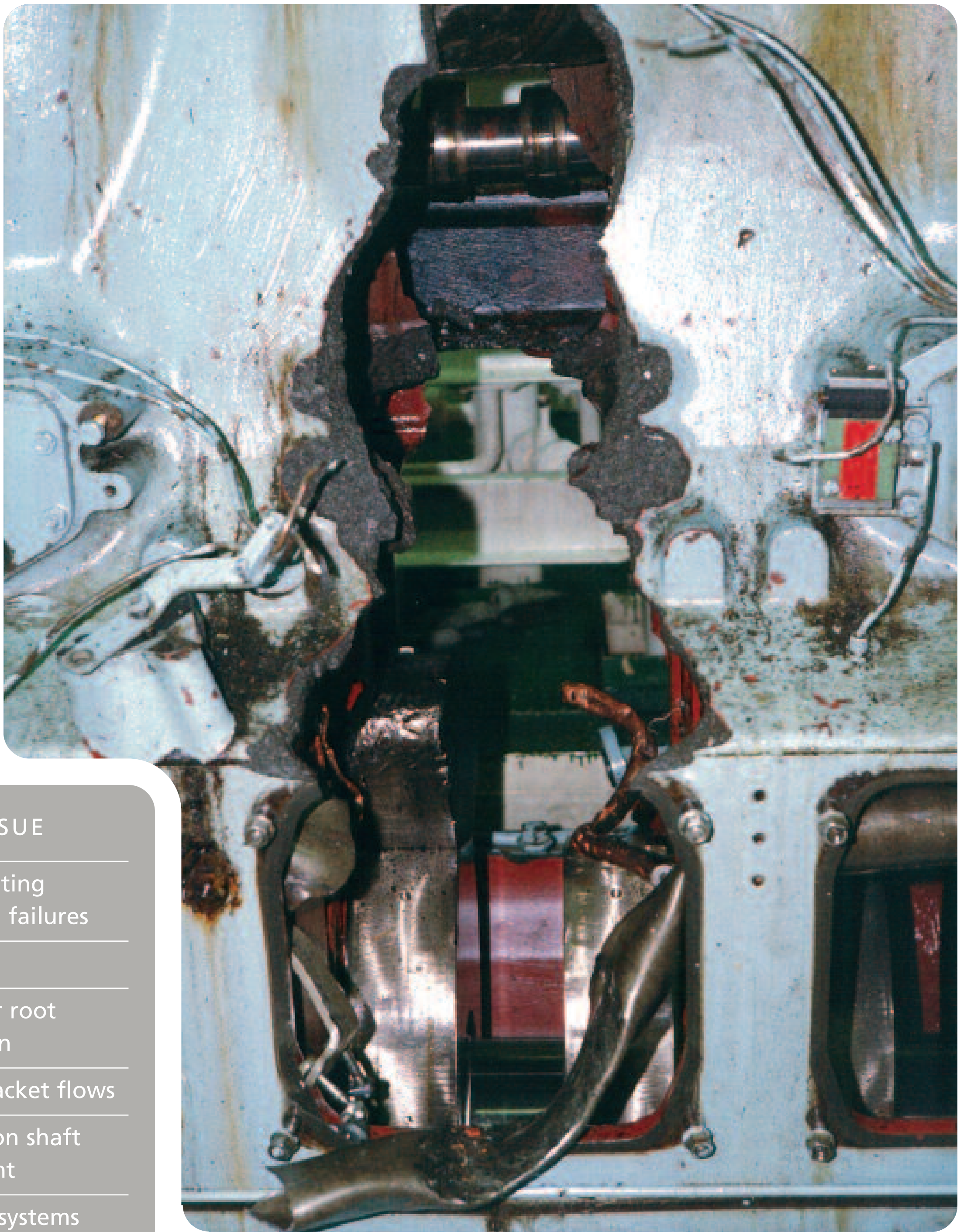
TECHNICAL MATTERS

APRIL 2006 Issue 1

Welcome to the first issue of *Technical Matters,* a publication containing a series of case studies covering technical problems and their solutions, produced by Lloyd's Register EMEA's Technical Investigations.



IN THIS ISSUE

Lloyd's Register

Investigating technical problems

For nearly 60 years, Lloyd's Register EMEA's Technical Investigations has provided the industry with cost-effective solutions to technical problems.

Shipowners, operators and shipyards can all experience technical problems with ship structures and machinery at any stage of a ship's lifecycle. Such difficulties can prove to be costly due to downtime, delay and the expense of repair. Establishing root causes and taking preventive and, where necessary, corrective measures are therefore of great importance.

Over the years, the marine industry has amassed a wealth of experience and expertise concerning technical failures and problems, and many well-recognised solutions now exist as industry standards and will no doubt continue to do so as long as knowledge is shared and retained within the industry. However, as ship technology continues to develop, with vessels growing in size and diversity of function and the carriage of ever-more sophisticated systems and equipment onboard, it is inevitable that new technical challenges will arise. Other factors that impact the industry's collective ability to prevent and deal with failures include the loss of traditional skills, corporate memory loss and old-fashioned human error.

