# Marine Machinery Failures

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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. Lloyd's Register of Shipping, with its world wide network of surveyors surveying some 11,000 ocean going merchant ships, is in a unique position to know what happens to ships' machinery in service and how the more important casualties are dealt with. It is right that some of this information should be fed back into industry, and one of the ways in which this can be done is for the Society's Chief Engineer Surveyor to present a paper from time to time describing some of the serious cases that have come to his notice.

The present paper, which is by no means comprehensive, deals mainly with cases which impinge on design, construction and materials. Some of the failures mentioned in the paper were due to causes which may not have arisen previously; others are of a type reasonably well known but which nevertheless continue to recur, either because their importance is not appreciated or the lessons that they teach have not been sufficiently studied.

## ENGINE ROOM EXPLOSIONS AND FIRES

In December 1951 Lloyd's Register of Shipping first issued Rules concerning the provision of spring loaded valves or other devices to relieve crankcase pressures in the event of an internal explosion. Prior to this date several costly explosions had occurred with severe loss of life. These Rules applied only to ships constructed after this date, and by 1954 such devices would have been fitted in all new construction built to the Society's class. However, during the last six years there have been 30 explosions in the crankcases of main oil engines which have caused the deaths of 12 persons and injured 13 others. In the same period there have been 22 explosions in auxiliary engines, and in one case the engine room was completely destroyed by fire.

Inquiry into these cases showed that where serious damage and loss of life had occurred there were three contributing factors: either

1) explosion relief devices were not fitted, the ships having been built prior to the Rule requirement; or
2) such devices had been fitted but were not effective, due either to inadequate design or incorrect fitting; or
3) the devices were not flame shielded.

In the large slow running marine Diesel engine the normal crankcase atmosphere in daily service consists of air and oil globules which are not explosive. Explosive conditions arise when the oil is vaporized by heat from some source so that a white mist or smoke collects in sufficient quantity to form an explosive mixture, and explosions occur when the mist is ignited, generally by the same hot spot which vaporizes the oil. In the majority of cases the hot spot arises in the running gear—bearings, pistons, thrusts, etc.—but there are the odd cases where flame reaches the crankcase from some other circumstances, such as a burnt piston, and there is one record of a scavenge fire being followed by a crankcase explosion in association with defective diaphragm glands.

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Where engines are provided with explosion relief valves, of adequate size, and the crankcase doors are securely fastened, no damage to the engine should result from an explosion, although considerable local damage may by then have been suffered by the overheated parts of the engine, which were the cause of it. The installation of a smoke detector makes possible the avoidance of both the local damage and the explosion.

Smoke detectors not only indicate that the temperature of some part of the running gear of an engine is above its normal service level, and thus enable suitable action to be taken before serious damage to the overheated part can result, but at the same time they call the attention of the engine room staff to the existence of abnormal conditions in the crankcase long before these approach the stage at which an explosion could occur, thus eliminating a serious hazard to personnel which, in adverse circumstances, may result from the flame discharge from the relief valves at a crankcase explosion. Unfortunately, there is as yet no practical flame trap available for these large engines, and flame guards or deflectors, though essential fitments, are not always effective in dealing with large volumes of flame. Where deaths and injuries have resulted from crankcase explosions they invariably have been due to burns.

A recent case of explosion and fire in the engine room of a tanker resulted from an entirely different cause. In this case, although steam and exhaust lines for cargo oil heating coils were fitted leading to the cargo tanks, the coils themselves were purposely omitted. Since these lines were open ended, the opening of certain valves in the lines, accidentally or otherwise, would allow explosive gas from the tanks to pass into the observation tank in the engine room. This, in fact, happened on more than one occasion and heavy explosions resulted with severe loss of life. As a result of this, the Society now requires that on new ships where heating coils are omitted, the steam and exhaust lines must be blanked off forward of the machinery space bulkhead.

On the general question of fires in engine rooms it cannot be too often stated that every precaution must be taken to prevent leakage of oil in and around such places where heating or sparking (as from electric motors) could cause a fire or explosion.

TURBINE ROTORS

A problem which, judging by recent records, continues to cause anxiety is that of bent turbine rotors. While much attention is now being paid to the provision of satisfactory glands and adequate lubrication, since 1955 there have been eighteen cases of bent rotors in main machinery in ships classed with Lloyd's Register. Of these, sixteen were in British built turbines. In the same period there have been five auxiliary rotor failures. Reasons advanced for the earlier troubles of this kind included improper "start-up" and "shut-down" conditions causing rapid changes in temperature in the metal of the rotor, and in some cases severe thermal shock due to inadequate drainage prior to starting up. Rubbing in the glands during manœuvring due to insufficient axial clearance also accounted for some of the trouble.

In the majority of the more recent failures, however, the bending of rotors appears to be due to other causes associated with defective workmanship, and breakdowns occurring during the operation of the machinery, such as, for example, the loss of lubricating oil.

Defects in lubrication systems have been responsible for other serious failures of turbine rotors. Fig. 1 shows an h.p. rotor that was badly grooved during the maiden voyage of the ship, the grooving being caused by foreign matter that had been left in the lubricating oil system. Steel cuttings, weld spatter, bolts, and even a steel wedge, were found in the oil passages.

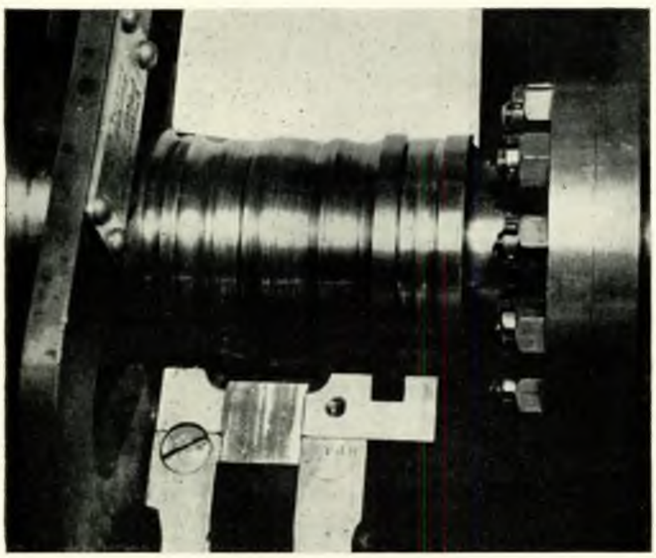


FIG. 1—*Grooving of an h.p. rotor shaft*

A more serious case is shown in Fig. 2 where the rotor shaft of an h.p. turbine had been almost completely severed by the grinding action of an accumulation of carbon in the oil labyrinth box. The carbonization had initially clogged up the drain holes in the labyrinth box and in consequence carbon build-up steadily increased. The maximum operating temperature at the oil gland was measured as about 275 deg. C. Laboratory experiments with the turbine lubricating oil revealed that in small quantities it would carbonize very rapidly at a temperature of about 230 deg. C. The primary cause of failure appeared to be paucity of oil circulation through the gland. Small quantities of oil passing through the drain holes at 275 deg. C. would carbonize rapidly and restrict the oil flow and a build-up of carbonaceous matter was then inevitable.

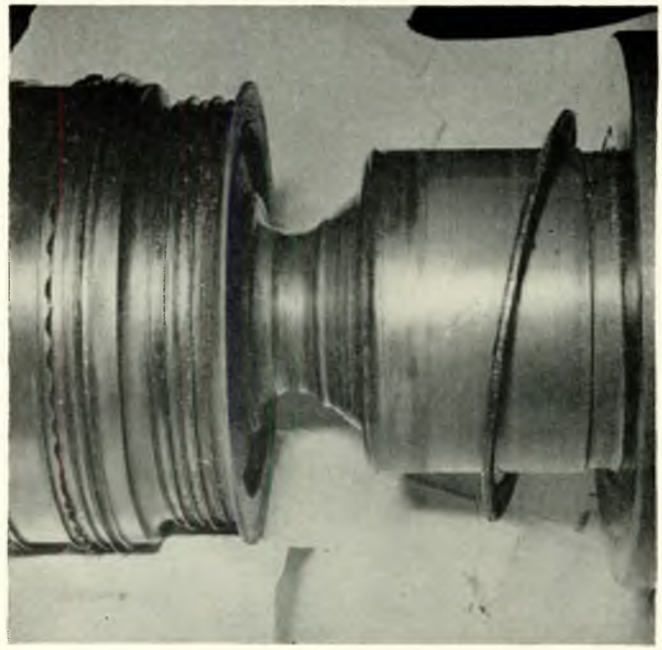


FIG. 2—*H.P. rotor shaft worn through carbonization in labyrinth box*

Enlarging the holes and using an oil that carbonized less rapidly at high temperature have prevented a recurrence of this trouble. A similar failure occurred in a sister ship where the thrust end of the shaft with its two journals was completely severed from the turbine rotor body, and when the installation was shut down the shaft was seen to be red hot. At least two other cases of this trouble are known to have occurred.

Occasionally an undesirable operating condition is discovered during trials or in service when this might have been avoided at the design stage. An example concerns the heavy vibration of an h.p. turbine due to the critical whirling of the ahead and astern rotors (Fig. 3). Subsequent calculations

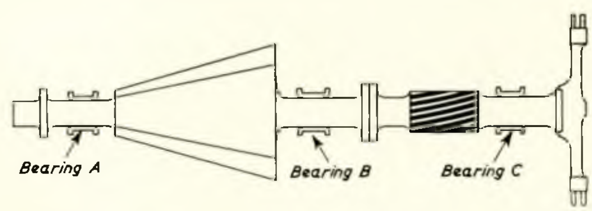



FIG. 3—*Configuration of whirling h.p. rotor assembly*

showed that the critical whirling frequencies of the ahead and overhung astern rotors were practically identical and occurred near the service speed. Remarkable improvement was obtained by reducing the length of the bearing carrying the astern rotor (Bearing C) and lowering the frequency of the overhung system, which was now detuned in respect of the ahead rotor. Bearing loads remained within the permissible limits and the damping in the system produced by the pinion and bearings was sufficient to limit the whirling of the astern rotor to insignificant proportions.

A spectacular failure occurred to the main turbo-alternator set of a tanker. During a dock trial when the speed was being gradually increased to check the overspeed trips of the turbine, increasing vibration was observed at the forward pedestal bearing. With the set operating at about 4,000 r.p.m., sparks were thrown from this bearing and the unit disintegrated (Fig. 4). The turbine and generator shafts were broken and

FIG. 4—*Main turbo-alternator failure*

there was considerable consequential damage in the engine room.

Examination of the damaged parts showed that the shafts had bent to fracture. The bolts in the coupling between the shafts had also failed either under combined tension and bending or by shear. It appeared that bending about the coupling had been responsible for the failure of some of these bolts, the remainder being sheared as a result of torsional overload. When a shaft having a high angular velocity starts to bow, the deflexion increases as the shaft continues to rotate and retains its position and sense of direction. Although confirmed by metallurgical examination, the deformation suffered by the shafts and the coupling before fracture was evidence of the satisfactory qualities of the materials of these parts.

There was no evidence that the set failed as a result of gross overspeeding. Most of the components of the thrust and governor drive were recovered from the debris. It was noted that the security of the thrust assembly had depended upon a tab washer which locked the retaining nut (Fig. 5). The conditions of the threads in this nut and on the shaft indicated that the nut had slackened back before the shaft was damaged (Fig. 6). It was considered that this was the primary cause of the breakdown. As the tab washer was not found, despite a careful search of the wreckage, it could not be

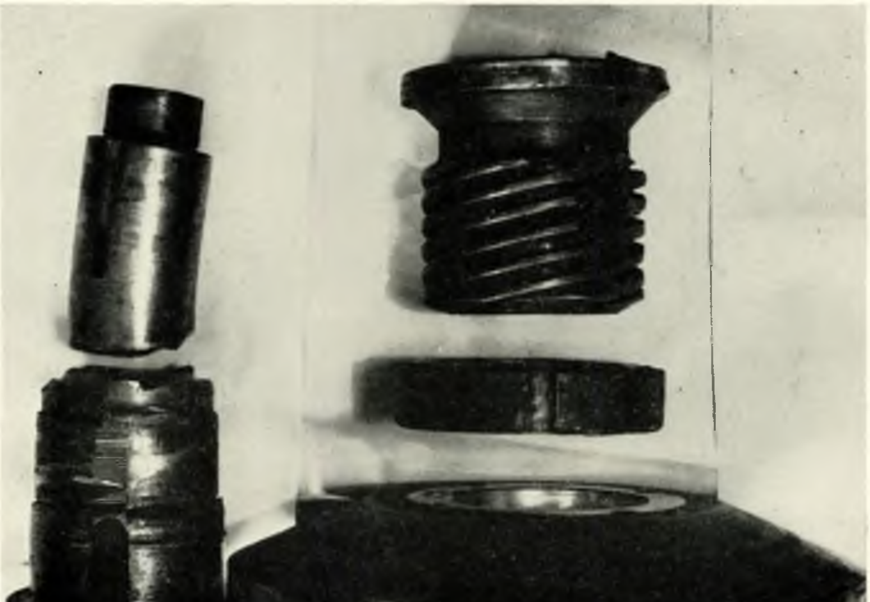


FIG. 5—*Damaged governor drive of turbo-alternator*

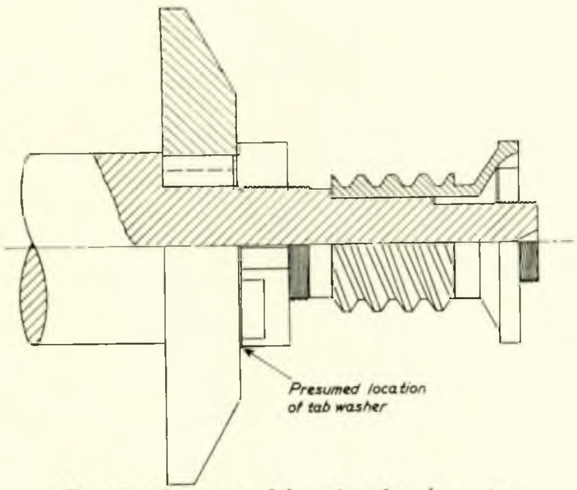



FIG. 6—*Governor drive of turbo-alternator*

established whether this washer had failed during service. It is interesting to note that different locking arrangements for the thrust retaining nut were used on some similar sets. As a result of this accident steps were taken to check the security of the thrust assemblies on all sets of this type.

