Paper read at the Institute on Tuesday, 7th December, 1971

MARINE MACHINERY FAILURES

B. K. Batten, M.Sc., C.Eng., M.I.Mar.E.*

In 1960, the late Mr. H. N. Pemberton, then chief engineer surveyor to Lloyd's Register of Shipping, introduced a paper to the Institute by saying:

"... knowledge of service experience, especially in regard to machinery defects, is of considerable value to both marine engine builders and shipowners, particularly if this knowledge can be utilized in improving design or materials or in preventing mal-operation of machinery".

There is still no better statement with which to begin a similar paper after some ten years more experience, during which the Society has carried out over 1100 specialist investigations besides numerous metallurgical examinations of failed components.

The present paper covers a range of problems from the unusual to the commonplace and, while by no means a comprehensive catalogue of the Society's world-wide activities, illustrates there are still many lessons to be learned in design, reliability, and plain engineering sense.

Mr. Batten

INTRODUCTION

The technical investigation department of Lloyd's Register of Shipping has the opportunity of seeing numerous interesting machinery failures. Lest this should lead to an attitude of complacency, it is important to realize that each one of these failures is a cause of distress to owners, builder or shipper. Invariably, the discovery of the reason for these failures is the result of the combined thought of many people, and often many disciplines, and the necessity of keeping an open mind cannot be over-emphasized. However, one thing is certain, failures do not just happen; there is *always* a reason: "Acts of God" there may be, but they are invariably clearly God begotten, and the phrase was never intended to cover man's ineptitude or material insufficiency.

Investigators make mistakes—the best will admit it and take a new course of investigation; and without this frank and open approach shared between owners, builders and investigators, many important facets of machinery breakdown are concealed. It is a salutary thought that after a further ten years of progress engineers have to contend with largely the same problems—vibration, alignment, material fatigue etc. Until one is prepared to look for failure points in the grass roots of design and fabrication practice, one cannot hope to see a reduction in these unfortunate incidents.

PROPELLERS, SHAFTING AND STERNGEAR

Controllable Pitch Propellers

In view of the comparative complexity of controllable pitch propellers, it is natural to expect that the failure rate is higher than for fixed pitch propellers. Considering all c.p. propellers built to Lloyd's Register class between 1960 and 1969 with engines developing 5000 hp and over, the defects from all causes (excluding contact damage) amounted to 40, an incidence of 11·1 per 100 years of service.

Defects have been largely divided among operating mechanism failures, blades or their securing bolts and blade seals.

Two interesting cases investigated concerned failure of a propeller blade pivot and of a control system. Examination of a broken blade pivot on one vessel led to the examination of similar rings on other vessels which were also found cracked (Fig. 1). The rings were made from S.G. cast iron, and the cracks originating from the fillet between the boss and the

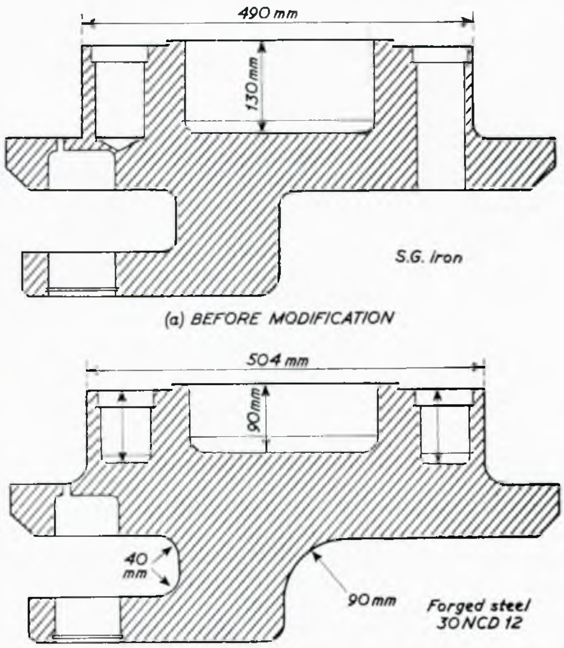


FIG. 1—*Variable pitch propeller blade pivot rings*

* Principal Surveyor of Technical Dept., Lloyd's Register of Shipping.

flange had fatigue characteristics. Metallurgical examination showed that while there was extensive galvanic corrosion pitting in the fillets, the fatigue cracks had not originated from these points, but from apparently notch-free surfaces. The structure of the cast iron contained numerous zones where inoculation had not been effective and, while this is normal in fairly large castings, their incidence in this pivot was higher than normal. Changing these pivots to a carbon steel of 0·32ton/in² U.T.S. and increasing the fillet radius (Fig. 1b) cured the problem.

Failure of the control of a KaMeWa propeller system occurred at the moment a ship was entering locks. A slow ahead movement was given by bridge control, followed, almost immediately, by a stop order. It was then noticed that the ship was gathering speed ahead, and excessive ahead propeller pitch was indicated. The officer on the bridge de-clutched the engines and rang for an emergency full astern. This had been anticipated by the watchkeeper who, on seeing the indication of unusually high propeller pitch, changed the pitch manually, too late to prevent bow contact.

After clearing the filters, the propeller mechanism was tested by the remote control systems for about 1½ hours. Apart from a little dirt in the fine mesh operational element of the change-over filter, the remaining filters in the line appeared to be clean.

There were two systems which could affect propeller pitch, a pneumatic control system and an oil hydraulic operating system. To maintain zero pitch, a constant pressure of 14 lb/in² was required in the pneumatic system, and should this fall, the propeller would adopt an ahead pitch. In all aspects this system appeared to be effective.

The oil supply to the telemotor was bled from the propeller pitch-changing hydraulic system and passed through two filtering stages before reaching a control orifice at the entrance to the differential control piston. The balance pressure on this piston was regulated by a needle valve, the position of which was controlled by the pneumatic system. The arrangement was such that a reduction of pressure supply to the differential piston resulted in an ahead pitch movement of the propeller.

Analysis of the sludge found in the oil drain tank showed, beside sand, steel and rust, other particle sizes down from 0·003 in to 0·0001 in which could pass through the 100 micron mesh (0·004 in), and also the pencil filter protecting the first strangler hole. These small soft and friable particles composed of fibres and paint flakes could accumulate at the nozzle entry during manoeuvring periods, and rapidly upset the delicate balance in the hydraulic system. Clearly, the cleanliness of such oil is of paramount importance, as is the necessity to avoid, at design stage, areas where build-up of contaminants and restriction of oil flow could occur.

MAIN SHAFTING AND STERN GEAR

While the purist may regard shafting only in relation to propulsion, it is an integral part of the machinery of a ship, and failures occurring in this area can clearly not be ignored.

Whirling of shafting occurs in certain shafting systems, especially if weardown or misalignment has reduced or removed the load from the forward stern tube bearing. In one such vessel where whirling had been measured to occur within the speed range of 70 to 90 rev/min, the screwshaft was found cracked both in the keyway and under cavitation damage on the liner. This liner had been sand cast with a number of core plugs fitted, Fig. 2, one of which had presumably come out, as the surface of the tailshaft was exposed through the eroded bottom of the core hole.

A paper on this subject was presented to the Institute in November[1].

Excessive weardown of port and starboard after sterntube bearings on a twin screw passenger ship led to the Society checking the alignment conditions of both sets of shafting using strain gauges, and measuring the dynamic bending stresses in one of the tailshafts whilst in service.

The change in amplitude and phase of the first order bending which was coincident with certain pitching motions of the ship indicated a change in the vertical position of the screwshaft after point of support. The arbitrary selection of the after

FIG. 2—*Pitted tailshaft liner*

point of support is significant, as it affects all subsequent alignment and frequency calculations. It is the Society's normal practice, based on experience, to assume this point to be one third along the after sternbush measured from the after end. The measurements on this ship suggested that there was intermittent excessive loading of the after end of the after sterntube bearings due to deflexion of the bossing arms in certain operating and sea conditions, which could have been the cause of excessive weardown, and could, under certain running conditions, lead to short term whirling of the screwshaft. The calculated bodily displacement of the shaft to produce the bending stress measured earlier in the screwshafts was approximately 0·165 in and it was suggested the cantilever movement of the bossing arms might be of the same order.

Vibration stresses had been measured in bossing arms at sea on previous occasions, but no clear relationship between deflexion and stress had been established. It is reasonable to assume, however, that shaft deflexion is not caused by a parallel vertical movement of the bossing structure, but by a hinged cantilever action with maximum deflexion at the after end. This would mean the longitudinal axis of the bearing being angularly displaced relative to the shaft axis resulting in a periodically reduced effective load-carrying bearing area.

The strain gauge investigation was divided in two phases:

1) Jacking the port bossing arm to determine the relation between bossing deflexion and bending stresses in the tailshaft and bossing arm;
2) Measuring the stresses induced in the bossing structure under a given deflexion, and interpolation of the measured stress level under operating conditions to derive the associated shaft vibratory bending deflexion.

The results showed first that the bossing deflexion, both vertical and horizontal, increased linearly with load, suggesting the bossings were behaving as simple elastic beams.

The bending stresses in the tailshaft did not increase linearly with deflexion; in fact, there was a reduction in stress in the horizontal plane due to an inboard movement of the shaft contrary to the outboard movement of the bossing arm. This suggested that as the after end of the sterntube was lifted, the relaxation of weight on the forward sterntube bearing allowed the shaft to move across (inboard) within the clearance of the bearing, and with the reducing rate of increase of vertical strain, that the shaft, which had been turning ahead (outboard) had, in fact, climbed the side of the dry bearing.

Calculations were now made of the theoretical increase in shaft bending moment with bossing deflexion, and while the experimental results showed the shaft moving horizontally after a vertical bossing deflexion of about 1 mm, the calculations showed that for the same deflexion the forward sterntube bearing would be, in fact, unloaded and the shaft free.

Comparison of these measurements with readings of shaft bending moment variation taken at sea, showed that total bossing deflexions over 5 mm were being experienced, corresponding to a maximum stress of $\pm$ 8 tons/in^2 on the inside of the bossing arm.

Cracked liners due to material fault or, more rarely, arduous operating conditions, still account for a number of tailshaft failures.

Another large passenger liner completed in April 1969, reported after only six months' service, excessive weardown on the port aft sternbush. The rate of wear was over ten times as much as on the starboard sternbush, being 0·140 in on the port side as compared with 0·012 in on the starboard side in the same period. Jacking the shafts to measure the reactions on the after plummer bearings, showed, as expected, a significant increase in load on the port aft plummer bearing, and a calculated reduction in forward sternbush load.

By February, 1970, the port after plummer bearing load reached 70 000 lb, whereas that on the starboard after plummer bearing remained at 41 500 lb against originals of 44 500 lb and 43900 lb respectively.

Before drydocking, measurements of tailshaft bending stresses were taken at sea which showed a predominantly first order stress vibration with smaller amplitude changes associated with propeller blade forces. Bossing vibrations were slight. A high frequency stress variation in the port shaft was felt to be due to whirling of the muff coupling between tailshaft and the first intermediate length.

The port tailshaft was drawn. The liner was found fractured in many places in an entirely random pattern, (Fig. 3), and the synthetic resin bush severely overheated and charred thus blocking all water channels. A metallurgical examination showed that the liner had failed under cyclic thermal stressing and that the tailshaft itself was fractured under the liner.

FIG. 3—*Thermal cracking of a liner*

The most likely causes of the bush failure were:

a) Water starvation due to inadvertent shutting of the service stop valve, though both shafts were supplied from a common line, branched through unmistakably clear gauge glasses and any serious water shortage would certainly have damaged the forward end shaft seal;
b) Hygroscopic swelling of the bush material producing insufficient bearing clearance, though both shafts were stated to have had the same initial clearance.

It is certain that the blockage of the stern bush water channels must have occurred early in the ship's life. The continual stopping and starting characteristic of a cruise ship could have induced severe thermal cycling requisite to cause cracking of the liner. If the correct cause is obscure, the moral is obvious; gauge glasses and thermometers etc., however remote from the hub of activity in the comfort of the control room, are not there just for amusement; they are meant to be used intelligently.

GEARING

In 1966 the Society's gearing rules were modified to take account of the greater use of surface-hardened gears and to bring oil-engine driven gears within the rule framework.

Since that time numbers of gearing failures have been investigated by the Society. Two tankers propelled by twin Diesels developing a total of 10 080 bhp, suffered main wheel rim fractures, and whereas a material fault was considered at the time to be the cause of failures, it was decided to obtain a clearer picture of tooth load distribution. Short-base electrical resistance type strain gauges at the forward and after end of the face width in the main wheel tooth root radii showed bending strains as the teeth passed through the port and starboard meshes in succession. Gauges were also placed on the rim to detect meshing strains. Signals from all these gauges within the gear case were brought out by telemetric means. Torque in the port pinion shaft and intermediate shafting, together with the axial movement of the gearcase and shafting, were recorded, as was the radial displacement of the main wheel shaft in its bearings. A static shaft alignment curve was also plotted.

The trials showed that wheel tooth bending stresses at the after end of the main wheel mesh were nearly double those measured forward. The repeated stresses varied cyclically over the least number (5) of main wheel revolutions which corresponded to a whole number of pinion revolutions (18) with a maximum of 13·5 ton/in^2 at the port mesh. These indications were confirmed by the variation in rim stresses, and later measurements revealed some of the high frequency strains that have to be resisted by the shrink fit.

The variations in tooth bending could not be expected basically to increase the possibility of tooth failure, though it is known that irregular cylinder pressures can considerably amplify torsional vibration stresses. The cyclic variation corresponded with the successive inter-meshing of the same combination of teeth on pinion and wheel though whether this was attributable to skewing of the gear elements or to a local meshing error causing an acceleration of the system could not be determined. Axial vibrations were also present which would contribute to the general wear and tear of pinion and shaft assemblies. These factors, together with a 500 Hz tooth contact frequency and the reduced damping of a system carried in ball and roller bearings, would have accounted for the fretting and slackening of pinion bearing securing arrangements found in these gears. This would, in turn, lead to changes in alignment and gear meshing and hence higher tooth stresses, and it was concluded that all these factors taken together with the stress-raising defect found in the material had been sufficient to bring about failure.

This throws considerable responsibility on the gear manufacturer being able to accurately predict the running alignment of gears, having due regard to ship loading and movement of both gearbox and seatings in the hot full torque condition. This is especially so if pinion shaft alignment cannot be adjusted in the usual manner, i.e. by lateral movement of bearing.

Fractured teeth in the main reduction gear wheels of a 15 000 hp motor cargo ship, leading to axial slip of the rim under the helical tooth forces, inspired a detailed investigation (Fig. 4). Alignment of the main wheel to the shafting was satisfactory and local malalignment between the main wheel and the two engine pinions was suspected. In particular, interest was directed toward the movement of the pinions under clutch plate forces. In this case, phased readings

FIG. 4—*Cracked main wheel rim*

of the tooth bending stresses showed the pinion to be "tumbling" under the action of a radial clutch force generated by a slight displacement of the clutch plates when engaging. This could in fact be stopped by de-clutching one pinion, and then re-engaging when both pinions were running at the same speed. Measurements taken at the main wheel tooth roots showed superimposed stresses due to this movement. Re-design of the clutch-pinion bearing supports eliminated the clutch engagement effect and allowed greater control of pinion alignment and hence more uniform tooth loading.

In May 1971, the Society amended its rules to bring the tooth loading of epicyclic gears in line with those for parallel shaft gears. Up to that time the rules legislated only for parallel shaft gears and special consideration was given to epicyclic designs. Experience with one design of epicyclic gear in land installations had shown that gears with tooth bending stresses of up to more than twice those in comparable marine parallel shaft gears gave long and successful service. For the assessment of allowable loading, the planet or star wheel, which is, of course, essentially an idler gear, has its allowable loading reduced by a factor of 1·6 to take account of the reversal of load on the tooth flanks, and the load per unit face width has been taken as the mean of that for the sun and the annulus, which latter, being softer, generally had a reduced face width compared with the sun pinion. Recently, the annulus face width has been increased to parity.

There are currently two main types of steam turbine gearbox involving epicyclics. The original designs had a single primary epicyclic mesh for each turbine branch. Subsequently, some installations with a large overall reduction ratio on account of low propeller rev/min were fitted with a double epicyclic reduction in the H.P. branch, the first reduction generally being a star gear. Most other epicyclic gear meshes are of the planetary type. Examining the proposed tooth loadings for contact stresses, it would seem that under normal operating conditions with nitrided teeth there should be no risk of pitting, even for the sun pinion—planet mesh. The only danger of surface damage in the annulus—planet wheel mesh would be if there was interference between the planet wheel tip and annulus gear root. This could normally only occur if the planet moved outward relative to the annulus and is thus partially dependent upon the flexibility of the annulus and its attachments. To minimize this possibility, increased tip relief has been given to the planet gear teeth.

Towards the middle of 1970 a series of failures commenced in four types of higher powered sets. In the first two types failures were confined to the annulus rings of planetary type units and was mainly in the L.P. first reduction gears. In the second two types fatigue breakages occurred in the teeth of the H.P. first reduction star wheels. As a result of the star wheel failures the Society imposed, in December 1970, a power, and consequently tooth loading restriction on these gears, pending the fitting of stronger H.P. 1 gears of coarser pitch.

In order to understand more of the behaviour of these gears under running conditions, the Society has co-operated with the turbine and gear manufacturers to take measurements of torque in the L.P. quill shaft and the outer coupling ring of the first reduction (planetary) gearing, telemetry techniques being employed. From these measurements it was concluded that the vibratory torques measured should not be responsible for the annulus ring failures, assuming equal torque to be transmitted by each of the planet wheels. With this in mind, measurement of the relative movements of the various gear elements when under operating conditions became important, the radial and torsional stiffness of the axial cross-section of the annulus rings having a direct bearing on the planet wheel tooth contact loads. Further work is planned in this field.

Experiences with these gears is the subject of a paper to be given to the Institute in April, 1972, so will not be examined in detail here.*

COUPLINGS AND CLUTCHES

Sometimes couplings are maligned as well as mal-aligned but it seems the inclusion of a flexible coupling in an engine–gearbox line gives excuse for laxity. A classic example concerned a dry cargo ship with a 5-cylinder 6000 bhp main oil engine driving the line shafting directly at 154 rev/min through a rubber tyre type flexible coupling. During the maiden voyage there was trouble with fuel pump suction pipes necessitating operating the engine on four cylinders for eleven hours. Shortly afterwards the forward tyre of the coupling ruptured. Torsionals were naturally suspected, but on checking the shafting considerable mis-alignment was found which necessitated completely re-chocking the main engine and thrust block, well over half an inch being machined off the main engine forward chocks. Subsequent trials showed satisfactory torsionals at shaft speeds over 50 rev/min on both four and five cylinders.

The importance of careful alignment of any shafting system cannot be over-emphasised, and particular attention should be paid to envisage what will happen in the "hot" full power condition. Many problems arise where, for example, gearboxes rest with one end on a cofferdam and the other on a lubricating oil drain tank, and it is strongly advocated that alignment should be checked "hot" whenever possible.