Lloyd's Register EMEA's Technical Investigations capability was established in 1947 to explore marine failures and to research technical problems with a view to improving the Rules. Since then, it has evolved into a unit which not only carries out this original function, but also provides sophisticated technical investigations services to owners, operators and yards, helping them to find cost-effective and technically-sound solutions to their engineering problems.

The scope of these investigations covers a wide range of ship systems and structures, including:

- propellers, and other underwater elements such as rudders, fins and A and P brackets
- shafting systems, including sterntubes and intermediate (plummer) bearings
- gearing systems
- engines, including crankshafts
- hull structures, both global (hull girder) and local (panels), due to wave and related impacts such as bow and stern slamming
- superstructure items, such as accommodation blocks, masts, decks and bridge wings.

Over the past three years we have investigated over 200 failures for over 100 different clients, onboard both Lloyd's Register-classed and non-Lloyd's Register-classed vessels. This experience has shown that the majority of recent problems affect LNG ships, passenger ships and oil tankers. The most common technical difficulties experienced relate to shaft alignment, sterntube bearings, gearing, vibration and noise, engines and podded propulsors.

To prevent lessons learned from being lost, we have brought together a set of case studies covering a wide range of commonly experienced technical problems, along with their solutions, drawn from the publication of a comprehensive technical paper celebrating the 50th anniversary of Technical Investigations originally published in 1997. It is hoped that by learning from and remembering the past, similar failures can be avoided in future.

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CASE STUDY 1

Rudders

SUBJECT VESSEL TYPE
Container ship

PROBLEM
Rudder plating erosion

A wide range of rudder or steerable duct actuating force, cavitation and integrity problems can occur from time to time. In the case of rudders, these may take the form of bearing failures or erosion of the rudder plating, the latter due either to cavitation developed by the rudder itself or from the cavitation entrained in the helical slipstream from the propeller. Steerable ducts can also have accentuated vertical seaway loadings and, in common with fixed ducts, can suffer from erosion on their inner plating. Actuating force problems can occur with either form of steering device.

Typical of such an investigation was the full-scale measurement with strain gauges of rudder stock moments on the semi-balanced spade rudder of a container vessel. This assignment was undertaken in order to establish the reasons and to provide a basis for remedial action for both an under powering of the steering gear and a poor response to the helm from the ship. Rudder torque and stock bending strain measurements were made and the results of this trial's programme were compared with computed loadings. These were derived from the surface pressure distribution over the rudder horn and blade by means of one of Lloyd's Register's lifting body vortex panel codes.

For this computation, the incident flow to the rudder was evaluated from a lifting line propeller analysis procedure. The rudder model was verified initially using the model test data and, subsequently, calculations were performed for the container ship rudder in the slipstream of the propeller for comparison with the measured full scale data. It was found that good agreement between the measured and predicted loading was obtained and that the form of the rudder stock torque relationship for a 35 to 35 degree rudder angle zigzag manoeuvre was well represented.

The same modelling procedure was then used to predict the effect of increasing the rudder chord length by 20%. The predicted maximum torque increased by more than 150%, while the side force increased by approximately 10%. A similar percentage increase in rudder torque was subsequently re-measured on board the ship when the rudder extension was fitted.

LESSON
Rudder problems can be solved by a combination of full-scale measurement and theoretical approaches. Small changes in dimension can lead to large changes in applied torque.

CASE STUDY 2

Propeller root cavitation erosion

SUBJECT VESSEL TYPE
Fast displacement craft

PROBLEM
Deep cavitation erosion in the propeller blade root

Propeller root cavitation problems can be among the most complex to solve. In this region, along with tip vortex dynamics, rigorous mathematical analysis is currently of little assistance for design or analysis purposes.

In some cases, when scant attention has been paid to the propeller design environment or where ship layout difficulties have arisen and high shaft angles or a poor choice of advance angle have resulted, erosion problems can occur which are difficult to resolve. One such case is that of a fast displacement craft where deep cavitation erosion in the blade root regions was produced after only 30 minutes running at full power.

The problem was identified as originating in the angle of attack variations at the blade roots due to the propeller shaft's inclination. **Figure 1** shows the full-scale cavitation on the propeller blades when operating at high speed as observed through windows in the hull. Of interest is the cavitating structure at the trailing edge of the root cavity which was in the root erosion region.

An air injection system was devised based on previous successful experience by Lloyd's Register's Technical Investigations. This proved to be moderately successful in that it allowed the vessel to enter restricted service while a more permanent solution was found. Full-scale observation of the air injection process on the ship provided key information for a subsequent series of model tests.

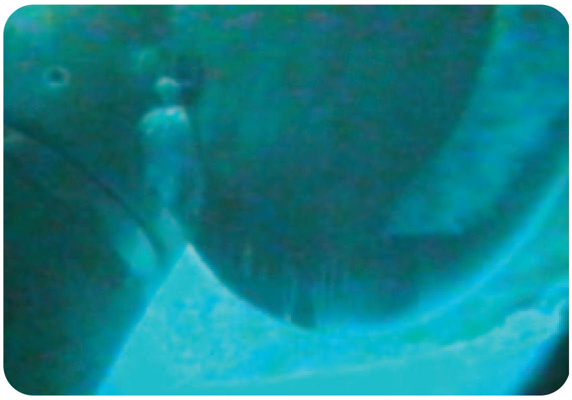


Figure 1

The video images showed that due to the high pitch ratio of the blades, in excess of 1.7, this conventional approach did not allow sufficient air to be held in the cavities and hence could not fully cushion the cavity collapse process, producing the erosion damage.