Irrespective of the type of machinery it is necessary to ensure that locking arrangements for threaded components are efficient. Tab washers, split pins and locking washers and wires should be replaced after the threaded connexions have been separated and before reassembly. There should be no freedom for the nut to move and so damage the locking device. Undersized split pins and tab washers, slack locking wires or wires looped around a nut so that slackening is not prevented, can have disastrous consequences.

GEARING

There have been a number of cases reported over the last few years where gear wheel rims have moved axially and sometimes circumferentially on their centres. In one particular case, during sea trials, two out of four primary wheel rims moved $1\frac{1}{2}$ in. axially and, judging by the surface marking on the wheel centres, had made several revolutions on the centres to get there (Fig. 7). There was no question of oil starvation to the bearings or gear mesh as temperatures had remained normal and the gears had been carefully run in. The designed interference fit was about 0·8/1000 inch per inch of diameter (1/1200 of diameter) which gave a theoretical nominal factor of safety of over 10. It appeared that the most likely cause of the failure was an insufficient interference fit during the construction of the gears. However, gaugings taken from the rim and centre showed that the interferences were correct. It was noticed on examination that while the bore of the rims was very smooth, the surfaces of the spheroidal graphite iron centres were very rough turned. The nature of the material and surface finish may not have been conducive to an effective shrink grip, the centres being of a softer material, and it is most probable that there were considerable areas not in contact. In this instance a cure was effected by dowelling a plate both to the centre and the rim, and this proved quite satisfactory.

The efficiency of shrinks is governed by several factors. Surface finish and condition on assembly are important as the coefficient of friction between oily surfaces is 20 per cent under that for a shop dry surface and one-third of that for chemically dry surfaces. However, interference fits of 1/1200 of diameter are in common use with shop dry surfaces and appear to be satisfactory, giving an adequate margin of safety. The interference of 1/500 to 1/700 as laid down by the Society for crankshaft shrinks would, of course, be too severe in these cases in view of the comparative thinness of the rim and the fact that teeth have to be cut in it when assembled.

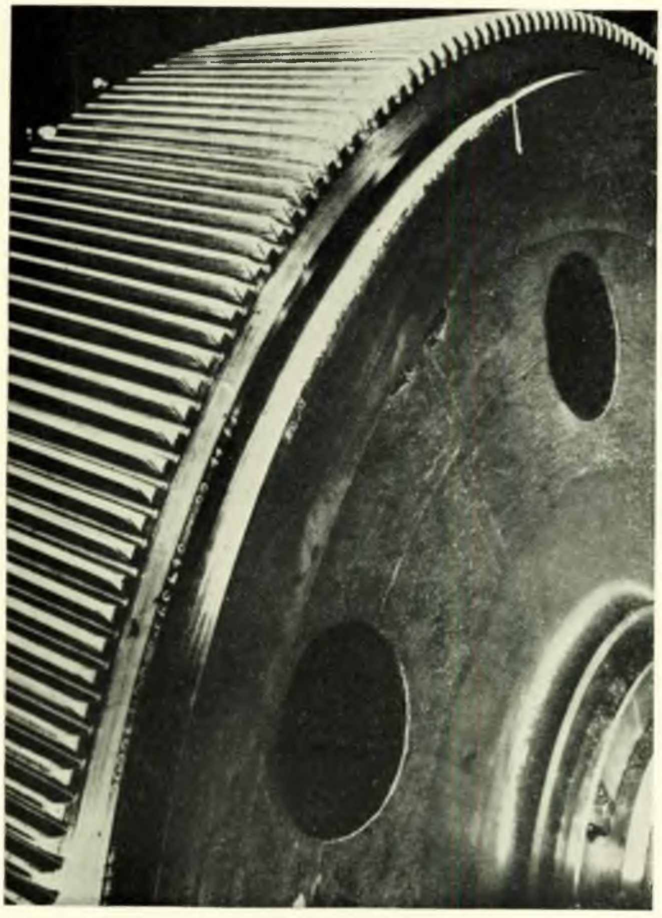


FIG. 7—*Slipped main gear wheel rim*

FIG. 8—*Rim marking from un-filled dowel hole*

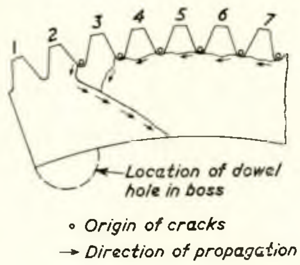



FIG. 9—*Direction of crack propagation*

It is essential that if the shrunk-on rim is to have a uniform hoop stress, the mating surfaces of both rim and centre are unbroken. Examination of a fractured pinion shroud which had less than 900 hours in service revealed no material defects, but a stain on the bore of the rim showed the location of a half dowel hole in the centre (Fig. 8). One edge of this stain was in line with the origin of the crack in the root of tooth number 2 (Fig. 9)—which was the initial point of breakdown. Clearly the unfilled half-dowel hole on the wheel centre periphery would permit local flexing of the rim, and this could result in failure by fatigue at the stress concentration represented by the radius at the root of a tooth. New gears with solid mating surfaces have given no further trouble in the ship.

Although some 54 per cent of reported cases of gearing damage concerns the pitting and scuffing of the teeth, there is a further 24 per cent of cases involving complete fracture of the teeth, a number of which were associated with defective material. A recent investigation of a portion of a fractured pinion tooth revealed that although the quenched and tempered alloy steel possessed satisfactory tensile properties, the material had a coarse dendritic structure. This structure indicated a low resistance to shock, some of the primary dendrites extending for the full depth of the tooth (Fig. 10). In this case the fragment broke out of the forward helix of the l.p. second-reduction pinion, passed through and severely damaged the gears. While the nature of the steel may not have been the primary cause of the failure, the structure was undesirable for the material of a high strength, highly loaded pinion. Fortunately it was possible to machine $4\frac{1}{2}$in. off the pinions and main wheel so the ship was able to proceed on her voyage at reduced speed.

Cracked wheel rims became too prevalent in Germany immediately after the war. On eight ships of 18,000 to 28,000 tons, cracked main wheel rims were reported after only six months to two years in service. In some cases replacement rims from the same source also failed. The steels were of a type that was not widely used for the manufacture of marine gear wheel rims. It was a medium carbon, 1 per cent manganese, 1 per cent silicon type in the toughened condition. Examination of these manganese silicon steel rims showed fine

FIG. 10—*Gear tooth showing dendritic structure*

inclusions on the machined outer surfaces, and supersonic tests confirmed the presence of further inclusions in the body of the steel.

Comprehensive tests were carried out on two of these rims. They showed that the steel had a U.T.S. of 41-45 tons per sq. in. in the circumferential direction, with a yield of 26-28 tons per sq. in. Bend tests showed about 30 per cent failure. Fig. 11 shows a magnetic crack test taken on a section of a rim, with an associated sulphur print showing extensive segregations in way of the cracks. Fig. 12 shows a magnetic crack test on a prepared bend specimen.

Large ingots of this manganese silicon basic open hearth steel are susceptible to segregations which are detrimental to rim forgings made therefrom. To avoid further failures, sulphur printing the ends of the wheel rims as well as pinions is now a Rule requirement of Lloyd's Register of Shipping. In doubtful cases magnaflux tests are also carried out after hobbing.

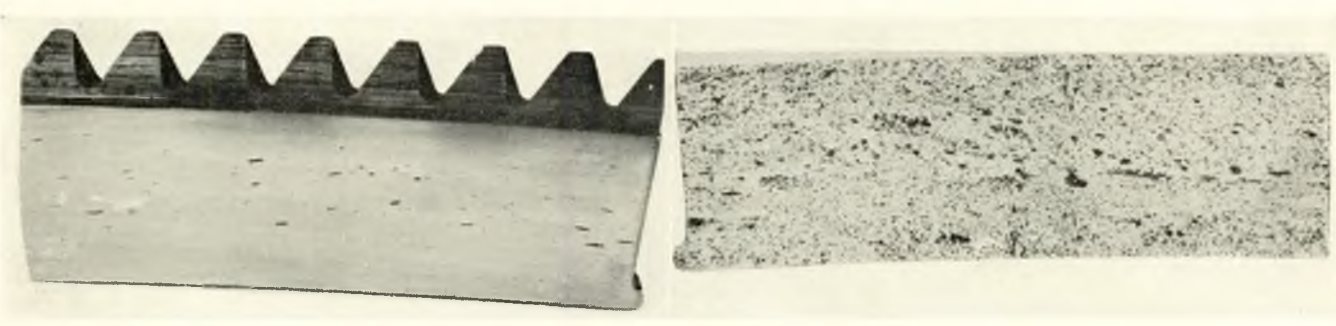


FIG. 11—*Segregation in gear wheel rim*

FIG. 12—*Magnetic crack test on pieces cut from gear rim*

WELDING

The welding of pressure vessels necessitates a high degree of quality control, which was first established when Lloyd's Register introduced Rules for Welded Pressure Vessels in 1934. Prior to that date there were no rules or regulations issued by any authority in the United Kingdom governing the quality of pressure vessel welding, and there can be no doubt that the standards laid down by the Society prevented serious accidents occurring due to defective welds. Accordingly, the author can point to no weld failure of a boiler drum in service where the welds have been made under Lloyd's Register Class I conditions.

The position is by no means so satisfactory where welds that may be termed "incidental" or even "indiscriminate" are concerned. Indiscriminate welding of attachments to boiler shell plates is to be discouraged and the sealing of riveted joints by a light fillet weld could lead to serious trouble. In one case a seal weld between the double butt straps and shell plate of a Scotch boiler was used instead of caulking. When this boiler was put under hydraulic pressure the circumferential stretching that took place threw a considerable load on the small fillet weld, and this, combined with initial cracking in the fusion zone, caused the boiler to rupture catastrophically. The shell plate tore along the lower edge of one of the riveted longitudinal butt straps for the full length of the boiler. Part of the small sealing fillet weld laid along the edge of the strap is shown in Figs. 13 and 14. This failure clearly demonstrates that small fillet welds should be avoided in riveted joints where they impose undesirable restraint.

The problem of cracking in Diesel engine bedplates of welded construction is one which has concerned the Society for some time, and in 1958 Rules were formulated in an endeavour to overcome the difficulty. In most of the cases reported the cracking had been confined to welds in transverse girders of the bedplates. The defect was common to a large number of engine types constructed by many different engine builders, and it was clear that the trouble was widespread. The new Rules introduced some measure of control of bedplate fabrication without placing too much restriction on design. Joints in components carrying main gas loads are now required to be continuous full strength welds. Intersection of welds and abrupt change of plate section have to be avoided, but perhaps the three most important requirements are accurate preparation of plate edges for welding, stress relieving heat treatment of bedplates when welding is completed, and the quality of plates, sections, forgings and castings to be used for welded engine structures.