This situation existed on two sister ships, each with two Pielstick Diesel engines driving a single screwshaft through electro-magnetic couplings and single reduction gears. It was found impossible to set the couplings in the "cold" condition, so as to keep the air gap within the maker's limits when the couplings were energized. In fact, the necessary clearances in the main engine and pedestal bearing rendered it difficult to set up these couplings in the first place. The situation was partially eased by stiffening the self-aligning pedestal bearing, but the Society's final recommendation was the fitting of new bearings throughout with clearances such that the magnetic coupling air gaps could be held within the required limits.

A clutch failure leading to a train of other troubles occurred on a roll-on/roll-off container vessel having a 14 000 bhp 8 cylinder non-reversing oil engine running at 375 rev/min and driving a 3-bladed c.p. propeller through an hydraulically operated friction clutch. The vessel was not to Lloyd's Register Class. See Fig. 5.

During manoeuvring two Diesel-driven alternators supplied electrical power, while on passage power came entirely from the main engine driven alternator.

Automatic disengagement of the main engine-driven alternator from the bus bars when changing to the independent supply was incorporated within the electrical system and led to a temporary black-out, itself overcome automatically.

The clutch, propeller controls and main thrust bearing were housed within a common casing. To engage the clutch, pressure oil was admitted to spaces between the inner cases, and to disengage, this pressure was released and further pressure oil admitted to spaces at opposite ends of the inner cones.

The history of the vessel revealed an inexplicable clutch seizure during the maiden voyage, followed by choking of the

* Jones, T. P. 1972. "Design, Operating Experience and Development Potential of Epicyclic Gears".

main lubricating oil filters, mainly with fine particles, and the whole oil charge contaminated. This trouble recurred, when the main engine crankshaft was found to have moved forward damaging the locating bearing and the timing gear train. The aftermost friction lining of the clutch was worn, while the forward lining was in good condition.

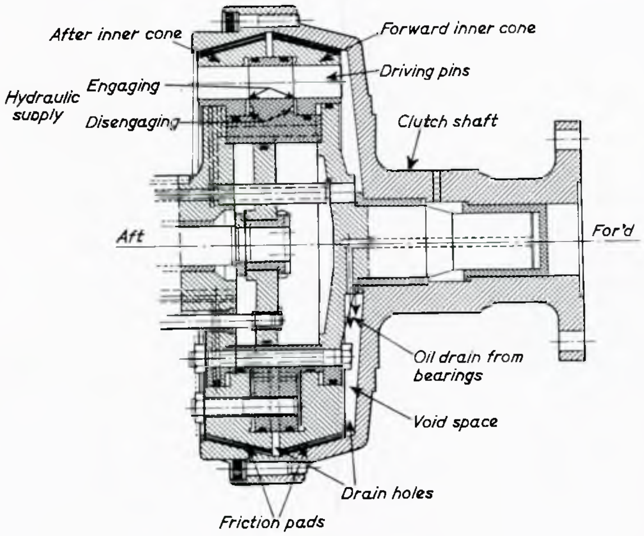


FIG. 5—*Clutch arrangement*

When changing from manoeuvring to passage alternators, the blackout was long enough to cause a marked drop in clutch oil engaging pressure, and automatic re-starting of the oil pump could bring a sudden application of full load torque on the clutch. Since the design was such that any residue of drain oil in the void space in the clutch was at this point thrown out between the forward lining and the clutch shaft, it was apparent that full load torque could be thrown suddenly on the aftermost lining after every blackout. It was apparent that a major re-appraisal both of clutch design and of the electrical system was required.

A catastrophic failure, this time of an H.P. turbine primary pinion forward flange led to fire, totally gutting the engine room and fortunately, no one was seriously hurt. Between the time of building and the failure the flexible coupling had been renewed. At that time a small crack was found in the face of the H.P. pinion coupling flange which was *stated* to have started in the rim of the flange, and now passed down through a bolt hole toward the centre of the flange. In view of the owner's commitments and non-availability of spares, the flange was considered to remain efficient for a limited period. Failure finally occurred after a further $11\frac{1}{2}$ days steaming.

From the outset it was clear that the engine room fire was consequential on the flexible coupling failures, which allowed lubricating oil to spray on the H.P. turbine. The remainder of the significant damage in way of the H.P. turbine and pinion unit was confined to five components. See Fig. 6.

At the instant of failure the watchkeeping engineer succeeded in closing the main stop valve. The sequence of damage was clear. After the initial failure of the pinion shaft at its forward flange, events would have moved rapidly. It seems likely that as failure of the pinion shaft occurred across the final holding area, which was offset from the central axis of the pinion, the pinion flange turned out from the centre axis still restrained by the stiffness of the now distorting flexible coupling. It must be remembered that the fore and aft float of the pinion was restrained by it being a double helical gear, and the float of the H.P. rotor restrained by its thrust pads. Therefore, the pinion flange and coupling sleeve would be prevented from flying out until the rim of the flange had cleared the end of the pinion shaft, and this would account for this shaft end and the pinion flange back-face being hammered. How long this took one cannot say, probably no more than seconds, during which much of the energy of the turbine would have been absorbed in belling out the after turbine bearing.

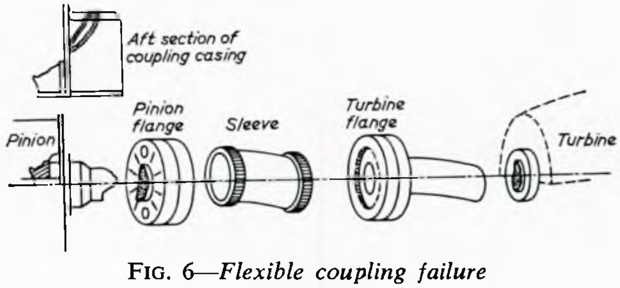


FIG. 6—*Flexible coupling failure*

The pinion flange finally broke on upward swing, coming out and carrying with it the flexible coupling cover. The coupling sleeve appeared to have worked itself out of the turbine coupling ring in two or three revolutions, heavily distorting the teeth on its way. The H.P. rotor was undoubtedly still turning at this moment but must have seized shortly after, causing the whirling coupling flange and shaft to break. On opening the H.P. turbine later, it was seen that at the moment of failure of the rotor shaft, it must have been nearly at standstill as the damage was confined to a small part of the shrouding of the aft two rows of blades.

All the evidence at this stage of the investigation pointed to an inward propagation of the crack in the pinion flange as being the origin of failure.

The metallurgical examination showed not one, but three distinct origins of failure. See Fig. 7:

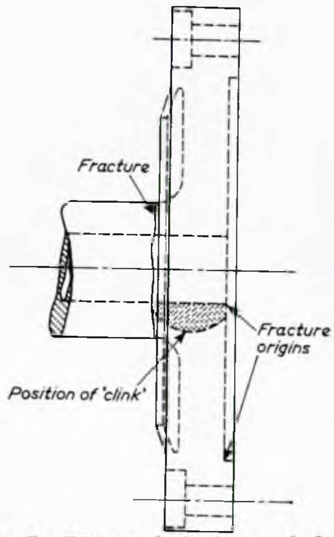


FIG. 7—*Pinion shaft forward flange*

i) Through a point on the lip of the bore and on the face of the flange, that had propagated round the shaft so as to sever it completely;
ii) A second crack having an origin adjacent to a sharp step forming the rim of the flange;
iii) A longitudinal crack extending from the bore surface of the central hole and penetrating the flange radially for a distance of 1·2in. This was a brittle fracture having the appearance of a clink in the original forging, most of which had been removed in the boring operation.

It was concluded that the initial crack propagation probably emanated from this large latent defect in the bore, possibly formed during heat treatment of the pinion forging.

The moral of this failure, would, with hindsight, be that things are not *always* as they seem—a truism which can be applied many times in investigating machinery breakdowns.

CRANKSHAFTS

It is disturbing to realize the incidence of crankshaft failure in the sixties, as a percentage of those at risk, was almost as great as in the fifties. One of the reasons for this may be found in the development of the medium speed Diesel for marine use, where the defect picture would be influenced by teething troubles on this type of engine. Apart from this, we are still faced with sharply radiused fillets, finely angled oil holes and similar stress raisers, all of which contribute to the reduction of the fatigue strength of the crankshaft. The highest incidence rates come, as might be expected, from "slipping" of built-up shafts, and there is recent evidence to suggest that the factors of safety against slip for such shafts are much lower than the theoretical values.

A major breakdown concerned the starboard oil engine of a twin, 14-cylinder, V-engine installation, developing a total of 11 480 bhp. The crankshaft failed without prior warning during normal service, though two weeks earlier oil mist detectors had operated the alarm but no signs of overheating had been found. Damage was confined to cranks 4–11 and 5–12 (from aft), crank 4–11 being fractured across both webs—a most unusual feature. See Fig. 8. Several balance weights were subsequently

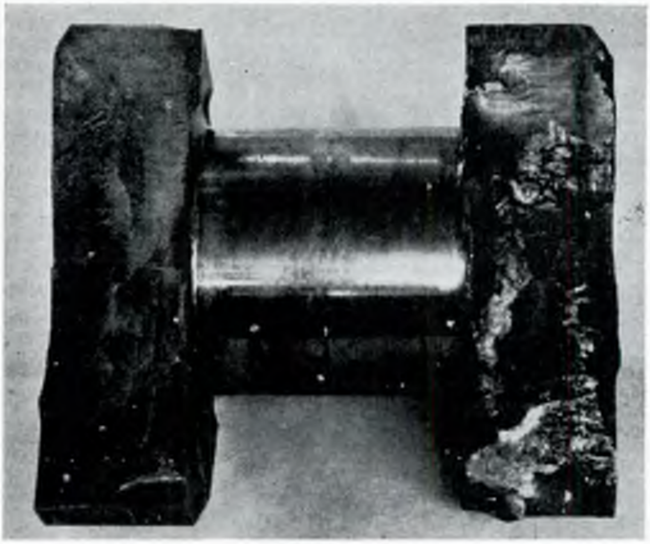
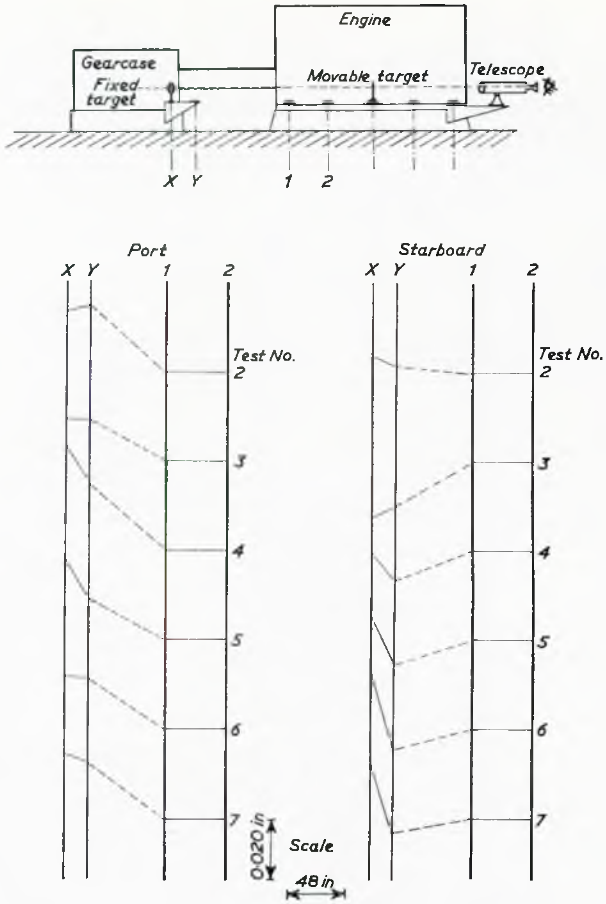


FIG. 8—*Failure across two webs*

found slack, and the crankshaft was seen to have been contacting the top halves of bearings Nos. 1, 2, 3 and 4 (from aft). This suggested misalignment, probably between engine and gearcase. Checking of the port engine showed a lateral discrepancy of 1/32 in due to an error in a coupling aligning jig, and crankshaft deflexions on both engines, had apparently been considerably greater than normal.

The appearance of the surfaces, together with the fact that cracks had been found in another fillet radius, suggested that in addition to normal service loadings the crankshaft had been subjected for some time to a bending moment which had been reasonably uniform over four lengths on each side of the centreline of No. 3 unit; and that the cyclic stresses arising from rotation of the crankshaft under this bending moment were primarily responsible for the fatigue failure.

From extensive alignment measurements using telescopic and proximity transducer methods it was concluded that significant changes in alignment took place not only due to temperature changes in the engine, engine seating and double bottom structure in way of the lubricating oil tank, but also due to changes in cargo distribution in holds and deep tanks. Heating of the double bottom lubricating oil tank alone gave engine to gearbox vertical and transverse alignment changes of 0·030 in. Fig. 9 gives some alignment changes under different loading conditions. It was also found that under certain conditions of engine hog, the journals lifted from the bearings and gave spurious crankshaft deflexions. It was concluded that these large alignment changes



Test	Condition	Draught Forward	Draught Aft
2	Vessel fully loaded, engines cold	18ft 7in	19ft 5in
3	Discharging cargo No.5 D. tank in	13ft 8in	19ft 4in
4	Discharging cargo No.5 D. tank out, No.6 in	13ft 11in	16ft 0in
5	Discharging cargo No.6 D. tank out	14ft 7in	13ft 8in
6	Discharging cargo mainly No.1 hold (forward)	13ft 3in	13ft 3in
7	Discharging cargo mainly No.1 hold (forward)	12ft 2in	14ft 1in

FIG. 9—*Gearcase alignment changes relative to after end of main engine*

FIG. 10a—*Mag × 100 non-etched—embedded impurities in fracture zone*

FIG. 10b—*Point of origin of fatigue fracture*

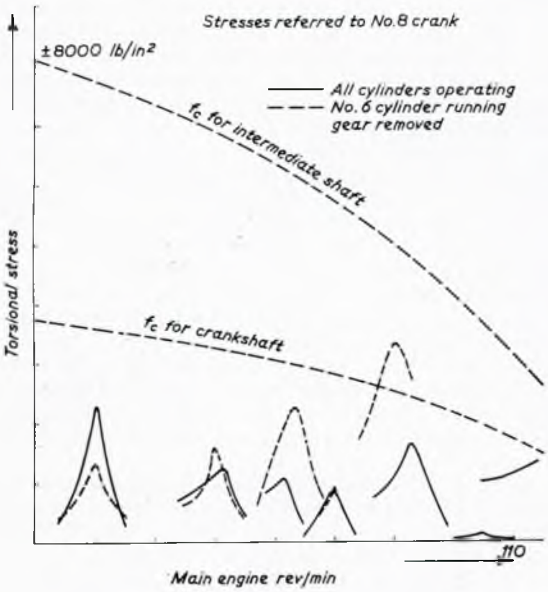


FIG. 11—*Crankshaft torsional stresses with and without No 6 cylinder running gear removed*

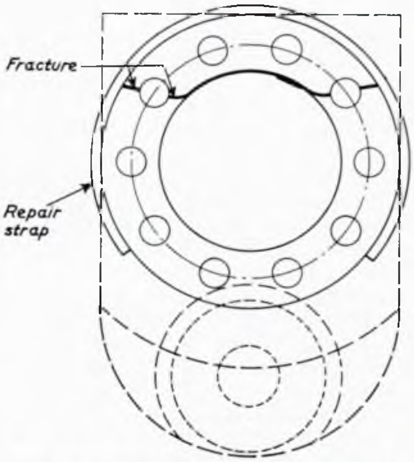


FIG. 12—*Emergency repairs to crankshaft*

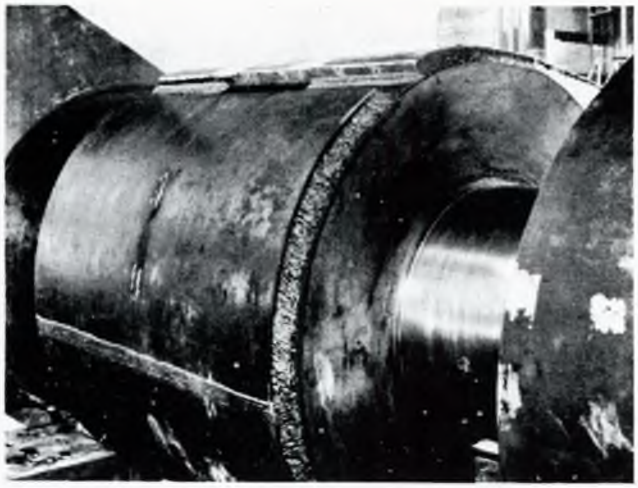


FIG. 13—*Emergency repairs to crankshaft*

would have allowed the crankshaft to run eccentrically, and that this could have been responsible for slackening of the balance weights which had clearly been working on their webs for some time. Torsional vibration was shown not to be associated with the failure.

A 19 000 bhp 9-cylinder oil engine crankshaft failed in the forward web of No. 8 crankthrow after approximately 200 million power strokes. Immediately prior to fracture the engine had been running for 360 h at 90 rev/min with No. 6 cylinder unit entirely "hung-up" (2×10^6 cycles).

Ultrasonic testing during manufacture had shown a number of impurities with embedded flaws in the zone where the fracture started, Fig. 10, but these were, at that stage, not considered to be serious.

Subsequent torsional vibration records, with and without No. 6 cylinder "hung-up", showed the engine had, in fact, been running for 2 million cycles on a 1-node fourth order critical (Fig. 11) and this, coupled with the stress raising effects of the internal flaws, had initiated the fatigue failure of the crankshaft.

Occasionally crankshaft defects are discovered before major damage results, and the investigation department were recently asked to advise on temporary repairs for two such cases. In the first, a crack was found extending round the fillet radius of a six-cylinder crankshaft forward half coupling flange, and passing out to the rim. (Fig. 12). Previous damage had been confined to the slipping of No. 4 forward side rod crankshrink three years earlier. Crankshaft deflexions still seemed reasonable and so a strap was welded round seventy per cent of the coupling flange to contain the cracked portion. This ship was able to return under restricted speed from Lobito to Europe.

In the second case, the No. 7 crankpin of a 9-cylinder engine was found cracked for 300 mm along the forward fillet radius on the underside of the crankpin and out at 45° into the pin for a further 5·9in. It appeared to be a typical fatigue failure initiated under combined bending and torsion. The backs of the adjacent bearing shells showed fretting and scoring, and No. 9 main bearing had been changed four times in two years for cracked white metal. The No. 7 crankpin bearing had been changed three times in 16 months. Crankshaft deflexions appeared normal. The temporary repair (Fig. 13) was to fit a 1·7in thick steel plate, which had been rolled to fit the No. 7 crankthrow and then cut into four pieces. These, after "hanging-up" No. 7 unit, were welded entirely around the throw and enabled the vessel to sail from Karachi to Cadiz for permanent repairs.