Model tests were used to develop an alternative air supply configuration and machined holes in the blade roots at approximately the one-third and two-third chord positions at the 0.3R radius. Through an iterative process, a suitable air-flow rate was determined such that a minimum speed loss was achieved while maintaining intact stencil ink coatings on the model blades, which were used to assess the tendency to cavitation erosion.

The machined holes in the blade roots appeared to delay the onset of erosion and increased the speed at which air injection would need to be started. In addition, the holes provided a mechanism by which air could transfer from the pressure side to the suction side and ensured that air was resident in the root cavities for a substantial part of each blade revolution.

The vessel is now in normal service and, after the equivalent of one full year of high-speed running, some minor erosion damage is still present but it appears to have stabilised.

LESSON

Cavitation problems can be solved much more readily when the problem area can be viewed either by window or boroscope. Possible solutions include re-profiling, drilling of pressure-relief holes, flow pattern changes and/or air injection.

CASE STUDY 3

Shaft bracket flows

SUBJECT VESSEL TYPE
Coastal ferry

PROBLEM
Cavitation and vibration caused by poor A bracket shape

Hull surface pressure and vibration measurements carried out during sea trials on a small twin-screw coastal ferry indicated that the shape and positioning of the A brackets, together with the design of the propeller, were likely to be the main causes of unwelcome cavitation activity on the propeller blades and consequent levels of structural vibration. In this case levels of 125 mm/s on the after deck were regularly being reached. A re-designed, highly skewed propeller replaced the original propeller, which reduced the vibration levels considerably, but not to an acceptable level. Through windows cut into the hull to allow observation, cavitation was seen to increase locally as the propeller blades passed behind the outboard arm of the A brackets.

Because of the scaling complexities of using model tests to understand the flow field around the A brackets, the flow field was analysed using a computational fluid dynamics (CFD) code in combination with Technical Investigations' propeller analysis codes. **Figure 2** shows a view of the solid body representation and a part of the computational grid used in the simulation model.

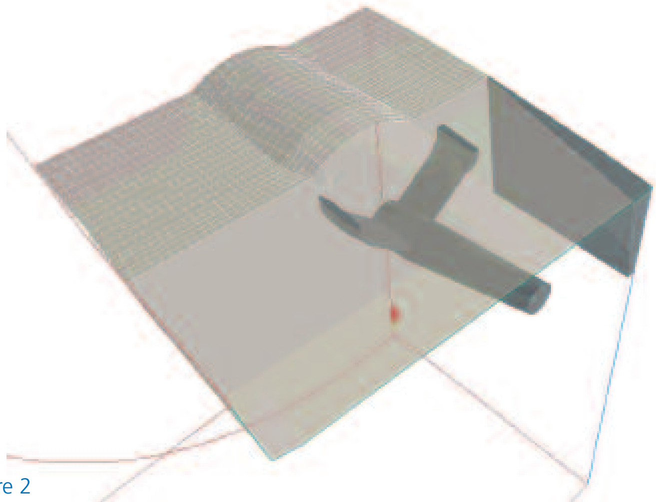


Figure 2

The simulations showed that the A brackets were contributing significantly to the wake field disturbance and could be better aligned to the local flow directions. However, to achieve such an alignment would have required the vessel's twin-screw shafting system to be mechanically re-aligned, which the shipbuilder was reluctant to undertake if other methods could be developed to solve the flow problem. A compromise was achieved by re-adjusting the leading and trailing edge shapes to form an S-shaped cross-section of the A bracket. The system was then re-analysed with the modified bracket geometry in order to confirm improvements. Contours of velocity ratio at several planes downstream from the brackets allowed the quality of the axial wake entering the propeller plane to be assessed.

Figure 3a shows velocity contours and transverse components at a location half a bracket length behind the trailing edge of the original brackets. Results for the re-aligned bracket geometry, **Figure 3b**, showed a marked reduction in the axial wake peak behind the outboard A bracket arm (outlined on the two figures), indicating that vibration and noise levels might be reduced by adopting such a design. The diffusive effect of the partial propeller tunnel in conjunction with the blockage effect of the bracket cross-section was also considered to have caused the formation of a separation pocket on and behind the upper bracket arms.

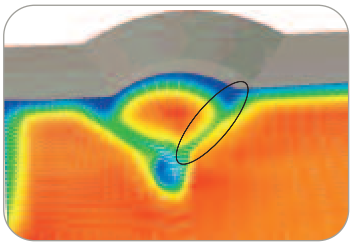


Figure 3a

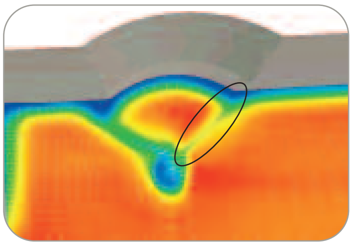


Figure 3b

The CFD calculations further showed that this latter tendency could be reduced by rounding both the leading and trailing edges of the upper bracket arms, this design aspect not having featured in the builder's original A bracket design. Subsequent vibration measurements following the modification of the A brackets confirmed that an acceptable vibration characteristic had been achieved.