Experience has shown the advantage of a number of features in welded engine structures which are not, however,

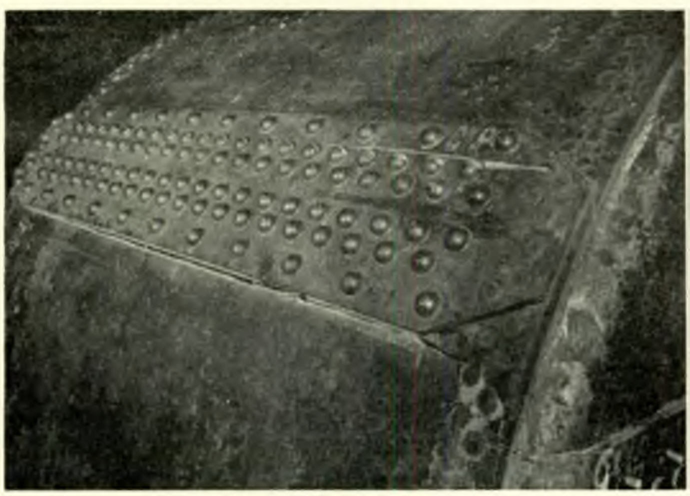


FIG. 13—*Cracking of boiler shell due to welding*

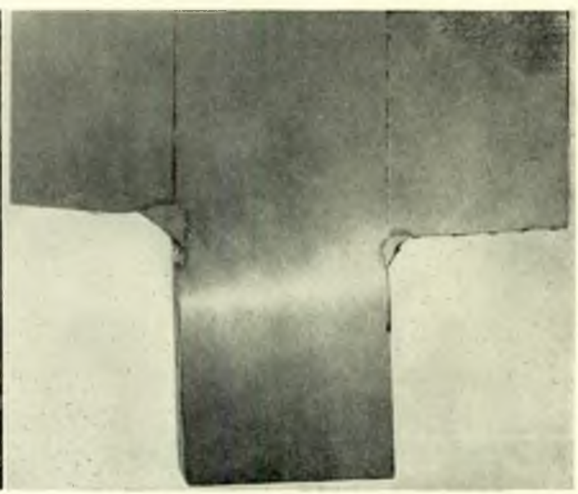


FIG. 14—*Section across butt-strap, showing fillet welds*

universally adopted. The size and strength of cast steel bearing housings have considerable influence on the structural efficiency of the bedplate, and where such a casting has been designed as an integral part of the transverse girder, then the possibility of fracture is considerably reduced. Such castings should be formed with web extensions which are butt welded to the flange and vertical web plates of the transverse girder.

Double-weld butt type joints should be adopted wherever possible in view of their superior fatigue strength. In large bedplates, double-plated transverse girders have an advantage over single-plate girders, especially in way of the bearing housings, since a much stiffer girder is thus obtained and better support given to the bearing. Again, if long through bolts are used, passing through the cast steel bearing housing and securing the upper entablature to the bedplate, a more even distribution of static and gas loads will be achieved. Access holes should be as small as possible in order to avoid cutting out too much material that would otherwise help in carrying the loads. If access holes are flame cut their edges should be finished off by machining or grinding so as to have no rough spots where a crack might originate. Figs. 15(a) and (b) illus-

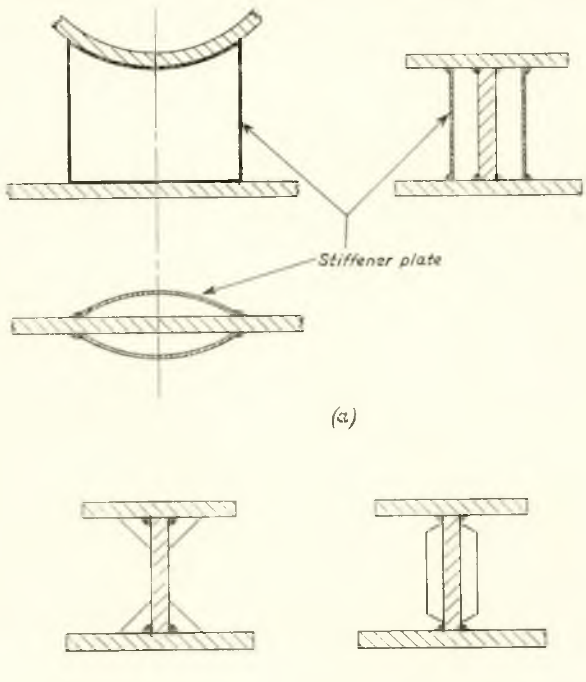



FIG. 15—*Stiffener of fabricated girders: a) inadvisable; b) preferable*

trate a principle concerning the attachment of stiffener plates to fabricated girders which have to carry bearing loads. In Fig. 15(a), curved stiffener plates are shown welded to the web plate of a transverse girder. These stiffeners are welded not only to the web plate but also to the top and bottom flanges of the girder, and therefore, besides acting as stiffeners, are called upon to take a share of the direct load. Where stiffeners are intended merely to reinforce the web of a girder, they should not be tight between the flanges, but as in Fig. 15(b). In all bedplate design and construction where it is intended to replace castings by welded fabrications in order to save weight, it is essential that the saving should not be made at the expense of rigidity and stiffness.

Although the standard of welding on new construction is usually high, the same cannot always be said concerning weld repairs. Too frequently indiscriminate welding is used to repair damage caused basically by poor design—where the effect of welding will only magnify the risk of repeated failure.

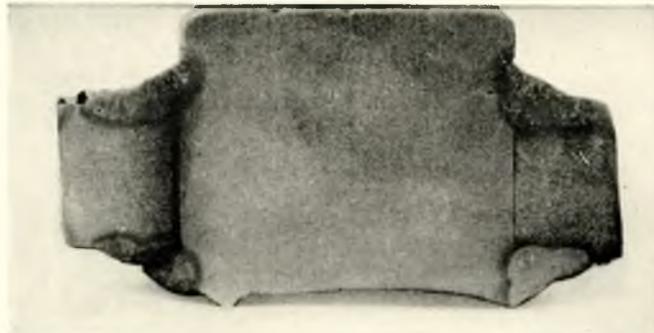


FIG. 16—*Welded boiler stay*

In considering any weld repair, the poor fatigue strength of casual welds must be appreciated. Welds should only be positioned in regions where the dynamic stresses are of a low order. Where welding is carried out on a large component, preheating is essential to avoid the cracking that can accompany a rapidly quenched weld. This is especially true when welding some low alloy carbon manganese steels where, unless a preheat of about 200 deg. C. is applied, cracking can be expected in the heat affected zone.

An example of unsatisfactory welding, this time on new construction, concerns the longitudinal stays in the auxiliary boilers of a 10,000-ton tanker. These stays in the steam space were welded, inside and out, to the front and back end plates. All the stays in the port boiler and four in the starboard boiler fractured in way of these welds. Mechanical tests and sulphur prints taken from the parent material showed it to be of satisfactory quality. It will be seen from Fig. 16 that the clearance between the stay and the hole in the shell plate was almost completely on one side of the hole. In all cases the fillet welds showed a considerable amount of undercutting and contained root cracks (Fig. 17). There is no doubt that the unsatisfactory quality of the welds was responsible for the failures.

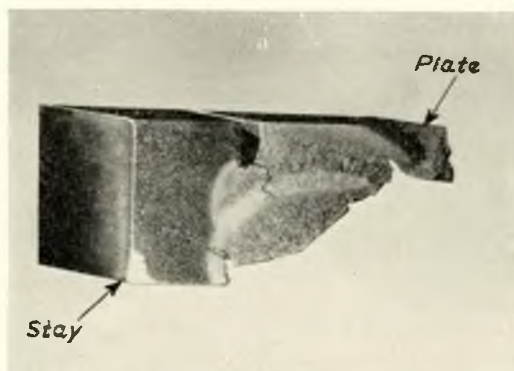



FIG. 17—*Undercutting at boiler stay fillet weld*

Tack welds for brackets, lugs, etc., and accidental stray flashes, are a potential source of failure and care should be taken that these do not occur on normally highly stressed areas. Several major failures have been reported as originating from such welds. Where tack welds are used for location purposes they should either be placed so as to be incorporated in the final weld or should be carefully machined off after serving their purpose.

It has hitherto been considered inadvisable to repair castings by welding, especially where they are subjected to dynamic stresses such as occur in crankwebs and top end beams. With recent improvements in welding materials and techniques, there is a greater possibility of effecting satisfactory repairs, and

Lloyd's Register are at present carrying out tests by which the conditions of acceptance of such repairs may be established. It goes without saying that the highest quality welding will be demanded under stringent control conditions.

FLAME CUTTING

The chief danger arising from flame cutting of plates prior to welding is that notches or cracks are frequently left in the overheated zone. Fig. 18 shows a section of plate 1-in. thick that had been flame cut and which fractured along its free edge whilst being welded along its opposite edge to another plate. Two cracks can be seen originating at irregularities on the flame cut edge. Similar cracks have occurred during the bending of plates having flame cut edges.

The effect of the process is to increase surface hardness and induce residual stresses which aggravate the possibility of cracking due to stress concentration at the notches in the flame cut edges.

FIG. 18—*Cracking of flame cut plate*

For plates which have low notch toughness at ambient temperatures, the tendency for small cracks to propagate is greatly increased. It is essential that plates and slabs that have been flame cut to size should not be either cold formed or put into service before the damage is rectified. This also applies to flame cut openings, etc., in plates and slabs. Lack of attention to rectification has resulted in the failures of crankwebs, transverse beams of oil engines and boiler shells. The extent of the damage by flame cutting depends on the thickness of the section and the type of steel. For thin mild steel plates the removal of $\frac{1}{8}$-in. from the flame cut edge by either machining or careful grinding is usually sufficient. For thick slabs, at least $\frac{5}{8}$-in. should be removed by machining. However, to avoid cracking between flame cutting and machining operations, thick slabs and alloy steel plates should be heat treated immediately after the cutting operation. The best practice is to charge the flame cut component while still hot into the furnace.

CRANKSHAFTS

The failure in service of crankshafts usually occurs at mechanical notches such as fillets and oil holes. Failures are mainly of fatigue type and arise due to local increase of the alternating stress to the endurance limit of the material. The effects of mechanical notches on the fatigue strengths of steels have been discussed and investigated by many authors, and improvements in design and manufacture have greatly reduced the incidence of such troubles due to notches alone.

Inadequate radiusing of fillets and oil holes proved to be the origin of many fatigue cracks in crankshafts, and the Society now requires that on solid forged cranks, the radius of the junction of webs with pins or journals shall be not less than 5 per cent of the shaft diameter. This figure has been derived from fatigue tests carried out by the Society who have also investigated the radius at the lip of an oil hole. Though not a Rule requirement, a lip radius of not less than a quarter of the hole diameter is suggested.

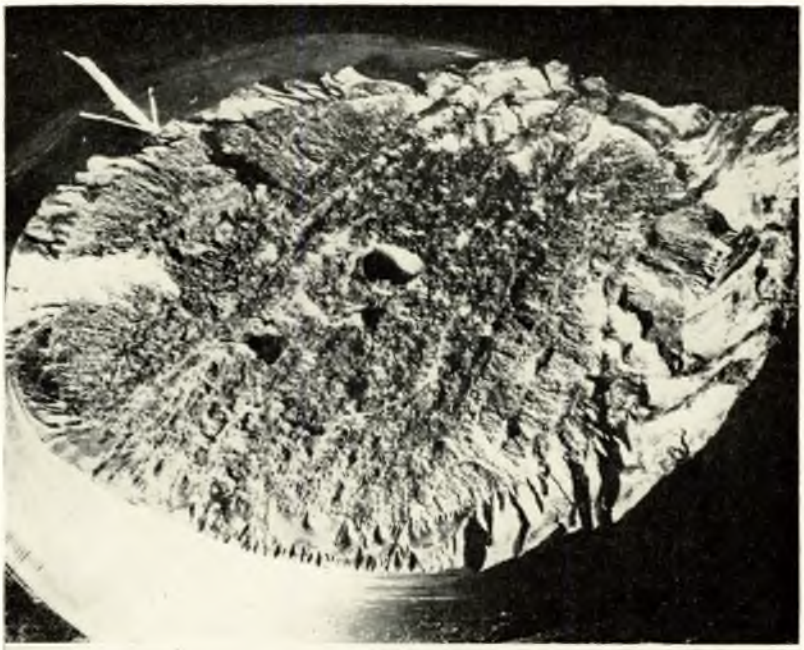
Normally the factor of safety in the design of a crankshaft will take care of shape features and accepted additional loads that may arise during service. But cases have occurred where the shape of a crankshaft has appeared to be reasonable but the shaft has proved to be unduly sensitive in service. Such sensitivity can occur where the size and power of a standard design of engine are increased and the component parts of the engine are geometrically similar to those which have been proved successful in the smaller engines. The modulus of section of crankwebs, for example, may be adequate by rule and yet due to some feature in design, such as a recessed fillet between web and pin, or web and journal, the effective fatigue strength of the web may be reduced and prove to be inadequate.

