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Solution of machinery problems by field investigation

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SYNOPSIS

This paper describes some recent investigations of the Technical Investigation Department (TID) of Lloyd's Register of Shipping (LRS). Interpretation of results and opinions are those of the author and not necessarily those of LRS. In general the investigations undertaken by TID involve a combination of theoretical analysis, metallurgical examinations and field investigations which brings together specialists of different disciplines.
Extensive use is made of modern field measurement techniques to define the nature of problems and in many cases this approach has provided the essential new information to solve problems. The paper focuses mainly on this aspect of investigations, but briefly refers to wider matters as appropriate.
The introduction to the paper generalizes some of the experience gained and describes considerations which must be borne in mind during investigations. The body of the paper describes a number of case studies, mainly in marine work, which are intended to illustrate the range of problems frequently encountered on ships. Many of them have arisen due to inherent design features, the adverse effect of which was exacerbated by additional factors arising in service.

INTRODUCTION

TID operates as a self-contained group offering an impartial service which can be independent of classification but sometimes is complementary. Frequent use is made, however, of the back-up provided by metallurgists, the main frame computer and the data bank of service failures. Investigations cover most of the problems commonly experienced on ships.

Any problem referred by a shipowner or builder to an outside organization for investigation normally contains features which render it unusual. It is also likely that it has been investigated already in some depth by those directly involved before calling in outsiders.

In such situations it is essential to produce new objective information by experimentation, measurements and analysis. Field measurements provide one of the most effective ways of obtaining objective information.

It has been found that many of the techniques employed for this purpose have uses which extend beyond trouble-shooting. For example the ability to measure torque and shaft speed accurately means that power can be measured. Similarly the strain gauge techniques used to investigate shaft alignment and gearing problems are powerful tools to optimize shaft and gear alignments in newbuildings or after repairs, and the methods used for machinery health monitoring have been adapted for balancing *in situ* .

Formulation of investigations has many pitfalls as the full nature of the problem to be investigated is often not known in advance and it is a human characteristic to seek the answer to a problem from the results of actions, appropriate or otherwise, taken to investigate it. Field measurements often require great skill and dedication in difficult conditions, but all the effort may be wasted if the wrong quantities are measured or the right quantities are analysed wrongly. Thus great care must be taken when planning measurement programs to formulate them to test the major hypotheses effectively.

Where fractures or cracking of components are involved, metallurgical tests are made as a matter of course during investigations of the cause to establish the failure mechanism and to confirm that material properties are satisfactory. It is fortunately now unusual to find significant deficiencies of material outside the regions affected locally by welding or overheating from accidents. Failure modes can involve outright rupture, brittle fracture or fatigue, but the latter is the most common form in the case of machinery or shafting failures. Fatigue failures of machinery components can lead to extensive secondary failures and in many cases it takes a trained eye to discern the fatigue failure of, say, a big end bolt, among the more eye-catching consequential damage.

As a useful first check of whether a failure is due to design features or some factor related to the particular installation the statistical failure records and rule calculations are checked routinely. Even if given a valid sample, which is not always possible, statistics may require interpretation and despite the modern computer data bank, time-consuming recourse to the original source data may be necessary.

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Difficulties can arise in defining the applicability of classification approval calculations. As is well known, classification rules do not cover every aspect of machinery and ship design and they employ criteria which are based on well-established practice and are applicable to a range which may not cover unusual design features. Sometimes design criteria are therefore not as firmly established as one would wish, particularly where new design concepts are involved. In such cases recourse has to be made to a variety of analytical techniques, often involving complex computer-based calculations.

A considerable number of the problems investigated by TID involve the interfaces between different components. Alignment-induced problems, vibration of machinery on the seating, interaction with the shafting or even the entire ship are the more obvious examples. In practice an orderly sequence of office-based investigations, followed by field measurements, if deemed necessary, is rarely possible due to the usual pressures of the shipping industry and the various stages of an investigation may well run concurrently. Often the field investigation precedes the other work, making the task of formulating a realistic measurement program even more arduous.

Dynamic measurements on a failed unit are not practical until repairs have been effected which may have destroyed vital evidence, but such tests are often the only feasible avenue, particularly where no similar installations exist.

With torsional or axial shafting vibration problems this approach is generally realistic as vibration characteristics depend largely on features which are not greatly affected by replacement or repairs. Such problems are usually similar on sister ships, while those related to alignment are more typical of a particular installation and thus may be lost after repairs.

Uniquely, among shaft vibration problems, lateral vibration of shafting can be affected strongly by alignment, particularly if sensitivity to bearings becoming unloaded due to misalignment is common to a class of ships. External vibration characteristics of machinery are generally similar on sister ships in the absence of gross defects or significant differences of construction which are found surprisingly often on nominally identical ships. In general the hull vibration characteristics of sister ships are similar, but the presence of sharply tuned hull girder resonances may mean that the differences render one ship marginally acceptable while the other gives problems. In cases involving human comfort and perception, many subjective factors come into play and it becomes difficult to relate these from ship to ship.

CASE STUDIES

(1) Stern tube leakage of 61,000 ton bulk carrier led to a succession of builder's trials, resulting finally in an investigation by TID to find the cause and to rectify the problem. As leakage was up to 200 litres per day in the deep draft condition the problem was far from trivial and had been investigated already by alignment checks and calculations. Although the calculated critical frequencies of lateral and axial shaft vibration were well above the operating speed it was decided that the possibility of excessive shaft vibration at the stern tube gland should be investigated experimentally. On the basis of such tests axial vibrations were eliminated from the study as they were normal, but lateral vibration amplitudes of the tailshaft at the after seal approached the diametral bearing clearance and led to various unsuccessful attempts to stabilize the vibrations by means of realignment. It was finally concluded that the

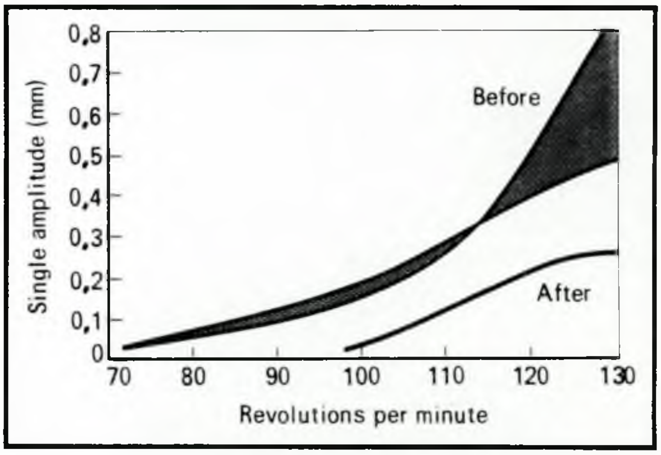

Fig. 1. Effect of bearing change on tailshaft vibration at stern seal

lateral vibration critical was closer to the service speed than calculated and that instability of the shaft within the bearing caused the leakage. An increase of effective bearing stiffness, to raise the natural frequency of the shaft system and a reduction of bearing clearance to increase damping were proposed. To effect these changes the stern tube bearing, which was of a resin-impregnated asbestos type with a fairly large diametral clearance, was replaced with a conventional white metal bearing of smaller diametral clearance, but without further alignment changes. This resulted in a considerable reduction of shaft vibration and the seal leakage rate was reduced from 200 litres per day to negligible amounts.

Central to this investigation was the ability to measure vibration of the shaft relative to the after-seal carrier. This task was accomplished with non-contacting eddy current transducers which were suitably waterproofed. The principal results of the investigation are summarized on Fig. 1.

The vessel was one of a class and a number of sister ships were also susceptible to intermittent high stern tube leakage, leading to a change of stern tube bearing on some of the vessels.

(2) Lineshaft fractures on two ships out of a fleet of four similar cargo ships within a time span of a few weeks suggested that a fatigue mechanism was involved but the age of the ships (8–9 years) ruled out a simple single fatigue stress mechanism, generated by the dynamics of the shafting system, machinery or propellers. The failures had many common features as both fractures had occurred at the tailshaft coupling flange of the intermediate shafts and the water-lubricated tailshafts were cracked at the coupling flange. The fractures had propagated from multiple origins mainly in a manner indicative of torsional loading (Fig. 2), but in detail some differences were noted. One ship had excessive stern tube wear down and at the origins the orientation of the cracks appeared to indicate torsional and bending fatigue loading. On the other ship, with normal wear down, there were no indications of bending fatigue at the crack origins. On both vessels the tailshaft couplings and adjoining shaft sections were deeply pitted by corrosion attributed to stern tube leakage. Similar corrosion was also noted on the other undamaged ships.

