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PRACTICAL APPROACH TO SOME VIBRATION AND MACHINERY PROBLEMS IN SHIPS

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The Engineering Research Department of Lloyd's Register of Shipping enjoys the privilege of having presented to it for investigation, with world-wide terms of reference, the more abstruse problems in connexion with shipping. Some 150 investigations are dealt with annually. They naturally cover a very wide field indeed, but there are many problems common to a large number of vessels, wherever built or engined, and some of these have been chosen for inclusion in this paper with a view to disseminating such first hand and well tried experience as may be helpful in avoiding repetition of similar troubles in other existing or proposed vessels.

Vibration problems relating to hulls and machinery are by far the commonest form of trouble dealt with, and several examples are included. Certain aspects of vibration problems in ships are considered, and how these problems are recognized and dealt with in existing ships is indicated. Design stage recommendations are also made. Propelling machinery, shafting and propellers are treated from the aspect of vibration, and a few recurrent troubles, not necessarily directly associated with vibration, are included as a matter of practical interest. Notable amongst these is the technique used for the successful remedy to top end bearing trouble in large oil engines.

The paper is in two parts, the first dealing with the hull and the second with propelling machinery.

HULL VIBRATION

Although much work has been done on the problems of hull vibration, and many excellent papers and articles published, it does not appear that the practical aspects of the problem are appreciated as widely as they might be. This is not meant in any way to be a facetious statement but, as it is most important that hull vibration problems be understood in their broader implications for an appreciation of what is referred to later in this paper, the main principles must be dealt with and appreciated before proceeding further. The basic conceptions are usually simple and straightforward and, if they are grasped, the whole field of hull vibration snaps into focus with relief and clarity.

(1) All ships built do, in fact, have many vibration criticals. It is possible to produce vibration of a severe order in practically any hull, given the necessary pulsating force at the required number of pulsations per minute. The vibration may be in the vertical or transverse plane or it may be torsional or of a highly localized character.

(2) There may be up to four, or occasionally more, natural frequencies of the hull in each plane which may be excited in the speed range—these are known as critical frequencies.

(3) For any type of vibration involving the length of hull, the ends of the vessel, i.e. stern and bow, will be positions of large vibration amplitude. There will also be other positions along the length of the vessel where no vibration will be felt, and intermediate positions where vibration may be of appreciable magnitude. These intermediate points of no vibration (nodes) and of large vibration (anti-nodes) will vary both in number and position as the various types of vibration (modes) are passed through. Fig. 1 gives a schematic idea of what might be experienced in a 450-ft. cargo vessel when excited into vertical and transverse modes of vibration.

(4) For the majority of ships a fairly narrow range of service speeds is used in operation and it is necessary that, under the varying conditions of loading that may be encountered in service, no serious hull vibration criticals are present in this speed range.

(5) The exciting forces, i.e. the pulsating forces applied to the hull and causing hull vibration, are usually from the propelling machinery or the propeller.

(6) The lower modes of vibration, i.e. the 2-noded modes (vertical and transverse) are most commonly excited by primary unbalance forces or couples in the machinery, or by a damaged or badly pitched or badly balanced propeller. The frequency of the vibration will coincide under these conditions to exactly once per revolution of the shafting.

(7) The higher modes of vibration, i.e. 3-noded modes and above, may be excited by the following:—

(*a*) Secondary unbalance (forces or couples) in reciprocating machinery.

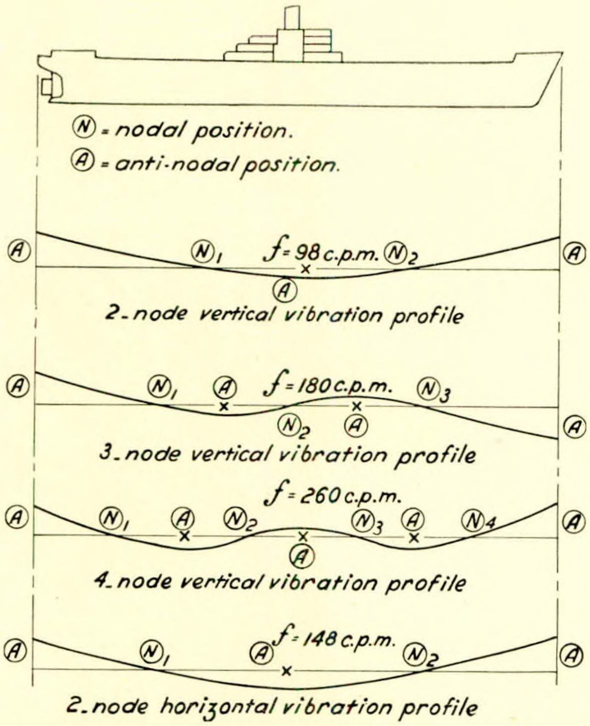

FIG. 1—*Hull vibration profile: cargo vessel*

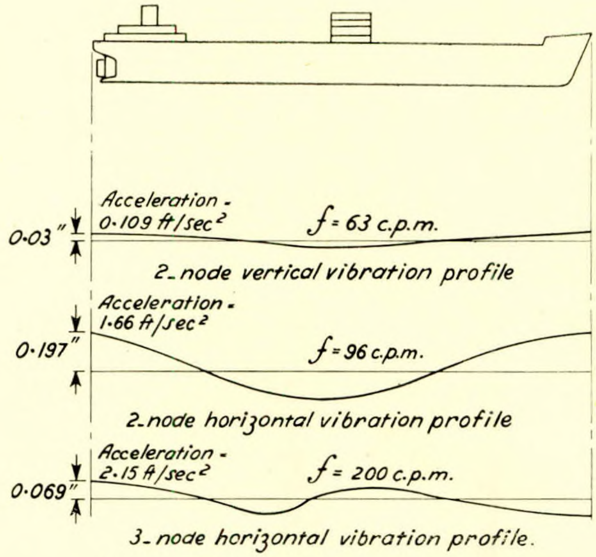

FIG. 2—*Hull vibration profile: tanker*

$$\text{Acceleration (ft./sec.}^2) = \frac{\text{Single amplitude of vibration, in inches}}{12} \times \left(\frac{2\pi \times \text{frequency}}{60}\right)^2$$

(Frequency in cycles per minute.)

(*b*) Propeller—at propeller blade frequency, i.e. number of blades × shaft r.p.m.

(*c*) Gas forces or reactions in the main engine.

(8) The amplitudes of vibration are often quite large. For the 2-noded mode of vibration in, say, a 500-ft. vessel, an amplitude of $\pm \frac{3}{4}$in. at the stern or bows is not abnormal when excited by the primary unbalance in the unbalanced or partly balanced main reciprocating engine. But amplitudes are not necessarily, in themselves, a criterion of the severity of the vibration, both from the subjective point of view, i.e. its effect on personnel, and from the damage effect on the local structure of the vessel or instruments and other gear attached to the structure. Acceleration, which considers amplitude and frequency, of the local vibrating structure of the vessel, possibly gives a clearer indication in deciding whether a vibration is acceptable or not. Here it must also be borne in mind that a vibration occurring just outside the running speed in a new vessel will, in the course of time, be lowered in frequency and may coincide with the running speed. It is obvious that hull vibrations occurring appreciably below the running speed are of little consequence unless they are very severe.

(9) The foregoing remarks relate to hull vibration criticals under the normal conditions of deep water, i.e. at least five × draught of vessel, under the keel. In very shallow water the various critical frequencies and amplitudes, particularly for the vertical modes of vibration, may be severely affected. The transverse modes are likewise affected in restricted waters such as the Suez Canal. These are, however, abnormal conditions of operation, usually at greatly reduced revolutions, and must be accepted as such, even though with propeller-excited modes of vibration, the exciting forces may be considerably increased. Astern running can, in certain circumstances, produce heavy vibration of the hull, but here again prolonged running does not occur, and the hull critical can be avoided by alterations to speed.

(10) Referring again to accelerations (paragraph 8 above), it will be found that the higher modes of vibration are usually propeller-excited and, although producing relatively small vibration amplitudes, may be of considerable annoyance and may cause damage to the structure or attached equipment, the resulting accelerations being quite high (Fig. 2).

(11) At this stage, only the principal modes of hull vibration have been considered, i.e. treating the hull as a floating beam set into vibration by a pulsating force. Fig. 2 gives a series of hull vibration profiles obtained from a tanker. A consideration of the actual stresses set up in the hull, assuming no stress raisers or discontinuities, would indicate that stresses caused by the elastic deflexion of the hull due to the vibration are normally insignificant when compared with the hogging and sagging stresses due to the loading and buoyancy distribution and the dynamic stresses which occur in a sea-way. The normal deflexions from hogging to sagging may amount to 1 inch per 100 feet length of hull, which means that in a 500-ft. vessel 5 inches may be reached.

(12) No mention has yet been made of the considerable dynamic magnification effects produced by resonance in local structures. The overhung bridge space on certain tankers is one which lends itself to quite severe vibration amplitudes excited by what may be almost imperceptible vibration on the strength deck. The frequency is commonly in the 400 c.p.m. range of propeller-excited hull criticals. Certain precautions can be taken in order to reduce the dynamic magnification due to local resonance. Masts, samson posts, piping, boilers, etc., and other decks and structures attached to the hull structure of a ship, come under this category.

(13) The transverse rocking modes, particularly associated with large oil engine installations, can also cause local vibration. The problem of engine seatings is a complex affair insofar as it is difficult to predict with usable accuracy the critical frequencies of the transverse rocking modes of vibration of the machinery about its seatings. Two approximate methods of calculation are given in references [1] and [2]. The exciting forces for this mode of vibration are usually the transverse guide reactions of the cylinder gas loads. It will also be appreciated that engine forces in the vertical plane may also be considerably magnified if the structure supporting the engine is such as to produce resonance in the running range. Here the engine may be considered as being mounted on a flexible diaphragm attached to the framing of the vessel. Fractures have occurred in frames and brackets attaching the tank top structure to the structure of the hull[3].

(14) Resonance of the rudder about the rudder head,

excited into vibration by the propeller race, has been known to be a source of hull vibration, where this critical frequency coincides with one of the principal modes of hull vibration[4]. A rudder vibrating at the service speed, whether this coincides with a hull critical or not, will be a source of constant trouble, producing frettage and wear in the trunnion bearings, hammering of the gearing (if gear driven) and rapid wear of pintles and the structure of the rudder itself.

(15) In the case of twin screw vessels, the same remarks apply generally. There is the qualification, however, that for engine- and propeller-excited vibrations in twin screw machinery, the vibrations build up and die away as the machinery or propeller forces causing the vibration run in and out of phase, as one set of machinery will run slightly faster or slower than the other, a condition which is unavoidable without special synchronizing gear. This "beating" effect is, if anything, rather more trying to personnel than a sustained vibration.

(16) Torsional vibration of the shafting has nothing to do with hull vibration.

APPRAISEMENT OF THE VIBRATION PROBLEM

EXISTING SHIPS

Any vibratory condition on board ship has to be considered from one of two aspects. The first, whether the vibration is so severe as to constitute a danger to the hull structure or equipment attached thereto, irrespective of where the vibration occurs in the speed range. The second, whether the vibration occurring in the service speed range is of sufficient nuisance value to personnel that it must be dealt with, in spite of the fact that the accelerations produced are small enough not to constitute a hazard either to the structure of the vessel or to attached equipment. It is possible, however, in most cases either to reduce the accelerations to negligible proportions, in which case the position of the critical in question in the speed range is of no consequence, or to reduce the acceleration to an acceptable value and to remove it from the service speed in cases where a further reduction of the acceleration might be a very costly or impracticable procedure. In any case, before proceeding with the remedy, it is necessary to know the frequency, the accelerations at the worst antinode (usually the stern of the vessel) and, also, the positions of the nodes, i.e. a vibration profile of the hull (Figs. 1 and 2). This, of course, will apply to all the criticals occurring in the service speed range at the various draughts which represent the normal service conditions of the ship.

The problem is then broken down into engine- and propeller-excited criticals, which are dealt with as shown later in the paper.

The necessary data may be obtained by using instruments. Intelligent observation can, however, go some way towards a determination of the frequency, mode of vibration and probable cause, and is certainly worth considering.

Both methods are detailed below.

Observations (for any particular draught)

The following facts should be established by careful observation:—

(*a*) *Engine revolutions at which the vibrations complained of are at a maximum.* This is often fairly straightforward as vibration criticals are usually sharply tuned and well defined. A very slow run-up through the speed range is necessary and it is advisable to come back down the speed range and re-check the first observations of each critical speed which corresponds to those speeds at which the vibration is worst.

(*b*) *Direction of the vibration, i.e. whether it is in the vertical or transverse plane.* Good positions to observe any of the principal modes of hull vibration, which are the commonest and most strongly excited, are in the steering flat and forecastle. Telephone communication is frequently provided from the steering flat either direct with the engine room or via the bridge to engine room and to the forecastle head, which is helpful in keeping track of the r.p.m.

(*c*) *Frequency of the vibration.* At each of the critical speeds the number of vibrations per minute should be counted, preferably against a stop watch. It is quite possible to count up to 500 per minute by using the simple device of stabbing a pencil dot on a piece of paper for every ten vibrations. The engine revolutions must, of course, be maintained as steadily as possible and there should be little or no movement on the rudder. Fairly perfect weather conditions without swell should be chosen for these tests.

(*d*) *Rudder vibration.* A note should be made of the speed at which any hammering of the steering gear occurs.

(*e*) *Tailshaft vibration.* The speed at which hammering of the tailshaft occurs in the stern bearing should also be noted.

With this information it is possible to recognize the cause of the vibration and to say whether the vibration is propeller-excited or engine-excited, also whether the rudder is resonant and contributing to the vibration.

The table below gives for convenience the major forces which either may be expected to excite the principal modes of vibration of the hull or may set up local vibration. The frequencies are multiples of "N"—the r.p.m. of the main machinery.

TABLE I.—EXCITING FORCES

Propeller		Machinery	
1×N	Due to lack of balance or blade out of pitch	1×N	First order unbalanced forces or couples in reciprocating engine or serious unbalance in main wheel in the case of geared drives
2×N	Badly pitched blades		
3×N	With 3-bladed propeller	2×N	Secondary unbalance forces or couples in reciprocating engine
4×N	With 4-bladed propeller		
5×N	With 5-blaαed propeller	$\frac{n}{2}\times N$	Where n=number of cylinders in a 4-stroke engine or 2-stroke engine
6×N	With 3-bladed propeller	$n\times N$	

Higher orders have been known to set up vibration but mostly of a localized nature. In cases where speed variations of the main machinery do not affect the vibration—which will be of a localized form—the source can sometimes be traced by shutting down various auxiliary machinery, starting with the auxiliary generators one by one, and observing what happens after each attempt. This can often best be done in port with a quiet ship. Dealing with high frequency local vibration, however, can be a fairly involved problem, requiring precise vibration equipment for indicating the relative phase and amplitudes at various positions in the local structure. It frequently happens, particularly in larger superstructure vessels having a number of decks, that local vibrations at some engine or propeller order are excited locally in the vessel at a speed which does not correspond to that at which a principal mode of vibration of the hull is excited. It is important to know this, and the methods described above should enable local critical frequencies to be determined. Obviously, if the vibration is not a local vibration but merely an anti-node of one of the principal modes for the hull, then local stiffening will not provide the best solution, which will in fact be one of dealing with the exciting force. If the vibration is local, however, then local stiffening may be the best solution; but the extent and positioning of stiffening may have to be decided as a result of fairly precise measurement with instruments, as suggested above.

In the case of tankers in particular, where it is possible by means of quite small modifications to the ballasting to eliminate from the running speed what might be a serious vibration, it is necessary to know the positions of the nodes and anti-nodes along the hull, before such modifications can be recommended with confidence. In order to do this, the critical engine revolutions for the particular hull vibration should be set and accurately maintained throughout the test now to be described—and here again there should be good weather conditions and little or no movement of the rudder. Starting from aft and working forward, the positions of zero vibration and of maximum vibration should be chalked on the strength deck as these are observed.

This procedure sounds both complicated and impracticable, but it is surprising how good a vibrometer is the human body—it will often be possible to get agreement to within ± 5 feet from two or three independent observers, provided the weather conditions are favourable for the test. With pitching and rolling of the ship, a test of this sort is quite impracticable, as the vibration characteristics of the hull are varying from moment to moment, even neglecting the simple fact that it is, practically speaking, impossible to maintain steady r.p.m. under such conditions. The procedure of using ballast to tune out a vibration critical is referred to in the next column under the heading " 'Tuned' Ballasting".

Measurements

It is not proposed to describe the instruments which are commercially available for recording hull or local vibrations, as they are now widely known and of great variety. Most of the well known scientific instrument makers have their own particular models on the market—mechanical or electronic or a combination of both.

The *modus operandi* is similar to that already described, but with many refinements as one would expect. Records can be taken simultaneously in either plane from a number of positions distributed throughout the length of the vessel, which will enable vibration profiles for the hull to be plotted out for each mode of vibration with a minimum of time and effort. A timing trace and revolution marking trace enable a precise determination of the critical r.p.m. and relative phases to be made.

Twin Screw Vessels

Broadly speaking, the procedure in the case of twin screw vessels is identical, with the complication, however, that some positive means should be available for indicating that the engines are either synchronized or have such a very small speed difference as will enable sufficient time for the vibratory amplitudes to build up. The hull vibration will be observed to build up to a maximum and then die away as the propellers run in and out of phase, and for the purpose of any hull vibration tests it will be seen that it is necessary to keep the engines synchronized to produce their maximum excitation for as long as possible.

A very crude but simple device which was used with remarkable success for operating the machinery in near-synchronism is shown in Fig. 3. The idea was to keep the pointer on the card as stationary as possible. With a little skill in operating one manœuvring valve for setting the speed, and the other as a synchronous slave, it was possible to keep the vessel vibrating steadily for long periods at a time on a dead straight course.

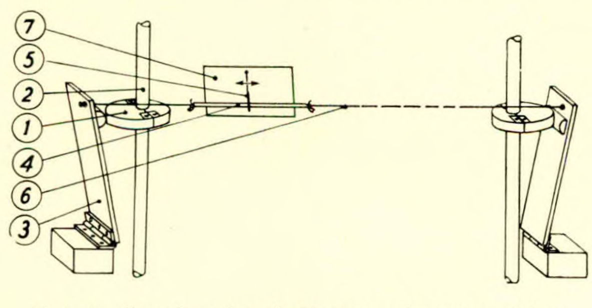

FIG. 3—*Synchronism indicator twin screw drives*

Hardwood eccentrics (1) rigidly secured to tachometer drive shafts (2). Hinged plywood levers (3) with a bakelite rubbing strip are held in contact with the eccentrics by a fishing line (6) and a length of aeroplane elastic (4). A needle pierces the elastic at its mid length and serves as a pointer against a stiff card (7). When the two engines are in synchronism, the pointer is stationary.

"Tuned" Ballasting

In tankers a vertical or transverse hull vibration critical in the region of the service speed can be "tuned" right out of the range by adjustment of the ballast. To do this it is necessary to know the nodal position in way of the last three tanks just forward of the poop. Careful observations of the positions of the maximum vibration amplitudes (anti-nodes) may be sufficient for this purpose.

Then, adopting the simple expedient of placing weight on an anti-node to lower the frequency, and removing weight from an anti-node and placing it on a nodal position for raising the critical, a frequency variation corresponding to several revolutions can be obtained by concentrating on these three tanks alone. Fig. 4 serves to demonstrate this fact simply.

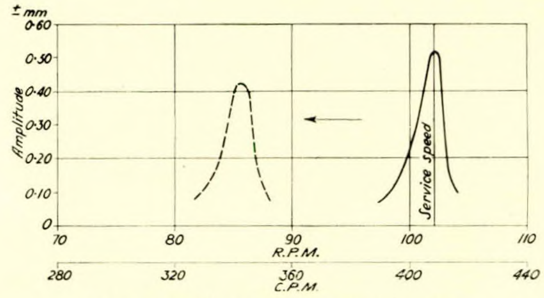

FIG. 4—*Ballast tuning of vibration*

Ballast condition

Normal	Tuned
3, 5, 7 centre and wings full	3, 5, 7 centre and wings full
	8 wings full
9 centre full	9 centre two-thirds full

A strong propeller-excited critical on the running speed in the loaded condition of a large tanker is a more serious matter, as little or nothing can be done with ballasting. Here a change of revolutions may be necessary. As a temporary measure, cropping the propeller blades up to 7 per cent of the diameter will serve two purposes, the first being that the engine speed may be increased in approximately direct ratio with little or no loss in efficiency, the second being a reduction in the exciting forces due to increased aperture clearances. Whether the arrangement may be regarded as temporary or permanent will depend on the propeller, i.e. whether it will be found to have developed cavitation erosion or not at the six months' guarantee drydocking. But this fact may be predicted with some certainty from an examination of the propeller's cavitation performance from the design data and from observed results when at full power in a quiet seaway. Aural evidence of cavitation is fairly recognizable.

The effects of the ballasting of the peak tanks are usually not of great value as they are small when compared with the cargo tanks, and in any case it is not convenient often to run with ballasted peak tanks.

The cargo vessel does not readily offer itself for tuned ballasting owing to widely varying cargo loading commitments. In any event, the foregoing remarks are qualified by the fact that the load distribution must be watched when considering a tuned ballast in order to avoid excessive longitudinal stresses in the hull and, in order to achieve the tuning effect of the modified ballast arrangements while keeping longitudinal stresses low, it may be necessary to load up or unload as the case may be, corresponding anti-nodal and nodal positions giving either an overall deeper or lighter ballast, preferably a deeper ballast than normal.

Reducing Propeller-excited Vibration

Fig. 5 gives recommendations for minimum aperture clearances, which have been found in practice to produce for any existing propeller and hull form, minimum excitation of the hull and rudder by propeller forces. It follows that while the propeller itself and the lines of the after body of the hull and rudder are the important factors influencing propeller forces, it is suggested that a ship which is subjected to severe propeller-excited vibration will most probably be found to have

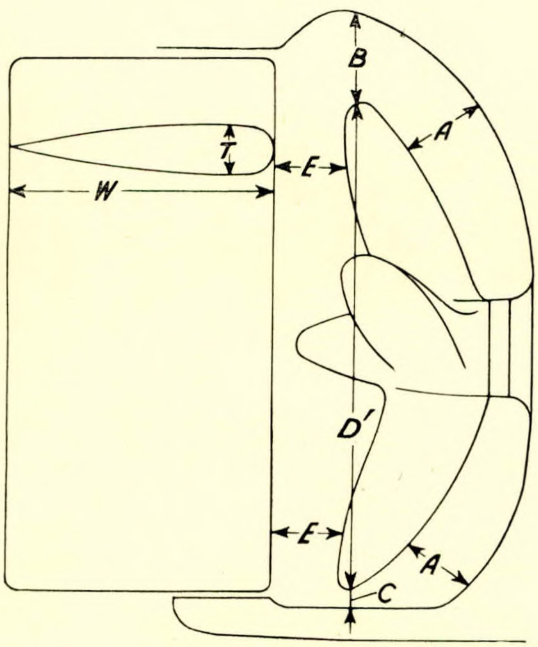

FIG. 5—*Single screw aperture clearances*

Position	Clearance
A =	$0{\cdot}15 \times D$ feet
B =	$0{\cdot}10 \times D$ feet
C =	$0{\cdot}03 \times D$ feet
E =	$0{\cdot}08 \times D$ feet or T feet, whichever is the greater(17)

aperture clearances less than those indicated in Fig. 5.

In an existing ship there is, of course, a very serious limitation as to what can be done to increase aperture clearances, but it is remarkable what considerable improvements can be obtained by comparatively small modifications to existing apertures.

The problem might be tackled along the following lines:—

(*a*) *Propeller Blade Trailing Edge to Rudder or Rudder Post Clearance.* Where this clearance is much smaller than the clearances shown in Fig. 5, a considerable reduction in the propeller-excited vibration amplitudes can be produced by comparatively small increase in this clearance.