Today the majority of crankshaft failures and faults are due to combinations of service conditions (in some cases with material defects), many of which could have been avoided. The factors which influence the performance of crankshafts or crankshaft components, in addition to stresses induced by the reciprocating forces and torque variation, are effects of malalignment, bedplate and hull deflexion (particularly in heavy seas), excessive vibrating stresses, surface corrosion, inherent metallurgical defects and residual stresses resulting from forging or casting, and which, for some reason or another, have not been removed by heat treatment.

The occasional case does arise where a failure is entirely due to defective material which could not be detected by normal inspection of the finished machined shaft. A defect of this kind occurred in the crankshaft of an 8,500-h.p. oil engine. The shaft failed through a crankpin during the maiden voyage and the fracture (Fig. 19) contained multiple origin fatigue cracks extending round the entire periphery of the crankpin to a maximum depth of about 2 inches. The major portion of the fracture was coarsely crystalline with pronounced directional features (Fig. 20). A sulphur print through a section of the pin (Fig. 21) showed severe bands of segregation which outcropped on to the working surface of the crankpin. Tests also showed that the pin had poor mechanical properties. The steel was generally dirty, containing numerous globular oxides, complex slag inclusions and sulphide particles. These undesirable features were the primary cause of the failure of this crankshaft.

Unfortunately this is not an isolated case and the Society has experienced others in which poor quality forgings have proved to be the cause of failure. Faulty material cannot be tolerated in such a component, and ultrasonic testing is strongly recommended to ensure that forgings with major internal defects do not go into service.

It has been felt for some time that in cases of oil engines operating with detuners and dampers at the forward end, there is a very real risk of damage occurring should the detuner or damper fail to perform its function for any reason, especially as such a breakdown may pass unnoticed for some time. On one ship last year the starboard main engine suddenly developed excessive vibration and heavy knocking, and though this was felt throughout the engine, it appeared to be more pronounced in the vicinity of the detuner. The outward cause of the trouble was lack of oil in the detuner due to the accidental leaving off of a small lubrication pipe. The lubricant supply was restored and the engine ran satisfactorily.



FIGS. 19 and 20—*Fractured crankpin, showing crystalline nature of central area*
FIG. 21—*Sulphur print from crankpin*

Shortly following this it was decided to take torsiograph records on two sister ships with similar installations to establish the cause of a knock which had developed in the detuner. During the course of these tests it was proposed to measure the movement of the floating mass relative to its flywheel. This test was carried out with the lubricating oil cut off from the detuner. At 50 r.p.m. a heavy first order knock developed, apparently originating in the detuner. The severity of this knock so increased with speed of the engine that at 75 r.p.m. the test had to be discontinued. On replacing the oil supply, the engine behaved satisfactorily up to full speed. The torsiograph records showed that the two-node frequency was of the same magnitude for tests with the detuner in operation and with the detuner mass removed, but there was a substantial increase in amplitude in the latter case.

In the light of these results it appeared that the detuner was acting as a vibration damper. This was also evident from the adverse effect on the vibration characteristics of the lack of lubricating oil which, in theory, should have little influence upon the detuner other than reducing wear. The presence of oil was obviously having a damping effect on the vibration. Moreover, in a sister ship where a more viscous lubricating oil was in use, the engine knock was noticeably less severe. Further tests were carried out with the damper supplied by gravity feed with a very much heavier oil. The results of these tests showed that not only was the knock and timing chain chatter eliminated, but the torsional vibration stresses were reduced, enabling a previously applied barred speed range to be removed.

### CONNECTING RODS

There have been several cases in which the connecting rods of some oil engines have fractured longitudinally from the edges of oil holes in either the top or bottom forked ends. Investigations have shown that these fractures were due to fatigue and were not associated with defects in the steel. Fig. 22 is a sketch showing a fracture at the top end of a rod. In this and in the other cases the fatigue cracks originated at sharp notches, represented by either unradiused oil holes and/or an adjacent tapped hole on the transverse axis of the fork.

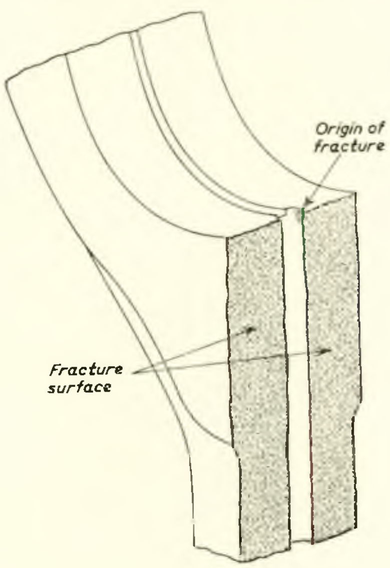



FIG. 22—*Typical connecting rod failures*

The roots of the forks are subject to fairly high levels of alternating stress. Sharp notches and large diameter oil holes are undesirable features at these positions. Where possible, oil holes should stop short of the fork at the bottom of a rod. This practice eliminates the stress concentration. When it is essential to drill the oil hole through into the root of the fork, either at top or bottom end, the diameter of the hole should be kept to a minimum and the lip should have a generous radius. This radius should be greater than half the diameter of the bore of the hole and, where possible, equal to that diameter. The bore of the oil hole should have a smooth finish to a depth of at least 3in. below the surface of the fork.

### FRETTING CORROSION AND FRETTING FATIGUE

The problems that arise with fretting are excessive corrosion and wear, and fatigue failure at relatively low levels of vibration stress. Although these are inter-related, excessive fretting corrosion can occur without failure of a component by fatigue; on the other hand, corrosion may appear to be almost negligible in some cases of failure by fretting fatigue. Fretting occurs where there is relative movement between mating surfaces of engineering components. It is possible that the result, i.e. excessive corrosion and wear or fatigue cracks, depends on the degree of relative movement and the level of vibration stress at the surface of either component.

Fretting may be due to the following processes, either singly or in combination, depending upon the nature of the surfaces:

1) Cold welding together of high spots on the rubbing faces, followed by the breaking of these welds and the separation and subsequent oxidation of the resultant particles.
2) Surface slip, alternating in direction and leading to attrition on a molecular scale; the products of attrition being, in the case of steel, oxidized to a hard abrasive oxide which continues the wearing action.
3) More severe abrasion where the products, receiving considerable cold working and hardening, assist in the process of fretting by gouging and roughening fresh surfaces.

This, then, is primarily a mechanical process, and is not to be confused with pure corrosion, which is an electrochemical process. Fretting, or scoring of surfaces in contact, leaves cavities of two distinct shapes. The first, a shallow depression, is the result of the abrasive oxide debris spreading out from the initial site of the attack; the second, a small but deep hole formed where the debris has become trapped and the attack is continued down into the metal.

Fretting corrosion is frequently found in flexible couplings. Inadequate or contaminated lubrication is sometimes a source of trouble, especially with fine tooth couplings, which require as much lubrication as the gears themselves, and the teeth should be cut and formed to the highest standard. Out of 39 reported cases of coupling defects due to fretting, wear, pitting, etc., 25 of the couplings have had to be renewed. Where at the same time lubrication has been improved, no further trouble has been reported.

Mal-alignment is also a major cause. There is no doubt that special care in the initial alignment of flexible couplings is important in view of the differential expansion which takes place between the turbine and gear casing. Careful honing and scraping have often been sufficient to ease the hard spots in a coupling and so eliminate fretting. In one persistent case the coupling teeth were phosphated to give a hard contact surface and this was found to be most effective.

Coupling bolt failures are often associated with fretting corrosion. In one recent case a failure occurred in the coupling bolts between the intermediate and screwshaft when the oil engine governor failed to function in heavy weather and the engine overspeeded. The shaft coupling faces were badly fretted and torn in way of the bolt holes. Examination of the bolts showed evidence of fretting and corrosion on the surface of the shank and the bolts had fractured inside the screwshaft flange. Fig. 23 shows a typical fracture surface in which the origin of fracture had occurred at one point at the surface of the shank. It is considered that failure of the bolts was due to fretting corrosion fatigue. The fatigue crack had probably been present for some time and the breakdown of the governor

FIG. 23—*Coupling bolt failure*

control of the engine led to a sudden increase of stress in the bolts.

A more common type of failure occurred in two sister ships where coupling bolts failed on the main engine centre coupling. In the first ship considerable trouble had been experienced with slackening and breaking of these coupling bolts, and examination of the coupling showed that the faces were pitted and fretted all over the mating surface to a depth of 20/1000in. Mal-alignment between the two halves of the crankshaft could have been a contributory factor. In the sister ship all eight bolts in the coupling failed whilst the vessel was at sea, fracturing across the parallel portion of the bolts (Fig. 24). On examination it was evident that the original fractures were of the fatigue type, and failure of the bolts was therefore attributed to fretting corrosion fatigue. The dynamic and shear loading of the coupling was not excessive and no cracks were found in either the head fillets or at the roots of the bolt threads—both places of high stress concentration. On one side of the coupling the cracks had developed over a considerable period and as they progressed so the coupling would have become less secure and fretting corrosion would increase at the remaining bolt surfaces.

In fretting fatigue, which occurs in members already highly stressed, the abrasive debris leaves small pits, which, acting as stress raisers, are sufficient to create local stresses above the fatigue strength of the material. Fatigue cracks are formed and propagate from these pits.

Fig. 25 shows part of a major breakdown of a scavenge pump piston. The radial cracks into the piston bore were secondary to the primary cracking which had occurred circumferentially. There was no indication that the cracks had spread from any surface defects or stress concentrations. The material of the piston was satisfactory. Fretting, however, was clearly visible on areas of the piston surface which had been in contact with the flange of the rod. Laboratory tests on an unbroken piston securely bolted on to a base showed by means of sand patterns that the piston had a natural mode of vibration corresponding roughly to the running speed. It is possible that excitation of this mode in service caused fretting at the bolted face and was the primary cause of cracks developing in the piston. There was little fretting oxidation but failure had been promoted at low levels of vibration stress and at the borders of lightly fretted areas.

Many proposals have been made to prevent, or at least postpone, the onset of fretting corrosion, and these may be conveniently subdivided under three headings, namely:

a) *Materials of Surfaces in Contact*
   1) The use of rubbing surfaces of different materials.
   2) One component case hardened.
   3) One component nitrided.
   4) One surface plated with cadmium, nickel or tin.
   5) The use of phosphate coatings.
   6) The use of a gasket to take up small relative motion without metal pick-up.

b) *Fit and Finish of the Surfaces*
   7) Increase compressive load to reduce sliding motion.
   8) Have rubbing materials of different finish.
   9) Use unmachined or roughened surfaces.
   10) Shot peen one surface.
   11) Surface roll one component.

c) *Lubrication of the Surfaces*
   12) Allow an increase of relative motion until film lubrication is obtained.
   13) Improve lubrication by oil grooves and seal off component to exclude air.
   14) Use special lubricants, i.e. aluminium soap, molybdenum disulphide, animal fat, etc.

FIG. 24—*Fretting corrosion failure of coupling bolts*

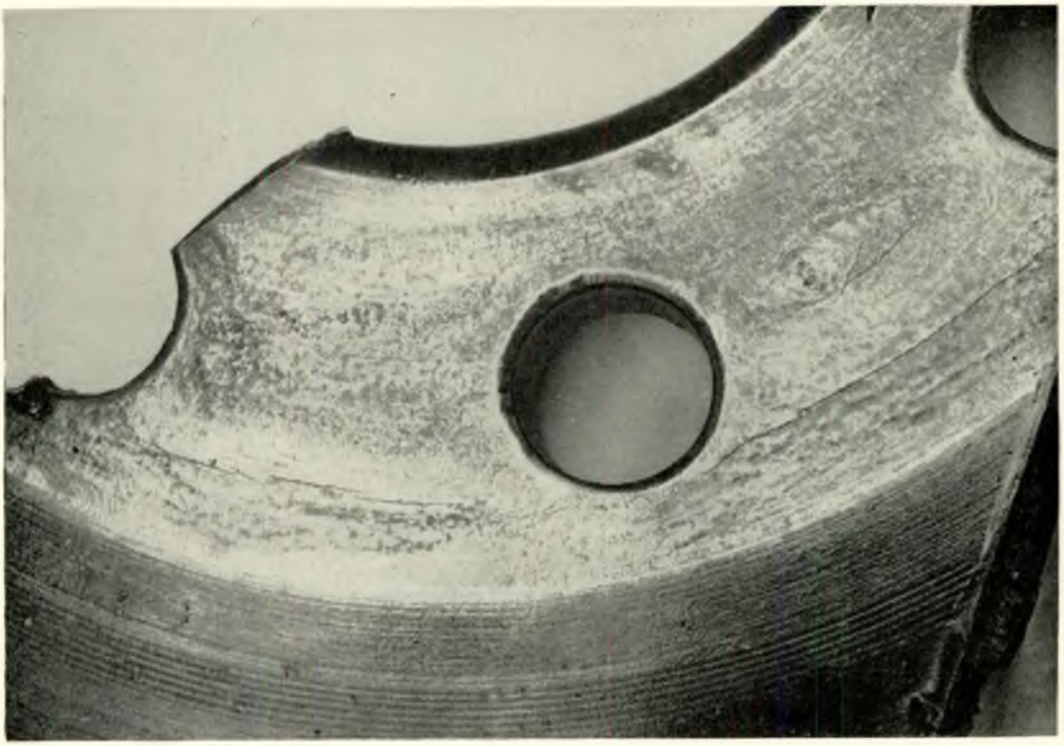


FIG. 25—*Cracked scavenge pump piston*

Obviously the selection of one or more of these preventive measures is governed by design and operating conditions. In any event, it is doubtful whether roughened surfaces have any value except in cases where relative movement is normally slow and small. Surface rolling, nitriding, phosphating, have all been used successfully, as has electro-deposition of nickel. Where larger relative motion may be accepted, gaskets of P.T.F.E. (Polytetrafluoroethelene) can be used. Lubricants in the molybdenum disulphide range, either rubbed or sprayed on to surfaces, have proved remarkably effective. It should be noted that mercuric ointment, in itself of moderate value, must not be used against brass or bronze as the mercury rapidly forms a zinc amalgam and destroys the bearing surfaces.