Torsional vibration characteristics were measured on one of the unaffected vessels, but there is no reason to suppose that they would have differed significantly on the other ships. Further tests covered dynamic bending of the tailshaft and intermediate shaft and lateral vibrations. On the basis of these measurements lateral shafting vibration was eliminated from the investigation, but it was found that the vibration torques were high when passing through the 1-node mode critical and

Fig. 2. Multi-origin intermediate shaft fracture

also near service speed under conditions of simulated poor combustion balance. It was concluded that the failures were due to the combination of high cyclic torsional stress and the deleterious effect on fatigue strength of the corrosion in the stress concentration areas of the tailshaft-to-intermediate shaft couplings. The multi-origin nature of the cracks was thought to reflect the detrimental effects of corrosion.

Excessive wear down on one of the vessels is considered to have been a contributory factor. In repairs these factors were taken into account. Alignment of all ships was checked and adjusted as necessary. The coupling flanges and fillets were machined on the remaining original shafts to remove the stress concentration effects of pits and to restore good surface finish. The fractured or cracked shafts were renewed.

A recommendation was made to coat the affected portion of the shafts and couplings to inhibit further corrosion. Further recommendations covered maximum allowable wear down and operation to restrict exposure to high torsional vibration.

This case pointed to a combination of design features and operational factors as the causes of the failures. It is unlikely that acting in isolation, one single factor would have caused the failures.

The field work to support the investigation involved telemetry to measure intermediate shaft torque and bending under dynamic conditions and the use of a strain gauge technique to measure shaft alignment which was supplemented by optical alignment telescopes. Extensive use was made of computer calculations of static and dynamic shafting characteristics and the metallurgical examinations contributed significantly to the case.

(3) It is unusual for lateral vibrations of the propeller shaft to be troublesome in the normal oil-lubricated stern tube bearing installations on single-screw ships. One of the rare examples concerns a bulk carrier where severe intermediate shaft and after-plummer bearing vibrations were reported on ballast passages in heavy weather. After about 1 year's service it proved difficult to maintain the tailshaft coupling bolts tight. The bolts were of a tapered design and their effectiveness to keep the coupling tight under lateral vibrations of the shafting was questioned, particularly as evidence of fretting at the coupling interface and the bolts was found. It was also postulated that the vibrations were mainly excited by transient partial immersion of the propeller in heavy weather.

Through systematic full-scale measurements it was found that with decreasing draft at the after end the dynamic after-plummer bearing load reduced progressively and that lateral vibrations increased in line with the reduction of bearing load and the loss of propeller immersion. The phase relationships of the vibrations indicated a vibration mode with the shaft supported on the single stern tube bearing and the forward plummer bearing which loaded the tailshaft coupling significantly and caused vibrations of the lightly loaded after-plummer bearing. Calculations indicated that with the after-plummer bearing unloaded the system was close to a transverse vibration resonance at service speed which was consistent with the observed shaft instability. To improve matters several steps were taken. Alignment was adjusted to increase the load on the after-plummer bearing within the constraints of the existing stern tube and engine alignment. The tailshaft coupling bolts were changed to a parallel shank design and they were hydraulically tightened to increase coupling interface pressure thus enhancing the effectiveness of the connection. Finally, deeper draught and greater propeller blade immersion in heavy weather were identified as major beneficial factors in reducing the tendency to shaft vibration in heavy weather.

The field work component of this investigation depended on measurements of the bearing loads and shaft bending moments under static and operating conditions using strain gauges and radio-telemetry and the lateral vibrations were measured with non-contacting probes.

(4) Torsional vibration failures of intermediate shafts are unusual. The relevant vibration modes can be predicted with some confidence using calculations which are based on extensive experience concerning excitation, damping, fatigue strength of the material and the empirical factors used to define crankshaft stiffness and entrained water at the propeller. Rare problems can arise in situations which involve new features of design or operation. A fairly recent case involved a bulk carrier fitted with a low-speed 2-stroke engine and a controllable-pitch propeller set in a steerable duct. The vessel had a narrow operating speed range, with the engine idling speed set above, but close to, the 1-node mode of torsional vibration which generates high vibratory torques in the intermediate shafting. The shaft line also incorporated a slotted oil distribution shaft to serve the controllable-pitch propeller. After about 1 year's operation cracks which were orientated at 45 degrees to the shaft axis were found to be propagating from the slots of the oil distribution shaft.

Metallurgical tests were conducted on the failed shaft and extensive torsional vibration and bending moment measurements were made. It was concluded that the shaft had failed in the areas of maximum stress concentration under torsional fatigue stress, produced either by repeated traversing of the torsional critical with no pitch on the propeller or inadvertent no-load idling at too low a speed, possibly in the presence of governor hunting or a combination of the two. Alignment was found not to have been of significance in the failure. In view of the findings it was considered that a sister ship was also at risk.

Options to deal with the problem included raising of the idling speed and traversing the critical with pitch on the propeller as this had been found to inhibit high transient vibration torques. The alternative finally selected for both vessels was to modify the torsional vibration system with a damper/detuner and after some fine adjustments of the damping this proved satisfactory. A summary of the principal results of this investigation which involved a considerable number of torsional calculations is given in Fig. 3.

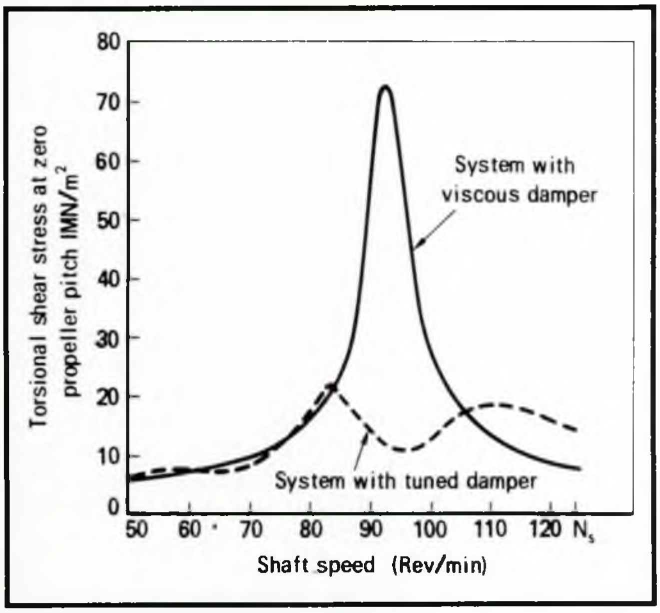

Fig. 3. Effect of changing damper on torsional vibration stresses

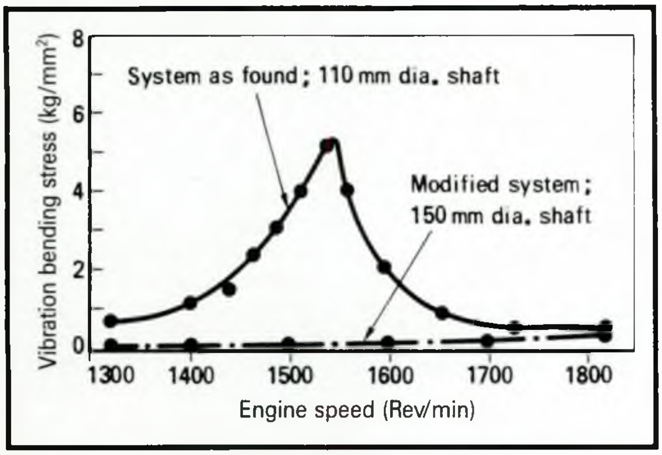

Fig. 4. Whirling-induced bending stress in intermediate shaft

(5) Given a suitable mass elastic system it is possible for any shafting to be excited into lateral vibration resonance. Excessively large bearing span-to-diameter ratios have been known to give problems and on a few occasions Cardan shafts with universal fork-type joints were involved. An interesting case concerns a tug boat fitted with a high-speed engine which propelled a Schottel unit through an intermediate shaft line. The shafting included short Cardan shafts at the engine and the Schottel unit. These were connected to a solid intermediate shaft which was supported with two roller bearings and was inclined from the engine room floor plate level up to the input gear of the Schottel unit at the stern.

During the original trials this shaft vibrated so badly that it bent. A replacement shaft of increased diameter was fitted on the recommendation of a local consultant, but high bearing vibrations persisted. A more thorough investigation based on dynamic measurements showed that the vibration was largely due to second-order whirling vibration of the shaft under excitation by the Cardan shaft joints. As was expected it was reduced to some extent by turning the Cardan joints at the two ends at right angles to each other, but at the cost of increasing second-order vibratory torques.

To prove that the shaft was moving in a near whirling orbit, the bending moment in the shaft was measured in two positions on the shaft which were circumferentially spaced 90° apart.

To deal with the problem it was finally decided to increase the intermediate shaft diameter from 110 mm to 150 mm and to reduce the overhang length beyond the bearings with subsequent satisfactory vibration results. Fig. 4 presents the reduction of the vibratory bending moment in the shaft. It should be realized that the bending signal transmitted by a strain gauge from a rotating shaft contains two orders of frequency, namely the true order $\pm$ 1. The mathematics describing this phenomenon are straightforward and have been published in several papers.