A rather puzzling failure concerned a small twin 9-cylinder oil engine, in a twin screw car ferry. On the maiden voyage minor teething troubles occurred with one or two broken holding-down bolts and studs around the main engines, but they produced a satisfactory speed of 1200 rev/min. The line shafts were driven through flexible couplings and combined clutch/reduction gearboxes. Shortly after entering service, and 3 months after delivery, the starboard engine crankshaft broke through the oil hole in No. 8 journal.

On dismantling the engine, cracks running at 45° to the axis of the shaft were found to pass through the oil holes of two other journals, and torsionals and misalignment were suspected.

During dismantling it was also noted that the rubber of the flexible coupling was cracked and the flange bolt holes on both driven and driving slide elongated and also that the flange between the engine and short intermediate shaft was fretted.

Misalignment, coupled with the overhang of a rather heavy flywheel was uppermost in most people's minds and the engine builders sought to measure the bending stresses in the intermediate shaft on a similar installation.

Torsional calculations were verified and found satisfactory. By this time the port crankshaft was also found fractured in similar fashion in three journals.

At this juncture the two most important points to note were:

1) That both shafts had failed in three, but not entirely the same three, journals;
2) The fractures ran through the oil holes and, on the one completely broken section, the crack origin ran from the bottom of the radius of the oil hole where it became a parallel bore (Fig. 14).

On examining the two replacement crankshafts by magnetic crack detection and with intrascope inspection of the oil hole bores, one of these shafts was rejected for a crack beginning at the root of a journal oil hole radius and extending 0·12in down the bore.

This, and the discovery of further cracks in shafts held in stock, threw new light on the whole problem. The shafts had a U.T.S. of 57·1 tons/in² and, after final machining including total finishing of the oil holes, were induction hardened on the pins and journals to a depth of 0·12in. It was the practice in this process to close the oil hole entry with a copper plug which was then hammered over to fill the radiused area so as to avoid sharp thermal gradients. This was clearly not successful, as metallurgical examination of some of the other cracks showed the origin to be at the point where the hardened zone met the oil hole radius.

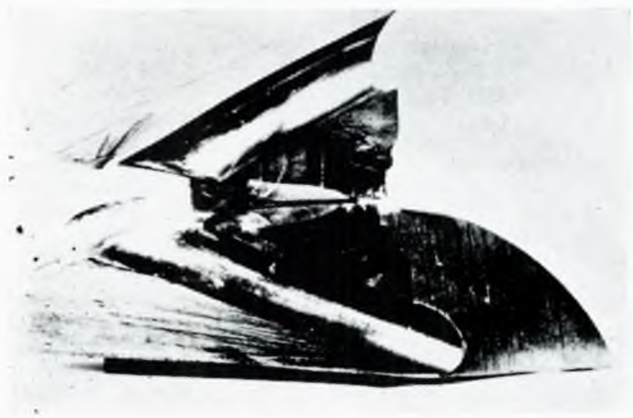


FIG. 14—*Fracture at crankshaft oil hole*

The forgemasters modified the hardening technique for these shafts and subsequent stress measurements taken when the vessel returned to service showed normal torsional and bending stresses in the crankshaft.

HOLDING-DOWN BOLTS AND CHOCKS

In his previous paper the late Mr. Pemberton laid particular stress on the failure of threaded connexions, and there are signs that a lesson has been learnt. This is no reason for congratulation, however, as the incidence rate of failure from bolts as prime cause should be near to zero. Incorrect fits and inadequate pre-stressing remain predominant causes of failure, and it is disturbing to find in how many cases no guidance is given as to the setting-up stress required in bolts holding valuable rotating and reciprocating machinery to its foundations.

A large 6-cylinder oil engine installation developing nearly 14 000 bhp had suffered from repeated loosening of main engine holding-down bolts and chocks.

Several attempts had been made by the owners to remedy the trouble, including:

a) Fitting moment compensators (second order harmonic-balancers) at each end of engine;
b) Fitting an axial vibration damper to the forward end of the engine;
c) Fitting integral chocks common to inboard and outboard bolts at the forward end of the engine.

The chocks, which were tapered inboard to outboard by design, were also tapered fore and aft, the forward chocks being ¼ in thick and those at the after end ¾ in thick. It would seem the move to integral chocks was to prevent them rotating on the seating to a slack position.

Measurements were made of engine vibration, the stress in a holding-down bolt and the relative athwartship movement between the port and starboard sides of the engine seating just forward of the engine. Readings were taken at various speeds with the ship in ballast and various loaded conditions.

The results showed that while the level of vibration round the engine was appreciable, it could not be considered excessive.

The strain gauged holding-down bolt was tightened by applying an hydraulic pressure of 3·9 tons/in² to the nut, which was then torqued by hand using the recommended toggle bar. The static results showed the bolt was not being adequately tightened. Even with the correct procedure, the stress in the bolt after release of pressure varied from 7670 to 1730 lb/in². The dynamic tensile stresses in the bolt showed a first order variation, nearly sinusoidal, of ± 1700 lb/in² on a static stress of 6700 lb/in². At the lowest level dynamic effects could cause the bolt to become slack and allow the chock to rotate. It seemed that this loss of stress on release of tightening pressure was attributable to the worn and badly fitted surfaces between chocks and seating. Spherically seated washers could not have helped in this case as, on release of tightening pressure, they would have tended to move and re-distribute the interface pressure, producing a relaxation in bolt stress. Individual chocks were haphazardly rotationally displaced, thus nullifying the two-way taper of the original fitting and making for point loading of the chock.

Complete re-surfacing of the tank top and bedplate flanges and fitting of substantially parallel chocks were recommended.

Continuous failure of holding-down bolts of the main engine of a fishing vessel was found to be solely due to poor design and bad practice. This vessel, not to the Society's class, had main engine seatings which formed an integral part of the tank top with the result that the bolts passed through the seating to the bunder tank. A cofferdam had been built round the bolts, but openings into the cofferdam were provided in isolated places only, and led to a mixture of through bolts and studs being used to secure the engine. The through bolts, where used were threaded at each end, with one nut on the lower end backed by a split pin. In some cases there was a gap between the back of the nut and the pin, allowing the nut to slacken back. A lock-nut was fitted on the upper end only.

The studs did not penetrate to the full depth of the seating and there was no landing shoulder for the stud at the tank top, which could lead to a bending moment acting on the stud as it was tightened down. The holding-down bolts (studs) were fitted in pairs, each bolt having its own chock. There were only four fitted bolts—bunched together in the fourth and fifth chocks from aft on the starboard side of the engine.

The conclusions were obvious, and an improved bolt design and method of securing were recommended. Further the difficulty of accurately fitting two chocks within 7·9in meant that, of each pair, invariably only one chock was truly tight.

The concept that a long enough stud is bound to be strong enough is dangerous thinking when it is realized that the run-out of the threads to the shank particularly without a stress relieving groove at this point, will introduce stress concentration which can more than double the nominal stress level.

TURBINES

Turbine problems, apart from isolated cases of unbalancing "whirling" of H.P. rotors, have been largely confined to blading. The unfortunate blade vibration problems in *Queen Elizabeth 2*[2] have already been well presented to the Institute, and much has been learned from these failures.

On another ship, blade failure on the final stage of a rotor was traced to the binding wire. The blades were of chromium stainless steel while the wire was non-magnetic austenitic stainless steel. Metallurgical tests showed that cracks had originated both at sharp notches at the blade/wire junction and at "pits" in the steel and had propagated into the wire and the brazing metal (Fig. 15). They were associated with poor adhesion and flaws in the brazing metal considered due to the use of an austenitic stainless binding wire. There had also been insufficient heat to keep the brazing metal liquid until it had filled the capillary space. Changing to a stainless iron wire was recommended.

Priming during a change of steaming conditions was concluded responsible for thermal bend in an H.P. rotor of a 15 000 shp set, leading to hammering out of the forward bearing and consequential damage to the Curtis wheel and glands. If the centre of gravity of this rotor had been displaced by as little as 0·004 in then at a propeller shaft speed of 40 rev/min the centrifugal force would have been one and a quarter times the weight of the rotor which would have tended to lift and drop back into its bearings. During the period of heavy vibration the vertical flange bolts of the forward bearing keep had slackened and the keep tilted. The whipping rotor, now running in greatly enlarged clearances at some 3300 rev/min (the turbine was not immediately shut down) led to loss of clearance between nozzle box and the Curtis wheel shrouding which was "machined" away, and this would have further aggravated the thermal bend.

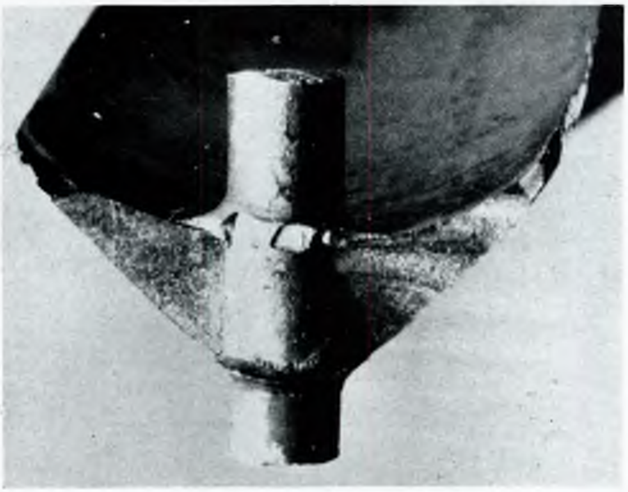


FIG. 15—*Blade failure in way of blading wire*

The fact that this rotor was apparently allowed to run for a time with heavy vibration seems a strong argument for some form of vibration monitoring system being a standard fitting on rotors. Where the control room is remote from the turbine flat and, in particular, sound-proofed, the author is concerned that the "intuitive feel" of the onset of rotor vibration is no longer readily available to the watchkeepers.

Failure of an H.P. turbine rotor due to fretting and bending fatigue occurred on a tanker. The H.P. rotor was originally fitted with a removable thrust collar. This being found slack on trials, was replaced by a shrunk-on collar with a keyed connexion. The thrust pads were carried in a spherical cage. The fracture, which occurred within the shrink, appeared to be due to fatigue under the influence of rotating bending forces, propagating from a point of stress concentration due to fretting corrosion. While it was certain that the theoretical bending stresses on the collar due to steam thrust were small, the source of the damaging stresses remained obscure. There was a possibility that the spherical thrust pad support cage could have "locked" misaligned, though subsequent tests showed it to move smoothly. The forward end supports, too, distorted under heat, though again the extent of distortion was uncertain. Repairs involved machining back the forward end of the rotor and shrinking on a new forward end with an integral thrust collar.

BOILERS

With the gradual passing of the Scotch boiler, major failures in boilers seem to be decreasing, though defects in water-tube boilers have maintained the same level since 1947.

In the years 1966 to 1969, 135 main boilers were fitted in classed ships, and up to 1970, 73 of these boilers had suffered repeated defects. The greatest number of reported defects were to tubes, mountings and superheaters (15–20 per cent of defects) with 9 per cent of defects attributable to drums and their welded attachments and 2 per cent to headers. Feed systems accounted for 7 per cent of defects.

An interesting water tube boiler failure on a tanker concerned a fracture in a main boiler steam drum tube ligament. The boiler normally operated at 1070 lb/in^2. The boiler was shut down to enable leaking joints on the economizer header doors and on some mountings to be repaired. During hydraulic test on completion, one of the after side wall/roof tubes was leaking in way of the steam drum. On close examination the drum was found fractured along the tube ligament between adjacent tubes. (Fig. 16.)

It was seen that the tube had not been fully expanded in the drum, there being a visible clearance on the external surfaces of the drum between the tube and its hole, extending around 50 per cent of the tube. The boiler history showed this tube had been removed some 3 months previously for access purposes, and there was then some difficulty in sealing the new tube. The tube holes were themselves good, though the mid-position groove in the holes had rather sharp corners.

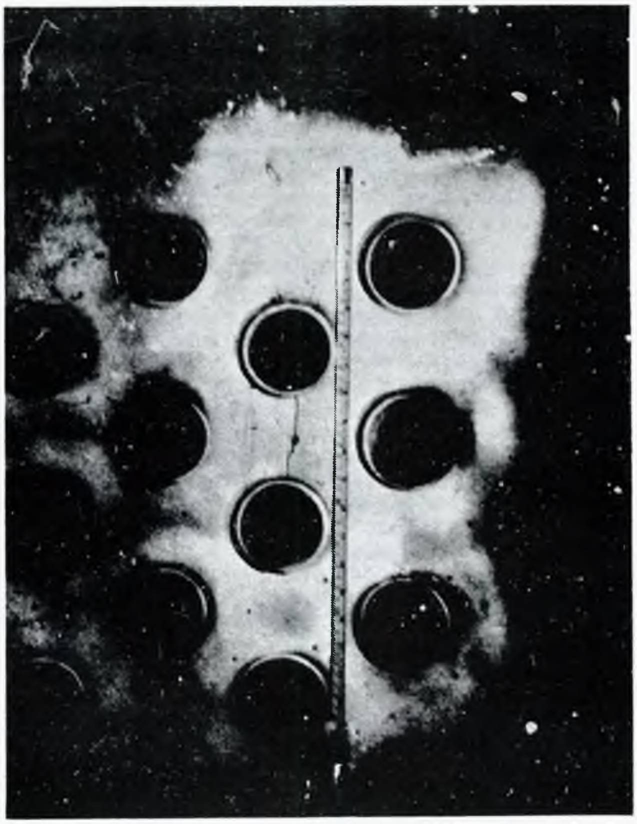


FIG. 16—*Ligament crack in a main steam drum*

Metallurgical examination of chippings taken out of the drum and containing the crack showed it to be a transcrystalline fatigue crack. It appeared that the failure had been initiated by overstressing of the drum material due to excessive pressure when rolling with the expander. The stress concentration induced at the edge of the holes when combined with the stress due to service conditions and the stress due to the inevitable work-hardening round the holes, could produce total stresses approaching the yield point of the material.

Repairs were effected to the ligament by cutting out the crack and welding under rigidly controlled conditions, the vessel subsequently returning to service.

A tragic accident associated with an exhaust gas economizer occurred recently on a ship where the economizer acted also as the silencer, there being no by-pass in the main engine exhaust system. To complete a survey, safety valves required adjustment under steam but as these had been temporarily removed for repair, the surveyor placed a subject against class that the boiler was not to be used until the valves were refitted and tested.

There was a change of chief engineers on board and the new chief decided to take the opportunity to put an hydraulic test on the economizer and the safety valve chests were blanked off for this purpose. Unfortunately, these blanks were not removed after the test, nor was care taken to ensure the economizer was properly drained, and some water was left behind. A few hours after sailing this boiler exploded up through the bridge causing a fatality.

An interesting vibration problem concerned certain rows of generating tubes of an auxiliary boiler which were always leaking. The boiler, as a whole was steady, but the tubes were moving 0·06 in peak to peak at fourth and eighth order of main shaft speed. Clipping the tubes so as to alter their natural frequency cured this leakage.

STEAM ENGINES

Damage to steam reciprocating engines comes less to our notice now than in the past. Troubles still occur with piston seizures resulting in cracking of the cylinder wall, generally attributable to inadequate lubrication, particularly, as in one case, where the vessel had been running at slow speed while anchored in adverse weather conditions. Under these conditions with low steam flow the turbulence in the L.P. cylinder may be insufficient to distribute the lubricating oil effectively, and with normal no-load back pressures the steam will be much drier. Cylinder lubricators of a type which use live steam to inject oil as an atomized vapour into the steam spaces would be an advantage in such cases.

MISCELLANEOUS

There are still a large number of defects which arise through neglect of elementary precautions either in practice or in design. Heavily wiped and damaged crosshead bearings of a large Diesel engine during sea trials were attributed to metallic and sand particles circulated with the lubricating oil. Several of the filter elements were found to be not fully adjusted, allowing a small quantity of oil to by-pass the filter. In addition, the lubricating oil low level alarm had not been installed and the engine had been running with low sump level. Inspection of the sump tank revealed several kilos of rusted steel deposits, rags and casting sand, some of which were clearly carried through the system.

Rotor spiders of Diesel driven alternators often fail by fatigue along the axial edge of the spider arm/shaft welds. Fluctuating stresses induced by torsional vibrations though usually themselves acceptable, are magnified by the stress concentration effects of the welds. Design of spider arm connexions should take cognisance of these stresses in addition to the normal steady driving forces.

Heavy loading associated with the fuel pump operation of some engines has led to severe pitting on gear teeth in the camshaft drive. Timing is usually such that the loads consistently fall on certain teeth, and in some cases, with the increase in bearing clearance, gear hammer occurs. See Fig. 17.

Pitting of crankshaft journals in a group of generators was traced to local high frequency vibration of the supporting platform. The damage appeared to be confined to whichever set was lying idle, and it was concluded that this was in fact fretting between the oil dry surfaces of journals and bearings.

EXPLOSIONS

A problem, unfortunately still very much in reality, is that of explosions, both of crankcases and elsewhere. Although modern monitoring equipment has greatly lessened the chance of these occurring by enabling machinery to be stopped before critical temperatures are reached, there are still cases where this would not have been the total answer to the problem.

One such case involved a 4000 hp motor vessel where, due to heavy weather, the no-volt trip on the only generator in use was thrown out, totally blacking-out the ship. On re-starting the engine, black smoke was seen coming from the scavenge ring blower and a crankcase explosion soon followed.

FIG. 17—*Selective pitting on camshaft drive gears*

Examination showed one of the blower idler gear wheel bearings to have seized and rotated in the wheel housing thus blocking off the oil supply. It seemed that at the blackout the electrically driven lubricating oil, fresh water and salt water cooling pumps had stopped but the main engine ran for a further four minutes, during which the idler bearing was starved of oil. On re-starting the engine, this bearing rapidly overheated and the explosion ensued.

Another explosion took place in the pump room of a tanker during discharging. A major factor in initiating this explosion was the failure of one of the cargo pump bearings which led to the wear rings acting as bearings lubricated by crude oil. Evidence showed that this condition had existed for some time, but with loss of flow through the pump, excessive heat was generated to cause a breakdown of lubricant at the wear rings.

Further, omission to strictly follow the makers' instructions resulted in this pump failing to prime, which would account for there being the air necessary to support an explosion. Examination of the other cargo pumps revealed incipient signs of similar bearing troubles.

CONCLUSIONS

A catalogue of failures can make depressing reading, especially if one is aware that the examples described no more than touch the surface of the whole problem. Yet, if there is one profound truth that comes from the cast-off pieces of tortured metal that lie in the wake of failures in service, it is one that we should be eager to learn—and then to start all over again.

The plain fact is that many of the failures should simply not have occurred. When working not only with metal but with people, although in 99 per cent of the cases no one is hurt, every failure is a potential killer.

One cannot afford to dismiss failures in design or fabrication as accidents, but should regard them as short-comings in human effort, resolving that, so far as it lies within ones power, they shall not recur.

ACKNOWLEDGEMENTS

The author wishes to express his thanks to many people in the shipping industry who have talked freely about their experiences and in particular to colleagues and members, past and present, of the technical investigation department of Lloyd's Register of Shipping.

