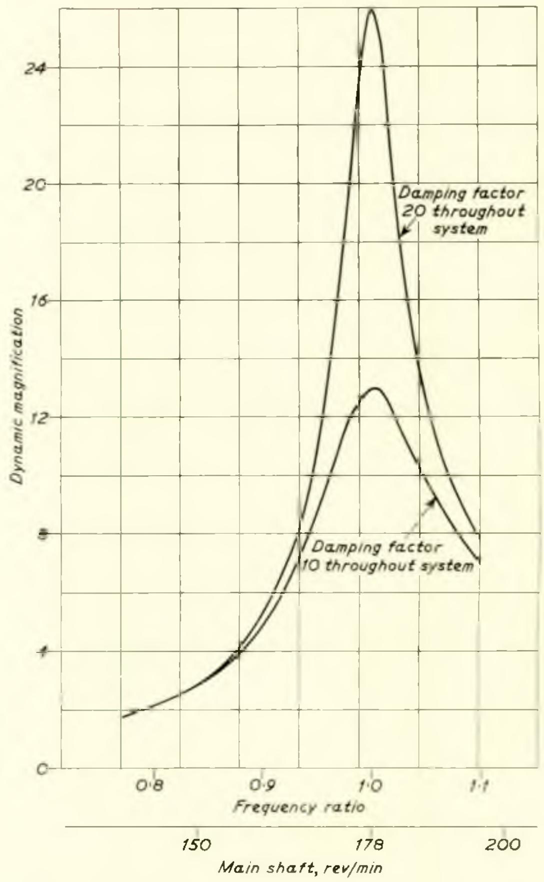


FIG. 9—*Axial resonance curves shaft E—Showing effect of change of damping factor on response curves*

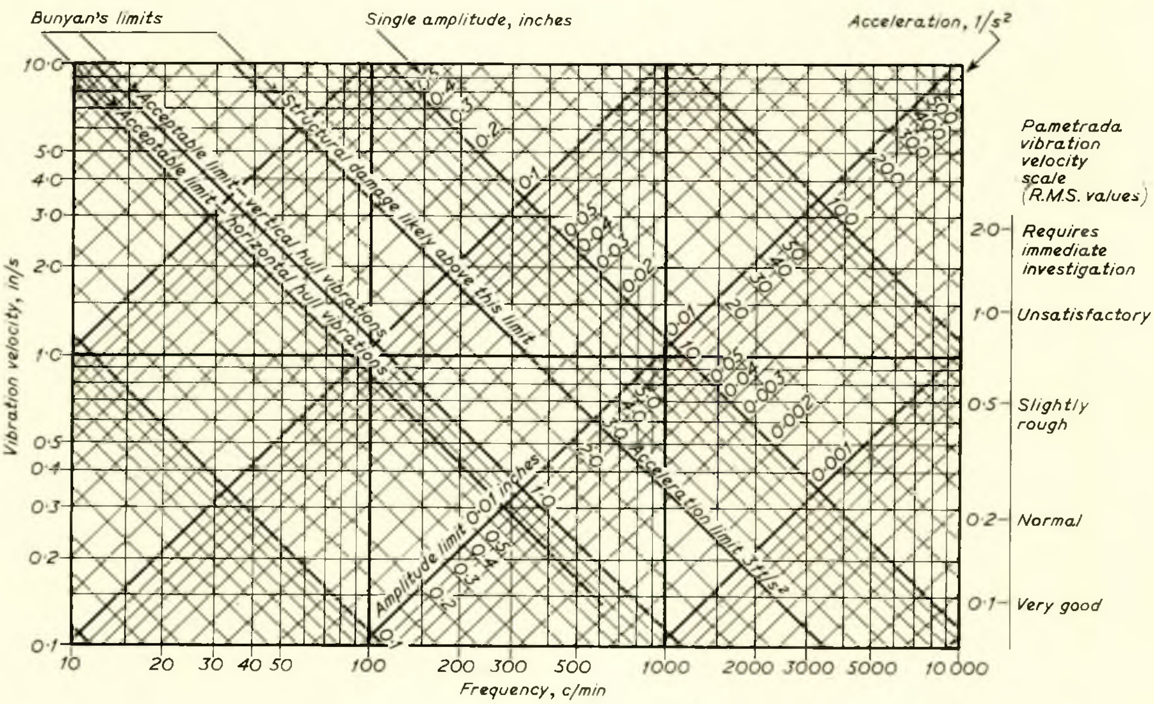


FIG. 10—*Vibration amplitude, velocity, acceleration chart*

In the case of torsional vibrations, tooth separation in the gearing is the main consideration and a plot of vibratory torque against steady torque is usually made for the secondary quillshafts. Tooth separation in the L.P. train is usually indicated at the I-node mode and this can be tolerated, since it occurs well down the speed range. Tooth separation at the II-node mode, however, would not be accepted and steps would be taken to tune the system to reduce the vibratory torques.

If this was not possible then a barred speed range would be imposed though within the experience of the authors this has rarely occurred. All recent cases involving excessive vibration torques and rough running have involved old construction using "foreign" designs of main machinery. The remedy in such cases to reduce the vibratory torque at the II-node mode has usually been to reduce the H.P. (and I.P. where fitted) quillshaft diameters to values made possible by changes in the classification society rules. If the I-node mode vibration levels (or tooth separation) were troublesome the same approach could be used by reducing the main shafting diameter between tunnel bearings, though the authors have not had need to recommend this change to date.

For axial vibrations, consideration of the forces involved suggests that avoidance of thrust reversal alone is not a sufficient criterion for satisfactory operation; vibration amplitudes and accelerations at the thrust block and main gearwheel must also be considered. Rigby[24] found that accelerations of 6ft/s^2 at the gearwheel caused extensive fretting and wear of the flexible couplings in multi-screw naval vessels and suggested that amplitudes of 0·01 in on straight runs would be acceptable. Based on measurements made in large turbine-powered passenger liners, Couchman[9] suggests acceptability criteria of 0·01 in and 3 ft/s^2 respectively for the amplitude and acceleration at the gearwheel for non-resonant axial vibrations, but with the rider that thrust variations must be given equal consideration and should not exceed $\pm$ 30 tons. These limits of amplitude and acceleration are indicated in Fig. 10 which is a log-log plot relating amplitude and acceleration to velocity for a given vibration frequency. Also included in the diagram are Yates vibration velocity levels for turbine machinery, Bunyan's limit for hull structures and acceptable limits for horizontal and vertical hull vibrations.

TABLE IV—AXIAL VIBRATION STUDY SHAFT G—SUMMARY OF DATA

1)	Weights:	Tons
	Propeller:	
	Propeller weight	32
	Entrained water (50 per cent)	16
	Nut-threaded shaft	2·5
	Taper boss	2
		52·5
	Tailshaft (41 ft × 29¼ in):	
	Tailshaft weight	42
	Liners	6
		48
	Muff Coupling:	
	Muff coupling weight	6·5
	Intermediate Shaft (23$\frac{7}{16}$ in diameter):	
	Weight per foot	0·6541
	Thrust Shaft (14 ft long):	
	Thrust shaft weight	14·6
	Thrust Assembly:	
	Thrust block	13·5
	Thrust seating	3
		16·5
	Gearing:	
	Main wheel and shaft	50
2)	Flexibilities:	
	Tailshaft:	
	Flexibility per foot	$0{\cdot}59525\times10^{-3}$ (in/lb$\times10^{-6}$)
	Intermediate Shaft:	
	Flexibility per foot	$0{\cdot}9271\times10^{-3}$ (in/lb$\times10^{-6}$)
	Thrust Shaft:	
	Flexibility per foot	$0{\cdot}8741\times10^{-3}$ (in/lb$\times10^{-6}$)

Axial Vibrations

Design Study Twin-screw Passenger Liner

A preliminary design study of propeller-excited axial vibrations was made for a twin-screw passenger liner (shaft G).

As is generally the case, the total length of shafting was fixed, but the number of propeller blades and the position of the thrust block and, hence, the combined thrust block and seating stiffness were still to be determined. The study was made in two parts. The possibility was first examined of raising the axial critical above the maximum shaft speed by moving the thrust block aft and making the thrust seating very stiff. This was not possible and so the second stage of the design study was to examine the effect of the number of propeller blades on the axial vibration of this system. A comparison was made of propellers with four, five and six blades.

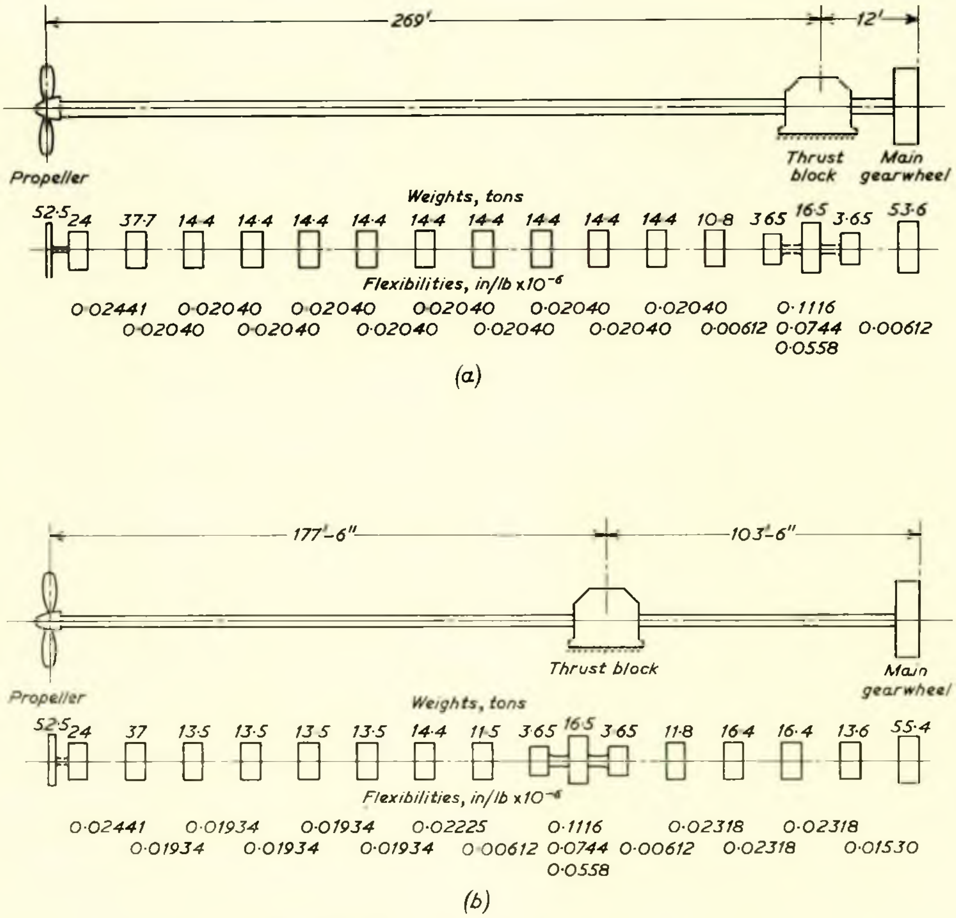
Shaft System: The basic shaft system is shown in Fig. 11. Fig. 11(a) shows the shafting with the thrust block in the forward position adjacent to the main gearwheel and Fig. 11(b) shows the same system with the thrust block moved 91ft. 6in aft. The equivalent mass-spring systems used in the calculations are also included in Fig. 11 and the data from which they were derived is presented in Table IV.

Thrust Block and Seating Stiffness: At this stage of design there was insufficient information to attempt to calculate the thrust block seating stiffness, indeed it was hoped that the design study would indicate the seating stiffness required. From measurements made by B.S.R.A. on a number of passenger liners it was estimated that a combined thrust block and seating stiffness in the range 4500-6000 tons/in would apply with the thrust block in the forward position and a maximum combined stiffness of 8000 tons/in was possible with the thrust block moved aft. Calculations were made, therefore, using combined thrust block and seating stiffness values of 4000, 6000 and 8000 tons/in.

Propeller Excitation: Thrust variation at the propeller varies with the number of propeller blades, hull form, sea conditions etc. as already discussed. For the purpose of the present study a thrust variation at the propeller equal to $\pm$5 per cent of the steady thrust was assumed in all cases. This represents the maximum thrust variation that has been derived from experimental data on a basis of prolonged maximum variations for a four-bladed propeller behind bossings in a calm sea. The thrust variation with a five-bladed propeller will be less than $\pm$5 per cent and may even be as low as half this value, whilst the variation with a six-bladed propeller will not be more than that of a four-bladed propeller under similar conditions.

Damping: In all the turbine-powered vessels examined by B.S.R.A., in which the machinery and thrust block seating were located amidships, the dynamic magnification measured was found to lie in the range 6-10 with the majority in the range 9-10. Damping, therefore, was introduced into the calculation to produce an effective dynamic magnification of 10.

Axial Resonant Frequencies: The calculated axial resonant frequencies are summarized in Table V. The frequencies given in v/min, have been related to the main shaft speed for four,



(a) Thrust block forward adjacent to main gearwheel
(b) Thrust block moved 91ft 6in aft

FIG. 11—*Axial vibrations shaft G preliminary shafting*

five and six-bladed propellers. Two frequencies are given in each case, one for the thrust block in the forward position (F) and the other for the thrust block moved aft (A). Combined thrust block and seating stiffnesses of 4000, 6000 and 8000 tons/in are included and for the softest seating, 4000 tons/in, the second mode frequencies are also given. The lowest main-shaft speed at which the second mode could be excited was 204 rev/min with the thrust block moved aft and a six-bladed propeller fitted. This was still sufficiently removed from the maximum shaft speed and thereafter only first mode vibrations were considered.

Amplitude and Thrust Variations: The axial vibration amplitudes (in × 10^{-3}) and thrust variations (± tons) at the thrust block position were calculated throughout the speed range for the various shafting arrangements and these have been used as a basis for comparison.

Effect of Position of Thrust Block: A summary of the amplitude and thrust variations at the thrust-block position for the shafting with the thrust block in the forward position, adjacent to the main gearwheel, and for the thrust block moved 91ft 6in aft is given in Fig. 12. A four-bladed propeller was assumed in each case. For a given thrust block and seating

TABLE V—CALCULATED AXIAL VIBRATION FREQUENCIES
SHAFT G

Combined thrust block and seating stiffness (tons/in)	Mode	Frequency (V/min)		Main-shaft speed (rev/min)					
				Four blades		Five blades		Six blades	
		F	A	F	A	F	A	F	A
4000	1st	494	567	123	142	99	113	82	95
	2nd	1408	1224	352	306	282	245	235	204
6000	1st	542	650	136	163	108	130	90	108
8000	1st	571	706	143	177	114	141	95	118

F = Thrust block forward; A = Thrust block aft.

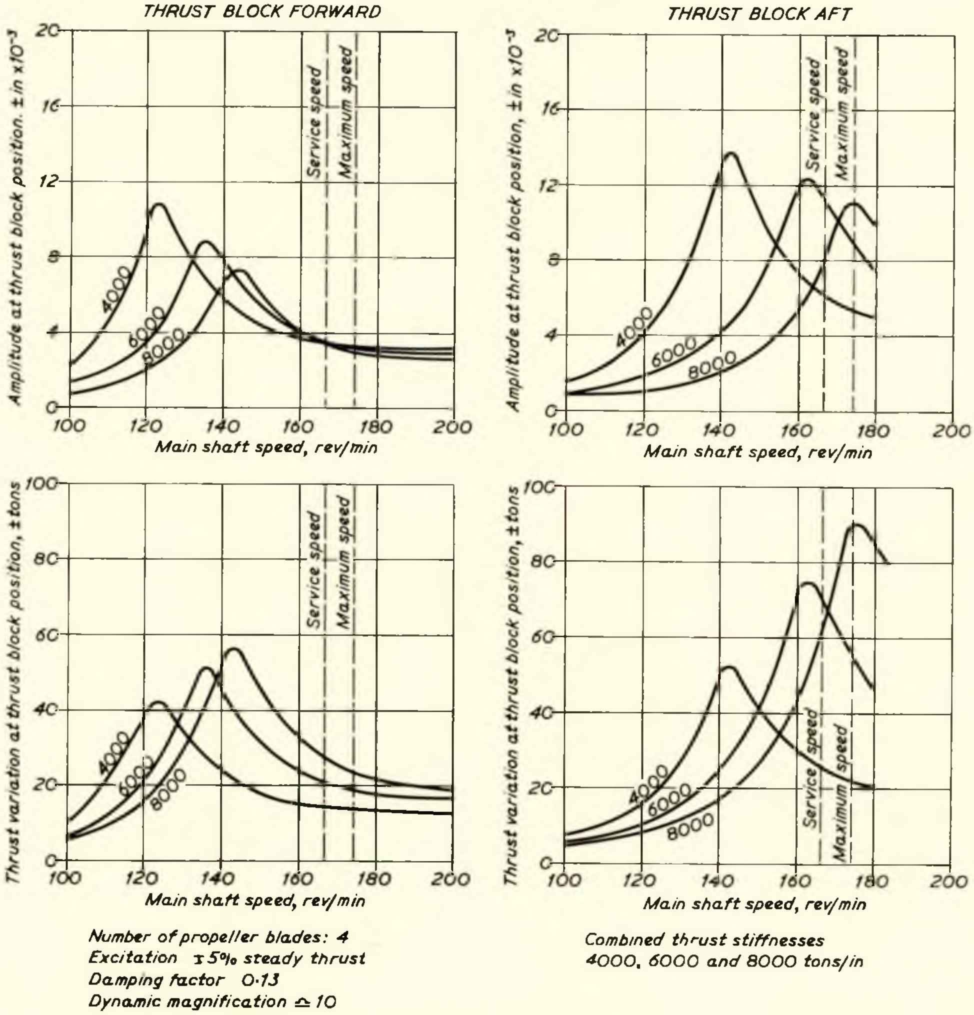


FIG. 12—*Propeller-excited axial vibrations shaft G—Comparison of the system with the thrust block in the forward and aft positions for combined thrust stiffnesses of 4000, 6000 and 8000 tons/in*

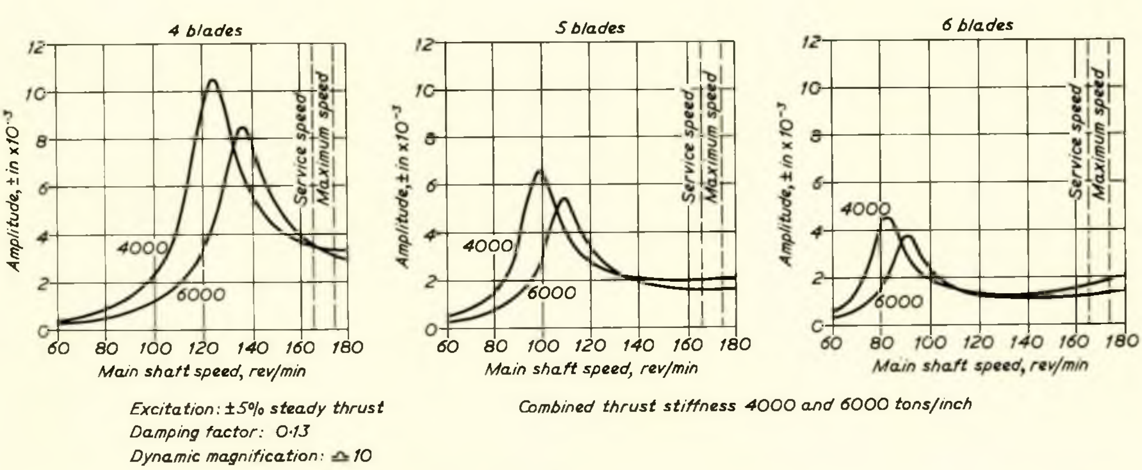


FIG. 13—*Propeller-excited axial vibrations shaft G—Effect of number of propeller blades on amplitudes at thrust block position thrust block forward*

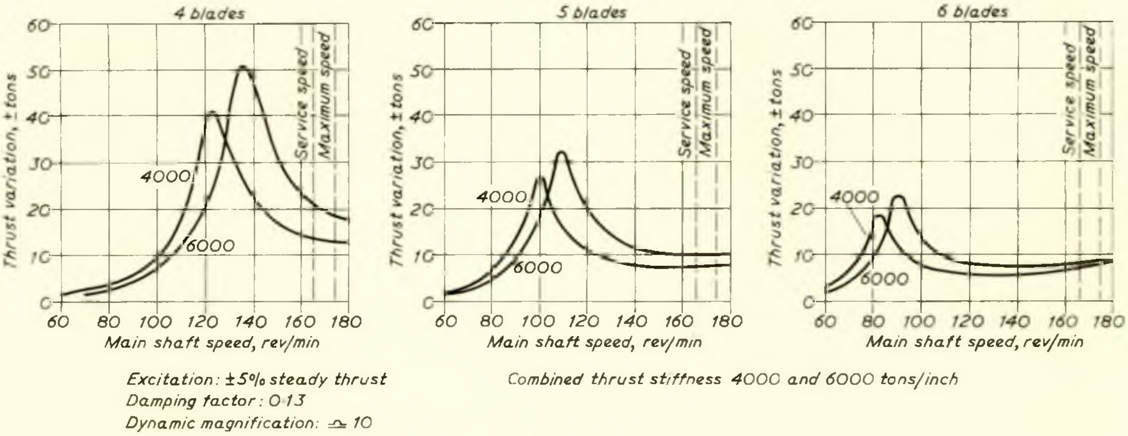


FIG. 14—*Propeller-excited axial vibrations shaft G—Effect of number of propeller blades on thrust variation at thrust block position, thrust block forward*

stiffness, moving the thrust block aft raised the frequency, but, even with the maximum combined stiffness of 8000 tons/in, the increase was not sufficient to raise the resonant frequency sufficiently clear of the maximum shaft speed. This being the case, it was then the aim to move the resonant frequency lower down the speed range, i.e., to keep the thrust block forward and increase the number of propeller blades.

Effect of Number of Propeller Blades on Amplitude and Thrust Variation: Figs. 13 and 14 summarize the effect, at the thrust-block position, of the number of propeller blades on vibration amplitude and thrust variation respectively. The curves all refer to the shafting system with the thrust block adjacent to the main gearwheel and to a combined thrust block and seating stiffness of 4000 or 6000 tons/in.

In general, the resonant peaks occurred earlier in the speed range and were smaller in magnitude as the number of propeller blades was increased.

The resonant frequencies, 494 and 542 v/min for combined stiffnesses of 4000 and 6000 tons/in, both fall into the range 400-550 v/min for which the main gearwheel amplitude and acceleration and thrust variation should be considered to determine acceptability[9]. Although the values plotted in Figs. 13 and 14 refer specifically to conditions at the thrust-block position, with the thrust block adjacent to the main gearwheel, values at the gearwheel differ by less than 1·5 per cent of those at the thrust block and hence the curves can be considered to represent conditions at the gearwheel also.

A detailed examination of Fig. 13 showed that a four-bladed propeller produced amplitudes of between 8·5 and 10·5 in $\times$ 10^{-3} at resonance with an amplitude of 3·5 in $\times$ 10^{-3} at service speed. The peak amplitudes at resonance for five and six-bladed propellers were 5·5 to 6·5 and 4·5 to 5·5 in $\times$ 10^{-3} respectively with amplitudes at service speed of less than 2·0 in $\times$ 10^{-3}. The corresponding maximum accelerations for four, five and six-bladed propellers were respectively 2·3, 1·5 and 1·2 ft/s^2. These values were all acceptable, but the figures relating to a four-bladed propeller were equal to the upper limit of acceptability given in reference (9).

Fig. 14 shows similar results obtained for the thrust variations. The maximum values at resonance for four, five and six-bladed propellers were respectively $\pm$40 and $\pm$50 tons, $\pm$25 and $\pm$30 tons and $\pm$15 and $\pm$25 tons, the two figures in each case referring to combined thrust stiffnesses of 4000 and 6000 tons/in. At service speed, the four-bladed propeller produced a thrust variation of $\pm$15 and $\pm$20 tons whilst the five and six-bladed propellers were both less than $\pm$10 tons.

These figures when compared with the suggested maximum thrust variation of $\pm$30 tons showed the four-bladed propeller to be unacceptable, the five-bladed to be just acceptable and the six-bladed propeller to be preferred. From this investigation it was concluded that the four-bladed propeller was unacceptable, but that the five and six-bladed propellers were acceptable. Of these, the six-bladed propeller appeared to be preferable to the five-bladed propeller, but only on the assumption of equal excitations at the propeller. Only by determining the wake field of the hull and calculating the thrust variation for the five and six-bladed propellers could a final assessment be made. In the event, a six-bladed propeller was chosen.

Design Study Twin-screw Diesel Ferry

A similar preliminary design study was made of a twin-screw Diesel ferry (shaft S) fitted with a four-bladed Liaaen "Navy type" controllable-pitch propeller. An alternative to the original arrangement which had the main thrust block integrated with the aft end of the main gearbox was considered but eventually rejected. The investigation is summarized in Fig. 15.

From the resonant frequencies plotted in Fig. 15 against thrust and seating stiffness it was evident that the original system provided a better margin between full speed (310 rev/min) and the axial critical speed for the combined seating stiffness range 1000-2000 tons/in applicable to this type of vessel and at the same time was still adequately above the idling speed of 155 rev/min. Calculations for the original system were then made of the thrust fluctuation at the thrust block and of the amplitude at the main gearwheel in each case at 310 rev/min and for the combined thrust block and seating range 1000-2000 tons/in.

Propeller excitation of $\pm$5 per cent total thrust, i.e., $\pm$2 tons for an assumed maximum thrust of 40 tons at full speed (310 rev/min) was assumed at propeller blade frequency for these calculations and the results are included in Fig. 15.