The problem of fretting fatigue is that while the condition may be alleviated, no satisfactory preventive measures have come to light, other than the elimination of relative movement or the reduction of the vibration stress to very low levels. Cold working of the surfaces in contact, while not preventing the formation of fatigue cracks, slows down their rate of propagation. Of lubricants, molybdenum disulphide in suspension, sprayed on to a warmed surface, appears to be the most satisfactory. However, care must be taken that the use of lubricants does not lead to loss of grip. It must also be appreciated that the coating of this compound will wear and must be renewed at intervals to provide continued protection.

### FAILURE OF BOLTS AND STUDS

The failure of a threaded connexion is perhaps one of the most common occurring in marine practice. The consequences of such a failure may be comparatively innocuous. On the other hand, they may be extremely serious and lead to heavy damage and complete breakdown. It is for these reasons that great attention should be paid to the condition of all bolted connexions. It has already been shown that fatigue cracks can occur at relatively low levels of dynamic stress in the presence of fretting. With steel to steel contact, only very small vibrating movements are necessary to bring about fretting. In the case of bolted connexions, even where malalignment between the components is present, the trouble can usually be attributed to an incorrect fit of the bolt in the coupling hole, together with insufficient prestressing of the bolts. The fit of the bolt in the coupling hole should not be so slack as to allow relative movement to take place between the surfaces in contact, nor should it be so tight that the bolt is inelastic by reason of the grip. An interference fit of 0·0003in. to 0·0005in. per in. of bolt diameter is reasonable, and a prestress in the bolt of about 7 ton/sq. in. should ensure that the bolt remains tight under all normal circumstances.

FIG. 26—*Corrosion fatigue failure of tie rod*

Failures of engine tie rods and stays do occur quite frequently and from a variety of causes. It would be invidious to select one particular defect for criticism, but the cases mentioned below typify those brought to the notice of the Society.

The first concerns the failure of a tie rod in an engine developing 4,400 h.p. Fig. 26 shows the appearance of the fracture, which was of fatigue type, with multiple origins coming from the root of a thread at the end of the rod. The crack had propagated on a surface which was concave to the end of the rod, and had extended across almost the complete section before final separation of the parts. This indicated a low general level of stress on the rod. The material was of good quality and the root radius of the threads was well formed, but the surface of the threads was corroded. The pattern of fracture is not unusual and is indicative of unidirectional bending at a point of severe stress concentration. The local stress at this point would have been sufficiently high to allow cracks to have originated at the corrosion pits in the root of the thread. Once formed, these cracks would then have propagated slowly under the influence of the working stress.

The second, and admittedly less usual, case concerned a fractured tie bolt from a Diesel engine of low power. The broken bolt was found to contain cracks which had originated on the longitudinal axis and propagated radially outwards. These cracks did not penetrate to the surface of the bar but it was noted that they had started to repropagate by fatigue. On cutting a section from the bar, nicking and breaking it, the fracture was found to be crystalline and of fine grain size. This implies that the cracks occurred before the bar was heat treated.

A longitudinal section through the centre of this bar revealed marked carbon segregation. The range of hardness of the bar corresponded to a 48-ton steel in the outer fibres, while in the centre the material was akin to a 60-ton steel. It was concluded that the carbon segregation was the cause of the defects in the bar and the cracks originated in the eutectoid (high carbon content) zone. This segregation was present in the original ingot and could have been due to the inclusion in the bar of the segregated zone just under the "pipe". It was considered that only a few bars in the batch would have been similarly affected, but the engine builder was put to a considerable amount of trouble in checking the tie-bolts on many other engines.

A failure which caused extensive damage took place in several ships of 6,000 tons and involved the breaking of four 1-in. diameter studs securing the piston cooling service assembly to the engine crosshead. To cite one case, resultant damage included a bent connecting rod, distorted bedplate, bent and seized crankshaft and sundry lesser items, and the vessel had to be towed in for repairs. Examination of the studs in one instance showed that three of the four had failed by shear at a point corresponding to the junction between the assembly and the crosshead, while the fourth stud had failed in a manner typical of bending fatigue and it would appear that this stud had failed first. The fracture had occurred at a position on the threaded portion at the face of the nut and another complete fracture lay a quarter of an inch within the nut itself. The fractures had as their origin the roots of the thread, which were themselves well formed. No other defects in material or workmanship were revealed, and it was concluded that, due to insufficient hardening-up on the studs, the major portion of the load had been carried by one stud that was subjected to fluctuating bending stresses. A case, possibly, where half an hour's attention would have saved several months' costly repair.

Fig. 27 shows the half bearings and fractured bottom end bolts from an auxiliary oil engine. Both bolts had fractured at the junction of the tapered section and thread "run out", but while bolt A had fractured under a high "static" bending stress, bolt B had failed by fatigue under a high level of bending stress; thus bolt B was the first to fail. Further, in bolt A, the split locking pin was missing and the nut was slackened back several turns. The tops of all the threads below the nut were damaged. It was concluded that this fatigue failure was brought about initially by the slackening of the nut in bolt A. This would have introduced immediately an excessive dynamic bending stress into the system acting upon bolt B, which failed at the weakest section—the junction between the tapered portion and the "run-out" of the thread. Bolt A was distorted and failed under the high stress placed upon it following the fatigue fracture.

Another case in which consequential damage was severe concerned the fracture of a set of top end bearing bolts. Four

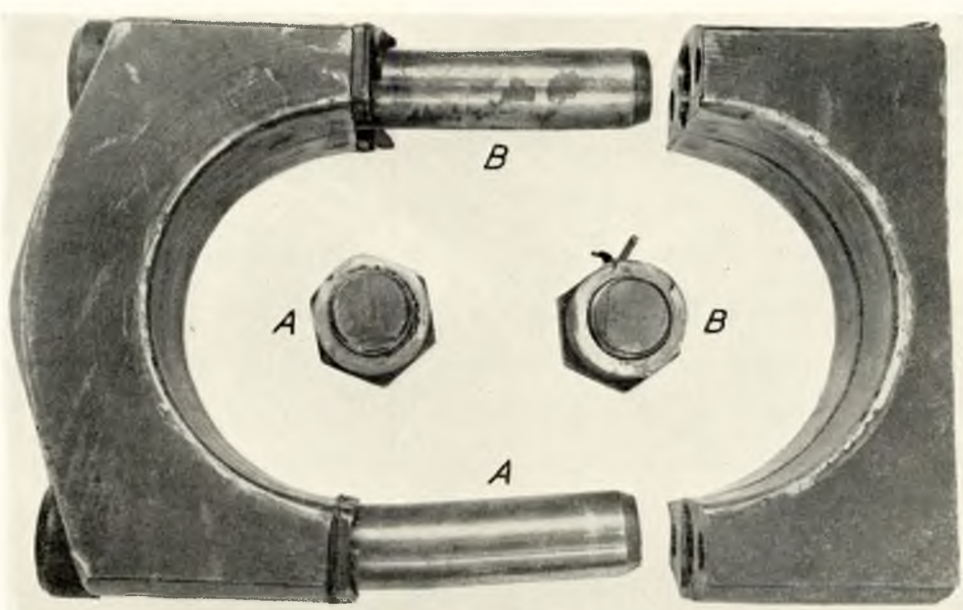



FIG. 27—*Fractured auxiliary bottom end bearing*

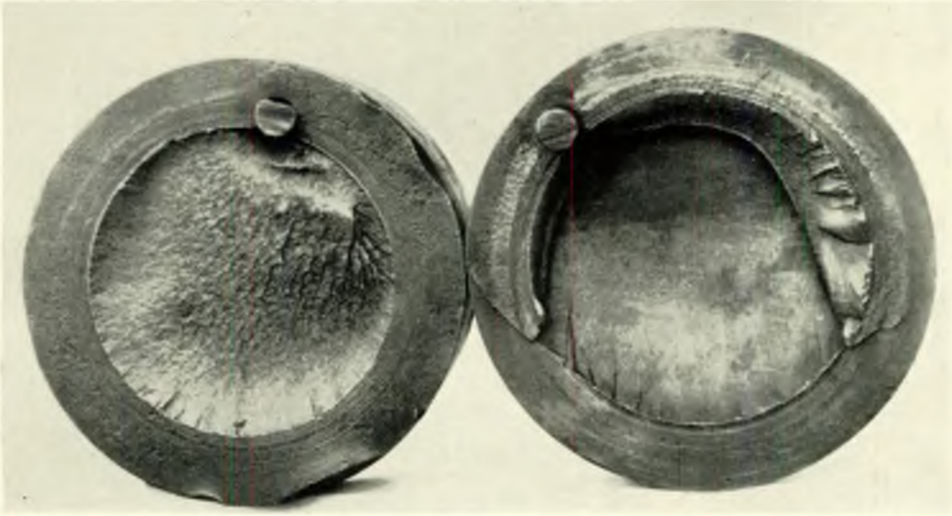


Fig. 28—*Top end bearing bolts*

bolts had fractured due to fatigue, two of which are shown in Fig. 28. Examination of the fracture surface showed that the cracks originated not in way of the dowel pin but all round the root of the fillet between the head and shank. This type of failure was indicative of a sharp stress concentration, and this was borne out by the very sharp fillet radii at the roots of the heads of the bolts. The importance of an adequate radius at sudden changes of section subjected to high fluctuating loads cannot be over-emphasized, since the stress concentration factor of a sharp notch is remarkably high. This has also arisen in other recent cases in which fractures originated at the roots of badly cut threads. The radii at the roots of these threads were negligible and the fatigue cracks followed these stress concentrations for several complete turns. Three breakdowns which were attributed to this fault caused serious damage to the machinery.

The author is aware that failures of threaded connexions have been discussed on previous occasions but the serious failures which continue to occur due to inadequate attention to these components cannot be disregarded. The principal faults may be summed up as follows:

1) Inadequate pretightening or loss of preload due to either inefficient or inadequate locking arrangements.
2) Misalignment of the head or nut imposing excessive bending stresses on the threads or other unavoidable notches.
3) Inadequate root radii at either the head or roots of the threads, and in some cases the thread "run-outs".
4) Corrosion at the roots of the threads.

The cures for these defects are self-evident. Although it has been shown that the form rolling of threads results in a marked increase in fatigue strength of threaded connexions, the application of this practice will not avoid failures due to the above faults.

Fig. 29—*Furnace cracking due to oil contamination*

FIG. 30—*Cracks in the furnace corrugations*

OIL CONTAMINATION

Oil in its rightful place is essential but in the wrong place it can lead to serious and extensive defects. Within a few days of going into service, six furnaces in the main Scotch boilers of a 5,000-ton ship failed. Cracks were found on the water side of the furnaces running circumferentially at the bottom of the corrugations. The cracking was of a "crazy" type, as shown in Fig. 29, and in order to determine the depth of the cracks specimens were cut from the corrugations and flattened out. They were found to be deep and their penetration is clearly shown in Fig. 30. The cracks were predominantly transgranular and contained corrosion products. They originated at mild stress raisers on the surface of the furnace plate, such as shallow corrosion pits, indentations, laps and grinding marks. It was an obvious conclusion that these cracks were the direct result of corrosion fatigue. Whilst there was ample evidence of oil being present in the boiler water and the engine room log books revealed an excessive consumption of cylinder oil, the furnaces themselves were found to be comparatively clean. It may be surmised, therefore, that the deposits on the furnaces would be disintegrated and removed by the expansion and contraction of the furnaces induced by the successive deposition and removal of the deposits. These local temperature fluctuations undoubtedly provide a thermal fatigue condition. This case is a unique example of the combination of corrosion and thermal fatigue.