REFERENCES

1) Toms, A. E. and Martyn, D. K. "Whirling of Line Shafting" November 1971 Vibration Symposium *I. Mar.E.*

2) Fleeting R. and Coats, R. 1970 "Blade Failures in the H.P. Turbines of R.M.S. *Queen Elizabeth 2* and their Rectification. *Trans. I. Mar. E,* Vol. 82 p. 49.

Discussion

Mr. G. McNee, B.Sc., Member of Council, I.Mar.E., said that when he had been invited to open the discussion on the paper the thought had occurred to him that a marine engineer must have a certain masochistic streak in his make up. He spent the working day on engineering problems of various types, but this was not enough, he must go to the Institute in the evening to hear about the problems of others. There were two reasons for this. One was that it was comforting to realize that other people were having problems too; and the other was to learn from their experience and, what was more important, to take action before it was too late.

The author was right when he drew attention to those people who could benefit from a paper on marine machinery failures, and it was a pity that more such papers were not presented for discussion. Designers should learn from the information given, but unfortunately, as could too often be seen in the end product, the lessons had not been learnt.

The author had started his paper with propellers and shafting. The controllable pitch propeller was a very good instance of the dilemma that confronted the engineer who had to make the decision as to its use. On the one hand there was the normal propeller which, unless it received contact damage, could in most cases be expected to last for the life of the ship; on the other, there was the c.p. propeller, working in troublesome conditions, which could not be repaired except in dry dock where conditions for cleanliness were poor, particularly for some of the intricate control gear. No matter how good the maintenance system, the c.p. propeller must fail on occasion. In making the decision, some assessment of possible down time due to fault had to be made, and in this the author had tried to give guidance by showing an incidence of failure of 11·1 defects per 100 years. Unfortunately, through no fault of the author as the full information was not at his disposal, this was not sufficient.

In the two examples of failures given, the first was a fault which could cause little interruption in service, whereas the second could reach disastrous proportions as not only the owner, but many other parties could be involved, with consequent escalation of cost. These examples were sufficient to show that the frequency of failure figures by themselves were not enough and what was really required was a total cost of the disorganization caused by the number of failures. Due to the workings of the industry, it was unlikely that figures of this nature would ever be produced, but it could be seen how much easier the engineer's job would be if communications within the industry were better. Two of the examples shown under "Main Shafting and Stern Gear" also indicated the same lack of communication.

Some twenty years ago, in Mr. McNee's company, similar cavitation damage had been found on tailshaft liners in twin screw ships which had been cured by altering the practice of never rewooding the forward sterntube bush, which had not been thought to be necessary, to rewooding at regular intervals. The second example went even further back. The possibility of core plugs in tailshaft liners coming out or slackening had been well known in his company before he joined it, as specifications going back to before World War II specifically prohibited their use. These examples, he said, added emphasis to the author's remarks. After ten years of progress engineers had to contend with largely the same problems. It was indeed, as the author said, a salutary thought.

Another field where communications were ineffective was between user and designer, but in this case the fault was not entirely with the user, as on many occasions, although experience was proffered or faults were brought to the attention of manufacturers, there were times when there seemed to be a marked reluctance to change.

The author mentioned alignment, another frequent cause of trouble, as every user knew. At the present time the manufacturer was so intent on reducing the size and increasing the speed of his particular product, in order to reduce the rate of price increase, that he landed the user with some difficult alignment problems which could not be solved without the expenditure of much time and money. Yet these problems had to be solved, otherwise there was the onset of fatigue, with perhaps failure, and an even heavier expenditure to renew broken parts. The author had given some good examples, there were plenty of others, but there was no need to add to the depressing picture.

At the beginning of the section on holding down bolts and chocks, the author had referred to the stress Mr. Pemberton put on the failure of threaded connexions. It was suggested, however, that there might be a certain amount of wishful thinking in the statement that "there are signs that a lesson has been learnt". Screwed connexions were still a source of failure, and most of them could be traced to bad workmanship. A connected subject was fillets (Fig. 7 was a very good example of this), and one of the difficulties here was that even when the designer had provided nicely rounded fillets they seemed to disappear between drawing office and shop floor when the turner used a sharp nosed tool. If manufacturers could give more attention to inspection many unnecessary failures could be averted.

In a paper of such interest it was tempting for the opening speaker to comment on all the most interesting points. He had resisted the temptation, but would very strongly emphasize the important points the author had made, as had the late Mr. Pemberton before him, on the value of service experience. Without it no designer, manufacturer, or operator could work efficiently. Design should not be based on design, but on the service experience of design.

All engineers had to learn the lessons, and sometimes they could be salutary, which experience taught. For this, the prime requirement was a free and open communication of the type which the author had provided through his paper.

MR. J. A. DUNCAN, M.I.Mar.E., said that most of the failures described in the paper were of a "one off" character, as distinct from a repetitive or persistent nature, and their various causes were traceable to faulty design or manufacture (including defective materials), or to negligent operation of the machinery in question. Most of them could be described as severe and most had been expensive in terms of repairs and time. At the end of the paper the author had said that many of the failures should not have occurred, and he agreed, but the question was how were they to prevent them occurring. Everyone knew the difficulties of trying to provide for every eventuality. He would be interested to hear the author expand his views on this matter.

In the section "Propellers, Shafting and Stern Gear", the author related the incidence of failures in a number of controllable pitch propellers to a period of 100 years of service. He noted that none of the propellers in question had a life of more than ten years. Although the total number of defects was given as 40, it was not possible to work back to determine the total number of propellers involved, as there was no information on whether any propeller had more than one defect. He had seen this method applied to other components and had always found it difficult to understand. With one or two notable exceptions, no ship could expect to have a life of 100 years, and therefore to take such a period of time as a standard of comparison seemed misleading. As far as he could see (and he admitted that his understanding of this might be deficient) it did not clearly tell one what degree of risk of failure would be associated with such a type of propeller if chosen for a particular ship with a normal life expectancy of 20 to 25 years. He would be grateful, therefore, if the author would explain why he used this method of evaluating failure rates, and how he related his result to normal ship life expectancy.

Referring to the first twin screw shaft alignment problem described in the section "Main Shafting and Stern Gear", the author had stated that it had been deduced that deflexions in the bossing arms were in excess of 5 mm, giving rise to stresses of ± 8 tons on the inside of the bossing arm. Earlier in the report of this case the author had referred to possible displacement of the shafting of the order of 0·165 in. There, unfortunately, with this brief information, the author had left them, like the shafts, in intolerable suspense. Could Mr. Batten tell them what view was taken of these flexings and what action, if any, had been considered and taken.

Mr. Duncan said it was remarkable how much abuse machinery would tolerate at times without failure. Many years ago, he had been involved in repairs to the starboard shafting system in a twin screw oil tanker. In this case there had been no sudden major failure, minor trouble had persisted for a number of years, mainly with the stern gland and to a lesser extent with coupling bolts which tended to work slack. The vessel had been brought to dry dock for annual repairs and the opportunity was taken to deal with the stern gland trouble, the tailshaft being drawn for this purpose. The condition of the tailshaft and the various parts showed no defects which would account for the trouble, so the alignment of the shafting from the engine coupling to the outside of the stern tube had been checked. This had been done in the first place by centring the tailshaft in the sterntube by means of closely fitting rings at each end, and then aligning the thrust shaft and intermediate shaft couplings parallel, coupling to coupling, from the engine to the tailshaft, with the shafts free to turn on rollers. When the shafts had been set up as described, it had been found that the intermediate shaft after coupling flange had been approximately $\frac{3}{8}$ in high to the tailshaft coupling flange. Two courses of action were open—rechock the main engine or refit the stern tube. The latter course, after discussion, had been chosen. Before drawing the sterntube, the shafting had again been aligned through it, in true line (the neck and stern bushes being drawn to permit this). This had been done as a check, and also to enable proof marks to be made on the outside of the boss of the A bracket. Fig. 18 illustrated the condition when the shafting had been set up as described. Repairs had consisted of drawing the sterntube, welding in a new piece of plate in the after peak bulkhead, welding up the landing in the eye of the "A" bracket to a sufficient amount to permit correct centring of the tube, and then re-boring all on the correct centre line. Thereafter, the sterntube, tailshaft etc, were all re-fitted.

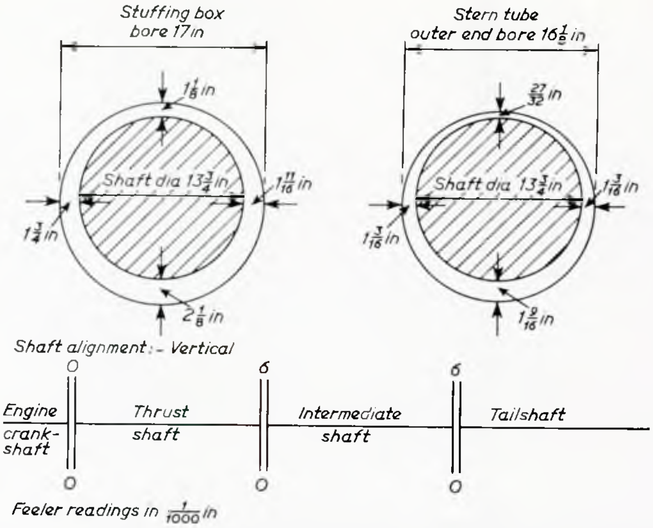


FIG. 18—*Starboard shaft alignment—Intermediate shaft and tailshaft lined up on rollers through sterntube*

As to the cause, the only event traced in the ship's history which could credibly be held responsible had been a major re-rivetting repair, about 20 years previously, to the engine bedplate supporting girders, built on the tanktop. At that time the engine had been removed completely to permit this reconstruction, and had subsequently been re-chocked. It was conceivable that the misalignment had been established then.

COMMANDER P. D. V. WEAVING, R.N., M.I.Mar.E., said that, as one who had been intimately concerned with the epicyclic gears mentioned, he agreed entirely with the author that shortcomings in human effort had played a part in the difficulties they had experienced. In addition, nearly 300 high power epicyclic gears were giving completely satisfactory service in over 100 large ships and some of these gears had been in service for six years. These comprised about 40 different designs.

The author had mentioned that trouble had occurred in two of the H.P.1 gear designs fitted in the triple-reduction gear arrangement. Some members might not be familiar with this arrangement, which was shown diagrammatically in Fig. 19. It would be noted that the H.P.1 star gear was connected to the H.P. turbine through a comparatively short quill shaft with a gear type flexible coupling at each end.

Fig. 20 showed an actual H.P.1 and H.P.2 gear of about 17 000 hp.

It was important that gears were of adequate strength and that the nominal tooth bending stresses should not be too high, also that other factors affecting reliability should receive the same attention. The author had referred to the need for good alignment when describing a parallel shaft gear failure, and also under the heading "Couplings and Clutches". His remarks were equally applicable to epicyclic gear installations. Failures in two particular designs of H.P.1 gears had led to the introduction of the power restriction mentioned in the paper. One of the designs had been fitted in only two ships. New, stronger gears had been made, back-to-back tested at full load and fitted in the ships within eight months of the power restriction.

In the case of the other design, many more ships had been involved, and the power restriction of 5–10 per cent was being removed as and when the new, stronger gears were fitted.

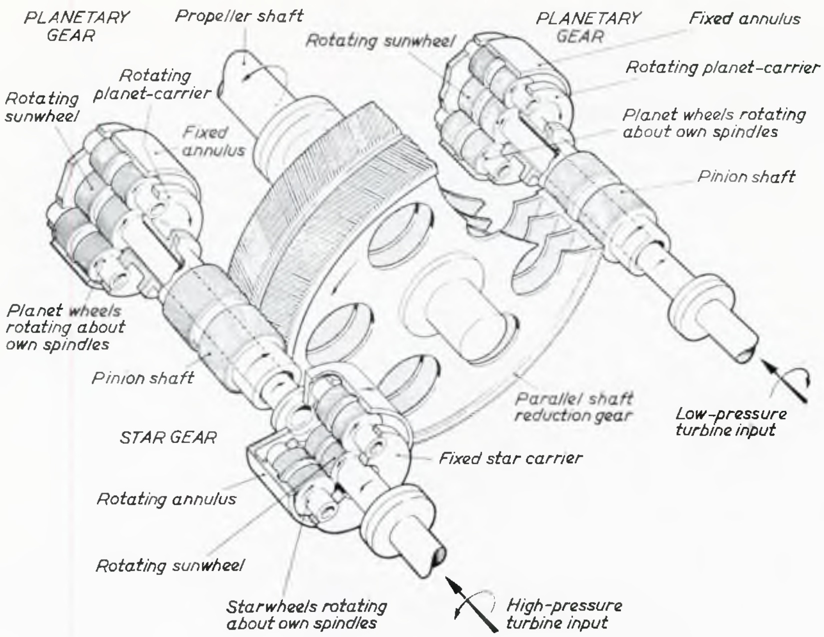


FIG. 19—*Triple-reduction gear arrangement*

In view of the large numbers of epicyclic gears now being used it seemed logical and convenient that Lloyd's Register should have one set of rules for both epicyclic and parallel shaft gears.

MR. J. MCAFEE, Vice-President. I.Mar.E., said that they received many papers at the Institute on new engines, but very few papers on break-downs. How refreshing it would be if at some time some engine builder would come and tell the Institute what really happened to the engine that he had so vividly described to them five or ten years earlier. He only knew one engine builder who had had the courage to do this, and it had been a most refreshing experience to hear him.

As the author had said, an open mind could not be over emphasized, when dealing with investigations of any kind. During the afternoon he had turned over a notebook of his in which he had jotted down cases over the years and had come across a case where a ship had been towed 1000 miles or more because it had been believed that the propeller was about to drop off. When the ship had finally been docked it had been found that the knock that everyone had heard had been in the turning gear. He was quite sure that Mr. Batten and his colleagues would have diagnosed better than that.

He said he would also like to think that in at least two cases in the paper, what he always considered to be the first law of machine design had been very clearly illustrated; this was "make the radius as large as possible". This should be written up in every workshop and drawing office.

With regard to troubles with screw shaft liners, over twenty years ago he had investigated two sister vessels suffering from

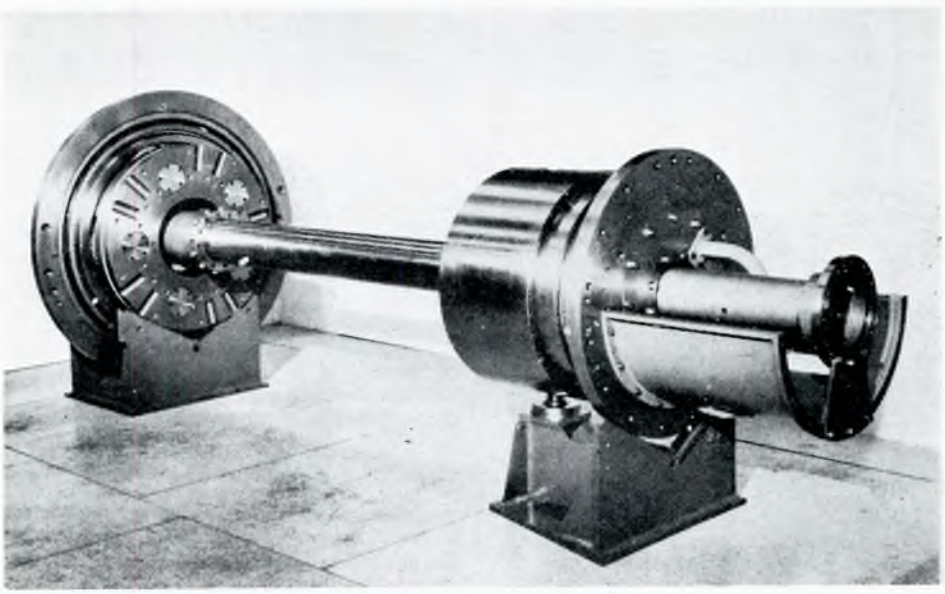


FIG. 20—*H.P.1 and H.P.2 gear of about 17 000 hp*

rapid cavitation wear of shaft liners and bushes. The erosion of the liners had been in clearly defined longitudinal strips, one per blade, indicating that the whirl was propeller excited. Change of propellers had cured the trouble. It might be that with the increased aperture clearances now prescribed, propeller excited whirling was no longer probable.

The author had referred to tailshaft failure arising from "arduous operating conditions". This seemed to imply something outside normal conditions. What, in fact, had he in mind? Mr. McAfee still thought that the stern gear was the weakest part of the ship.

Many years ago he had made an evaluation of screw shaft defects from records kept over a period of about 30 years, and had found that each year over 10 per cent of all shafts examined had been replaced for one reason or another. This was a very high rate, and clearly indicated the weakness of the stern gear. He would like to think that the situation had improved but suspected that whilst there had been factors, such as better inspection, better aperture clearances etc., which had probably tended to improve the situation, there had been other factors which had operated adversely, such as increased speeds, powers and weights of propellers, often combined with less rigid after bodies. He would be interested to have Mr. Batten's views on this.

Some startling incidents with new ships over recent years, including those mentioned by the author, still gave the impression that the stern gear was the most vulnerable part. The reference to the need for taking into account alignment of machinery in the full power hot condition was topical in view of the ingenious chocking system recently described for the new Doxford engine*.

With regard to built-up crankshafts, the author had suggested that the factor of safety against slip might be lower than theory predicted. Was it not a fact, however, that mechanical damage apart, when slip occurred the usual reason had been that the correct shrinkage allowances had not been made during construction? The gauging of these allowances in manufacture required a special expertise not possessed by the average superintendent or surveyor.

In spite of his opening remarks about the size of the radius, it would seem that the crack in the crankshaft shown in Fig. 12 originated not in the fillet but in the bolt holes. Undersized reamers and oversized bolts driven home by strength alone must be the cause of many similar failures.

Mr. P. Mack, M.I.Mar.E., said that re-appraisal of the basis of design was governed by financial and commercial considerations which sometimes dictated a compromise rather than a complete revision. This was certainly the case with existing machinery where owners often had to resort to a palliative rather than a permanent cure. The author had rightly pointed out that it was very necessary to keep an open mind when dealing with casualty investigations, and it was with apologies to hindsight that he said he would like to amplify briefly three of the subject failures.

Mr. McNee and Mr. Duncan had referred to a statistical analysis based on the costs of machinery failures, and it was very difficult to give an analysis of failures when one had to consider costs, because so often costing of a failure was not related to an actual failure but to the consequences of that failure. This was particularly relevant in the case of the well known c.p. propeller failure, discussed in the paper, which was attributed to fouling of the control system strangling orifices. There had, however, been many unrecorded instances of pitch changing failures which had come about for many different reasons. These were not heard of because the consequences were not catastrophic. These incidents generally occurred during manoeuvring in confined waters when the Master's attention was concentrated on the ship's movement and heading. At each movement of the bridge control lever, therefore, it was generally not the practice to observe that the pitch indicator had responded as required. In the event of derangement of the control system, the ship might gather way in the wrong direction, and critical time might therefore be lost before the master appreciated the situation. The engine room watchkeeper on the other hand, was unable to determine the pitch response in relation to the bridge demand because he was not provided with a repeat of the bridge control signal. It was a surprising fact that in the majority, if not all, c.p. propeller systems, that an audible bridge/engine room wrong way alarm was not provided. In most cases a duplex gauge could monitor the bridge control and propeller pitch feed back signals and fulfil this long established and essential function which seemed to have been forgotten in the present technological age.