The results obtained showed some advantage in increasing the shaft diameter, but the gain was small compared with the difference due to thrust stiffness. The thrust fluctuation at worst was still under 20 tons, less than half the steady thrust and was as low as two or three tons for a seating stiffness of 1000 tons/in. Vibration amplitudes at the main gearwheel, however, were high. The maximum acceptable amplitude, corresponding to 3 ft/s^2, is just less than 0·003 in and corresponds to a thrust stiffness of 1000 tons/in. It was agreed, however, because of the low usage factor for this class of vessel, to accept a higher amplitude at the gearwheel, but to limit the combined thrust stiffness to 1500 tons/in. In addition it was recommended that provision should be made to fit a Michell resonance changer if found necessary. These were the results of the preliminary investigation but, as the design developed further changes became necessary to overcome tailshaft deflexion difficulties and at this stage the design was still not finalized.

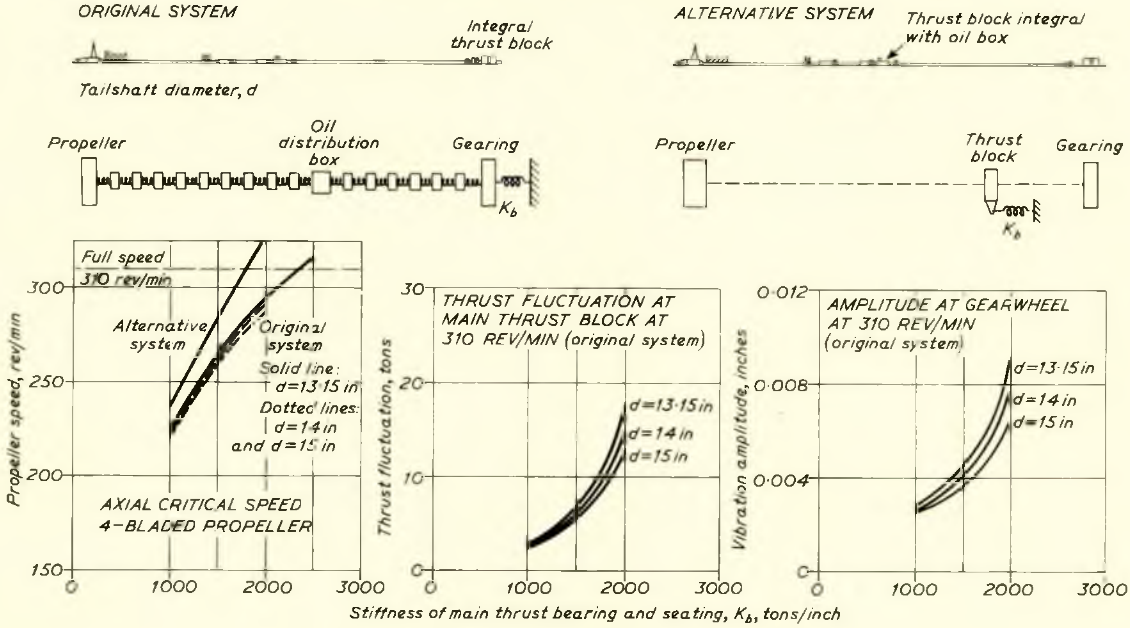
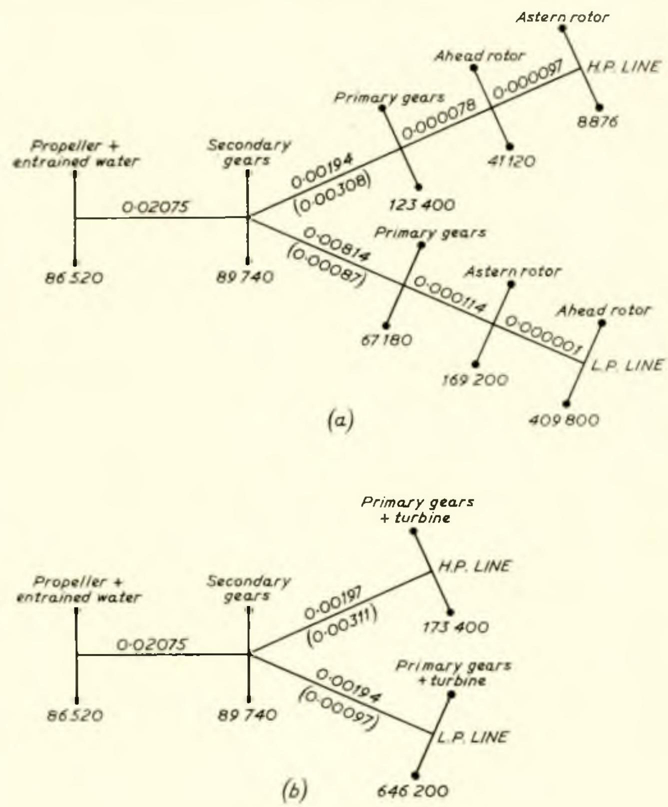


Fig. 15—*Propeller-excited axial vibrations—Shaft S*

Torsional Vibrations

A very useful aid to the designer in assessing the torsional vibration characteristics of a geared-shaft system is the admittance diagram. This diagram is simple to construct and was described by Yates[26], but from several contributions to the discussion on his paper it was obvious that the real value of the diagram had been missed. It is hoped that the following design study will illustrate some of the uses of the diagram as a design chart.



(a) Full system; (b) Reduced system
Inertias in lb ft sec². Flexibilities in rads/lb ft 10^{-6}
All values are referred to main shaft speed
Figures in parenthesis refer to H.P. quillshaft diameter 6·5 in and L.P. quillshaft diameter 8·5 in

Fig. 16—*Torsional vibrations—Shaft T*

Design Study for an 85 000-dwt Tanker

A design study of propeller-excited torsional vibrations was made during construction for a single-screw 85 000-dwt tanker (shaft T). This is a steam turbine installation incorporating the Pametrada standard combination of D/M30 turbines coupled to a PSG25/110 dual-tandem gearbox.

The shaft system was originally designed with 7½-in diameter H.P. and L.P. quillshafts and is shown diagrammatic-

TABLE VI—TORSIONAL VIBRATION STUDY SHAFT T—SUMMARY OF DATA

	Full system	Reduced system
Moments of inertia (lb ft s²) referred to main shaft		
Propeller + entrained water	86 520	86 520
Secondary gears	89 740	89 740
H.P. line:		
Primary gears	123 400	
Ahead rotor	41 120	173 400
Astern rotor	8 876	
L.P. line:		
Primary gears	67 180	
Ahead rotor	169 200	646 200
Astern rotor	409 800	
Flexibilities (rad/lb ft × 10^{-6}) referred to main shaft		
Main shaft	0·02075	0·02075
H.P. line:		
Secondary quillshaft	0·00194 (0·00308)*	
Primary coupling	0·000078	0·00197 (0·00311)*
Rotor	0·000097	
L.P. line:		
Secondary quillshaft	0·00184 (0·00087)*	
Primary coupling	0·000114	0·00194 (0·00097)*
Rotor	0·000001	

*Figures in parenthesis refer to H.P. quillshaft diameter of 6·5 in and L.P. quillshaft diameter of 8·5 in.

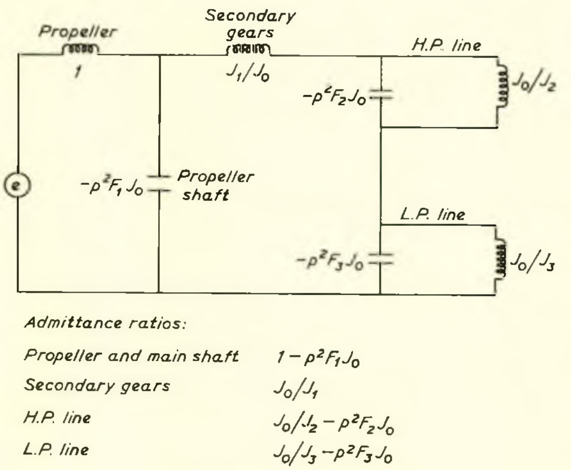


Fig. 17—*Equivalent electrical circuit for reduced system showing admittance ratios*

ally in Fig. 16. The upper diagram shows the full eight-mass system and the lower diagram shows the reduced system obtained by combining the inertias of the ahead and astern rotors with the primary gears. All inertias and flexibilities have been referred to main shaft speed and the flexibility figures shown in parenthesis refer to H.P. quillshaft diameter of 6·5 in and L.P. quillshaft diameter of 8·5 in.

The equivalent electrical circuit for the reduced system together with admittance ratios is shown in Fig. 17. The admittance of an element is the inverse of impedance, i.e., propeller J_0, impedance jpJ_0, admittance $\frac{1}{jpJ_0}$. The admittance ratio, g, is then the ratio of the admittance of the element to the admittance of the propeller, i.e., H.P. turbine J2, admittance $\frac{1}{jpJ_2}$ admittance ratio $\frac{J_0}{J_2}$. The admittance diagram (Fig. 18) has been constructed from the admittance ratios of the elements. For the reduced system these are linear functions of frequency squared and were simply plotted as straight lines in the diagram.

A rapid estimate of the torsional critical frequencies can now be made from the admittance diagram. Resonant frequencies correspond to values of p^2 for which the impedance of the circuit is zero. From the admittance diagram, bearing in mind that small admittance means large impedance, it is seen that the I-node critical is determined principally by the propeller and shafting and the L.P. line and is given approximately by the value of p^2 for which their admittance ratios are of equal magnitude, but opposite sign. The true value of p^2 will be somewhat lower than this estimate due to the influence of the H.P. line and secondary gears, but the error is usually quite small. The II-node critical is fixed mainly by the H.P. and L.P. lines and again corresponds approximately to the value of p^2 for which their admittance ratios are equal in magnitude, but of opposite sign. The true value of p^2 will again be smaller than the estimated value, since the influence of the secondary gears is greater than the propeller and shafting. Using the admittance diagram in this way the I-node and II-node criticals for the reduced system were estimated to occur at 57·5 and 103·0 main shaft rev/min respectively. These estimates compare with 57·3 rev/min and 102·0 rev/min obtained from a detailed analysis of the full eight-mass system.

The propeller-excited vibratory torques in the H.P. and L.P. quillshafts, for the system as designed assuming an exciting torque equal to 8 per cent of the steady torque, were plotted against the steady torques in the quillshafts (see Fig. 19). Tooth separation is indicated in the L.P. line at the I-node critical, but, because this is well down the speed range, would be acceptable. The vibratory torques at the II-node critical, although they do not indicate tooth separation, were nevertheless not acceptable. These torques were reduced by reducing the diameter of the H.P. quillshaft to 6·5 in and increasing the diameter of the L.P. quillshaft to 8·5 in (modified system "a"), i.e., the system was "tuned". On the admittance diagram this is equivalent to bringing the intercepts on the p^2 axis of the H.P. and L.P. line admittance ratios closer together as indicated in Fig. 18. This, however, still left the II-node critical at 102 rev/min compared with a maximum service speed of 105 rev/min and although the indicated vibratory torques were small this situation was considered unsatisfactory. By reducing the L.P. quillshaft diameter to 7·5 in again (modified system "b") the II-node critical was shifted down the speed range to 91 rev/min and whilst the estimated vibratory torques were again greater than system "a", system "b" was considered to be the better arrangement and was finally adopted.

Although the digital computer is a valuable aid in the full analysis of the final shafting arrangement and for estimating

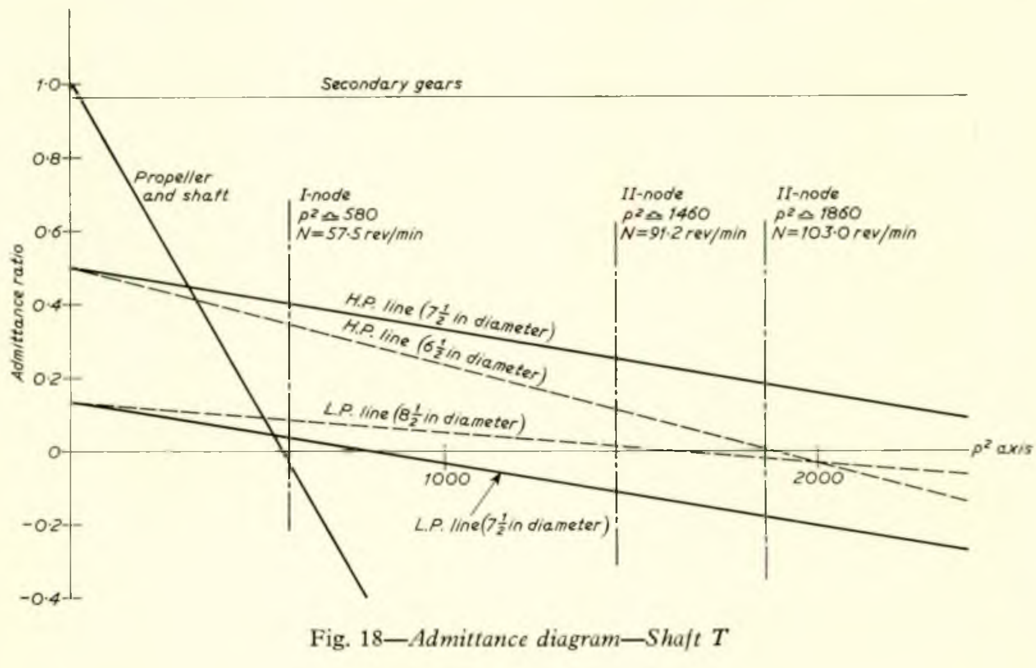


Fig. 18—*Admittance diagram—Shaft T*

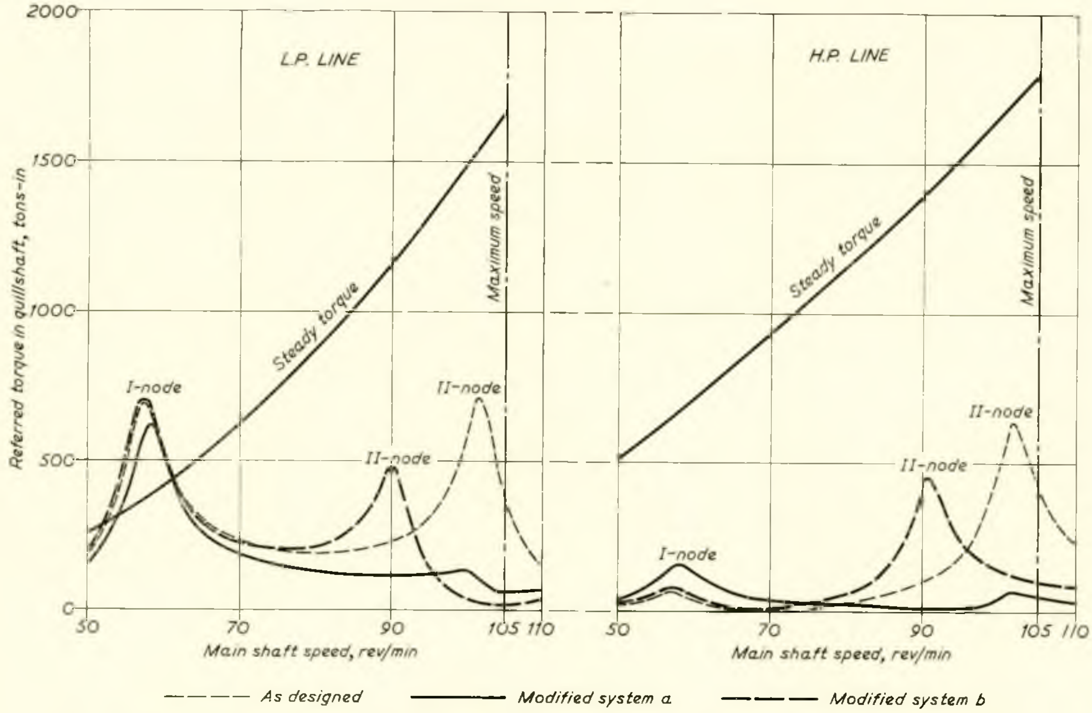


Fig. 19—*Estimated steady and propeller-excited vibration torques in quillshafts—Shaft T*

vibratory torques in the system, much of the early design work can be done more quickly using the admittance diagram. In addition to providing a quick estimate of the critical frequencies, it also presents a graphical picture of the effects of changing the inertia or flexibility of any element of the system and as such is truly a design chart.

4. THE SHAFT SYSTEM

Supporting the Shaft System

The paper has not, as yet, considered how shafting is supported in the ship since, apart from stiffness characteristics of the main thrust block and its seating, and the damping occurring in the various bearings, the axial and torsional vibration characteristics will be unaffected.

Journal bearings are required to carry the deadweight of the shafting and its attachments, without being individually overloaded during any foreseeable service condition, including hull deformations due to distribution of cargo or heavy seas, or because of the influence of normal weardown of other bearings in the system. Furthermore, they should be so spaced that shaft whirl will not occur, either in the intermediate shafting or at the tailshaft assembly. Additionally, a main thrust bearing is required so that thrust generated by the propeller may be applied to the hull and so propel it.

In the broad view, a choice must be made between hydrodynamic bearings and those using rolling elements. In the larger vessels, representing most of the personal experience of the authors, oil-lubricated white-metalled journal bearings have been used for all intermediate shafting and, in recent years, also for stern-tube duty. Smaller vessels, particularly the ferry group represented by shafts Q, R and S in Fig. 2, have now adopted the grease-lubricated split roller bearing for intermediate shafting. Naval vessels (shaft K) also use this type of bearing but in a modified flange-mounting form permitting attachment to bulkheads rather than to special seatings. Roller bearings of the solid-race type require removable flanges to be used on shafting and have been used recently in large bulk carriers fitted with the "Algonquin" design of stern gear[8]. This arrangement retains a conventional water-lubricated rubber-stave bearing adjacent to the propeller and therefore is not as complete in its adoption of roller bearings as was the *Owasco* Class of U.S. Coastguard cutters commissioned about 1940[27]. In this latter group a "Syntron" seal was fitted to exclude sea water and very satisfactory service has been given[20]. More recently U.S.S. *Timmerman* was fitted with roller races in the two "A" brackets supporting the starboard shaft, the lubricant being piped through the struts from header tanks above the waterline.

Roller bearings of both the split and solid types possess the useful characteristic of being essentially zero clearance and, therefore, capable of carrying loading reversals without the need to absorb a diametral clearance. This is an asset where transverse vibration may occur and tends to compensate for the lower damping capacity which is a further characteristic of these bearings.

Roller bearings offer a low starting torque and are free from wear on turning-gear conditions, but have a specific life which is inversely proportional to the cube of load and to shaft speed, so that life is directly influenced by the service operating conditions. Programmed replacement during the life of the ship may be necessary for most classes of operation. "False-Brinelling", arising from vibration caused by auxiliaries when the main shaft is stationary, is also a possibility, but by improving the micro-geometry and surface structure of the races and rolling elements, bearing manufacturers have done much to eliminate this problem.

Main-shaft bearings are essentially constant-load/variable-speed units and therefore resemble turbine bearings rather than gear bearings. With white-metalled bearings this means that the permissible specific loading is determined primarily by wear resistance when the shaft is turning at low speed, as during the starting transient or when the turning gear is engaged. In extreme cases the normal ring-oiled lubricating method may require supplementing during this form of use. Higher specific loadings may also mean a greater power loss in an individual bearing, this being determined by the coefficient of friction within the oil film and is therefore dependent upon mean film temperature as well. Consequent overheating of the plummer block is readily avoided by including a cooling coil in the oil sump. While most of the shaft systems shown in Figs. 1 and 2 have plummer blocks operating at specific loadings of 50-60 lb/in^2, some of the more recent systems have bearings operating satisfactorily at twice this level. This has been due to two main factors—the need to increase the span between bearings, while for a given shaft diameter no corresponding increase in white-metal area has been made. This latter effect stems from the retention by British and Continental bearing manufacturers of L/d ratios of about $\frac{2}{3}$. Because of the small shaft deformations involved there seems to be a good case for the future adoption of ratios of 1-1·5, as used across the Atlantic.

Selection of white-metal tunnel bearings involves the choice of using the bush-type construction or the tilting-pad unit, the latter normally being reserved for heavy duty conditions as may exist adjacent to the stern tube where heavy transverse vibration could be encountered. Bearings at this position are normally loaded in their bottom half (shaft Z is an interesting exception), but additional contact marking in the top half is commonly encountered when vibration is serious. For this reason the diametral clearance of this particular bearing is often reduced to about half that of the tunnel bearings, so that the permitted vibration amplitude is thereby reduced.

Spacing of Tunnel Bearings

In the London discussion of the paper by Andersen and Zrodowski[1], Commander A. J. H. Goodwin concisely stated the Admiralty policy that maximum shaft flexibility should be sought by using the least number of bearings consistent with suppression of shafting whirl. The practical effects of this are evident in Figs. 1 and 2 by comparing ships J, K and N with the remainder. Whereas bearing spans of 30 shaft diameters occur with the naval vessels, ferries use spans of about 20 diameters and the remaining merchantmen use 6-12 diameters. While the use of hollow shafting in naval vessels helps to justify their great spans, the simple fact remains that most merchant vessels use far too many bearings.

For a uniform shaft supported on several bearings such that all spans are equal and the shaft ends are free from angular restraint, the whirling speed N_c, rev/min is given by[10]:

$$\frac{N_c s^2}{K} = 1905 \times 10^4$$

where s = span between bearings (inches),
K = radius of gyration (inches) of the shaft section.
For a circular section, diameter d, $K = \frac{d}{4}$

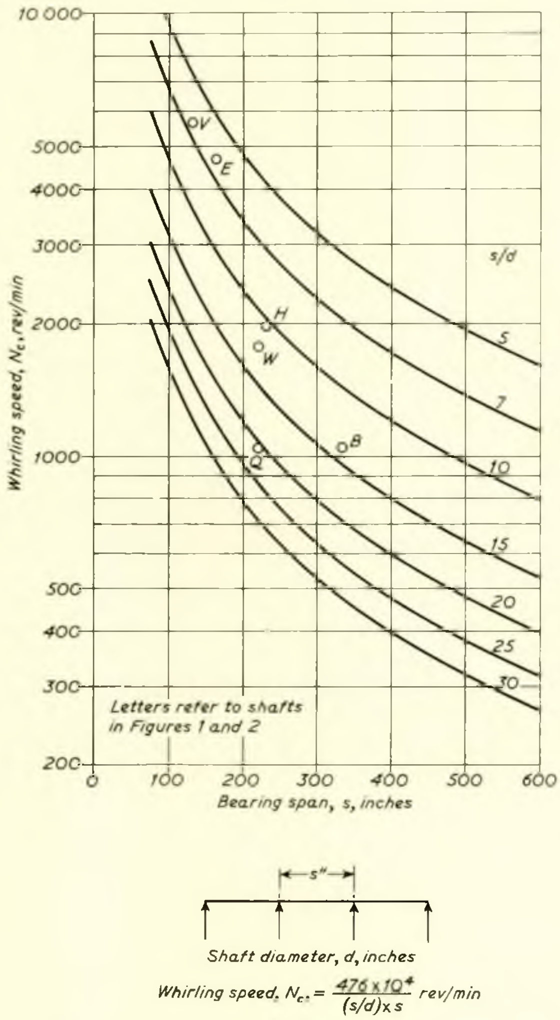
and $$N_c = \frac{476 \times 10^4}{(s/d) \times s} \text{ rev/min.}$$

This relationship can then be evaluated and plotted as in Fig. 20 to give guidance for preliminary layout of shafting systems. In practice the bearing spacing at the main engine and tailshaft is much closer than at the tunnel section and shaft diameters are also larger. Therefore, it will usually be found that this method will underestimate the whirling speed of the system. As design progresses these departures from uniform spacing and diameter as well as the influence of concentrated loads such as torquemeters, turning wheels and oil control boxes (for controllable-pitch propellers) can be taken into account using, for example, the Prohl method to consider the system complete from the main wheel to the propeller.

Certain of the shafts have been plotted in Fig. 20 to indicate the margin on whirling with existing merchant installations. Shaft Q, with a safety factor of 3·73, but using a three-bladed propeller, may raise the query as to whether whirling can be excited by propeller excitation at blade frequency. From experience with the channel steamers and naval vessels this does not seem to be so.

In many cases the close spacing of bearings arises because of the misconception that proper alignment of the shafting can only be done if there are two bearings per shaft length. Many of the systems shown in Fig. 1 include lengths having only one bearing, and in shafts K and S completely unsupported lengths appear. Clearly no difficulty is introduced in either case. Commander Goodwin discussed this aspect also[1], pointing to the use of temporary supports as an installation aid. Subsequent removal of bearings during, say, a tailshaft survey imposes the need to replace them again. This operation need not be any more difficult than if two bearings were on each length since the original location of the temporary support will already be known.

Where unsupported shaft lengths are included two approaches are possible—two temporary supports may be used or the length may be bolted up to one of its adjacent lengths of shafting so that one very long shaft length is being handled. An extreme example of this latter technique was used to line-in shaft B; all three lengths of intermediate shafting were bolted together to form a single length which was supported in the two plummer blocks during the alignment procedure.



Assuming: 1) Equals spans
2) No end restraints

FIG. 20—*Whirling speeds of intermediate shafting*

Main Thrust Bearings

Main thrust bearings have generally been placed adjacent to the main wheel in turbine-powered ships, twelve of the shafts depicted in Figs. 1 and 2 having it fitted to a separate thrust shaft while six have it integrated in the gearbox. The two remaining turbine-powered shafts have the main thrust block located well down the tunnel. These numbers do not really indicate the percentages of each type at sea since, for example, shafts N and U each represent about one hundred shafts in service while shaft E is one shaft of one twin-screw vessel.

Integrating the thrust block in the gearbox offers an obvious simplification of the forced lubrication system and also reduces the cost of seatings. As has been adequately discussed elsewhere[11, 24] thrust seating stiffness tends to be reduced with this arrangement and it should not be adopted on installations where axial vibrations are likely to be troublesome, unless a full investigation justifies an integral design.

For smaller ships with lower power ratings, the integral design thrust block has been of the "horse-shoe" type, as is commonly used with direct-coupled Diesel engines. While this design is cheaper than the "all-round" design and simplifies the operation of removing a main wheel or crankshaft, there is an unfortunate tendency to create substantial bending moments at the thrust collar. Two factors contribute to this. First, the summation of the loading carried by the individual pads is displaced several inches below the shaft axis due to the "missing" top pads and secondly, casing distortion under this thrust loading tends to magnify the effect by unloading the upper pads and overloading those at the bottom. Also, casing distortion itself is greater for this type as is evidenced by the results of a recent testing programme conducted by Pametrada, in conjunction with B.S.R.A.