It should not, of course, be necessary to wait for such defects to develop before realizing that oil is finding its way into the boilers. Excessive consumption of cylinder oil and oil deposits in feed filters should give ample warning of the danger.

Figs. 31(a) and (b) show the damage that can be caused to bearing surfaces by contamination of lubricating oil. These engines were running on boiler oil and due to wear on piston and scraper rings combustion gases had escaped into the crankcase. The damage had developed within a period of two years. The oil in the crankcase was found to be acidic and to have a high sulphur and water content. The oil purifiers had also become severely corroded. Such cases emphasize the importance of maintaining lubricating oil in a clean and acid-free condition and in preventing combustion gases from entering crankcases.

CONCLUSION

A paper dealing with failures makes somewhat depressing reading. Ideally, perhaps, there should be no failures in marine

FIG. 31(a)—*Damage to journal and crank pin due to contaminated lubricating oil*

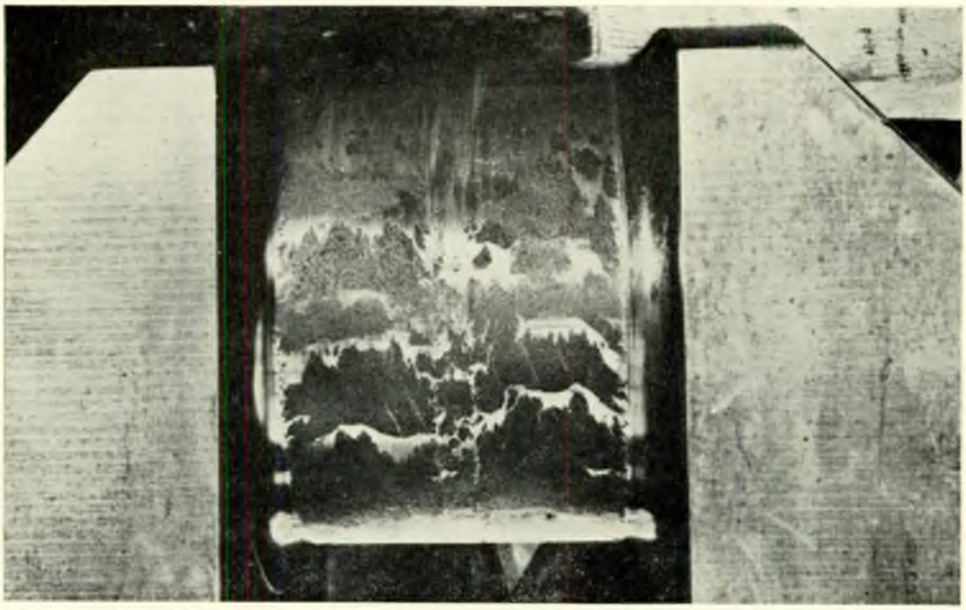


FIG. 31(b)—*Damage to journal and crank pin due to contaminated lubricating oil*

machinery. The few examples given in this paper show that this ideal is far from realization, and that machines, like men, are not infallible.

The important consideration is that engineers should appreciate the significance of the failures that occur so that they can guard against them in the future.

ACKNOWLEDGEMENTS

The author's thanks are due to various shipowners and firms for their agreement to the use of certain information. He also acknowledges the help he has received from colleagues, particularly from Mr. G. P. Smedley, B.Eng., B.Met., and Mr. B. K. Batten, M.Sc.(Eng.).

# Discussion

Mr. C. C. POUNDER (Vice-President) said that he did not agree with the author's statement that a paper dealing with failures made depressing reading. On the contrary, his only criticism of Mr. Pemberton's excellent paper was its brevity.

A discerning man, reading the paper attentively and thoughtfully, would absorb more useful knowledge from it than would be possible from a dozen volumes on elastic theory. An engineer's experience had not to go very far before he realized that an academic study of the theory of elasticity was no guide to the nature of machinery failures encountered in service. He did not desire to be misunderstood on this point; he was not decrying studies of elastic theory—he would be false to his own past if he did.

The author, in his conclusions, wrote that ideally, perhaps, there should be no failures in marine machinery. Ideally, perhaps he was right; but, as a shrewd down-to-earth Surveyor-in-Chief, Mr. Pemberton knew that, if there were no failures, engine builders in general, and Lloyd's Register in particular, would be exposed to criticism for building to requirements which were too exacting. In other words, if there were no failures, designs would be criticized for being too heavy; if there were too many failures, designs would be criticized for being too weak. Accordingly, somewhere between the two extremities, a *modus vivendi* satisfactory to reasonable men of all shipping interests had to be found. This was where the skill of Mr. Pemberton and his staff became apparent.

The paper began with a long reference to engine room explosions. As this subject came first in the paper, it seemed reasonable to assume that it probably came first also in the thoughts of the author. Mr. Pemberton indicated very clearly the present state of knowledge and much could be said in supplementation of his remarks. Attention would however be confined to one aspect.

Explosions did not arise instantaneously. Somewhere a hot spot was formed that grew and developed; and the physical evidence for the hot spot normally became apparent to touch and smell long before there was a bang. His interest in crankcase explosions was aroused in 1928, when a fatality occurred in a Continental-built installation. In succeeding years there were other explosions in Continental-built engines. It was not until the lapse of nearly ten years that he was brought face to face with a crankcase explosion in a Belfast-built ship.

A painstaking analysis of evidence concerning explosions had shown that in many instances an engine component, be it a trunk piston, or a camshaft thrust bearing, or an emergency governor sleeve, had been known to be hot, not for minutes only, but sometimes for as long as three and four hours. How engineers down in a machinery room, faced with a hot engine component that, despite all efforts, had continued to grow hotter and hotter over a relatively long time, could expect to reverse the temperature trend was puzzling. It was possibly a tradition inherited from reciprocating steam engine days, when piston rods ran hot and the engineer's duty was to swab the rods until they became cool.

The size and the approximate number of self-closing relief doors were now determined by the survey authorities, and, if the doors were maintained in a free-lifting condition, they would open with pressures normally not exceeding about 1lb./sq. in., perhaps less. He agreed with the author's statement that there was, as yet, no really satisfactory flame trap available for large engines.

Over the last twelve years or so, a well-known firm had developed an oil mist detector which, if properly applied and maintained, could give complete confidence to the engine room staff. In recent years another form of oil mist detector had been evolved, also excellent in conception and arrangement. If oil mist detectors were coupled to automatically controlled carbon dioxide drenching systems, there could be complete safety. But if a detector only operated a red light, or an alarm bell, the personal equation was still involved. It was to be hoped that, at an early date, the application of oil mist detectors would be made compulsory. The number of crankcase explosions per annum might be small, having regard to the number of engines in service, but no comfort was to be taken from probabilities and average numbers, because nobody could say with certainty that none of his engines would ever, at any time, have a hot spot. Unremitting vigilance, therefore, was the price of safety.

Non-inflammable lubricating oils, also plastic water lubricated bearings, had been investigated, but technical difficulties and high costs had made their respective applications impracticable.

The author correctly said that, if engines were provided with explosion relief valves of adequate size and the crankcase doors were securely fastened, no damage to the engine should result from an explosion; moreover, the installation of a smoke detector made possible the avoidance of explosion.

Regarding the author's reference to bent turbine rotors: it was not always clear to what extent systematic thermal stability tests were justified for ordinary commercial work.

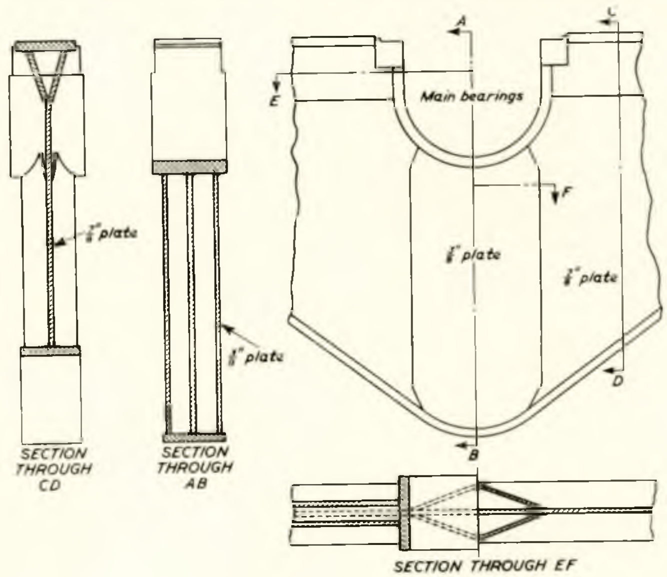



FIG. 32

As with so many other things, it was a matter of weighing probabilities. At one extreme no heat treatment was given beyond that due to any important forging; at the other extreme there was the full but costly two-stage procedure. There was also the differentiation to be made between carbon steel rotors and alloy steel rotors.

Referring to Diesel engines: Mr. Pemberton advocated cast steel main bearing housings having web extensions which were butt welded to the vertical cross-girders of bedplates. Experience had confirmed the value of this construction in ensuring down-welding. For comparison with Fig. 15(a): Fig. 32 showed a design for main engine bedplates which, for a time, had a measure of popularity. It seemed simple and effective, but inherent welding problems were responsible for its discontinuance.

Laminations in mild steel plates *seemed* to be relatively more numerous now than formerly. But this impression might not be statistically supported. The widespread use of welding might be responsible for revealing laminations which would otherwise not be detected. The author's opinion on this point would be valuable.

On the subject of failures of threaded connexions, the author summed up the principal faults under four headings. As a corollary to the first heading, namely that inadequate pre-tightening, or loss of pre-load, was due either to inefficient or to inadequate locking arrangements: many instances could be quoted in which loss of pre-loading had been due to the flattening of the many small ridges to be found on ordinary machined surfaces. Slackening occurred, but the locking devices remained unaltered. As regards engine tie rods: during the first twenty-five years of experience with four-stroke single-acting crosshead engines he did not recall more than two failures. Then suddenly, in one shipowning company, at least half-a-dozen rods failed within the space of a year or two. As the fractures seemed to be failures by violence rather than by creep, the tightening-up procedure became suspect. In due course the phase passed and there were no more failures. Probably the tie rods had been tightened on the same principle as if they were, say, bottom end bolts.

In presenting a paper of about fifteen pages the author showed sound judgement, because the most hard-pressed of men could find time to assimilate its contents. Mr. Pemberton would show even sounder judgement, however, if the present paper became the first of a series from his pen. A busy man? Of course!

Mr. Stewart Hogg, O.B.E. (Member of Council) asked the audience to bear with him after such an eloquent speaker as Mr. Pounder. Although he had listened to many papers in the Institute, he believed that there were few papers that would command more interest to those afloat. He said that he was mainly interested in machinery failures due to the incompetency of individuals, and that he classed machinery failures under four headings: (a) design, (b) materials, (c) manufacturing processes and (d) incompetency. He was of the opinion that the number of lives lost through machinery failures was not as bad as it might be, although he added that Mr. Pounder had rather made him shudder.

"Why are there not more breakdowns in vessels at sea when the engine room crews' total years of experience in many vessels is so lamentably low?" was a question which he was often asked; his reply was that modern engines and manufacturing methods had greatly improved, also that the materials used were much more reliable than in former years. These modern conditions reflected credit on Mr. Pemberton and his staff. A number of generators came to grief due to failure of the bottom end bolts, and while Mr. Pemberton had given various reasons for the failures, he (Mr. Hogg) was of the belief that these bolts were very inaccessible and consequently were inadequately tightened up, thus imposing too much strain on the front row bolts which in due time led to failure. He felt that this could often be attributed to the fitter or ship's engineer in the erecting shop or the ship, and therefore could come under the heading of incompetency; although on reflexion he felt that the designer should take some responsibility for the inaccessibility of these bolts.

While on the subject of threaded connexions he mentioned a series of main engine piston rod failures, and while he admitted that those mishaps belonged to the past he believed that the remedial action taken at that time—namely, more accurate screw cutting for rods and nuts and a more scientific method of tightening up the nuts on the rods—had still not solved the difficulties and that rods still fractured from time to time; however, he felt that it was such problems that kept members interested in their daily tasks.

At this point he said he would like to raise one or two questions, but that some of his points regarding crankshaft failures had already been covered by Mr. Pounder. He wanted to know if any experiments had been made on the fitting of oil drenched gauzes internally in way of explosion doors; he believed people were interested in this but he did not know the shape of these gauze fittings, and he thought perhaps Mr. Pemberton might have some more information on the matter.