LESSON
Hydrodynamic problems can be solved through a combination of theoretical and experimental methods. CFD is a powerful computational technique that can be used for a wide range of problems and avoids the disadvantages of scale-model testing.

CASE STUDY 4

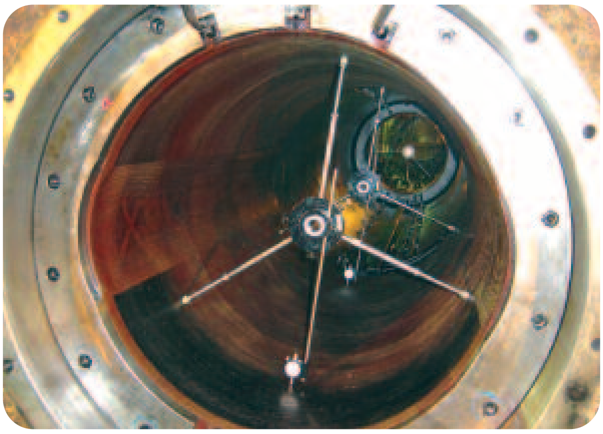
Propulsion shaft alignment

SUBJECT VESSEL TYPE
Bulk carrier

PROBLEM
Alignment failure due to unloaded bearings

Detailed predictions of hull deflections and their effect on shaft alignment are rarely included in design calculations. Additionally, the designers of direct-drive diesel engines specify permissible limits for the static forces and moments that can be imposed on the engine crankshaft by the propulsion shafting in order to achieve positive downward static loads on the engine bearings. The span between the engine main bearings is short, and these alignment tolerances represent stringent requirements in the same manner as the more familiar differential load limits specified for the output gearwheel bearings of main propulsion gearboxes.

Many of the above aspects were evident during the investigation of the propulsion shaft alignment on a large bulk carrier. Design bearing loads and engine alignment were specified and the installation procedure was prepared to achieve these values. It was assumed that the alignment would be changed only by the relative thermal rise of the engine.



Inherently, shaft systems with a single plummer bearing (as was the case on this ship) require an especially careful consideration of the initial alignment to accommodate the changes that can be expected on vessels with a large displacement variation between the ballast and loaded conditions. In this instance there was insufficient load on the sterntube forward bearing, and the engine alignment was too close to the hogging limit. Correction of a hogging moment at the engine flange could not be achieved by raising the plummer bearing without unloading the sterntube forward bearing. The scope for adjustment was further exacerbated by installation errors which resulted in a relatively high static load on the plummer bearing.

As the vessel's draught increased the respective acceptance criteria were progressively compromised in that the sterntube forward bearing became unloaded; the plummer bearing became overloaded; and the hogging moment imposed on the engine crankshaft increased beyond the design limit and the No. 7 main bearing was unloaded.

A fundamental re-alignment of the engine was required to restore realistic practical alignment margins. It is, however, a salutary lesson to report that these margins were subsequently eroded in part by the permanent alignment changes that occurred due to the relaxation of hull structure residual stresses during the vessel's maiden voyage and an initial encounter with adverse weather.

The static shear force and bending moment at the engine output flange can be verified only by strain gauge measurements and it is suspected that many installations are operating satisfactorily with shaft alignments that may be outwith the permissible values because the limiting parameters are not necessarily reflected by unsatisfactory crankweb deflections.

The execution of the engine re-alignment would have been an especially interesting experience for surveyors of the past who struggled to optimise engine bedplate and crankweb deflections by the selective adjustment of chock thicknesses. Their work was based on micrometer measurements taken from a taut piano wire judiciously threaded through the crankcase above the crankshaft in a way which avoided the connecting rods. They may have been surprised to find that the structure of the six cylinder engine installed in the bulk carrier was sufficiently stiff that it could be supported only at its four corners without jeopardising the crankweb deflections.

LESSON

Shafting systems usually require careful alignment, especially when the shaftlines are short and stiff. Unloading and/or overheating of bearings can be a warning of impending failure.

CASE STUDY 5

Gearing systems

SUBJECT VESSEL TYPE
LNG ship

PROBLEM
Gear tooth failures

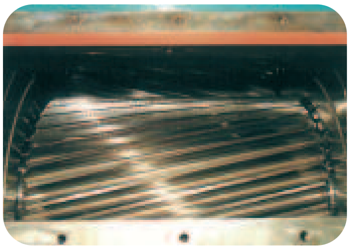


Figure 4

Despite improvements made in the design arid manufacture of propulsion gears and an almost universal adoption of case-hardened teeth, infrequent but persistent failures continue to occur. Throughout our experience, the common failure modes of tooth fractures and surface damage caused by either scuffing or fatigue have been reported. Such repetition must be attributable to the mechanical complexity associated with tooth-meshing conditions and the interrelated influence of elasto-hydrodynamic lubrication.
The following examples illustrate the importance of the findings derived from metallurgical examinations and the intractable nature of some problems when compared to an obvious immediate cause of a similar tooth fracture.