The original shafting, it is believed, had failed under a first-order whirling resonance and the first investigators were unlucky enough to raise the critical frequency of the shafting sufficiently to have it excited by second-order excitation after the change. In view of the known characteristics of Hooks joints this possibility could have been anticipated, but it must be realized that the calculations of this type of vibration are not easy and the TID investigation involved extensive mathematical modelling of the system before the design of the final shafting. This approach is often very powerful but pre-supposes competent access to relevant computing power.

The investigation relied on telemetry equipment to measure bending and torque on rotating shafts, and lateral and torsional vibration calculations were used to arrive at the final shaft dimensions.

(6) While torsional and axial shafting vibrations can exist as separate entities with their own natural frequencies and mode shapes, it is possible for them to combine to produce sometimes unexpected effects. The axial vibrations at the free end of the crankshaft induced by a torsional critical are well known.

A different case of interaction involved a 50,000 ton chemical tanker propelled by a direct-drive 5-cylinder long-stroke low-speed engine driving a four-bladed propeller at a service speed of 95 rev./min. Following reports of excessive fifth-order longitudinal vibrations on sea trials in the accommodation and on the main engine, it was recommended to brace the main engine longitudinally to the ship's structure at the cylinder head level. The recommendation was based on a hypothesis that the accommodation was excited mainly by the seating reaction to dynamically magnified main engine pitching motion. More measurements were carried out after these modifications when it was found that the original accommodation vibration problem had been solved. From measurements

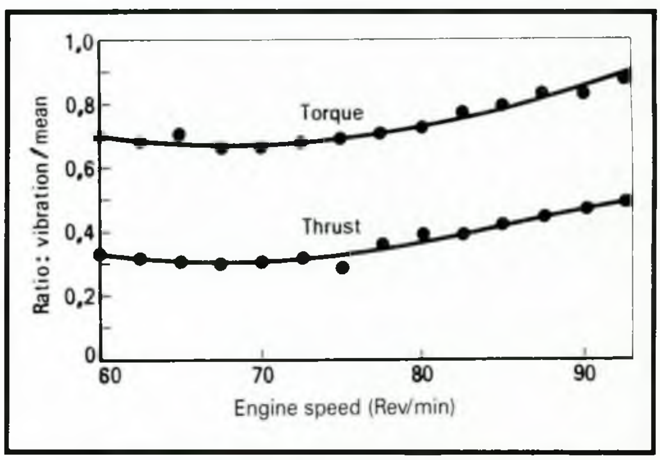

Fig. 5. Fifth-order torque and thrust vibrations

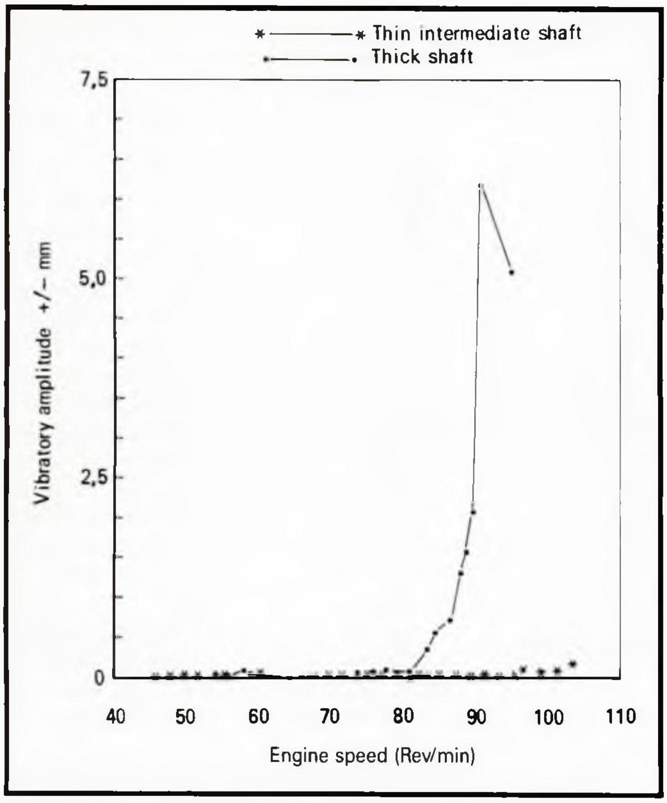

Fig. 6. Vibration of bottom transverse members

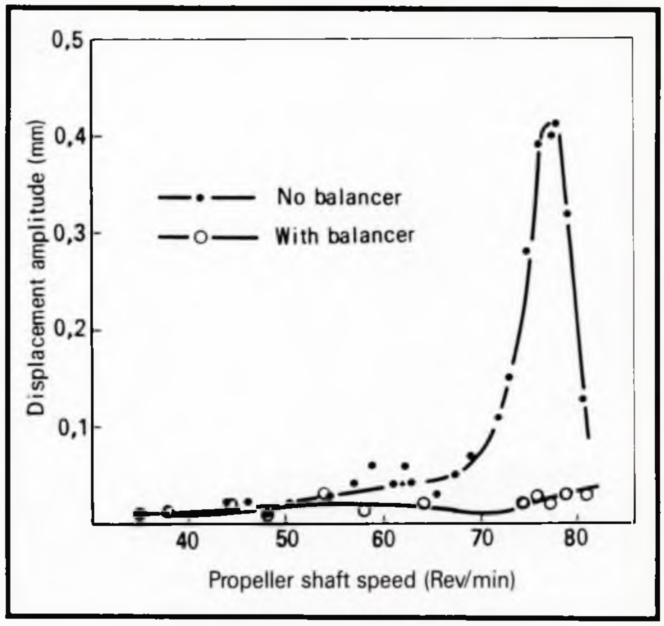

Fig. 7. Effect of fitting secondary balancer

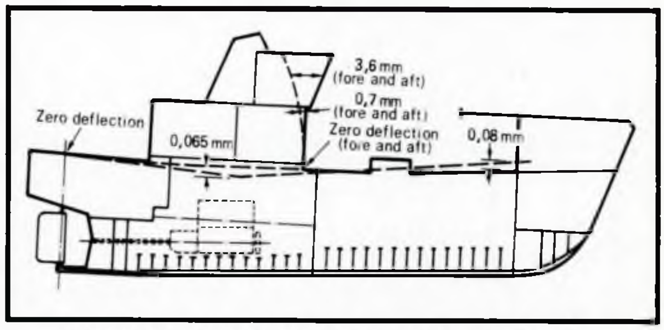

Fig. 8. Vertical vibration response of a trawler

of higher-order resonances, firing order excited torsional and axial line shafting criticals were deduced to be present some 30 % above operating speed and consequently they involved only minor dynamic magnification within the running speed. These forced firing order torque variations in the intermediate shaft were within normal allowable limits, although approaching the mean torque at service speed, and the corresponding thrust variations in the intermediate shaft were approximately half of the mean thrust. The phase relationship of the torsional and thrust vibrations was consistent with excitation of the thrust vibration by vibratory torque-induced speed variations of the propeller. These results, which were confirmed later also on other vessels of the class, are summarized on Fig. 5.

Prior to the bracing of the main engine the thrust variation, which is reacted at the engine thrust bearing, must have induced dynamically magnified longitudinal and pitching motion of the engine which gave rise to the problems in the accommodation. It was recognized at the time that the problem could have been solved in other ways. For example repositioning of the 1-node torsional critical below the service speed by the use of a reduced-diameter high tensile strength intermediate shaft was considered, but thought unnecessary at the time in view of the scale of the modifications involved and the simplicity of fitting the axial stays to the engine.

Some time later extensive cracking at the boundary welds of the bottom transverse members was found in the cargo tanks of a number of ships of the class and was identified by measurements as having been caused by longitudinal vibration of the bottom transverses when the tanks were full, due to excitation from the thrust fluctuations. Exploratory stiffening of the bottom transverses and provision of an axial crankshaft damper failed to deal with the problem and it was finally decided to resort to redesign of the shafting system which entailed fitting of a reduced-diameter intermediate shaft and torsional damper. This modification reduced vibrations in the cargo tanks to negligible amounts (Fig. 6).

It also allowed removal of the longitudinal engine stays, some of which had cracked in service and resulted in improved habitability in the accommodation. This prolonged investigation was based on the measurements of shafting torsional and axial vibrations and thrust variations in addition to external vibration checks on the main engine, in the accommodation and in the cargo tanks. Extensive calculations were made to confirm that the system was left in a safe state and to match the conceptual mathematical models of the dynamic characteristics of the shafting and the structural elements to measurements. By necessity the investigation brought together specialists in shafting dynamics, structural analysis and field investigation. The successful solution vindicated the aim of the investigation which was to reduce excitation in preference to large-scale structural stiffening which was believed to be an inefficient approach in this case.

(7) Balance characteristics of diesel engines can give rise to habitability problems. For example, 6-cylinder 2-stroke engines have an external unbalanced secondary couple and with typical low-speed engines this can excite the hull girder or superstructure on many ships.