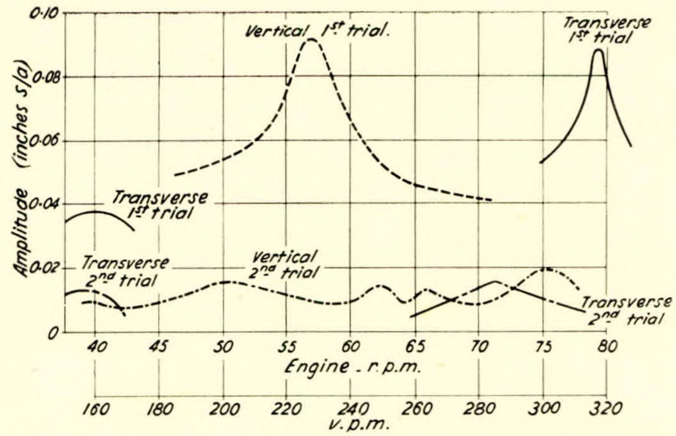

FIG. 6—*Reduction of hull vibration by increasing propeller aperture clearance*

This fact can perhaps be best demonstrated by quoting actual full scale results. Three sister ships had each suffered rapid tailshaft failures and, in one case in particular, the axial vibration of the shafting was such as to produce repeated failure of the holding down bolts in the thrust seatings. The hull vibration, which was found to be propeller-excited, had a resultant acceleration of 7·6ft. per sec.^{2}, and was reduced to 1ft. per sec.^{2} merely by increasing the propeller trailing edge clearance, which was only a matter of 4¼ per cent, by 3½ inches, the leading edge to propeller post clearance being ample. This was accomplished by making the new tailshaft 3½ inches shorter. The results of identical vibration tests before and after this modification are shown in Fig. 6. Similar arrangements were made on the other two sister vessels, which have now given trouble-free performance for roughly two years.

In other cases the increased trailing edge clearance has been obtained by cutting away part of the leading edge of the finned rudder post, and in one case by also trimming back and fairing 2½ inches off the trailing edges of the blades in way of the tightest portion of the aperture. In no case has there been a measurable loss in efficiency but a remarkable improvement in the overall steadiness of the hull and machinery.

(*b*) *Propeller Blade Tip to Sternframe Arch Clearance.* Here again a considerable reduction in propeller excitation is obtain-

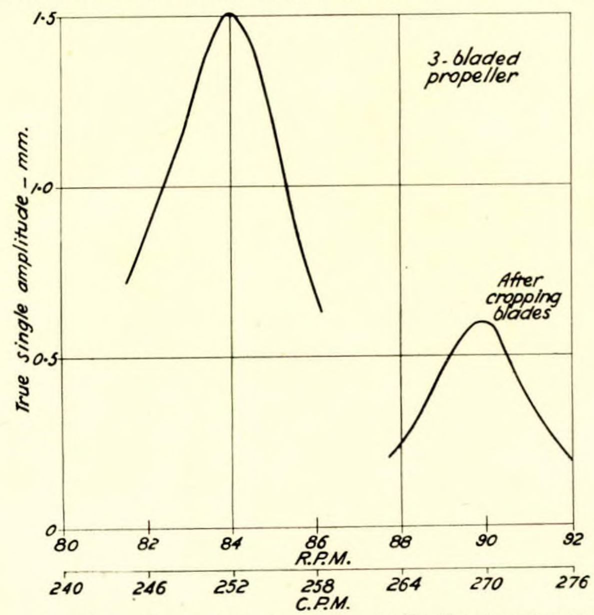

FIG. 7—*Propeller-excited transverse hull vibration (tanker, 26,000 tons displacement)*

able by an increase in the clearance. Apart from reducing the propeller diameter, there is practically little that can be done. This expedient has in fact been tried in many cases with the most satisfactory results. Fig. 7 shows the improvement in a serious critical on a tanker obtained by cropping $0{\cdot}045R$ off the propeller blade tips. As this operation means that the engine r.p.m. will be increased approximately proportionately for the same power, a factor which must be taken into account is whether the unwanted hull critical is just above or just below the service speed. If above, and tuning out by ballasting is not possible, as would be the case for a critical occurring in the deep draught condition, then it may be disadvantageous to increase propeller r.p.m. as it will bring the running speed nearer the critical, and the vibration, although considerably reduced in severity, may still be undesirable. If the critical is below the running speed, then the modification is advantageous as the running speed will be raised.

(*c*) *Propeller Pitch.* Some variation in the propeller pitch is, of course, admissible where a new propeller is to be fitted and the range is sufficiently wide to avoid a critical which is found to occur on the running speed in, say, the deep draught condition. The choice of the number of propeller blades offers another means of tuning out a vibration critical from the running speed.

DESIGN STAGE MODIFICATIONS

There are a number of methods of calculating the frequencies of the lower modes of vibration (5, 6), and in general the more accurate methods involve the determination of the sectional inertia, which depends on the scantlings. Since, however, the machinery must be decided upon very early in the design stage, it is necessary to make as accurate a calculation as possible of the natural frequencies of vibration before details of the scantlings are available.

The following formula for the 2-noded vertical mode has been evolved for this purpose: although it is based only on the dimensions and can, therefore, be rapidly evaluated at the inception of a design, it will be found to be reasonably accurate.

$$\text{2-node vertical } f_{\text{c.p.m.}} = \frac{K}{L^n}\sqrt{\frac{D_E\, d}{C_b\,(B + 3{\cdot}6\, d_l)}}$$

where K = 48,700 for tankers (longitudinal or combination framing);
34,000 for transversely framed cargo ships;
38,400 for cargo ships with longitudinally framed deck and bottom.

n = 1·23 for tankers;
1·165 for cargo ships.

C_b = block coefficient = $\dfrac{\Delta \times 35}{L\,Bd}$

Δ = loaded displacement corresponding to d.
L = length between perpendiculars.
d = load draught moulded.
d_l = mean draught for condition considered.
B = breadth moulded.
D_E = effective depth as defined by Todd and Marwood(5) or depth moulded to strength deck where there are no effective superstructures.

The following approximate relationships appear to exist between the frequency of the various modes relative to the 2-node vertical frequency:—

	Vertical Cargo	Vertical Tanker	Horizontal Cargo	Horizontal Tanker	Torsional(7) Cargo	Torsional(7) Tanker
1-node ...	—	—	—	—	0·66	—
2-node ...	1·0	1·0	1·44	1·54	1·6	—
3-node ...	1·8	2·1	2·6	3·1	2·1	—
4-node ...	2·6	3·2	3·4	4·3	—	—

It will be appreciated, of course, that, as already indicated, comparatively slight variations in the load distribution, while hardly affecting the trim or the mean draught, can considerably affect the natural frequencies, and this is particularly the case with the higher modes of vibration—which in the case of the larger vessels are the ones that matter most because of their proximity to the running speed.

Another factor which must be borne in mind is that with the ageing of the hull the overall stiffness is reduced and this brings about an overall reduction in the critical frequencies.

It will thus be evident that the problem of avoiding hull vibration criticals in the choice of engine and propeller frequencies is one of some complexity, as the conditions are so variable. A more positive and practical approach is to limit, if possible, all exciting forces to values which will ensure that the hull vibration amplitudes—or rather accelerations resulting therefrom—are acceptable. The following limits are suggested for acceptable accelerations recorded at the largest anti-node of the hull vibration usually occurring at the stern of the vessel.

Vertical vibration ... ... 1·0ft. per sec.^{2}
Transverse vibration ... 0·75ft. per sec.^{2}

The effects of hull torsional vibration are automatically allowed for in these limits, which also apply to the vertical and transverse components of this mode of vibration. Accelerations above 3ft. per sec.^{2} are liable to cause damage at points of high stress concentration in the structure.

From experience of some sixty hull vibration problems, it can be said that the acceptable values suggested above represent a condition which may be regarded as normal. At these accelerations the problem is purely a psychological one of nuisance value to personnel, and does not normally assume any significance from a consideration of vibratory stresses imposed to the hull structure. There is, however, the qualification that accelerations of this order can set up local vibration amplitudes of appreciable dimensions producing, through resonance, local accelerations of much greater magnitude, in which case some local stiffening is probably the remedy.

General Design Considerations

From subjective considerations, much can be done beforehand in the choice, position and method of installation of cabin and bathroom fittings, doors, cupboards, binnacles, gyro compass, etc., and the type and rigidity of division bulkheads in accommodation, particularly that situated aft, to minimize the drumming, rattling and creaking which can be produced by quite small accelerations. Such effects create a most exaggerated impression of the severity of the vibration, and careful attention to these small and comparatively insignificant details makes a vast difference to the overall comfort and efficiency of personnel, who have to live under these conditions for days on end.

The support and securing of main and auxiliary boilers—particularly where mounted aft of the main machinery on large tankers—is a matter requiring careful attention. Serious damage to superheater and generator tubes and brickwork has been produced by propeller-excited local resonance of the boiler about its supporting structure. It should be remembered that considerable transverse rigidity is necessary, yet adequate allowances must be made for the considerable differential thermal expansions that occur.

The attachment of ancillary gear, such as feed heaters, strainers, small pumps, valves, chests, etc., and heavy spare gear to large flat areas such as bulkheads should be avoided, as the amount of stiffening that may be required to reduce local resonance is out of all proportion in cost and inconvenience for the job it has to do.

It is accepted that the sea trial of any new "first off" ship will always indicate that a certain amount of staying and stiffening will be required to piping, switchboards and machine tools in the workshop, together with other items in the machinery or accommodation spaces.

This is particularly the case in large single screw tankers where design details have often been scaled up. Severe vibrations have been encountered during the sea trials, which have produced widespread damage to the local internal structure of the after peak. In one case some twenty tons of steel had to be fed through the after-peak manholes, faired and welded in place. This, of course, produced a spectacular improvement in the general bedlam in the after part of the vessel. Incidentally, the propeller aperture clearances of this vessel were not good.

Every effort should be made to avoid large flat areas in the shell plating in way of the ship's stern. Such areas, which are subjected to the pulsating wake of the propeller and propeller forces and couples, may vibrate with considerable amplitude at propeller blade frequency, producing an alarming sensation of rough running. It is worthy of serious consideration whether there is not a very good case for increase in the scantlings of the structure and shell plating in way of cruiser sterns of large single screw vessels.

Propeller Aperture Clearances

As can be inferred from the preceding remarks, the propeller forces play a vital part in the problem of hull vibration. The propeller and sternframe and rudder design call for the closest attention.

As a result of model tests, followed up by full scale tests on a number of vessels, the aperture clearances shown in Fig. 7 have been arrived at as a good guide towards achieving minimum propeller-excited hull vibration with the present designs of sternframes and rudders. The propeller performance is not measurably affected. It is generally accepted that a thick streamlined rudder will give a better performance than a thin rudder(8). Unfortunately this type of rudder is a trouble maker as regards vibration and local erosion pitting, and for this reason a much larger trailing edge clearance is called for. Here again it should be emphasized that it has been shown(8) that no loss in efficiency or performance will occur with these increased trailing edge clearances. It should be mentioned that reduced clearances, in addition to increasing propeller excitation, will also produce heavy pitting of sternframe and/or rudder. An interesting case of acute erosion of the rudder nose with a slim rudder is shown in Fig. 8. This is possibly due to breakdown of the flow, particularly on the suction side of the rudder. The matter is being investigated.

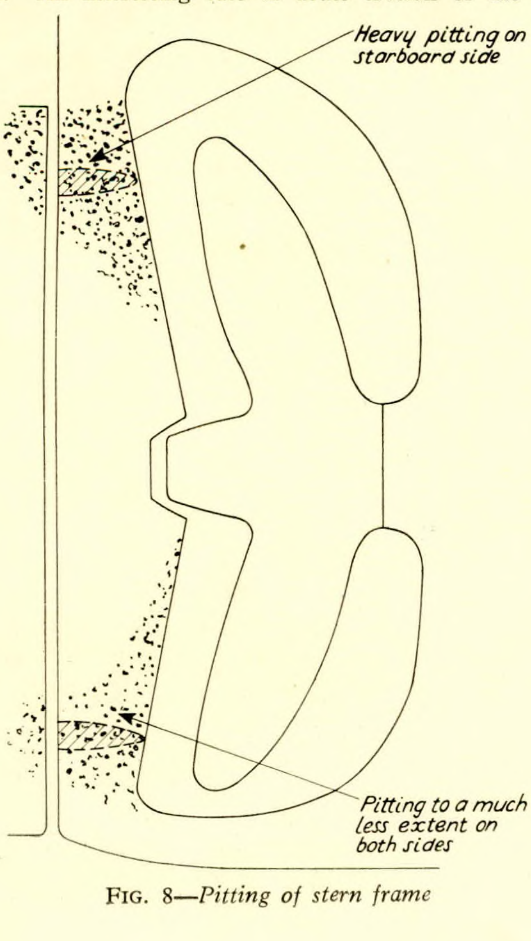

FIG. 8—*Pitting of stern frame*

The propeller tip to arch clearance should not be much less than 10 per cent to avoid excessive increase in the transverse forces transmitted to the hull.

The leading edge to propeller post clearance is decided on the score of the wake, and while the length of the vessel should accordingly enter into the assessment(9), the value of 15 per cent of the diameter will be found to be satisfactory for all practical purposes, as we are considering only the normal run of single screw vessels. With a fine after body, as would be the case of a high-powered high-speed vessel, this value could be reduced somewhat.

Propeller Forces

An attempt has been made from investigations on two large tankers of roughly similar dimensions to deduce from hull vibration measurements the actual propeller forces. The results are shown plotted in Fig. 9. It will be seen that the forces are large with both a three- and a four-bladed propeller.

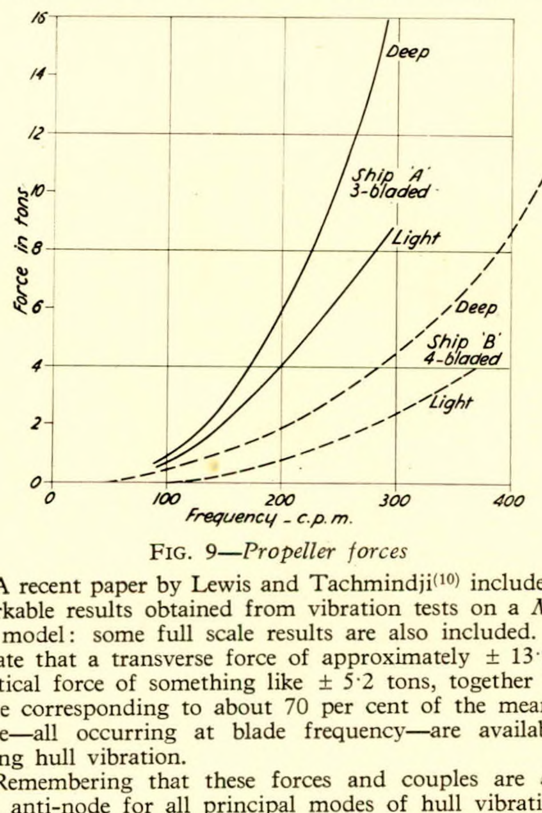

FIG. 9—*Propeller forces*

A recent paper by Lewis and Tachmindji(10) includes some remarkable results obtained from vibration tests on a *Mariner* class model: some full scale results are also included. They indicate that a transverse force of approximately $\pm$ 13·9 tons, a vertical force of something like $\pm$ 5·2 tons, together with a couple corresponding to about 70 per cent of the mean shaft torque—all occurring at blade frequency—are available for exciting hull vibration.

Remembering that these forces and couples are applied at an anti-node for all principal modes of hull vibration, i.e. the most effective point for producing the maximum effect, the magnitude of the problem can be appreciated.

As a rough guide to the accelerations of the hull which might be expected, the following formulæ have been deduced from a number of results. The formulæ are in terms of force in tons to produce on resonance, unit acceleration, i.e. 1 ft. per sec.^{2}, in the stern of a vessel of welded construction of any displacement (Δ tons) when excited by a force situated at the propeller.

2-node vertical	$\pm F = 0{\cdot}100 \times 10^{-3} \times \Delta$ tons
3-node vertical	$\pm F = 0{\cdot}125 \times 10^{-3} \times \Delta$ tons
2-node transverse	$\pm F = 0{\cdot}131 \times 10^{-3} \times \Delta$ tons
3-node transverse	$\pm F = 0{\cdot}144 \times 10^{-3} \times \Delta$ tons

In the case of an all-riveted construction, the above forces may be approximately doubled for the same unit acceleration.

Engine Balancing

The balancing of large reciprocating engines is almost standard practice today. One does, however, meet the odd case—usually resulting from torsional vibration considerations—where a crank arrangement has been chosen for satisfactory firing order for torsional vibration yet is a bad one for balance. One classic case of a geared medium-speed engine resulted in a primary internal couple of 720 tons ft. vertically, and 260 tons ft. transversely, which produced recurrent and serious trouble with the holding down bolts.

It is not a straightforward matter to suggest limits for forces and couples, since there are many factors involved, such as the position of the engine in the ship, and the size of the ship, its construction, etc., which, of course, considerably influence the effects of engine unbalance.

Balancing out first order forces or couples is a fairly simple business where the hull critical is excited at the service speed at one loading condition, which is usually the case. A single balance weight fitted to the line shafting as far aft as possible—the further aft the smaller the weight—has been employed with success on a number of occasions. The method of calculating the balance weight necessary is given in reference [12].

The elimination of second order vibrations has also been effected by modifying engine reciprocating masses where practicable; where impracticable, as a last resort, balance weights driven from the shaft at twice engine revolutions have been fitted.

Transverse Rocking Modes

In large oil engines it can happen that the rocking modes of vibration of the whole engine about its seating are excited by the crosshead guide reactions at or in close proximity to the running speed. The amplitudes can be large enough to cause serious concern and, while little can be accomplished by additional stiffening to the engine seatings, a simple solution has been provided by transversely staying the engine to the ship's side through struts, which in some cases have incorporated a shear pin to minimize damage that could be caused to the engine if the ship's side were struck.

Judging by the number of large engines which have been so dealt with—all tanker installations—it would appear that the stiffness due to the fine lines of the hull structure in this region is responsible. Engines stayed in this manner have proved to suffer no adverse effects, and though the procedure is regarded in the nature of a compromise, it does provide a simple and practical solution to the problem.

Resilient Mountings

It is rather surprising that there has not been more use of resilient mountings for auxiliary generators in particular. For years the engines of motor cars have been "floating" on rubber, and yet it is very seldom indeed that one sees such applications on ships. It is remarkable what can be achieved in the elimination of structure-borne noise by resiliently mounting Diesel machinery. Insulation of machinery spaces against air-borne noise is, however, quite a common practice in passenger ships.

MACHINERY VIBRATION

It is not intended to go into detail on the matter of the torsional vibration of shafting, which is now a routine matter with most of the major classification societies. A thorough investigation of the torsional vibration characteristics of the shafting system is automatically made in the plan stage of machinery intended for classification, as is well known, and the permissible stresses are such as ensure safety of the shafting in service. There are, however, certain constructional and design details arising from a consideration of the vibration problem which have given recurrent trouble due to the effects of working and vibratory stresses, and which may be recorded for the sake of completeness.

This section has been divided into three parts, the first embracing reciprocating machinery, the second geared turbine plant, and the third transmission shafting and propellers.

RECIPROCATING MACHINERY

There are still far too many failures in the bolts securing dynamically stressed parts. While in many instances one is led to the belief that failure has occurred as a direct result of the personal element, there are also some glaring evidences of design deficiency.

Bolted Connexions

The tightening of dynamically stressed bolts, particularly large bolts, is largely a matter of guess work. If it could be brought home that it is merely a question of time whether a bolt will fail when subjected to a dynamic load which may be a mere fraction of the tightening load, then a useful purpose would be served. Tests on large bolts to fatigue failure under direct reversed stress carried out by the British Shipbuilding Research Association[13] have shown that the fatigue strength of quite a well designed forged mild steel bolt is only of the

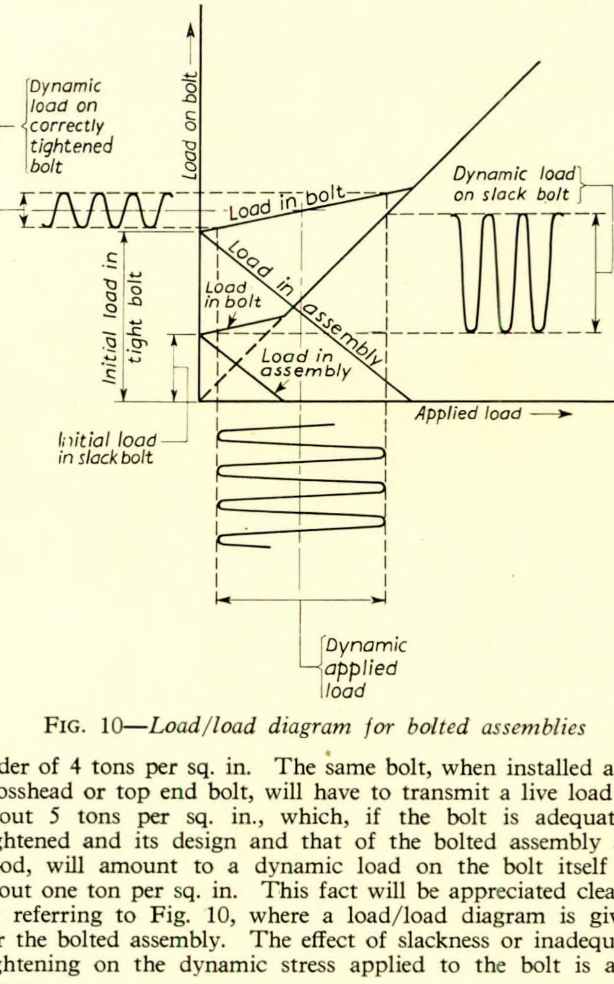

FIG. 10—*Load/load diagram for bolted assemblies*

order of 4 tons per sq. in. The same bolt, when installed as a crosshead or top end bolt, will have to transmit a live load of about 5 tons per sq. in., which, if the bolt is adequately tightened and its design and that of the bolted assembly are good, will amount to a dynamic load on the bolt itself of about one ton per sq. in. This fact will be appreciated clearly by referring to Fig. 10, where a load/load diagram is given for the bolted assembly. The effect of slackness or inadequate tightening on the dynamic stress applied to the bolt is also shown.

There are several factors which can reduce the elasticity of the bolt relative to that of the bolted assembly to a dangerously low value, producing rapid fatigue failure of the bolt with often disastrous consequential damage. Some of these factors are enumerated below: —

(1) Excessive cramping of the crosshead or top end design for space considerations, producing a bolted assembly which is dangerously elastic compared with the heavy bolts required to transmit the gas load. Resource has been made by some, under these circumstances, to the use of high tensile bolt material, without improving the design details. This has not improved matters as the advantage of the higher yield of such material is largely lost due to its increased notch sensitivity and rapid crack development. Wherever practicable the bolt should be at least four times more elastic than the bolted assembly.

(2) Large areas in contact with the butting surfaces of the

bolted connexions between the line of bolts make the degree of accurate fitting which is necessary almost a tool room job. Relief of this area in contact, except in the immediate vicinity of the bolt, will increase the safety factor appreciably. This applies to coupling bolts anywhere, including the rudder palm coupling.

(3) The use of a large number of shims between the halves of bearings produces the same hazard referred to above. A single steel shim, ground parallel to size, is the best arrangement.

(4) Correct tightening of dynamically stressed bolts is a most vexing problem, particularly with large bolts. Varying frictional forces, which can be reduced, however, by modern extreme pressure lubricants, limited access for the flogging hammer, crude indicators for tightness determined by the rotation of a nut, incorrect assessment of the relative elasticity of the bolt to the bolted assembly, introduce factors of uncertainty in the vital parts of an engine. The simple poker micrometer gauge shown in Fig. 11 provides a direct reading of the actual stress conditions in the bolt.