Use of a separately-mounted main thrust block adjacent to the gearbox brings with it the practice of using a thrust shaft which can be reversed to present in turn both faces of the collar for transmission of ahead thrust—as in ship B. With the high reliability of this component there seems little justification for retention of this practice in the future, and the asymmetric thrust shafts shown for ships A, L, M and W may reappear in future ships.

The separately-mounted thrust block also introduces the need to seal the pressurized central chamber against excessive oil leakage at the shaft periphery at each end. British practice in this regard relies upon long white-metal seals which also offer the prospect of high load-carrying capacity as journal bearings. Before being used for this purpose, however, the main thrust block should preferably be some 6-12 shaft diameters from the nearest bearing for the reasons generally described in[1, 17, 19]. At the time of writing, the authors have not encountered any case of gearing failure which was directly attributed to the close proximity of the main thrust block although, on occasion, there has been a suspicion that this may well have been a contributory factor. It should be borne in mind that these references are all strongly influenced by experience with U.S. naval gearing where hobbed and shaved gear elements of 75-80 tons/in^2 alloy steel are used exclusively. So that the size of these elements may approach that which is possible with surface-hardened gears, as used in British naval practice (shaft K and also certain elements in shaft N), gear ratings have been raised to a level where little capacity is left for the acceptance of overloads arising from otherwise minor influences. This is particularly true in the case of tooth root-stress levels and, for most recent U.S. naval designs, tooth breakage failures appear to have been accepted on early ships in the hope that intelligent analysis will safeguard the remainder of the class.

Selection of size of main thrust blocks is largely determined by specific loading of the white-metalled pads and for merchant ships this is normally limited to a maximum of about 300 lb/in^2 which leaves a very substantial overload capacity to handle the transient loads which occur during manœuvring. Naval thrust blocks use significantly higher specific loadings and can, therefore, be much smaller for a given maximum thrust value. By integrating the thrust block at the forward end of the main wheel, a further reduction in overall dimensions is also possible. At this position the eye of the thrust collar is not required to carry propeller torque and can therefore be much smaller in diameter than at any other alternative position.

Thrust values for a particular type of ship show a good correlation with shaft torque rating and Fig. 21 has been included to indicate the general relationship against this parameter and also to compare the thrust bearing capacity for different types of ship.

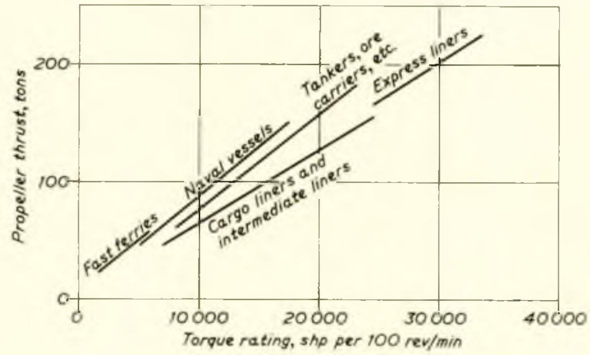


Fig. 21—*Maximum thrust values*

Tailshaft Design

While the fundamental duties of a tailshaft are simple, namely, to support the propeller, transmit torque to it and to transmit propulsive thrust to the intermediate shafting, many other factors, which arise mainly from environmental conditions, add considerable complication to the rational design of a tailshaft and its supporting bearings. Apart from the need to give satisfactory performance under steady-load conditions, where bending and torsional stresses arising from propeller mass and steady torque are fairly readily calculable, a high degree of dynamic load is introduced due to wake variations across the propeller disc. While much can be done to estimate the general levels of these cyclic loadings once the wake field surrounding the hull is known, they cannot as yet be applied with confidence to detail design of the tailshaft and its bearings, but should still be kept in mind as an indication of what may happen in service.

Many of the obvious features of the tailshaft are due to hull type and method of shaft support. For example, the fast ship necessarily has fine lines and a long tailshaft is therefore required if its forward end is to be reasonably accessible from inside the ship. An even greater length is normally required for multi-screw propulsion due to the influence of athwartship displacement of the shaft upon the axial location of its penetration of the shell-plating lines.

Enclosing the tailshaft in a bossing permits retention of the conventional stern-tube assembly, as is evident with most of the multi-screw ships included in Figs. 1 and 2, but bossings considered as beams cantilevered from the main hull possess various modes of vibration, the natural frequencies of which can coincide with propeller blade frequency in the higher power ranges. Several cases have been experienced of hull, bearing and tailshaft damage arising from such resonances.

The alternative use of "A" brackets, as well as providing better support for the shaft, also allows free access to most of its length. A coupling flange can therefore be placed at an outboard position, as with naval ship K and has also been done in merchant ships, e.g. *Canberra,* where the tailshaft length can be greatly reduced. Further recent merchant use of "A" brackets includes *France* and certain channel steamers, encouraging the belief that this trend may continue.

In Fig. 22 four tailshafts have been drawn to a common scale for ease of comparison of their principal features. To demonstrate some of the variations which occur this group comprises shafting for:

a) an express liner;
b) a channel steamer;
c) a single-screw merchant tanker;
d) a faster single-screw ship of much finer hull form.

Prior to lining-in the intermediate shafting the propeller is fitted and this is the condition illustrated. Propeller weights for a), c) and d) are in the range 25-35 tons and the overhung moments they produce about the effective point of support in each aft bush are in similar agreement. If the corresponding moment of shaft weight, forward of the effective point of support, exceeds the overhung moment then the shaft bears down on its foremost bearing, otherwise it will gain additional

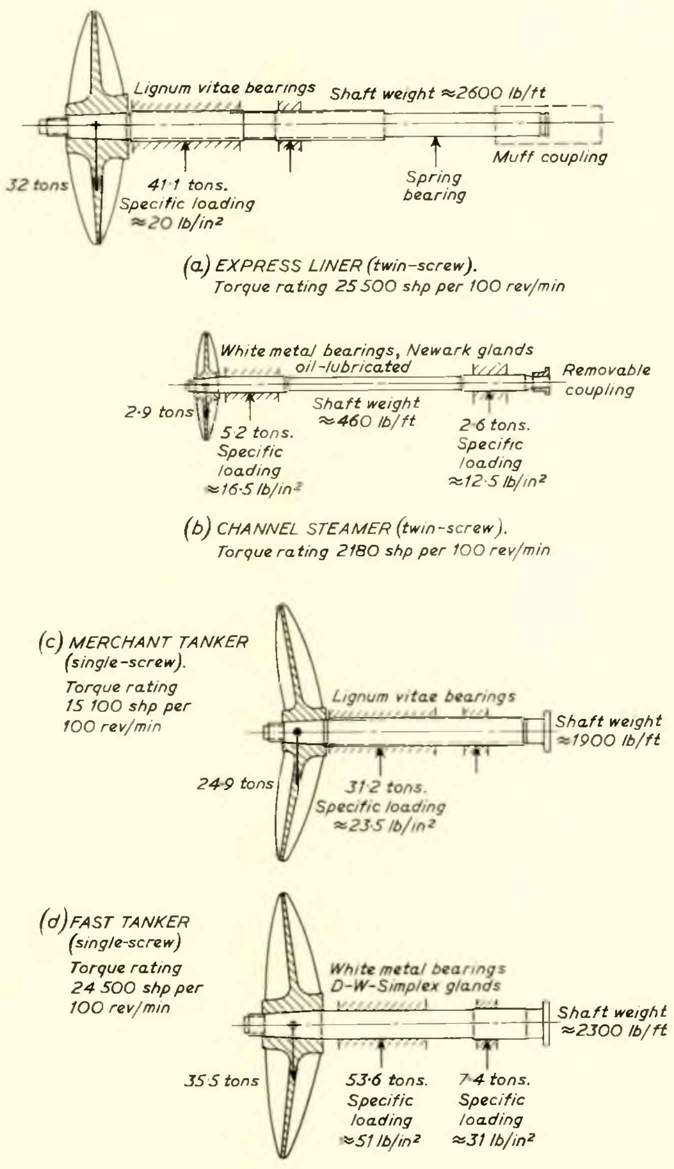


Fig. 22—*Comparison of various stern-tube arrangements (bearing loads allow for buoyancy when immersed)*

restraint from the top half of this bearing. Shaft a) successfully achieves the former condition by virtue of its extreme length and the shorter shafts, c) and d), require application of a jack to obtain bottom loading as, for instance, in setting the flange at its reference position for aligning the intermediate shafting. After the shaft system is fully coupled, weight must then be transferred from the intermediate shafting to the tailshaft flange to sustain bottom loading in the absence of the jack and this transfer is necessarily produced by offsetting the tunnel bearings below the stern-tube bearings. For tailshafts a), c) and d) the deflexion curve will be concave downwards between bearings.

Shaft b), being fitted with a very light propeller, is subjected to only a very small overhung moment and therefore also achieves downward loading on the forward bush without dependence upon jacking or transfer of intermediate-shafting weight. In this instance, the deflexion curve sags at both free ends and also between the bearing due to the dominance of shafting weight in this span. A more favourable agreement is therefore obtained between shaft curvature and white metal even though no special design features are used. Unfortunately, this benefit is largely wasted because of the very low weights which have to be supported in this class of ship.

Specific loadings of stern-tube bearings, until recently, were not design parameters but, rather, the consequence of classification requirements for length to diameter ratios (L/d) of 4 for the normal materials and lubrication methods. Successful application of certain oil-seal designs to white-metalled stern-tube bushes caused these rules to be revised permitting an L/d ratio of 2·5 for this particular category, and more recently again they have been modified to permit specific loading to be the design criterion while L/d now becomes the derived quantity.

For different ships having the same torque rating and, hence, shafting diameters, the aft-bush length has not varied greatly where water-lubricated materials have been used. Nominal specific loadings have ranged from 20 to 30 lb/in², the lower value being for fast ships having light propellers and the higher value for tankers and similar ships. These levels have held for ships using lignum vitae staves, as do many of those shown in Figs. 1 and 2, and also for plastics-laminates as in ship V. Bearings of a fibre-reinforced cresyllic resin are fitted to ship K and are returning a most satisfactory performance while having a nominal specific loading in excess of all but a very small number of oil-lubricated white-metalled bushes. It presents the successful outcome of a development programme comprising successive small-scale and full-scale rig tests, seagoing trials in a single ship and then general adoption into new construction in combination with a rational alignment procedure.

Bearing clearances for lignum vitae and the synthetic materials must necessarily be large to allow for swelling during initial absorption of water, and weardown is usually permitted to exceed ¼in before re-lining is attempted—a figure which can give six to eight years life in multi-screw ships, but may be exceeded in as many months with slow single-screw ships, particularly if installation has been defective. As weardown proceeds and clearance increases, the occurrence of tailshaft whirl is encouraged. Ships fitted with oil-lubricated white-metalled stern-tube bearings are consistently showing weardown rates of 0·001in per annum, but in order to do so a good oil seal is mandatory, it must be correctly fitted, and good alignment must be obtained between tailshaft and bearing surface. Initial clearance for this type of bearing can be about half that used for the water-lubricated bearings.

Propellers come in an apparent infinity of shapes and sizes, but the factors which mainly concern this present thesis are the weights, inertia, position of centre of gravity, and number of blades. The first two are shown in Figs. 5 and 6 for a large group of post-war ships and it is clear that weight is largely determined by torque rating and type of vessel. The latter point influences approach velocity, this being highest at the wing screws of fast naval vessels and lowest for slow, single-screw vessels having a high block coefficient. Development of nickel/aluminium bronzes and copper/manganese/aluminium alloys with improved fatigue strength has also had a considerable effect in reducing weight by permitting smaller blade thickness. Built-up propellers achieved a brief popularity in the early post-war period as a means of minimizing delays in certain Asian ports where the absence of normal port facilities dictated the use of barges for loading and unloading. Under these conditions the loss of a propeller blade was comparatively frequent and the built-up construction promised a simple method of blade replacement. The substantial increase in weight is evident, and several cases of fatigue failure in the blade roots occurred as a consequence of inadequate aperture clearances.

Installation and Withdrawal of Shafting

Early in the development of a new ship, an arbitrary decision must be made as to whether the tailshaft will be arranged for withdrawal inboard or outboard—the choice being relatively simple for single-screw propulsion, but verging more on personal preference where multiple screws are involved in combination with bossings. The general case for multi-screw ships will therefore be discussed first.

Where "A" brackets have been selected as the means of supporting the outboard shafting and the tailshaft is terminated at an outboard position, as with ship K, no great complication is imposed by using either a solid flange or a removable muff coupling at this position. A solid flange does, however, compel forward withdrawal of the tailshaft itself, but the bearing bushes must be jacked clear of the "A" brackets onto the tailshaft first, so it can be angled to clear the hull form.

When bossings are chosen, there is still a free choice between using a solid tailshaft flange or a muff coupling, the two types being shown respectively on shafts D and E. A solid-flange compels forward withdrawal of the shaft, but also enables a shorter tailshaft to be used—which may be compulsory anyway since it must now be fleeted into and out of the ship. Consequently, at least one length of intermediate shafting must be removed to provide withdrawal space and, because of the poor accessibility at the tailshaft flange and perhaps at the next flange as well, this length will probably be the second or third length from the tailshaft. It therefore becomes necessary to remove the holding-down bolts for tunnel bearings on two or three lengths of intermediate shafting. One length is then displaced transversely and the tailshaft plus one or two intermediate shaft lengths is then drawn forward for inspection of the liner etc. If shore-repairs are found necessary the tailshaft flange-joint is then broken and the tailshaft fleeted out of the ship. On replacing it, the reverse procedure is followed, but with the necessary extra operation of checking alignment of the tunnel bearings which are disturbed. Such a routine is—to say the least—formidable.

Muff couplings have, therefore, appeared in certain merchant ships such as E, F and G although being generally regarded as naval practice—see the removable couplings fitted to ships, J, K, M and N. While they do offer relief from the previous procedure, detail design requires careful attention to minimize the occurrence of fretting in the coupling halves and bolts which may hinder re-opening the muff at any required time. A further penalty is the necessarily longer tailshaft to bring the muff coupling to a position where accessibility is not too difficult—see Fig. 22(a). Forgings exceeding 80 feet in length can be obtained if required, although this length would almost certainly be used only for naval vessels. In service, a long tailshaft carrying a heavy muff coupling can be advantageous in providing a substantial downward loading on bearings at the forward end of the tailshaft. It will, to a large extent, cancel out the influence of the overhung propeller and make the system less sensitive to the load-fluctuations produced by the propeller traversing the wake field. For long-shaft installations, such as shafts G and H, further consideration would need to be given before selecting a muff coupling because of the effect the extra mass would have in lowering the axial critical frequency of the system. If, on balance, adoption of the muff coupling was still favoured then careful detail design must follow if a minimum weight design is to be obtained.

A further application of the muff-coupling principle is seen in shaft S, where different circumstances apply. Here a controllable-pitch propeller had been chosen to enable the best characteristics of a Diesel engine to be exploited. Because of the hub internals appropriate to c.p. propellers, a conventional "cone and key" connexion cannot be considered and a flange connexion is now universally used for such installations. The outboard flange therefore impels withdrawal of the tailshaft outwards and consequently requires a removable inboard coupling.

The exceptional performance of tailshafts of this design, from the point of view of freedom from fatigue cracks etc., is now a matter of record and the first reaction is to associate this wholly with adoption of the flanged connexion to the c.p. propeller. However, it should be remembered that these tailshafts are not designed to the same rules as those for solid propellers. This is necessarily so because the combination of greatly increased weight and overhang for the c.p. propeller produces a massive increase in bending moment at the critical section of the tailshaft. Further, the aperture normally provided is more generous in its clearances and could be expected to produce a more favourable wake distribution.

All the foregoing comments apply, to a greater or lesser degree, to single-screw vessels, although, in the case of ships having a high block coefficient, good accessibility to a solid-flange is possible, even with very short tailshafts as in ships T-Z, which are all oil tankers. For this reason, tailshaft withdrawal inboard is possible after the adjacent intermediate shafting length has been cleared and much less disturbance is involved than with the multi-screw ship. In many ships of this present type the shafting system comprises only the main engine, main thrust shaft, one intermediate shaft length and the tailshaft as in shafts X, Y and Z.

Tailshaft Whirl

Tailshaft whirl has been discussed in great detail by Jasper[12, 13] and three approximate methods of predicting whirling speeds were proposed. The simplest of these is adequate for the present purpose of highlighting the design features requiring particular attention, though the method is of very limited accuracy unless used in conjunction with coefficients obtained by measurement of whirling speeds of similar shafts. For the first order critical whirling speeds, Ω_{N1}, i.e. whirling of the shaft in the direction of rotation of the shaft, with no node between the end of the shaft (considered built-in) and the bearing (assumed rigid) the formula is:

$$\Omega_{N1} = \sqrt{\frac{EI}{mb^2\left(\frac{b}{3} + \frac{l}{4}\right)}}$$

where: E = Young's modulus;
I = second moment of area of shaft cross-section;
m = the mass of the propeller;
b = the length of overhang;
l = the length from the bearing to the built-in end of shaft.

For the shafts shown in Figs. 1 and 2, the expression $\left(\frac{1}{3} + \frac{l}{4b}\right)$ does not vary significantly hence:

$$\Omega_{N1} \propto \sqrt{\frac{D^4}{mb^3}}$$

where D is the shaft diameter.

Thus the whirling speed, as a first approximation is directly proportional to (shaft diameter)^2 and inversely proportional to (propeller mass)$^{\frac{1}{2}}$ and (length of overhang)$^{\frac{3}{2}}$ The important parameters, therefore, are shaft diameter, length of overhang and propeller mass in that order. Two of these, length of overhang and propeller mass are adversely affected by the use of aft-raking propeller blades. These have been adopted mainly on single-screw ships to reduce wake effects. Is the improvement obtained sufficient to offset the increased risk of shaft whirl and possible hull vibration problems, the increased first cost of the propeller and the reduced efficiency of the propeller in stopping the ship? A reappraisal of the use of aft-raking propeller blades is perhaps necessary.

Transverse Flexibility

During the last decade considerable effort has been directed towards improved understanding of the transverse bending characteristics of propulsion shafting. The incentive to do so has been elimination of any redundant bearings shown to be present in older systems by the appearance of "hot" or unloaded bearings under certain conditions of hull loading or of weardown in the stern-tube bearings.

Richards[23] in 1954 presented results for the static bearing loads for the shaft system of a small Diesel-powered tanker. The problem was solved using the Principle of Three Moments applied to a continuous beam of variable cross-section supported on many bearings standing on a rigid base. Calculations were made with all bearings in line and also for specified amounts of weardown of the bearings.

Later Kosiba *et al*[17] reported on experience gained at the Boston Naval Shipyard in investigating and rectifying shafting problems occurring in several U.S. naval vessels. Unloading of bearings, excessive rates of weardown, shafting whirl and gearing misalignment had all been encountered and corrected by reference to the transverse bending flexibility of the particular shaft system. The Hardy-Cross relaxation method was used to determine static bearing loads for the system and a computer programme had been written to expedite these calculations. Antkowiak subsequently refined this method by introducing the influence number method to obtain a mathematical model of the transverse flexibility characteristics of a particular system. The new bearing loads produced by a given combination of offsets from the in-line position could then be evaluated directly using matrix algebra thus:

$$\{F\} = \{F_o\} + [C]\{\Delta\}$$

where: F = the array of bearing loads caused by offsets Δ;

F_o = the array of bearing loads when all bearings are in-line;

C = the array of influence coefficients for the system.

All geometrical aspects of the shaft system including shaft diameters, lengths, bearing spacing and positions of concentrated loads are encompassed by the C matrix. Magnitudes of the concentrated loads, shaft weight and any changes therein which do not also change the geometry of the system enter into evaluation of the F_o matrix. The new computer programme also included the classical method of successive integration from shear force to give bending moment, slope and deflexion at each point so adding greatly to the power of the analysis.

A different approach[31] was published in the U.S.S.R. at about the same time. This was a method of assessing the overall flexibility of the system in terms of a K factor where:

$$K = 0{\cdot}95 \times 10^{-6} \times \frac{s^3}{d^2}$$

and s is taken as the mean span of those three adjacent spans in the system which have the shortest total length. Both s and d are expressed in inches.

Limits on bearing span were given as not less than $\frac{78}{\sqrt{d}}$ nor more than $\frac{125}{\sqrt{d}}$ for intermediate shafting of diameter 8-20 inches. These limiting spans correspond to fundamental whirling speeds of 1280 and 304 rev/min respectively. A description is also given of successful attempts to halve the number of plummer blocks on certain existing ships.

Later, Andersen and Zrodowski[1] described the application of Antkowiak's method to large merchant ships as well as giving further examples involving U.S. naval vessels. Particular aspects of gearing misalignment were also treated in detail including the influence of changes in static loading on the main wheel bearings due to stern-tube weardown and also because of thermal lifting of the main wheel.

Availability of several computer programmes now cleared the way to large-scale analysis of transverse flexibility of shafting systems, and Lehr and Parker[19] described such an investigation on families of stylized shaft systems in order to determine general limits on desirable spacing of tunnel bearings. The conclusions reached in this study were that minimum bearing spacings should be 14 shaft diameters for shafts from 10-16 in. in diameter and 12 shaft diameters for shafts from 16-30 in. in diameter. In the discussion of this paper further interesting merchant and naval investigations were quoted by Andersen.

More recently Mann[20] has described further applications of transverse flexibility analysis under the name of Fair Curve Alignment—a title which is apt and probably stems from the Kosiba paper. The examples given[20] relate to U.S. Coastguard ships, being similar in scale and characteristics to the small twin-screw ships in Fig. 2.

Case Histories

For the last few years the authors have been engaged in similar applications of flexibility analysis to shafting fitted in both new and old construction powered by geared turbines, and geared or direct-coupled Diesels. Because the foregoing literature adequately describes the methods employed, further comment will be restricted to practical experience obtained during this period.

Shaft B is an example of single-screw turbine propulsion, having a separate main thrust block adjacent to the main wheel and with two white-metalled bearings fitted in the stern tube. Three sister ships were involved, and the investigation was not initiated until the lead-ship had been launched and the gearbox roughly positioned. The principal features of this alignment were:

1) bearing No. 8 was not slope-bored;
2) the bore of No. 7 was offset below No. 8, so enabling the full length of the latter to be used to carry load;
3) both No. 6 and No. 5 bearings were offset below the datum, the respective amounts being approximately $\frac{1}{4}$in and $\frac{1}{16}$in, thus allowing shafting weight to bend the tailshaft to contact the bottom half of No. 7 bearing applying a significant load, a condition which was greatly assisted by the relatively wide gap between the two stern-tube bushes;
4) to increase shaft flexibility near the main wheel the forward seal in the main thrust block was eased to prevent it carrying load—this was done after the full system had been coupled;
5) bearing No. 4 was retained as a journal bearing because Nos. 5 and 6 could not be re-sited closer to it; however, a smaller offset was used so that the shafting now swept up in a faired curve to meet the main wheel at its previous general level;
6) all three lengths of intermediate shafting were bolted together forming a single length supported in two bearings during installation;
7) with the propeller fitted, the forward end of the tailshaft was forced down to give a prescribed load in No. 7 bearing by using an hydraulic jack near the flange; the gap and sag method of alignment was then commenced using the tailshaft flange as reference;
8) as a final check on the fully coupled system, the loadings on bearings Nos. 5 and 6 were verified, using a double-jacking technique at each in turn.

The two sister ships which followed differed in some details from this procedure, but in all three the subsequent seagoing experience has been very favourable with a notable freedom from vibration in service and being easily turned when at anchor.

Shaft T is from an 85 000-dwt crude oil carrier and the investigation was initiated just prior to sea trials, with the result that no modifications could be implemented prior to entry into service. Reports were subsequently received of very heavy vibration at the stern, requiring a severe limitation on speed if structural damage was to be avoided. To ascertain which bearings at the aft end were actually loaded double-jacking tests were done on bearing No. 7 during the voyage preceding Guarantee drydocking, showing that this bearing itself was loaded, but bearing No. 8 was not. Conceivably the wide span between No. 7 and the effective point of support in No. 9 had lowered the whirling speed of the propeller/tailshaft system into the running range. The opportunity was therefore taken, after drydocking, to lower bearings No. 7 0·040in and No. 6, 0·015in., which unloaded No. 7 bearing and transferred load to No. 8 bearing. Service experience since this change has shown the vibration amplitude at the steering flat to be more than halved and the residual level is now clearly defined as a hull vibration probably excited by pressure forces.

Shaft R was investigated early in construction and found to be very flexible because of the wide relative spacing of the bearings. This, combined with the very light propeller, meant that no advantage was to be gained by introducing a faired curve, and the final alignment was therefore done to bring all bearings in line.

Shafts Y and Z, being driven by direct-coupled Diesel engines, introduced the problem of how to handle the flywheel which is required to improve torsional vibration characteristics. Further improvement had been obtained by using high-tensile intermediate shafting which increases the flexibility both in torsion and in transverse bending. Shaft Z posed the greatest problem with its 11-ton flywheel but, during shop trials, measurements were taken of the spread of crankshaft webs as the flywheel was mounted and removed. No significant change was found on this or later engines and shipboard alignment was therefore based upon leaving the forward end of the intermediate shaft hanging free while the engine coupling-flange was faired-up parallel and concentric to it. In this way there is no transfer of shear force or bending moment from one to the other. This principle was also used for shaft Y.

A further item of interest concerning shaft Z involves the departure from the normal practice of seeking downward loadings on all bearings. In this instance, a single-bush white-metalled stern tube was to be fitted so that the usual forward bush was replaced by a plummer block in the tunnel. The very heavy propeller, which is characteristic of the relatively slow, or blunt, ship, accentuates the tendency for this forward bearing to be either lightly loaded, or even unloaded, in service. Agreement was then obtained to a proposal to load this plummer block in its top half, dropping this bearing below the optical sight line until the elastic curve of the tailshaft matched the axis of the concentrically bored stern-tube bush to give optimum utilization of the white-metal surface. Minor difficulties were subsequently encountered on sea trials, being largely due to insufficient rigidity in the plummer block keep casting. Seagoing experience since then has been reported to be trouble free. In using this form of loading it should be noted that both adjacent bearings will be more heavily loaded than if the normal approach was used, as will be evident from considerations of static equilibrium. However, it is possible to utilize the full length of a long stern-tube bearing more easily and local concentrations are avoided.