At the cost of added complication an engine-driven balancer can alleviate such problems. A typical example concerns a 26,000 ton chemical tanker fitted with a 2-stroke engine developing 11,000 bhp at 81 rev./min. By fitting a secondary

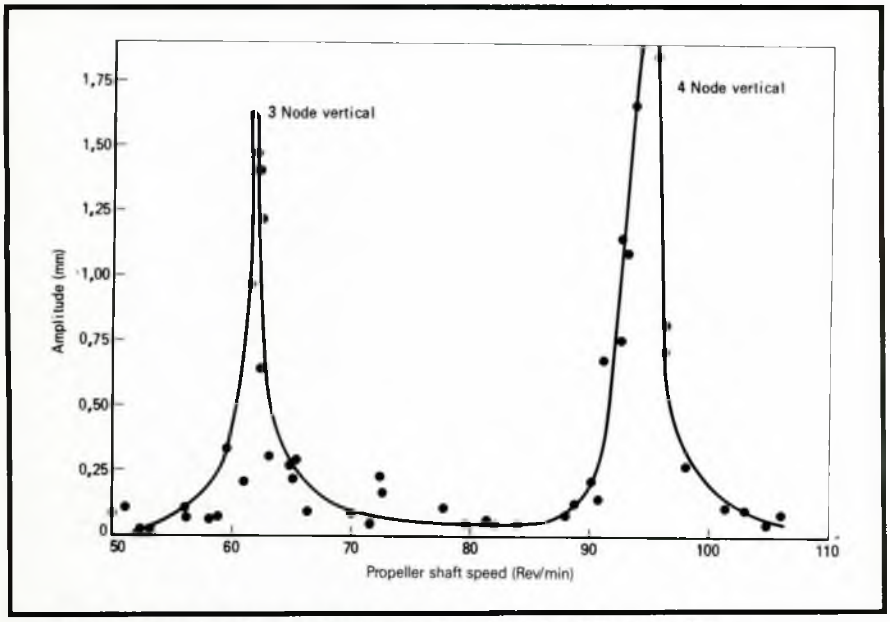

Fig. 9. Second-order excited hull vibrations in different vibration modes

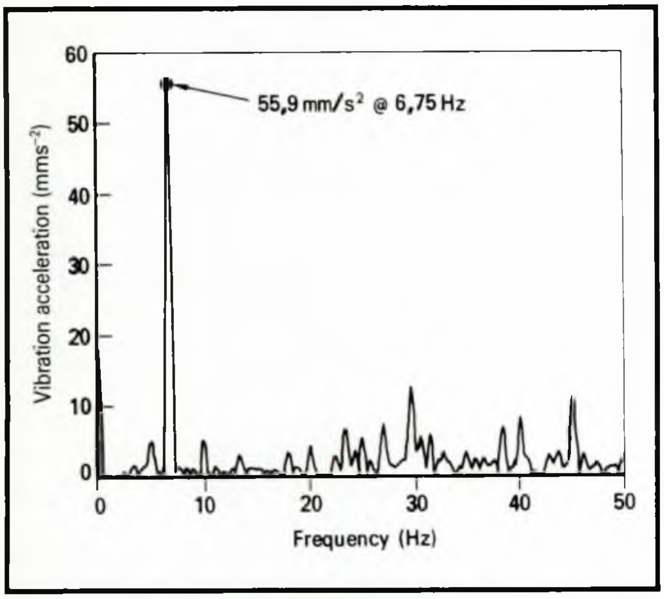

Fig. 10. Transverse vibration at deck 10 before balancing engine no. 3

balancer, troublesome accommodation vibrations were eliminated (Fig. 7), and main engine vibrations were greatly reduced. In another case, involving secondary unbalance of the main engine on a small vessel propelled by a high-speed engine, the accommodation above the engine room vibrated excessively at second engine order in diaphragm fashion, but was rendered acceptable by fitting pillars from the tank top to the accommodation (Fig. 8). In this particular case stiffening provided a more efficient solution than a reduction of excitation by the provision of a secondary balancer. Engine-driven secondary balancers are not entirely free of mechanical problems and in many cases may be difficult to retrofit after a ship has been completed. In several cases, involving secondary unbalance, electrically driven Lanchester-type balancers have been provided. They are usually placed in the steering gear flat which is a suitable antinode to counteract hull girder vibrations with opposing mass forces.

Hull girder criticals, in common with most structural vibration, involve very little damping and thus they develop over a very narrow speed range to considerable amplitudes, with low vibration levels on either side of resonance. As the critical frequencies are affected by the draft of the ship the potential overall speed range in which they are of concern is far wider than the critical itself, sometimes giving rise to apparently contradictory reports. Some owners are able to operate vessels with potentially troublesome hull girder vibrations by judicious adjustments of speed and more rarely of ballast distribution without the complication of secondary balancers. Clearly, objective knowledge of a vessel's characteristics can be of help in these matters. Results of a typical vibration investigation on a 25,000 ton tanker are presented in Fig. 9.

(8) Diesel engine primary mass or couple unbalance-generated problems are much less common, but a few cases are known where problems have arisen although the engines were theoretically balanced.

On a large cruise ship a first-order vibration problem generated by rigidly mounted 6-cylinder medium-speed engines was tackled at source by providing engine-mounted counterweights in two planes to oppose the vibrations. It was possible to halve the vibrations fairly readily by such means (Figs. 10 and 11). The magnitudes and angular positions of the weights were determined by a field balancing technique which has been used on turbo-machinery.

It is not obvious at first sight why reciprocating engines with theoretical external first-order balance should have excited such vibrations. Indications are that the excitation may have been due to a build-up of manufacturing tolerances on the rotating parts and on the proportion of the reciprocating component which rotates effectively with the crankshaft, but an internal couple may have been a contributory cause.

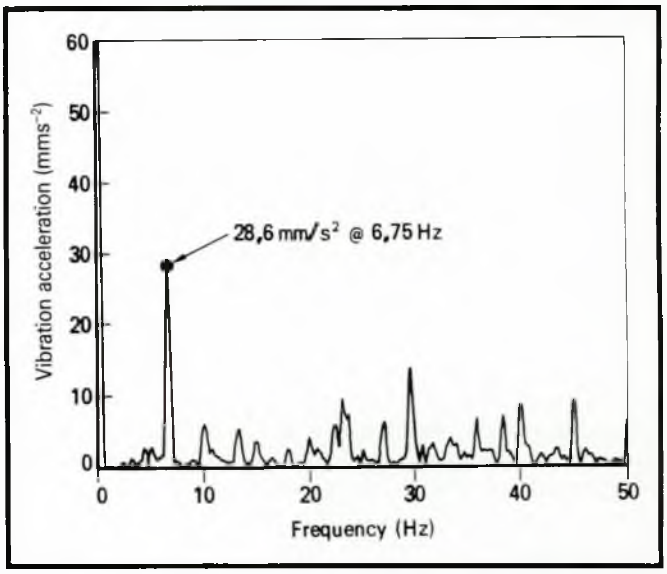

Fig. 11. Transverse vibration at deck 10 after balancing engine no. 3

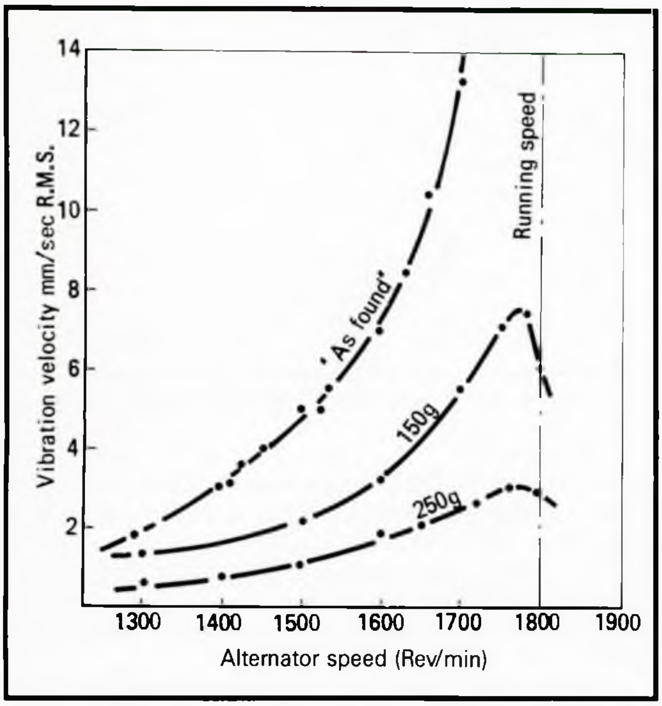

Fig. 12. Field balancing of turbo-alternator electrical rotor

(9) First-order vibration of turbo-machinery is, of course, normally excited by small unbalanced masses arising from manufacture or in-service deterioration, although other effects can be present. It is unusual for such vibration to be of significance other than on the affected machine where the main concern is with bearings and the danger of initial rubbing of glands and interstage labyrinths leading to permanently bowed rotors. If bearing vibrations are referenced to a trigger point on the rotor it is possible to field balance turbo-machines in their own bearings using an influence coefficient method. With rotors which operate above their first critical speed, three-plane balancing is required in theory but in practice two-plane or even single-plane balancing may suffice, thus avoiding the need to lift casings. Access to suitable balancing planes for fitting test weights or removing mass requires prior thought and effort.