Figure 4 shows scuffing damage on the case-carburised pinion of a single reduction main propulsion gear train. The characteristic 'tiger-stripe' pattern is thought to reflect small undulations generated by the grinding process, and the extent of the damage suggested a tooth-load distribution which was biased more heavily towards the forward end of the helix. The reason for the observed tooth misalignment was confirmed by measurements of the gear wheel attitude in its bearings taken under running conditions which indicated a tilt induced by deflection of the integral main thrust bearing on its seating.

Nevertheless, this behaviour was similar for three other identical gearboxes which ran without a problem in the same service application. The most plausible explanation of the scuffing was thought to be a temporary deterioration of the lubrication perhaps caused by an ingress of water, but as in many cases where operational shortcomings are suspected as being a principal contributory factor, the failure hypothesis can be difficult to substantiate.It may be of interest that subsequent careful running-in of the scuffed tooth surfaces re-established satisfactory operation.

Fatigue cracks which lead to tooth breakages invariably originate on the tooth root surface or on the flank in areas of pitting and exfoliation damage. The isolated single-tooth failures shown in **Figures 5a and 5b** are less common where fatigue cracks have grown from sub-surface origins. In both cases, cracks were initiated close to the transition between the core material and the carburised case - a position where the expected residual stresses would be tensile. Other similarities included an absence of metallurgical defects and a heat treatment procedure which resulted in a satisfactory case depth hardness profile and distribution of carbides.

The failure shown in **Figure 5c** could be easily explained by the remaining evidence of debris that had been entrained in the mesh. This was found as a small accretion of steel cold welded to the tooth flank in the area of the sub-surface crack origin. The chemical composition of the alloy confirmed that it was a foreign body. Conversely, the similar single-tooth fracture of a different propulsion pinion remains unexplained; all the more so because the crack origins were some one-third of the face width distant from the end of the tooth, as shown in **Figure 5b**.

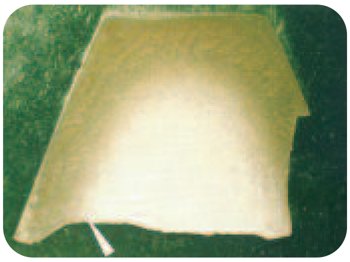


Figure 5a

Figure 5b

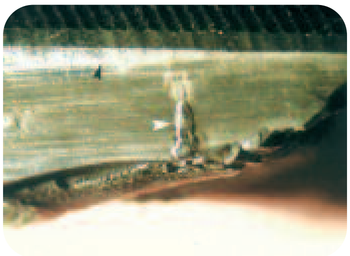


Figure 5c

Problems have also arisen on less precisely manufactured gears. The rack and pinion jacking mechanism on a self-elevating offshore drilling platform suffered from excessive plastic deformation of the tooth surfaces. Strain gauge measurements taken in the tooth roots and an analysis of the contact stresses led to new pinions being fitted with a greater number of teeth.

LESSON
Gear tooth meshing and wear is a complex and sensitive subject. Practical experience is a highly desirable attribute in related investigations.

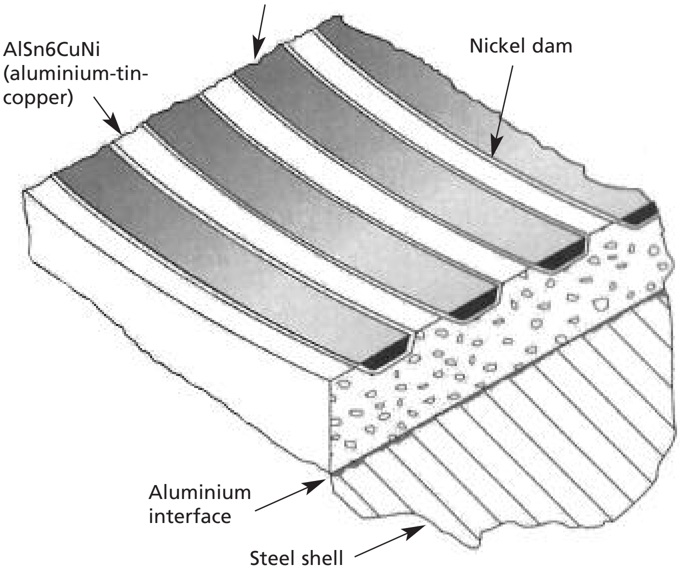
CASE STUDY 6

Diesel engine bearing failures

SUBJECT VESSEL TYPE
Passenger ship

PROBLEM
Cracking of bearing materials

Typical of the periodic problems that arise is the fatigue cracking of soft 'whitemetal' or 'babbitt' bearing materials. These lead or tin-based alloys have a poor inherent fatigue strength which is further impaired by modest temperature rises. Their ability to tolerate lubricating oil contamination and imperfection between the journal and bearing surfaces is, however, an indispensable attribute. In medium-speed engines such materials are used as a thin surface overlay to protect the stronger copper-lead or aluminium tin alloy bearing material from incipient pick-up and potential seizure.