Likewise, the H.P. turbine primary coupling failure referred to was catastrophic principally because of the consequential fire. The gutting of the engine room was produced by the initial vaporization and rapid combustion of only several hundred gallons of lubricating oil which escaped from ruptured rubber bearing supply pipes in circumstances where neither the gravity tank run down valve could be closed, nor the lubricating oil pump stopped from a remote position. Although such provisions were not statutory, builders and owners would be aware that lubricating oil could be equally as hazardous as fuel oils, and that under casualty conditions fuel and lubricating oil gravity tanks situated within the light and air casings could hazard the ship.

The author had referred to the alignment of medium speed vee-engine crankshafts and spurious crankshaft deflexions. About 18 years ago, a particular London owner had suffered a spate of crankshaft failures across the webs. The ships' records of crankweb deflexions indicated that the alignments were as good as the builder could have produced and well within the limit of tolerance of 0·003 in. In the first instance failure had been attributed to stress concentration at a deepish machining score in the journal fillet radius aggravated by the close proximity of the journal/crankpin oil hole. After the third successive failure, however, and a very full crankshaft alignment investigation, it had been concluded that the accepted method of taking dial gauge deflexions between the webs was suspect. These medium speed engine crankshafts were relatively stiff and the running gear relatively light, as compared to a slow speed engine, and it had been found that removal of a bottom half main bearing produced a relatively small change of deflexion which in any case was well within the limit given in the instruction manual. Subsequently deflexions had been taken at each cardinal point by first jacking down the adjacent main bearing journals. Having regard to the increasing intensity of tank top loadings, combined with the use of resilient mountings and thin shelled bearings, he felt that it was time medium speed engine builders provided fixed entablature reference blocks which could be utilized for checking alignment in the manner indicated by Fig. 9 under stationary and possibly also running conditions.

He would be obliged if Mr. Batten could enlarge upon the proximity transducer method of alignment to which he had referred.

Mr. A. R. Hinson, A.M.I.Mar.E., said that the reference to the incidence of defects for controllable pitch propellers—11·1/100 years of service—was best considered in comparison with figures derived in a similar way for other items of machinery; for example, 24·4 for water-tube boilers, 17·7 for turbines, 6·4 for gearing and only 0·95 for fixed pitch propellers. A fixed pitch propeller was a relatively simple piece of machinery, hence the incidence of defects was low, whereas, water-tube boilers and turbines were relatively complicated and operated under arduous pressure and temperature conditions. It would seem that an incidence of defects of 11·1 was not unduly high for equipment as complicated as a controllable pitch propeller.

With reference to the failure caused by dirt in the system, there was a tendency for manufacturers of hydraulic equipment to fit filters on the return line into the hydraulic reservoir and, on the assumption that once the tank was clean it stayed clean, no filter, or only a coarse filter was fitted on the suction side of the pump. A filter on the discharge side of the pump had much to recommend it. It also helped to prevent contamination if the tank filter had a fine mesh screen for straining the fluid when filling, and also if the tank was relatively narrow and deep with

* Butler, J. F. and Crowdy, E. P. 1972, "The Doxford Seahorse Engine", *Trans.I.Mar.E.*, Vol. 84, p. 73.

sufficient capacity to enable dirt and air to separate out. Hydraulic engineers agreed that dirt was the biggest enemy of good operation.

With regard to the alignment of magnetic couplings, Mr. Hinson assumed that the coupling air gaps referred to were radial gaps. It was often difficult to measure these gaps accurately, but unless the radial gap was equal all round the coupling, the magnetic forces were out of balance since the force was inversely proportional to the square of the length of the gap. Thus, if the gap at the bottom was twice that at the top, the magnetic pull downwards was greater than the upward force, and this could lead to increased misalignment and increased out of balance force.

The trouble with the hydraulic coupling illustrated in Fig. 5 was interesting, but the author did not say why the main engine crankshaft moved forward. Could it be that the drain holes shown at the bottom of the drawing choked and oil pressure built up in the void space? The oil pressure would be increased by the centrifugal force on the oil in the space and would push the clutch shaft forward and cause the aft friction pads to take an increased share of the load.

The author's views on fatigue failure of crankshafts were always worth studying and it would have enhanced the bibliography if he had referred to his own paper* which he had given to this Institute.

With regard to the H.P. rotor of the 15 000 shp set, which failed allegedly during a change of steaming conditions, similar failures had occurred when reducing steam pressure after a long run at full power. These failures might have been caused by water in bled steam connexions or gland steam systems being drawn into the turbine with subsequent thermal shock. Efficient non-return valves should be fitted to prevent water entering the turbine through connexions of this type.

As the author had stated, vibration monitoring of turbines was to be recommended, and Mr. Hinson added that a device to monitor the axial position of the rotor was also well worth while.

---

* BATTEN, B. K., 1958, "An Introduction to Fatigue in Marine Engineering." *Trans.I.Mar.E.*, Vol. 70, p. 331.

Correspondence

MR. F. A. MANNING, B.Sc., A.M.I.Mar.E., in a written contribution, said that during the past five years the Institution of Mechanical Engineers had held a number of symposia on component failures in engineering plant. It had been implied in the papers presented at these meetings that 60 per cent of defects arising in service could be firmly apportioned to design shortcomings, with troubles due to bad workmanship being minimal. The equipment under consideration had covered a much wider spectrum than marine engineering although marine engineering had always been included in the subjects under discussion.

From the author's wider knowledge of the 1000 or so cases his department had dealt with in the past ten years, did the overall analyses of cause of marine plant failures confirm this general picture? Mr. Manning's own knowledge of machinery failures certainly seemed to confirm Mr. Batten's views that the old problems died hard and for some inexplicable reason lessons learned seemed soon forgotten. Also, in his experience, due to commercial pressures, designers were very reluctant to admit to error, leaving the unfortunate operator of plant to believe that bent steam turbine rotors were an act of God and not an act of inadequate gland clearances, and also that every fatigue failure, be it a cracked plate or turbine blade, was a completely new and puzzling phenomenon to be relished.

If indeed design errors were the principal cause of today's troubles, as suggested by published data, did the author now think that his final remarks at the meeting advocating an extended shakedown trial would seriously cure this fundamental defect in the engineering process.

In Mr. Manning's opinion, if the identified problem was in the design stage then its proper cure should be applied to the design office.

The *Queen Elizabeth 2* turbine blade failure was almost a classic example of a vibration mode which could not have been detected in service from outside the machine since the clamped-pin flap mode was in equilibrium, with no external reactions in the shaft prior to failure, unlike, for example, a one nodal diameter disc mode which could exhibit a sympton of primary shaft unbalance at outset.

As stated in his written contribution to the paper by Fleeting and Coats* it had then been common practice at C.E.G.B. for some time to request all vibration data theoretical and experimental for every single blade row on new machines to be submitted for approval. The submission of this data was an essential condition of contract. This pre-supposed that somewhere in the electricity supply industry an efficient (and also expensive) design vetting department existed. The subsequent significant reduction in blade failures in modern plant during the late 1960's spoke for itself.

Mr. Batten had quoted from the scriptures, Mr. Manning quoted from John Ruskin who said that people who considered only price and not value were the lawful prey of those who purveyed shoddy goods.

FIG. 21—*Erosion-corrosion pitting of propeller shaft liners*

---

* FLEETING, R. and COATS, R., 1970 "Blade Failures in the H.P. Turbines of R.M.S. "Queen Elizabeth 2" and their Rectification." *Trans.I.Mar.E.*, Vol. 82, p. 49.

In the purchase of large capital plant, such as ships and their machinery, "value", in most cases, was related to reliability, and the onus fell fairly equally on everyone concerned to take nothing for granted, especially the design of an untried plant. Mr. Batten's department must have been in a position of great strength and envy when a prospective shipowner sought advice on plant reliability. Having this latest failure data to hand to feed back into design or production did not seem consistent with the author's frank admission that progress over the past ten years in eliminating common faults had been disappointingly slow. What was the true reason for this state of affairs?

Mr. Hinson had made a plea for more supervisory gear—and in particular fitting high vibration trips on turbine machinery. Over the past ten years Mr. Manning had investigated a number of failed turbine rotors due to a permanent bend resulting from

FIG. 23—*White metal damage due to loss of contact at bush*

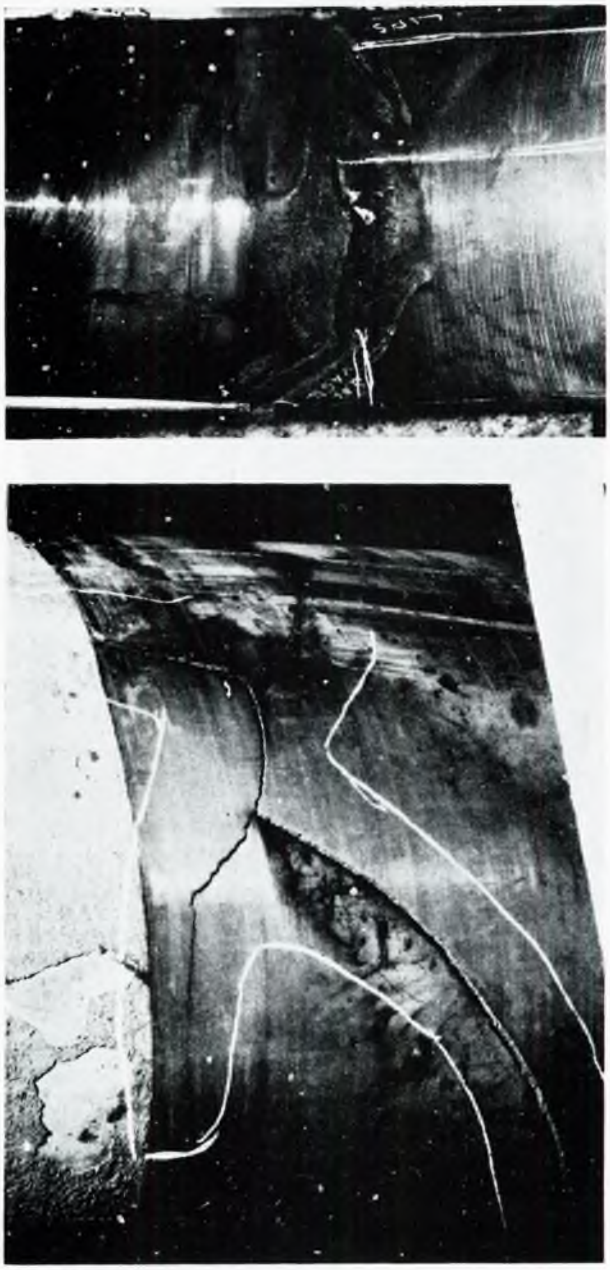


FIG. 22—*Fractures of propeller shaft liners*

gland rubs. Those incidents occurring in power stations where continuous monitoring of bearing vibration levels was provided produced interesting data. Firstly the majority of incidents had taken place during transient operation such as speed or load changes. The time period from onset of noticeable increase in vibration above the norm to an off-scale reading was very short of the order of ten seconds and with the self-destructive type of failure, once the rub was established, he would not have thought it likely to have averted disaster with a high vibration trip. He would agree with Mr. Hinson that a rapid trip was desirable to minimize consequential damage—but with respect he would not share his optimism as regards safeguarding the rotor from a permanent bend with the aid of this device.

At present engineering contracts were still very slow to introduce a penalty/bonus clause, referring precisely to guaranteed levels of availability of service duty. In the past such liquidated damage clauses had related solely to thermodynamic performance and late delivery of plant. In his opinion financial motivation was a strong one to persuade erring managements that poor availability of engineering plant was not going to be the accepted way of life for the 1970's.

Would Mr. Batten venture to add some further advice on how best in the long term this basic problem could be tackled to reach a successful conclusion.

MR. G. C. VOLCY, M.Sc., M.I.Mar.E., wrote that he entirely agreed with the author that, to overcome the failures affecting different parts of the propulsive plants of ships, it was absolutely necessary to combine the thoughts of many people from several disciplines. A common effort should be made by several organizations working in shipbuilding and shipping. He, and Bureau Veritas, would like to complete the author's cases of machinery failures with some which were previously investigated by the trouble shooting team of their organization, for which work he was responsible. Several publications had been written by their staff so he would only mention the essential problems they had met which were similar to those mentioned in the paper.

The author's conclusions concerning main shafting and stern gear were very similar to those published in 1967 (1). In fact, the case of the broken liner (Fig. 3) and of the pitted tail shaft liner (Fig. 2) looked very much like Figs. 21 and 22 (Figs. 3 and 4 in reference (1)). Figs. 23 and 24 (Figs. 6 and 7 in reference (1)) showed that the loss of contact of the forward bush of the stern tube had provoked destruction of white metal and this was due to the old fashioned straight alignment. These troubles were overcome by application of the rational shafting alignment developed

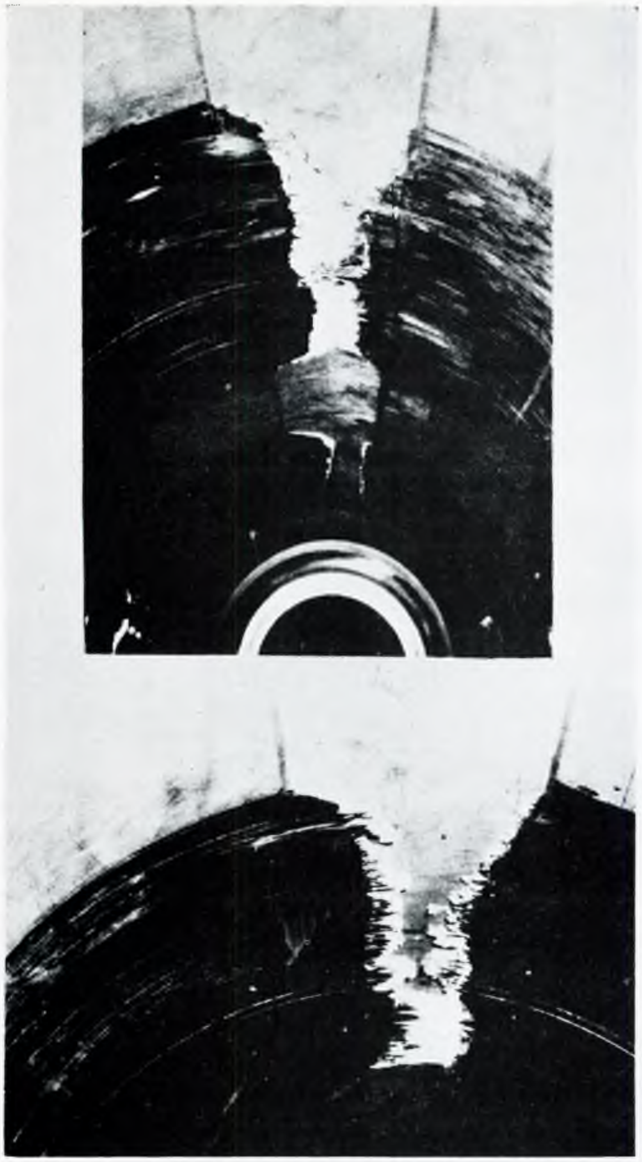


FIG. 24—*Upper part of white metal bush found seized after sea trials*

by Bureau Veritas ten years ago. Fig. 25 (Fig. 52 of reference (1)) showed a liner previously pitted due to the straight non-rational alignment which was machined (by the lignum vitae) when the negative reaction in way of the forward bush had been cancelled. The contact pattern of the whole length of the lignum vitae which could be realized by such a curved rational alignment was shown in Fig. 26 (Fig. 51 of reference (1)).

One of the reasons for troubles occurring on stern gears was not only the non-rational straight alignment but also the presence of the conventional stern tube including two bushes, aft and forward. This conservatism caused many problems and they recommended shortening or even suppression of the stern tube making the length not greater than the length of the aft bush. In order to keep the tail shaft free of vibration a convenient location for its forward support, replacing the old fashioned forward bush of the stern tube, should be realized by a vertically movable plummer block. Of course, the calculation of the vibratory behaviour of this part of the shafting should also be previously executed.

Mr. Batten's remark concerning troubles occurring on gearings, pitting, spalling, heavy wear and broken teeth, should be ascribed in most cases not only to metallurgical properties of the metal or of machining of the gearing but also to exterior influences such as hull deformation due to loading and sea waves, as well as to the detrimental influence of old conventional thrust bearing architecture. This problem had been investigated by his society for about 12 years and the results of these investigations had been published[2,3]. Gearing trouble having the pattern of pitting and heavy wear was shown in Figs. 27 and 28 (Figs. 1 and 2 of reference (2)).

He agreed with Mr. Batten's deduction that several cases of troubles concerning the crankshafts of main engine bed plate and double bottom steel-work deformations were due to loading conditions as well as to thermal dilation. Over prudence with allowable stresses of crankshafts had provoked the over stiffening of the crankshafts leading to many unnecessary troubles, first with bearings and later with the crankshaft itself. His experience had led him to the conclusion that the main bearings were the safety valves of the crankshaft. In fact, it was the main bearings which deteriorated first due to the abnormal behaviour of the crankshaft, so attention must first be drawn to the behaviour of the main bearings. If the overload was not taken from the crankshaft in time it resulted in deterioration by seizure or hammering of the white metal of the main bearings, then the crankshaft could break in bending. Such cases were shown in Figs. 1 to 7 of reference (5). (Figs. 3, 4 and 6 are shown here as Figs. 29, 30 and 31).