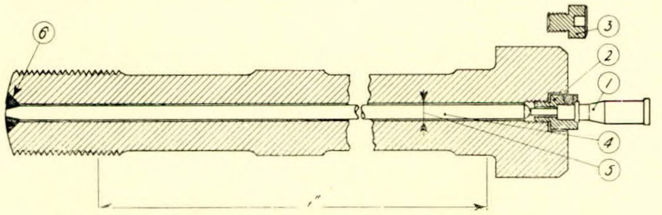

FIG. 11—*Micrometer gauge for indicating bolt strain (=0·0006 × L inch for M.S. bolts)*

(1) Standard 0-½ inch micrometer head; (2) brass bush screwed ½-in. Whit. into bolt head; (3) steel ½-in. Whit. socket screw for plugging hole; (4) ¼-in. diameter ground silver steel rod; (5) $\frac{9}{32}$-in. diameter hole drilled the length of the bolt; (6) rod inserted and welded or brazed there.

Recurrent failures of bottom end bolts 1⅛-in. diameter in an auxiliary engine indicated, by gauging, that what the fitters regarded as being correct tightness by the "feel" was, in fact, above the yield point for the material.

In a recent investigation of the recurrent failure of the studs securing the detachable blades of a built-up propeller, the poker gauge technique was used as a check upon the tightening strain put up by the fitters doing the job. It was found that the 3-in. diameter studs, which were mild steel with brass nuts, were severely overstrained. The tightening strain was nearly four times what was required and well into the plastic range for this material.

(5) The vital need for conscientious crankcase inspection, following up all dynamically stressed bolts, particularly in a new engine and also after explosions at starting up.

(6) The shorter the bolt length, the greater is the hazard, for the simple reason that the elastic strain between yield (which may start a crack at the second thread), correct tightness and dangerous slackness is very small. For this reason coupling bolts, and in particular crankshaft coupling bolts which may have to take considerable bending, should not be driven in so hard as to be inelastic, due to the heavy holding grip of the bolt holes which extend practically the length of the bolt. The bolt holes should be reamed and lapped round and the ground bolt should have an interference fit of the order of 0·0005in./1in. diameter.

Short dynamically stressed bolts should be avoided wherever possible. A substantial collar extension to the nut is a simple expedient whereby the effective elasticity of the bolt may be increased.

(7) Holding down bolts should preferably be bolts, and not studs, and the chocks small, with the fitting area concentrated around the stem of the bolt only; otherwise there is a real danger of introducing serious elasticity into the bolted assembly. Adequate and rigid side and corner chocks should be provided, and hard and rigid contact with the bedplate flange maintained by folding wedges, which should be positively locked against working loose.

Crankshaft Stresses and Alignment

The stresses were recently measured in the crank web to pin and journal fillets of the crankshaft of a large oil engine while running under load. Electric resistance strain gauges were used for the purpose. The tests were instigated as a result of an investigation of the failure of a similar crankshaft after a short life in service. The following points were emphasized: —

(1) The resultant bending stress in the pin fillet at a point in line with the centre line of the web was about 2·5 times that occurring in the same fillet but displaced 90 degrees.

(2) The effects of torsional vibration may be additive depending on the phase of the vibration relative to the crank throw.

(3) The effects of axial vibration may be additive for the same reason.

(4) Malalignment stresses in either the vertical or transverse plane are directly additive. Stresses resulting from malalignment in the vertical plane are the more significant insofar as the peak bending stresses due to the gas loads occur in a plane approximately vertical to the crankshaft centre line.

(5) The position of the maximum crankpin bending stress unfortunately occurs at a point in the crankshaft forging where the amount of forging reduction is a minimum and, due to the large masses of metal which surround this point, it is also a position where the structure of the metal is coarser and the internal locked-up stresses due to differential cooling from the forging and heat treatment temperatures are normally a maximum.

(6) It therefore follows from (5) that additional stress concontration effects such as tool marks and corrosion pits in this critical region should be avoided at all costs.

The extent of torsional vibration stresses in any dynamic system are limited under normal conditions. The effects of bad weather in ballast may be serious, however, especially if the machinery is driven hard and allowed to race on to a serious critical normally well removed from the running range[14]. It may be of interest to mention that the reason for certain auxiliary crankshaft failures occurring after five to seven years' service was simply the fact that the engine tachometers were relied on to indicate the r.p.m. An error of ± 10 per cent was found in the tachometers—unfortunately some of the machines working at higher speeds were running on the flank of a strong critical, with serious results in the case of three. Accurate hand tachometers issued to the chief engineers with instructions as to frequent use are expected to be the means for the cure.

The stresses resulting from axial vibration are not, under normal conditions, of any significance, unless excited at a strong torsional critical.

But the effects of malalignment may be serious. There are many failures of large crankshafts on record which have been directly attributed to malalignment, a condition which may develop slowly or rapidly in service.

The checking of the alignment of the Doxford crankshaft is somewhat complicated by the fact that, owing to the flexibility of the shaft and the freedom given to the journals by the spherical bearings, it is not a straightforward matter to interpret centre crank web deflexion readings in terms of journal alignment, as is the case with the normal type of crankshaft. If well kept records of deflexions and bridge gauge readings are available, dating back to those taken after the sea trials, some fairly reliable deductions of crankshaft alignment can be made in normal circumstances.

Satisfactory operation of the Doxford shaft is quoted by some operators with centre crank web deflexion readings of the order of 70-thousandths of an inch, but it is not clear how these readings were taken. It should be stated here that scavenge

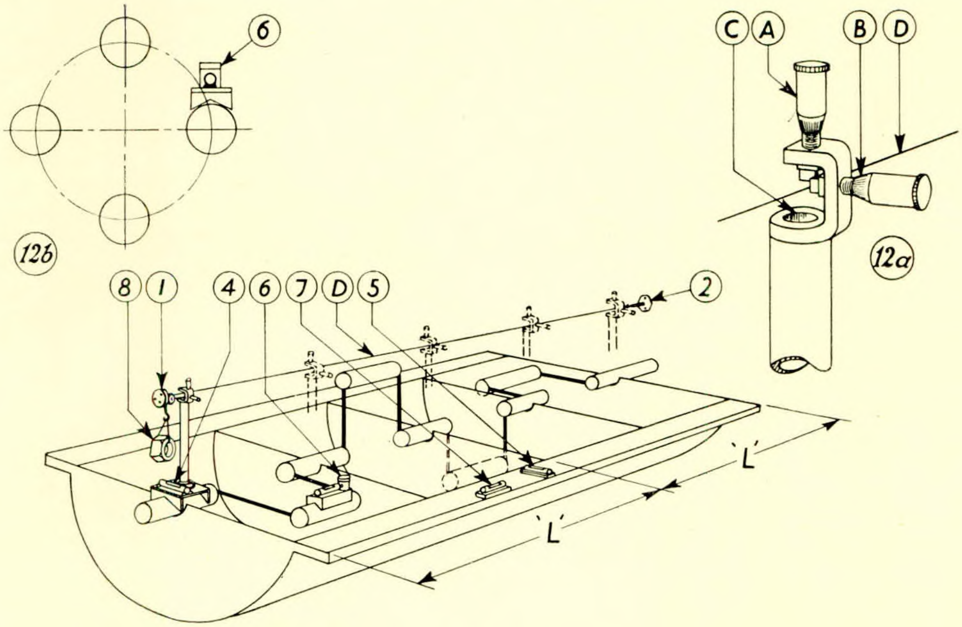

FIG. 12

Alignment of Journals

Piano wire 0·018 inch (*D*) anchored to endplates of engine entablature over pulley at one end (*1*) and fixed block (*2*) at other; height of wire set to be approximately the same at each end and in approximately the same vertical plane as crankshaft centre line by micro readings. Micro staff set approximately vertical by spirit level (*4*) to be read only when master level (*5*) reads "zero". Master level left undisturbed throughout tests—note micro readings for only just making contact with wire when bulb (*C*) lights up. Repeat for X_2 X_3 X_4 X_5, etc.

Checking Crankpin Errors

Take clinometer readings (*6*) when mounted on Vee block on pin—the clinometer to be read only when master level (*7*) reads zero—take readings on the four quarters as shown in Fig. 12(b).

Important

(1) Weight (*8*) = 42lb.
(2) Spirit levels (*4*) (*5*) (*6*) (*7*) sensitivity = 0·001 per ft.
(3) Piano wire is 0·018 inch diameter.
(4) Sag of wire to be allowed for = $\frac{L^2}{7{\cdot}25}$ thousandths of an inch where L = half length of engine (feet).

crank deflexions, which are always negative in a correctly aligned engine, should not exceed 12-thousandths, the gauge being zeroed with crankpin on bottom centre and gauge pintles in line with the outer edge of journal away from crankpin.

There have been instances where the alignment in the transverse plane has had to be reckoned with, and it may be of interest to consider a simple direct reading method which has been used by the Department.

Figs. 12 and 13 give the details of the apparatus and method, which are readily understood, as they do in one sense or another typify normal practice in many engine builders' erecting shops in this country and abroad. The provision of datum anchorage points on the after-peak bulkhead and on the forward engine room bulkhead at the time of building would provide a permanent and ready check of the line shafting and crankshaft alignment throughout the life of the ship.

Top End Troubles

It does occasionally happen that an engine develops trouble with the top end bearings, the symptoms being that white metal is progressively hammered out of the forward and after crosshead bearings. The trouble is usually met with after remetalling of the crosshead bearings. It is a rare occurrence but, as it can be a problem not necessarily of alignment of the running gear to the crankshaft, nor of the crankshaft itself, and in which the usual methods of approach may not produce the cure, it is considered to be of interest and value to indicate a method which has produced satisfactory results in several cases.

The following factors singly or collectively may bring about this defect:—

(1) Lack of lubrication, particularly where the clearances in other bearings are large, thus starving the bearing in question.
(2) Malalignment of crankshaft, due to unequal wear down, or transverse or vertical relative displacement of parts of the bedplate due to ship or engine vibration or from other causes unknown.
(3) "Helical" or "conical" errors of the crankpin (see Fig. 15)—original sin increased by the wear pattern of the pin, or distortion produced by heavy explosions, or by the engine being brought up solid with the cylinder full of oil or water from a leaky fuel valve or cracked liner respectively.
(4) Malalignment of the guide face relative to the crankshaft caused by grounding, vibration or distortion of the entablature for the reasons described in item (3) above.

The method used with the Doxford engine will be described in some detail.

Before doing any gauging on the pin it may be necessary to file and stone the high band which usually wears around the pin in way of the oil hole. Micrometer readings should indicate the extent. A variation of 0·003 inch lack of parallelism may be expected, however, in a new pin of about 17 inches diameter.

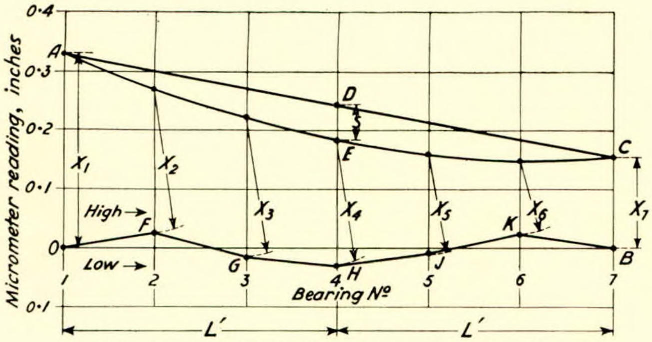

FIG. 13—*Vertical alignment of crankshaft*

Note:

(1) Position of bearing centre line plotted out to scale—1, 2, 3, 4, 5, 6, 7, etc.

(2) Actual micrometer readings X_1 and X_7, i.e. first and last bearing, plotted OA and BC.

(3) Join A and C (set wire so that $OA = BC$ approximately).

(4) At point corresponding to mid length of crankshaft D, drop down $DE = S$ inches = the sag of the wire. $S = \frac{L^2}{7 \cdot 25}$ thousandths of an inch where L = half length of engine in feet.

(5) Draw in smooth curve AEC showing sagging wire.

(6) From intersections of curve AEC with bearing centre lines, drop $X_2\,X_3$, etc., actual micrometer readings.

(7) Then O, F, G, H, J, K, B is actual bearing alignment; those above OB are high, below OB are low.

(8) *Important.* Wire is 0·018-in. diameter piano wire, tension 42lb.

The gauging method shown in Figs. 12 and 13 may be used for obtaining the following information:—

(1) The extent of the "helical" and "conical" error in the centre crank pin.

(2) The alignment of the crankshaft journals in the vertical and transverse planes.

The alignment of the guide face relative to the longitudinal

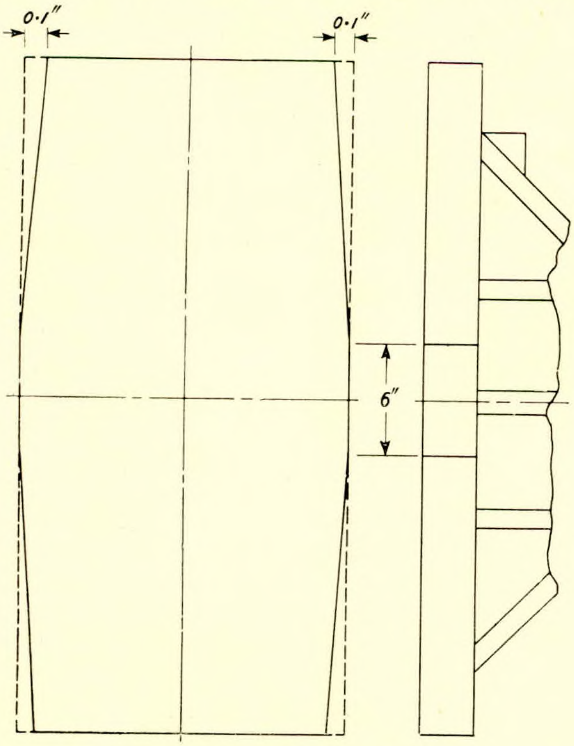

FIG. 14—*Articulation of crosshead slipper*

(Note: Centre portion left parallel and remainder machined away 0 inch-0·1 inch)

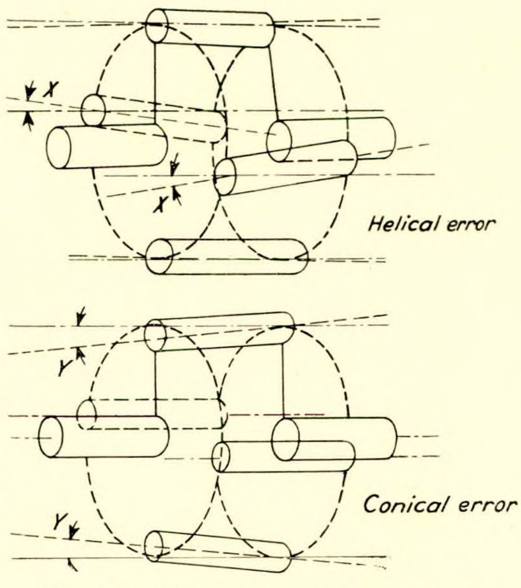

FIG. 15—*Schematic diagram of crankpin errors*

centre line of the crankshaft should also be determined. This is easily done by stretching a wire extending over the faces of the two adjacent guides and measuring by means of an internal micrometer the variation between the wire and the forward and after portions of the face in question.

The bottom-end spherical bearing, when in good condition and properly bedded and relieved in the horns as shown later, will cope with all errors which tend to produce a torsion in the connecting rod. The symptoms of such torsion are shown up by a hot spot in the crosshead bearings towards one side of each bearing, often to the front of the engine in the forward bearing, and to the back of the engine in the after bearing. Malalignment of the crankshaft in the vertical plane or guide malalignment, with or without crankpin errors, depending on the extent of each, will set up a fore and aft weaving action of the bottom end, and the action of the spherical does not affect this issue, as the whole bottom-end assembly moves bodily fore and aft along the crankpin at each revolution.

The bright marking of the webs by the rubbing strips on the cheeks of the bottom-end bearings is an indication of this process, which loads first the forward top-end bearing and then the after top-end bearing. The crosshead pin cannot follow this movement as it is constrained by the crosshead slipper in the guide. A steady hammering out of the bearing metal then occurs fairly rapidly, particularly so with new metal which has not had time to produce a hard working surface.

Opening out the guide bar side clearances will have little effect, in fact it tends to make things worse, possibly due to the increase in the resultant fore or aft guide reactions which maintain the cheek of the slipper in harder contact with the fore and aft guide bars.

Considering positive micrometer spirit level readings of the helical error to be those which would produce crankpin deflexions in the same sense as those resulting from the full load torque of the engine, i.e. the forward web leading the after web of the crank in the direction of ahead rotation, which, incidentally, amounts to approximately + 0·005 inch in the length of the centre crankpin, then the following limits will produce satisfactory results in service:—

Total helical error ... ... = 0·010in./ft.

Conical error ... ... = 0·005in./ft.

The crank journal alignment should not exceed 0·015 inch between adjacent bearings in either the vertical or transverse planes. The parallelism of the guide face should not exceed 0·003 inch in the width of the face.

Remedy I—if time and facilities are available.

(1) Adjust alignment either by re-metalling and/or re-setting

chocks, jacking bedplate up, down, or transversely as necessary.

(2) Re-check crankshaft errors as some of these may be affected by the new alignment.

(3) If crankpin errors are still excessive, then recourse must be made to filing and lapping to a sleeve or to machining in place. Several engine repair firms now have machining facilities for doing this job accurately and neatly in place. In any method, it is a safeguard to file on several reference patches at the forward and after extremities of the working surface of the pin before commencing the corrective operation.

(4) Set guide face true if necessary.

(5) Re-machine or re-bed spherical as necessary, the bedding to be uniformly good on forward and after sections of upper spherical, washing away bedding about 0·010 inch for an arc of 40 degrees on either side of horizontal joint. After re-assembly it should be possible to swing the top end of the rod by hand through a complete circle.

Note: Care should be taken to ensure that the circumferential edges of the bedded spherical surfaces of the spherical bearing, also the edges of the oil grooves, are not left sharp by the bedding operation. These edges should be well rounded to ensure that oil reaches the bedded surfaces. A sharp edge acts most efficently as an oil seal and instances have occurred in which frettage at the spherical surfaces has been produced very rapidly, causing partial seizure; after rounding off these edges the bearings ran perfectly.

(6) Re-surface crosshead pin as necessary and re-metal bearings. Re-surface piston rod spherical if necessary. Check alignment of bore of bearings with each other and that they are normal to the centre line of rod.

(7) Re-assemble engine and run up to 60 or 80 r.p.m., or as high as moorings will permit, for four or five hours or more without stopping. It is possible that one or other of the top ends will heat up while finding its running alignment. It should then cool off and all should be well.

Remedy II—when time is very limited and facilities scanty.

The following is the procedure after checking and correcting excessive crankshaft malalignment, if any:—

(1) Remove crosshead slipper and relieve cheeks as shown in Fig. 14. This will allow the crosshead pin to follow the movements of the bottom end.

(2) Re-metal crosshead bearings and re-surface crosshead pin and piston rod spherical as necessary, observing the same precautions as in (6) above.

(3) Re-surface and carefully bed spherical as in (5) above.

(4) Re-assemble engine and run as indicated in item (7) above.

TURBINES AND GEARING

As a paper on the subject of gearing is to be given by a colleague in the 1955-56 session, a few remarks along essentially practical lines only will now be included.

Claw Couplings

The fine tooth claw coupling appears to be gaining favour in this country; in fact, it is almost standard practice in the United States. One of its main advantages is from a vibration point of view. Some turbine blading failures have in the past been directly attributed to coarse tooth claw coupling excitation, brought about by malalignment between turbine rotor and primary pinion, which is almost unavoidable. By increasing the number of claw teeth, the frequency of the excitation has been raised, producing increased damping and, in addition, a much lighter and more compact design.

The fine tooth claw has, however, brought troubles of its own in its train. It is noticeably susceptible to frettage, but given hard material for both male and female members (250 and 350 Brinell respectively) and adequate lubrication, together with other design refinements, it is entirely satisfactory.

An interesting case of heavy vibration at turbine rotor r.p.m. felt throughout the engine room, was caused by frettage and wear of the fitting or centring strips in the claw coupling on one vessel. It was so serious that the owners were convinced they had a bent rotor.

Built Up Primary Gear Wheels

A type of construction employing loose end plates bolted to the forged-on flanges of the wheel shaft and to the wheel rim flanges has had a number of failures caused by fatigue failure of the bolts themselves, the nut or the bolt head passing through the mesh, or by failure of the pinion teeth due to high loading produced by malalignment of the rim caused by slackness on the end plates.

In some cases the bolts were not a good fit in the holes, with a consequent concentration of load on some of the bolts and, to make matters worse, the nuts were locked by welding to the end plates—a risky procedure because, with the heat and stresses of the welding, which was unnecessarily heavy, the nuts were distorted, and any pre-stress which may have been there after the tightening was promptly destroyed.

Gear Noise

In three particularly noisy double reduction war-time-built main geared units of fabricated construction, a reduction of the order of 16 decibels in the gear noise was obtained by the simple expedient of lagging the gear case (to almost the depth of the stiffening ribs) with a heavily loaded plastic compound and filling in with light sheet iron closing plates. (16 decibels represents a reduction of 40/1 in the sound intensity.)

PROPELLERS AND SHAFTING

Propellers

In December 1949 a strain gauge technique was used in measuring propeller blade and tailshaft stresses under service conditions. Trials were carried out on m.v. *Jason II*, a cabin cruiser, which were reported by Dr. Dorey in his Presidential paper to the 1951 International Conference of Naval Architects and Marine Engineers at Newcastle in July 1951[15].

However, it was not until 1953 that, through the courtesy of Shell Tankers, Ltd., and the Esso Petroleum Company, measurements were made on an ocean going vessel under normal service conditions. A T.2 tanker was made available for these tests, which required the boring of a 2-inch diameter central hole down the length of the tailshaft for bringing the gauge

TABLE II.—HARMONIC COMPONENTS OF PROPELLER BLADE STRESSES

Harmonic	1	2	3	4	5	6	7	8	9	10	11	12	16	17
'Sam' ship—light % stress	40·8	7·2	7·2	4·0	7·2	5·5	2·9	0·9	3·3	1·3	3·2	3·5	2·4	4·0
T.2—deep % stress	30·0	17·8	0·8	7·7	6·1	5·0	4·9	6·5	3·7	3·6	3·1	4·2	1·2	1·2
T.2— light % stress	41·0	16·0	3·0	8·0	15·0	2·1	5·5	2·5	2·7	2·3	7·5	2·9	1·1	2·8
Chryssi(11) % thrust	29·2	62·0	10·2	18·5	2·3	7·5	1·1	2·5	—	0·5	0·15	0·3	—	—

leads into the ship. The instruments used are described in reference [15]. A few months later a similar investigation was made on a "Sam" type Liberty ship.

The results of these investigations, briefly summarized below, include a harmonic analysis of typical records obtained of bending stresses at the blade roots. The tests on the T.2 tanker were made in the deep and light ballast draught, while on the "Sam" ship tests were made during a run in very light draught only. Also included for comparison are results of a harmonic analysis of blade thrust derived by Panagopulos[11] for the *Chryssi* from the wake distribution at the propeller disc of a four-bladed 22-ft. diameter propeller.

It will be noted that in the case of the "Sam" ship, the second-order component is small, the reason being that the propeller was immersed only up to the boss. In deeper draughts this component would be considerably increased, as will be seen from the T.2 results, which were for a propeller immersed at light and deep draught. The blade stresses recorded were as follows: —

"Sam" ship ... = ± 5,000lb. per sq. in. (failed in service)
T.2—light ... = ± 3,250lb. per sq. in.
T.2—deep ... = ± 2,750lb. per sq. in.