He then mentioned boiler shell plate cracks and said he believed that riveted double butt straps were on the whole satisfactory. In his experience only two boilers out of several hundred had developed leaks from the butt strap joints which could not be stopped by a caulking chisel. In one case of leakage from the rivets and the plates of a double butt strap joint, the outer butt strap was removed and a whitish powdery deposit was found, together with a number of star cracks from the rivet holes. The boilermakers said the cracks were due to the chemical action of the white powder, and that this was produced due to operating conditions, whilst the shipowners' representatives insisted that the primary cause was over-pressure by the hydraulic riveter being used to push together badly shaped plates, and that the deposit would not have been present but for the leakage.

He felt that efficient inside caulking was all that was required, for to stop the leakage from the outside would not prevent corrosive elements getting between the plates. Seal welding of either rivets or shell from the outside should be unnecessary and was a practice that should be discouraged.

The Chairman (Mr. W. R. Harvey, Vice-Chairman of Council) suggested that a possible reason for the seal welding of butt straps in place of normal caulking was the scarcity of good craftsmen riveters, this class of work being used less and less as welded construction was adopted for all other types of boilers. He said that in his own company they no longer had any riveters skilled in this type of work.

Mr. T. W. Bunyan, B.Sc. (Member of Council) prefaced his remarks with a warmly worded tribute to the great contribution to the technology and practice of marine engineering being made by Lloyd's Register of Shipping, to whom he owed a debt of gratitude for the assistance which the Society had so readily given on a number of problems.

The fact that it was possible to obtain a factual report prepared by a competent engineer, who might have had to go at very short notice to the ends of the earth for his information, was of very great value indeed to shipowners and others. He found every part of the paper of great interest as he himself had been steeped in that atmosphere for many years. He said that he still felt the odd twinge when he rang up the Society's Investigation Department to be told that Mr. So-and-So had just left for Japan or New York.

While it was very sad that so many disasters overtook marine machinery every year, and it was not difficult to visualize the consternation caused at head office by the arrival of a badly worded cablegram announcing the bald facts, it was some comfort—rather like cold tea—to realize that these things did occur in other highly efficient and well run organizations.

He expressed alarm on seeing Fig. 18 and requested the author's assurance that it was in fact a bad design of weldment in which high tensile contractional stresses were induced.

He found the author's remarks about fretting corrosion most illuminating, which suggested that some form of fretting inhibition could with advantage be applied to the large end of the propeller cone, which, if confined to the sensitive region, the first inch or two at the large end would not impair the grip of the propeller boss on to the cone.

The havoc resulting from the failures of dynamically stressed bolts must always be a feature of a paper of this kind, and he wished to underline the values for interference fits and pre-stress given in the paper on page 388; but, in spite of all that was said and published they still had to live with that formidable school which was doubtful of any bolt requiring less than the crushing stress to drive it as being a satisfactory indication of fit.

Mr. Bunyan said that Fig. 12 was a classic example of incompatible steel making, particularly when it was considered in the context of the multiple disasters which these defects caused, defects which were difficult to detect until extensive sulphur printing was insisted upon, a matter involving a very high grade machine finish to the cheeks of the wheel rims which were normally left fairly rough.

It was quite common practice to arrange that the calculated primary whirling critical speed of turbine rotors was at about 65 per cent of the running speed—there were sound technical reasons for this. Under these conditions a worn claw coupling could make itself apparent by setting up a violent vibration at the whirling critical, and there had been instances where rotors had been removed for balancing to cure this condition, only to find that the rotor was in fact in perfect balance but the mysterious vibration was still present on starting up. Renewing the worn couplings had of course been the cure; *Canberra* had no claw couplings between the propulsion turbines and alternators—a semi-flexible bellows coupling served the same purpose, with the advantage that there were no parts to wear eccentrically.

He concluded his remarks with the sure hope that Lloyd's Register would go on from strength to strength and looked forward to the day when shipowners and others could approach the Society's mathematics division to run out a theoretical problem on their digital computer.

Mr. P. Jackson, M.Sc.(Eng.) (Member of Council) joined in complimenting Mr. Pemberton both on his paper and on the work of the Statistical and Investigation Department of Lloyd's Register of Shipping. Referring to the question of explosion doors and the remarks of Mr. Hogg, who stated that Doxford were now fitting explosion doors with layers of oil wetted gauze to prevent the passage of flame into the engine room when the explosion doors lifted, he would be pleased to let Mr. Hogg and Mr. Pemberton inspect these first experimental doors and have particulars.

Mr. Jackson stated that the last time he spoke on the subject of crankcase explosions some few years ago, on the occasion of Mr. Pounder's paper*, he had stated that Doxford had never had a crankcase explosion; since then there had been three very minor explosions though in no case was there any damage to the engine nor was anybody hurt. The first of these explosions had been caused by the choking of the special filter which filtered the oil to the thrust block and to the failure of the oil supply to the thrust block. This was actually worse than having dirty oil to the thrust block. The lack of oil had caused the white metal on the thrust pads to run and then the bronze pads had thrown off sparks, causing a minor explosion at the aft end of the engine. In another case, a short piece of piston ring had been broken off and found its way between the piston skirt and the liner of the old type engine, and, being scrubbed about, had worn a hot groove in the liner which had ignited the lubricating oil again, causing a very minor explosion and lifting one of the crankcase doors.

He would like to say a few words about the failures of gearing rims, although this was not really his subject. He had come across one instance some two years ago where the metal between the cast iron centre and the rim was worn to a powder. It was well known that the machining of a rim caused crushing of the metal on the extreme periphery and this was known as the Beilby layer. In the particular case of this gear rim, the Beilby layer had been reduced to powder and the gearing had slipped. The powder was wiped off and the surface of the cast iron centre slightly filed down to the undamaged metal and the original rim was re-shrunk on to the centre.

Turning to flame cutting, he agreed with Mr. Pemberton on this and it was the practice at Doxford to machine the edges of all the plates to the extent of 10 or 12 mm. prior to welding. Examination had shown that flame cutting removed the surface cracks on the flame cut edge, which extended into the steel plate by no more than 5 to 7 mm., so that machining away 10 to 12 mm. produced an edge free from cracks.

The particular case quoted in the paper of a crankshaft failure (Figs. 19 and 20) was the same example that Mr. Atkinson and he* had dealt with six weeks previously. They had, however, given a somewhat different explanation of the cause of this failure but a combination of the two explanations was interesting.

On the subject of detuners and detuner damping, he did not think that Mr. Bibby would have accepted the suggestion that his detuner functioned as a damper and as such it must be inefficient. On the Bibby detuner there was a clearance of about 1 mm. between the spring and the surfaces of the fixed and moving rims and when there was no oil in the detuner there was a definite knocking noise. It was the practice to supply oil to the detuner to keep it quiet and he could imagine that in certain cases a cold high viscosity oil would be more effective.

With regard to the example of the split connecting rod shown in Fig. 22 of the paper, he believed that this was an example from a two-cycle auxiliary type engine which he had seen some 3 or 4 years ago. Although he had seen laminations and segregations in steel parts, he found it very puzzling to know why the connecting rod, as shown in Fig. 21, should have cracked in that particular way—longitudinally up the rod. He thought that there must have been longitudinal laminations in the material and that the fitting of the big end bearings must have been such as to cause hammering at the point of the origin of the fracture, but he was not entirely satisfied with this explanation.

He then discussed the scavenge pump piston failure shown in Fig. 25 and said that this had been due to vibration of the scavenge pump piston, which had a natural frequency with 4 nodes of about 440 per min., and, with a double-acting piston running at 110 r.p.m., the frequency of the fundamental load variation was 220 per min. This particular piston had been vibrated magnetically with sand on the surface by the same technique as was nowadays adopted for turbine discs. The failure had been cured by putting a flange on to the surface of the piston which extended out towards the periphery and this had had the twofold effect of strengthening the attachment and further increasing the frequency of vibration.

Finally, he referred to the cases of crankpin and journal corrosion (Figs. (31a) and (32b)), and stated that these had been caused by the sludge from the combustion of high viscosity fuels getting down into the crankcase and, unfortunately, mixing with water leaking from the water cooling system of the pistons. This sludge had a high sulphur content, sometimes as high as 8 per cent, and thus a dilute sulphuric acid was formed. This had occurred with the old trunk piston type of engine but it had been entirely corrected with the diaphragm engine, where there was a distinct separation between the piston and the crankcase.

* Pounder, C. C. "Some Current Types of Marine Diesel Engines". Proc.I.Mech.E., Vol. 160, p. 312.

* Atkinson, R., and Jackson, P. 1960. "Some Crankshaft Failures: Investigations, Causes and Remedies". Trans.I.Mar.E., Vol. 72, p. 277.

Mr. W. McClimont, B.Sc. (Member) said that Mr. Pemberton had very rightly pointed to the value to builders and to operators of knowledge of service experience. This was particularly true of information concerning machinery defects which had been investigated in a systematic manner, using all the techniques of detection available to the engineer today. The investigational resources of Lloyd's Register were known to all, resources of the greatest value not only in terms of facilities, but equally important of skilled personnel.

A paper dealing with failures need not be depressing reading; if there were no failures in marine machinery one would have good reason to suspect that there was no longer any progress in marine engineering. He admitted that he might be biased, however, because the attainment of the ultimate ideal in marine machinery would leave him looking for another job.

He confessed to a measure of disappointment in the content of Mr. Pemberton's paper; one or two of the cases mentioned by him hardly seemed to earn a place in the record and one was left re-reading a paragraph or two, looking for the moral of that particular tale. He was of the opinion that a few dates might have helped to get the picture in better perspective, though he realized nevertheless that dates could be tricky and that one often had to keep things as anonymous as possible.

Turning to a few of the cases mentioned by Mr. Pemberton, Mr. McClimont endorsed the remarks made concerning crankcase explosions, particularly regarding smoke detectors. Most people were aware of the work done by the British Shipbuilding Research Association in the past decade; since the safety of life had been involved, the results had been widely circulated. It had been very gratifying to know that the views of Lloyd's Register were so much in line with their own.

Several features of the notes on gear wheel rims that had moved were puzzling and this was an instance of the moral of the story not being clear. It was said that the nature of the material might not have been conducive to an effective shrink fit, but he was sure that spheroidal graphite iron was quite usual. It was said that it was probable that there were considerable areas not in contact. He asked how this was deduced and if it were possible to suggest a method of ensuring that it did not happen again. The surfaces of the centres were described as very rough turned and he wondered if Mr. Pemberton could indicate the order of the roughness quantitatively and, further, if he cared to indicate quantitatively the order of surface finish he considered good practice for a fit of this kind.

Looking for guidance on how to ensure that failures would not recur, the gear tooth in Fig. 10, with a coarse dendritic structure, prompted him to ask what steps should be taken to avoid this.

He felt it would be interesting to be able to fit Fig. 17 to the appropriate weld on Fig. 16. There was a remark about the clearance between the stay and the hole in the shell plate being almost completely on one side of the hole, but he could not see the importance which Mr. Pemberton attached to this mal-alignment.

He believed he was correct in saying that the fatigue tests carried out by Lloyd's Register on the radius at the lip of an oil hole were part of a programme of research undertaken by the Society for B.S.R.A.; the lip radius of a quarter of the hole diameter was not generous and might reasonably be a half.

Ultrasonic testing of crankshaft forgings was a good thing but needed skilled personnel, not so much for operation but for interpretation of the results; otherwise forgings would be scrapped for things that did not matter and some of the most serious faults might well be missed.

When discussing fretting fatigue he felt it was well to emphasize that relative movement might be very small; he did not know Mr. Pemberton's experience but his own was that fatigue failure was more frequently associated with small but deep holes rather than shallow depressions. Later, however, Mr. Pemberton spoke of small pits, and also, speaking of a scavenge pump failure, he referred to fatigue failure with little fretting oxidation, so his experience might well be the same as his own. It was worth emphasizing that some of the worst cases of fatigue failure arose in conjunction with a few, small, unimpressive pits. In his view these pits were a sympton rather than a cause, the cracks causing the pits rather than the pits being the point of initiation of the cracks.

Turning to the coupling bolt failures, first in the line shafting and secondly in the crankshafts, he felt that the explanation of the mechanism of failure was sound but once again he asked what they should do about it. There was a partial answer later on page 388, and he felt that the recommendation of a pre-stress in coupling bolts of about 7 tons/sq. in. was a good theoretical solution, but the problem remained of how to obtain a reliable measure of the pre-stress, especially with an interference fit on the bolts.

Reverting to the more general discussion of fretting fatigue, he was rather doubtful whether the life of a coating produced by spraying molybdenum disulphide suspended in any base would be long enough to be of practical value.

The author had listed four faults that gave rise to failures of threaded connexions and said that the cures for these defects were self-evident, but, as he (Mr. McClimont) had already suggested, he felt that the first, inadequate pretightening was not such a simple matter. Torque was not a good criterion; for instance, for the same stud, or set bolt, diameter but with different detail design the pre-stress might vary by 100 per cent for the same torque, and significant variations in the relationship between pre-stress and torque arose from variations in thread finish and the lubrication between threads.