From the behaviour of these and other ships the authors agree with the comment[5] that smooth running was obtained with every single-screw ship in which the forward stern-tube bearing was loaded. In practice, this has sometimes been done by including a separate "spring" bearing on the tailshaft as for shafts E, P and U. While past shipyard practices have been surprisingly difficult to unearth, it is safe to say that many ships completed in British and European shipyards, at least, had the tailshaft set concentric in the forward stern-tube bush. The massive weardown normally permitted on wooden and similar bearings would allow the tailshaft to drop gradually until contact with this bush did occur. In some ships this situation has been deliberately avoided by lifting the first inboard bearing to restore the clearance at this bush. Under these conditions the exact axial location of this plummer block becomes particularly important because of its immediate influence upon the whirling characteristics of the propeller/tailshaft system.

Finally, in all cases where the gap and sag method of alignment is used, two important features should be borne in mind:

a) at any particular bearing the shaft slope will change after the coupling bolts are fitted and tightened in the full system; this change will normally be small, but can sometimes be sufficient to cause heavy corner loading due to canting the shaft in the bearing; a further operation is therefore required to tilt the bearing housing by the same amount without changing its offset;

b) standard plummer blocks, whether of cast iron or cast steel, are very flexible, even when loaded in the bottom half; measurements obtained with systems so far encountered show that the shaft can sag approximately 0·006 in due to this effect when the specific loading approaches 100 lb/in^2; this becomes important when there is a considerable change in loading on a particular bearing between the uncoupled condition, when gaps and sags are checked, and the fully-coupled condition which obtains in service; so long as this flexibility is known, the movement of the bearing housing can be anticipated.

ACKNOWLEDGEMENTS

The authors have drawn freely on many papers published on aspects of marine propulsion and wherever possible the reference has been given. Much of the service experience, however, has been gained via the member firms of Pametrada, in particular Messrs. Hawthorn Leslie, and Wallsend Slipway and Engineering Co. Ltd. They are also indebted to Stone Manganese Marine for the variable wake analysis (Fig. 8), the Boston Naval Shipyard for the shaft alignment programme, the Ministry of Defence (Navy) for data on naval vessels, and to Lloyd's Register of Shipping. The authors also acknowledge the assistance received from Mrs. M. Tasker, B.Sc., with the computer calculations included in the paper.

Finally, the authors are grateful to the Director and Council of Pametrada for permission to publish the paper and for help and encouragement, without which the paper could never have been written.

REFERENCES

1) ANDERSEN, H. C., and ZRODOWSKI, J. J. 1960. "Co-ordinated Alignment of Line Shaft, Propulsion Gear and Turbines". *Trans.I.Mar.E.*, Vol. 72, p. 135.

2) ARCHER, S. 1951. "Contribution to Improved Accuracy in the Calculation and Measurement of Torsional Vibration Stresses in Marine Propeller Shafting". *Proc.I.Mech.E.*, Vol. 164, p.351.

3) ARCHER, S. "Propeller-excited Vibration; Five Blades or Four". Lloyd's Register of Shipping.

4) AUSLAENDER, J., and LEWIS, F. M. "Virtual Inertia of Propellers".

5) BAKER, E. August 1964. "Current Tailshaft Liner Problems". *Trans.I.Mar.E., Indian Division Supplement*, No. 10, p.44.

6) BREHME, H. 1954. "The Effects of the Number of Propeller Blades on Ship Vibrations". *Schiff und Hafen*, Vol. 6, p.662.

7) BURRILL, L. C., and ROBSON, W. 1962. "Virtual Mass and Moment of Inertia of Propellers". *Trans.N.E.C.I.E.S.*, Vol. 78, p.325.

8) CAMPBELL, G. T. R., and LASKEY, N. V. September 1963. "Stern Gear Design for High Powered Single-screw Tankers and Bulk Carriers". *Trans.I.Mar.E., Canadian Division Supplement*, No. 13, p.45.

9) COUCHMAN, A. A. J. 1965. "Axial Shaft Vibration in Large Turbine-powered Merchant Ships". *Trans. I.Mar.E.*, Vol. 77, p.53.

10) FELGAR, R. P. and MACDUFF, J. N. 1957. "Vibration Design Charts". *Trans.A.S.M.E.*, Vol. 79, p. 1459.

11) GOODWIN, A. J. H. 1960. "The Design of a Resonance Changer to Overcome Excessive Axial Vibration of Propeller Shafting". *Trans.I.Mar.E.*, Vol. 72, p.37.

12) JASPER, N. H. 1956. "A Design Approach to the Problem of Critical Whirling Speeds of Shaft-disk Systems". *International Shipbuilding Progress*, Vol. 3, p.37.

13) JASPER, N. H. 1956. "A Theoretical Approach to the Problem of Critical Whirling Speeds of Shaft-disk Systems". *International Shipbuilding Progress*, Vol. 3, p.183.

14) JOHNSON, A. J., and MCCLIMONT, W. 1963. "Machinery Induced Vibrations". *Trans.I.Mar.E.*, Vol. 75, p.121.

15) VAN MANEN, J. D., and KAMPS, J. 1959. "The Effect of Shape of the Afterbody on Propulsion". *Trans. S.N.A.M.E.*, Vol. 67, p.253.

16) KANE, J. R., and MCGOLDRICK, R. T. 1949. "Longitudinal Vibrations of Marine Propulsion-shafting Systems". *Trans.S.N.A.M.E.*, Vol. 57, p.193.
17) KOSIBA, R. E. *et al.* 1956. "The Alignment of Main Propulsion Shaft Bearings in Ships". Paper read to the New England Section of S.N.A.M.E.
18) KUMAI *et al.* 1961. "Measurement of Propeller Forces Exciting Hull Vibration by Use of Self-propelled Model". *Reports of Research Institute for Applied Mechanics*, Vol. IX, No. 53.
19) LEHR, W. E., and PARKER, E. L. 1961. "Considerations in the Design of Marine Propulsion Systems". *Trans. S.N.A.M.E.*, Vol. 69, p.555.
20) MANN, G. 1965. "Analysis of Shafting Problems Using Fair Curve Alignment Theory". *Naval Engineers Journal*, Vol. 77, No. 1.
21) O'BRIEN, T. P. "Propellers". *International Marine Design and Equipment.*
22) PANAGOPULOS, E. P. 1950. "Design-stage Calculations of Torsional, Axial and Lateral Vibrations of Marine Shafting". *Trans.S.N.A.M.E.*, Vol. 58, p.329.
23) YELLOWLEY, G., and RICHARDS, J. E. 1954. "Marine Shafting Alignment". *Trans.N.E.C.I.E.S.*, Vol. 70, p.343.
24) RIGBY, C. P. 1948-49. "Longitudinal Vibration of Marine Propeller Shafting". *Trans.I.Mar.E.*, Vol. 60, p.67.
25) YATES, H. G. 1949. "Vibration Diagnosis in Marine Geared Tubines". *Trans.N.E.C.I.E.S.*, Vol. 65, p.225.
26) YATES, H. G. 1955. "Prediction and Measurement of Vibration in Marine Geared-shaft Systems". *Proc. I.Mech.E.*, Vol. 169, p. 611.
27) *Motor Ship,* August 1946.
28) ARCHER, S. 1949. "Screw Shaft Casualties—Torsional Vibration and Propeller Immersion". *Trans.I.N.A.*, Vol. 91, p.J56.
29) GATEWOOD, A. R. 1950. "Some Notes on Propeller Shaft Failures". *Trans.S.N.A.M.E.*, Vol. 58, p.753.
30) VAN MANEN, J. D. and WERELDSMA, R. 1960. "Propeller-excited Vibratory Forces in the Shaft of a Single-screw Tanker". *International Shipbuilding Progress,* Vol. 7, p.371.
31) YATSENKO, V. June 1957. "The Problem of Flexibility for Marine Shaft Lines". *Morskoi Flot,* Vol. 17, No. 6, p. 21. October 1962. "Measurement of Forces acting on Line Shaft Bearings in a Ship in Service". *Morskoi Flot,* Vol. 22, No. 10, p. 25.

APPENDIX

TWO ALGOL COMPUTER PROGRAMMES TO DETERMINE THE AXIAL AND/OR TORSIONAL CHARACTERISTICS OF A TURBINE-GEARED SHAFT SYSTEM

Two programmes written in Algol for the KDF/9 computer have been used in the axial and torsional vibration investigations described in this paper. These programmes are essentially two linear electrical network programmes, one of which determines the resonant frequencies of the circuit and the other evaluates the current and voltage drop in any element of the circuit.

The mechanical system is converted into the equivalent electrical circuit using the well-established electrical analogy method in which the electrical analogue of velocity is current (see for example reference 26). The complete circuit is then made up of a voltage generator, *e*, of negligible low internal impedance connected to any combination of basic L,C,R circuits as indicated in Fig. 23. These basic circuits have been found sufficient to describe all geared-shaft systems so far considered, e.g. a single mass can be represented by a basic parallel L,C,R, circuit with C = 0.

Frequency Programme

This programme determines the resonant frequencies (impedance zero) and anti-resonant frequencies (impedance infinite) of the equivalent electrical network. Damping is ignored (i.e., R = 0 for all basic circuits) since the amount of damping in most mechanical systems has very little effect on the resonant frequencies. This simplifies the calculation which can then be carried out in real arithmetic. The search begins at a prescribed frequency, P_0, and continues at fixed increments, P_1, in frequency until a change in sign of the circuit impedance is detected. Iterations are then carried out until the frequency corresponding to the zero or infinity has been determined to the required accuracy, P_2. The frequency is then identified and printed out in radians/second and vibrations/minute.

By using impedance and admittance ratios the calculation is non-dimensional and any consistent system of units may be used, e.g.:

	W tons	F in/ton	g in/s^2
or	J ft lb	F rad/lb ft	g ft/s^2.

If mass units are used for the inductance elements then the gravity constant is put equal to unity.

Table VII gives the layout of the data tape. It is headed by a title punched between two colons. The gravity constant g and scale factor, c, for the capacitance elements follow the title. k_1 is the number of basic L,C circuits which make up the circuit for the complete system. L_0 is the reference (propeller) mass or inertia and is followed by k_1 sets of T, L and C where T identifies the type of basic circuit according to Fig. 23 and L and C are the inductance and capacitance making up the basic circuit. The basic circuits are listed in the order in

TABLE VII—FREQUENCY PROGRAMME DATA TAPE LAYOUT

:Title:

g	=	Gravity constant (=1 if mass units used);
c	=	Scale factor for capacitance elements;
k_1	=	No. of basic L, C circuits;
L_0	=	Reference (propeller) mass or inertia;
T	=	Type of basic circuit T = 1 series T = 2 parallel T = 3 "tee" T = 4(o) multiple "tee" — k_1 sets of elements;
L	=	Elements of basis circuit
C	=	
P_0	=	Starting frequency;
P_1	=	Increment in frequency for initial search;
P_2	=	Accuracy to which frequency is required;
m	=	Maximum number of iterations before search is abandoned;
K	=	Programme control integer: K=1 New data accepted beginning with g; K=2 New data accepted beginning with h_2 i.e. number of first basic circuit to be changed and k_2 number of last basic circuit to be changed followed by new values of T, L, C etc.; K=3 New data accepted beginning with P_0; K=5 End of programme.

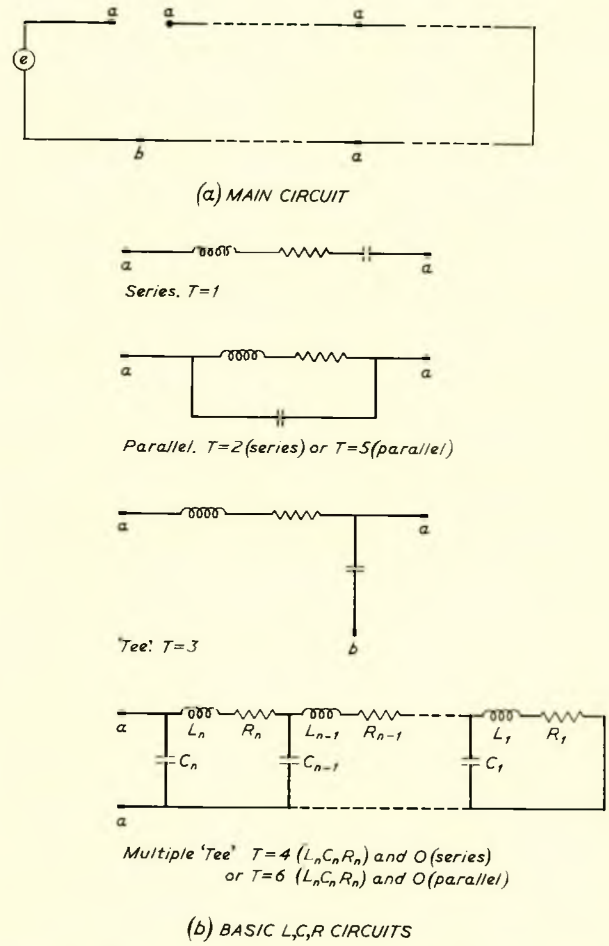


Fig. 23—*Types of electrical circuits which can be solved by general network programmes*

which they occur in the complete circuit and in the case of multiple "tee" type circuits the elements are listed in the order T_n L_n C_n, T_{n-1} L_{n-1} C_{n-1} etc., with $T_n = 4$ and all other T_{n-1} etc. = 0. The frequency data P_0, P_1 and P_2 in radians/second come next followed by m, the maximum number of iterations to be carried out before the search for the frequency is abandoned. The data is completed by a programme control integer, K, which determines the subsequent action of the programme.

The complete programme as written for the KDF9 computer is as follows:

```
GENERAL NETWORK 1→
begin library AO, A6, A14;
open (20); open (30);
write text (30, [[p]DJOPO2*GENERAL*NETWORK***
RESONANT*FREQUENCIES[cc]]);
begin real g, LO, PO, P1, P2, A1, A2, A3, B1, B2, B3, c;
    integer h1, h2, k1, k2, k3, m, f, K, i, j;
    integer array T[1:100];
    real array L, C[1:100];
    switch LABEL: =L1, L2, L3, L4, L5, L6, L7;
procedure IMPEDANCE (x, y);
    value x;
    real x, y;
    begin real z, a, S1, S2;
        integer i;
        y: =z: =0;
        a: =x×x;
        for i: =k1 step −1 until 1 do
        begin
            if T[i]=0 then z: =1/(1/(z+L[i])−a×C[i])
            else
            if T[i]=4 then
            begin
                y: =y+1/(1/(z+L[i])−a×C[i]);
                z: =0;
            end else
            if T[i]=1 then y: =y+L[i]−1/(a×C[i]) else
            if T[i]=2 then y: =y+1/(1/L[i]−a×C[i]) else
            if T[i]=3 then y: =L[i]+1/(1/y−a×C[i]) else
        end;
        y: =y+1;
    end;
    f: =format ([sssndddd.dd]);
L1: copy text (20, 30, [::]);
    h1: =1;
    g: =read (20);
    c: =read (20);
    k1: =read (20);
    L0: =read (20)×g;
    if L0<0 then L0: =read (20)×0·0481×g−L0;
    h2: =h1;
    k2: =k1;
L2: for i: =h2 step 1 until k2 do
    begin
        T[i]: =read (20);
        L[i]: =read (20)×g/L0;
        C[i]: =read (20)×L0×c;
    end;
L3: P0: =read (20);
L6: P1: =read (20);
    P2: =read (20);
    m: =read (20);
    k3: =0;
K4: A1: =0;
    write text (30, [*FREQUENCY******RESIDUE[css]
                    RAD/SEC******IMPEDANCE[c]]);
    for j: =0 step 1 until m do
    begin
        A2: =(P0+P1×j);
        IMPEDANCE (A2, B2);
        write (30, format ([ndddd.dddd]), A2);
        write (30, format ([ssss+d.dddd10+ndc]), B2);
        if A1=0 or B1×B2>0 then
        begin
            B1: =B2;
            A1: =A2;
        end
        else
        begin
            B3: =B2;
            A3: =A2;
            goto K1;
        end
    end;
    write text (30, [[c]NO*ZERO*FOUND*AFTER*
        MAXIMUM*NUMBER*OF*ITERATIONS[cccc]]);
    goto L4;
K1: A2: =(A1+A3)/2;
    IMPEDANCE (A2, B2);
    write (30, format ([ndddd.dddd]), A2);
    write (30, format ([ssss+d.dddd10+ndc]), B2);
    if A3−A1<P2 then goto K2;
    if B1×B2<0 then goto K3;
    A1: =A2;
    B1: =B2;
    goto K1;
K3: A3: =A2;
    B3: =B2;
    goto K1;
```

```
K2: if (B1×B2<0 and abs (B2)>abs (B3)) or
    (B1×B2>0 and abs (B2)>abs (B1)) then
    begin
        write text (30, [[c]ANTI-RESONANT*
                                  FREQUENCY]);
        k3: =k3+1;
        if k3=6 then K: =4 else K: =7;
    end
    else
    begin
        write text (30, [[c]RESONANT*FREQUENCY
                                            [5s]]);
        K: =4;
    end;
    write (30, f, A2);
    write text (30, [***RADIANS/SEC]);
    write (30, f, A2×9·549297);
    write text (30, [***VIBRATIONS/MIN[cccc]]);
    goto LABEL[K];
L4: K: =read (20);
    if K=2 then
    begin
        h2: =read (20);
        k2: =read (20);
    end;
    if K=6 then P0: =A2+10×P2;
    if K=1 then write text (30, [[p]]);
    goto LABEL[K];
L7: P0: =A2+10×P2;
    goto K4;
L5: close (20);
    close (30);
end;
end→
```

Note: All basic Algol words should be underlined.

Amplitude Programme

This programme determines the amplitude and vibratory thrust or torque in selected elements of the system for a prescribed range of frequencies. Provision has been made to deal with both propeller-excited and pinion-excited vibrations and for the exciting thrust or torque to be constant, or proportional to the square of the frequency.

Table VIII gives the layout of the data tape, which is similar in form to the data for the frequency programme but with additional data included. For propeller-excited vibrations the steady thrust (tons) or torque (tons inches) T_0 at N_0 rev/min is included together with the excitation fraction, E, i.e., the fraction of steady thrust or torque exciting the vibration. In the case of pinion-excited vibrations the pinion mean radius (inches) gear ratio and pinion eccentricity, E_{cc} (inches) are given.

Two constants, N_1 and N_5 follow, defining the exciting frequency, p (rad/s), in terms of the mainshaft speed, N (rev/min), i.e., $p = N_5 + 0{\cdot}10472\, N_1 \times N$ where (rad/s) = 0·10472 (rev/min). Thus for propeller excitation $N_5 = 0$ and N_1 is the number of propeller blades.

The programme allows two of the basic L, C, R circuits (parallel and multiple "tee") to be placed in parallel as well as in series in the main circuit (e.g. T = 5 or 6 in Fig. 23). This was found necessary to construct the equivalent circuit for pinion-excited vibrations.

The basic L, C, R circuits are numbered according to their order in the complete circuit and are listed in this order in the data layout. P(i) then is the number of a given basic circuit for which amplitudes etc., are required. The range of main-shaft rev/min for which these are required is defined by N_2, N_3 and N_4 such that:

N_2 is rev/min at beginning of range;
N_3 is rev/min interval between points;
N_4 is rev/min at end of range.

Table VIII—amplitude programme data tape layout

:Title:		
h_4	=	$h_4=1$ axial, $h_4=2$ torsional;
h_5	=	$h_5=1$ propeller excited, $h_5=2$ pinion-excited;
h_6	=	$h_6=1$ excitation constant, $h_6=2$ excitation proportional to $(\text{frequency})^2$;
g	=	Gravity constant (=1 if mass units used);
c	=	Scale factor for capacitance elements;
k_1	=	No. of basic **L**, **C**, **R** circuits;
L_0	=	Propeller mass or inertia;
R_0	=	Propeller damping factor;
if $h_5=1$,		i.e., propeller-excited vibrations:
T_0	=	Exciting thrust (tons) or torque (tons inches) at N_0 rev/min;
N_0	=	rev/min;
E	=	Excitation fraction;
if $h_5=2$,		i.e., pinion-excited vibrations:
PMR	=	Pinion mean radius (in);
GR	=	Gear ratio;
E_{cc}	=	Eccentricity (in);
N_1		Constants defining excitation frequency
N_5		as function of rev/min, i.e. $p = N_5 + 0{\cdot}10472\ N_1 \times N$;
T		
L		k_1 sets of data describing basic
C		LCR circuits (see Fig. 23);
R		
h_3	=	Number of basic LCR circuits for which amplitudes etc. are required;
P(i)	=	i=1, 2, . . . h_3, number of each basic LCR circuit for which amplitudes etc. are required;
N2	=	Main shaft rev/min at beginning of range;
N3	=	Interval of rev/min between points;
N4	=	Main shaft rev/min at end of range;
K	=	Programme control integer
		K=1 New data accepted beginning with :Title:;
		K=2 New data accepted beginning with h_2 i.e. number of first basic circuit to be changed and k_2 number of last basic circuit to be changed followed by new values of T, L, C, R etc.;
		K=3 New data accepted beginning with N2 i.e. new speed range;
		K=5 Same as K=1 except results begin on new page;
		K=6 End of programme.

At the end of the data is the programme control integer, K, which determines the subsequent course of the programme.

For each point in the speed range, the vibration frequency is output in radians/second and revolutions/minute together with the exciting thrust (tons) or torque (tons inches). For the propeller and each selected basic circuit the amplitude (inches or radians) and thrust (tons) of torque (tons inches) absorbed by the mass and in the corresponding shafting is printed out.