There appears to be a considerable divergence between the harmonic analyses of the measured blade stresses and the harmonic components of the blade thrust quoted from Panagopulos's paper[11]. While the differences in the wake pattern, the design of blade, and aperture clearances must undoubtedly account for much of the divergence, it is difficult to reconcile the small first-order component in the derived thrust.

The tests, however, serve to focus attention once again on the importance of the wake pattern as it affects the dynamic stresses on propeller blades. Heavy weather conditions might double the values quoted and underline the necessity for easing down the engine r.p.m. under these conditions. Recommendations were made for increasing the blade root thickness in the case of the "Sam" ship propeller.

On the matter of the blade root, it is now generally accepted that there is a considerable falling off of the order of 20/25 per cent in the physical properties of the manganese bronze in the region of the root, particularly at the back of the blade. Some of the new high tensile non-ferrous propeller alloys which are now coming ito favour, while evidencing a somewhat lesser falling off in physical properties in the critical root section, do, however, have the advantage of an almost 100 per cent increase in the notched (Wöhler) fatigue strength in salt spray which, from design considerations, is a most important factor. The reduction in the elongation and proof strength in thick sections as against thin sections is also superior with the new material, and the increased hardness may offer some advantages where cavitation is concerned.

Built-up Propellers

Naturally the same remarks as itemized above apply to built-up propellers, which are still fitted today. The problem, however, is considerably aggravated by the weakness or discontinuity introduced by bolting a blade on to a boss. There are also certain hydrodynamic deficiencies.

Certain obvious precautions, however, will reduce the hazard of stud failures: —

(*a*) The blade butting face should be correctly bedded to the landing in the boss. The palm should be free in the tapered recess. The butting faces should be relieved except in way of the bolts and allowing an abutment of $1\frac{1}{2}D$ around the centre of the bolt-diameter D.

(*b*) The studs, preferably good quality wrought iron, should be as long as practicable.

(*c*) After setting the propeller pitch, half the number of studs should have correctly machined bronze filling pieces to act with the studs as dowels for locating the blade torsionally.

(*d*) The studs should be perpendicular to the plane of the landing in the boss and the spot facing in the blade palm should be parallel to the abutting landing on the palm.

(*e*) The greatest care should be taken in hardening up the nuts to ensure that the studs are not over- or under-tightened. See also page 9 and Fig. 11, where a positive tightening strain indicator is shown.

(*f*) The recess in the hub and blade could with advantage be pumped up with a plastic sealing material containing zinc.

TABLE III

Tailshaft stresses	Light ship			Deep ship		
Vibratory stresses due to hydrodynamic forces	±4,850	±3,400	(16)	±4,250	±1,450	(16)
Reversed bending due to propeller weight	±1,850	±1,500		±1,850	±1,500	
Total vibratory bending stresses	±6,700	±4,900		±6,100	±2,950	
Vibratory torsional stress	±1,500	±2,110		±1,040	±1,200	
Maximum principal bending stresses	±7,000	±5,650		±6,200	±3,400	

Tailshaft Stresses

The following tailshaft stresses were recorded on the T.2 tanker on the fourth-order torsional vibration critical (at 85 r.p.m.). The stresses due to hydrodynamic forces were deduced from the bending stresses recorded on the propeller blades. Corresponding values, but corrected for tailshaft diameter, obtained by Rupp[16] on another T.2 (critical at 90 r.p.m.) are included for comparison.

The peak-to-peak tailshaft stresses recorded in the case of the "Sam" type ship in light ballast amounted to ± 6,900lb. per sq. in.

A paper by Panagopulos and Nickerson[11] includes the results of a comprehensive investigation along identical lines carried out on the tailshaft of a super tanker, the s.s. *Chryssi*. The test covered a complete voyage during which some very heavy weather was encountered. The results indicate that under heavy weather conditions with the ship running into a strong sea—gale force 7—and pitching heavily, the calm water stresses may be doubled. The results also demonstrate that under certain conditions of loading, corresponding to a light ballast draught in which the blade tips were just immersed, an almost critical draught could be reached, in which the deep draught stresses were almost doubled. The results as indicated in the paper are somewhat compromised by the fact that the records were made while the vessel was ballasting in coastal waters, and there may have been a shallow water effect introduced.

The paper also includes a unique comparison of the tailshaft stresses occurring with a five- and a six-bladed propeller fitted to the same ship, in which stresses with the five-bladed propeller are almost double those recorded with the six-bladed propeller under identical conditions. A remarkable fact.

The tailshaft problem is obviously still very much in evidence and, while a great deal of investigation is under way in this country, Europe and the U.S.A., the trends are that the solution is one of dealing effectively with the wake distribution at the propeller disc, which is the predominating factor. This has actually been known for many years, from the simple fact that tailshaft failures seldom occur in twin screw ships. Until the solution, which may even involve a most unorthodox re-design of the single screw aperture, is established, there is much that can be done to reduce the hazard of fatigue failure of the tailshaft.

(1) As the wake distribution is the predominant factor, careful attention to aperture clearances along the lines already mentioned (Fig. 7) will do much to reduce the effects

of wake. Lewis[10] has shown that axial (forward) propeller aperture clearances of the order of 30 per cent of the propeller diameter will reduce the vibratory forces to 25 per cent or less of those obtaining with the same propeller working in an aperture having an axial clearance of 12½ per cent.

(2) The design of the tailshaft, using the necessary precautions for minimizing the stress concentration effects of the hub, the key and keyway, are now well known and widely adopted.

(3) Efficient sealing arrangements against the ingress of sea water at the large and the small ends of the cone are vital factors.

(4) Careful assembly of the tailshaft and propeller without lubricant and allowing a positive drive up of 0·003 inch $\times D$ (where D = diameter of large end of a 1/12 cone in inches), will ensure that the grip of the cone will take most of the drive, thus reducing the stresses at the key—a point of high stress concentration.

(5) Propeller design in which pitch variation is utilized in reducing the effects of wake variation, thus reducing the eccentric thrust of the propeller.

(6) The vital necessity for reducing r.p.m. in rough weather where there is a tendency towards racing. Hammering of the tailshaft in the stern bearing can be very serious in a ship being driven hard while pitching.

Shafting Alignment

A creaking line shaft coupling or a hot line shaft bearing are both good indicators of malalignment. It will be found that the symptoms will either be present in the light or the loaded draught. These symptoms suggest that a continual frettage action is occurring in the couplings and bolts, which may reach serious proportions in time. Normally shafting alignment is seldom a serious problem in midship machinery installations—where alignment is seldom critical. Installations with as much as 1½ inches well distributed malalignment, recorded in the light condition, have given trouble-free service, but the sense of malalignment has been such that for the loaded condition it would be less. In tankers the setting of the after plummer block bearing is critical to thousandths of an inch. In one case such a bearing was rapidly broken up because it was set 15-thousandths too low.

The Department has now used, for three years, on a large number of jobs, an alignment technique in which neither couplings nor bearings are disturbed, and a complete shafting alignment check can be made on a midship machinery installation in an afternoon. An aft-end job would take half this time. The method has already been fully described[15].

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Discussion

Dr. W. Ker Wilson said that he greatly appreciated the privilege of opening the discussion. Mr. Bunyan's paper was a worthy addition to the many valuable papers that had come from the Engineering Research Department of Lloyd's Register of Shipping. Many people would recall with gratitude that, largely owing to the initiative of Dr. Dorey, this department was established twenty years ago, at a time when several very ugly problems were confronting the marine engineering industry, to apply scientific method to the investigation of machinery breakdowns in classed ships. The phrase "scientific method" might have fallen heavily on some ears. It might have suggested, for example, the possibility of conflict between academic thoroughness and the urgency of day-to-day engineering activity. Academic thoroughness was in itself a very praiseworthy and rightly esteemed quality, but there was a danger that it might lead to endlessly increased refinement of both theoretical approach and experimental procedure until, in the end, effort drifted asymptotically to rest from sheer exhaustion. Of course, nothing of this kind had occurred in the Engineering Research Department of Lloyd's Register of Shipping. As Mr. Bunyan's paper abundantly testified, the challenge of every problem was accepted with a very lively appreciation of the academic and practical implications, and a solution was sought in the simplest possible terms. Many great scientists had shared this liking for simplicity. One recalled, for instance, Newton's famous pronouncement, "Nature affects not the pomp of superfluous causes". Not so long ago a very eminent person defined an engineer as being a fully educated scientist. Perhaps he had in mind the engineer's liking for simplicity.

With regard to hull vibration, he could endorse from personal experience Mr. Bunyan's remarks about the nuisance created by loose fittings and squeaky doors and pipes. The remedy seemed simple enough, and consisted chiefly in tightening up all loose connexions and making more liberal use of rubber and other sound-deadening materials.

In his list of basic conceptions, Mr. Bunyan ruled out the possibility of hull vibration arising from torsional vibration of shafting. Was this strictly true of geared installations where there was a possibility of disturbances being transmitted by torque reaction at the gearbox?

The information on the effect of propeller aperture clearance was most valuable. Similar investigations leading to similar conclusions had been carried out in aeronautical engineering.

A very interesting experiment was carried out some years ago at Farnborough in which, starting from practically zero, the clearance between the airscrew blades and the leading edge of the wing was gradually increased. This experiment indicated that there was an optimum distance at which the input energy to the wing was a maximum. This suggested that, in cases where it was not possible to increase clearance, some alleviation might be obtained by reducing the clearance.

Mr. Bunyan whetted the appetite, but he did not satisfy it with his remarks on engine balancing. It was customary to determine the magnitude of the unbalanced forces and couples of an engine in the design stage, sometimes with great elaboration. But when the figures were available, no one really knew what to do with them. Some guidance on permissible values would be very welcome, especially in cases where the selection of engine firing order was itself nicely balanced between the requirements of shaft torsional and hull flexural vibrations.

It was reassuring to note Mr. Bunyan's remarks about the significance of stresses resulting from axial vibration of the crankshaft under normal conditions, unless they were excited by a strong torsional critical. This confirmed the impression one gained from reading the literature on the subject of axial vibration of the crankshaft; but it would be useful if Mr. Bunyan would explain in fuller detail the meaning of the phrase "under normal conditions". It was often the exceptional circumstances that were of the greatest interest.

With regard to the practice of staying an engine to the ship's side to reduce the amplitudes of the rocking modes of engine vibration, he gathered that the cases quoted in the paper were of tanker installations. It would be useful to know whether similar successful results had been obtained with midships installations. His own experience of troublesome transverse vibration was on a twin-screw midships installation where the two engines vibrated against one another like the prongs of a tuning fork. The expedient of staying one engine to a casing on the adjacent shipside was tried but this only resulted in severe damage to the casing plates. The cure was eventually found by fitting cross-ties between the tops of the two engines.

Mr. J. F. R. Ellison (Member) said that the paper gave in a concise form some useful information on the salient features of design which must be carefully studied if the hull and machinery were to be free from troublesome vibrations which, as the author stated, were not only annoying to personnel but might well be disastrous.

He would not comment on the first half of the paper, as it mainly concerned the naval architect, except to mention in passing—with reference to Fig. 8—an example of severe stern post erosion coupled with transverse vibration at the stern of three single-screw vessels with which he was concerned some years ago. This was attributed to the fine clearances between the propeller blades and the aperture. Both these troubles were almost entirely eliminated in a further three similar vessels which were subsequently built, having the aperture clearances very similar to those quoted in Fig. 5.

With regard to the use of resilient mountings for generator sets, in his experience the flat slab type of composition mounting required constant following up on the holding down bolts, especially if a severe torsional disturbance was passed through when running up to speed, and in the end the resilience was lost. The flexible pipes, which were necessary for all services when using resilient mountings, were also a source of trouble.

The more modern rubber bush type of mountings might be more satisfactory, and at least one maker had adopted these for three-point mounting of medium- and high-speed sets. It might be advantageous to use these where passenger accommodation was near the machinery space, as in cross-Channel vessels. In large motor vessels the annoyance to passengers was caused more by mechanical noise transmitted up the exhaust pipes and through their attachments to the engine casing. This was where sound *and* heat insulation connexions were very necessary, but he had yet to find one that was really efficient.

Turning now to bolted assemblies, he agreed with the

In most cases that he had seen, taking, for example, one of the palm bolts securing the lower end of a Doxford side rod to the crosshead, the failure had been due to the head of the bolt not bearing solidly all round, or the fillet had been fouling the bevelled edge of the hole; thus the bolt had been able to work slack and fail through fatigue. The remedy, of course, was the simple, nay, elementary, precaution of testing round the bolt head with a feeler gauge. Having each of the large bolts throughout an engine drilled and fitted for the use of the micrometer gauge illustrated in Fig. 11 would add considerably to the cost and in his opinion was not justified.

A number of engine builders used a hydraulic jacking device for applying a specified tension on tie bolts while the nuts were screwed up, and also for applying a specified torque on piston rod nuts. He considered that more advantage should be taken of this technique, particularly the direct stretching method, as the error due to friction on the threads was ruled out.

The small bolts which were most likely to give trouble were those fitted to generator engine bottom end bearings. The use of fine threads on these bolts made it quite possible for a strong man—using a spanner with about 24-inch leverage—to stretch them beyond the yield point. It was therefore most essential to use some form of outside micrometer gauge to measure the extension, or here again to employ the hydraulic technique. A spanner using this principle was described in a paper* read before the Institute in 1946 by Mr. G. R. Grange.

The use of castle nuts for bolts such as these was in his opinion a menace, as in order to insert the split pin they were either slacked back or over-tightened. A locking pin with hardened steel cup-shaped point to bite into a collar extension of the nut was much to be preferred.

The author discussed top end trouble with Doxford engines and suggested that the chief reason was fore and aft oscillation of the bottom end, which had a side clearance of 10 mm. This was quite a usual figure for large marine engines. While this might be one reason for the trouble, he did not explain the cause of this oscillation. Unequal wear down or conical error would and did cause the bottom end to run to one side only and thus the top ends ran out of alignment with the crosshead. This became apparent immediately spare top end bearings were fitted, as one side often hammered out right away. Some of the main bearings in these engines would always wear down rapidly to a certain figure and then remain more or less static, particularly Nos. 2 and 5 in a five-cylinder engine, thus it was most difficult to maintain perfect alignment; this, however, was allowed for by the spherical bearings. The correct bedding in, and adjustment of the centre bottom end spherical, was most important, as if this unit was locked either by too fine adjustment or the hammering of the metal of the connecting rod into the oil grooves, then any conical error of the crank-pin tended to throw the rod over alternately fore and aft at the top end, thus firing up the side bearing strips of the top end and finally hammering the metal of the bearings themselves. The machining of the guide slipper in the manner described in Fig. 14 might alleviate the trouble, but would allow excessive movement of the piston rod spherical, which might lead to trouble with the piston cooling inlet and outlet pipes. The latter, being very short and stiff, were liable to prevent the free working of the spherical and cause the piston to bear hard on one side in an athwartship direction.

In conclusion, he would like to congratulate Lloyd's Register for the invaluable research work they were carrying out under the able leadership of Mr. Bunyan. Their efforts were proving of great benefit both to shipbuilders and shipowners alike.

DR. J. E. RICHARDS, Wh.Sc. (Associate Member) said that the author had set himself a difficult task in covering so much ground in one paper. He was to be congratulated on the result.

* Grange, G. R. 1946. "Engine Room Layout with Special Reference to Pipework". Trans.I.Mar.E., Vol. LVIII, p. 107.

It would be most unwise, however, to assume that all vibration troubles were so clear-cut as those mentioned in the paper. The statement was made in the first paragraph that "The basic conceptions are usually simple and straightforward . . .". Personally, he questioned this statement, and in order to illustrate the point he proposed to refer to the results of some tests carried out by the British Shipbuilding Research Association.

The basic assumption adopted in the study of ship vibration was that a ship behaved as a simple beam and every part of a transverse section of the hull vibrated with the same amplitude.

Vibration tests were carried out on a standard 10,000-ton d.w. dry cargo ship and the results showed that, for this particular vessel, this basic assumption was not true for the higher modes of vertical vibration. Vibration amplitudes were recorded on a ship section at the mid length of No. 2 hold, which was near an antinode of both the 3- and 4-node modes of vertical vibration, and the results showed that the amplitude at the centre of the tank top was three times that of the side girder when the ship was vibrated in the 3-node vertical mode, and ten times that of the side girder when the ship was vibrated in the 4-node mode of vertical vibration. The bottom structure was vibrating as a panel with appreciable amplitude and this might appear rather surprising, but it should be remembered that, when the ship was vibrating, the water was in motion and this was equivalent to a loading of the bottom with entrained water of roughly the same mass as the loaded displacement of the hull.

The results showed that the natural frequency of the bottom of the hold was of the order of 300 to 400 c.p.m. and it was interesting to note that for higher modes of vertical vibration, the bottom vibration would be out of phase with that of the hull and consequently, although the 4-node frequency was about 240 c.p.m., the 5-node frequency would probably be very much higher, perhaps 600 to 700 c.p.m. This was because, for hull frequencies above the natural frequency of the bottom, the amplitude of the bottom would be less than that of the side shell, and the effective mass of entrained water would be small.

This phenomenon was probably a feature of the horizontal vibration of dry cargo vessels as well as the vertical vibration because there was an entrained water effect on the side of the vessel when vibrating horizontally and the strength of the ship side in bending was very much less than that of the bottom.

In oil tankers the transverse bulkheads were not so widely spaced as in dry cargo vessels, and the presence of longitudinal bulkheads would also reduce this drumming effect, except in way of the engine room. In the engine room of a tanker the effective stiffness of the bottom structure was reduced by the inclination of the sides to the vertical. In the tanker mentioned by the author the deep frames had parted from the tank top of the engine room and the shell plating had cracked, due to the vessel being operated for a long period at an engine speed corresponding to the 2-node vertical mode of vibration; the amplitude of hull vibration being appreciable.

From these results, one might conclude that it was not feasible to calculate the frequencies of the higher modes of vibration with any certainty. Although the table given on page 104 was very useful, he himself was of the opinion that the results were not sufficiently accurate for use in design.

The vibration at the bottom of a vessel as a panel was a subject of some importance, because of the comparatively low frequencies. In some types of small Diesel-engined passenger vessel, it might be necessary to recommend the fitting, in certain circumstances, of resilient pads to the top of the engine room pillars to isolate the bottom vibration from the superstructure.

A higher frequency vibration was liable to be more troublesome in tankers than in cargo ships, and in the case of a 30,000-ton tanker, the 6- and 7-node modes of vertical vibration could be distinguished easily. The frequencies were so closely spaced near the normal engine speed that some degree of magnification of vibration could not be avoided.

With regard to propeller aperture clearances, he would like to ask whether the width of the blade at the position of minimum clearance was a factor which should be taken into account in deciding what this clearance should be. It would appear that with the increase in size and power of single-screw vessels it might become necessary to adopt six-bladed propellers to reduce the various troubles which could arise from the action of the propeller.

Mr. B. Hildrew, M.Sc. (Associate Member) said that in a paper covering such a vast amount of ground, it was only possible to comment on those points which were perhaps liable to arouse controversy.

It would be useful if the author could place an upper and lower limit on the use of acceleration as a guide to the acceptability of vibration to (*a*) the human frame, and (*b*) the ship and engine structure and their local parts. For example, a ship having a natural frequency of 60 c.p.m. and 1in. single amplitude at the stern had an acceleration at that point of about 3ft. per sec.^{2}. This was not dangerous. A steam pipe vibrating at a single amplitude of 0·002in. at a frequency of 1,500 c.p.m. had an acceleration of about 3ft. per sec.^{2}. This was not dangerous either. However, amplitudes of about 0·02in. at a frequency of 400 c.p.m. also had an acceleration of 3ft. per sec.^{2} and were extremely uncomfortable. Such vibration was liable to excite considerable local panel vibration aft and on the bridge. Local vibration on the bridge was becoming common in welded ships, and recurrent damage to radar and to other navigating equipment was frequently reported.

It was impossible to estimate accurately the 4-node hull frequencies, and accordingly in new construction of this type particular care should be taken that the secondary couple especially was kept to a minimum. The reason was that in a hull structure with a 4-node mode of vibration the engine was close to a node, and the secondary unbalance force would not have much effect, whereas the secondary couple could have a considerable effect.

As stated in the paper, the engine builder chose his crank arrangement to reduce the torsional criticals in the engine operating range and he often introduced a large secondary couple in the engine in so doing. If the hull structure was such that the natural frequency of the 4-node mode occurred in the region of 200 c.p.m. and the engine speed was about 100 r.p.m., then a very unpleasant vibration could occur at full engine speed.

During the past year, six or seven vessels vibrating in the 4-node mode of hull vibration had been examined. The cause of the vibration had always been insufficient aperture clearance which gave a 4/engine revolution hull vibration, or secondary unbalance of the main engine which gave a 2/engine revolution hull vibration. The development of the large ore carrier-tanker had apparently produced hulls which had a 4-node vertical frequency in the deep condition of about 210 c.p.m.

Once the ship and engine were built, the possibility of influencing to any great extent the secondary unbalance by modifying the engine reciprocating masses was remote. One could not add to or subtract from the masses enough weight to have any effect on this couple.

The usual academic solution was the provision of balance weights to operate at twice engine speed, one at either side of the engine, but it was not a practical proposition in the engine room of a ship. In fact, the only practical solution was to fit a finer pitch propeller or to crop the blades of the old one and so permit the engine to operate at higher revolutions.

Insofar as the table on page 104 was concerned, where the author gave the approximate relationships between the frequency of the various modes relative to the 2-node vertical frequency, he (the author) should point out that it depended to some extent on the construction of the ship. It was doubtful whether a welded and riveted ship could be judged by the same factors.

With regard to aperture clearances, the greater clearance a propeller could have in an aperture the better. Naval architects had assured him for years that such was not the case, but the conclusion that tight apertures were conducive to efficient propulsion was arrived at when the errors in model work were much greater and when the high block coefficients of tankers were less prevalent. He felt certain that as techniques in tank work on tanker models were improved, they would tend towards greater clearances all round, fore and aft. This statement was based on full-scale work carried out on board ships and recent model work in Sweden and Holland would seem to support it. In consequence, it was probable that the author's remark on page 111, suggesing a complete re-design of the single-screw aperture, would one day be realized —and in the not too distant future.

Referring to "tuned" ballasting, where vibration occurred in the ballast condition it was possible to alter the hull frequency by modifying the ballast arrangement. As stated in the paper, the last four tanks seemed to be those having the greatest effect.

To take an example, he himself was on one tanker recently in ballast where, owing to secondary unbalance of the engine at full speed, it was extremely unpleasant to stand in the steering compartment. Unfortunately—or perhaps fortunately—there was propeller-excited local vibration on the bridge at full speed which caused recurrent damage to the navigating equipment. In consequence it was decided to reduce the propeller diameter by cropping the blades. A week later he went on the ship with an instrument to measure the vibration caused by the secondary unbalance. He went out on the ship in very good sea conditions. When he tried to repeat the condition with instruments at identical draughts fore and aft he could not get the hull vibration at 212 r.p.m. Instead, amplitudes of the order of 0·01in. at 215 r.p.m. were obtained. An examination of the ballasting plan indicated that on the second trial 800 tons of ballast in No. 8 centre had been distributed into Nos. 9 and 7 centre and adjacent wings. Accordingly, ballast was run down to No. 8 and with 400 tons transferred the amplitude had increased to 0·04in. at 213 r.p.m. Again, it was very uncomfortable in the steering compartment.

He wished to make two points relating to this technique: In the first place, the movement of the ballast did alter the natural frequency of the hull. Secondly, even a small movement of the ballast did seem to affect the damping characteristic of the hull. He would like the author's comments on this last point.

With regard to the same ship, the fact that the propeller was cropped eliminated the bridge vibration, but rather to everyone's surprise the ship was found to go a knot and a half faster than on her trials.