A recent case concerns a replacement turbo-alternator fitted on a VLCC. The machine vibrated excessively despite numerous attempts at rectification which included realignment and low-speed balancing ashore. Using field balancing techniques it was found possible to get the machine to operate smoothly (Fig. 12). It was also found that the synchronous speed was close to a first-order lateral vibration critical of the alternator shaft line and thus was sensitive to unbalance excitation. To render the machine less sensitive to unbalance the lateral critical frequency of the system was displaced above the operating speed by removing a spool piece which had been introduced to accommodate the alternator on the existing seating.

This investigation, in common with all field balancing work, depended on being able to relate the phase of bearing vibrations to the rotor and was greatly eased by use of a dynamic signal analyser with this capability.

Essential analytical work was involved in the calculations of the lateral vibration characteristics of the machine and the assessment of the effects of modifications.

(10) Transverse vibrations of 2-stroke low-speed main engines on their seatings can give rise to a number of problems but in most cases they can be controlled by stays of various types which by preference should provide some damping. In a disturbingly large number of cases where problems are experienced the cause can be linked to failure to provide suitable anchor points for stays on the ship's structure.

Connecting stays to the mid-span of pillars or fitting them at a steep slope to meet a flat are typical examples of this type (Fig. 13). Improvements of such arrangements after sea trials are associated with considerable costs, and modifications therefore often end up as a poor compromise.

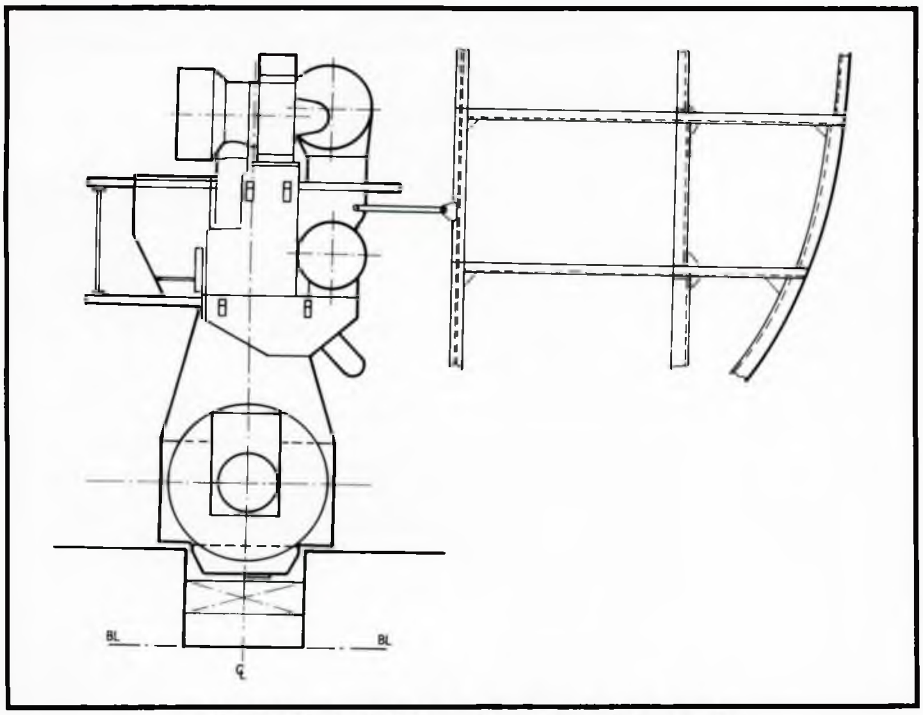

Fig. 13. Undesirable engine stay arrangement

Resilient mounting of engines avoids such effects, but at the cost of increased loads on the engine frame and sump and several, on face value, minor, but in practice, significant, difficulties. The operating speed range will be quite severely restricted by resonances below full service speed and on start-up and run-down several criticals which result in transient high vibration amplitudes have to be traversed. The various service connections to the engine and the securing arrangements of engine-mounted attachments have to be able to take these transient large excursions which may be up to several millimeters.

Nevertheless it is frequently possible to achieve a worthwhile reduction of vibrations, not only in the engine room, but also in the accommodation where sometimes the reactions to high engine vibration cause problems. Assessment of vibrations on ships in relation to habitability is now generally made on a basis of vibration velocity.

The curves produced by ISO which have also been adopted as a British Standard have found general acceptance (Fig. 14)

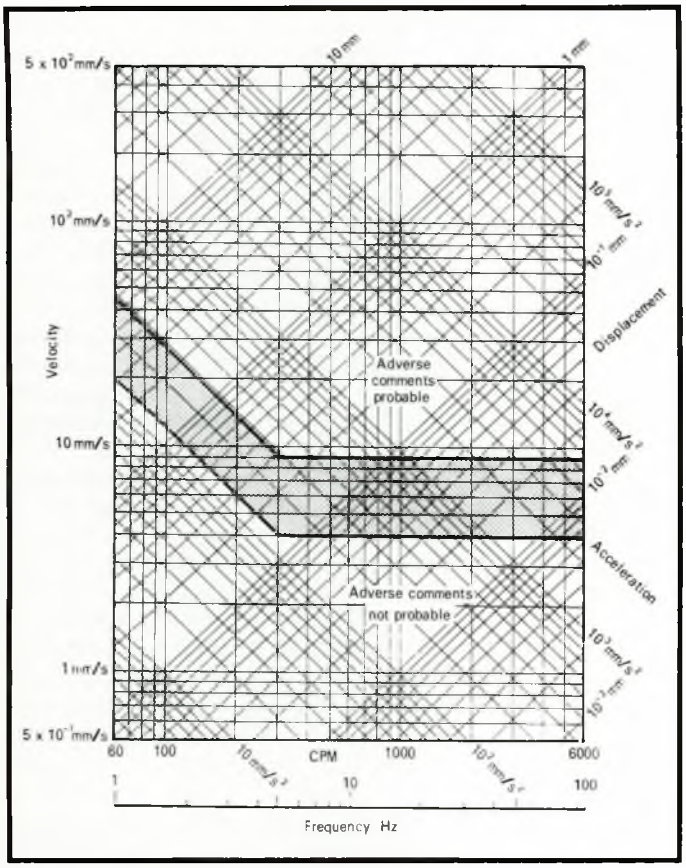

Fig. 14. Guidelines for the evaluation of vertical and horizontal vibration in merchant ships (peak values)

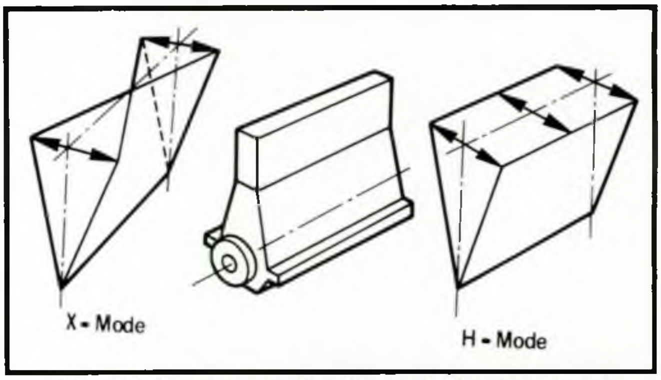

Fig. 15. Transverse vibrations of diesel engines

and form the basis of vibration guidance values of Lloyd's Register.

Implicit in the use of such criteria is harmonic analysis of complex analogue signals and it must be appreciated that the analysis technique can have a great effect on the answer which is obtained. For example, in the case of vibrations with strong-amplitude modulation which are often associated with propeller cavitation, the use of a long time span for root mean square (r.m.s.) analysis has an averaging effect which can reduce repetitive peak values by factors wildly in excess of the commonly quoted crest factors of 2 to 3. It may be argued that this practice represents the energy content of a vibration more correctly, but in the experience of the author this is not how vibration is experienced subjectively.

In general machinery-excited vibrations are remarkably repetitive and therefore results are less sensitive to analysis technique.

Engines can vibrate in a number of vibration modes, the most common transverse modes being the X- and H-mode respectively (Fig. 15). Excessive pitching or fore and aft vibration in a shear mode are less common but it has been noted that longitudinal engine vibration problems appear to have become more common with the latest generation of long-stroke engines with relatively low numbers of cylinders.

It is normal to use the amplitude of engine vibrations at the cylinder head level for assessments of vibration severity. No single acceptable level can be quoted but in the experience of TID the values shown in Fig. 16 are representative. The upper band of the shaded band represents VDI while the lower band is based on Japanese experience. The upper bound of the band is often associated with problems of engine-mounted attachments, such as pipes, turbocharger connections and manifolds.

(11) The final case study presented in this paper concerns a power station. The example is presented because it illustrates use of several different options to control engine vibrations and their effect on the environment.