The complete programme as written for the KDF9 computer is as follows:

```
GENERAL NETWORK 2→
begin library A0, A6, A14;
open (20); open (30);
write text (30, [[p]DOPO2*GENERAL*NETWORK[cc]
        VIBRATION*AMPLITUDES*AND*VIBRATORY
        THRUSTS*AND*TORQUES*IN*A*GEARED*
        SHAFT*SYSTEM[cc]]);
begin    real S0, S1, S2, S4, g, c, T0, N0, E, N1, L0, R0, I0,
              I1, I2, N2, N3, N4, N5, PMR, GR, Ecc;
    integer h1, h2, h3, h4, h5, h6, k1, k2, k3, i, j, K, f1, f2, f3;
    integer array T, t, P[0:100];
    real array L, C, R, D, Z, ZI, ZR, ZP, ZPI, ZPR, GPI,
                                                GPR[0:100];
    switch LABEL: =L1, L2, L3, L4, L5, L6;
procedure COMPIMP (x, y);
    value x;
    real x, y;
```

```
begin real a;
    integer i, k, n;
    k: =0;
ZR[k1+1]: =ZI[k1+1]: =0;
    for i: =k1 step -1 until 1 do
    begin
        D[i]: =C[i]×x↑2;
        if T[i]=1 then
        begin
            ZR[i]: =L[i]-1/D[i]+ZR[i+1];
            ZI[i]: =ZI[i+1]-R[i];
        end else
        if T[i]=2 then
        begin
            a: =1/((L[i]-D[i])↑2+R[i]↑2);
            ZR[i]: =(L[i]-D[i])×a+ZR[i+1];
            ZI[i]: =ZI[i+1]-R[i]×a;
        end else
        if T[i]=3 then
        begin
            a: =1/(ZR[i+1]↑2+ZI[i+1]↑2);
            GPR[i]: =ZR[i+1]×a-D[i];
            GPI[i]: =-ZI[i+1]×a;
            a: =1/(GPR[i]↑2+GPI[i]↑2);
            ZP[i]: =sqrt(a);
            ZR[i]: =GPR[i]×a+L[i];
            ZI[i]: =-GPI[i]×a-R[i];
        end else
        if T[i]=5 then
        begin
            a: =1/(ZR[i+1]↑2+ZI[i+1]↑2);
            GPR[i]: =ZR[i+1]×a+L[i]-D[i];
            GPI[i]: =-ZI[i+1]×a+R[i];
            a: =1/(GPR[i]↑2+GPI[i]↑2);
            ZP[i]: =sqrt(a);
            ZR[i]: =GPR[i]×a;
            ZI[i]: =-GPI[i]×a;
        end else
        if T[i]=0 and k=0 then
        begin
            k: =1;
            ZPI[i+1]: =ZPR[i+1]: =0;
            n: =i+1;
        end;
        if T[i]=4 or T[i]=0 then
        begin
            k: =k+1;
            a: =(ZPR[i+1]+L[i])↑2+(ZPI[i+1]
                                        -R[i])↑2;
            ZP[i]: =sqrt(a);
            GPR[i]: =(ZPR[i+1]+L[i])/a-D[i];
            GPI[i]: =-(ZPI[i+1]-R[i])/a;
            a: =GPR[i]↑2+GPI[i]↑2;
            ZPR[i]: =GPR[i]/a;
            ZPI[i]: =-GPI[i]/a;
            Z[i]: =1/sqrt(a);
        end;
        if T[i]=4 then
        begin
            t[i]: =k-1;
            k: =0;
            ZR[i]: =ZR[n]+ZPR[i];
            ZI[i]: =ZI[n]+ZPI[i];
        end;
        if T[i]=6 then
        begin
            t[i]: =k;
            k: =0;
            a: =1/(ZR[n]↑2+ZI[n]↑2);
            GPR[i]: =ZR[n]×a+L[i]-D[i]
                                        +GPR[i+1];
            GPI[i]: =-ZI[n]×a+R[i]+GPI[i+1];
            a: =1/(GPR[i]↑2+GPI[i]↑2);
            ZP[i]: =sqrt(a);
            ZR[i]: =GPR[i]×a;
            ZI[i]: =-GPI[i]×a;
        end;
    end;
    if h5=2 then y: =sqrt(ZR[1]↑2+ZI[1]↑2)
    else y: =sqrt((ZR[1]+1)↑2+(ZI[1]-1/R0)↑2);
end COMPIMP;

procedure CANDV(n);
    integer n;
    begin
        for i: =1 step j until n do
        begin
            j: =1;
            if T[i]=1 then
            begin
                if P[k3]≠i then goto REPEAT;
                I1: =I2: =I0;
                goto X1;
            end;
            if T[i]=2 then
            begin
                if P[k3]≠i then goto REPEAT;
                Z[i]: =1/sqrt((L[i]-D[i])↑2+R[i]↑2);
                I2: =I0×Z[i]×D[i];
                I1: =I0×Z[i]×sqrt(L[i]↑2+R[i]↑2);
                goto X1;
            end;
            if T[i]=3 then
            begin
                I1: =I0;
                I2: =I0×ZP[i];
                I0: =I2/sqrt(ZR[i+1]↑2+ZI[i+1]↑2);
                I2: =I2×D[i];
                if P[k3]≠i then goto REPEAT;
                goto X1;
            end;
            if T[i]=4 then
            begin
                if P[k3]≥i+t[i] then
                begin
                    j: =t[i];
                    goto REPEAT;
                end;
                I1: =I0×Z[i]/ZP[i];
                I2: =I0×Z[i]×D[i];
                if P[k3]≠i then goto REPEAT;
                goto X1;
            end;
            if T[i]=0 then
            begin
                I2: =I1×Z[i]×D[i];
                I1: =I1×Z[i]/ZP[i];
                if P[k3]≠i then goto REPEAT;
                goto X1;
            end;
            if T[i]=5 then
            begin
                I2: =I0×ZP[i];
                I0: =I2/sqrt(ZR[i+1]↑2+ZI[i+1]↑2);
                I1: =I2×sqrt(L[i]↑2+R[i]↑2);
                I2: =I2×D[i];
                if P[k3]≠i then goto REPEAT;
                goto X1;
            end;
            if T[i]=6 then
            begin
                I2: =I0×ZP[i];
                I0: =I2/sqrt(ZR[i+t[i]]↑2+
                                        ZI[i+t[i]]↑2);
                I1: =I2×sqrt((L[i]+GPR[i+1])↑2+
                                (R[i]+GPI[i+1])↑2);
                I2: =I2×D[i];
                if P[k3]≥i+t[i] then
```

```
                begin
                    j: =t[i];
                    goto REPEAT;
                end;
                if P[k3]≠i then goto REPEAT;
                goto X1;
            end;
X1:         write text (30, [[33s]]);
            write (30, format ([6snd]), i);
            write (30, f2, I1/S2);
            write text (30, [[8s]]);
            write (30, f1, if T[i]=2 or T[i]=5 or T[i]=6
                then I1×S4/sqrt(L[i]↑2+R[i]↑2)
                else I1×S4×sqrt(L[i]↑2+R[i]↑2));
            write (30, f3, if D[i]=0 then 0 else I2×S4/D[i]);
            k3: =k3+1;
            if k3>h3 then goto X2;
REPEAT:     end;
X2: end CANDV;
    f1: =format ([nddd.ddssss]);
    f2: =format ([6sd.ddddd]);
    f3: =format ([6snddd.ddc]);
L1: copy text (20, 30, [::]);
    h1: =1;
    h4: =read (20);
    h5: =read (20);
    h6: =read (20);
    g: =read (20);
    c: =read (20);
    k1: =read (20);
    L0: =read (20)×g;
    R0: =read (20);
    if L0<0 then L0: =read (20)×0·0481×g−L0;
    h2: =h1;
    k2: =k1;
    if h5=1 then
    begin
        T0: =read (20);
        N0: =read (20);
        E: =read (20);
        N1: =read (20);
        N5: =read (20);
        S0: =E×T0×2240/L0;
        end else if h5=2 then
        begin
        PMR: =read (20);
        GR: =read (20);
        Ecc: =read (20);
        N1: =read (20);
        N5: =read (20);
        S0: =Ecc/(GR×PMR);
    end;
    if h5=1 and h6=2 then S0: =S0/(0·1047198×
                                        N0×N1)↑2;
L2: for i: =h2 step 1 until k2 do
    begin
        T[i]: =read (20);
        L[i]: =read (20)×g/L0;
        C[i]: =read (20)×L0×c;
        R[i]: =read (20);
        if T[i]=2 or T[i]=5 or T[i]=6 then
        L[i]: =R[i]×R[i]/(L[i]×(1+R[i]×R[i]));
        R[i]: =L[i]/R[i];
    end;
    h3: =read (20);
    for i: =1 step 1 until h3 do P[i]: =read (20);
    if h4=1 then
    write text (30, [[ccc22s] EXCITING[35s]THRUST*
        VARIATION [c4s] FREQUENCY [7s] THRUST
        (TONS)****POSITION**AMPLITUDE [15s]
        (TONS) [c] RAD/SEC***R.P.M. [31s] INCHES
        [12s] MASS [10s] SHAFTING [c]])
    else if h4=2 then
    write text (30, [[ccc22s]EXCITING[35s]TORQUE*
        VARIATION [c4s] FREQUENCY [6s] TORQUE
        (TONS.IN)**POSITION**AMPLITUDE [14s]
        (TONS.IN) [c] RAD/SEC***R.P.M. [31s]
        RADIANS [9s] INERTIA [9s] MASS [10s]
        SHAFTING [c]])
L3: N2: =read (20);
    N3: =read (20);
    N4: =read (20);
    for S1: =N2 step N3 until N4 do
    begin
        S2: =N5+S1×N1×0·1047198;
        S4: =S2×L0/2240;
        COMPIMP (S2, Z[0]);
        write text (30, [[c]]);
        write (30, f1, S2);
        write (30, f1, S1);
        if h6=2 then I0: =S0×S2 else if h6=1 then
                                        I0: =S0/S2;
        write (30, f1, if h5=2 then I0×S4×Z[0] else I0×S4);
        if h5=1 then
        begin
            I0: =I0/Z[0];
            write text (30, [[7s]0]);
            write (30, f2, I0/S2);
            write text (30, [[8s]]);
            write (30, f1, I0×S4×sqrt(1+R0×R0/R0);
        end;
        write text (30, [[c]]);
        if h3=0 then goto L4;
        k3: =1;
        j: =1;
        CANDV(k1);
    end;
L4: K: =read (20);
    if K=2 then
    begin
        h2: =read (20);
        k2: =read (20);
    end;
    goto LABEL[K];
L5: write text (30, [[p]DJOP02*GENERAL*NETWORK
        [cc]VIBRATION*AMPLITUDES*AND*
        VIBRATORY*THRUSTS*AND*TORQUES*IN*
        A*GEARED*SHAFT*SYSTEM[cc]]);
    goto L1;
L6: close (20);
    close (30);
    end;
end→
```

Note: All basic Algol words should be underlined.

Discussion

MR. W. MCCLIMONT, B.Sc. (Member) said that problems involving propulsion shafting systems had undoubtedly been among the most troublesome of recent years and a paper ranging over the many considerations which arose in the transmission of torque from the prime mover to thrust at the screw was particularly welcome. The authors were to be congratulated on a well-presented picture covering all aspects of a complex subject; or nearly all aspects, since one general criticism was that the paper was understandably biased towards turbine installations and some of the considerations which arose in motor-ship shafting systems had been somewhat scantily treated or overlooked.

The authors had been comprehensive in their studies of published work and generous in their list of references. That was, however, a little inhibiting when attempting to offer critical comment, as either one's colleagues or oneself were quoted in support of so many of the statements. If Mr. McClimont contradicted anything he might have said at the Institute before, he pleaded as justification more recent knowledge.

The opening paragraphs of the paper were indeed gloomy and, while nobody would wish to minimize the serious nature of transmission problems, the speaker pointed out that those difficulties were not nearly so prevalent as suggested. That was true even of tailshaft replacement. The paper concentrated on crack detection at an early stage as a significant step forward, but the speaker submitted that more attention to adequate pull-up of propellers on their cones and vastly improved sealing arrangements had reduced the incidence of cracking and that was surely a more laudable achievement than earlier detection?

Turning to the section of the paper which was a survey of existing shaft systems, Mr. McClimont said that reference was quite correctly made to the use of the intermediate shafting diameter as the basis for fixing the diameter of all other shaft lengths in the system. The authors then went on to quote Lloyd's Register Rules and to observe that the constant C had been reduced over the years, as illustrated in Fig. 4 of the paper. They had omitted to mention, however, that when C was reduced, the relationship was also changed, the net effect being to maintain, in general, the required diameter of screwshaft. From some of the submissions received, it would appear that that point had not been appreciated by everyone.

Mr. McClimont suggested that hollow shafting had more future than the authors' observations would indicate.

In the second section, dealing with the propeller as source of excitation, reference was made to axial vibration of a dry cargo ship with which the speaker was concerned and particular stress was placed on the weight of the propeller. That was puzzling as the effect of the mass of the propeller would be confined essentially to lowering the frequency of resonance and that by some four to five per cent, or approximately 6 rev/min. The propeller was of an outdated type, built-up, with in fact an oversized boss 5 ft 8 in diameter, on a 19 ft 6 in diameter propeller, and it had abnormally bad cyclic thrust variation characteristics. By some estimates it might have been as high as plus or minus 25 per cent of the mean thrust. The solid propeller by which it was replaced, with markedly beneficial effect, had additional skewback on the blades which would have the effect of creating a smoother transition of each blade into the varying wake conditions encountered during each revolution; in addition, modifications to the pitch distribution were designed to unload the tips of the blades and hence reduce the thrust variations on those parts of the blades subjected to the greatest variation of wake.

Fig. 8 was interesting, but its value was quite limited due to the data on which it was based. It was stated that wake surveys had been used which were representative of the respective hull forms; presumably derived from model testing. Mr. McClimont was dubious whether model tests would prove much help in predicting full-scale cyclic fluctuations which he suggested would prove to be of lesser magnitude than predicted by models.

Two statements in that section worried him. The frequency of the pressure forces on the hull was stated to be of the same order as the natural frequencies of the hull when considered as a free-free beam, but that was virtually meaningless since no reference was made to the modes involved. Excepting certain small vessels, the lower modes of hull vibration lay below the frequency of the pressure forces which occurred, of course, at propeller blade frequency. Hence, in considering propeller-excited vibration they were fairly high in the frequency spectrum, where the hull response characteristics were distinguished by the disappearance of real nodal positions and concurrently the disappearance of clearly-defined resonances, with amplitudes increasing at the aft end and dying out at the forward end. In considering hull vibration, therefore, the selection of a number of propeller blades should be concerned more with the magnitude of the pressure forces than with their frequency.

The other troublesome statement was the reference to the effective point of support in the aft stern-tube bush. Assumptions were often made as to where that was located, but not normally, he felt, with such assurance. The effect of alignment was too significant.

The electrical analogue method to which the authors referred in Section 3 was one for which they had long expressed partiality which the speaker had not always shared, although it interested him when it was expounded by Manley in 1941. However, the speaker was a little nearer to having become a convert since the availability of computer programming had enabled the method to develop its full potential. It was unfortunate, however, that in presenting the two computer programmes as appendices, the electrical equivalents of the mechanical system were not set out. There was admittedly a reference given and some mention of equivalence appeared on page 185 of the paper, but few mechanical engineers thought freely in electrical terms and they required all possible assistance.

Couchman did indeed find that he got his best correlations when he assumed sliding of flexible couplings as designed, but the speaker's interpretation of that remained that sliding only occurred under the high acceleration conditions approaching resonance and that, over most of the speed range, the flexible couplings did not isolate the turbines and primary gears from the axial shuttling of the main shaft system. Indeed, Couchman's acceptability criteria might well be devised to ensure that the couplings did not slide. Frankly, Mr. McClimont had always regarded flexible couplings as alignment devices and not axial vibration absorbers, a role for which he felt they were particularly poorly suited.

The absolute value of the plus or minus 30 tons suggested by Couchman as an upper limit for thrust variation had always appeared to him to be illogical. For the design study of the twin-screw passenger liner (shaft G) it represented something of the order of plus or minus 25 per cent of steady thrust at resonance with a four-bladed propeller and that appeared unduly restrictive, particularly when one also considered that the maximum instantaneous thrust involved would only be a shade over 70 per cent of the maximum designed steady thrust of the thrust bearing.

Mr. McClimont said that so far he had touched on three sections of the paper only and, even in those, he had omitted many observations that he would have liked to have made. He felt he dared not weary the meeting with more on the later sections and would confine himself to noting that in considering tailshaft whirl the authors' had rightly concentrated on Jasper in preference to Panagopulos and Nickerson, but had wisely indicated the very limited accuracy. Recently, his colleague, Mr. A. E. Toms, had produced what Mr. McClimont considered a more rational expression, for which he had prepared a computer programme; it had given promising initial results. Now they were looking for further cases of recorded tailshaft whirl on which to try it out. It was sometimes difficult to decide whether it was fortunate or unfortunate that they were rather rare—one experienced a certain conflict of interests.

The authors had done a useful service in bringing, within the space of 28 pages, so much information concerning the published studies of the past 20 years into the phenomena associated with marine propulsion shafting systems, but one could not but feel disappointed that they had not been more critical in their approach and expressed more of the views which they and their organization must have had on the relative values of other workers' expositions. One looked, too, with an equal sense of disappointment for data on many of those problems which one would have expected to have been obtained by their organization in the course of the last 22 years, much of which one would have felt to be of a nature that could have been made public.

Mr. P. W. R. Blake congratulated the authors on the extremely clear presentation of the paper and the fund of practical as well as theoretical detail.

Regarding the damping effect of split roller bearings, Mr. Blake's company had recently had an interesting experience with two 30 000-ton dry cargo vessels. These had single four-bladed propellers as built and excessive vibration was experienced in the stern of each ship. The shaft system consisted of oil-lubricated white-metal bearings. Recommendations were made to change to a five-bladed propeller, but that had the effect of transferring the vibrations from the structure to the shaft. The maximum amplitude of the vibrations was 1 mm or approximately 0·040 in.

A close-clearance Cooper split bearing was assembled in the aftermost position at the anti-node of that vibration which was then reduced to 0·15 mm or about 0·006 in. So far these vessels had been running for over 15 months without trouble.

There was the authors' reference to false-Brinelling arising from the vibration of auxiliaries when the main shaft was stationary. From the experience of his company, it could be said that false-Brinelling, even if it did occur, and they had no evidence of that, did not seem to cause any shortening of life. Presumably it would be the unloaded rollers which would vibrate and that would affect the top half of the outer race, but it was usually the bottom half, or loaded area, of the outer race which suffered from rolling fatigue first.

Mr. Blake was most interested in the comments about the close spacing of bearings and fully agreed with the authors' remarks that most merchant vessels used far too many bearings. This was odd from an economic point of view as well as the fact outlined by Lehr and Parker (reference (19) in the paper) that a setting error of plus or minus 0·010 in might produce a variation in load on the bearing of up to 50 per cent at short centres, but at the minimum ratios given by Lehr and Parker, only about 10 per cent variation might be expected. He said that his company had experienced specific cases where it had been necessary, for the correct loading of the bearings, to remove at least one plummer block. They made a plea for practical measurements of the loads on the aftermost and one but aftermost bearings on various classes of ship. This could be done by means of load cells mounted beneath the bearing pedestals. They had made an approach to B.S.R.A. a few years ago, but unfortunately the researches needed did not fall into any of the association's programmes at the time. In really rough seas the loading could be quite high.

Mr. A. R. Hinson (Associate Member) said that at the foot of page 182 of the paper it was stated: "each blade has been replaced by a radial lifting line or vortex, so that, neither the influence of skewback nor the averaging effect of a finite chordal blade width has been taken into account. Also, wake surveys have been used which are representative of the respective hull forms".

To neglect skewback was to neglect a factor which influenced the rate at which propeller loading changed as the blades passed through a region of high wake variation. If the blade had a large amount of skewback, it sliced through the region of high wake variation progressively from the root to the tip and thus the shock, and hence the propeller excitation, was reduced. To neglect skewback would tend to increase the theoretical excitation.

To neglect the averaging effect of a finite chordal blade width would also tend to increase the theoretical excitation and the answer again became pessimistic.

Could the authors state the approximate percentage total increase in theoretical excitation due to these two assumptions?

With regard to the utilization of wake surveys which were representative of hull form, Mr. Hinson would be grateful for a definition of the term "hull form" when used in that context. By "hull form" did the authors mean block coefficient only, or had they taken into account the shape of the afterbody? Stuntz *et al** gave some indication of the complexity of the problem of defining "representative wake survey" for any particular hull form. It seemed doubtful whether it was possible to say anything more than, "the finer the ship, the less variation in the wake component".

The tailshaft designer had no more than a rough idea of the magnitude of the forces against which he was designing. The authors' calculations for Fig. 8 could do much to remedy that and Mr. Hinson asked whether they could publish in their reply, a typical example of their calculation.

It was unfortunate that Table II had been so computed that it presented the five-bladed propeller in so favourable a light. If there was a villain of the piece it was the five-bladed propeller, especially when fitted to a tailshaft running in a lignum vitae stern tube. That did not mean that Table II was not generally correct, but it represented a half-truth. The omission of any reference to bending moment could be misleading.

It could be even more misleading if it was used in association with the formula for the critical speed of tailshaft whirl given on page 197 of the paper.

That formula and similar formulae for predicting tailshaft whirl were of little use when dealing with five-bladed propellers. This was due to the large variation in the bending moment which could occur with five-bladed propellers and which could cause forced tailshaft whirl. Forced tailshaft whirl occurred when the variation in the bending moment was

*Stuntz, G. R. *et al.* 1960. "Series 60—The Effect of Variations in Afterbody Shape Upon Resistance, Power, Wake Distribution and Propeller Excited Vibratory Forces". *Trans. S.N.A.M.E.*, Vol. 68, p. 292.

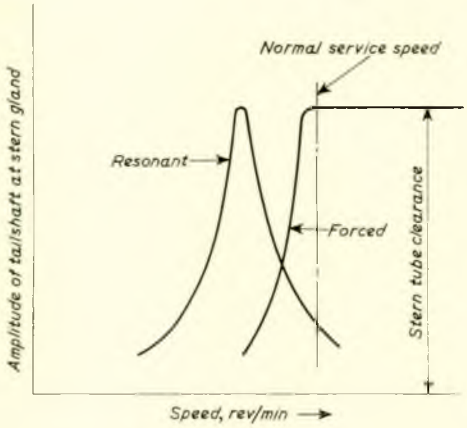


FIG. 24

large enough to lift the propeller and shaft clear of the aft end of the stern tube each time a blade passed the stern frame.

Resonant tailshaft whirl and forced tailshaft whirl had different types of curves (see Fig. 24), but in an existing ship, if the peak of a curve occurred at full speed, the important question as to whether or not the vibration was resonant, could not be answered from the curves alone.

The flattening out at the top of the curve indicated that the shaft was hammering through the bearing clearance and that was limiting the amplitude of the vibration. One characteristic of forced tailshaft whirl was that calculation based on a formula of the kind given on page 197 of the paper indicated that the theoretical resonant whirl was remote from the frequency at which whirl was occurring.

Changing the number of propeller blades did not necessarily eliminate forced tailshaft whirl. If a five-bladed propeller was forcing the vibration, changing to one with six blades would probably give better results than one with four.

Slight reduction in excitation for any propeller would probably be accomplished by pitch reduction towards the tips, reduction in area towards the tips, reduction in propeller diameter of up to ten per cent and increased skewback. It was generally beneficial if skewback, measured at the propeller circumference, was not less than eight per cent of the diameter. It also helped if the blade rake was increased. Although the authors stated "length of propeller overhang and propeller mass are adversely affected by the use of aft-raking propeller blades", it should not be forgotten that raking the blades aft took the tips into a region of lower wake variation, as they were further from the deadwater immediately behind the stern frame.

MR. R. J. MILTON, in a contribution that was read by Mr. R. F. Hamlin, said that the paper was of considerable interest to his company, an interest which they had held for a considerable time. The use of roller bearings in the intermediate shaft positions was fairly common and no undue difficulty, to their knowledge, had been experienced. Perhaps the major factor was that the loads on those bearings, relative to shaft size, were low and that stemmed from the fact that the shaft size was controlled by the transmitted torque, making the bore of the bearings very large for the loads involved. Invariably, therefore, roller bearings fitted were of a light series and nominal predicted fatigue life became very generous, making their replacement for fatigue failure reasons most infrequent.

This factor was again fairly significant as a counter to the false-Brinelling effects mentioned in the paper. To their knowledge very little trouble seemed to arise from this effect for shaft bearings and, indeed, even in the much more difficult rudder stock bearing applications that had now been overcome by the use of preloaded bearings and the use of oil as the lubrication medium.

That brought the contributor to some points concerning the thrust block which figured quite prominently in the paper. It had been noted that thrust block stiffness had an important influence on the shaft axial vibrations and in general it appeared to be a case of the stiffer the better. In the past this did not seem to have been adequately considered and his company's response now to that situation would be to give some serious thoughts to the possibility of fitting a pair of thrust roller bearings, preloaded against each other. He would be delighted to receive the authors' comments, as to whether this would be seriously considered by the industry. The effect of such a proposal would be to approximately halve the deflexion or double the stiffness. This referred only to deflexions within the bearings, deflexions within the castings had to be given separate consideration and his company would welcome the opportunity to give that some serious study, particularly now that they had some figures to go on.

An aspect upon which they had often pondered was the use of roller bearings in the stern tube position. They had for a long time thought that the transverse vibratory troubles stemmed from the fact that the propeller mass was inadequately supported at that point and the paper tended to confirm that this was the case.

They had noted reference to five feet overhang and thought that roller bearings would provide a more positive position for the reaction and that two feet could be knocked off that dimension. The point of concern was, as always, the problem of adequate lubrication, but they believed that the successful development of modern sealing arrangements made this a practical proposition. Once again shaft size related to shear was high and a bearing manufacturer was unlikely to be embarrassed if asked to find a bearing with a generous fatigue life capacity. It was interesting to observe the increase in popularity of white metal for the stern bearing—a move which surely had to rely on the seals being efficient? Reference to a weardown figure of 0·001 in per annum, providing good alignment and lubrication was maintained, was impressive but they thought it possible to improve upon this by employing self-aligning roller bearings and, incidentally, better control on the shaft that way was beneficial for the seal. They knew of more than 50 ships of Continental origin fitted out in this way between 1953 and 1963, all without any failures, although admittedly the size of the ships was small.

Oil-injection type muff couplings were becoming increasingly popular and had now quite a long history, particularly in Continental-based ships, and Mr. Milton confirmed that fretting would theoretically occur in that type of coupling, as indeed in every other type of dry coupling, but that did not seem sufficient in practice to create any serious problems. Fretting was considerably worse in cases which suffered continuously reversing torsional loads, or cases where the couplings were subjected to severe bending moments. In tailshaft applications neither of those conditions applied to any great extent. In muff couplings manufactured by his company, the normal form was to phosphate the inner sleeve as a counter to that fretting effect.

COMMANDER A. J. H. GOODWIN, O.B.E., R.N. (Member) said that it was interesting to note the movement in merchant ships towards fewer plummer-block bearings and hence a more flexible shafting system, the limit only being set by shafting whirl.

In the formula given on page 194 of the paper, the qualification was given that it applied when all spans were equal. It might be of interest that, in the design of the shaft K in Fig. 1, there was a great temptation to fit the first bearing aft of the gearcase at a bulkhead 50 ft 9in from the after gearing bearing. The reasons for this were that the bulkhead immediately aft of the gearing was too close for a bearing without risk of upsetting the share of loading between the gearcase bearings and there was insufficient room between the gas turbines for a bearing built-up from the bottom. Use of the Admiralty formula, which had a similar basis to that used in the paper, but allowed for a hollow shaft, suggested a span limit of 46 ft. However, it was argued that the smaller span of the shafting aft and the very short span of the gearing bearings made the assumption of simple support unrealistic and the 50 ft 9 in span was used, completely satisfactorily in the event.

Now that computer programmes for whirling were available, it was possible to exceed the recommended spans in certain cases with greater confidence. In the then lack of such a programme, they had made provision for fitting an additional bearing at the forward bulkhead, but did not have to resort to that. More recently the programme had shown margins varying from five per cent to thirty per cent on the whirling speed due to stiffer neighbouring sections.

In considering outboard shafting, there seemed to be a difference of opinion as to whether the mass of entrained water reduced the critical speed, or whether its resistance to motion increased it. Had the authors any views on that, or better still, any practical results? It seemed to be a point that model tests could satisfactorily resolve.

Referring to the torsional vibration study on pages 191 and 192, it seemed surprising that system (b) was chosen rather than (a), where a minor change in dimensions would have tuned the H.P. and L.P. lines to cross the p^2 axis at zero admittance and propeller excitation would not have reached the turbines at all.

One other small point was that near the bottom of page 190, should not the wording have been "to fit a resonance changer to the Michell thrust block?"

MR. R. COATS felt that the paper was essentially a practical one, with a strong theoretical foundation and with a wealth of detailed information which had been gathered together over a number of years. As the authors had now both left the marine industry it was, perhaps, as well that their swan song should record that information for the benefit of posterity.

His impression, on reading the paper, was that there was so much information that it was easy to overlook the very real achievements that had been made in the recent past in the subject of shafting design and installation, in which the authors had played a very considerable part.

The torsional and axial vibration aspects were now well understood and the values for excitation, damping, entrained water allowance, and seating stiffness estimations gave reliable results. Mr. Coats supported the authors in their contention that the electrical analogue method had great advantages over other methods. It enabled vibration amplitudes and torque, or thrust variations, to be assessed with great facility, and also enabled the effect of changes in the system to be readily appreciated.

In the section on main thrust bearings, reference was made to the "horse-shoe" type of pad carrier, and to the substantial bending moments created thereby at the thrust collar. It might not be generally appreciated that, if a separate thrust collar was fitted, as was often the case with a cast iron or built-up main wheel centre, that bending moment introduced a tendency for the securing nut to unscrew and care had to be taken, both in the type of screw thread used and in locking it securely against relative turning between nut and shaft.