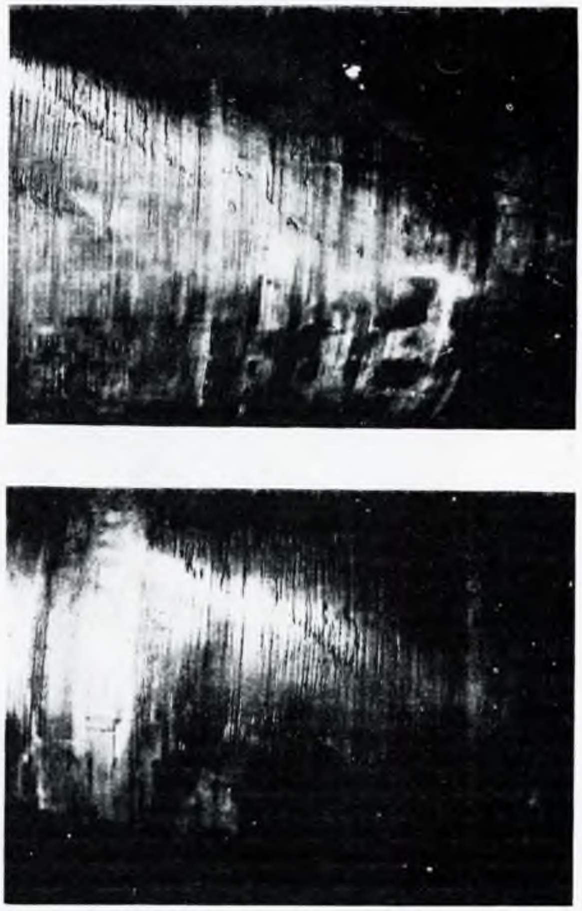


FIG. 25—*Erosion-corrosion pitting process of propeller shaft liner arrested by rational alignment*

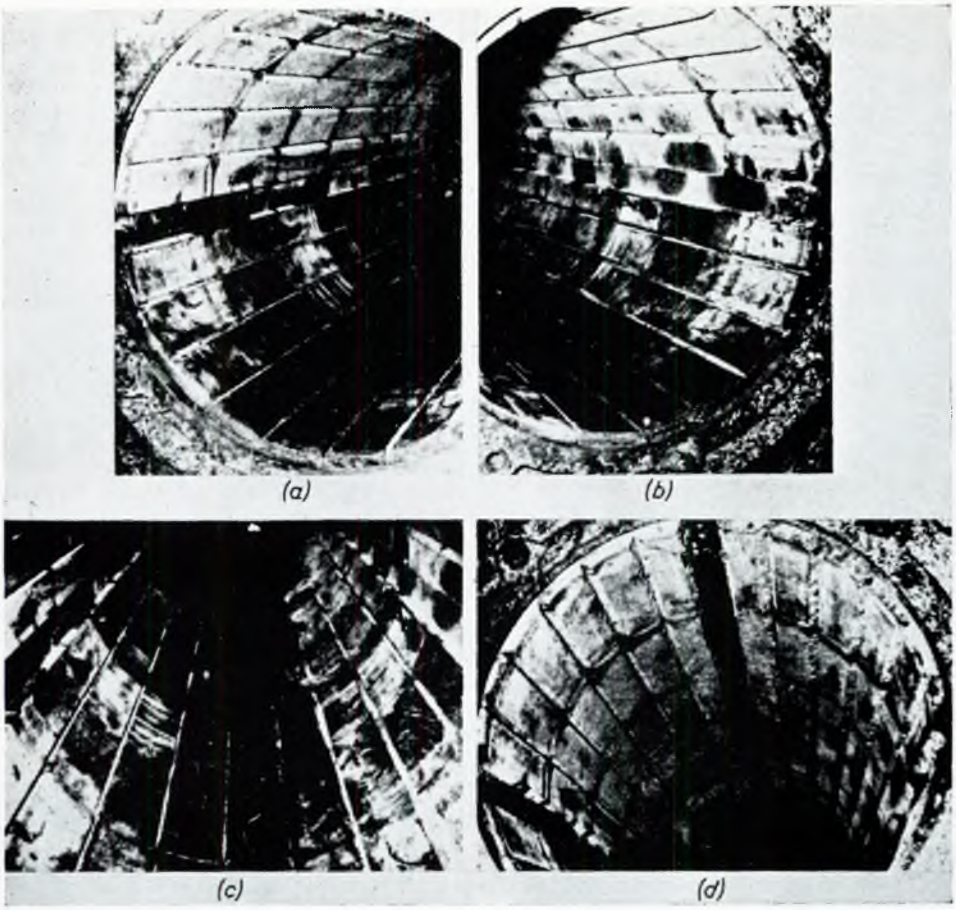
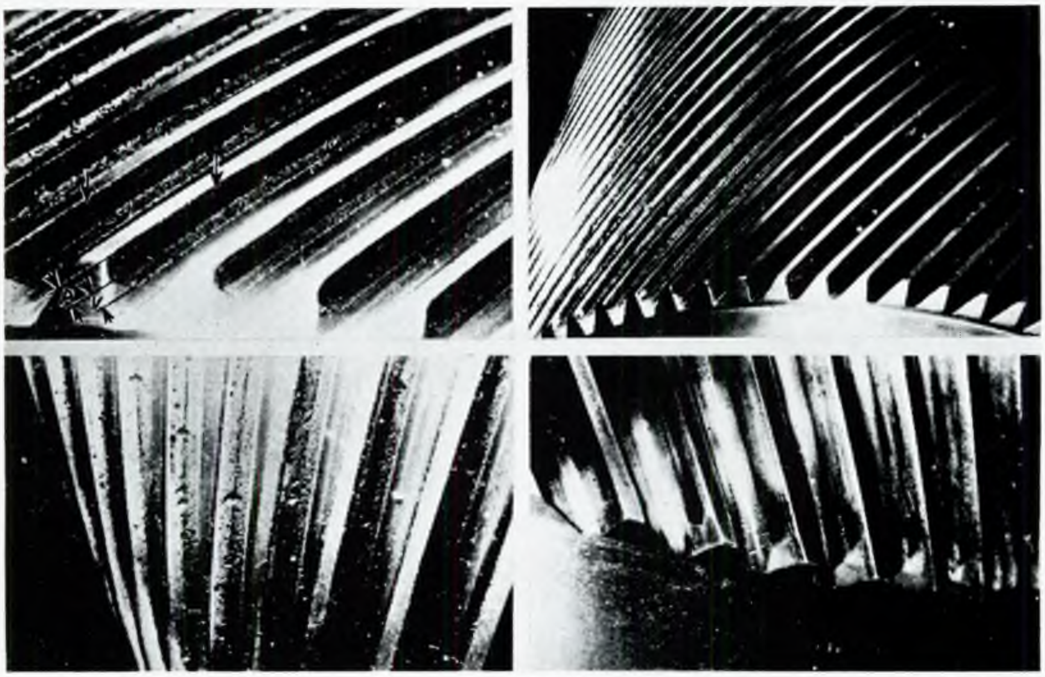
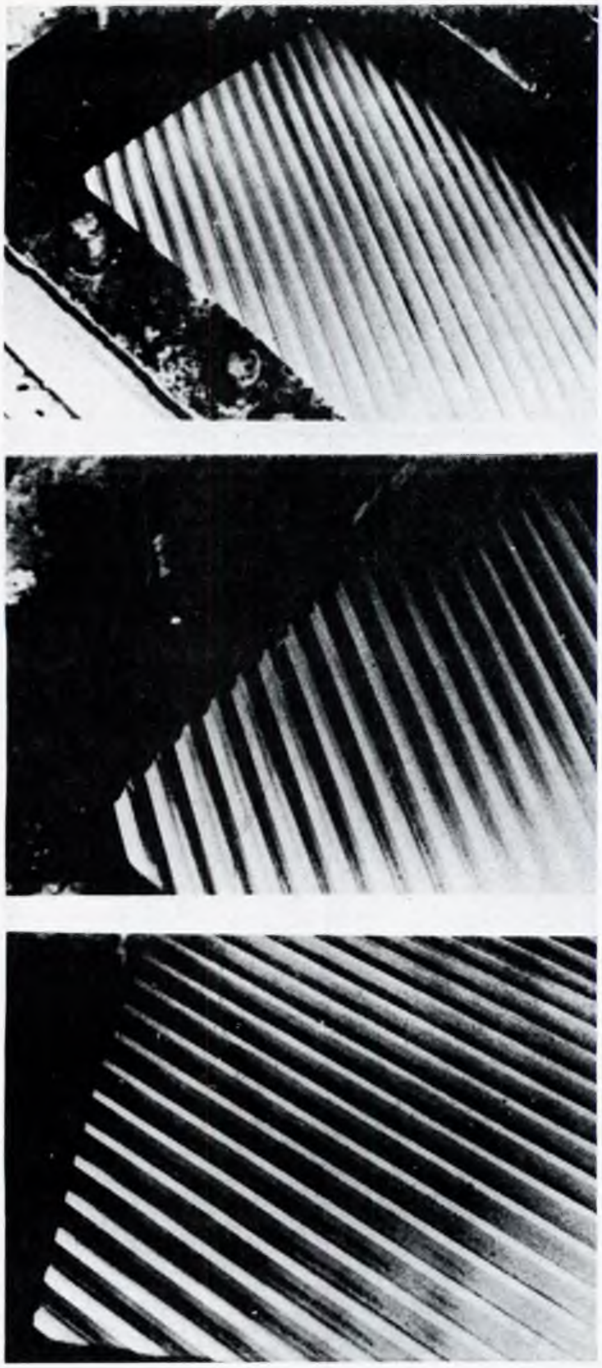


FIG. 26—*Slope bored aft stern tube bush (lignum vitae) after a year of operation—(a) aft extremity of bush viewed towards port, (b) aft extremity of bush viewed towards starboard, (c)forward lower portion of bush (slope bored), (d) forward upper portion of bush*



17 600 shp *n = 108 rev/min*

FIG. 27—*Pitting, spalling and heavy wear of after helix caused by tilting of thrust block and a non-rational alignment of the shafting supports*



17 000 shp *105 rev/min*

FIG. 28—*Pitting, spalling and heavy wear of forward helix caused by double-bottom flexibility tilting of thrust-block and a non-rotational alignment of shafting*

Such cases as those mentioned above were, according to their researches, due to the straight alignment of the crankshaft of the main engine during fitting out of the vessel at the quay. When the ship was loaded the deformation of the steel-work and bed plate occurred provoking vertical displacement of the bearings as could be seen in Fig. 32 (Fig. 9 of reference (5)). If, to this static deformation, they also added the wave loading, the crankshaft very often could not follow such a deformation of the bearings. The freely flying crankshaft journal vibrated as shown in Fig. 33 (Fig. 10 of reference (5)). Such behaviour of the crankshaft led to the mentioned overstressing, the general idea of the stress was presented in Figs. 11 and 12 of reference (5). The free flying crankshaft journal led to the breaking of the oil film and the destruction of the white metal, as shown in Figs. 4, 5, 6 and 7 of reference (5). These troubles with bearings were overcome several years ago by application of the curved alignment of the crankshaft. A very satisfactory study was near completion which would greatly facilitate the execution of such a curved alignment.

It was not surprising that such cases as those presented by Mr. Batten in Figs. 8 and 10b occurred.

He asked the author for his opinion and outlook regarding application to engine chocking of a product called Philadelphia Resin.

He said that Bureau Veritas were always ready to put at everyones' disposal the results of their research work in order to ensure the trouble free running of the vessels which were "to be sailed instead of to be repaired".

REFERENCES

1) VOLCY, G., 1967 "Forced Vibrations of the Hull and Rational Alignment of the Shafting." *Nouveautés Techniques Maritimes.*
2) VOLCY, G., 1969 "Damage to Main Gearing Related to Shafting Alignment." *Proc. Imas 69.* p.15/16
3) VOLCY, G., 1969 "Influence of Hull Deformation on Shafting Alignment and Behaviour of Marine Gearing." 7th Round Table Conference on Marine Gearing, Finspong.
4) VOLCY, G., 1968 "Actual Behaviour of the Marine Reduction Gear and Alignment Conditions of the Line Shafting." *Noueautés Techniques Maritimes.*
5) BOURCEAU, G. and VOLCY, G. 1966 "Some Aspects of the Behaviour in Service of Crankshafts and their Bearings." ATMA.
6) VOLCY, G., 1970 "Vibrations of Crank and Thrust Shafts and Damage to Bearings." *Nouveautés Techniques Maritimes.*

MR. I. T. YOUNG, M.I.Mar.E., wrote that it seemed rather naive that loadings for epicyclic gears should have been fixed, on their first introduction to marine service, on the basis of a comparison between land-based epicyclic gears and marine propulsion parallel shaft gears. Land-based parallel shaft gears were appreciably higher rated than their marine counterparts.

Those who had had experience of epicyclic gears in whatever application, especially with double-helical teeth, would know that tooth strength had always been their weak point. The reversing load on the planet teeth, as Mr. Batten pointed out, automatically reduced their bending strength in the ratio of 1·6 to 1.

With double helical teeth, the apex wander could be a critical factor which had to be catered for, in addition to pitch error differences between planets, by the flexibility designed into the annulus ring and its supports. This was especially so when the number of planets exceeded three and the freedom of location of the sun could not assist.

His own firm's experience with such gears was limited to shore based installations, but it had been repeatedly obvious that off-design conditions which could have been accepted quite happily by conventional pinion/wheel installations could be quite disastrous to some epicyclic gears. There was a quite natural desire to keep the number of teeth in the planets and sun reasonably large, but their experience had shown that 22 or 23 teeth were frequently appropriate in order to take advantage of the surface load carrying capacity.

In one installation, the tooth pitch had successively to be increased from three to four and five module to combat a succession of tooth failures in generating plant. The final tooth bending stress figures were 85 per cent of the bare allowable figure under Lloyd's Register Rules for parallel shaft propulsion gears.

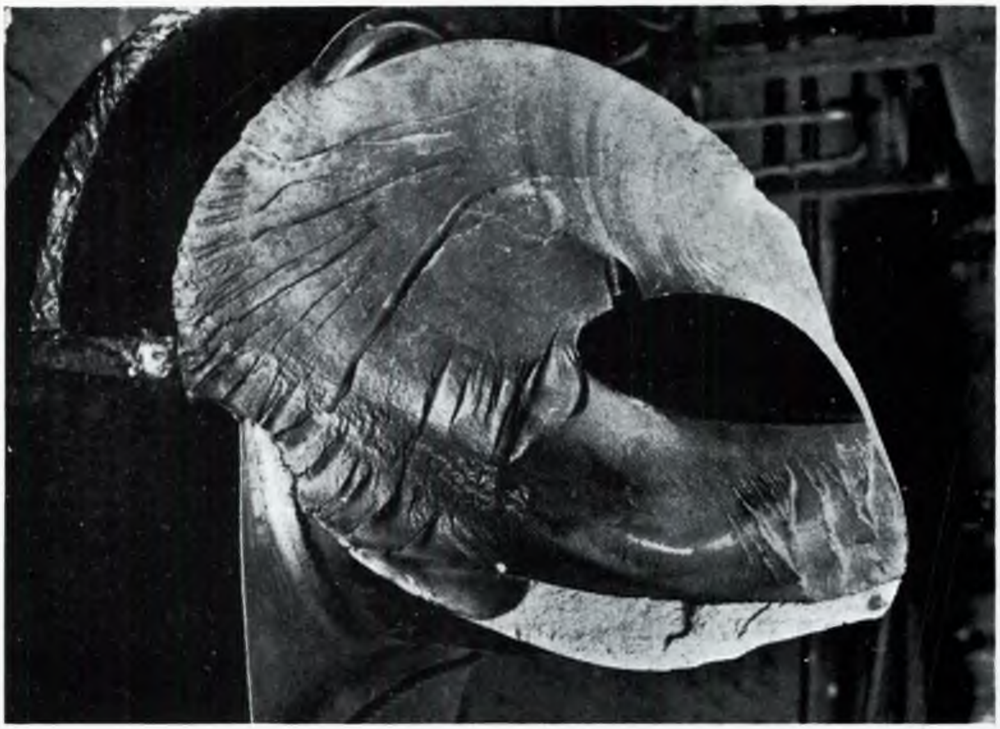


FIG. 29—*Fracture caused by torsion bending*

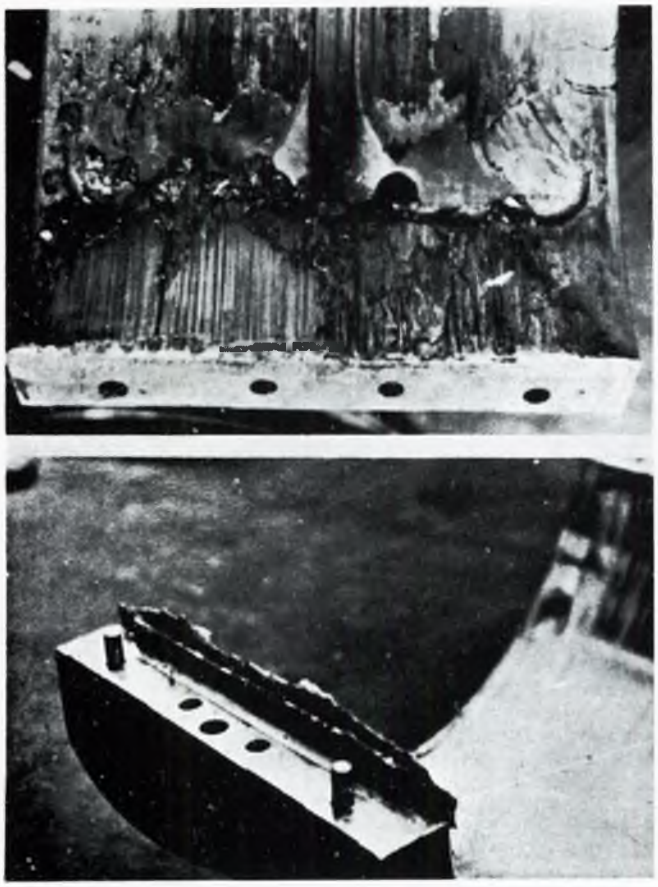


FIG. 30—*Damage to shells of seized bearings*

FIG. 31—*Bearing shell damaged as a result of hammering*

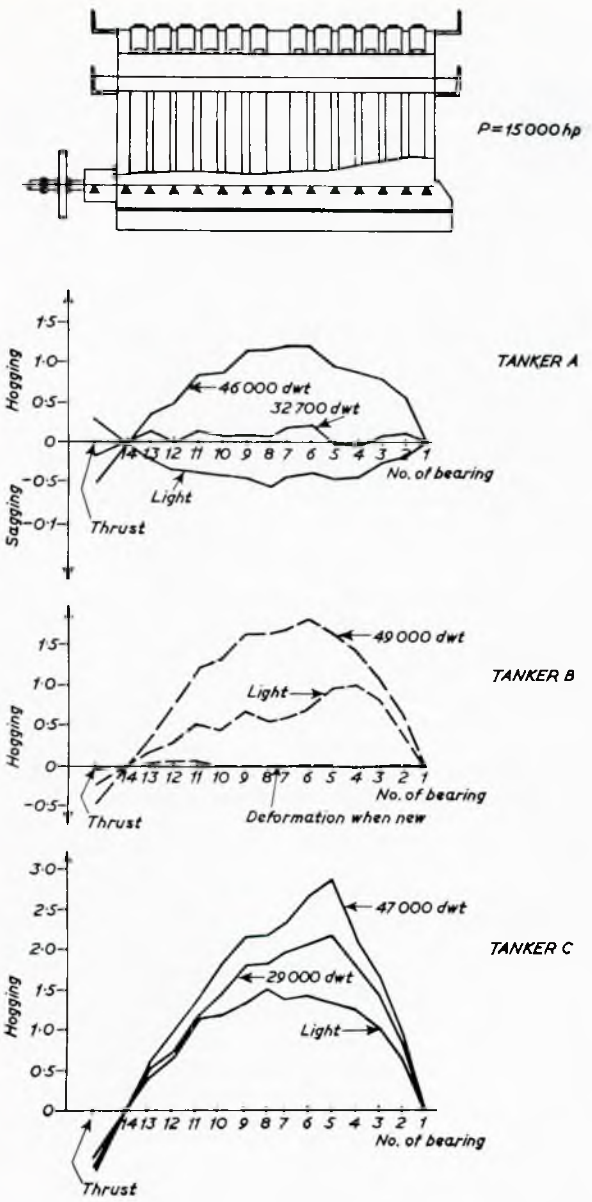


FIG. 32—*Deformations of engine bedplates in tankers while loading*

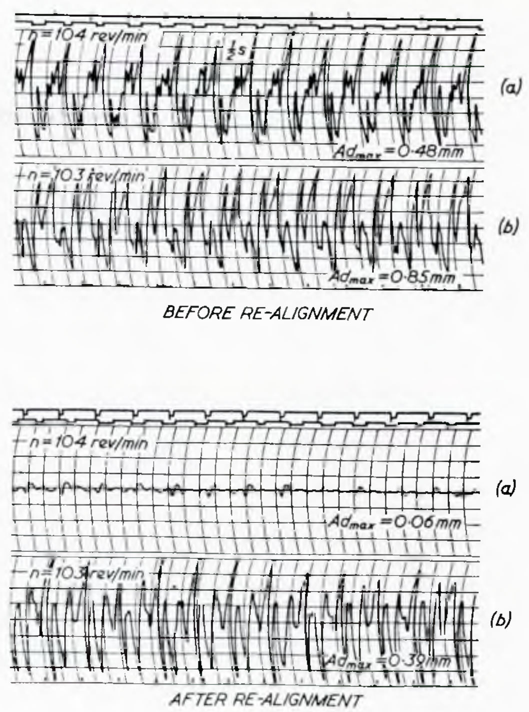


FIG. 33—*Recordings of the vibrations affecting a crankshaft with two alignment variants (a) vertical vibrations of a journal, (b) longitudinal vibrations of a web—P = 15 000 hp*

Author's Reply

Mr. Batten thanked Mr. McNee for opening the discussion and emphasizing the importance of taking action before it was too late. It could not be stressed too often that while one naturally, though not deliberately, tended to take pleasure from other people's problems—the next time it could be you.