With regard to the transverse rocking modes of the engine, the frequency of vibration was usually the number of firing strokes per revolution multiplied by engine r.p.m. These frequencies tended to excite local panel vibration in accommodation and on the bridge.

As stiff an engine seating as possible with good anchoring to the hull in the construction stage was the answer. Once the ship was built, the provision of struts was the only simple solution. Whilst it could be effective, it was usually a very inconvenient addition to the engine room.

With respect to gear noise, there was one point that often struck him, though the author had not mentioned it. Whilst notice was taken on a ship if the captain's door rattled, the engineer on watch did not get the same consideration. Noisy gears could be very wearing on personnel. To spend a four-hour watch close to a noisy gear case and to come back with the sound still singing in one's ears was prostrating physiologically, even though the gearing was not wearing and it was not dangerous mechanically. The author's suggestion that the gear case should be lagged in order to reduce the noise, especially if this represented a reduction of 40:1 in the sound intensity, was a sound investment on the part of any shipowner. The additional work obtained from the ship's engineers due to the improvement in environment would pay for the modification in a few weeks. There is no apparent reason why an owner should not specify a maximum noise level at the manœuvring platform for his new construction.

With regard to top end trouble, Remedy II on page 110 was only applicable if the spherical type of bearing was fitted. Top end trouble with the orthodox top end necessitated the full treatment along the lines specified in Remedy I and was a long job.

DR. A. J. JOHNSON congratulated the author on the clarity with which he had presented his various points: it made the paper very pleasant to read.

He had a few queries and suggestions with regard to the first part of the paper. In the first place, there was a natural tendency in the opening sections of the paper for the specialist in the subject to over-simplify the problem. Other speakers had already drawn attention to this. One was left, for instance, with the impression that after the necessary observations had been made one was in possession of a clear picture of the vibration characteristics of a vessel. He had frequently attended trials with first-class measuring equipment only to find a very complex situation which demanded great ingenuity to define in the simple terms of the paper.

It would be helpful if the author could take the section on observations a little further to deal, for instance, with such matters as: The procedure in measurement of hull torsional vibration which, to the uninitiated, might be easily confused with transverse modes. The distinction of cause when propeller blade frequency coincided with engine secondary excitation, or when first order forces from a propeller coincided with primary engine forces. It was not always a practical proposition to uncouple the propeller from the main engines to make sure of these points.

He would like to disagree with the author's statement on page 102 about the human body being a good vibrometer until such time as Mr. Bunyan had devised a means of calibrating the individual. The human body was very sensitive but not quantitatively.

He had one or two points with regard to propeller-excited vibrations (page 103). Mr. Bunyan made positive recommendations for minimum aperture clearances. Everyone appreciated that something of this sort was required, but a large number of variables were involved in this problem. In order to make an appraisal of the results in Fig. 5, it would be necessary to have at least a summary of the techniques, the data and the reasoning of which it formed the ultimate conclusion.

It was stated later that the figures were based upon model experiments and full-scale work. Perhaps Mr. Bunyan would be good enough to supply the data for the build-up of the results.

Fig. 6 was an intriguing diagram. In moving the propeller forward to give increased clearance between the trailing edge and the leading edge of the rudder, one might expect a reduction in transverse vibration. Such a result was, in fact, indicated in the diagram. At the same time, however, the parameter B was being reduced and it might be natural to expect the vertical vibration to be somewhat larger. Fig. 6, in fact, showed this to be reduced even more than the transverse. Perhaps Mr. Bunyan would comment on this.

In such data as was presented, it would be interesting to have some ship particulars, where the measurements were made and possibly the loading conditions pertaining to them.

A number of speakers had drawn attention to the anomalies of Fig. 7. When attempting to isolate a single fact such as, in this case, the cropping of the propeller tips, it was essential to keep all other things equal. There was obviously a great difference in the loading condition between these two tests before and after the cropping. If this ship was fairly large, as it appeared to be, it probably represented a difference of several thousand tons of water. One other point that had not been mentioned previously by other speakers was the immersion of the propeller blade which greatly influenced the magnitude of the exciting forces.

In case Mr. Hildrew's remarks on this section were taken too literally, it should be pointed out that the ship must have been in dock before the tips were cropped. Therefore, she undoubtedly left with a much cleaner bottom. Allowing for a little exaggeration on the figure of 1½ knots, this fact might possibly account for the increase in speed.

With regard to the approximate formula on page 104, new methods for estimating hull frequencies were always welcome, but the accuracy of a formula could only be judged by its application to a large number of ships upon which accurate experimental data were available with regard to hull frequencies. The author must have these data in order to arrive at the coefficients. It would save potential users much time if the degree of accuracy expected could be judged by a table of figures or, alternatively, by the scatter of points on a graph.

He had not studied in detail the composition of the formula, but he was a little surprised that it did not include the direct values of the moment of inertia and displacement which were available in the design stage. These might be considered much more reliable than derivations from basic ship dimensions. C_b as defined also incorporated the load draught moulded. It was difficult to understand why the mean draught condition considered was not used, since C_b was a variable quantity with the draught.

With regard to the use of effective depth, D_E, he was of the opinion that it might lead to serious error. The method was based on solid rectangular sections and assumed simple beam theory to be strictly applicable. The effects would inevitably be overestimated since the scantlings of a deckhouse were lighter than those of the main hull girder, and it was known that the interaction between the hull girders and the deckhouse was frequently such as to give appreciable departures from the beam theory as applied to the hull structure.

There was a puzzling statement at the top of the second column on page 105 concerning tip clearances. He failed to understand why the propeller tip to arch clearance greatly influenced the transverse forces transmitted to the hull. He thought that the tip clearance might be more important in relation to vertical vibration while the trailing or leading edge clearance was more vital in transverse vibration. Perhaps the author would enlarge upon this point.

He would like to know what modes of vibration were depicted in Fig. 9 and exactly how the magnitudes of the forces were derived. He did not see how, without a very complete understanding of the damping characteristics of the vessel, one could get primary forces without the use of a vibration exciter.

MR. BRYAN TAYLOR, B.Sc.(Eng.) (Member) referred to tailshaft stresses. In considering Table III, he had been struck by the great difference between the total vibratory bending stresses measured and the cyclic bending stresses set up by the weight of the propeller. It would be seen that the total vibratory bending stresses approached four times those due to the propeller weight.

In this connexion he noted that the American tests had shown that under heavy weather conditions the stresses could reach twice the values obtaining in calm weather. He enquired whether the figures given in Table III referred to calm weather conditions; if they did it meant that one could expect alternating bending stresses in the tailshaft of the order of ± 13,000 to 14,000lb. per sq. in. under certain conditions. A further consideration of these figures suggested that if this were the case the margin of safety of shafts dimensioned according to Classification Society rules could in certain circumstances be reduced almost to zero.

Little was known about the fatigue strength of very large shafts, but it seemed probable that the fatigue limit under reversed bending would be of the order of ± 30,000lb. per sq. in. Thus, if the strength reduction factor caused by stress raisers such as the keyway, the effect of the key loading or of the clamping of the boss, reached a value greater than two, the actual stress in the shaft would exceed the limiting stress range. In that case, one would expect more frequent tailshaft failures than actually occurred. It would appear, however, that as more information was gathered together on this subject, it might be necessary to add refinements to the rules of the Classification Societies to allow for the various factors that influenced tail-shaft stresses in high-powered single-screw ships.

One point the author did not make in his list of suggestions on pages 111 and 112 for reducing the possibility of fatigue failure in tailshafts was an increase in the diameter. In (4) he suggested that a positive drive of the propeller up the taper of 0·003 inch per inch shaft diameter should ensure that the grip of the cone would take most of the drive, thus reducing the stresses at the key. Approximate calculations suggested that one would need considerably more than that drive in order to transmit the torque by friction. But it was of little use to suggest that the propeller should be driven up any given amount if it could not be done in practice. From experience, it would seem that even to drive a propeller 0·002 inch per inch diameter of shaft up the taper beyond the shop fitting mark was about the limit that could be obtained with the normal wedges. This was the case with a large propeller but for a small one it might be comparatively easy.

It seemed to him that in order to transmit most of the torque to a large propeller by friction an alternative method of fitting the propeller would be needed, either heating the boss before tightening up or perhaps some hydraulic means of pressing on the propeller. He would emphasize that this appeared to be one of the best ways of reducing the stress concentration at the propeller tailshaft assembly.

One small point occurred to him on reading the section dealing with the alignment of crankshafts. With regard to the apparatus illustrated on page 108, he understood from the paper that the staff was set up vertically by means of a spirit level at each journal in turn. Could the author give some indication of the accuracy with which the apparatus could be set up from one point to the next by means of the spirit level alone? He would have thought it possible that the horizontal errors at the micrometer might well equal the errors one was trying to measure.

COMMANDER(E) J. I. T. GREEN, O.B.E., R.N.(ret.) (Member) said he had no criticism of the paper, but rather hoped to underline a few points.

He was as surprised as Dr. Ker Wilson when he first read that torsional vibration had nothing to do with hull vibration. He thought he understood what the author meant, but anyone who had been to sea in the old type of submarine would say that there was a link between these two, which in combination was known as "going over the cobbles".

So many of the vibration problems referred to in the paper had their origin in the propeller and shafting that he wondered whether it would not be a good thing to have a simple instrument available which could be set up in the ship for trial purposes for measuring the cyclic variations in torque. Such an instrument had been made in the British Iron and Steel Research Association laboratories, using slotted discs made from armature stampings, a light and a selenium cell. The discs were attached to split flanges fitted temporarily on the shaft. The current passing through the cell was picked up by a cathode ray oscilloscope and the cyclic variations could be very easily picked up. This was strictly a laboratory instrument and the principle was well known. He would be glad to hand over the description and sketches to the author of the paper for his consideration.

On the question of bolts, since he had left the sea service he had found the same lack of understanding among engineers on the relationship between the stiffness of the bolt and the stiffness of the seating. He then made a plea for a simpler form of diagram than that shown in Fig. 10, which he found difficult to follow.

B.I.S.R.A. had been particularly interested in the question of tightening bolts in rolling mills. Very large forces were exerted on the bearing cap bolts of the roller tables. When large ingots were dropped sideways on to the rollers, forces of over 60 tons were recorded by strain gauges on the bolts. The first step in the investigation was to measure the initial tightening load on these bolts. It was found that 8 tons was all that could be imposed on a normal type of bolt, but that after four working days this load had entirely disappeared. An improved design of bolt gave an initial load of 33 tons which dropped to 23 tons in the same period. The peak loads arising from sudden external shocks in the two cases were 60 tons and 30 tons respectively. A report of these tests would appear shortly in the I.S.I. Journal.

MR. J. B. GRIFFITH, B.A. (Member) said he would like briefly to outline a problem he had come up against. Mr. Bunyan and some of the speakers in the discussion were already aware of it. He would conclude with a shocking generalization and see whether he could get Mr. Bunyan to agree with it.

Since the war, his company had been replacing a fleet of river steam tugs by Diesel tugs. The Diesels had all been two-stroke direct-coupled engines, and in the early days the company built tugs of powers roughly equivalent to the steam tugs which had been performing the duty before. They had no difficulty at all with powers up to about 500 b.h.p. but over that h.p. they ran into certain difficulties which roughly took the following form.

The hull form of these tugs was, of course, very restricted by considerations of draught and overall length. Because of the direct-coupled engine, the shaft line was of necessity very low and in order to meet the requirements of draught it was necessary to cut away a lot of the "meat" in the floors beneath the engine, so as to accommodate the sump of the engine. In addition, the floors were further cut away, perhaps unwisely, towards the side frames in order to accommodate the bunkers, which were a separate entity.

There was trouble in those tugs in the 700 h.p. range, due to rocking in the engine. This was undoubtedly brought about by weakness (indicated) with the result that each of the girders was moving up and down in opposition. As one went down, another moved up, causing a horizontal movement on the top of the engine. The force exciting that vibration was due to torque reaction in the cylinders.

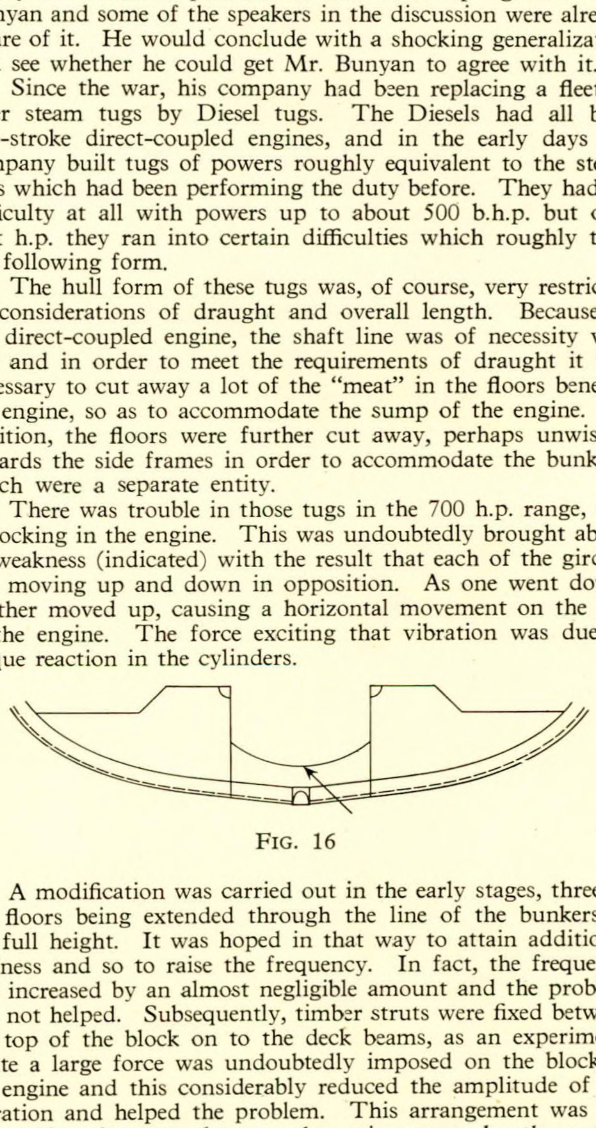

FIG. 16

A modification was carried out in the early stages, three of the floors being extended through the line of the bunkers at the full height. It was hoped in that way to attain additional stiffness and so to raise the frequency. In fact, the frequency was increased by an almost negligible amount and the problem was not helped. Subsequently, timber struts were fixed between the top of the block on to the deck beams, as an experiment. Quite a large force was undoubtedly imposed on the block of the engine and this considerably reduced the amplitude of the vibration and helped the problem. This arrangement was not acceptable, of course, because these river tugs, by the nature of their work, were always liable to receive heavy blows from other craft, which would throw back an undue load on to the engine. A mitigation of the difficulty was eventually secured by the use of wire rope ties from the engine through bolts on to the side of the vessel. Special arrangements were provided so that the tension of the ties could never exceed a quantity which was thought to be safe.

After they had had difficulty with the direct-coupled engine, the company investigated the frequencies at which they would be likely to run into difficulty with other types of direct-coupled engine. They found they were likely to meet severe engine-excited hull criticals at some stage within the range, depending on the mass of the engine and the height of its centre of gravity.

The rash generalization on which he would ask the author's opinion was: for a river tug with its very stringent limitations of draught, and so on, was it fair to say that when the brake horsepower was over approximately 550, a geared drive was essential in order to raise the crankshaft sufficiently high and to provide enough strength in the floors on the centre line to overcome this problem?

MR. P. D. FRASER-SMITH, B.Sc., said, with reference to "tuned" ballasting of tankers, that the author rather gave the impression that almost any hull vibration occurring in the ballast condition could be removed from the service speed range by judicious redistribution of the ballast. But bearing in mind that longitudinal bending moments must be kept within reasonable limits, the longitudinal distribution of inertia could not normally be altered sufficiently to have any great effect on the natural frequencies of the lower modes of vibration. This method of altering the critical frequencies was, in fact, more effective the higher the mode of vibration.

For example, while one might expect to alter the 4-node critical by as much as 70 or 80 c.p.m., which would be equivalent to 20 r.p.m. if the vibration was of the fourth order, 2-node criticals could not be altered by more than a few cycles without a radical change in draught or an excessive increase in longitudinal bending moment.

Secondly, the author had stated that little or nothing could be done with ballasting to eliminate a strong propeller-excited critical on the running speed in the loaded condition. Here he was presumably referring to the redistribution of cargo and not ballast. It must be agreed that there was generally insufficient cargo capacity to permit of much redistribution. However, if the ship was engaged in the carriage of heavy oils, there would generally be several cargo tanks empty in the loaded condition. This might enable sufficient redistribution of cargo to be made to alter the critical frequencies at least of the higher modes of vibration.

Longitudinal strength must take precedence over vibration considerations, and Mr. Bunyan's proviso on this aspect could not be over-emphasized. In some conditions, a dangerously high longitudinal bending moment could be induced by filling one tank and emptying another. Some form of longitudinal bending moment calculation should therefore be carried out before resorting to redistribution of ballast.

With regard to propeller clearances, Mr. Bunyan had referred to some model experiments recently carried out in Sweden which indicated that the propulsive efficiency was largely unaffected by propeller clearance. But in the experiments in question only one hull form was employed—that of a 15-knot cargo ship. While the conclusions drawn therefrom were no doubt largely true for the majority of cases, he had recently heard of some tests on large tanker forms which did indicate that propeller clearances had a noticeable effect on the propulsive efficiency. Nevertheless, it must be agreed that clearances should always be designed to obviate propeller-excited vibration rather than give maximum propulsive efficiency, if these two considerations conflicted with one another.

Correspondence

DR. J. F. ALLAN remarked that, as the title stated, this paper dealt with a practical approach to some problems of ship machinery installations and it contained many results of the author's wide experience in investigating vibration and other problems arising from these. It was not proposed to go into the paper in detail, but there were some points in it which had arrested the attention of the writer.

First of all, the propeller clearances indicated in Fig. 5 were important and were more or less in line with those published by the writer in reference 17. It was probably worth commenting that the necessity for clearance E to be at least equal to T to avoid vibration trouble with thick rudders arose not so much because of the thickness of the rudder as because of the full round nose usually associated with this type of rudder. If the rudder were reduced to a fine leading edge, the clearance could be reduced to about 8 per cent of the diameter.

The instances given in Figs. 6 and 7 illustrating the effect on vibration of propeller clearance in the aperture were certainly impressive, but the question of cutting propeller diameter by 5 per cent or 7 per cent was one which must be entertained with great caution, as this might have a rather serious effect on the propeller efficiency. Individual cases would have to be considered on their merits.

The formula given on page 104 for the frequency of 2-node vertical vibration seemed to be an odd departure from the fundamental form of $f = \beta\sqrt{\frac{B.D_E^3}{\Delta_1.L^3}}$ In view of the amount of work which had been carried out in studying the variation due to several factors of the constant β in the above formula, it seemed a retrograde step to introduce an inverse linear function to an empirical index in the "constant" part of the equation and an arrangement under the square root sign which was not dimensionally correct. The proposed formula had been applied to four specific cases covering different types of ship, and in comparison with the Todd-Marwood formula it gave a much poorer agreement with the actual frequency in every case. The approximate relationships given on page 104 between the various modes of vibration might be a useful guide but they should be treated with considerable reserve. It might be remarked, for instance, that 1·54 was rather a precise approximation.

The statement on page 105 that the use of a thick streamlined rudder would give a better performance than a thin rudder was scarcely true and was not in line with their experience at the National Physical Laboratory. The evidence in Lindgren's paper was most interesting and tended to show that a good streamlined rudder form of 15 to 20 per cent thickness ratio gave the best performance. Thick rudders of the type referred to by the author were usually approaching 25 per cent thickness ratio and frequently had a round nose as shown in Fig. 5 of the paper. This was not a good streamlined form, and in their experience caused a loss in performance of some two or three per cent on such rudders.

The author made reference on page 111 to the problem of tailshaft failures and remarked that these were largely confined to single screw ships. This was an interesting fact and seemed to line up with the much more violent variations of wake which were typical of the single-screw ship compared with twin-screw designs, especially if the propeller clearances in the aperture were not too adequate. Proposals had been made at one time and another to design the underwater afterbody ahead of a single-screw propeller as near as possible in the shape of a solid of revolution, and there was no doubt that by this means a much more uniform flow to the propeller could be achieved. Combined with a somewhat unorthodox overhung rudder mounting, an interesting design could be achieved on these lines, but so far as the writer knew such a design had never been built.

PROFESSOR J. W. BONEBAKKER wrote that vibrations in existing sea-going merchant vessels were sometimes insupportable for the crew, or dangerous for the ship structure. Lloyd's Register would have accumulated complete information about quite a number of actual cases. Could the most frequent of these be grouped under a few headings, such as, for instance:—

1. Type of vibrations.
2. Type of vessel (tanker, or vessel with machinery aft).
3. Size and proportions (range of length or tonnage, $L:D$).
4. Loading condition (distribution of W.B. in ballast condition, trim)?

The answer to this question might single out those vessels for which vibration calculations should be made in the design stage.

Could the qualification "insupportable" and "dangerous" (vibrations) be clearly defined quantitatively?

They were writing and speaking about vibration calculations, but was there a generally accepted method for these calculations? Were there still one or more uncertainties, and how did the results of these calculations compare with actual records?

MR. R. W. CROMARTY (Member) thought Mr. Bunyan was to be congratulated on covering so large a field so concisely. His title, "Practical Approach to Vibration Problems", encouraged him to offer the following as a contribution to the discussion, but his remarks were not altogether solutions to vibration problems; rather, they were a practical approach to vibration problems—in other words, problems which had been thought to have been due to certain vibrations.

In marine engineering they had, like other professions, their fashions and fads. When a part of an engine failed, it was fashionable today to blame "vibrations". "Latent defect" was somewhat out of fashion. But many years ago, when a Doxford engine side rod failed, those who first investigated the trouble asserted that no vibration then known to science could be the cause of the failure. It was subsequently proved that such a vibration was known to science and was the cause of the failure. The late Mr. Keller mentioned this case in his paper, "Torsional Oscillations in Marine Shafting" (Liverpool Engineering Society, January 1934).

When in later years his company had an epidemic of bolt and other failures in war and post-war built engines, the first thought was that they might have stumbled into another series of mysterious vibrations "unknown to science". However, the solution proved to be more matter of fact. Mr. Bunyan had mentioned some of these in his paper, i.e. those due to incorrect hardening up of bolts. Now that bolts were set up to a given elongation, this trouble had been eliminated.

It might be asked why the troubles experienced appeared in war and post-war built engines. The complete answer was not straightforward, but the ever-increasing tendency of builders to "assemble" an engine was undoubtedly part of the answer. By "assemble", was meant machining engine parts in comparatively large numbers to jigs, etc., and assembling them together to form an engine instead of carefully marking off the parts, machining them to the marks, bedding them carefully together and part by part fitting them to form an engine.

As an example of this tendency to assemble parts direct into an engine was the case of a Doxford engine in which the top end bearings "squeezed" out during the shop trial. This was not due to any of the many reasons given by Mr. Bunyan, but entirely due to the builders' practice of boring out the bearing a few thousands of an inch larger than the pin diameter and not bedding the bearings to the pins. A similar procedure probably accounted for the failure illustrated in Fig. 17.

There were those who asserted that marine Diesel engines need not be erected on a test bed and given shop trials, but erected direct into the ship from the machine. These "up-to-date" methods of erecting engines were praiseworthy only insomuch as it was in a way an effort to offset increasing costs

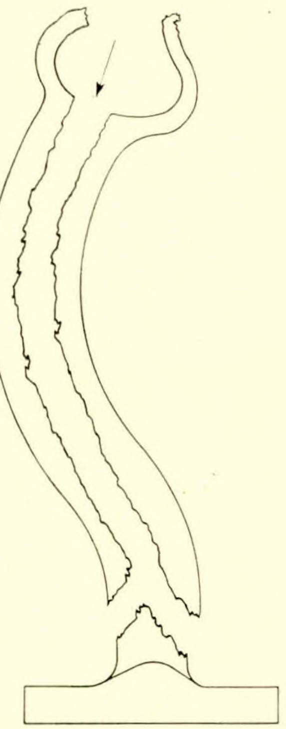

FIG. 17—*Generator connecting rod in service—fractured as shown from top end eye down towards foot*

and, at the same time, increase production, but this method unfortunately brought in its train "acceptable machinery discrepancies".