With regard to fair curve alignment, the authors had advanced the techniques beyond the work published in the references, with very satisfactory results. In his younger days, Mr. Coats spent a fair amount of time working out gap and sag values for shafting in connexion with naval installations. Such calculations could not be afforded for merchant applications and probably they were not essential for the powers prevailing at that time. However, those calculations were performed on the ruling that all the bearing centres had to be on the optical sight line and no attempt was made to offset from that line with the object of improving load distribution. Modern computer techniques not only took the drudgery out of that task, but reduced the cost so that the technique could properly be applied to all installations.

Mr. Coats thought the value of that part of the paper would be increased if a typical gap and sag diagram could be included for the benefit of the uninitiated.

Reading the section on non-uniform wake distribution and the offset bending moment arising therefrom, it was clear that this was a developing art on which there was not too much information at the moment. It was not clear, from the case histories, whether the offset moment was taken into account, or not, but Mr. Coats felt the answer was probably not. Could the authors agree, particularly in cases where the propellers were light and the shafts relatively flexible, that the moment would cause a significant change in the slope of the shafting in the stern-tube bearings when going from zero power to full power?

MR. A. ROSE, B.Sc. (Associate Member) said that the authors had given a very valuable addition to the literature on the design of shafting systems. Probably the most valuable part was contained in the appendix which gave the computer programmes for the vibration characteristics.

The programmes given in the appendix were based on an electrical analogy of the shafting system. Since an electrical analogy was available would the authors comment on the relative merits of using a digital computer and an analogue computer for the purpose of vibration characteristics? At first sight, the analogue seemed to have the advantage of the fact that the effect of design changes could be seen readily, particularly by the non-mathematician. Also certain assumptions which might be doubtful (thrust block seating for instance) could probably be investigated more easily on an analogue.

On the subject of bearings, one or two points arose. The authors had said that the greater the specific loading, then the greater the power loss in the bearing. That was only true if the increase in specific load was due to an increase in load on the same sized bearing. If the same load was supported by a shorter bearing then one would expect a reduction in power loss.

As the authors had pointed out, the loading on a plummer block was largely determined by the turning-gear performance of the bearing. However, loads varied due to cargo distribution and hull movement and, because of that, plummer loads had to be kept (nominally) low.

The loads of 100 lb/in^2 quoted by the authors were acceptable on both counts and there seemed little need to increase the L/d ratio from $\frac{3}{4}$ to $1\frac{1}{2}$. Bearing geometry was at an optimum from a lubrication aspect at L/d equals $\frac{3}{4}$ and, perhaps more important in that instance, the cost of the bearing would be approximately doubled for the increased ratio.

The authors had compared horse-shoe main thrusts with all-round designs to the detriment of the former. It should be pointed out that the two performed different duties, in that in the present range of Michell blocks, which were used in the majority of British ships, all-round blocks were used only above 20-in shaft diameters, while the horse-shoe type was used only below 21 in. Even in the range where they overlapped there were differences in loading.

If the 20-in blocks were compared under normal load for each block, then the deflexion of the all round block was 0·014 in and the deflexion of the horse-shoe was 0·013 in, based on B.S.R.A. stiffness figures.

The authors suggested rather radical changes in existing thrust block design in that they preferred all-round design and also seemed to regard journal bearings in the thrust block with disfavour. Since that design had been successfully used for many years was it only to the large sizes now coming into use that their remarks applied? If so, could the authors give a size of shaft above which they felt design changes were most needed?

When the thrust was integral with the gearbox great care had to be taken in stiffening the gearbox. In one case reported to Mr. Rose's company, the gearbox deflected 0·06 in under thrust load.

The figures given for plummer-block deflexion seemed rather high. As it was such an important figure, would the authors give details on how it was measured? Was it on board ship, where seating and chocking arrangements could influence it, or was it on a heavy bedplate in an engine works?

Discussion

Mr. P. Atkinson said that he would like to confine his remarks to the section dealing with Fig. 8.

There had been considerable criticism of the validity of the results produced and to a certain extent this was justified, although he did not think that the authors had attempted to suggest otherwise.

His company were responsible for the development of the work and subsequent computer programme leading up to the results as shown in Fig. 8. He thought it was important to emphasize that only by use of the computer was it possible to present results of the type shown in the paper.

Even so, certain assumptions were necessary and of these the use of the line in preference to the surface treatment of the wake appeared to be the most important. This was shown clearly by the two examples in the paper, the higher values of thrust and torque fluctuations in the case of the twin-screw vessel as compared with the single-screw tanker would not be expected in practice. For the twin-screw vessel the line approach over-emphasized the effect of the local wake concentrations in the boss region. The choice of wake distribution for the calculation was as predicted from model survey but the resultant mean value was modified to agree with the ship service condition.

On this basis he did not think that the omission of skew in the calculation would affect the order of fluctuations as shown.

Finally, referring again to Fig. 8, the datum point for the twin-screw case should be the upper vertical and not the boss axis.

Correspondence

Mr. W. M. Brown, B.Sc. remarked, in a written contribution, that it was interesting to note that the authors employed the voltage-force electrical analogue method for torsional and axial vibration calculations. It was further noted that for axial vibrations the method followed convention by dividing the shafting into a number of masses and springs and did not consider the shafting to be a continuous system. This could be conveniently done since the shaft was simply a long bar subject to pressure disturbance, the analogue of which was the loss-less transmission line and, in fact, one computer programme which took cognizance of this had been operational for a number of years, producing results similar to those obtained with more conventional methods.

In connexion with the authors' comments regarding the various methods of allowing for entrained water effects, it would be realized that use of the British Ship Research Association empirical formulae for computing the axial stiffness of thrust-block seatings automatically meant employing an entrained water allowance of 50 per cent. However, since there was little logical basis for the use of fixed percentage allowance and technically the methods of Burrill and Robson (reference (7) of the paper) were more attractive, it might be worth while to modify the B.S.R.A. equations by substituting back into the relevant Holzer tables entrainment allowances calculated by the methods of Burrill and Robson.

The design study on the axial vibrations of the twin-screw passenger liner (shaft G) made some interesting points which were worthy of comment. The first of these was the main conclusion that the vessel should be fitted with a six-bladed propeller and, while agreeing that this was the best choice so far as the shaft system was concerned, it was relevant to enquire what effect this would have on the hull. So far as one could determine from Figs. 12, 13 and 14, the maximum power speed corresponded to about 175 shaft rev/min and, in consequence, the maximum beat frequency of the shaft brackets would be 1050 c/min. This frequency was in the range of the second order criticals which would be expected of the shaft brackets for a ship of this type and could give rise to objectionable amplitudes of bossing and hull vibration. This could perhaps be overcome by adopting "A" frames for shaft supports, but this would probably increase the power requirements of the vessel. Furthermore, with a ship of this type, a rigid and, therefore, costly superstructure could be envisaged as necessary to accommodate a beat frequency of 1050 c/min. These problems could, of course, be overcome if a four-bladed propeller was employed and it was therefore worth while to examine the authors' reasons for discarding this choice. The main reason given for discarding the four-bladed propeller was that, with the thrust block 91 ft 6 in aft and with a combined stiffness of 8000 tons/in, the vibration characteristics were unacceptable. In the first case, 100 ft aft was probably nearer the optimum position for the thrust block of this particular vessel and the statement that the combined stiffness of 8000 tons/in was the maximum possible was certainly open to debate. He suspected that the figure of 8000 had been obtained as the maximum quoted by Couchman (reference (9) of the paper) for his ship A.V.1 (shaft E in the present paper) and it was therefore important to establish what was incorporated in this figure. Couchman's figure was based on a seating stiffness of 17 500 tons/in and the stiffness of a standard A.R. 1440 thrust block having a measured stiffness of 14 700 tons/in. However, for shaft E the thrust was 160 tons compared with 210 tons for shaft G and it could therefore be expected that the thrust block of the latter design would be an A.R. 1730. Although he had no figures for the stiffness of the A.R. 1730 thrust block, it would certainly have a rating in excess of 20 700 tons/in which was the measured value of the A.R. 1600 thrust block.

Furthermore, he could see no reason why a seating built in the shafting tunnel should not have a stiffness of up to 25 000 tons/in. In consequence, it was considered that instead of 8000 tons/in combined stiffness the maximum possible, with a vessel of this type, would be in the order of 13 000 tons/in. With such a stiffness and a thrust block 100 ft aft it was suggested that satisfactory axial vibration characteristics could be achieved with a four-bladed propeller. This would, of course, reduce the risk of exciting harmful vibrations of the bossing and hull which he considered to be a serious objection to the proposal presented in the paper.

However, if this should prove to be the case, then no doubt it would be desirable to fit a resonance changer to the system for operation under rough weather conditions, especially when steaming at maximum power.

Dr. P. A. Milne, B.Sc. (Associate Member) wrote that the authors suggested that the design of a shafting system would not take place until late in a contract. This was not necessary when computer techniques were available. A preliminary shafting arrangement could be prepared at the same time as the preliminary machinery arrangement. By applying simple rules, the spacing of bearings and the diameters of shafting could be established. In most installations the rules for spacing would provide sufficient flexibility for alignment and avoid shaft whirl. It was then possible to check the torsional and axial vibration characteristics of the system to see if they were acceptable. Detail design could be carried out at a later stage and selected calculations repeated or extended if there was any margin of doubt.

Apart from the calculations mentioned in the paper, the detail design of stern gear needed revision. Methods of attaching, or retaining, propellers, interference fits, surface finish, design of tailshafts and stern tubes were often based on previous designs.

On larger vessels local aft end or "fantail" vibration had caused difficulties. Attempts had been made to predict the resonant frequencies by considering the structure aft of the forward engine-room bulkhead to be a built-in beam and then applying corrections. This work suggested that the problem could be avoided by using lower propeller revolutions and the smallest number of propeller blades. This introduced another factor into the choice of propeller design.

At the beginning of a contract, the axial frequencies of a shafting system could be calculated. In some cases they lay outside the running speeds of the shaft or occurred at speeds well away from the normal service values. In these cases, was there a value of the difference between the resonant frequency and the service speed above which the vibrations could be considered insignificant? It would not then be necessary to check the accelerations and amplitudes in the system. Had the authors considered axial vibrations in slow-speed direct-coupled Diesel installations? In this case the stress was calculated, but there was some divergence of opinion on the need to carry out an analysis of this type.

In most shipyards, the alignment was established in the light ship condition at a fitting-out quay. In one vessel, the forward bush in the stern tube was loaded during fitting out and then checked in service. It was still loaded in the ballast condition of the vessel, but carried no load in the loaded condition of the vessel. Stiffer structure in way of the shafting system and more flexible systems helped to reduce this problem. The change in hull form with load could, however, still be significant and correction factors for aft end installations should be considered when the alignment was checked in the light ship condition.

When a shafting system was being aligned on a Diesel installation, the load of the flywheel could be shared between the adjacent engine bearing and the first plummer block. The aft coupling on the engine would be jacked up in the ship until the load equalled the support required from the shafting system.

The gap and sag at the coupling would then be arranged to provide this support. This would avoid overloading the last bearing in the engine.

MR. M. HARPER (Member) wrote that without doubt there were available experience and design techniques which enabled a good design of shafting system to be achieved. The paper did, however, indicate several areas in which more data were required: stiffness of hull and appendages, stiffness of bearing and thrust seatings, wake forces and pressure forces. The need for close liaison between marine engineers and naval architects, particularly in the early stages of design, was clearly indicated.

The authors' comments on the application of hollow shafting in merchant vessels appeared to present a specious argument and yet it did not appear impossible to obtain the benefits of hollow shafting whilst preserving adequate transverse stability by reducing superstructure moment.

The results of the axial vibration study for the twin-screw passenger liner were very interesting. The range of resonant frequencies for the assumed values of combined thrust block and seating stiffnesses showed that more precise values for this stiffness could be useful. Nevertheless, for the thrust block forward there was a reassuring margin between the service speed and the nearest resonant frequency.

He agreed with the authors that flexible couplings should be assumed to function as intended, when considering axial vibrations, and he shared their regard for the value of the admittance diagram.

Could the authors please elaborate on their preference for an L/d ratio of 1 to 1·5 for plummer blocks?

The authors presented a good case and, he felt, a correct one, for a reduction in the number of tunnel bearings in merchant vessels. However, it would be interesting to hear owners' views, as he was sure there would be a very cautious approach to any marked change from established practice in the area.

A separately-mounted main thrust lent itself to the provision of a reversible thrust shaft at little extra cost and, surely, provided a fair degree of extra insurance in the event of damage to the ahead face. Did he understand that, even when a separately-mounted main thrust was incorporated, the authors did not consider the reversible thrust shaft worth while?

MR. H. J. ADAMS observed, in a written contribution, that in recent years, as stated in Section I of the paper, a common arrangement in merchant tankers and bulk carriers with Diesel engines right aft was to have a tailshaft, one intermediate shaft and a thrust shaft. In spite of the considerable strengthening of the after end, which was a recent feature, vibration had been experienced in the region of, and above, the engine room. This vibration appeared to be dissociated from the natural periods of the hull and was frequently evident to a high degree in the wheel-house and other raised structures. The source of this vibration appeared to be, at least partially, the propeller and this source might be coupled with engine-excited forces.

This vibration appeared to have become a major problem in the type of installation quoted in the previous paragraph and was particularly unfortunate in view of the numbers of delicate instruments which now formed part of the equipment of a ship and which were invariably situated aft. From a hull and operational aspect, the all-aft arrangement was the most attractive for large single-screw bulk carriers and tankers.

The authors stated that, to eliminate propeller excitation, two conditions were required, a radially uniform flow past the propeller, or a propeller with an infinite number of blades. The latter was not a practical solution and assuming, as was implied, that a propeller design which would cope with a variable wake without exciting periodic forces was not possible, the conditions suggested that approach to the problem should be on the lines of modification of hull form forward of the propeller to produce as even a wake as was possible.

Intensive research into hull form aft was necessary and it might be that radical alterations would be required. The basic shape aft, in single-screw ships, had changed little since the days of sail and it was suggested that some radical rethinking was necessary. It was agreed that the fitting of a new after-end form might increase the cost and, since reduction or elimination of vibration was difficult to assess in terms of money, it would be necessary, in order to make any proposal attractive to owners, that it should show an overall improvement in propulsive efficiency. Was this impossible and might it not be a natural result?

MR. A. R. HINSON (Associate Member), writing in a further contribution, asked if the authors would give their opinion of the following formula for tailshaft diameters for five-bladed propellers:

$$D = [4{\cdot}6 + 2(C_b - 0{\cdot}65)] \sqrt[3]{\frac{\text{shp}}{\text{rev/min}}} \text{ inches,}$$

where C_b was the block coefficient.

E. F. Noonan* had given details of strain gauge measurements at sea. Summarized, the results were:

	Sea condition	Ship condition	Maximum stress due to vertical bending moment lb/in^2
1)	Calm	Ballast	3000
2)	Calm	Loaded	3000
3)	Moderately rough	Ballast	4000-5000
4)	Rough	Loaded	7000

Hence the maximum stress in rough weather was 2·3 times that encountered in calm weather.

From model work carried out by van Manen and

*Noonan, E. F. 1961. "Propeller Shaft Bending Stresses on the *Esso Jameston*". *Jnl. A.S.N.E.*

Wereldsma† it could be shown that, for a conventional stern arrangement in loaded condition at 100 per cent power, the largest alternating stress component was that due to the vertical bending moment and, for a five-bladed propeller, was +55 per cent to −10 per cent approximately when expressed as a percentage of the average propeller torque.

Hence, it could be argued that the maximum rough weather vertical bending moment for a five-bladed propeller would be $2 \cdot 3 \times 0 \cdot 55\ T_{av} = 1 \cdot 25\ T_{av}$.

Assuming that the conversion factor from model to full scale for five-bladed propellers‡ was 0·8, then the vertical bending moment could be taken as $0 \cdot 8 \times 1 \cdot 25\ T_{av}$ or T_{av}.

If these assumptions were used and a maximum permissible bending fatigue stress of 5000 lb/in^2 was adopted, the tailshaft diameter obtained was greater than that given by current practice.

It seemed evident, that as methods became available for more accurately determining the forces generated by the propeller, the permissible stress levels taken for design purposes would have to be reviewed in order that full advantage might be taken of these methods when they were perfected.

Mr. Langballe (Member) wrote that he would confine his remarks to the problem of shaft whirling, his experience of which seemed to be at some variance with that of the authors.

Det norske Veritas had devoted some attention to tailshaft whirling and the operational difficulties which might arise due to this mode of vibrations. In this respect, it was his opinion that a large number of cases of serious tailshaft and stern-tube troubles might be attributed to resonances and flanks of whirling vibrations. Contrary to the authors' experiences, which suggested first order whirling as the most probable mode, some theoretical and experimental investigations seemed to indicate that the fundamental, blade-frequency excited, whirling vibrations were likely to cause trouble under certain circumstances, as he hoped to demonstrate.

Jasper§ had established a complete theoretical method suitable for the analysis of marine shafting. It took into account the vertical, horizontal and angular bearing stiffnesses, the shear rigidity of the shaft, the variable cross-section of the shaft and the diametral moment of inertia of the propeller, Based on a finite element method, the eigen values of the equations of movements were readily found by a digital computer solution for any practical mode of vibrations.

Several uncertainties as to the system parameters entered into this otherwise rigorous treatment:

a) bearing stiffnesses;
b) location of the points, or areas and distributions of support of the tailshaft in the stern tube;
c) mass of entrained water by lateral vibration and vibration around the propellers' diametral axes.

However, to evolve experimental values, the mathematics had to be kept straight.

Usually, lateral vibrations of the propeller shaft had to be measured as stresses and deflexion inboard of the aft peak bulkhead, where the amplitude might be very small. Great care had therefore to be exercised in providing for a sensitive and noise-free electronic measuring system and instrumentation. In a number of measurements undertaken, the fundamental blade-frequency criticals were located well above the normal shaft speed and the natural frequencies had therefore to be determined by the relatively feeble overtones, that was to say, ±8th, ±12th orders and so forth, for a four-bladed propeller.

A typical case was shown in Fig. 25, where there was extreme wear-down of the lignum vitae, cracking of the shaft cone in way of the forward end of the propeller boss (no sea-

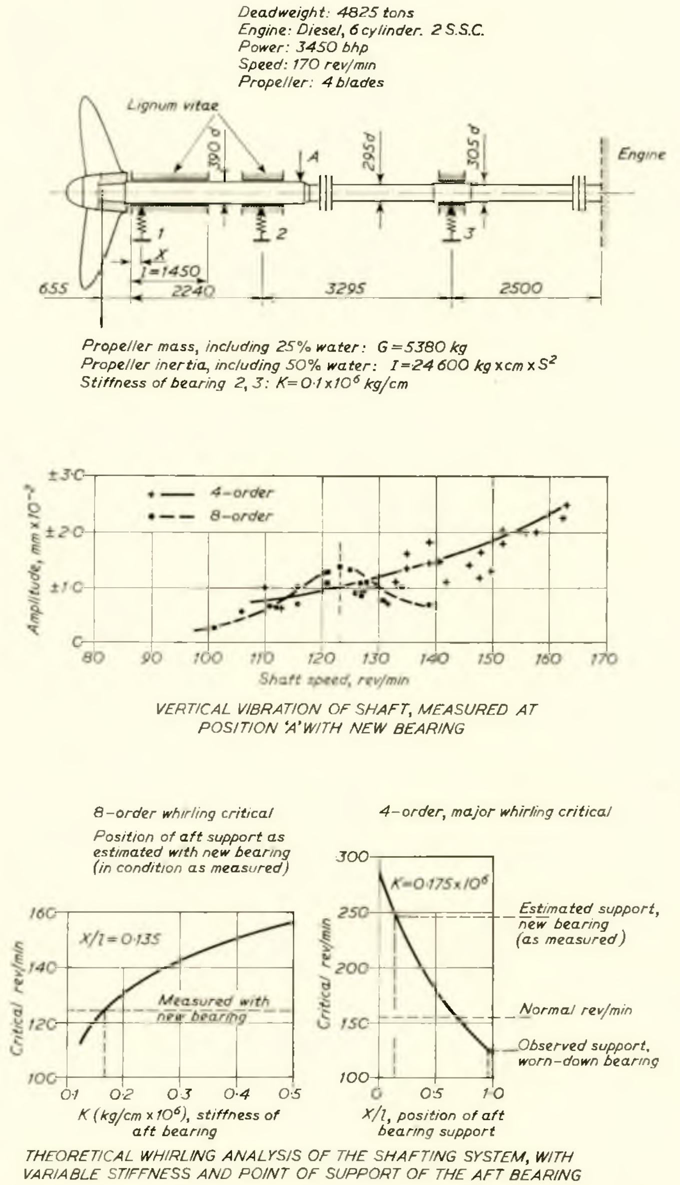


Fig. 25—*Analysis of whirling vibration in medium-size shafting*

water corrosion or fretting could be detected) and cracking of the bronze liner. In the re-wooded and re-aligned state, in which the measurements were taken, the 4th order major critical, the flank of which appeared in the upper speed range, was far, and safely, above the normal revolutions. With the aid of the measured 8th order resonance, some information given by Jasper and the determination of the bearing positions by the strain-gauge method, the parameters a), b) and c) were estimated.

A subsequent computer calculation showed that, together with progressive wear of the lignum vitae and forward moving of the aft effective point of support of the tailshaft, the 4th order critical would fall to around normal speed, causing violent vibrations in the stern-tube region and overstressing of the shaft. In the last phase, only the fore section of the lignum vitae and the glands had carried any loads, which was apparent upon inspection of the bronze liner. The procedure used in the analysis of the shafting system should be self-explanatory upon studying Fig. 1. However, it should be noted that the stiffness of bearings Nos. 2 and 3 as well as the shafting ahead of bearing No. 3, was of little importance.

†van Manen and Wereldsma. 1960. "Propeller-excited Vibratory Forces in the Shaft of a Single-screw Tanker". Netherlands Research Centre, T.N.O. 37M.

‡Nitzki, A. 1959. "Resistance and Propulsion of High-powered Single-screw Vessels". *European Shipbuilding*, No. 3.

§Jasper, N. H. 1956. "A Theoretical Approach to the Problem of Critical Whirling Speeds of Shaft-disc Systems." *International Shipbuilding Progress*, pp. 183-198.

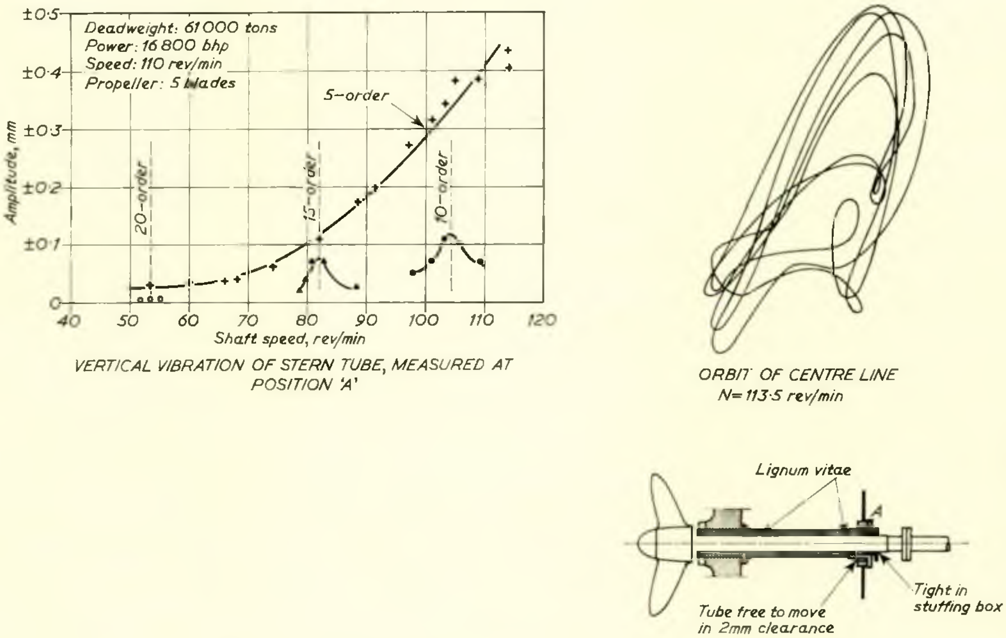
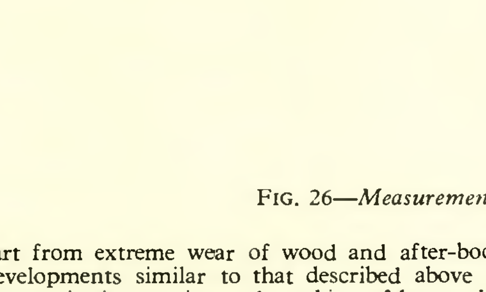


FIG. 26—*Measurements of whirling resonances in large shafting*

Apart from extreme wear of wood and after-body vibrations, developments similar to that described above had contributed to cavitation erosion and cracking of bronze liners and even loss of the propeller. In the case shown in Fig. 26 the flange connexion between the stern tube and bulkhead failed, due to dynamic overload of the forward bearing section as the major whirling critical came down because of wear of the aft bearing section. As would be noted, the whirling orbit was rather elliptical, but seemed to become circular at the resonances. By analysis of the dynamic measured bending stresses in the shaft both forward (+) and counter-whirl (−) seemed to be possible.

The development in the two cases cited was accelerated by an intentional slope boring of the stern tube. In Mr. Langballe's opinion, this procedure was out of place for lignum vitae bearings, which should be aligned straight.



ASSUMED TOTAL BEARING STIFFNESS OF AFT BEARING: 10^6 kg/cm

Order	Critical rev/min Counter	Forward
4	157	178
5	127	141
6	107	116

FIG. 27—*Theoretical analysis of whirling criticals in large shafting with white-metal stern-tube bearings*

With the almost universal adaptation of white metal, oil-lubricated, stern-tube bearings in new ships, the possibility of troubles of the kind mentioned was reduced. However, the arrangement of shafting alignment should be given the closest attention. To his mind, it seemed inadequate that the alignment of these stern-tube bearings, which was rather critical due to their great length, should be left to the accuracy of such contrivances as the stern-tube boring gear. Rigid control, at least, was essential. It would seem a substantial step forward in this respect if a design for an adjustable stern tube could be evolved.