Two 8-cylinder 12 MW low-speed diesel engines operate side by side in a machinery hall which is laid out much in the manner of a large twin-screw engine room. Each engine and alternator set is mounted on a large concrete block set into the ground and the installation is rather close to residential areas and public buildings. Initially the vibration amplitude at the cylinder head level had caused concern with the integrity of

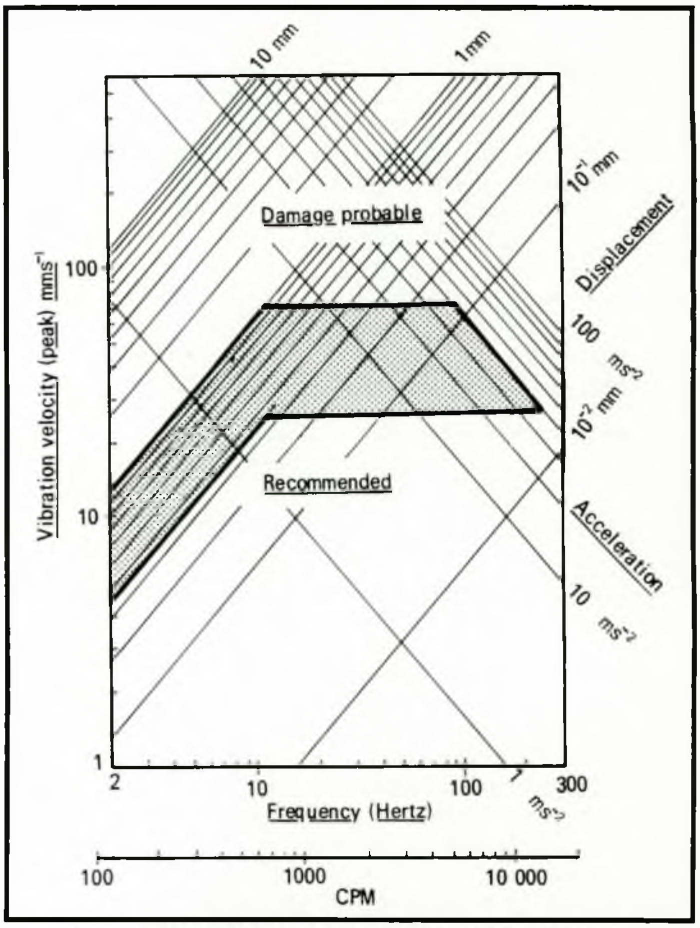

Fig. 16. Reciprocating engine vibration severity

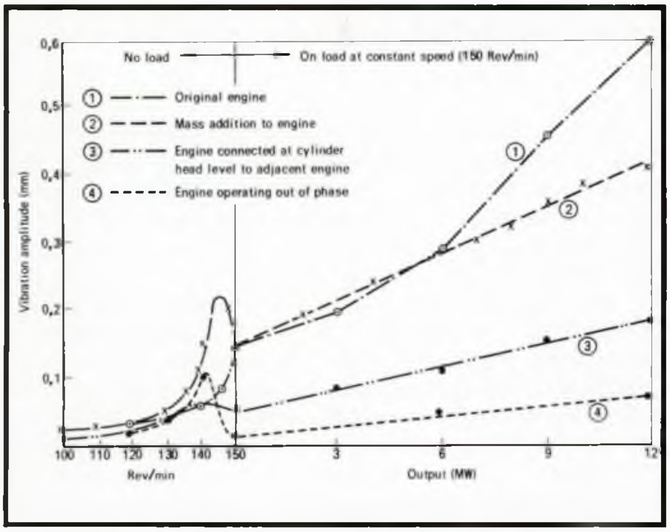

Fig. 17. Free end transverse vibrations at cylinder head level (fifth-order)

machines and their attachments. Measurements showed that fifth-order transverse vibration in the X-mode was involved with a critical close to, but above, the synchronous speed. As the engines are operated at a constant speed the engine builder added mass to the engines at the cylinder head level, thus reducing the critical to below the synchronous speed. The modification reduced the transverse vibrations at cylinder head level by about one-third. While this was considered satisfactory from the point of view of the engines and attachments, complaints of ground-borne vibrations were then being received from the neighbourhood making further reductions of vibrations desirable. At this stage the engines were connected with braces at the cylinder head level and in twin operation arranged to be mechanically out of phase with respect to the major fifth-order vibration frequency. In this configuration the vibration amplitude at cylinder head level was reduced to approx. one-third of the original value, but further improvements were effected by rephasing the engines relative to each other to put the two dominant vibration components (fifth- and third-order) into antiphase thus reducing the allowed relative angular phase relationship of the two crank shafts to only one. This modification reduced vibration levels to 10 % of the original value (Fig. 17). The vibrations transmitted into the environment were still measurable at this stage but roughly conformed to values suggested in the British Standard on Building Vibrations.

The investigation involved fairly simple measurements of vibration amplitude, frequency and phase, and the evolution of the modifications effected depended largely on an understanding of the vibration mode shapes involved and of the excitation mechanisms. This involved extensive calculations including finite element modelling.

The problem arose because of the combination of low-speed engines being placed in close proximity to residential buildings and soil characteristics which transmit vibrations readily.

It was recommended that future engines on the site should be medium-speed types mounted on resiliently supported concrete blocks.

CONCLUDING REMARKS

The case studies presented in these notes are only a small selection of cases on record. It is hoped that they illustrate that most problems are amenable to rational investigation. Solutions involve a variety of methods of investigation and correction and they frequently require extensive analysis.

It is often stated that engineering problems arise as a result of human frailty. The examples presented in this paper conform to this in the widest context and in many of the cases an inherent design weakness was exacerbated by operating factors or poor workmanship. With hindsight some of the problems might have been foreseen but it cannot be forgotten that the economics of ship building preclude an aircraft-type design approach. As a consequence many ships will continue to bear the hallmarks of floating test beds despite vastly increased design efforts of the advanced builders.

In this setting, through well-planned investigation and analysis of problems the technical knowledge in the industry is enhanced allowing progress to greater reliability and availability of ships and machinery.

Discussion

D. McKinlay (LR) Mr. Kunz is to be congratulated on producing an excellent paper which will be an asset to the Transactions of the Institute.

Vibration of one form or another has been with us since ships were propelled by mechanical means and the case studies reported in the paper show how modern ships are producing their own crop of problems. It would seem that Mr. Kunz has also found vibration problems on engines mounted on large pieces of concrete and he is to be congratulated on using his marine experience of the low-speed engine to solve problems in a different environment.

It is essential that the experiences reported in the paper are used in the design of new ships to ensure that they are not repeated. In this context, perhaps one of the more important investigations is that of case no. 6 of the paper. Stiff shafts were introduced on these ships to avoid barred ranges in the running speed of the machinery. To avoid one problem associated with the operation of the ship, we encounter another which affects the structure of the ship. This case is also an object lesson when structural failure is experienced; that is to tackle the problem at its source and not to attempt to alleviate damage by stiffening.

The accurate resolution of vibration problems has improved enormously with the availability and power of modern instrumentation. Mr. Kunz has made it clear that it takes experts, with the many different types of transducer available, to be able to use this equipment and give correct practical results. However, the real dividend for shipbuilders and owners occurs when these experts, together with others, using the power of the modern computer, study the design of new ships to ensure as far as possible that vibration problems and failures are avoided at the conceptual stage.

The use of diesel engines on cruise vessels has also resulted in a number of problems and it is interesting that Mr. Kunz has developed a field balancing technique which he shows can result in reduced vibration levels in the accommodation spaces. I wonder if Mr. Kunz could give some guidance on the range of installation types which could benefit from this treatment? Would Mr. Kunz also indicate if he would advise a rebalance after a major engine overhaul?

I am grateful for the effort that Mr. Kunz has made to produce a paper which contains a great deal of valuable information which will be of value to many of us in our daily tasks.

F. Kunz (LR) Mr. McKinlay asks if rebalancing of diesel engines would be recommended after a major refit on machines where trim balancing has been applied. There is at present insufficient evidence to be categoric on the matter and what is required in each specific case is an assessment of whether an overhaul has led to an increase of perceived vibration. On the vessel quoted in the paper a vibration monitoring point has been installed at a location which is sensitive to the particular form of vibration.

P. Morel (Bureau Veritas) We wish to congratulate Mr. Kunz on his paper which is of particular interest to people engaged in making *in situ* measurements. The importance of making actual dynamic measurements when seeking solutions to many machinery problems cannot be over-emphasized. We also support Mr. Kunz in his advice on the use of other techniques such as metallurgical analysis.

As demonstrated in the first case cited in the paper, calculations can, for various reasons, lead to mistakes; reasons such as insufficient definition of modelization, unknown parameters such as damping or exciting forces, and inaccuracy of the boundary conditions, and only the measurements themselves are able to take account of actual data.