Incidence of failure expressed in terms of defects per hundred years of service, though somewhat confusing to begin with, did in fact provide a very handy way of setting out the problem, especially as one was able to relate the size of a fleet and its expected life to service years. For example, if a component had an incidence rate of ten per hundred years then an owner with four ships, each with an expected life of say 25 years, might expect to meet ten defects in his fleet of this particular component during their total lifetime, i.e. an average of 2·5 defects per ship. To compare incidence of failure rates of various types of machinery, as Mr. Hinson had done, was somewhat misleading as these figures did not necessarily indicate either the type or the seriousness of the defect involved. Thinking over the range of defects generally experienced it was therefore perhaps true to say that an incidence of failure rate of 11·1 c.p. propellers could be regarded as at least as serious as that of say 24·4 for watertube boilers, many of the defects of which could be dealt with on the spot.

The problems of communication within industry had engendered many papers both to this and other Institutions, but the author suspected that theory was not always borne out by practice.

Ideally every draughtsman should see every product which came from his pen, but in practice, of course, this was sometimes hard to realize. However, some interchange of thought and opinion between drawing office and shop floor would undoubtedly be of benefit.

On the question of alignment the author felt that engineers were not working sufficiently closely with the naval architects. The fault undoubtedly lay with both sides, but it was of increasing importance that there should be a very real understanding between hull and machine men as to the precise limits within which each was able to operate.

Mr. Duncan had given some interesting examples from his experience, and one of the delights of such a paper was perhaps that it invariably brought to light some interesting occurrences hitherto tucked away in obscurity.

The author apologized to Mr. Duncan for leaving him in the air regarding the flexing of the bossing arms on the passenger liner. The stresses being experienced on the bossing arms were not considered to be excessive, and subsequent service experience showed that the re-alignment of the shafting had produced quite a reasonable improvement in the rate of weardown. As a final measure it was decided to change the tailshaft liner material from an aluminium bronze to a softer manganese bronze, and an acceptable final rate of wear in the stern bearings was achieved. Mr. Duncan had also asked about other ways of prevention of failure and the author would have thought the reply to this was fairly obvious. There was no way of countering the unexpected, such as a hidden material defect, but there were far too many defects which could be traced back to lack of thought, care or attention in the original design, fabrication or inspection of the failed component.

The author was glad Commander Weaving had reminded them of the effectiveness of so many epicyclic gear units. Living so close to failures one tended to close one's eyes to those units which were operating satisfactorily in service. The author sincerely believed that they had probably beaten the problems on epicyclics to a large degree, and thought that Commander Weaving's company had been extremely helpful in all the work they had done together in analysing this very relevant and immediate problem.

Mr. McAfee's contribution was, as always, both entertaining and highly relevant. The author was glad that Mr. McAfee had raised the problem of screwshafts for they lay in a much neglected end of the ship. Nowadays engineers seemed to monitor everything they could lay their hands on. In the Royal Navy coloured lights were provided for almost every machine so that, at a glance, one could tell whether the vibration level was acceptable or not. In our Merchant Service engineers now sat in comparative comfort with a printer chattering away on one side, and warning lights, all satisfactorily glowing green, on the other side, but the after end was left to carry on on its own. It was the author's firm opinion that much more attention would have to be paid to aft end problems if catastrophic failure was to be prevented, because the borderline between satisfactory running and failure was becoming very sharp indeed. If one cared to examine the pattern of gearing defects on large ships over the last five years there would seem to be a shadow lying across them not unconnected with aft end movement. It was the author's opinion that not enough attention was being paid to the rigidity of the shafting system as a whole when compared with the rigidity of the aft end of large ships, and that in fact shaft and gear alignment could not tolerate the movement of the aft end of these ships under certain sea and operating conditions.

Mr. McAfee had asked about the shaft whirl. The author felt this was a dangerous word, as shaft lateral vibration would be far more descriptive. This was indeed something of a problem, although recent advances in knowledge had enabled a far more accurate prediction to be made of troublesome excitations of the system.

Mr. Mack's remarks were welcomed and he had made one very significant point. He had mentioned finance. The author felt that engineers must be very clear in any approach to their financial directors as to what was important and what was not important at the design stage of ships. So much could be looked at during design or trials which would prevent very expensive failures later on. When one thought of the cost of taking a ship out of regular service nowadays, particularly the large super tankers or container vessels, one realized it was nothing short of phenomenal. Perhaps two or three days of measurement and consideration at an earlier stage might save a lot of heartache later on. Mr. Mack also queried the use of proximity transducers for alignment. This was perhaps misleading as they were of most use in measuring relative movement between two parts of a component.

Mr. Hinson had raised some interesting points and the author hastened to remedy his omission regarding the bibliography.* Referring to Fig. 5 which showed a clutch arrangement, Mr. Hinson had asked why the main engine crankshaft moved forward. The reason for this was quite simple in that since the aft linings only of the clutch had worn then, with the clearance of the cone taken up, the forces due to the engaging pressure of the aftermost inner cone would be opposed by the thrust shaft, but those acting on the forward inner cone would not be so opposed and movement of the clutch shaft as a body in a forward direction would occur.

The written contributions of Messrs. Volcy, Manning and Young were appreciated, and most of their queries had already been answered. The author was glad, though not surprised, to see that Mr. Volcy's ideas on aft end problems coincided with his own. Mr. Volcy had much experience in this particular field and had some very valuable contributions to make. Mr. Volcy had asked one question concerning Philadelphia Resin. In the Society's experience this had proved to be a reliable chocking material once a satisfactory initial alignment had been achieved. Perhaps the latter needed to be emphasized as once chocked, re-chocking became an expensive and tedious procedure.

In conclusion, the author said he would like to emphasize that there was still an awful lot to learn from the errors and breakdowns of the past. Too often failures and breakdowns were treated as remarkable wonders to be exhibited and discussed over a coffee and then shelved thereafter. Engineers had a great responsibility to look at each and every failure and to think whether something could be done to prevent it happening again.

The author was always reminded of the Scripture passage which said: "Whatsoever thy hand findeth to do, do it with thy might".

---

* Batten, B. K. 1958. "An introduction to Fatigue in Marine Engineering". Trans.I.Mar.E., Vol.70, p. 331.

CORRIGENDUM

It is regretted that, in Part 7 on page 241, the contribution made by Lt. Cdr. R. Hales, R.N., M.I.Mar.E., is shown as being made by Mr. K. C. Hales.

Papers read at the Institute on Wednesday, 19th January, 1972

SOLID STATE MOTOR STARTERS

G. W. Pritchard* and B. Highe B.Sc.*

Mr. Pritchard

The development of static motor starting contactors and other electronic switching means are discussed, existing production types and their utilization considered and explained. The control methods and their application to maritime problems, where the application cannot be met by conventional mechanical switching, are examined along with reliability, servicing and future usage.

INTRODUCTION

An electronic switch, relay or static contactor is best described as a means of switching without the use of any mechanical moving parts other than the controlling means which, for practical purposes, may be the normally accepted push button switches.

These have now been so modified that they consist of either long-life magnetically operated enclosed reed switches, pressure switches or magnetically operated solid state units.

With the advent of integrated circuit logic and computer control, the starter contactor may be controlled by this aforementioned means, being ideally suited for this form of remote control and sequencing.

Whilst it is possible to include the use of solid state switches as circuit breakers, insufficient work has yet been carried out on the switching of potentially high short circuit currents by this means, to be able to offer constructive discussion on the practical application of the results.

Over the past decade in particular, much serious development, research and practical application has endorsed the obvious advantages of various applications using solid state switching.

Considerable background information extending over many years into the long term reliability has been accumulated and whilst no standards exist for these as for conventional mechanical switches and contactors, the inherent reliability associated with solid state controls has encouraged many industrial users to experiment with the many types marketed. These have given extremely satisfactory results, leading to the incorporation of solid state switching of motors and other functions throughout complete plants.

This paper deals largely with the more practical considerations of design and utilization of static switching and associated problems.

The basic component of an electronic switch is either a power transistor or a multilayer device known as a thyristor. Other devices have been manufactured using such names as "Triacs" or "Quadracs" but they may be briefly described as an electronic switching device which is triggered into a conducting state by the application of a low level signal.

STANDARDS OF SAFETY

It is not possible to give an absolute standard of safety for the solid state switch, but it is possible to examine those applicable to their mechanical counterpart for some guidance.

It must be impressed at this juncture that in electronic switching no circuit disconnexion occurs by mechanical isolation of the current path as in conventional switches, but is merely the introduction into the circuit of very high resistance by limiting the electron flow through the crystal structure within the thyristor. The physics of this phenomenon are outside the intended discussion of this paper.

Where solid state controls are to be used, the application engineer must consider the total circuit and allow for isolation further back to source in his circuit.

In most practical cases (see Fig. 1) the manufacturer has

FIG. 1

provided an "off load" isolator switch interlocked with the access door on his equipment, or, as shown in Fig. 2, isolation can be obtained by the removal of input HRC fuses.

Since all semiconductors have a known forward voltage drop, they dissipate heat, and the junction temperature of these semiconductors must be kept within the maximum temperature as given by the manufacturer's data. This means that they must be mounted upon heat dissipation platforms or heat sinks having a known dissipation rate and in the majority of cases, care must be taken to ensure the convection of air over these heat sinks and to consider the ambient working temperature where the finned heat sinks are protruding from the cabinet (Fig. 2).

Electrical isolation of a high order between the heat sink

* Booth Engineering Limited

and the semiconductors must be achieved without the loss of heat transfer. In the case of the motor starter shown in Fig. 2 this is accomplished by the use of mica sheet between two blocks

FIG. 2

giving at least 2·8 kV proof minimum between the semiconductor and earthed heat sink.

Difficulties are encountered where the electrical supply, either to the power semiconductors or the control logic is harmonically rich or subject to transient voltage spikes, as without carefully designed protection within the contactor or switch, they may be liable to inadvertent switching into the "on" state. A later section deals with this aspect of design.

In general, acceptable safety standards can be achieved and checked in the final test before application.

BASIC DESIGN CONSIDERATIONS

It is first necessary to define and outline the operational limitations both electrically and mechanically. The starting point must always be the main power semiconductor devices.

Manufacturers of these semiconductors produce specifications to which they are manufactured or selected.

For a standard 415 V a.c. r.m.s. 50 Hz supply, the peak inverse voltage rating of the power semiconductor device must be in the order of $2\frac{1}{2}$ times the applied voltage, higher transient voltage spikes of long or short duration are eliminated by the methods described in the paragraph on protection.

The i^2t characteristic curve allows for the contactor to be rated correctly for the maximum designed switching capability. The di/dt rating is important where the switched load is capacitive.

D.C. APPLICATIONS

A typical circuit for a d.c. contactor is shown in Fig. 3.

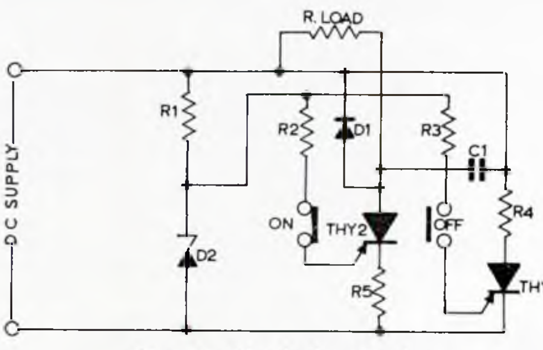


FIG. 3—*Simple d.c. circuit*

Upon "switch off" a smaller thyristor is fired commutating the main thyristors off by means of the charge in capacitor C, producing a reverse voltage in excess of that which is applied.

The controlling potential at lower voltage in this case where alternative voltages are not available, can be derived from series resistors. This can give rise to large wattage and heat dissipation.

Care must always be taken to ensure that the charge time of capacitor C is not anticipated by the switching rate.

The simple formulae for capacitor design is given by the equation:

$$C > t\ \text{off} \,/\, R'L\ \text{uf}$$

where t off is the turn off time of the power thyristors in microseconds.

Further Considerations in the D.C. Case

It is possible in the d.c. case to incorporate a system whereby the load carrying thyristor is not fired if the circuit resistance or load is below an acceptable figure, i.e. a short circuit condition.

Further, if a short circuit should occur after switching, the rate of rise of current may be limited and the contactor switched off before excessive current flow takes place. This can be controlled within the order of two milliseconds. Again it is possible to switch at high frequencies or rates with an adjustable on to off ratio such that control of mean output voltage is achieved. This lends itself to motor speed control, battery charging, etc.

SWITCHING A.C. CIRCUITS

With normal a.c. switching, the conducting thyristor is turned off by phase reversal every half cycle. Thus the design is simplified since no provision has to be made to force turn off the thyristors. In multiphase circuits the main considerations are in arranging for the thyristors to conduct at the correct time. For general purpose design, that is for inductive, resistive or capacitive loads, the gate firing of the thyristors must be arranged to occur over 180° of the half cycle for each thyristor.

Gate Firing Circuits

The simplest method is to derive the gate firing signal from the in phase voltage supply. This, however, is generally limited to a firing period of 170° by the voltage amplitude required to fire the thyristor.

The gate characteristics must be studied from the thyristor

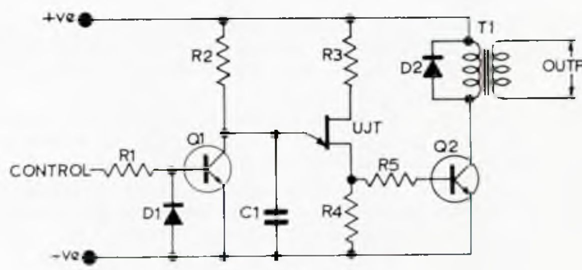


FIG. 4

manufacturer's detail. Pulse or d.c. level firing is generally the more acceptable since the full 180° firing can be obtained at the correct amplitude. Single phase and 3-phase firing circuits are shown in Figs. 4 and 5 (gate firing circuits). A simple circuit for a starting contactor for the a.c. induction motor is shown in Fig. 6.

The stator windings are supplied via three thyristors fired into a conducting state in the correct firing order by means of the thyristor gate control circuit. Inverse parallel diodes to each thyristor provide the return current path.

The contactor is rated to accommodate the initial starting surge of the motor which is in the order of six to eight times its full load current.

The gate firing circuit can be fed by automatic logic control from any process or level control signal, which may be remote and of low magnitude, as discussed at the beginning of the paper.

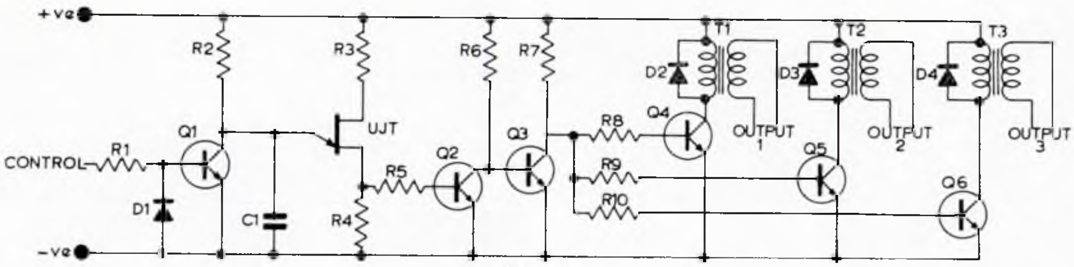


FIG. 5

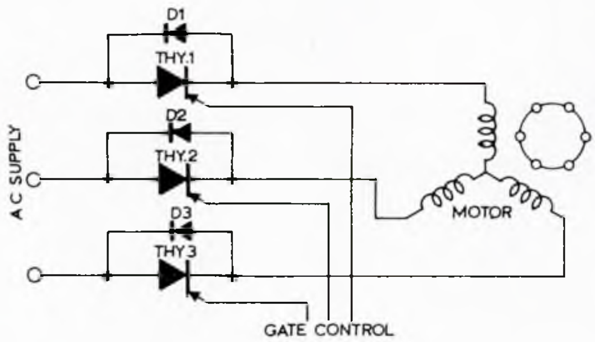


FIG. 6—*Simple a.c. circuit*

The gate firing circuit can be controlled to give an adjustable firing rate to the thyristors. This will cause the thyristor to fire "on" for a short period, the gating signal being delayed (see Fig. 7). The rate is increased by means of a "ramp" circuit which

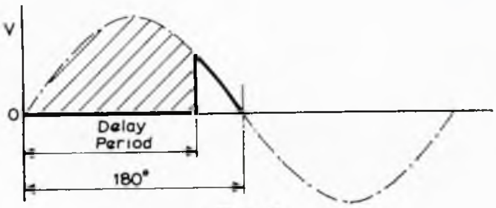


FIG. 7

progressively reduces the delay period until the thyristor is fully conducting and a complete sine wave is present. By this means the starting current to an SC induction motor may be controlled within 300 per cent full load current and a smooth take up of, say, a winch load effected.

The initiate signal for this form of control can be manual or automatic.

Further Considerations in the A.C. Case

Where the a.c. motor is to be repeatedly switched on and off, the period may be shortened by the incorporation of electronic d.c. braking.

This involves the firing of one of the thyristors at a controlled phase angle relative to the supply voltage wave form. The resultant injection of d.c. gives rapid motor deceleration.