A theoretical analysis of whirling critical evolutions for large-diameter shafting with a white-metal stern-tube bearing was shown in the table in Fig. 27; (the stiffness and load distribution of the aft bearing were estimated). As would be noted, major propeller-excited whirling criticals in the normal speed range might exist with a six-bladed propeller under the assumptions which were made.

Evidently, there was a need for research in the field of tailshaft whirling and it would be most gratifying if a general interest—as strong as that which was shown in the fields of torsional and axial vibrations—could be aroused in such theoretical and experimental investigations. More information on the stiffness of white-metal tailshaft bearings, dynamic magnifications, entrained propeller water, influence of stern-tube alignment and forced whirling vibrations would be of particular interest.

MR. J. H. ATTWOOD (Member) wrote that he would be glad if the authors could give their considered opinion on the following points.

Referring to the variable wake analysis (see Fig. 8), the thrust and torque fluctuations, for a single-screw tanker (fully-framed aperture) were quoted as 15·5 per cent and 5·0 per cent respectively. He would be grateful for the authors' opinions as to whether, and if so, how, the design of the shafting system was affected by the thrust and torque reversal experienced when the propeller was turning astern with ahead way on the vessel. Reference to a paper entitled "Backing Power of Geared-turbine Driven Vessels" by E. F. Hewins, H. J. Chase and A. L. Ruiz would appear to indicate that, with the propeller turning astern and ahead way on the vessel, the maximum astern thrust and torque could approach, over a limited period of time, 70 per cent of the ahead valves.

The authors had included an allowance for entrained water at the propeller in their calculations, but he would ask whether a further allowance should be made to allow for accelerating and decelerating forces at the propeller, caused by the vessel pitching in a seaway. For example, with a wind force varying between Force 4 and Force 8, combined with seas varying from moderate to rough, accelerations of up to $0 \cdot 37g$ had been measured in the vicinity of the propeller.

Mr. R. J. Milton wrote in a further contribution that there appeared to be a little confusion as to the definition of self-aligning bearings and he would like to define what he had in mind during the discussion. There were essentially two types of self-aligning bearing, those that relied on the outer ring being fitted in a sphered housing and those which had their rolling elements operating on a sphered outer track.

Of the two, he agreed that the spherical-housing type had to be settled in its correct position during assembly and that, with the flexible marine structure, it would be difficult to avoid a degree of edge loading on the rollers. However, the spherical roller bearing with its barrel-shaped rolling element operating on spherical outer track had quite different properties and edge loading in the rollers was just not possible. This bearing could not only withstand initial malalignment but also changing malalignment due to changing loading conditions. Very conveniently, its use, particularly within the stern tube position, would enable the precise point of load reaction to be accurately predicted.

Authors' Reply

The authors said, in reply, that they were grateful to all those who had attended the presentation of the paper and in particular to those who had contributed to the discussion which followed. They were also pleased to acknowledge the written contributions received.

The authors were pleased to have Mr. McClimont's critical appraisal of the paper though they could not agree with all of the comments made. They could make no apology for presenting a paper biased towards turbine installations since it was based on their experience at Pametrada. Direct experience of motor-ship shafting systems had been gained only in recent years and was mostly confined to the problem of shaft alignment.

It was not the authors' intention that the opening paragraphs of the paper should suggest a high frequency of shafting and bearing failures, but that those failures which had occurred had pinpointed the troublesome features of shafting systems. They agreed that more attention to pull-up of propellers on their cones and better sealing arrangements had contributed to a reduction in the incidence of cracking and that prevention was certainly better than cure. Even when the seal was quite effective, however, and the cone free from any sign of corrosion, fatigue cracks could still occur in this area. The authors had encountered this in the case of shaft D and Mr. Langballe reported a similar experience in his written contribution to the paper.

Mr. McClimont had rightly pointed out that the classification society's changed requirements for intermediate shafting diameters shown in Fig. 4 did not apply to tailshafts. Fig. 4 was relevant when discussing axial and torsional vibrations, since the tailshaft was only a minor part of the complete shaft system, but was obviously irrelevant in reference to tailshaft vibrations.

The use of hollow shafting was covered in the reply to Mr. Harper.

In referring to the axial vibrations of a dry cargo ship with which Mr. McClimont was concerned, the authors did not agree that particular stress was placed on the weight of the propeller. This was just one of a list of five features given as relevant to the axial vibrations of this vessel.

Comments on Fig. 8 were made by several contributors and these would all be covered in the reply to Mr. Atkinson.

Mr. McClimont's comments on the nature of higher order hull vibrations were most interesting. The authors, however, still considered their statement, that the frequency of propeller pressure forces on the hull were of the same order as the natural frequencies of the hull, to be fair comment. It was based on typical mainshaft speeds for large merchant ships of 100-120 rev/min which, combined with a four-bladed propeller, gave rise to exciting frequencies of 400-480 rev/min. These approximated to the 5th and 6th order vertical and horizontal, and 1st order torsional, hull vibrations.

The electrical analogy method for solving mechanical systems was a standard method described in many text books and the application of the method to marine geared-shaft systems had been fully covered elsewhere[26]. The authors, therefore, did not consider it necessary to go into details of the method other than to define the system used. They were, however, more concerned with the particular advantages of the method when designing shaft systems since these advantages did not seem to be fully appreciated by everyone [see for instance the discussion of reference[26]]. The use of the admittance diagram as a design chart was, therefore, discussed in detail in the hope that its value might become more widely known. The authors were pleased to note that Mr. McClimont was now nearer to being a convert to the electrical analogue method, but would have preferred the conversion to have been attributed to the admittance diagram rather than the availability of a digital computer.

The authors were in agreement with Mr. McClimont in regarding flexible couplings as alignment devices rather than axial vibration absorbers. Results of an investigation made in a large tanker indicated that the flexible couplings were unlikely to slide even at frequencies approaching turbine frequency. They did feel, however, that the fine-toothed coupling was a much maligned component which did a reasonable job of work under particularly arduous conditions.

Mr. Blake's case history of aft-end vibrations on two dry cargo vessels underlined the need for adequate support at the inboard end of the tailshaft. In this instance it was obtained via a zero-clearance bearing, the effectiveness of which was evident in the greatly reduced vibration level recorded. Damping characteristics of the roller bearing would contribute very little to this improvement, the major effect being achieved by the reduced bearing span at this point, since it was most probable that the tailshaft was originally set concentric with the forward stern-tube bush and was, therefore, not supported at that point. The solution in this case should be compared with the authors' approach to shaft T (Fig. 2) described under "Case Histories". A similar reduction of bearing span was obtained by lowering the aft most tunnel bearing until contact occurred at the forward stern-tube bush.

The uncertainty of changing the number of propeller blades was also brought out in Mr. Blake's example. Since the additional bearing was a successful cure for the shaft vibration problem experienced with the five-bladed propeller, one wondered whether it might also have reduced the hull vibrations previously experienced with the four-bladed propeller.

The authors' direct experience of false-Brinelling was confined to small diameter solid-race ball and roller bearings, in which the indentations were generally present around the full circumference of the races. In one example these indentations were about 0·05-in deep, although figures of 0·001 to 0·005 in were more typical. Where such damage did occur a great reduction in life would certainly follow and it was, therefore, good to learn from Mr. Blake (and Mr. Milton) that the problem did not occur in the present context.

The authors appreciated the difficulties of concisely defining "hull form" and, in reply to Mr. Hinson, they had in general used the term as synonymous with block coefficient. Perhaps a better term would be evolved eventually, but at present the only alternative was to quote a multitude of parameters which then confused rather than clarified the picture. The problem of representing mathematically the wake flow and its effect on the propeller was indeed complex and even the simple generalization made by Mr. Hinson that "the finer the ship, the less variation in the wake component" was not always true. Based on a statistical investigation of measurements of shaft torque and propulsion force fluctuation of about forty different ship models carried out at the Dutch Ship Testing Station, J. D. Van Manen[32] concluded that no systematic relationship could be found between the amplitude of the force fluctuations and the main parameters of the ship's shape, i.e. block coefficient, prismatic coefficient or ratio of propeller diameter to ship's length. Figures were given for the thrust and torque variations for four and five-bladed propellers over a range of values of the prismatic coefficient of the after part of the ship and these were followed by the statement that finer and thus mostly faster ships might give rise to very much stronger force fluctuations.

Table II was quoted directly from reference (6) in the paper in which no reference was made to variations in bending moment and the table in isolation did indeed tell only half the story of what happened at the propeller. However, the authors did not suggest that the number of propeller blades should be determined from a consideration of Table II only. On the contrary, the main aim of the authors' paper was to present as complete an account as possible of the many factors to be considered when designing a satisfactory shafting system and, in particular, Fig. 8 was included to describe fully the variations in forces and moments which occurred at the propeller. This figure surely emphasized that Table II was limited in application to the prediction of the axial and torsional behaviour of the system and gave no indication of the lateral vibrations which might occur.

It was not clear what Mr. Hinson meant when he claimed that Table II could be misleading when using Jasper's formula to calculate propeller whirling frequencies. This formula was concerned only with the estimation of frequency and required no assumptions about the magnitude of the exciting forces.

Mr. Hinson was wrong in thinking that Jasper's formulae for tailshaft whirl did not include the case of tailshaft whirl at propeller blade frequency. The equation given was a simplification of the more general formula:

$$\Omega^2_N = 2 \times \frac{Q_2 \pm \sqrt{Q_2^2 - Q_3\, Q_1}}{Q_3 \cdot Q_1}$$

where:
Q_1 = $(\delta p \theta m - \delta m \theta p)$;
Q_2 = $(m \delta p + G \theta m)$;
Q_3 = $4mG$;
m = mass of propeller;
δp = static deflexion of propeller due to unit load;
δm = static deflexion of propeller due to unit moment;
θp = static rotation of propeller due to unit load;
θm = static rotation of propeller about a transverse axis due to unit moment;
G = $\tau_d (1 - kh)$ an effective inertia;
τ_d = mass moment of inertia of propeller about a diameter;
τ = polar mass moment of inertia of propeller;
k = ratio τ/τ_d = 2 for a thin disc;
h = ratio ω/Ω_N;
ω = spin velocity of the shaft (rads/sec);
Ω_N = natural angular whirling frequency (rads/sec).

The effective inertia term, G, was the device by which blade frequency resonances were included. Values of G for different orders of forward and counterwhirl were given in Table IX.

In Fig. 28 the calculated tailshaft whirling frequencies for shaft D (Fig. 1) had been plotted against main shaft speed. This diagram illustrated the effect of the gyroscopic influence of the propeller on the higher order whirling frequencies and at the same time showed the much greater effect the order of whirl had on the critical propeller shaft speed. It was this latter effect which was the more important, in particular when it was accompanied by the higher bending moments which characterized the five-bladed propeller.

With reference to Fig. 24, the distinction made by Mr.

TABLE IX
THE EFFECTIVE INERTIA, G, FOR DIFFERENT ORDERS OF TAILSHAFT WHIRL

Order	Forward whirl	Counter whirl
Zero*	τ_d	
First	$-\tau_d$	$+3\tau_d$
Second	O	$+2\tau_d$
Third	$+\frac{1}{3}\tau_d$	$+\frac{5}{3}\tau_d$
Fourth	$+\frac{1}{2}\tau_d$	$+\frac{3}{2}\tau_d$
Fifth	$+\frac{3}{5}\tau_d$	$+\frac{7}{5}\tau_d$
Sixth	$+\frac{2}{3}\tau_d$	$+\frac{4}{3}\tau_d$

*Stationary shaft

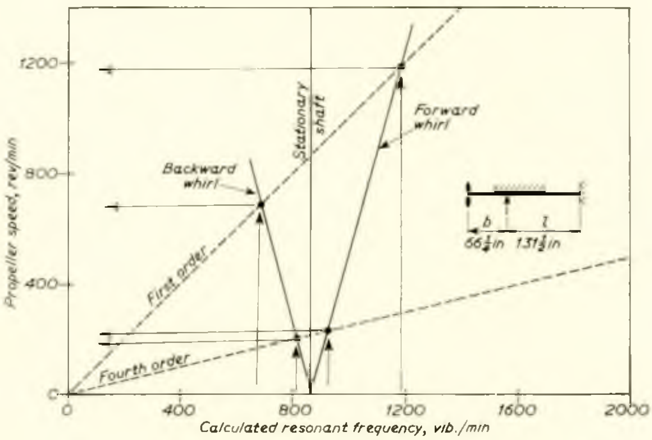


FIG. 28—*Tailshaft whirling diagram for shaft D*

Hinson between "resonant" and "forced" was rather puzzling. He suggested that it was only when the variation in bending moment was sufficient to lift the propeller and shaft clear of the aft end of the stern tube that forced whirl occurred. In vibration theory the term "forced" was used to describe the frequency or response of a system when vibrating at the frequency of an external exciting force. This was in contrast to the natural frequencies of the system which were a property of the system and independent of the source of excitation. Resonance occurred when the forcing frequency became equal to one of the natural frequencies of the system. The authors, therefore, suggested that forced tailshaft whirl occurred when the frequency of the force and moment variations at the propeller, i.e. propeller blade frequency, did not coincide with a natural frequency of the tailshaft. The frequencies indicated in Fig. 28 by the intersection of the first and fourth order radial lines and the backward and forward whirl curves corresponded to propeller speeds at which resonant vibrations would occur, in the first case due to mass unbalance of the propeller and, in the second, due to wake excitation at propeller blade frequency.

The "forced" curve in Fig. 24 suggested the rising flank of a resonance curve and the plateau indicated that the vibration amplitude had been limited by the bearing clearance. The remedy in such a case was as described in the reply to Mr. Blake, namely, to reduce the free span of shafting at the aft end and this usually meant achieving a good positive reaction at the forward end of the stern tube. Increasing the number of propeller blades would lower the propeller speed at which the offending tailshaft whirl occurred and a reduction would raise it. In the case reported by Mr. Blake shaft vibration occurring with a five-bladed propeller could not be treated by reducing the number of propeller blades because of the previous finding that hull vibrations were then encountered.

The authors were also very much aware of the improvement in wake effects obtained in the past by the adoption of backward raking blades. On page 197, however, they had pointed to the less favourable, and also less obvious, consequences of this practice.

The formula:

$$D = \left\{4{\cdot}6 + 2\,(C_b - 0{\cdot}65)\right\} \sqrt[3]{\frac{\text{shp}}{\text{rev/min}}} \text{ in}$$

for the diameters of tailshafts fitted with five-bladed propellers given by Mr. Hinson in a written contribution had been plotted in Fig. 29 in the form of a coefficient C against block coefficient (C_b). Fig. 29 showed that if the proposed formula

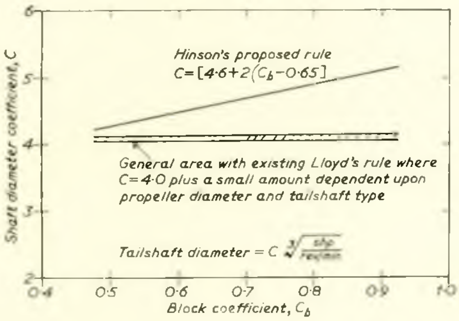


FIG. 29—*Hinson's proposed rule for tailshaft diameter when a five-bladed propeller is fitted*

was applied, all ships with a block coefficient greater than 0·5 and fitted with a five-bladed propeller would require tailshafts with diameters in excess of the current Lloyd's Register requirement. For the tanker group (i.e. C_b approximately 0·8) the increase in diameter was approximately fifteen per cent.

The value of such an amendment to the present Rules was debatable for two reasons. First, reference[32] had already shown that the greatest propeller excitations could be found with fine, fast ships and not necessarily in ships with a high block coefficient. Secondly, some doubt had been thrown on the value of oversized tailshafts so far as water-lubricated stern tubes were concerned by reference[35] which reported the survey observations of a large group of ships mainly fitted with five-bladed propellers. Whereas the mean expected life before renewal for those ships having Rule-size tailshafts was 10·5 years, for those having oversize tailshafts it was only 6·3 years. The smaller-size shafts failed typically by fatigue cracking, whilst the oversized shafts failed by reparable damage to the bronze liner. This did suggest that the oversize shaft would show a clear advantage if an oil-lubricated stern tube were also adapted.

Reference to Noonan's measurement of tailshaft stresses under different loadings, speeds and sea states brought to mind similar work by others [e.g. reference[36]] showing similar results. This particular aspect was also introduced in Mr. Attwood's contribution and the reply to him was also relevant here. These two problems, viz. the severe transverse force fluctuation arising with five-bladed propellers and the effect of pitching and other hull motions, seemed to lend themselves to the type of investigation summarized in Fig. 8 and the authors would recommend this approach to Mr. Hinson.

Mr. Milton's comments on the use of roller bearings in the intermediate shaft positions agreed with the authors' experience of this type of bearing used in channel steamers. Whether the use of a pair of thrust roller bearings, preloaded against each other, would produce a stiffer unit was not obvious. Such an arrangement would eliminate the axial clearance required with tilting-pad bearings, but this would only be relevant if thrust reversal occurred. In all but the most severe cases of axial vibration, the thrust would be unidirectional so

TABLE X

BREAKDOWN OF THRUST BLOCK AND SEATING STIFFNESS FOR LARGE SHIPS

Component	Stiffness 10^6 lb/in	Remarks
Fluid film	90–260	
Thrust bearing parts	13– 22 22– 36	Standard bearing Rigid bearing
Thrust Collar	75–120	
Housing	20– 35 35– 50 35–100	Thrust bearing forward of gear-box Thrust bearing aft of gear-box Thrust bearing far aft of gear-box and independently mounted
Foundation	3– 15 5– 20 20– 40	Thrust bearing forward of gear-box Thrust bearing aft of gear-box Thrust bearing far aft of gear-box and independently mounted
Total	2– 13	

that preloading would not greatly change the stiffness of the roller races. Table X showed a breakdown of the stiffness of a conventional tilting-pad main thrust block used in large naval vessels [reference[33]]. It was evident from Table X that the oil film under the tilting pads was a very stiff component and that most of the flexibility of the thrust block occurred in the housing and in the mechanical parts. Considering an equivalent, but hypothetical, roller bearing it was possible that the races themselves might have a greater stiffness than the tilting-pad assembly including the oil film, but the thrust housing itself would be appreciably larger for the new unit and, consequently, would be less stiff. Thus it was difficult to estimate whether the stiffness of the whole assembly would be greater or less than that of the corresponding tilting-pad unit. This, of course, could easily be determined by measurement on an existing unit before installation in the ship. The authors would, however, draw Mr. Milton's attention to the axial vibration study for a twin-screw Diesel ferry in which the alternative and, in this case, stiffer, system was rejected in favour of the original design.

The authors agreed with Mr. Milton that many cases of transverse vibration were the result of inadequate support of the tailshaft. While many ways were available for solving such problems, it was the authors' conviction that re-appraisal of the tailshaft bearings and their working conditions would generally show the direction for greatest improvement. As Mr. Milton implied, the roller bearing gave a zero-clearance feature, which was most attractive, and also offered a well defined point of support due to its short length and self-aligning property.

The recent change in Lloyd's Rules which allowed the stern-tube bearing length to vary while the specific loading was taken as the design parameter promised a useful reduction in the length of white-metalled bushes, particularly if an oversize tailshaft was envisaged. Under these conditions, the short bush shown in Fig. 30 might eventuate, in which case the roller bearing would show no reduction in overhang and might even appear less attractive because of its larger outside diameter.

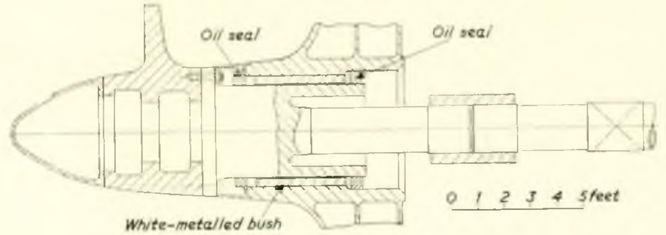


FIG. 30—*Proposal for stern tube and tailshaft assembly*

Since both types of bearing required the same type of oil-seal, there was no basis for preference on this score, but if sea water did penetrate the seal it would surely have a much more serious effect on the precision-machined components of a roller bearing unit than a white-metal bush. The normal device of fitting a gravity tank in the stern well above the water level, however, did ensure that any leakage would be of oil outwards and not water inwards.

The omission of a reference to use of the oil-injection type of muff coupling was an oversight, since the authors were aware of its extensive use in connexion with controllable-pitch propellers. It was interesting to learn that fretting was only a minor problem in this type of application, particularly if the inner sleeve was phosphated first.

Commander Goodwin's anecdote concerning the design of shaft G (Fig. 1) was a valuable contribution which supported the authors' speculations on the influence of closely spaced bearings at the ends of such systems. With regard to the use of computer programmes for the calculation of whirling speeds, the authors had used a Pametrada programme based on Prohl's method for their investigations, but, like Mr. McClimont, they had been unable to find any genuine cases of shaft whirling with which to test the results. The special case of whirling of the intermediate shafting was at present only likely to occur in naval ships, and such cases were not generally available to the merchant industry.

The authors had no experience of vibrations of outboard shafting supported by A-brackets and could only offer an opinion on the likely influence of the surrounding water. The effect would be twofold; the effective mass of the shafting would be increased by the entrained water and the motion of the propeller would be resisted by the water and would therefore be damped. The first effect would lower the critical speed, but the damping resistance would change the frequency very little, though it would cause a marked reduction in the amplitude of the vibration near resonance. The authors agreed, however, that model tests would resolve any difference of opinion on this matter.

The reason why system "b" was preferred to system "a", in the torsional vibration design study for an 85 000 dwt tanker, was that with the H.P. and L.P. lines tuned to zero admittance the II-node resonant frequency was still very close to service speed. Had the H.P. and L.P. lines for the tuned system crossed the p^2 axis at a value of p corresponding to a main shaft speed remote from the service speed then there was no doubt that the tuned system "a" would have been preferred.

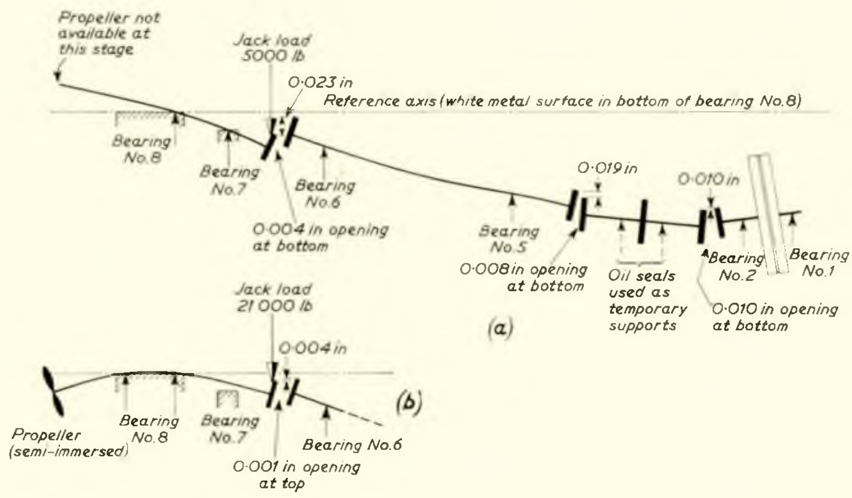
The reference to fitting a Michell resonance changer under "Design Study Twin-screw Diesel Ferry" was correctly worded in the paper but did perhaps require an explanation. This particular gear-box used a main thrust block of recent design, comprising two collars on the shaft and having the ahead and astern tilting pads held in common carrier rings located between these collars. In the original design these collars were too close together to permit a standard resonance changer to be fitted in the carrier ring. The recommendation made was to increase the spacing of the collars and hence the carrier ring thickness, so that a resonance changer could be installed at any future date if required.

The authors thanked Mr. Coats for drawing attention to another potentially troublesome aspect of the "horse-shoe" thrust block. The eccentric thrust vector was fixed in space and as the shaft rotated it produced a cyclic bending moment, at shaft frequency, which acted on the thrust collar. The rocking tendency so produced had, on occasions, led to fretting of the collar where it was clamped by the nut and the shoulder of the shaft. This could only happen when the collar was separate from the thrust shaft which normally meant a thrust block integral with the gear-box and not a separately mounted unit.

Mr. Coats' recollection of calculating gap and sag values emphasized that the principles had changed very little; only the methods used had changed with the advent of digital computers. The relatively low cost and extremely high speed of calculation of these machines made possible a very much more detailed investigation than was possible in the period to which he referred.

At Mr. Coats' suggestion Fig. 31 had been included to illustrate the gap and sag diagram, this particular one being for a 100 000-dwt tanker installation. In this instance the stern-tube assembly had been machined to include slope-boring of the aft most bush prior to the investigation. Also, because of a late change in propeller design the propeller was not available until some months after the shaft alignment was due to be completed. The shafting was, therefore, lined-up as shown in Fig. 31(a), but not coupled, and again checked for gap and sag values when the propeller had been fitted. It would be noted that the aft bush was considered to have two effective points of support for the normal service condition, but only one in the absence of the propeller.

The results of the calculations upon which Fig. 8 was based, were not embodied in the alignment instructions for the shafting concerned. In one case this was because the authors were not concerned with the ship in question and in the other because of an appreciation of the uncertainties included in the



(a) Shaft alignment used to locate bearings Nos. 1-6
(b) Predicted gap and sag at tailshaft flange if propeller had been fitted (subsequently verified before coupling up)

FIG. 31—*Shafting alignment instruction diagram*

calculations. Until such calculations had been confirmed by measurement, preferably at full-scale, they should only be used to indicate possible effects on the shafting and bearings. Subsequent inspection of these elements during tailshaft surveys might then reveal the occurrence of the predicted effects after which possible remedies could be applied. The particular case posed by Mr. Coats related to shafts such as Q, R and S (Fig. 2) which were fitted in cross-channel steamers. His prediction of a large change in shaft slope seemed very likely when the thrust and torque eccentricities for the twin-screw ship in Fig. 8 were considered, but it could be quite wrong to extrapolate in this manner. Unfortunately, although cross-channel steamers had been built at the rate of at least one per year for the last decade, there was still a vast ignorance of the wake profile associated with them. Because of the shallow-draught hull and small-diameter propellers which were characteristic of the channel steamers, it was quite certain to have different general propeller characteristics to the twin-screw vessel shown in Fig. 8. Until wake characteristics were measured for this class of hull and the variable wake analysis carried out for this hull/propeller combination, even the broadest speculation would be very dubious.