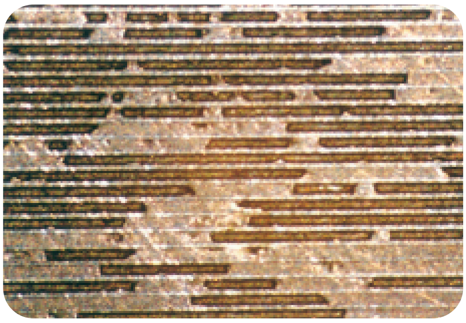


Figure 6

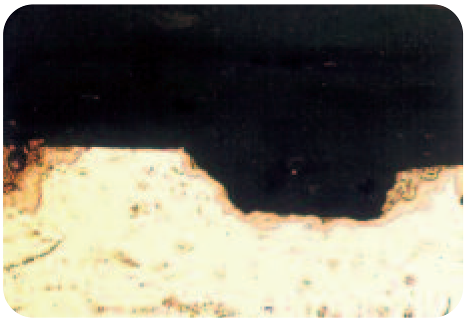


Figure 7

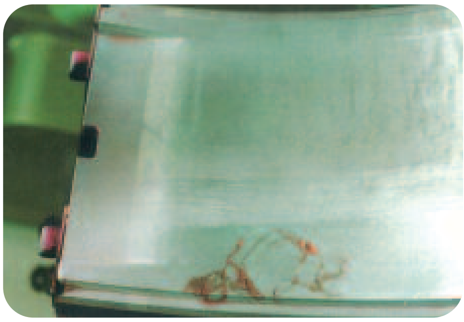


Figure 8

Removal of the overlay is detrimental and equipment manufacturers provide well-documented guidance regarding the extent of permissible wear.

The 'Rillenlager'-type bearing was designed to counteract overlay-wear problems for engines burning residual fuels; **Figure 6**. Fatigue cracking and consequent premature loss of the overlay from the grooves of such bearings have been found in a connecting rod application, **Figure 7**, which required a stronger but harder overlay material.

Similar cracking also occurs in the thicker tin-based alloy layers used in slow-speed engines where the same material is used for load bearing and bedding purposes. Experience of fatigue damage to these bearings suggests that cracking is influenced by oil film thickness. Cracks originate on the surface, although their growth and the detachment of fragments can lead to an initial impression that the metallurgical bonding between the bearing material and the steel backing shell has failed. Engineers should also be aware of the importance of the material microstructure in terms of the distribution and grain size of the respective alloy phases. In addition to fatigue cracks, **Figure 8** illustrates a mottled surface appearance caused by thermal ratcheting of a poorly cast tin-based alloy.

LESSON
Particular attention needs to be paid to good lubrication, avoidance of contamination and good quality control in engine bearings.

CASE STUDY 7

Shipboard vibration

SUBJECT VESSEL TYPE
Oil tanker

PROBLEM
Excessive vibration and tank cracking

A shipowner reported that, onboard one of its new ships, there were excessive levels of vibration in the accommodation and localised cracking of various tanks and support brackets due to vibration. The vessel's power was supplied by a five-cylinder, two-stroke diesel engine, developing 2,942 kW at 210 rpm, with the power being directly transmitted to a five-bladed, fixed-pitch propeller. An investigation was carried out by Lloyd's Register prior to the ship's drydocking. During the investigation, the linear vibration characteristics of the hull, superstructure and engine were quantified, in addition to the torsional characteristics of the main engine shafting system. These calculations showed that fifth-order vibrations, measured in the accommodation spaces at some operating conditions, were at a level where complaints regarding habitability were considered likely to occur.