By suitable arrangement of the cross connexions of the thyristors, motor reversal can be achieved without the use of moving parts.

RELIABILITY AND SERVICEABILITY

The nature of the static contactor or electronic switch, in that all component parts are solid state, means that a high degree of reliability is inherent provided that the electrical and mechanical parameters are understood and care is taken to ensure correct ratings are adhered to.

Since wear rate is not a consideration (as applied to mechanical switching means) the operating rate of switching can be discounted, within the limits of the specification. The failure rate can only be determined by a statistical analysis of the predicted failure rate of individual component parts.

Experience in the field indicates a maintenance free life expectancy measured in years.

The limitation in components viewed as an assembly are generally concerned with the life of capacitors, especially electrolytic capacitors which eventually deteriorate.

Care must be taken to ensure that where timing circuits are involved electrolytic capacitors are not used.

Sophisticated types of capacitor and other parts can be used to give even greater reliability, cost being the limiting factor. Where manually operated push button controls are used, the current carried by their contacts is of low voltage and magnitude. As the main supply is not carried by these switches, little contact burning or wear occurs.

Serviceability

Generally the static contactor or motor starter has a reduced number of removable parts. The printed circuit logic and gate firing circuits, fuses, power semiconductors and transformers are easily removed and replaced.

For a range of contactors say 10 to 150 amp/phase rating the construction and components are exactly the same with the exception of the power semiconductors (and the protecting HRC fuses) which are applicable to the chosen rating.

This means that for a given installation the carrying of spares is reduced considerably.

Fault Finding

The tracing of any fault in this type of equipment, although at first sight complex, has, in practice, resolved itself to the simple task of determining in which of the four sections the fault is occurring and then replacing the defective part.

PROTECTION

Protection of semiconductor switching devices can be sub-divided into two main categories:

a) protection against overvoltage;
b) protection against overcurrent.

These categories can be further sub-divided into long and short term effects.

Long term overvoltage protection is taken into account by specifying all switching devices with a voltage withstand capability of at least twice the peak supply voltage.

Short term overvoltages take the form of surges or "spikes" which are superimposed on the supply waveform and may vary in duration from a fraction of a microsecond to a millisecond and in amplitude from a few volts to many thousands of volts. It is a general rule that the longer the duration of the "spike" the lower its amplitude. Where the amplitude is less than the voltage capability of the switching device, then no protection is necessary. If, however, the amplitude is greater than this level then the "spike" must be prevented from reaching the switching device as it may cause permanent damage.

The first line of protection is a series resistor capacitor network connected across the switching device. This has the effect of absorbing and dissipating the energy in the "spike" and also steering a certain amount of the energy into the load.

Some form of voltage clamping component may be connected across the switching device, usually a non linear resistor or a selinium surge suppressor and in both cases the effect is to present a low resistance to voltages above a certain level. This level is usually about 75 per cent of the voltage which the switching device can withstand. Hence, any voltages above this level are effectively clamped to a safe limit.

Where high energy overvoltage surges are anticipated a series inductance is connected in the supply lines to the equipment and this has the effect of reducing the amplitude and rate of rise of the surges to a level at which they can be dealt with by the above-mentioned methods.

Long term overcurrent protection as in the case of a sustained overload on the circuit, or motor being switched is usually taken care of by use of a delayed overcurrent trip. This takes the form of a current transformer to sense current and a level sensing circuit followed by a fixed or variable time delay. The effect is to switch the contactor or static switch into an "off" state before damage can occur. This system can be reinforced by use of temperature sensing elements on the thyristor heat sinks giving a tripping signal should the equipment become overheated. This has the added feature of giving protection against too high ambient temperature or loss of cooling air.

Short term overcurrent protection as in the case of a short circuit across the output of the equipment takes the form of the fast acting, high rupturing capacity fuses. These fuses are designed with an energy let through or i^2t value which is very closely defined. Since the energy required to damage a given switching device is also carefully catalogued by the manufacturer it is possible to select a fuse which will protect the device adequately. Should the prospective fault current be extremely high then it may be necessary to interpose a series inductance in the circuit to limit this current to a level at which the fuse gives protection.

In motor starting applications care must be exercised by the equipment designer to ensure that his selected fuse will withstand the initial starting surge without failure and this often leads to switching devices being very much over-rated.

CONCLUSIONS

It can be seen by the circuit diagram of the complete three-phase starter contactor (Fig. 8) that it is essentially simple, using the least number of component parts and is therefore capable of

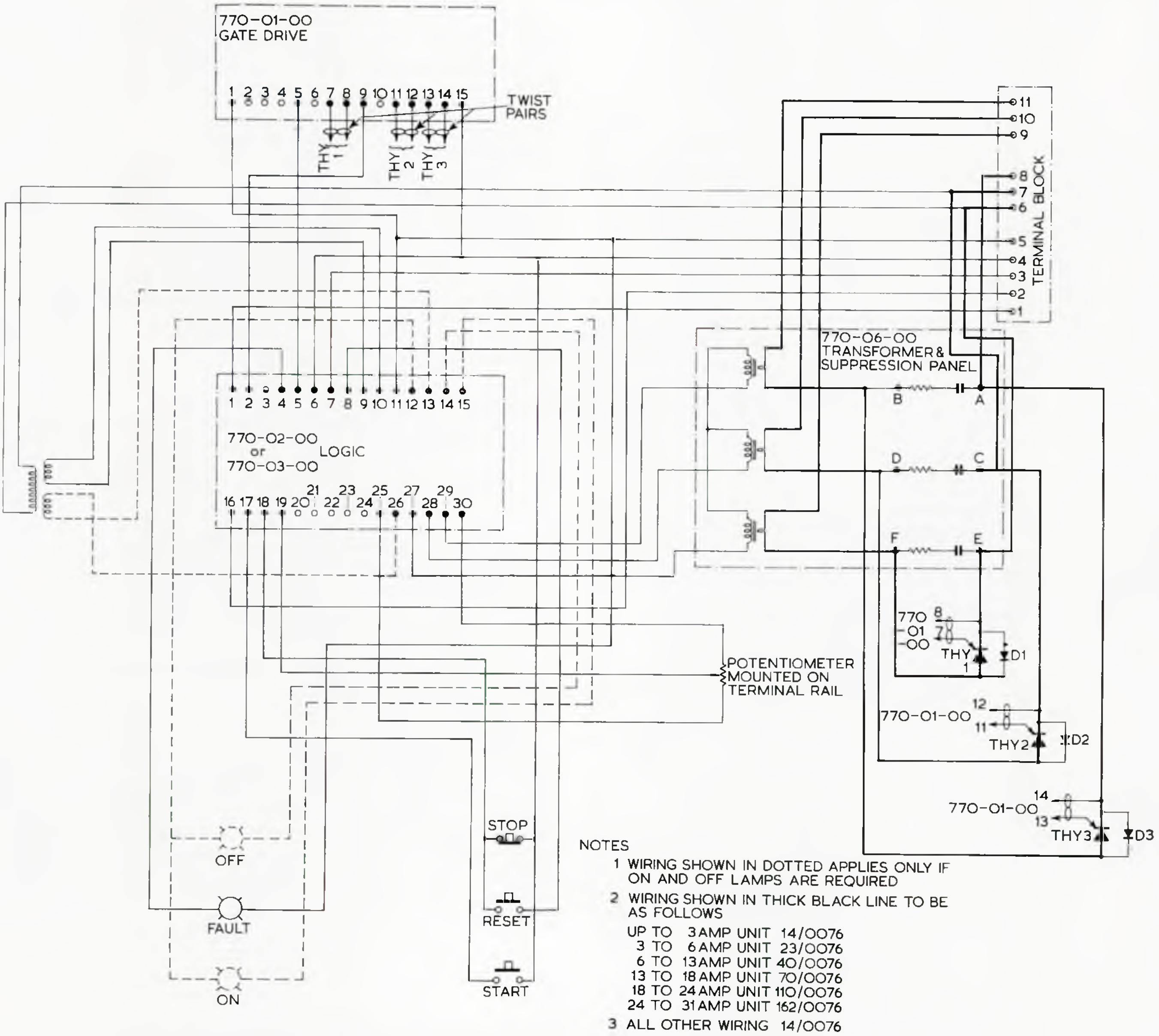


FIG. 8—*Static contactor circuit diagram Mk II*

easy service.

In many applications requiring rapidity of operation in environmentally disadvantageous conditions, or the need for long term reliability in areas where servicing may be a problem, the static contactor is replacing the conventional contactor.

Provided that the applications engineer gives careful consideration to their rating and utilization the static contactor should provide a limitless life.

At present the lower current ratings of these contactors are perhaps five or six times more costly than their mechanical counterparts. This is mainly due to the thyristors, which owing to their necessarily high electrical parameters are expensive. These will reduce considerably over the next few years as they are more easily manufactured. The present small quantity production of static contactors will increase, also reducing costs, which in time will allow a more competitive comparison with conventional starters.

Larger current ratings, particularly those offering ramp start facilities, compare favourably in price. This occurs about the three hundred amps/phase rating.

Disadvantages are mainly that a free flow of cooling air must be kept around the heat sinks of the contactors, especially in the large sizes.

It will, however, be conclusive that this form of motor starting has established its place in modern electronic control systems and has an ever increasing potential.

REFERENCES

1) JONES, D. V. 5 August, 1960, "Designing Solid State Power Relays". *Electronics*.
2) LANGFELDER, R. 22 July, 1960, "Design of Static Relays for Signalling & Control". *Electronics*.
3) BLAKE, R. F. 27 May, 1960, "Designing of Solid State Power Relays". *Electronics*.
4) PRITCHARD, G. W. 2 July, 1970, "Static Switches versus Contactors". *Electrical Times*.
5) GRIFFIN, A. and RAMSHAW, R. S. "The Thyristor and its Applications" (Chapman, Hall).

Discussion

MR. J. GILLESPIE, C.Eng., M.I.Mar.E., thanked the authors for an interesting paper. As far as he was aware, solid state motor starters had not been used permanently in any British-built vessel. Experimental starters had been fitted and operated very successfully for over six months.

The authors had commented on the effect of transient over-voltages on these devices. On any marine installation the incidence of voltage spikes was purely random and could be caused by the switching "on" of quite small motors. One known case of a 0·7 kW (1 hp) motor feed by a relatively long feeder had produced voltage spikes of over 2100 V.

It would appear from the information available that all marine static starters must be protected against spikes of this magnitude.

As voltage transients could be caused during any switching operation, could the authors give some indication of the attenuation which took place in a conventional marine radial supply system?

Comment was also made on the sensitivity of the devices to heat. In the marine environment, starters might be fitted in boiler rooms, or adjacent to boilers, and also on the open deck in direct sunlight. It would appear that static starters for these positions would have to be named in any specification as the starter maker was often unaware of the siting of any given starters.

With regard to the life of the starters, the authors had come right to the point when they stated that this was generally concerned with the life of capacitors. It was to be hoped that all manufacturers of these starters had the same outlook.

He would like to hear the authors' experience with regard to the leakage current with length of service. It had been stated that in theory the leakage would not increase with life in service. Was this so in their experience?

In Fig. 7 a method of starting a squirrel cage motor with reduced starting current was suggested which would also reduce starting torque. Using this method of starting a winch, as mentioned by the authors, he would like to have their comments on the harmonics which would be caused by this form of starting. It could be that the winch load would be the major part of the total load while the vessel was in port.

It would also be of interest to hear—using the voltage chopping technique—of the effect on radio communication. From the information available it appeared that maximum interference occurred when the thyristors were "chopping" at approximately 90°. When chopping at this angle, the interference which had been recorded covered the 0·015 MHz to 6 MHz band and, at or near the equipment, exceeded the limits set in BS 1597. As the limits given in BS 1597 appeared to be under review at the moment and the interference levels lowered, it would be interesting to hear of the experience of others, with radio communication interference.

Finally, on cost of the equipment, the last time this had been investigated it had appeared that a static starter would cost some three times that of a contactor type. Had this margin been narrowed?

MR. N. W. G. HICKS said that the authors had referred to controlling the gate firing circuit so as to give an adjustable firing rate to the thyristors. He agreed with the previous speaker that this would produce radio frequency interference and wondered whether the authors had had any contact with users in the marine world in order to obtain their view on this point.

Under the paragraph headed "serviceability" it was stated that the components for contactors from 10 to 150 amp/phase rating were similar with the exception of the fuses and the thyristors. He would like to know what other component changes were necessary below 10 amps and above 150 amps and why.

With reference to Fig. 8, concerning the circuit diagram, he would be interested to know why flexible cables had been used throughout, and not solid core.

MR. H. RUSH, G.I.Mar.E., said that his interest was mainly in three-phase starters for the squirrel cage motors. Like every other marine design engineer, he was always pleased to see a development which had prospects of increased reliability and, perhaps, less frequent maintenance requirements.

He posed a number of questions; the first of these was partly concerned with the ship's supply system. The ship's system was not an infinite bus system as one might expect in some industrial applications. Did the authors envisage any problems which might have to be taken into account by the designers of the a.c. generators, in particular because of harmonic distortion, where there was a large number of static starters, and also at the switch on or off of a large motor, which might form a substantial part of the ship's load? Secondly, would it be necessary for a squirrel cage motor to be derated for use with one of these starters?

The third question concerned the actual controls of the starters. It had been implied that solid state were susceptible to external interference and he wondered if it was necessary, when remote controls were used, to employ screened cable from the switching area?

Mention had been made of the necessity of adequate heat dissipation. Because of the emphasis that had been placed on this he assumed that the loss might be higher than in a conventional starter. Were there any figures available on this?

His company's policy was to make the major motor starters into a grouped starter arrangement. Was this still feasible with static or solid state starters, and did they have to take any particular cooling precautions?

He said that his knowledge of solid state starters was very limited, but from the information gathered from various sources, the only designs being marketed at the moment were really for the smaller sizes. This seemed strange because the larger sizes would perhaps appear to be more competitive and also give one the opportunity to take advantage of the ramp starting circuits.

With regard to protection, he noticed in the paper mention of protecting the starter from overheating by use of temperature detectors. Did the thermistor type of motor protection become an ideal application when using solid state motor starters?

Referring to standards, while it was accepted that there were apparently no standards in existence at present for these types of starter, he would like the authors to say whether there had been any consultation with standards organizations or with marine classification societies and, in particular, whether they had been carrying out any environmental testing to perhaps Lloyd's Register or Det Norske Veritas requirements.

Authors' Reply

Mr. Pritchard referred first to the questions put forward by Mr. Gillespie on high transients. It was known, in fact, that the transient problem relative to ships was probably higher than one would normally experience in industrial applications. In these particular cases, they had to look at the transient problem very closely out of necessity. In normal industrial environments, one could find arc welding going on adjacent to a machine tool controlled by static contactor, which led to very high transient conditions. In these particular cases they had normally managed to eliminate these conditions by the use of selinium surge suppressors and other suppressor circuits.

With regard to leakage, they had not found that in the period they had been making static contactors—over the last six years—that there had been an increase in leakage current. Provided the devices did not deteriorate—which was quite rare—they did not get any increase at all.

With regard to the problem concerned with ramp starting, he agreed that on voltage chopping this was fed back into the supply, and where this supply was being generated by an alternator, this could have some adverse effect on the alternator if it was not specifically designed for this. They had dealt with this problem very successfully in mine winching, where previously, trucks had been hauled with a wire hauser. The hauser would be hooked onto the truck, the motor switched on and one would get a direct on-line start which would pull the cable smartly. This had been a major cause of accidents in mines. With relatively large static contactors with ramp starting they were able to control this very effectively.

Radio interference was a general problem with electronic equipment and where the rate of switching was high, one was in fact chopping and interfering with the wave form. They had had many consultations with the General Post Office in regard to normal environments where this could cause interference either throughout the factory or in adjacent areas. They had fitted various forms of suppressors and had gone some way towards solving the problem. They had not looked very much at the application to ships other than some work done in conjunction with the Ship Research Association, when they had deliberately made a series of power controllers enabling them to chop wave-form without any suppression at all. These were apparently put in ships, in the galley, so that they did not interfere with the normal important functions of the ship and could be switched on and off to determine what effect they had in regard to radar and the ship's radio. He thought work was still going on in this direction, and they had not as yet received any results.

Turning to Mr. Hick's questions on serviceability, below ten amps one could mount the devices in a slightly different fashion. They could be mounted on the side of the container; it was not necessary to mount them on a heatsink assembly. At 150 amps, however, due to the ambient environment, it was sometimes necessary to force-cool the devices.

As far as cables were concerned, they invariably used flexible cable because when fitting contactors into smaller housings it meant that one was always endeavouring to get cables into awkward places. There was no reason at all why bus bars could not be used as in other electronic equipment. In fact they used bus barring quite extensively.

With regard to Mr. Rush's questions about losses in these equipments, it could be taken that on each thyristor there was a one volt drop. If the drop was multiplied by the current it would be understood from where the watts which had to be dissipated were derived. So there were, to some degree, more losses than with the conventional contactor.

As far as squirrel cage motors were concerned, they had never found it necessary to derate at all. Once switched on, it was just like straightforward mains supply. If, however, one was ramp starting fairly rapidly and pulling the starting current of the motor for a longer period, one would then need to look at the question of lifting the frame size to accommodate the extra heat derived therein.

Where remote controls were in use they had not found it necessary to use screen cabling, because the main control aspect was within the starter anyway. These control cables were really only switching lines. However, where one wanted a manually controlled ramp and the control means remote, it would be necessary to have a screen cable.

On thermistor protection, it was possible to use the signal from the thermistor in the motor directly into these contactors, since the logic was already contained therein and would accept it.

As far as standards and environmental tests were concerned, they provided equipment for use in hazardous areas—areas of high temperature. These equipments were supplied to such places as Kuwait, where the ambient temperature was very high. One could, by selection of the devices and by increasing the area of the heatsink, usually overcome these problems. They were able to operate in ambients of 60° and 70°C.

They had conducted tests for their own information, other than those which were being conducted by the Central Electricity Generating Board on similar equipment. He said they would very much like to see a standard or set of standards established, from which they would be able to work and obtain some form of guidance. Usually, once these things had been established, one could use and incorporate them to give better design.

Mr. Highe referred to the point made by Mr. Rush about the generator design, and said that when one was employing phase control, ramp starting was very valid, particularly in the ship where the winch load was a large part of the total load of the generator. The effect was that the harmonic currents caused heating in the generator. The Central Electricity Generating Board had, for land based application, laid down a standard of powers which might be phase controlled on a given supply; he would like to see a standard for ship applications. He believed that the alternator manufacturers could tell one how much load might be phase controlled on an alternator, but the main point about ramp starting motors as opposed to speed controlling motors, was that it was a short-term effect, and the starting time would only be extended to 10 or 15 seconds, which meant that any heating effects were minimal in the time concerned. Unless one was repeatedly starting winches or extending the starting times, the effect on the alternator would be minimal. It might, however, affect instrumentation, as these harmonic currents were not always measured by instrumentation, and could lead to trouble with metering and overloading from this aspect.