We would like to draw attention to a remark by Mr. Kunz in case study no. 10 concerning the effect of utilizing too long a time span in spectrum analysis. This can artificially reduce the 'official' vibratory values not only in the case of the presence of cavitation but also in every case when vibration fluctuations are present. This point was underlined several times at ISO TC108/WG2/SC2 when builders proposed to increase this sampling time to take account of the lowest harmonic for very low running diesel engines (60 rev./min).

Finally we would like to take this opportunity to point out that different experts may resort to different measurement techniques, governed by the judgment of the vibration severity.

F. Kunz (LR) The author is much indebted to Mr. Morel for his contribution. Established criteria for assessing the effect of vibration on humans such as ISO 6954 or ISO 2631 are founded on single frequency tests, and situations involving multiple frequencies were at first used with quite different analysis techniques to those coming increasingly into use. Scanning the analogue traces for the amplitudes of repetitive peaks and major components was the mainstay of analysis in the author's organization for many years, to be superseded later by digital Fourier analysis of visually selected traces of analogue signals. In some ways the nearest to this, when analysing tape recordings on real time analysers, is the peak hold function, but this tends to be too severe, enforcing the use of multiple averages from different parts of the signal. The subject badly requires rationalization, but until this is done there is room for manoeuvre with analysis techniques and with the definition of criteria for reasons which are not always strictly technical.

It is agreed that measuring techniques must be suited to the problem in hand which reinforces the need for a working hypothesis before testing. To ponder the questions why, what, where and how measurements should be made, and what use can be made of the readings, is always instructive.

J. S. C. Bloomfield (LR) Mr. Kunz has related some interesting and difficult cases, and the effective manner in which they have been investigated and solved, in as far as vibration problems can be solved, is entertaining reading.

It should be added that, in addition to vibration, noise measurements and noise damping are also specialities of Mr. Kunz's department.

Working in a consultant capacity we observe that many vessels are contracted with very little investigation being given to these problem areas. In the larger ships certain combinations spring to mind such as 7-cylinder main engines and 4-blade propellers, sometimes ending up with inelegant mechanical vibration dampers, all adding to the complexities of the installation.

If the vibration problem is solved the owner says no more but if vibrations of a particular frequency persists, the owner will encounter repairs and renewals earlier than anticipated and budgeted for, with a reduction in his profit levels.

Pipework and electrical connections are two areas where failure can be expensive and particularly if failures occur at critical points, for example in a canal or during heavy weather or on a busy loading and discharging berth.

When advising a client it remains important to impress upon him that investment 'before signature' to a contract nearly always brings worthwhile benefits.

Mr. Kunz generally works on current problems but it is felt he has opinions that would be useful 'before signature' and perhaps he would like to comment on this.

F. Kunz (LR) Mr. Bloomfield invites comment on the desirability of considering noise and vibration and past experience prior to signing contracts. It is clearly prudent and technically correct to consider all these factors. Often quite astonishing omissions are noted after the event in contract documents. With the technical staff of many companies reduced to very low levels and the range of potential problems if anything being more complex than ever, most owners would be well advised to call on expert assistance, but it may be said that caution is also required when choosing an expert.

Mr. Bloomfield draws attention to the combination of a 7-cylinder engine with a 4-blade propeller. In the early 1970s this combination caused vibration problems in a number of cases, but it has been found that this particular system responds to selective phasing of the engine and propeller.

F. A. Manning The author is to be thanked for making this most valuable contribution to the Institute's proceedings.

I have always maintained that to depart from 'scientific method' when attempting to solve serious technical problems, is for the investigator to court disaster.

It is appreciated that commercial pressure may tempt one to take a big step in the dark but in my opinion this should be strongly resisted unless there are special (non-commercial) circumstances.

Problems should be identified at the outset and relevant repeatable data should be gathered. Possible solutions based on proper consideration of the evidence and data obtained should be tested where at all practicable. Only then should the best solution be put into action.

Turning to case study no. 8 in the paper, concerning residual primary unbalance from a medium-speed diesel designated as 'balanced' but causing a vibration problem, it reminded me of an incident of which the following details may be of interest.

The vessel in question was a well-known ferry which participated in the Falklands issue. Machinery was twin-screw CP propellers driven through gearboxes by two 16 TM 410 diesels running originally at 500 rev./min. Approximately 2 months before delivery to the owners the ship hull was subjected to comprehensive vibration testing and the major hull modes were determined. At the time of testing the ship was floating freely with approx. 3 m of water under the keel. The measured value of the 3-node vertical mode was precisely 500 cycles/min.

At that time it was assumed that an increase in frequency for that mode would occur when the vessel was in deep water. Coupled with the known satisfactory primary balance of the engine, no problems were foreseen for normal service.

During sea trials however an irritating 500 cycles/min vertical vibration corresponding to an acceleration of 0.05 *g* was present throughout all decks in the vicinity of the bridge. Unfortunately this was the area of the first class passenger cabins, and it did not take long to show that this area was precisely over an antinode of the 3-node vertical mode of the hull girder.

At that time (1974) it was assumed that a primary internal couple was the source of trouble. The ship was required for immediate service so during the first 3 months of trade a series of experiments was undertaken which showed that a 10 rev./min reduction in engine speed reduced the vibration to less than 0.02 *g*. This was agreed as acceptable and a permanent modification to the engine/propeller control was satisfactorily carried out and the problem was then considered to be resolved.

In case no. 9 I feel that the author is unwittingly supporting a rumour which has persisted in the steam turbine industry for many decades; namely that vibration can be the main cause which ultimately leads to a permanently bent rotor.

For over 35 years I have had a morbid interest in steam turbine failures caused by bent rotors. My records cover more than 80 machines in both marine and power station units of all sizes up to 500 MW. With the exception of four or five machines which failed due to inadvertent and sudden entry of water into the glands, the remainder failed due to a combination of cylinder distortion and inadequate gland clearance.

The consequences of gland contact between fixed and moving parts of a turbine and the mechanisms by which a permanent bend is developed is elegantly described in a short paper by the late H. G. Yates.[1]

In conclusion, for some years (1960–1980) I was interested in transverse vibration of 2-stroke low-speed diesel engines which is the subject of case no. 10 of the paper.

I have been away from main stream marine engineering since 1980, but during the preceding period I did not encounter an H-mode resonance problem at service speed.

However, I encountered more than one severe X-mode, 5th order resonance problem with 8-cylinder engines. These X-mode vibrations showed minimal motion at the bedplate. This I assumed was due to the ever increasing depths of engine room double bottom space, some of which under big engines exceeded 3 m.

Can the author advise whether the modern long-stroke engine has now resurrected the H-mode problem, and what is the correct ratio of H- to X-mode problems now being experienced by the marine industry?

Reference

1. H. G. Yates, Some Interesting Ship and Machinery Defects – Their Investigation and Cure, paper 7: Thermal Straightening of Turbine Rotors, *Trans.I.Mar.E.(TM)*, vol. 66, p. 77.

F. Kunz (LR) Mr. Manning advocates a rigorous technical approach to investigations with which it is difficult to disagree. Nevertheless, provided one knows very clearly what is being done, a pragmatic approach has merit and in many cases is the only way forward. For example in case 6 of the paper, the precise mechanism of the cross-coupling between torque and thrust variations was not amenable to mathematical modelling at the time when the recommendations were formulated, although an empirical connection had been demonstrated experimentally.

The question of low-speed engine transverse vibration modes cannot be answered dogmatically. Current 5-cylinder engines vibrate transversely in the H-mode near service speed, but amplitudes on the engines are normally quite reasonable. Problems experienced have been mainly associated with poor design of stays and anchor points and with the transmission of vibration energy into the ship's structure.

With respect to the comments on turbine rotors it was not intended to imply that vibration by itself bends rotors and it is not quite clear how the inference was drawn. Local rubbing of the rotor at the glands has been the probable cause of the cases investigated by the author.

J. Harrison (Three Quays Marine Services Ltd.) The comments in case 8 of the paper applicable to hull girder criticals can be expanded on according to individual experiences. As mentioned the actual critical frequencies are affected by ship loading and draft and can be additionally affected, or even magnified, in shallow water conditions.

Experience in the late 1960s/early 1970s with successive series of ships, similar in many respects, but with CP propellers on the earlier vessels and solid propellers on the later ones, showed that unpleasant and possibly damaging vibration on the earlier vessels had in practice been overcome by operating at a different pitch and rev./min combination so as to avoid resonance. More detailed calculations in the design stage of the later vessels revealed the potential problem which required a different solution as the decision for the propeller type had already been made.

The experiences quoted in case 8 should bring into question the engine designer's philosophy in specifying tolerances in rotating and reciprocating mass balancing. It may be that the Classification Societies and hence the author's involvement in this should be extended. One particular engine designer, unusually, quotes a moment which might result from the limit of tolerances of moving parts. An explanation of how the internal moment, which is quantified for this 6-cylinder engine, could be a contributory factor would be welcome. When this situation is extended to engines on resilient mountings, what criteria are available, or should be used by the mount designer, to allow for his inevitable unbalance which is in the engine? Who should be responsible for specifying this?