Mr. Bunyan had, in his excellent section on Doxford engine top end troubles, indicated that "acceptable machinery discrepancies" could be the source of such troubles. A few which had come his way were illustrated in Fig. 18 and were actually found in engines having bolt failures. They were discovered after it was found necessary to put a spare bolt in the lathe to turn it down to size because it had been supplied oversize (unknown to the ship it was the builders' standard practice to supply such spares oversize). When this spare bolt was in the lathe, with the shank and round head running true, the screwed portion was running a minimum of $\frac{1}{16}$ inch eccentric and about $\frac{1}{16}$ inch tapered on the length of the screwed portion out of line with the centre line of the shank (or fitting portion). The bolt was about 3 inches diameter.

Instead of these bolts being machined and screw cut in one lathe, the shanks were "turned" to size in an automatic capstan and then taken to a screw cutting machine where they were clamped on to a bed to have the die run down to form the screw, just as if these important parts were lengths of gas pipes.

Such methods had been discarded by most engine builders. Just recently he was shown, with some pride, by a machine shop superintendent, their latest screw cutting machines which now dealt with this type of bolt between centres. These machines apparently did the job, and did it better, in a matter of minutes that would take a skilled turner hours to do. The bolt nuts were also threaded by these machines, but he was quite certain that no skilled turner would screw cut a nut without taking a cut across the bearing face of the nut, and this was what this "efficient" machine, and its semi-skilled operator did not do. Every nut from this machine was a potential fractured bolt.

Not only must they be on their guard against "acceptable machinery discrepancies" in new work, but in repair work against "acceptable fitting discrepancies".

With one of Mr. Bunyan's colleagues, he had recently to

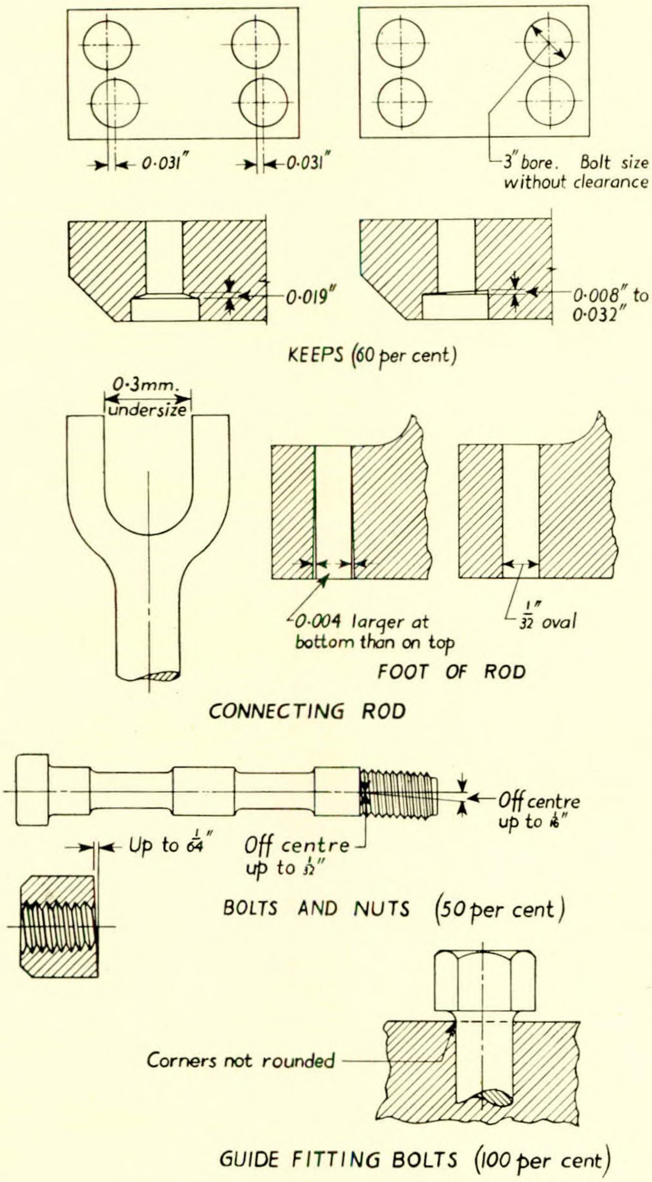

FIG. 18—*Machining discrepancies*

investigate why the top end bearings and guides in a pair of Diesel main engines were overheating in spite of the two lubricating oil pumps being supplemented by the spare pump, all running full out. To begin with, the problem was somewhat confused for the ship's personnel by one of the lubricating oil pumps (reciprocating type electric-driven through gear wheels) having bent its crankshaft. This was put down to "overloading". On investigation, the trouble was found to be due to a new pinion, the teeth of which had been cut ½in. pitch instead of No. 6 diametral pitch. The new pinion and original wheel (the teeth which were No. 6 diametral pitch) did not mesh properly and eventually the crankshaft, which carried the pinion, bent.

The lubricating oil to the main engines first went to the main bearings, then through holes in the crankshaft and pipes to the crankpins, and then up the centre of connecting rods to the top ends and guides. As the main bearings had been remetalled some two years previously, the trouble was thought in turn to be excessive working clearances, restricted oil ways, leaking pipes, leaking coolers, defective valves, and blocked pipes. These, and other suggestions, were speedily eliminated and the trouble established as entirely due to excessive scraping away of the white metal at the bearing horns, oil gutters in all bearings being too long, and the shims not properly fitted.

There was little doubt that when the main bearings were remetalled the white metal at the horns had been excessively scraped away to make refitting easier. Also, that when filing the radius round the end of the gutters, the file had cut across the ⅛-in to ¼-in. bearing surface between the ends of the gutters and the sides of the bearings. The shims, instead of being fitted as close to the pins as possible, had been cut to the shape of the so-called "oil reservoir" at the horns and the scraped away "bearing" surface at the side of the "oil reservoir".

The area of these unnecessary clearances in each engine totalled that of a pipe 1¼-in. bore. The supply pipe to each engine was 1¾-in. bore, so it was easy to appreciate why, even with three lubricating oil pumps working full bore, the top ends and guides were starved of lubricating oil and ran hot. The cure was to fill in the gutters and oil reservoirs and bed the white metal to give a bearing right up to horns.

No longer were three lubricating oil pumps required running at full bore (25 amperes each and delivering oil at 60lb. per sq. in.), but only two running comparatively light (10 amperes each and delivering oil at 40lb. per sq. in.).

No longer did the previously reported "overloaded" "undersized" top end and guides run hot. In fact, the guides which had been salt water cooled for many years had the salt water shut off and continued to run trouble-free uncooled.

Mr. Bunyan mentioned that in the design stage some factors might not be fully appreciated. This was brought home to him recently when a generator crankshaft fractured after about ten years' normal service. When the engine was designed, it was known that it had to pass through a major critical torsional vibration before attaining its running r.p.m. The first engine of this type was shop tested until the crankshaft fractured. From the number of reversals of stress needed to fracture this shaft, it was estimated that in normal service a crankshaft would last at least thirty years before fracturing, and so a damper was not considered essential. What had not been appreciated was that in service, due to wear and tear of working parts, the engine would run freer than on the test bed, hence when passing through the critical, the number of reversals of stress would be appreciably greater than the number recorded on the test bed. Also, the estimated number of "stops and starts" did not make sufficient allowance for stoppages to deal with leaking joints, etc. Dampers had now been fitted.

MR. G. DE WINNE mentioned that Mr. Bunyan had referred to the forces set up by a propeller when working in a single screw aperture and had given details of what he considered might be minimum clearances between the blade edges and the aperture. He would be grateful if Mr. Bunyan would indicate what he considered to be minimum propeller tip to hull, and shaft bossings to propeller clearances for acceptable vibration in twin-screw ships.

MR. CH. EVRARD wished to know whether there was any marked difference in the hull vibration, particularly at the after end of the ship, where riveted as against all-welded construction was used. He would also like Mr. Bunyan to indicate whether, in the case of cargo vessels, vibrations were influenced to an appreciable extent by the type of cargo carried.

MR. G. H. FAIRLEM, O.B.E. (Member) had read Mr. Bunyan's paper with great interest, and appreciated the practical approach to the problems concerned.

The trouble experienced with top ends, with particular reference to the Doxford engine, had at times been a source of considerable worry. The factors covering the difficulties were clearly emphasized in Mr. Bunyan's paper, although there had been occasions with his experience when all these factors had been eliminated, yet evidence of alternate loading, heating and metal squeezing still remained. One could only assume in this case that the connecting rod for some reason or other adopted under power conditions a position different to the true alignment indicated when the engine had stopped. Engine trials alongside the quay seemed to have the effect of exaggerating this condition and it was noticeable that under free running conditions the engine adapted itself more easily to the new alignment which it had taken under power loading.

He noted the suggestion made that the engine be run at 60 to 70 revolutions to allow parts to accommodate themselves after adjustment. He suggested that certain conditions would call for much lower revolutions initially depending on results, and wherever the opportunity occurred the trials should be carried out with the engine under free running conditions.

Much work could have been eliminated if these conditions had applied in the early stages of this top end trouble.

MR. N. V. GOEMAERE thought Mr. Bunyan had shown that considerable "tuning-out" of vibration criticals on the service speed might be accomplished by quite small redistribution of ballast in tankers and indicated that for the loaded condition there was little or no scope for such tuning. This was, of course, dependent on the gravity of the fuel carried, and in the case of dense cargoes there was considerable slackness in cargo tanks which could be used for "tuning" purposes. He would value Mr. Bunyan's comments on this matter.

MR. J. G. KONING wrote that some time ago somebody had told him that the propeller and its design were always the real causes of ship vibrations. This, of course, was right insofar that if the ship had no propeller and no engine there would be no vibrations either. It was, however, quite true that, for instance, in a single-screw turbine vessel where the engines and a good shaft alignment could not start vibrations, vibrations occurred due to the propeller revolving in the screw aperture. But could these vibrations be called "propeller-excited" and were they the result of a certain propeller design?

Most of these vibrations happened in ships with small clearances between the propeller and the aperture, and in modern high-powered tankers with rather high block coefficients. This led them to the real basis of the trouble, i.e. the wake distribution behind the vessel. Every wake field had a high wake peak in the upper part of the aperture. It was possible to design a propeller in accordance with the mean wake value of the whole screw disc; it was also possible to design every propeller section in accordance with the mean wake on the circle which is described by this particular section but, and this might sound rather pessimistic, it had up till now been impossible to design a propeller section which suited the high wake peak in the upper part of the aperture as well. This caused local cavitation, which practically could not be avoided, and vibrations. So the only thing that could be done at the moment was to reduce wake variations and local wake peaks as much as possible. This could be achieved by keeping the block coefficient of the vessel within certain limits, by fine ending waterlines in the aperture, by rather "U"-shaped sections in the afterbody, especially just in front of the propeller and by ample clearances between the propeller and the aperture.

Too many stern posts were too thick and too blunt, especially in the upper part of the aperture, and many clearances even should be larger than advised by Lloyd's Register.

He would be very much obliged if Mr. Bunyan would give his views on this aspect of the vibration problem.

MR. J. LENAGHAN found that the author's remarks, *inter alia*, were a statement of "do's and dont's" on vibration problems and measures to avoid them.

The recommendations for reducing propeller-excited vibration were worthy of attention, and if the proposed propeller clearances were followed they should in addition improve propulsive efficiency.

It was surprising in these days, when most hull forms and screws were tank tested, still to see in the stern arrangements of many existing ships how little regard had been given to the siting of the propeller within the aperture.

When designing single-screw stern frames, careful consideration must be given to placing the propeller in the aperture in order to provide adequate clearance between the trailing edge and the fore edge of the rudder, also at the same time every endeavour must be made to keep the propeller as far aft as possible in the aperture and well clear of the main hull; in other words, a good clearance forward as well as aft of the propeller was extremely important. The clearances indicated below Fig. 5 were good practice minima with perhaps one exception, "B", which, it was suggested, should not be less than $0{\cdot}125 \times D$ ft.

It was not easy with some of the present-day very wide streamlined rudders on the larger single-screw ships to provide a clearance equal to the maximum width of the rudder, between trailing edge and fore end of rudder. A rudder of 3ft. width and a clearance of equal amount between propeller trailing edge and the rudder could provide problems in supporting adequately the overhang of the stern frame heel piece.

He was surprised at the author's suggestion that there might be a good case for an increase in the scantling of the structure and shell plating in way of cruiser stern of large single-screw ships. Some of the problems in this vicinity had little to do with scantlings; they resulted largely from defects in design. Many vibration problems at the stern and elsewhere, particularly local vibration, never would arise if more thought were given to the structural arrangements and to the addition of local stiffening in the proper place at the proper time. It was an easy way out to think in terms of additional pillars and thicker materials as the cure for vibration. Neverthless, it was agreed that pillars properly disposed, together with continuity in the deck girders and other longitudinal members, undoubtedly would go a long way towards eliminating many vibration evils.

He did not entirely agree with the suggestion that large flat areas of shell plating in way of the stern should be avoided. Again, it was a question of design and in a properly designed arrangement such areas should behave quite satisfactorily, and no better example of this existed than the flat sides, etc., at the after end of many naval vessels. The structure of these ships had to resist and support much greater powers and faster turning screws than was the case in merchant ships, and for various reasons it was not always possible to avoid flat surfaces, nor could it be said that the scantlings anywhere in these ships, flat sided or otherwise, were substantial, but invariably the structural design was good.

MR. F. MCALISTER (Member) thought that this was a most vital paper affecting the satisfactory performance of ships at sea and provided it was realized that the vast majority of ships were reasonably trouble-free and that this paper dealt with a minority of such problems, one was then in a better position to appreciate Mr. Bunyan's excellent approach to each case.

No one person was better able than Mr. Bunyan, in his official capacity, to appreciate the difficulties arising from those diseases, such as vibration, affecting ships in service, as such diseases were always bound to occur where insufficient knowledge was available on any one particular case to predict with accuracy possible sources of trouble. He cherished Mr. Bunyan's wish that the information given in his paper might be helpful in avoiding repetition of similar troubles in the future.

He did not wish to enter deeply into the questions of hull vibration nor of engine vibration as these would, doubtless, be dealt with in a more capable manner by the experts in that field.

It had fallen to his lot, however, to spend many years in studying propellers and he had yet to hear of any case of vibration which could in any way be associated with the designs for which he had been responsible. Rather the reverse. Many hundreds of cases of vibration had been placed before his company and once the diagnosis had indicated that the existing propeller had been the likely source of vibration, the change in design had always proved eminently effective.

He agreed entirely with the clearances Mr. Bunyan advocated in Fig. 5 as similar values have been their guide for many years. No doubt in low powered vessels lower clearances than these could be shown not to have been conducive to any vibration, but in the modern higher powered vessels the clearances given were reasonable to ensure freedom from vibration.

Just one word of warning, however; if the tip clearance at the top of the arch were increased considerably above 10 per cent of the diameter, a cross flow through the arch might develop which would affect efficiency.

On the other hand, the leading edge clearance of 15 per cent could be increased always with benefit to the efficiency. If too constricted the proximity of the stern frame prevented the development of the normal rotational inflow and disturbed the propeller efficiency.

Mr. Bunyan described an outstanding case on page 102, where owing to vibration in a tanker the propeller was cropped by 7 per cent as a temporary expedient. As he showed in Fig. 7, this had the immediate effect of reducion of vibration. However, the revolutions must have increased by about the same 7 per cent, i.e. from 84 to 90 r.p.m. The efficiency must also have dropped by 3 to 4 per cent, and if the original propeller were anywhere near to cavitation breakdown the consequent loss in area and increased rotational speed must have precipitated the cropped propeller into a zone of slow erosion. They were not given the overriding features of this case and although the surgery was drastic and successful he trusted that a subsequent change to a good propeller giving the designed revolutions, high efficiency and freedom from vibration, was ultimately made.

In good propeller design many features conduced to quiet running, and varying pitch and a modest skew back, together with other requirements, all lent strength to the designer's arm in encouraging freedom from vibration.

Later in the paper Mr. Bunyan mentioned the "Jason" experiment and further recordings of propeller stresses he had made in actual tests on propellers at sea and he felt that a few words could be added on this subject which might amplify the author's comments.

The calculation of the stress in the root of a propeller under a given power, revolutions and speed in a fully immersed, steady motion condition was finite and precise. For normal propellers this stress, in his company's designs, rarely exceeded 6,000lb. per sq. .in. (2·7 tons per sq. in.) for bronze scantlings and this for a material which could confidently give a test bar up to 32 tons per sq. in. if not much more.

This nominal factor of safety of 12 might appear high to those whose work lay in other branches of engineering; but consider what the propeller had to contend with over a long life: —

1. Uniform motion in propelling the ship.
2. Dynamic forces in waves.
3. Racing in heavy weather.
4. Manœuvring, full ahead to full astern.
5. Striking ice or wreckage.
6. Unevenness of transmitted torque.
7. Fatigue consequent upon vibration.
8. Dynamic impulses from stern frame or shaft bossing.
9. Variation in casting stresses in material.
10. Treatment of propeller during handling, and probably many others.

Having allowed for No. 1 by careful calculation, Nos. 2 to 10 could only be allowed for by judgement, even after pondering deeply on such imponderables.

When, therefore, Mr. Bunyan mentioned that the blade stresses on a "Sam" ship were recorded as ± 5,000lb. per sq. in. and added in parenthesis that the propeller failed in service, he felt sure that it failed from one or more of the factors, Nos. 2 to 10, rather than from a continuous stress of ± 5,000lb. per sq. in.

As he had said elsewhere, 99·95 per cent (the figure was precise) of the propellers for which he had been responsible had been free from breakage and such high standards as the propeller industry had attained were not the result of good fortune but rather of a sound, logical and reasonably conservative policy which had well repaid the British shipowner.

He thought they should all appreciate the hard facts that Mr. Bunyan had given in this excellent paper and support him all they could in his further search for more and more facts on these problems.

MR. J. H. MILTON (Member) wrote that, to a marine engineer, the idea of a ship's hull vibrating throughout its length with nodes and antinodes, either vertically or transversely, was rather difficult to visualize, and his colleague, Mr. Bunyan, had explained these phenomena very clearly.

He thought, however, that it should be pointed out that for any particular mode of vibration, owing to the variation in stiffness and mass along the length of a ship's hull, the nodes were not equally spaced, and that the amplitude of vibration varied at each antinode—perhaps Mr. Bunyan would enlarge on this theme?

With regard to the methods for reducing the hazard of fatigue failure of tailshafts, it was noted that a positive drive up the cone of 0·003 inch $\times$ D (where D = diameter of large end of cone in inches) was recommended. This amount of drive up might be theoretically sound but in practice, as was often the case, variables crept in—firstly, at what point during the fitting of the propeller was the drive up measured from? This could vary according to whether the shaft was dropped vertically into the propeller in the fitting shop or whether the propeller was being fitted in dry dock with shaft in position. Secondly, fits always varied and whereas 3/1,000 inch per inch diameter might be correct for a propeller which was a perfect fit, how often was this perfection obtained? Thirdly, there was the variation in scantlings and material of the propeller boss. In his experience, having obtained an even marking, slightly heavier at the large end, it was usual to find that the wedges were "ringing solid" when, in the case of manganese bronze propellers for a shaft of, say, 15-17 inches diameter the travel up the cone from tight to solid was $\frac{1}{16}$ inch to $\frac{3}{32}$ inch. This was appreciably more than 3/1,000 inch per inch of diameter and would vary with the quality of the fit and the scantlings of the boss.

He had heard of one case where at each drydocking the manganese bronze propeller of a vessel was tried and hardened up—the extent of the movement of the nut on each occasion was so much that the owners' superintendent, becoming anxious, had the propeller removed for examination—nothing was found amiss; the boss, somewhat light, was stretching and creeping up the cone.

In view of the foregoing it appeared to him that to quote any definite drive of a propeller up a cone was pushing a theoretical ideal, and once a good marking had been obtained from the propeller on to the cone *with the key in position*, the extent of the hardening up procedure, even when gauged, was based on experience.

MR. E. PANAGOPULOS considered that Mr. Bunyan's paper contained information which would be found very valuable as a reference in connexion with the modern, high-powered, single-screw ships. The problem was receiving first priority in the United States, where the tendency to higher powers on single-screw had been more pronounced in the last few years.

The problem of hull and machinery vibrations had not by any means been solved as yet; neither did it appear to be near a final solution. A most striking example was that of a series of about ten similar tankers built from the same plans and by the same yards and identical in every respect, both with regard to hull and machinery design and construction. A few of these ships showed severe local hull vibration, affecting the machinery, whereas others did not. No ballast arrangement could bring these ships to follow a steady pattern, so it was found necessary to use five- or six-bladed propellers, as best suited each particular ship. No satisfactory explanation had been offered as yet for this puzzling phenomenon.

Referring to Fig. 5, showing the recommended propeller aperture, perhaps it should be stressed that each particular design case must be examined on its own merits, as any hard and fast rule might lead to trouble. It should be noted that the latest American cargo ship design—that of the *Mariner* class—had a stern frame open under the propeller and a suspended rudder with no bottom pintle. It seemed to be the opinion of some American shipbuilders that this stern design should be adopted for high power installations because it allowed for larger propeller clearances and resulted in better flow conditions before and aft of the propeller disc.

Referring to Fig. 9, could Mr. Bunyan explain how these propeller forces were measured and what force the curves represented, that is, vertical or horizontal.

Referring to tailshaft stresses, Mr. Bunyan stated that the stresses due to hydrodynamic forces were deduced from the bending stresses recorded on the propeller blades. Did this mean that no direct measurement of shaft stress was made, as was done in the case of the *Chryssi* and the *Obispo*[11, 16]? Table III gave comparable values of stresses obtained from reference 16, but Mr. Bunyan stated that these stresses were corrected for tailshaft diameter. What was meant by "corrected for diameter"? The s.s. *Mission St. Louis Obispo* was a T-2 tanker of 10,000 h.p. and not 6,000 h.p. as was the case with most of the commercial T-2 tankers; therefore the tailshaft diameter was bigger, but its relative size to the rule requirements would be about the same as for the 6,000 h.p. T-2 tanker. He mentioned this because there might be a confusion for shaft diameter correction as was done on the *Chryssi*[11], where the tailshaft diameter was increased by 4 inches above the rule requirements.

An extensive programme of tests was at present under way in the United States. Several shaft-propeller assemblies, incorporating various design features, were being tested and it was expected that the findings would be made public on completion of the programme. Also, further measurements of tailshaft stresses similar to those of the *Chryssi* and the *Obispo* had just been completed on a Victory ship under service conditions, and the stresses measured seemed to be in line with what was predicted as service stresses in reference 11 (bending stresses of about ± 10,000lb. per sq. in. or over).

Mr. Bunyan made six recommendations for minimizing the danger of fatigue failure of the tailshaft. The writer would agree to these recommendations but would like to add that all stress measurements under service conditions had indicated higher bending stresses than had hitherto been expected or suspected. A logical conclusion would follow that the tailshaft diameter should be increased, together with any other precautions that might be taken. This would be particularly recommended as long as the wake effect was quite unknown (especially under conditions of varied draught, trim and weather). Increasing the tailshaft diameter would at this stage be the most direct means of reducing the stresses. It could easily be understood that any change in the classification rules in that direction would cause many complications, especially regarding the seaworthiness of existing ships having the rule-size tailshaft. However, it might be advisable if the designer, of his own accord, "beefed up" the tailshaft; this had been the practice in some cases with American shipbuilders who had increased the tailshaft strength in bending by about 50 per cent to 100 per cent by increasing the diameter. It had been remarked that this would be a rather crude answer to the problem, taking the bull by the horns, so to speak, but sometimes this might be the only way of getting the bull back in the pen.