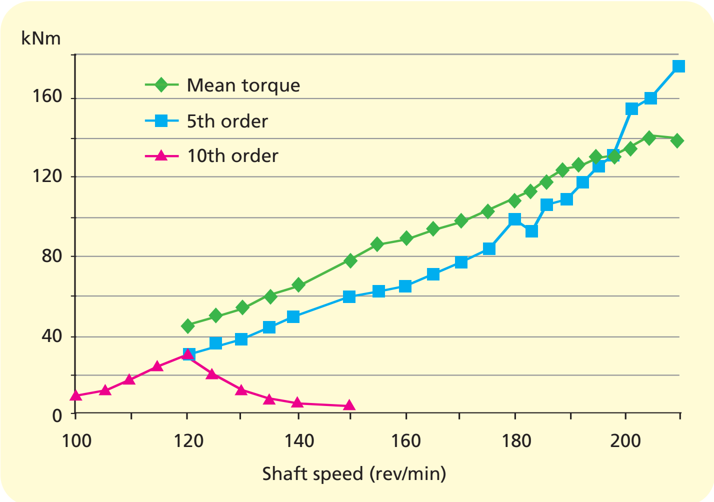


Figure 9

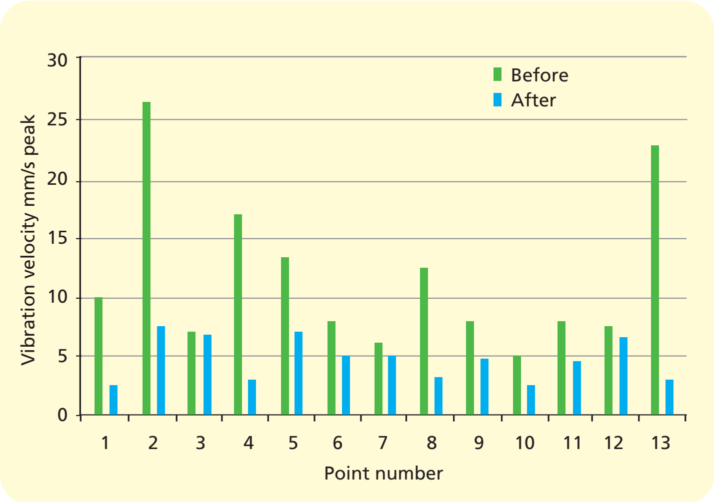


Figure 10

Furthermore, vibratory torques which were measured in the intermediate shaft exceeded the mean torque at higher shaft speeds, as seen in **Figure 9**. Associated with these vibratory torques was a corresponding increase in axial vibration of the intermediate shaft.

A one-node, tenth-order torsional vibration critical speed was measured at 130 rpm, indicating that the corresponding one-node, fifth-order critical frequency would occur at 260 rpm. This speed was only 24% above the main engine speed of 210 rpm and compared with a critical speed of 284 rpm calculated by the engine builders. Consequently, vibratory stresses and torques were larger than predicted as the operating point was further up the flank of the resonant response curve.

It was recommended that global vibration of the ship could be reduced by either a re-tuning of the shafting torsional vibration characteristics which would entail the installation of a torsional damper, a change of shafting and possibly a barred speed range or, alternatively, the installation of an electrically driven, independently mounted, fifth-order thrust compensator. The vessel's owners elected to adopt the latter recommendation and the modification was implemented. Subsequent vibration measurements indicated that vibration levels throughout the vessel had fallen to a level below those where complaints regarding habitability were likely as seen by **Figure 10**.

LESSON

A variety of answers exist to in-service vibration problems; an appropriate and cost-effective solution should be chosen, whether it be via reduction, attenuation or isolation.

CASE STUDY 8

Noise emissions

SUBJECT VESSEL TYPE
Cruise ship

PROBLEM
Noise from ship disturbing local residents

Residents living near a cruise ship terminal complained about the noise from one particular ship. Noise measurements were taken at a number of locations in the residential area, both with the ship at its berth and with the berth unoccupied. The overall 'A'-weighted sound pressure levels were considered acceptable but analysis of the noise signals when the ship was berthed showed a distinct peak at 37.5 Hz. This gave the noise a tonal characteristic which was the feature of the noise spectrum which the residents found annoying.

The source of the tonal noise was identified as exhaust gas pulses from the four diesel generators exiting the funnel, with the frequency of the pulses corresponding to the firing frequency of the six-cylinder, four-stroke diesel engines. Pressure measurements on the inlet and outlet sides of the installed exhaust gas silencers showed that frequency components less than 50 Hz were not attenuated effectively by the silencers.

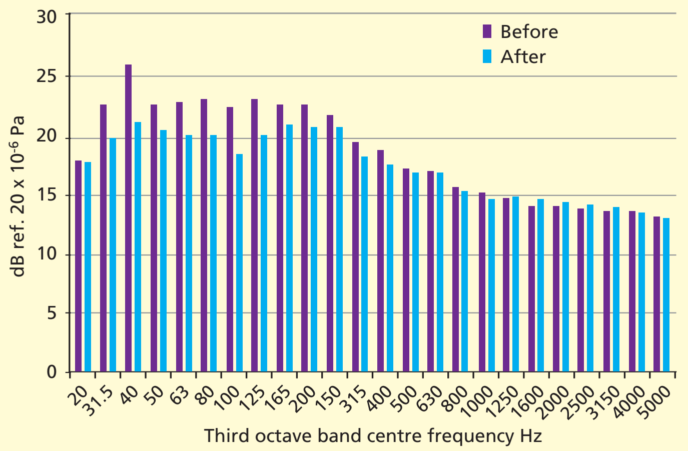


Figure 11

An assessment of the generator exhaust gas system dynamic behaviour was made using Lloyd's Register's 'MERLIN' diesel engine simulation program. The system allows the thermo-fluid dynamics of a particular system to be simulated against time, which permits a functional analysis of the design details to be undertaken. The predicted results showed good correlation with measurements, and further simulations were run to investigate the effect of fitting an additional plenum downstream of the existing silencer to act as a buffer to dampen the pressure pulses. These calculations demonstrated that such a solution would be effective.

The measurement and calculation results were then used by a silencer manufacturer to design a secondary silencer and these were fitted in each of the four exhaust trunkings. Follow-up measurements confirmed a significant reduction in low frequency noise components, **Figure 11.**

LESSON
Appropriate computer simulations can be extremely effective in aiding problem solving.

CASE STUDY 9

Crank bearing bolts

PROBLEM
Low-speed marine diesel engine failure

This investigation, although involving a land-based installation, has direct implications for the marine industry. Our surveyors are often called upon to attend the site of a mechanical mishap to conduct an investigation into the cause. Sometimes the cause is surprisingly simple, as in this case of a loose crank bearing bolt.