The subject of the analogue versus the digital computer raised by Mr. Rose merited a complete paper in its own right and was beyond the scope of the present paper. The authors had knowledge of one analogue computer which was successfully adapted to represent the torsional vibrations of a single reduction set of turbine machinery. Damping was represented by resistances in the form of potentiometers and in order to apply the correct value it was necessary to determine the resonant frequency of the individual elements in the system. The biggest problem in this particular case, however, was the wide range of values necessary to represent the system and all fifteen of the computer's amplifiers were required. To simulate a double reduction set of machinery would require more than fifteen amplifiers and thus the computer would be an expensive piece of equipment if it was only used for torsional vibration studies. The advantage of the method once the analogue was set up, however, was the ease with which the effect of changes within the system could be observed. Nevertheless, the authors felt that the combination of digital computer and admittance diagram was the quickest and cheapest way of carrying out a torsional vibration study at the design stage.

The reference to increased power loss with increased specific loading of plummer block bearings was made assuming a fixed L/d ratio and hence referred to an increase in load on the same sized bearing. This was a necessary consequence of wider bearing spans. The increased spacing, however, gave the extra transverse flexibility which rendered the system less sensitive to hull movements. This in turn meant that load fluctuations were less and higher basic loadings were possible without incurring higher peak loadings. Although the power loss per bearing would be increased, with fewer bearings in the system the total power loss should not be greater and might even be reduced.

Regarding increased L/d ratios, the authors had been influenced by the need on certain occasions to accept specific loading of 120-130 lb/in^2 with bearing spacings of only 10-12 diameters. Assuming that wider spacings would be encountered in the future, one must either be prepared to accept even higher specific loadings or else produce a more suitable design of bearing. While a range of 1·0-1·5 of L/d ratio was given in the paper, the authors' experience suggested that the lower value would be adequate for the immediate future, representing an increase of 50 per cent in load-carrying capacity for a given bearing diameter. While the authors agreed that for high-speed bearings such as those for turbine application a figure of $\frac{2}{3}$ was ideal, they operated under very different conditions to the plummer blocks in question. For the very low shaft speeds and with the use of ring-oiled lubrication (or equivalent) a fairly simple change of bearing surface geometry would give optimum lubrication for the longer bearings envisaged.

Further enquiry on current U.S. practice revealed that tunnel bearings with L/d ratios as low as 1·0 had been supplied for U.S. merchant and naval ships during the last two years. Such a value was extremely low for U.S. practice representing the latest step in a trend to shorter bearings. The authors' opinion that there was a need for a tunnel bearing which was longer than the standard British bearing was strengthened by this development.

The authors agreed that the "all round" and "horse-shoe" designs of main thrust block overlapped for only a small range of shaft diameter, but they still regarded the former as preferable for the reasons stated. Although the deflexion values quoted by Mr. Rose for the two designs were similar, the slopes of the deflexion curves were not equal. A comparison showed the "all round" design to be 55 per cent stiffer than the "horse-shoe" design and this was the basis of the authors' preference. It must be remembered, however, that most ships did not require special attention to the axial vibration characteristics of their shafting, in which case the stiffness advantage was of no consequence and, provided that the eccentric thrust vector was given due consideration, it would be logical to select the cheaper unit.

The thrust-block construction and the use of the white-metal seals as journal bearings should be determined by the location of the unit relative to the main engine. If the thrust was integral with the main engine then white-metal seals were not necessary. If the thrust was separate from the main engine and 6-12 shaft diameters from it, then the white-metal seals could conveniently be used as journal bearings. If, however, the thrust was separate, but adjacent to the main engine, then the seals must not be used as journal bearings and should be omitted or at least reduced in length.

The gear-box deflexion of 0·06 in under normal thrust load quoted by Mr. Rose sounded a most extreme movement. Without knowing the detail design of this box, e.g. plate thicknesses, location of stiffeners, chocking details etc., the authors could offer no useful comment other than that corrective action should be taken.

The deflexion values for plummer blocks given by the authors were all measured on bearings installed in ships. In all cases the measurements were secondary aspects of jacking tests made to determine the bearing loads. In one particular case an arm was welded to the deckhead above the bearing and fitted with a dial indicator touching the adjacent journal section. In this way the movement of the shaft was measured. Other dial indicators attached to the bearing keep and touching the journal section, both fore and aft, measured the relative movement of shaft and bearing housing. From the readings obtained it was ascertained that the bearing housing had little tendency to move relative to the deckhead and that the greatest movement was a shift of the bottom half of the bearing-shell relative to the bearing housing. With further confirmatory measurements made subsequently on other bearings the authors felt that it was time for a programme of deflexion tests on standard journal bearings to be started, similar to those previously done by Pametrada and B.S.R.A. on main thrust blocks.

The authors thanked Mr. Atkinson for his comments on the basis of the calculations summarized in Fig. 8 and would add that his efforts and those of his colleagues in pursuing this type of inquiry were praiseworthy in every way in spite of the criticism voiced during the discussion. The authors sympathized greatly with the difficulties which presented themselves in the form of inadequate knowledge of wake forms for particular types of hull, necessitating assumptions which must greatly influence the results of the calculation. The main purpose of including Fig. 8, however, was to illustrate the complex happenings at the propeller and in particular to demonstrate the variations in bending moments which were applied to the tailshaft, in contrast to the more common simplifying assumption of a fixed percentage variation in thrust and torque made when estimating the axial and torsional vibrations of the shaft system. It also indicated the kind of calculation which, with the advent of high-speed digital computers, was now possible, provided the necessary data were available. It was the authors' hope that, with this knowledge, more attention would be given in the future to the measurement of wake profiles, both by shipbuilders and shipowners.

It was interesting to note that Mr. Brown used a loss-less transmission line as the electrical analogue of the main shafting. This technique was used by H. G. Yates at Pametrada at least 20 years ago and was described by one of the present authors in the correspondence to reference[34]. This technique was useful when calculating the axial frequency longhand, but it lost this advantage once a computer programme such as that given in the Appendix was available. It was also difficult to include the damping associated with the shafting when using the transmission line method to estimate the vibration amplitudes or thrusts.

The method of estimating entrained water at the propeller given by Burrill and Robson was technically superior to using a fixed percentage allowance and, for torsional vibration calculations, the authors used values given by Burrill's method. For axial vibrations, however, there was little justification in using such a refined approach until the stiffness of the thrust block and its seating could be estimated more accurately than was possible at present.

With reference to the design study of the axial vibrations of the twin-screw passenger liner (shaft G), Mr. Brown had put forward a case for fitting a four-bladed propeller based essentially on stiffness values very much in excess of those assumed in the original study. In a previous case of a twin-screw passenger liner (shaft E) every effort, within economic limits, was made to produce the stiffest seating possible. The thrust block was moved two shaft lengths aft, which placed it well down the shaft tunnel, and the seating was stiffened. The result was a measured combined thrust block and seating stiffness of 8000 tons/in, compared with figures of 6000, 4100, 3510 and 4090 measured on four other large turbine-powered passenger liners[9]. Table X summarized the range of combined thrust block and seating stiffness measured on a number of large American ships[33] and again the upper limit was only 5800 tons/in. In view of these measured values the authors considered it extremely unlikely that a combined stiffness of 13 000 tons/in, as suggested by Mr. Brown, could be achieved in the case of shaft G. Even with such a figure for the combined stiffness and with a four-bladed propeller fitted, the system could not be considered satisfactory. Table XI gave

TABLE XI

PROPELLER-EXCITED AXIAL VIBRATIONS SHAFT G FITTED WITH FOUR-BLADED PROPELLER

Combined thrust block and seating stiffness, tons/in	Resonance		Conditions at thrust block at 180 rev/min	
	Frequency, v/min	Mainshaft speed, rev/min	Amplitude, $\pm$ in 10^{-3}	Thrust variation, $\pm$ tons
12 000	776	194	5·6	67·5
13 000	786	196·5	4·7	60·7
14 000	796	199	4·0	55·6

the results of calculations made for shaft G and, though the resonant frequency lay 10 per cent above the maximum shaft speed, the thrust variation at the thrust block at maximum speed was approximately $\pm$ 60 tons, just twice the limit proposed by Couchman[9]. Table XI also indicated how sensitive conditions at the thrust block were to changes in the stiffness in this range.

In reply to Dr. Milne, the authors had stated in the paper that the detailed design of the shaft system occurred fairly late in the development of a new ship but even so this was still usually well before a contract was placed. Nevertheless, the authors were in full agreement with Dr. Milne on the use of computer techniques at the earliest design stage and indeed had been greatly encouraged by him and his company in developing certain parts of the work reported in the paper. They were also in agreement on the need for revision of all aspects of stern-gear design to meet the needs of present day ships rather than merely scaling-up from past practice.

The avoidance of "fantail" vibration by adopting lower propeller speeds and fewer propeller blades was an additional factor to be considered when choosing a particular propeller. Whilst this approach was possible with steam turbine ships and medium-speed Diesel installations, it would not be very practical with direct-coupled Diesels due to their consequent loss of power at the lower speed, unless an oversized engine was installed.

The authors had not been concerned with axial vibration studies for ships powered by direct-coupled Diesels, but were aware of the need to consider a second source of excitation—the engine itself via the flexing of all the cranks. In view of the complications which arose when only propeller excitation was present, the authors were quite happy to leave this further complication to the designers of Diesel engines.

Alignment, as Dr. Milne had stated, was generally done following launching and while in the light-ship condition, although some work had been done on alignment prior to launching. In either case, movements occurred in the hull at full load, making the wider spacing of tunnel bearings essential to render them less sensitive to these movements. Extreme changes in bearing support conditions such as Dr. Milne reported would then be avoided. The marine industry was still collecting information relating to hull movements and it was hoped that sufficient would be known in the near future to enable likely changes to be predicted for a given class of ship.

Mr. Harper and other contributors had commented on the reported rejection of hollow shafting for recent passenger liners. This was not an expression of the authors' views, but an observation of events. Transverse stability investigations were clearly the concern of the naval architect and the problem was to re-design around this particular aspect. Assuming this could be done successfully, one must then face up to the extremely high cost of hollow shafting. This was largely the result of the current method of manufacture whereby the forgemaster pierced the billet and then hammered it out on a mandrel. It was the authors' belief that hollow shafting would be pro-

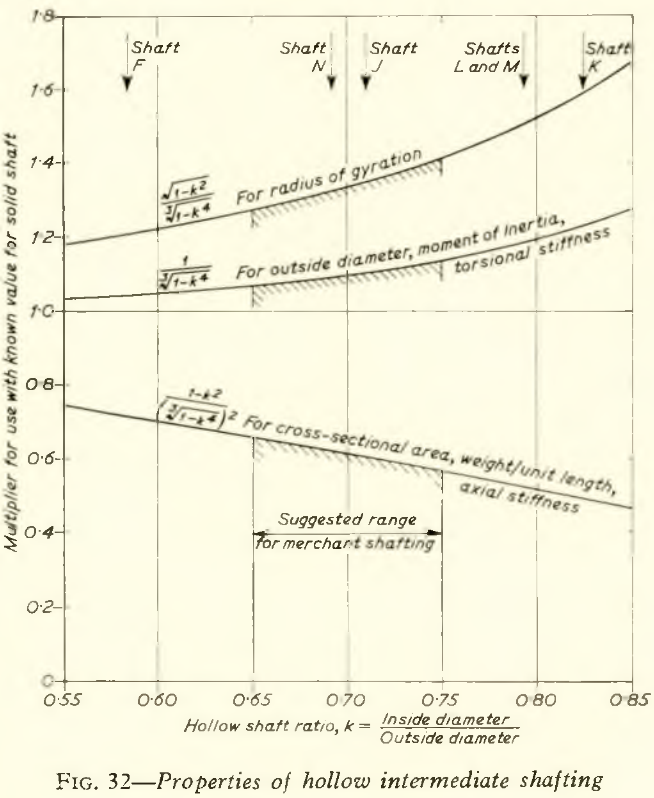


FIG. 32—*Properties of hollow intermediate shafting*

duced at an attractive cost only when high-speed trepanning was used, this observation being based upon their previous experience of the method of producing medium-size gun barrels. Because of the high cost of suitable trepanning equipment the facilities would necessarily be restricted to one or two centres in the U.K. and a very limited number of bore sizes used. For example, the range of bore sizes from $10\frac{1}{2}$ to 21 inches could be adequately covered by only six trepanning heads yet permitting the ratio of inside diameter to outside diameter to be maintained in the range 0·65 to 0·75. In this way most of the advantages of hollow shafting could be obtained while keeping tooling costs to a minimum. The consequent change in the various properties compared with those for the equivalent solid shaft were shown in Fig. 32. Further comments on the subject of longer tunnel bearings had been included in the reply to Mr. Rose.

With regard to the reduction in the number of tunnel bearings the authors felt that the owners' views were reflected in their increasing adoption of these measures.

The principal objection to the use of reversible thrust shafts was in the context of separate thrust blocks placed 6-12 diameters down the tunnel. To do this and keep the thrust shaft reversible required an additional very short shaft length to connect the thrust to the main engine and this in turn meant a further flanged coupling. If the extra length required was added to the thrust shaft, this additional coupling was eliminated but the thrust shaft was no longer reversible.

Mr. Adams' comments on "fantail" vibrations of ships having direct-coupled Diesel engines installed aft related to a fairly general experience and were complementary to Dr. Milne's contribution on this subject. Mr. Adams had suggested that the solution to the problem was in improved hydrodynamic shape of the hull at the aft end, whereas Dr. Milne was more interested in the hull as a structure having many local resonant frequencies. As Mr. Adams had stated, the sources of the vibrations were the propeller and the main engine. Improved hydrodynamic flow would reduce the former and substitution of turbines for Diesels would eliminate the latter. Recently developed variations of the Hogner stern and the asymmetric stern seemed to reduce the wake variations and claims were also made of improved propulsive efficiency which the authors were inclined to accept.

In reply to Mr. Langballe, the reference to first order whirling was only a convenient approach to illustrate a particular point and not a statement of the authors' experience of the relative importance of the different orders of whirling. The authors had experience of trouble similar to that described by Mr. Langballe in the case of shaft D (Fig. 1). In this case the first indication of a problem was also a catastrophic failure and, similar to Mr. Langballe's experience, an improvement was noted after re-wooding.

The difficulties in obtaining reliable measurements which led ultimately to the amplitude-speed curve of Fig. 25 were appreciated only too well. Jasper's original papers were developed in the context of naval shafting with proportions generally similar to those of shafts J, K and N (Figs. 1 and 2) which were characterized by very widely spaced bearings supporting the tailshaft and very light propellers. While Jasper evaluated the stiffness of bearings and their supports for this type of configuration, he was very much aware of the need to treat merchant shafting as a quite different case and advocated parallel investigations in this field, Mr. Langballe's contribution was, therefore, particularly welcome in helping to extend this field of knowledge.

The authors noted with interest the use of slope boring in the original installations for the ships concerned in Figs. 25 and 26. It was their belief that this practice should be avoided in the case of white-metalled bearings as well as wooden bearings, since other methods of achieving the same result were now available. The aim was to produce a bearing bore matching the slope of the deflected tailshaft and so use its full length to support the tailshaft assembly instead of having a heavy concentration of loading at the outboard end. This could be done in several ways e.g.:

1) sloping the bore of the bearing relative to its outside diameter (i.e. slope boring);
2) putting the same slope correction in the stern frame bore instead of 1);
3) tilting the shaft system so that its deflexion curve in the stern tube region was parallel to the bearing bore.

A complicated boring set-up was required to produce a white-metalled bearing bush satisfying 1) and this must be repeated for any replacement bearing subsequently fitted. The second alternative amounted to slope boring the ship itself, and was readily done during construction by elevating the boring target an appropriate amount. A concentrically bored bearing bush was then used throughout the life of the ship. The third method involved dropping the tunnel bearings below the axis of the stern-tube bush so that the weight of the intermediate shafting was applied to the tailshaft assembly, causing it to tilt until contact was established along the full length of the stern-tube bush. This method also had the advantage of requiring only a concentric bush in the stern tube. Methods 2) and 3) also obviated the need for an adjustable stern tube, the adjustment in the latter case being made on the shaft itself.

With reference to Fig. 27, the six-bladed propeller might cause the higher order critical speed to occur in the normal speed range, but it would be accompanied by a useful reduction in excitation as discussed in section 2 of the paper.

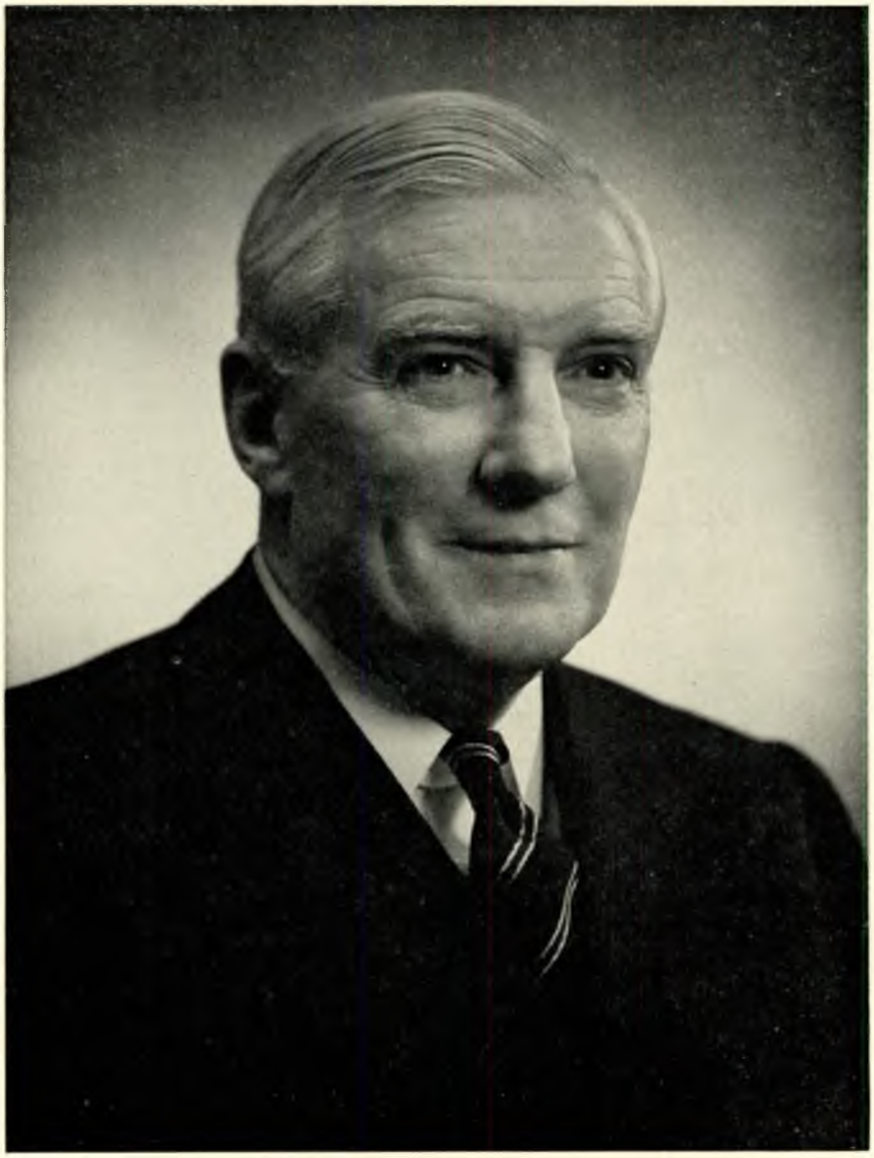
The authors were in full agreement with Mr. Langballe's comments concerning the need for continued work in the field of whirling of tailshafts in merchant ships and the consideration that it now ranked equally in importance with studies of torsional and axial vibrations.

The behaviour of shafting systems under crash-stop conditions referred to by Mr. Attwood was, fortunately, of minor interest because of the relatively short period of the ship's life involved. Almost all of the modes of failure, arising because of shafting vibrations, involved a fatigue mechanism of some kind and therefore required the condition to persist for a long period. The effects of pitching, however, were present for a considerable part of the life of the ship and although, to the authors' knowledge, this aspect had not previously been investigated, it deserved close study. Rather than considering accelerations, it would seem best to think in terms of an applied vertical velocity which varied cyclically with a period of about six seconds. This vertical component would then modify the inflow to the propeller producing an additional cyclic change in thrust and torque, but at a relatively low frequency (approximately one fortieth propeller blade frequency) and should, therefore, be of minor importance. Much more important under these conditions was the relative performance of Diesel engines and geared turbines where the much higher inertia of the latter caused a much smaller fluctuation in propeller speed which, in turn, permitted a higher propeller speed to be maintained with safety.

Once again the authors were grateful to their colleagues at Pametrada for assistance in preparing these replies and in particular to Miss M. Scott for her continued efforts at the typewriter.

REFERENCES IN REPLY

32) VAN MANEN, J. D. 1965. "Door de schroef apqewekte scheepstrillingen". *Schip en Werf*, Vol. 32, p. 346.
33) STERNLICHT, B. *et al.* 1959. "Review of Propeller Shaft Thrust Bearings". *A.S.N.E. Jnl.*, Vol. 71, p. 277.
34) JOHNSON, A. J., and MCCLIMONT, W. 1960. "Axial Vibration in a Single-screw Cargo Ship". *Trans.I.E.S.S.*, Vol. 104, p. 13.
35) HECK, J. W., and BAKER, E. 1963. "Marine Propeller Shaft Casualties". *Trans. S.N.A.M.E.*, Vol. 71, p. 327.
36) PRICE, R. L., NOONAN, E. S., and FELMAN, S. 1962. "Bending and Torsional Stresses in Propeller Shafting of U.S. Observation Island (EAG 154) in Smooth and Rough Sea". David Taylor Model Basin Report No. 1596.



VICE-ADMIRAL SIR FRANK MASON, K.C.B.

VICE-ADMIRAL SIR FRANK MASON, K.C.B.

Admiral Mason was educated at Ipswich School and entered the Royal Navy as a Special Entry Cadet in 1918. While still a Midshipman, he elected to specialize in engineering and, after two years' service afloat, at the end of which he was awarded his Engine Room Watchkeeping Certificate as a Sub-Lieutenant, he studied engineering at the Royal Naval College, Greenwich, and the Royal Naval Engineering College, Keyham. After completing these courses, he served another commission afloat as a Lieutenant and was then appointed to the Engineering Department of H.M. Dockyard, Malta.

On leaving Malta three years later, he was appointed to H.M.S. *Rodney*, in connexion primarily with her 16in triple gun mountings. For the next twenty years his career lay chiefly in the field of gunnery engineering, although the period included two commissions afloat, one as Senior Engineer of H.M.S. *Rodney*, the other as Engineer Officer of H.M.S. *[illegible]* the Flagship of the Rear-Admiral Commanding the Destroyer Flotillas of the Mediterranean Fleet.

In the early part of the Second World War, he was appointed to H.M.S. *Excellent* for ordnance and instructional duties; this was the first time that an Engineer Officer had served on the instructional staff of this famous establishment.

Appointed Fleet Gunnery Engineer Officer, Home Fleet, in 1943, he came into direct contact with the war-time problems of maintaining the efficiency of the armament under such arduous conditions as those experienced in escorting the Russian convoys.

In the following year he was promoted to the rank of Captain (E) and appointed to the Naval Ordnance Department at Bath, then preparing for Operation "Overlord" and the support of the future British Pacific Fleet. This was followed by a special appointment concerned with the reorganization of the training of artificer apprentices, resulting in a system of training which is still in force. Admiral Mason then became Chief Gunnery Engineer Officer and Deputy Director of Naval Ordnance (Material), after which he studied at the Imperial Defence College, being the first Engineer Officer to do so.

In 1950, he was promoted to the rank of Rear-Admiral (E) and became Deputy Engineer-in-Chief at Bath. This was followed by an appointment on the staff of the Commander-in-Chief, The Nore, which again brought him into direct contact with the problems encountered afloat. Finally, in April 1953, he was promoted to the rank of Vice-Admiral and became Engineer-in-Chief of the Fleet, which appointment he held until September 1957 when he was placed on the retired list.

He was made a Companion of the Most Honourable Order of the Bath in 1953 and a Knight Commander of the same Order two years later.

Admiral Mason holds a number of public offices. He is Chairman of the Steering Committee of the National Engineering Laboratory and a member of the Steering Committee of the National Physical Laboratory. In connexion with the latter, he is Chairman of the Froude Committee which is responsible for the policy of the Ship Division of the Laboratory.

He is a Director of H. W. Kearns and Co. Ltd., Power Jets (Research and Development) Ltd., Hilger and Watts Ltd., and English Electric Diesel Engines Ltd. and its subsidiaries, D. Napier and Son Ltd. and Ruston and Hornsby Ltd.

He was a member of the former Council for Scientific and Industrial Research and the Governing Body of the National Council for Technological Awards.

He is a Freeman of the City of London, a Liveryman of the Worshipful Company of Shipwrights and a member of the Smeatonian Society of Civil Engineers. He is a past Chairman of the Council of the Institute which he represented on the Engineering Institutions' Joint Council, now the Council of Engineering Institutions, serving as Chairman of the Education and Training Committee.

He is a Past President of the Institution of Mechanical Engineers.

Admiral Mason is a member of the Governing Bodies of Ipswich School and the Royal Naval School, Haslemere. Quite recently he has been appointed by the Corporation of Ipswich to the ancient office of High Steward of the Borough.