The example quoted in case 10 with the engine bracing at a mid-span of pillars may often be inevitable as the engine attachment position is predetermined by the engine designer and the position of engine room flats is governed by working heights and/or continuity of ship structure. The author's alternative solution to this problem would be appreciated.

It would also be of interest to have the author's view on the mounting configuration(s) most suitable to minimize noise and vibration transmission in vessels such as the several passenger, cruise or ferry vessels now under construction, where it appears all solutions are considered 'correct' by the parties involved, i.e. solid mounting, single flexible mounts, both vertical and 'Vee' configuration, and double flexible mounts.

F. Kunz (LR) Mr. Harrison questions how an internal couple could be a contributory factor to the excitation of hull vibrations. Theoretically such a couple should be absorbed within the engine structure and could only be exacted to influence excitation levels through the effects of the elastic deformation of these components on the supporting structure.

With resiliently mounted engines it has become clear that the engine frame, bed plate and sump are subjected to significant stresses and elastic deformations. In current practice engine transverse stays often end up at inconvenient heights, but the author remains convinced that better co-operation between ship and engine builders at an early stage could avoid the situation shown in Fig. 13 of the paper. There is no reason why provision should not be made to fit stays at different heights on the engines or why flats or casings should not be moved within reason given adequate thought at an early stage.

T. A. Wilkin (Three Quays Marine Services Ltd.) Some of the problems mentioned in the paper are familiar to most marine engineers; for example, main engine secondary out of balance. It is noted that the author has also considered it necessary to emphasize the need to match main engine transverse stays with adequate supporting steel work.

It would be of interest to know whether the problems investigated by LR's TID generally reflect progressive development leading to new problems or a loss of experience in the industry resulting in the need to re-learn some previous lessons.

The paper mentions three cases of lateral vibration (case studies 1, 3, and 5).

The problem discussed in case study 5 of an intermediate shaft supported in roller bearings whirling due to excitation from the Cardan shaft joints is unusual. Presumably, an investigation of the whirling characteristics of this system could ignore assumptions of mass and inertia at the propulsor and the effect of bearing stiffness. Could the author indicate whether the final bearing positions were chosen to achieve minimum static deflections?

Considering the more typical shaft systems mentioned in cases 1 and 3 the author notes in case 1 that it was concluded that the lateral vibration critical was closer to the service speed than calculated and in case 3 calculations indicated a system close to a transverse resonance with the after-plummer unloaded.

In the writer's experience the calculation of lateral shaft criticals has always approached black art rather than science and the current views of TID regarding the usefulness of such calculations, having regard to the assumption made in respect of entrained water, bearing support and stiffness, would be of interest.

Experience suggests that shafting systems with a bearing span/shaft diameter ratio of the order of between 14 and 20 can be satisfactorily aligned whilst avoiding lateral vibration problems. In the introduction to case study 3 the author indicates that it is unusual for lateral vibrations of the propeller shaft to be troublesome in the normal oil lubricated stern tube bearing installations on single-screw ships. This implies that problems are more usual on twin-screw ships. If this is the case is it due to a tendency towards longer bearing spans on twin-screw ships, and does TID experience confirm the usefulness of a rule of thumb regarding bearing spans as mentioned above?

The subject of shaft alignment is referred to in passing throughout the paper. The value of the paper would be enhanced if the authors could comment on the techniques currently employed by TID to measure static and dynamic alignment. A discussion of possible solutions to the problem of checking actual loads on the stern tube bearing of twin-screw ships would be of interest.

F. Kunz (LR) In reply to Mr. Wilkin, it is nevertheless correct to state that problems investigated by the author often arise from new design features which extrapolate beyond the framework of existing empirical design knowledge. Perhaps the advent of powerful direct calculation methods will change this situation, although at present they appear to be more often employed in explaining why things go wrong.

The bearing positions of the shaft described in case 5 were chosen to suit the natural frequency characteristics of the system. The calculations ignored the propulsor and bearing stiffness, but took account of the overhung mass of the cardan shaft, hooks joints and the shafting to the adjacent bearings.

Experience on modern cruise ships supports Mr. Wilkin's view that shaft spans of the order stated by him can give satisfactory service on twin-screw vessels, but it is noted that current practice on single-screw ships is to use much shorter shaft spans which place the criticals above the service speed.

Many techniques are available for static alignment checks with the vessel afloat and methods are chosen to suit the problem in hand. Where the shafting is connected and alignment in both planes or stern tube bearing loads are of interest, strain gauge methods are preferred to jacking, but on long and flexible shaft lines with a great number of bearings, alignment has been optimized sometimes by jacking. Gap and sag checks or lines of sight are employed in dry dock checks, as are other conventional, widely used techniques.

Load checks on the outboard bearings of twin-screw ships are difficult, particularly with the vessel afloat. To instrument the bearing struts and calibrate them is a possible method. Potentially this can provide static and dynamic loads, but has required a lot of advance planning and co-operation from the builder. It is also rather difficult to achieve good resolution due to the low strain levels in A-bracket struts.

On ships with exposed shafts between the hull and the A-bracket, underwater telemetry could be used to measure dynamic strain but this approach is not open on most modern cruise ships which have enclosed shafts.

It may be questioned why the bearing loads would be measured in the first place in view of the long shaft spans and good alignment tolerance of twin-screw vessels. If the concern is with instability of the shaft in the bearings and the potential detrimental effects on seals, then a more relevant measurement is that of dynamic shaft attitude within the bearing. This technique has a good prospect of success and the necessary resolution and it can be fitted in with a normal drydocking.

R. F. Munro For a number of happy and extremely interesting years I was a 'general practitioner' on the marine engineering side of the Classification Department of LR and so was enviably placed to consult many technically specialist colleagues at short notice when problems loomed.

The author and his colleagues in TID figure clearly in my recollection of these supportive gentlemen and I can vouch for the concentration and dedication they apply to their duties, with strong metallurgical back-up and direct access to an extremely powerful computer.

The paper presents 11 case studies which serve to whet the appetite for a sequel at an early date.

I wish to offer some comments on case study 2. In December 1960 a tanker entered drydock in Cardiff to undergo amongst other things, a routine screwshaft survey.

Under magnetic crack detection the coupling flange fillet of the screwshaft was found to have numerous cruciform cracks and following the use of exploratory machinery in the workshop the cracks were found to be so deep that the screwshaft had to be rejected.

The spare screwshaft was then carefully examined and found to contain the same defects, and could not be used.

The flanges and fillets of both of these shafts were severely corroded due to long exposure to cooling sea water emerging from the stern tube.

This led to attention being drawn to the condition of the aftermost coupling of the thrust shaft and, as could have been expected, this was also found to be severely damaged by torsional stress–corrosion fatigue and had to be rejected.

At this stage the ship's staff became aware that they were to enjoy a long Christmas and New Year holiday and the classification surveyor became unusually popular.

Attention was drawn to this matter in case 5 of the paper cited in ref. 1 below.

'Operators of ships having continuous liners on the screwshafts are strongly advised to consider protection of the hare steel shafts in the way of couplings from the effects of stress corrosion fatigue due to the continuous flow of sea water which is usually experienced as a result of the desire to keep the stern gland packing lubricated and cool.'

Since shafts of this type are still being specified this advice still stands.

Reference

1. J. F. Munro & P. E. Haynes, Some Marine Machinery Failures and their Causes, *Trans.I.Mar.E.(TM)*, vol. 97, paper 9 (1984).

F. Kunz (LR) Mr. Munro pertinently points out that some lessons have to be learnt over and over again. The risk of cracking of the unprotected forward sections of screw shafts due to excessive gland leakage remains real in all water-lubricated bearings. Many of the ships fitted with water-lubricated bearings are of the older vintage, and the experience gained in the early days of the ship's life may have been lost in repeated changes of ownership.

Since the preparation of this paper the author has been associated with two further cases of this type of cracking and the lesson may well be that it rests with the Classification Societies to provide the continuity in such and similar cases. With the changes moving away from the traditional types of manning this may become a more evident need as time progresses.

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PROPULSION SYSTEMS

WEGEMT TWELFTH GRADUATE SCHOOL

This graduate school is part of the WEGEMT programme of higher education in marine technology. The school is principally designed for engineers from the marine industries, but is also intended for postgraduate students. It has much to offer shipowners and those in shipbuilding industries, consultancies and classification societies. It will be clear from the content of this notice that there is a difference between such a school and a conference assembled for the reading of papers.

PROPULSION SYSTEMS

Successful operation of marine systems depends upon full understanding of basic design principles, coupled with a knowledge of modern techniques in computer-aided design, system design, data retrieval, condition monitoring, etc.

The school will provide a modern perspective to the appreciation of problems in propulsion system design at a high level and will also provide an insight into future developments. Lecturers have been drawn from a wide range of the marine industry, including engine builders and classification societies, as well as European universities where recent relevant research has been carried out.

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