Mr. Bunyan should be congratulated for this and other contributions to the practical facing of marine problems and for his ingenuity in devising simple but very effective improvised "gadgets" such as that indicated by Fig. 3. It proved that much useful data could be collected by using a few pieces of wood here and a string there, without need of elaborate instrumentation, which was often a prohibitive factor in vibration studies.

MR. W. SOZONOFF wished Mr. Bunyan to indicate whether the five-bladed propeller which appeared to be fitted in some single-screw drives was better or worse than a four-bladed propeller for exciting longitudinal vibration in shafting.

MR. K. V. TAYLOR, B.Sc. (Associate Member) presumed that the alteration in the various critical frequencies and amplitudes in shallow and restricted waters referred to in paragraph 9 of the section on hull vibration, was due to the changed hydrodynamic conditions around the hull which, in turn, affected the quantity of water which vibrated with the ship, that is, the "virtual" mass. Whilst it was appreciated that the remarks in the paper were applicable to large vessels occasionally operating in restricted water, he would be interested to know whether difficulty was experienced in ships which frequented rivers and canals, due to the inability to predict critical frequencies.

The remark regarding the cropping of propeller blades was interesting and one which he had not met with before, and he had wondered whether this fact could be verified from general theoretical considerations. This he achieved by using Froude's basis thrust equation; by keeping the speed constant he found that the condition for the same thrust was $\frac{dN}{N} = \frac{-5s}{1+s} \cdot \frac{dD}{D}$ where N was the speed of revolutions, s the slip, and D the diameter of screw. If s was taken as 25 per cent then the equation reduced to $\frac{dN}{N} = \frac{-dD}{D}$.

The aperture clearance necessary for a single-screw had been dealt with in detail but no reference had been made to hull propeller clearance in the case of twin-screwed ships. It was possible that little trouble was caused through the blades coming too close to the hull but Troost gave $0 \cdot 12L^{\frac{1}{2}} - 0 \cdot 4$ (ft.) for the minimum distance (L being the waterline length).

Quadruple-screwed ships were not specifically mentioned in the paper, but trouble could arise if the screw race from the forward propellers interfered with the after screws.

MR. F. A. VAN DYCKE had checked the author's formula for the 2-node vertical mode of vibration on a 450-foot tanker recently completed at his company's yard against measured results obtained during the trials. The 2-node calculated frequency (using the formula on page 104 of the paper) was 76 vibrations per minute against a recorded value of 78, and the 4-node vertical frequency deduced as indicated in the paper was 243 as against a recorded value of 236—a most commendable and practical method, using so little effort, particularly since the extensive Prohaska calculation took days to complete for no higher accuracy. Todd's formula was found to be just as good in the example quoted, giving a value of 77 cycles per minute. For avoiding strong propeller excitation the writer would follow the guidance given in Fig. 5. It was not always practicable to "tune" the ballasting or cargo distribution. He entirely endorsed Mr. Bunyan's remarks as to the avoidance of large flat areas in the shell below the loaded water line as a guarantee against local vibration and panting. He would strongly emphasize a careful attention to the stream lines in the after body of a single screw ship finishing in a sharp edge to the stern frame, which, together with the aperture clearances suggested in the paper, would go far towards reducing the wake variation which had to be somehow dealt with by the propeller.

On the matter of the five-bladed propeller as against the four, it would be better to accept the one or two per cent drop in efficiency of the five-bladed and have a quiet running ship.

In conclusion, it would be good, for the sake of completeness, if Mr. Bunyan would include details of the low frequency hull vibration exciter developed by the Research Department of Lloyd's Register; also of the shaft alignment technique which was referred to in the paper.

MR. C. VAN VLIJMEN'S first question referred to the problem of torsional vibration and in particular to the application of a flexible coupling in a propeller shaft line. In ships with the engine room aft, the diameter of the shafting would in general be such that also the one-node critical speed would be well above the working range of the engine. However, when for one reason or another this was difficult to realize, one might consider the application of a flexible shaft coupling between main engine and thrust block, by which the one-node critical could be put down to well below the working range of the engine. However, the two-node critical speed would not be altered very much by the fitting of a flexible coupling. Now it would be of interest to know whether there was any general rule, based upon experience, regarding the effect of a flexible coupling on the two-node stresses and critical speed. In this respect he might refer to the fact, mentioned by Mr. Bunyan, that the frequency of natural torsional vibration could be determined by calculation within narrow limits, but that the limits of accuracy for calculation of stresses had considerably greater margin. In the case of a two-node critical speed, this margin would be the more important, as the amplitude and in consequence the damping influence of the propeller would be relatively small.

His second question referred to the subject, which had already been raised in the discussion by Mr. Koning, of the clearance of the propeller in the aperture. He thought, as already posited by the speaker, that this clearance, especially the distance between propeller and rudder or rudder post, should not exceed certain limits, in order to avoid unfavourable influence on the propeller efficiency. Therefore it was very interesting to have just heard from Mr. Koning, who was an expert in this matter, a somewhat different opinion. However, they had learned from the paper read by Mr. Bunyan that systematic trials with various clearances had been carried out in order to establish the influence on the subject of vibration. In relation to the above it would be interesting to know whether, during these trials, attention had also been paid to the subject of the influence on the efficiency of propulsion and, if so, what had been the results of this experience.

MR. CH. WOUTERS wished to be informed whether there was any difference in the vibration characteristics of identical tankers when turbine driven and when motor driven. He also wished Mr. Bunyan to comment on why it appeared from service results that there was more trouble with tailshafts in the case of turbine driven tankers than with similar motor driven vessels.

MR. G. YELLOWLEY (Member) wrote that an analysis of a problem and the manner in which it was overcome was always of great interest to the practising engineer and they were indebted to the author for presenting the subject in such a practical way, which would enhance its usefulness to those who were called upon to resolve similar difficulties in their day-to-day work.

With regard to the problem of avoiding hull vibration it would be generally agreed, as the author stated, that the positive and practical approach was to limit, if possible, all exciting forces to values which would ensure that the hull vibration accelerations were acceptable, and it was noted that the suggested limits for vertical and horizontal vibration, i.e. 1·0 and 0·75ft. per sec.^{2} respectively, based on experience of ship vibration was somewhat higher than the figure of 0·3ft. per sec.^{2} suggested by Dr. J. E. Richards* based on a survey of existing knowledge on the human reaction to vibration. It was appreciated that it was not a straightforward matter to suggest limits for forces and couples due to the many factors involved but some guidance from the author's experience in which known out-of-balance forces had given rise to vibration trouble would be helpful. In the writer's experience, elimination of the free vertical force was most important and in cases of near coincidence with the hull frequency this involved a very careful assessment of the weights of the reciprocating parts.

The author's remarks on bolted connexions were very timely and the firm with which the writer was associated had given a great deal of thought to this subject and had instituted a testing rig, using strain gauges which enabled the correct tightening load for various sizes of bolts to be demonstrated to the personnel involved in such work.

The section of the paper dealing with crankshaft stresses and alignment was an important contribution to the subject and the remarks regarding the interpretation of centre crankweb deflexion readings in the Doxford engine corroborated a similar conclusion by the writer in a paper† on "Marine Shafting Alignment". In the latter paper a description was given of a simple method of alignment by "sighting occultation" developed by the writer's firm, which had the advantage of being readily applied to certain types of oil engine without removing main bearing keeps or any running gear and was in regular use for erection of engines in shop and subsequent installation on board ship and also for checking of alignment of engines in service; the method had been used in some fifty installations with complete immunity from bearing troubles.

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* Richards, J. E. 1951-52. "An Analysis of Ship Vibration Using Basic Functions". Trans.N.E.C.Inst. E. and S., Vol. 68, p. 51.

† Yellowley, G., and Richards, J. E. 1953-54. "Marine Shafting Alignment". Trans.N.E.C.Inst. E. and S., Vol. 70, p. 343.

Author's Reply

The author thanked Dr. Ker Wilson for his most encouraging remarks which, coming from one so eminent as an engineer and prominent in the field of vibration, were greatly valued. Dr. Ker Wilson was right in pointing out the possibility of a torsional vibration critical in a geared installation exciting a hull critical which, however, had not yet been met with by the author. In geared turbine installations, where the principal torsional vibration modes were propeller-excited, it would be difficult to distinguish the contribution to the exciting forces from the propeller, applied at an antinodal position for all fundamental modes, which might be large—depending on aperture clearances, from that resulting from the torsional vibration critical.

Dr. Ker Wilson's remarks on vibration set up by airscrew blades were most interesting and explained why the propellers were mounted so far forward from the leading edge in one of the most successful turbo propeller commercial aircraft, which was, in fact, particularly free from airscrew blade vibration.

On engine balancing, Dr. Ker Wilson would like to draw the author into stating limits for acceptable forces and couples, which was a difficult proposition as a number of factors were involved, the principal one being the position in the running speed of the engine-excited critical. If this critical were well removed from the service speed in a 400-ft. ship, then the forces and couples might be quite large, say, ± 6 tons primary force and 250 tons ft. primary couple, which would not be troublesome as a transient condition. On the other hand, a critical engine-excited on the service speed would require careful consideration, which involved a knowledge of the position of the nearest node of the mode of hull vibration involved. An interesting example of this type of problem was dealt with a few years ago, which involved a tanker 480 feet long, powered by a triple expansion engine. The 2-node vertical mode was excited sufficiently near the service r.p.m. to be unpleasant in the deep condition. This meant that a primary force of less than one ton and a primary couple of about 100 tons ft. in the engine itself was sufficient to produce hull vibration of appreciable proportions. A compromise was obtained by taking moments of the engine masses about the after node to give, with suitable crank web balance weights, the minimum vibration which was acceptable for both loading conditions. "Tuning" the ballast distribution was not a proposition for this mode. In regard to axial vibration under normal conditions, the author would amplify his remarks to say that an abnormal condition occurred when the axial torsional excited critical coincided with a propeller-excited axial resonance, which in one case dealt with produced extremely violent hammering of the thrust block. The cure in this case was to rotate the propeller relative to the crankshaft one bolt hole of the intermediate shaft coupling, thus throwing the two exciting forces out of phase. Regarding the transverse rocking modes of vibration, the majority of cases were, in fact, tanker installations, though the records did show a few midship installations which were similarly dealt with entirely successfully. Dr. Ker Wilson's experience with twin-screw machinery was interesting. There were two principal and distinct modes which were excited, the one being the "tuning-fork" mode where the engines rock antiphase and a "swaying" mode where they rock in phase. Tying the engines together would completely cure the "tuning-fork" mode, but would not affect the "swaying" mode, which required tying one or both engines to the ship's side. In one hull vibration test, using the hull vibration exciter, a violent "tuning-fork" mode was excited during a run up of the exciter for a plot of the vertical hull vibration criticals, the amplitudes of vibration of the engines being so severe that it was dangerous to run through this critical.

It was valuable to have Mr. Ellison's comments about an experience with strong propeller-excited hull vibration and sternpost erosion, which was almost entirely eliminated in further similar vessels having aperture clearances increased to the dimensions indicated in Fig. 5.

The author endorsed Mr. Ellison's remarks as to slab type resilient mountings, which tended to flow and needed constant following up. In the case of turbines mounted on semi-resilient pads, considerable malalignment could occur in time due to settling of the turbine feet into the pads, leaving the gear case relatively high as this was secured on solid metal chocks.

Mr. Ellison's remarks that unequal wear down (malalignment) and conical error caused the bottom end to run to one side only did not include the facts that conical errors were not necessarily of the same proportions on top dead centre as bottom dead centre, and appreciable malalignment also produced an axial bodily movement of the centre crankpin which the bottom end followed. Helical errors would also bring about this fore and aft movement of the bottom end. It was quite true that it often happened that, when a top end was remetalled, one bearing might rapidly warm up and hammer out slightly in finding its true running position—this caused little concern as, after the job had run in and the bearing put back to its normal running clearance, the unit would operate entirely satisfactorily for years. The author could not see how a tight spherical could set up weaving of the bottom end, but it had happened on more than one occasion that a seized bottom end spherical had caused hammering out of the top end white metal because of the torsion it transmitted to the crosshead pins resulting from crankshaft errors, wear pattern in the crankpin, and malalignment. The "lozenging" of the crosshead slipper was first carried out just over two years ago and the ship in question had been in constant service with no adverse complaints whatever. There had been several others dealt with in the same way, and here again no complaints had been received to date.

In reply to Dr. Richards, the author had to admit—as he had already implied in the paper—that there were some complex hull vibration problems which had to be faced from time to time, but by far the greatest majority of them lent themselves to a straightforward practical solution. The problem of local panting vibration of the tank top was certainly of interest, but it would be difficult to convince the shipbuilder that he had to stiffen up considerably the double bottom and provide heavier connexions to the ship's structure merely to reduce these amplitudes, when the cause of the vibration could be so simply dealt with. In the tanker referred to by Dr. Richards, in which damage had occurred to the connexions of the deep frames to the tank top, the force about the after node causing the vibration amounted to approximately 1·5 tons. This had been reduced to zero by the Research Department of Lloyd's Register, by re-balancing the engine for this mode of hull vibration. The hull vibration amplitudes were reduced from ± 0·2 inch to zero.

Propeller forces were influenced to some extent by the width of the blade and more significantly by skew back of the blade and blade pitch variation. A radial leading edge gave a more peaky excitation. At this stage the propeller aperture clearances shown in Fig. 5 did not involve consideration of the effects of blade shape or number of blades, but it was possible that these factors might be introduced at some future date when the experiments at present being conducted on this problem were completed.

Mr. Hildrew raised a point which should have been covered in the paper. The limits suggested for acceptable vibrations in terms of linear acceleration were meant to cover the frequency range spanned by the fundamental modes of vibration, as met with in service, i.e. up to about 600 c.p.m.; for higher frequency, and sonic frequencies, the basis of acceleration was not intended.

The author would endorse Mr. Hildrew's comments regarding secondary unbalance in main engines, which was a problem that could be most difficult to deal with in a practical and reliable manner once the engine had been built. Care should be taken in the design stage to reduce secondary forces and couples to a minimum.

Mr. Hildrew's remarks on the "tuning" of hull vibration by ballasting, indicating the possibility that a change in the damping characteristics was involved, was a factor that might be brought to light when this matter was fully investigated, a work which was now planned and ready to go into operation.

Dr. Johnson, like his colleague, Dr. Richards, was somewhat concerned with the over-simplification of the subject as treated in the paper. The author might perhaps be forgiven for leaning towards such a down-to-earth approach, which was a direct result of having had to find a practical solution, as quickly as possible, to a large number of diverse ship vibration problems every year. The successful results obtained in every instance gave the author confidence to proceed along these lines in the future. Practical experience would immediately decide, in the case of a hull vibration critical, whether it was caused by first order propeller forces or a first order engine excitation. The balancing calculations of the engine primary forces and couples would immediately indicate, from experience, what was required, and a check of the propeller while still afloat, but with the ship ballasted forward to expose at least half the blade, would establish by observation and measurement of the blade to rudder clearance whether the propeller was damaged or badly pitched. Usually, it would not be necessary to go as far as this, as the primary forces and couples would provide the clue. The same approach would be used for secondary unbalance. On the score of hull torsional vibration one would not go far wrong, in fact, in confusing it with a transverse mode—the remedy would be the same in the case of propeller-excited vibration, and would also be the same in the case of engine-excited vibration. Confusion with the transverse mode, however, should not occur as the two were clearly recognizable from the plot of the hull frequency spectrum, which was the first stage of any hull vibration investigation carried out by the Engineering Research Department of Lloyd's Register of Shipping.

Dr. Johnson's faintly amusing remarks as to the necessity of calibrating the individual when observations are made on board, missed the point entirely, insofar as any null indicator is not a piece of equipment that needs calibrating; all that is necessary is sensitivity which, however, he admitted as being one of the qualities of a human being. It was suggested in the paper that the observer would indicate positions on the deck where no vibration was felt, and was not required to express any opinion as to the amplitude of vibration, only as to its direction, i.e. vertical or transverse. Surprising as it might seem, this crude and improbable method worked sufficiently well to enable an important decision to be made while the ship in question was still at sea.

In reply to Dr. Johnson's demands for further detailed information as to the results of a research still in progress on aperture clearances, the author was not prepared to divulge this information at this stage. Regarding the necessity for keeping loading conditions identical when comparing vibration performance before and after modifications to the vibration generator, this was an important factor that was well appreciated by the author. As was indicated at the meeting, the chance to obtain records after modification to the propeller had to be accepted at short notice, and a short run only was available as the ship had to proceed on her normal voyage commitments. The results were included, however, as it was felt that it could be appreciated that, were the identical conditions of ballast obtainable, the reduction in vibration amplitude would have been even more marked. Dr. Johnson's comments on Mr. Hildrew's remarks should be corrected, as the comparison of speeds before and after cropping were with clean bottoms in both cases, i.e. results obtained on trials and after drydocking. Were it not the case, the author was confident that Mr. Hildrew would not have quoted the example.

The reason for the formula being included in the paper at all was to avoid the necessity for a knowledge of moment of inertia, which was not always available at the time when a consideration of the propelling machinery was made. Were sufficient information available at the time, it might be preferable to use a more elaborate method.

As with most of the approximate methods, and indeed with any known method, the addition of superstructure decks reduced the accuracy of the result considerably; the same was the case with the formula in the paper, which was as accurate as any of the approximate methods. It was a simple matter for Dr. Johnson to verify the results for himself.

Propeller tip to sternframe clearances varied considerably and influenced the transverse forces applied to the stern of a ship. The trailing edge clearance was also an important factor in producing transverse vibration.

The modes of vibration considered in Fig. 9 were transverse modes and the bases of the deduction were actual forces applied to an almost identical hull by means of the Society's low frequency hull vibration exciter.

In reply to Mr. Bryan Taylor, the author would state that the overhung weight of the propeller was not the parameter for determining the scantlings of tailshafts in Lloyd's Register's Rules. He agreed with Mr. Taylor that, as tailshaft failures were not the epidemic which measured results appeared to suggest they should be, there were factors difficult to allow for at the present stage of knowledge, which reduced the hazard in service. On the other hand there were factors which made the problem more hazardous—notable amongst these was the ingress of salt water, which reduced the life of a tailshaft very considerably. Any Classification Society's Rules were based on many years of wide experience, and the author would reassure Mr. Taylor that when service performance directed, the requirements of the Rules were modified accordingly.

The author was unaware of the size of the propeller to which Mr. Taylor referred, in which a drive up of 0·002 inch per inch only was possible beyond the shop fitting mark. It was possible, of course, that the shop fitting mark represented a considerable drive up to start with, in which case the difficulty was understandable. In practice it was not difficult to drive up 0·003 inch per inch diameter from the point where the propeller was just nipped up on the cone with the nut.

The author agreed with Mr. Taylor's comments calling for an alternative method of securing a propeller, preferably one in which no key was fitted, and sufficient friction grip was established to take any load occurring in service.

The accuracy of the taut wire gear described in the paper was such that independent observers could obtain readings to

± 0·001 inch. The lamp would light up and go out with a variation of 0·0005 inch. The transverse alignment would give the same accuracy if the staff could be set true in the vertical plane exactly. In the shop this was possible to ± 0·002 inch, making the total error ± 0·0003 inch. On shipboard the sensitive spirit level could only be used in quiet water such as a fitting out basin, preferably at night when the level on the base of the instrument must be set up when the undisturbed master level reads zero. It was near enough for practical purposes with the ship afloat at a berth to use a bar laid across the top of the base and having two clock gauges registering on the machined horizontal surface of the bearing pocket. Assuming the bearing pockets to be in the same plane—as they were in fact so machined and as set up originally in the ship—then it was possible to set the staff true to ± 0·003 inch. Where damage had occurred to the welding securing the bearing pockets to the bearing girders, the truth of the pockets should be first tried with a straight edge and feelers across the girder.

Commander Green raised the question of the effects of torsional vibration and hull vibration, which had also been raised by Dr. Ker Wilson and had been replied to above. The submarine vibration problem might well have been associated with torsional vibration—it was a special case and the author would not care to comment further. Details of the torsionmeter referred to by Commander Green had been examined and it was considered that, for alternating torque, the instrument might be suitable, but might be expected to be somewhat inaccurate for the measurement of mean transmission torques, which required no zero drift over many hours of operation before a zero check could be obtained, which was the more important application of torsionmeters.

The author regretted that Commander Green found the load/load diagram (Fig. 10) complicated. It was produced for the express purpose of indicating what happened when a bolt was incorrectly tightened—a load extension diagram tended to get out of hand at this stage.

The author was most interested in Commander Green's description of an experience with some large bolts securing the bearing caps of run-out tables in a rolling mill—it appeared that the shock loads were too large for the bolts to carry without separation at the bolting faces. It would be interesting to read the report on this investigation when it appeared in the B.I.S.R.A. Journal.

Mr. Griffith described an incident on one of his company's river tugs in which severe transverse rocking mode of vibration occurred near the running speed. The author was brought into the problem in the final stages and it was obvious that sufficient strengthening could not be provided where it was required, i.e. below the crankcase, without very considerable expense. The transverse staying of the engine was carried out on the author's recommendation and produced sufficient frequency raising to keep the running speed practically free from vibration. If the engine could have been raised 4½ inches and heavy tranverse girders fitted below the crankcase, connecting with the frames on each side, the same purpose would have been served. It would have meant an increased rake in the shafting. If these factors were not impracticable from service and maintenance considerations, then they could be incorporated in new buildings. The geared drive would certainly raise the speed at which a 550 b.h.p. engine could be operated, which from a hull vibration aspect might be more satisfactory; the reversing gearbox would also save considerable wear and tear on the engine.

It would be impossible to generalize on this problem, which was largely one of adequate stiffness under the engine, which would resolve itself as to whether the higher mounting, which appeared to be a significant factor, were acceptable or not.

Mr. Fraser-Smith was correct in saying that "tuning" by ballasting was only applicable to propeller-excited criticals; although the examples quoted in the paper were for propeller-excited criticals, this point might have been emphasized in the opening remarks to the section on "tuned" ballasting. He thanked him for pointing this out.

The limitations on the extent of "tuning" possible in the deep condition were dependent on the gravity of fuel carried in the cargo tanks—Mr. Fraser-Smith and Mr. Goemaere both raised this point, with which the author agreed.

The author thanked Dr. Allan for pointing out that the trailing edge clearance to rudder being equal to the thickness of the rudder need only apply where the leading edge was blunt and rounded as shown in Fig. 5, and need only be $0{\cdot}08D$ where the leading edge was fine. This fact, which was included in the Society's circular to the ports last year, had been overlooked when condensing the original manuscript of the paper. The author would hasten to reassure Dr. Allan that modifications to propellers were never made without careful consideration, not only of efficiency but also of the cavitation characteristics.

Dr. Allan took exception to the formula for the frequency of 2-node vertical vibration on the grounds that it was not dimensionally correct. However, it must be borne in mind that the constant K virtually had dimensions in the same way as in the Todd-Marwood and Schlick formulæ, i.e. it took account of Young's Modulus and the acceleration due to gravity. In this case, further units were implied in the constant by virtue of the substitution of a formula for I which included the length to a complex index, and this also accounted for the inverse function of length outside the root sign.

It might be pointed out, in connexion with the Todd-Marwood formula, that frequency varied as the square root of sectional inertia, but sectional inertia certainly did not vary as the square root of $B\,D^3$. The formula given on p. 104, however, as already mentioned, did include an approximate formula for I.