The engine which suffered damage was an eight-cylinder two-stroke marine diesel engine arranged to drive a large alternator. The first indications that it had a problem were loud knocking sounds. Unfortunately, before the engine could be brought to a stop, one of its connecting rods separated from the crank pin and emerged from the crankcase, causing substantial damage. The final state of the connecting rod can be judged from **Figure 12.**

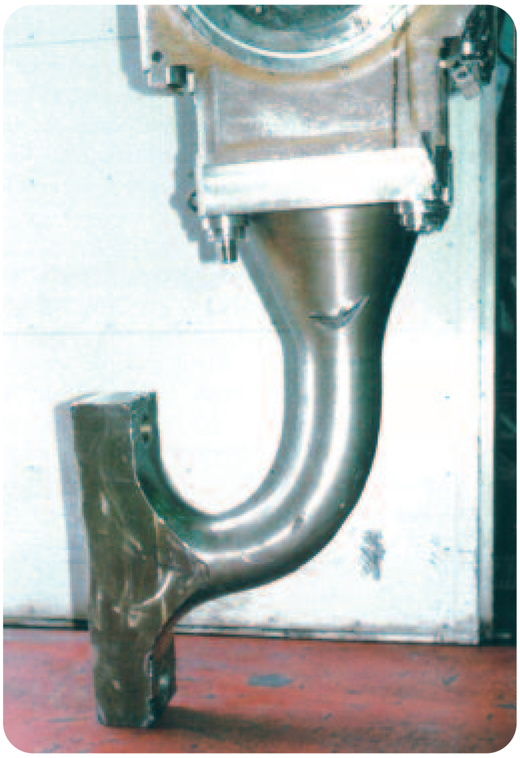


Figure 12

As the various damaged parts became accessible, possible alternative causes were tested for credibility. By this process, attention was increasingly drawn to the need to find the crank bearing bolts, which were not initially accessible. When they were retrieved, one of them was found to be virtually intact, but without its nut. This nut, its locking device and locking screw were all found separately. The nut threads for about one and a half turns nearest to the lower face were seen to have been damaged by axial shear.

About one quarter of the first turn, although present, had been separated from the nut around its circumference. The other bolt had broken under severe combined tensile and bending load and its nut, with locking device and screw, was still present on the broken end. Furthermore, a bolt in a different connecting rod was found to be 1 mm slack, with its nut locking device and locking screws correctly fitted.

This evidence clearly showed that the slackening and loss of the intact bolt was the initiating cause of the engine damage. In searching for the reason why the bolt became slack, the effects on the bolt of possible cylinder malfunctions were calculated and physical tests of similar bolts were conducted. The key finding from this work was that if the bolt had been properly tightened, the probability that sufficient torque could have been generated to loosen the nut against the frictional resistance of the threads and of the nut face was minimal. The torque due to tensile load, which normally acts to unscrew a nut, was found to be one eighth of the frictional torque resisting such movement. It was concluded, therefore, that the damage resulted because the bolt had not been properly tightened to the correct tensile load.

The routine maintenance procedures adopted for this installation involved periodic checks of the bolt tension. The method used was to slacken and re-tension the bolts and, as such, this procedure had the potential to introduce error.

LESSON
Human error often plays a part in failures. Using an experienced investigation team is often crucial is getting to the real root cause

CASE STUDY 10

Ship superstructure vibration

SUBJECT VESSEL TYPE
Cruise ship

PROBLEM
Excessive bridge wing vibration

A twin-screw, diesel-electric cruise liner reported noticeable vibration on the bridge wings from the time the ship entered service. The propulsion system comprised four medium-speed diesel generator sets supplying power to two three-phase synchronous- and frequency-controlled electric motors driving controllable-pitch propellers.

The vibration investigation was carried out while the ship was in normal cruising service. It was found that while the global vibration characteristics of the ship under both port and sea-going conditions were satisfactory, the bridge wing's vertical natural frequency was excited at first-order excitation frequency of the main engine. The vibration of the bridge wing varied in a complex way, with the port wing suffering to a greater extent.

High to severe vibrations were experienced when the ship was manoeuvring, at full speed with all four engines running and operating with No. 4 engine running.

It was found that the main engine's resilient mounting system was reasonably effective. However, the residual vibration energy transmitted to the ship at the engine's first-order frequency was sufficient to dominate the ship's vibration and to excite the bridge wing resonance. While the vibration severity was considered unlikely to cause structural damage in the short term, the effects on the operation of the local control console and the compass repeater were of concern.

It was therefore recommended that both bridge wing structures were stiffened in order to raise their natural frequency above the excitation range. As with all problems involving resonant conditions, care in implementing the remedial action was taken so as to avoid an undue response from other excitation frequencies.

LESSON
Even with good design and construction, in-service vibration (and noise) problems can arise. Understanding the nature of the problem will lead to a solution, by adjusting either mass, stiffness, damping and/or excitation force.

TECHNICAL MATTERS

These case studies, together with others, originally appeared in Lloyd's Register Technical Association Paper No.1, Session 1997-98. In view of their continuing relevance they have been reproduced here, in an updated and enhanced form.

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LIFE MATTERS

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