The author was surprised to hear that the Todd-Marwood formula gave better agreement with the measured frequencies in the cases investigated by Dr. Allan. It was possible that the ships in question were old ships (the inertias of ships built since 1950 were greater than those built before 1950 and the formula given on p. 104 was designed for new ships), or the ships might have had several tiers of effective superstructures, a feature which reduced the accuracy of any approximate formula. It was also a simple matter to quote cases where a comparison against measured frequencies of the results given by the two formulæ favoured the one given in the paper.

The author should have been more specific in stating what he had in mind when referring to a thick streamlined rudder, as he did not consider that rudders appreciably exceeding a thickness ratio of 20 per cent were fitted; usually they were nearer 15 per cent. The author thanked Dr. Allan for his interesting remarks on his proposals for achieving a more uniform flow to the propeller of a single-screw ship, and it was hoped that he would be given the encouragement to go ahead on a problem that was becoming, if anything, more acute because of the increasing size and powers of single-screw ships.

The author thanked Professor Bonebakker for his contribution, which raised a number of interesting points in regard to hull vibration.

I. Dealing in order as to their prevalence and severity, with the predominating types of vibration met with in service, the following information was included:—

(1) *Types of vibration*
- (*a*) Propeller-excited fundamental modes involving the length of the hull—vertical and transverse.
- (*b*) Engine-excited fundamental modes involving the length of the hull—vertical and transverse.
- (*c*) Local vibrations in way of the stern of the vessel.
- (*d*) Local vibrations in bridge structure affecting navigation instruments, radar and wireless telegraph.

(2) *Types of vessel*
- (*a*) Tanker—particularly over 500 feet in length.

(*b*) Vessels with machinery aft.
(*c*) Vessels with machinery mid-ships.
(*d*) Twin-screw vessels.

(3) *Size*

The larger tankers could be a severe vibration problem, particularly where apertures were tight, though there had been many coasters which had had severe main engine-excited vibration criticals which had to be dealt with.

(4) *Loading condition*

(*a*) The ballast condition was perhaps the worst condition, but fortunately with "tuning" of the ballasting it was possible to remove such criticals from the service speed.
(*b*) The load distribution of ballast should be in accordance with the recommendations of the Classification Society, but slight variations might be necessary for tuning in way of the last three tanks—care being taken to keep longitudinal stresses in the hull within approved limits.

The necessity or usefulness of vibration calculations was a difficult matter on which to generalize. In single-screw vessels, say, 450 feet and over, and indeed one might also include twin-screw ships also, the predominating exciting force was most often the propeller, at blade frequency, which excited the higher modes of vibration. These modes did not lend themselves to accurate prediction by calculation, particularly where the hull was complicated by the addition of superstructure decks. Attention to the single-screw aperture and tip to hull clearances in the case of twin-screw ships, however, should ensure little or no trouble from this cause. However, the point did arise where the two- and three-noded modes might be excited on the service speed range by first and second order engine forces and couples in the case of reciprocating main machinery, and in the author's view these cases should be thoroughly investigated, as the accuracy of the prediction for these modes of vibration was a practical proposition.

II. The qualification "insupportable", i.e. discomforting to personnel, might be applied to sustained vibrations in accommodation spaces exceeding in terms of local accelerations 1ft. per sec.^{2}. "Dangerous"—or rather vibrations which had been known, in welded construction, to produce damage to the structure at points of high stress concentration, exceeded approximately 3ft. per sec.^{2} locally.

III. Accuracy of vibration calculations was such that the two- and three-noded modes of vertical and transverse vibration might be predicted to a usable accuracy. For higher modes the results obtained were of doubtful value where complex hull forms were considered. The paper gave two references which were helpful in this matter.

Mr. Cromarty had the author's thanks for his most valuable contribution which, emanating from his own experience, underlined so much of what had been the author's experience also.

On this matter of production of large engine components, Mr. Cromarty's remarks emphasized the essential necessity for well defined machining limits, fits and tolerances, which with mass production methods must be much finer and more rigidly controlled by an inspection department well equipped with the necessary gauges, jigs and fixtures for the job. Such equipment must be constantly checked and re-checked against standards, in a well equipped and well planned tool room. Selective assembly and hand fitting were increasingly necessary where the machine tools producing such parts were either not to grade 1 standards of accuracy or even not entirely suited to the job in hand. Mass production methods required first class machines, inspection and planning, as the limits must be tight and consistently maintained. It was a very costly conception, as might best be exemplified by the motor car industry. It was the practice in some shops abroad to build up the main engine in the ship, without any pre-erection in the shop. Given that there was to be little or no hand fitting or correction of errors required in the process, i.e. that the various components had been machined and checked to fine tolerances, and also that a close and intelligent watch was kept on the distortion of the bedplate, which was progressively corrected while the weights were built into the engine, then it was possible to produce satisfactory results and to achieve sizeable economy in time and cost. But if this were not the case, the consequences might easily prove to be disastrous, as had been amply demonstrated in fact.

Mr. Cromarty's remarks on the machining of large dynamically stressed bolts and nuts deserved serious attention. It was startling to find, in these somewhat enlightened days, that such a lack of appreciation of the fundamental and vital requirements could go undetected. What made the situation even more deplorable was that the screw cutting machines referred to were apparently new machines.

Mr. Cromarty's description of the serious lubrication troubles experienced on a pair of main Diesel engines was valuable in drawing attention to the manner in which serious oil leakages could occur through indifferent fitting of bearings.

On the subject of engine damping, from the new to the "run in" condition of a heavy oil engine, an increase in the crankshaft vibratory stresses of about 20 per cent might be expected in, say, a 6-cylinder engine running at 450 r.p.m. In the case of large slow-speed engines, a much smaller difference might be expected. It would, of course, be appreciated that there was no hard and fast rule about this matter, as design details such as trunk piston or crosshead type, two- or four-stroke, rating, loading and type of bearings, crankshaft materials and type of construction, etc., were all factors which affected the damping in varying degrees, not to mention the external factors such as generators, shafting and propellers, gearing, etc.

Mr. de Winne raised the matter of propeller tip to hull clearances which were dealt with in the reply to Mr. K. V. Taylor on p. 130.

Mr. Evrard raised two interesting matters. The first question raised the point about the influence of the riveted type of construction on the vibration at the after end of the ship. For the same exciting force the vibration amplitudes might be expected to be about half those which would occur in the case of an all-welded ship. In a recent investigation, damaged welded attachments of the floors to shell plating in an after peak tank due to severe local vibration had been replaced by riveting angle bars to the shell and to the floors. This was found to introduce additional damping and reduced the vibration amplitudes at the after end about 35 per cent.

Mr. Evrard's second question, about the effect of the type of cargo carried on hull vibration amplitudes, could also be replied to by quoting an actual experience in which the amplitudes of vibration for the 2-noded vertical mode appeared to be very considerably affected by the damping properties of cargo. The higher modes showed a very much slighter reduction.

In reply to Mr. Fairlem, the author would say that his unfortunate experience with continuing trouble with a Doxford top end, after incorporating all the corrections and modifications suggested in the paper, was difficult to understand. This had never been the experience of the Department, who had not to date been beaten by any Doxford top end problem. The author would value a discussion with Mr. Fairlem. Experience had shown that one was less likely to succeed with a top end problem if, on reassembly, the job were run at too slow a speed. If the alternatives of a short sea trial or a dock trial were offered, the sea trial would, of course, be preferred if the ship could be got under way fairly rapidly and one was not faced with a long period of countless manœuvres at low revolutions before getting under way.

Mr. Goemaere had also raised the matter of "tuned" ballasting of loaded tankers which had been raised earlier by Mr. Fraser-Smith and which had been replied to on p. 127.

Ir. Koning had the author's full agreement in the remarks he made regarding propellers and propeller apertures of single-screw ships. As a propeller designer, Mr. Koning would be well aware of the particular design problems peculiar to the single-screw ship. The high wake peak at the upper part of the aperture was the crux of the problem, and with the conventional design of the after body of single-screw ships, its effect could, at best, be alleviated only by ensuring adequate clearances in the aperture. The use of "U"-shaped hull sections forward of the propeller disc was now common for the same reason. The considerable increase in the powers transmitted by a single-screw over the past two years made the problem of cavitation a very real one, and while there were certain precautions which might be taken by means of pitch variation, it frequently happened that the propellers of large single-screw ships were working well in or near to the cavitation zone at full power. Pitching and rolling of the ship served to accentuate the problem considerably, and unless care were taken to reduce revolutions in heavy weather, not only the propeller but the tailshaft also came in for serious punishment. The author was of the opinion that, with the very tight schedules under which tankers operate today, speed reduction for weather, with geared turbine machinery in particular, which gave no aural indication of distress, was only contemplated when the bridge considered that ship movements were reaching excessive proportions.

The author thanked Mr. Lenaghan for his constructive contribution and was most interested in his suggestion of a still further increase to 12½ per cent D for the propeller tip to arch clearance. The streamlined rudder of 3 feet thickness would only require a trailing edge clearance of this amount if it had a blunt leading edge; it should, however, be possible to fine the lines of the leading edge, in which case the trailing edge clearance could be considerably reduced.

Mr. Lenaghan's surprise at the suggestion in the paper for increased scantling of the structure in the cruiser sterns of large single-screw vessels was perhaps understandable. The author's bias in this direction was perhaps coloured by the fact that he never boarded a ship that was normal and satisfactory in every way, but rather was called in on a large number of problem-children every year. The author was always left with the impression that, particularly in larger single-screw ships, a general bedlam at full power in way of the stern was something that should receive more attention. In several instances the panting produced at propeller blade frequency was sufficient to cause local damage, and in these cases something had to be done in the way of additional stiffening after the trials, often a very costly and time-consuming process. He agreed with Mr. Lenaghan that much could be done by design and, indeed, in the case of the vessels referred to above, sister vessels would no doubt incorporate the additional stiffening which had to be fitted, in some cases a bit more also for luck. The most severe cases within the orbit of the author's own experience had been those with large flat areas in the ship's side stern plating. Mr. Lenaghan quoted naval practice and mentioned much larger powers and faster turning screws which implied twin- or multiple-screw ships which, of course, might not be the same problem.

Mr. McAlister's authoritative contribution enhanced the value of the paper and deserved the author's thanks. The cropping of propeller blades—also referred to by other contributors—was undertaken after careful consideration of the facts and wherever possible in co-operation with the propeller manufacturer. In some cases it was a temporary expedient to improve operating conditions until such time as a more suitable new propeller was obtained. There were a few instances, however, where cropped propellers were in service and giving satisfactory performance, both from propulsive considerations and also from a vibrational aspect, the latter being the cause for the modification in every case.

The author agreed with Mr. McAlister's remarks regarding the leading edge aperture clearance; it might well be, as a result of further work now being undertaken, that it would be desirable further to increase the clearance now recommended as a minimum at $0 \cdot 15D$. His remarks about the tip clearance at $0 \cdot 10D$ were interesting in that they conflicted with Mr. Lenaghan's statement that he would like tip clearances in excess of $0 \cdot 125D$. One could not but be impressed by the fine record of the insignificant number of breakages which had occurred in service with the propellers for which Mr. McAlister had been responsible.

In reply to Mr. Milton's first point, the author said he would agree that the positions of the various nodes along the hull of a vessel, vibrating in one of the principal modes of vertical or transverse vibration were, in fact, determined by the mass and stiffness distribution of the hull structure, which also affected the vibration amplitudes at the various antinodes. It was only with a uniform structure of constant sectional inertia and having a uniform load distribution that the nodes and antinodes tended to be equally spaced.

Regarding Mr. Milton's comments in connexion with the fitting of propellers, while it was agreed that practical experience was probably the best criterion for deciding whether a propeller was home or not, it did not detract from a quantitative appreciation of the problem. A propeller fitted to a new tailshaft or a new propeller fitted to an old tailshaft must be tried with blue marking preferably, but always thinly applied, to ensure a sound fit, which it was good practice to make slightly harder at the driving end. The amount of drive up had always been a debatable issue, which could be judged by the varying practice up and down the country.

Mr. Panagopulos had the author's thanks for his valued contribution, which raised many interesting points. It was hard to accept, however, that such a wide variation in vibration characteristics as he suggested could exist among ten identical tankers. There must, of course, be some definite and significant variation between the ships that vibrated and those that did not. In identical ships, variations were to be expected in the positioning of the propeller in the aperture and indeed in the dimensions of the aperture itself, which might explain some of the difference between the various ships. Differences, however, must be considerable to produce the very large variations in the exciting forces, which were concluded to be propeller forces. The author was not clear as to what sort of trouble Mr. Panagopulos had in mind when he referred to the aperture clearances shown in Fig. 5, clearances which had already been used on a variety of types and sizes of single-screw ships with very satisfactory results. The author had himself been considerably attracted by the *Mariner* type rudder and aperture, and agreed with Mr. Panagopulos that it had real advantages for high power single-screw ships. The service performance of this design was being followed with considerable interest. Regarding the propeller forces shown in Fig. 9, they were transverse forces derived from exciter tests and corresponding measured results of propeller-excited vibration. The vertical forces showed similar characteristics.

Relating to the T.2 tailshaft stresses, gauges were not applied to the tailshaft as it would have meant a significant alteration at the termination of the liner and to the sealing arrangements, which could not be entertained as the shaft was

TABLE III (RE-CAST)

Tailshaft stresses	Light ship			Deep ship		
Vibratory stresses due to hydrodynamic forces	±4,850	±2,650		±4,250	± 950	
Reversed bending due to propeller weight	±1,850	±1,150		±1,850	±1,150	
Total vibratory bending stress	±6,700	±3,800	(16)	±6,100	±2,100	(16)
Vibratory torsional stress	±1,500	±1,690		±1,040	± 900	
Maximum principal bending stress	±7,000	±4,440		±6,200	±2,440	

a working shaft. In the case of the "Sam" type vessel, the sealing arrangements were such as to provide sufficient length of bare parallel shaft as could accommodate the gauges.

The author thanked Mr. Panagopulos for pointing out that the *Obispo* was a 10,000 s.h.p. installation, in which case correction for shaft diameter was not justified when comparing the two sets of results. Table III had accordingly been re-cast and appeared on p. 129.

The author awaited with keen interest the results of the researches at present being conducted on a wide scale in the U.S.A. into the problem of tailshaft stresses and the single-screw aperture. This was vital information which would enable positive action to be taken in assessing the significance of the numerous factors involved, such as aperture clearances, wake effects, tailshaft stresses resulting from eccentricity of thrust and whirling, and the several other factors influencing the local stress condition at the tailshaft propeller assembly.

At this stage a general increase in the Rule requirements for tailshaft diameters, which would have to be very considerable if based on the information now available, was not contemplated as the relative importance of all factors was not yet defined and, indeed, the overall picture of service performance of tail shafting did not justify such drastic action.

Mr. Sozonoff had raised the matter of axial vibrations in shafting excited by a propeller having an odd number of blades compared with one having an even number of blades. The author would reply that it was quite probable that the propeller excitation in the axial direction from, say, a five-bladed propeller might be about 60 per cent of that from a four-bladed propeller, the frequency being in the ratio of 5 to 4.

In reply to Mr. K. V. Taylor's query in regard to the accuracy of prediction of hull frequencies for ships operating in canals and rivers, it was possible that in very shallow water, say, of the order of twice the mean draught of the vessel, the reduction in the calculated value might be as much as 15 per cent. The experiments carried out on the *Clan Alpine*(6) indicated that Professor Prohaska's approximate virtual added mass correction $= 1 + 2\,C_b\left(\frac{d^2}{hw}\right)$ (where d = mean draught, C_b = block coefficient and hw = total depth of water) gave a much larger change than that actually measured.

Regarding the hull to propeller blade tip clearances which were not included in the paper, Troost's value tended to be slightly on the large side; the preferred minimum clearance was given as follows,

$$\text{Clearance} = 0{\cdot}11\,L^{\frac{1}{2}} - 0{\cdot}36,$$

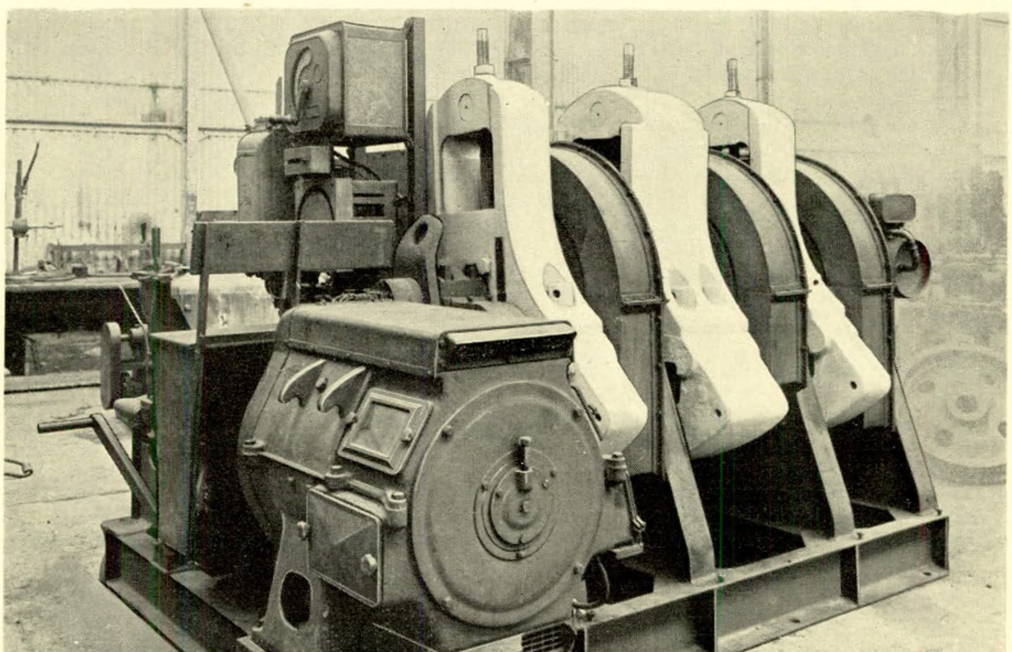
which is a later edition of the Troost formula. Quadruple-screw vessels, of which there were only two merchant vessels in service, were a special case. The problem of screw race interference was a factor which must be considered in the lay-out of the bossings. In the case of triple-screw naval vessels, very severe axial and hull vibrations were set up, due to the effect of screw race interference of the outer screws into the centre screw during turning. An investigation was made of the problem and the results had been published in a paper* by C. P. Rigby.

Mr. van Dycke had the author's thanks for his interesting remarks, which underlined several points made in the paper. The author had pleasure in including photographs (Figs. 19 and 20) of the Society's low frequency hull vibration exciter, and of the shafting alignment indicator referred to in the last section of the paper. Technical details of the exciter were included in reference (6) and of the alignment technique in reference (15), both references being found in the bibliography included at the end of the paper.

Ing. van Vlijmen raised two interesting matters. In reply to his first question, the author said that the fitting of a flexible coupling in the line shafting between the thrust block and the engine flywheel would only slightly influence the 2-node critical frequency of the shafting system, while considerably lowering the one-node frequency. In the calculation of 2-node stresses, were a generalized expression for engine damping applied to all types, this would give inaccurate values with many types, and in each case must be considered on its merits. If Mr. van Vlijmen would submit details of his proposals, the matter would be carefully considered, whether the ship was classed with Lloyd's Register or not.

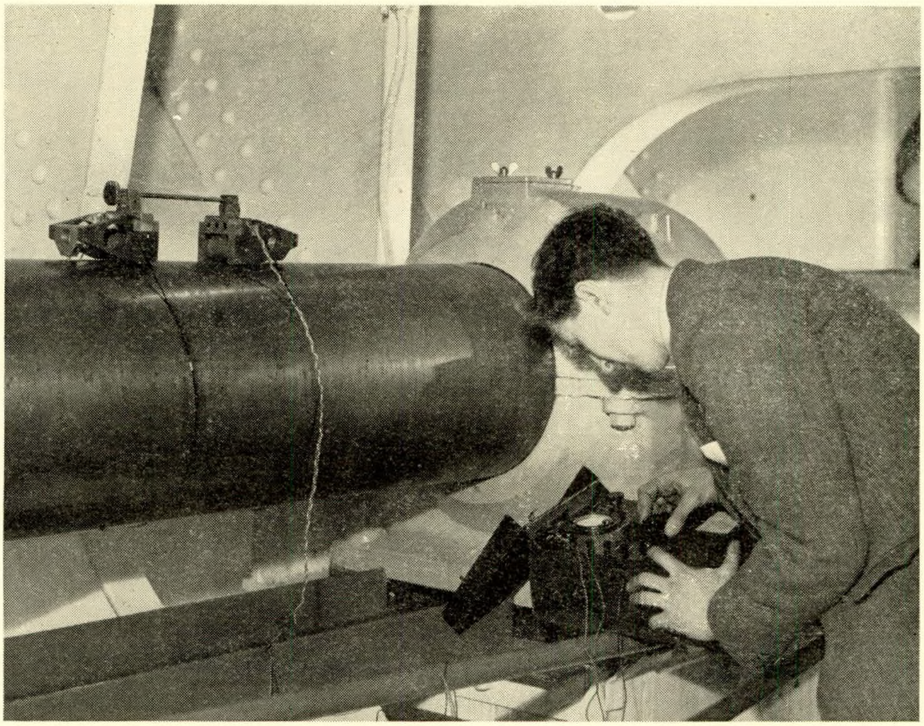
The model tests had shown that there was no significant change in the mean propeller thrust by increasing or decreasing

* Rigby, C. P. 1948. "Longitudinal Vibration of Marine Propeller Shafting". Trans.I.Mar.E., Vol. LX, p. 67.



By courtesy of the Institution of Mechanical Engineers

FIG. 19



By courtesy of the North East Coast Institution of Engineers and Shipbuilders

FIG. 20

the propeller blade to rudder clearance. On the other hand, the mean thrust was appreciably reduced as the leading edge clearance was reduced below the dimensions shown in Fig. 5.

Mr. Wouters had raised a most interesting point, suggesting that it appeared to him from service results of identical turbine and oil engine driven tankers that the turbine driven tanker suffered more in respect to tailshafts. Two factors might contribute to this, the first being that owing to the much greater rotational inertia of the geared turbine installation, which would considerably reduce the speed variation during pitching in heavy weather, there was the danger of applying vibratory stresses approaching 200 per cent of the calm water stresses[11] apart from heavy overloads of mean torque. The other factor, a psychological one arising out of that just mentioned, was that in heavy weather the greater tendency towards racing with the oil engine drive was a powerful inducement to the engine room staff to reduce the revolutions. As far as the author was aware, no stress measurements had been made on an oil engine driven tailshafting under service conditions.

Mr. G. Yellowley's valuable contribution, which had the author's thanks, raised several interesting points.

On the matter of permissible hull vibration, the values suggested for vertical and transverse vibration had been arrived at as giving a fair and reasonable practical limit which should not produce complaints, where attention had been seriously given to the choice and siting of fittings, doors, etc. On the matter of elimination of the free vertical force in reciprocating main machinery—this might be sufficient in most cases to reduce the vibration to acceptable proportions. It was a matter really involving the position of the machinery in relation to the adjacent node of the vibration—usually the 2-noded mode for first order excitation. Cases had occurred where dealing with the free force had not been sufficient and the method referred to in the paper had had to be employed. Mr. Yellowley had mentioned the matter of limits for forces and couples, which was raised also by Dr. Ker Wilson and had been dealt with earlier in the reply.

The author noted with great interest Mr. Yellowley's description of the test rig for demonstrating to his firm's trainee fitters the correct tightness in bolted connexions using various sizes of bolts. This was an excellent and practical approach to a vital matter and it would be well if such a good example were followed by other engine builders. The author was sure that Mr. Yellowley would be pleased to supply details to anyone interested. A simpler mechanical strain indicator, as shown in Fig. 11 of the paper, would avoid the need for strain gauging equipment and would serve the same purpose. The test rigs should incorporate an elasticity ratio of 4 for the bolt to the bolted assembly.