Reference was also made to the effect of inadequate root radii at roots of the threads. Whilst sharp corners in the root region caused high local stresses and reduced fatigue strength, the effect of reducing root radius to half that for the basic Whitworth or B.S.F. form was to reduce the fatigue strength by 15 to 20 per cent. He said that this was not a plea for poor roots but to put the matter of root radius in perspective; what was more important, he felt, was that the root radius of a bolt should be of smooth form and blend smoothly with the thread flanks. The root radius of a nut was not normally important unless it was a thin walled nut. At thread run-outs, one of the important things was to ensure that the run-out did not occur too sharply. He believed that another important point to watch was that after assembly the run-out should not be near the face of the nut, as this would superimpose the stress concentration due to run-out on the concentration at the first thread in engagement.

Finally, he was of the opinion that the observations on form rolling of threads were possibly a little misleading, as the form rolling of threads should help to avoid the production of threads with inadequate root radii; improvements similar to those for form rolling could also be obtained by root rolling of pre-cut threads.

Commander E. Tyrrell, R.N. (Member) was the next speaker and he began by joining with those who had congratulated Mr. Pemberton on the excellence of his paper. He felt the most important remark in Mr. Pemberton's paper was contained in the introduction where he stated that "Lloyd's Register of Shipping, with its world wide network of surveyors surveying some 11,000 ocean going merchant ships, is in a unique position to know what happens to ships' machinery in service and how the more important casualties are dealt with". It was particularly important that this knowledge be utilized in improving design or materials or in preventing mal-operation of machinery. He hoped Mr. Pemberton would forgive him when he said that it might well be used as a guide to shipowners on what they should buy.

Years of dealing with the machinery of merchant ships, both in shipowning and shipbuilding industries, had impressed upon him the importance of reliability in ships' machinery. A single machinery failure at sea could wipe out any savings made during the life of the ship by a decrease in fuel rate between one set of machinery and another.

Britain's competitors had not underrated the importance of reliability and he begged forgiveness for quoting part of a paper entitled "Economic Comparison of Steam versus Diesel for Oil Tanker Propulsion", written by R. T. Simpson of the International General Electric Company of America. In that paper the author showed failure rates of machinery on a percentage basis and remarked as follows:—"The number of casualties is compared with the number of tankers in service. During 1957 there were 1,010 casualties from all causes for a total fleet of 2,476 tankers, or an overall average casualty rate of 40·8 per hundred ships. Of the total of 1,010 casualties. 426 were from mechanical causes and 314 of these were due to main engine breakdowns. The casualty rates due to main engine failure were further analysed according to type of engine and age.

TABLE I

| Period of build | Steam | Turbo-electric | Motor | All tankers |
|---|---|---|---|---|
| Up to 1939 | 7·8 (77) | — | 12·6 (230) | 11·4 (307) |
| 1940—1949 | 7·1 (196) | 15·3 (425) | 16·6 (248) | 13·9 (905) |
| 1950—1957 | 13·0 (506) | 16·7 (6) | 11·4 (752) | 12·1 (1264) |
| All tankers | 11·0 (779) | 15·3 (431) | 12·8 (1266) | 12·7 (2476) |

Commander Tyrrell pointed out that it was interesting to note that the failure rate had not decreased at all; in fact, it had increased with the more modern ship, or had tended to from the original in 1939, which was the point raised by Mr. Hogg. He then quoted from the paper by Simpson again as follows:

TABLE II

The pertinent *main engine* casualty rates for modern (1950-1957) tankers are summarized as follows:—

| | Number of tankers | Casualty rate/100 | Variation from motorship rate, per cent |
|---|---|---|---|
| Motorships—all flags | 752 | 11·4 | — |
| Steamers—all flags | 506 | 13·0 | +14 |
| Steamers—U.S. flag | 53 | 3·8 | −67 |
| G.E.C. turbine—all flags | 59 | 1·7 | −85 |

"For U.S. flag steam turbine tankers built between 1950 and 1957, there were 53 with two casualties, making a rate of 3·8. With regard to tankers of various flags powered by turbines built by the General Electric Company during 1950-57, there was one main engine casualty in 59 tankers, for a rate of 1·7 per cent. The foregoing demonstrates that while there is a small difference in the main engine casualty rates for steamers and motor ships of all flags built between 1950 and 1957, there is a remarkably lower rate indicated for power plants involving American-built geared steam turbines generally and General Electric geared turbines in particular".

Commander Tyrrell felt that Mr. Simpson was advertising his own products by calling attention to possible deficiencies in those of his competitors. Mr. Simpson's remarks were bound to influence owners, particularly foreigners, against machinery built in this country. The only way to compete with advertising which denigrated their machinery was to ensure that the facts were available, and he thought members were fortunate in having Lloyd's Register, which could supply those facts; statistical information was available to enable them to produce failure rates for machinery made all over the world, and such was their status that these rates were accepted without question. He said that he would not go so far as to suggest that this information should be published for all to see, as that would be impossible, but he did think a useful purpose would be served if it were widely known that Lloyd's Register's certified failure rates could be made available to machinery manufacturers concerning their own particular machinery. Those manufacturers who had low failure rates would, of course, lose no time in advertising the fact. Owners would then form the habit of asking machinery manufacturers or designers for the Lloyd's Register certified rate. He realized that such a system would not be easy to operate; details of machinery failures were held in confidence by Lloyd's Register and were only available to those directly concerned, which was quite correct. Nevertheless, he thought the Shipping Federation would probably agree to information being collected to produce a percentage failure rate, if anonymity could be preserved.

MR. G. VICTORY (Member) said that they could argue all night on the fascinating examples of failure given in Mr. Pemberton's paper; however, he would confine his remarks to two of them.

Firstly, he felt that the undesirability of putting down a weld with a small gauge electrode on a medium carbon steel of substantial thickness without preheating deserved further emphasis. This fault had been mentioned in relation to Fig. 13 where the boiler shell cracked in way of the sealing welding. Surely, Mr. Victory continued, it was also one of the major faults leading to the fracture of main boiler stays and possibly to some of the bedplate failures. Here he thought it should be stressed that heat treatment after welding could be of very little use if cracks were already present due to hardening and contraction stresses in the heat affected zone.

He understood that the failure of welded stays had occurred in a number of boilers before the practice of welding those stays was discontinued, and it seemed unlikely to him that all those welds were of poor quality; perhaps it was another instance of a rapidly quenched weld made without preheating and in conditions of restraint. This treatment could be expected to give excessive hardening of the parent metal and lead to incipient cracks at the weld interface, such as were in fact found under some of the boiler stay welds which had not fractured. He asked if the author would say what precautions he would call for in the welding of boiler main stays, if such an arrangement were put forward for approval today.

Turning to the breaking of bolts in the bottom end and other bearings, it seemed obvious to him that the practice of stressing these bolts to a known degree should be extended. As Mr. McClimont had pointed out, the torque spanner method had its disadvantages but he believed there should be some way of specifying a definite elongation. However, there were two practical features that should be mentioned in connexion with this matter as they could reduce the benefits obtained by "pre-stressing". He said that if there were bruises on the nuts and threads of the bolt due to rough treatment, the nut might "settle in" after its original tightening and a bruised nut could also produce very severe bending stresses in the bolt. These faults might also be brought about by the use of numerous thin liners between the half-bearings. This practice seemed to be increasing, and the danger of a stress in the bolt being relieved by the liners bedding down after initial tightening, or of bent and buckled edges on the liners throwing the bearing askew and putting a bending stress in the bolts, did not seem to be appreciated everywhere.

Finally, he wondered if Mr. Pemberton could give any information as to the cause of the persistent top end bearing trouble which some ships had experienced, even when the alignment, bearing surface and combustion loadings were apparently satisfactory.

MR. W. F. JACOBS (Member) said he would confine himself to one point that seemed important.

Mr. Pemberton had said on page 389 that "A failure which caused extensive damage took place in several ships of 6,000 tons and involved the breaking of four 1-in. diameter studs securing the piston cooling service assembly to the engine crosshead". As in one case a ship had to be towed in it was not of minor importance. He quoted further from the same paragraph, as follows: ". . . . and it was concluded that, due to insufficient hardening-up on the studs, the major portion of the load had been carried by one stud . . . ."; he considered that was all right, except for a suspicion that there was a whole class of ships like that; but possibly the idea in mind had been to have a slight effect on design. The particular assembly described held on four studs and he wondered if they were placed in such a way that it was very difficult to tighten them up with an ordinary spanner. If there were a whole class of vessels like that, he hoped that once the trouble was known due precautions had been taken in the other ships.

Turning to page 378 of the paper, he quoted the following extract: "During a dock trial when the speed was being gradually increased to check the overspeed trips of the turbine, increasing vibration was observed at the forward pedestal bearing. With the set operating at about 4,000 r.p.m., sparks were thrown from this bearing and the unit disintegrated". He (Mr. Jacobs) wanted to know the normal service speed, and said that if it were a 60-cycle a speed of 4,000 r.p.m. was not excessive; if however it operated at 3,000, then 4,000 was excessive overspeed.

Mr. T. I. Fowle, B.Sc.(Eng.), who spoke from a background of experience with lubricating oil, wished to comment about the wear on turbine shafts. He endorsed Mr. Pemberton's remarks about rubbish left in the lubricating system; his company regularly received reports and sometimes photographs of masses of swarf left in the system, also nuts and washers, and only the other day he had been given a photograph of an oil pipe with a wad of paper wedged into it. These things occurred all too frequently.

Secondly, he referred to the wear on shafts due to carbonization of the lubricating oil in the oil glands due to excessively high temperatures. Mr. Pemberton had cited a case where the temperature had reached 275 deg. C., and Mr. Fowle referred to a Dr. Karl Wolf* who stated that if the temperature in a bearing housing exceeded 130 deg. C. it would become a "coke factory" and the carbon could cause heavy wear of the shaft. He said that he had made a few measurements on turbines not subjected to this trouble and the temperature of the oil gland was never greater than 120 deg. C., which was in line with Dr. Wolf's statement. Some simple laboratory tests also gave some support to this idea; they had put a few drops of oil on to strips of steel placed horizontally in ovens. The ovens were maintained at various temperatures for 24 hours, after which the condition of the oil film was examined. It was found that the oil film started to convert to lacquer under these conditions when the over temperature exceeded 130 deg. C. to 150 deg. C. There was thus not a great variation in the critical temperature between several kinds of turbine oil, considered in relation to the 275 deg. C. occurring in the case described by Mr. Pemberton. Although the laboratory conditions, which represented a thin film of oil stagnant for 24 hr. under conditions of high temperature, were possibly more severe than the conditions it was desired to simulate, the results seemed to be significant.

Just why some designs of oil baffle gave rise to such high temperatures had been uncertain. They had some evidence which suggested that the majority of the heat was radiated from the opposing end surface of the steam labyrinth gland. If this were so, a factor of some importance would be the cooling effect of the air passing between the oil baffle and the end of the labyrinth gland. Thus, designs where a free flow of air was permitted would be expected to operate at lower temperatures than those where air flow was restricted. It was interesting to note the author's view that poor oil circulation through the baffle itself was the prime factor. It would be very useful to know what the temperature of the oil baffles was after the modifications he described had been made.

* Wolf, K. 1951. Die Schmierung von Dampfturbinen. Published by Springer 1951.

Mr. J. H. Gooch, M.A., said that he would like to join with the other speakers in thanking Mr. Pemberton for a most interesting and useful paper. On page 384 of the paper the author referred to the failure of components due to defects in forgings which could not be detected by normal inspection methods. He was pleased to see that Mr. Pemberton recommended the testing of forgings to ensure that those which had hidden defects did not go into service.

Referring to ultrasonic testing, he said that his limited experience of this was that it was difficult to apply in ordinary marine engineering practice to a variety of forgings of widely varying shapes and sizes. X-ray testing was slow and expensive. Therefore, he had formed the opinion that a new technique needed to be developed for the detection of hidden defects in forgings. He thought that one development which might be of possible use was the extension to this work of gamma-ray reflecting equipment similar to, say, pipe wall thickness gauges. Mr. Gooch said he would appreciate Mr. Pemberton's views on this matter.

# Correspondence

Mr. H. D. Adam (Member) commented that from his vast store of information as Chief Engineer Surveyor to Lloyd's Register, Mr. Pemberton had given useful details of machinery failures, and in most cases the cause, effect and remedy were clearly stated, which was valuable in the avoidance of breakdowns.

He was sure they would all like to avoid crankcase explosion. They were only told that "in the majority of cases the hot spot arises in the running gear bearings, piston and thrust, etc."; in no case were they told which rubbing surface became hot.

Could the author please tell them if in any of the 52 cases mentioned the actual position and cause of the hot spot were found and what could have prevented any of these serious accidents? That information would be very useful.

The relief valves he mentioned no doubt stopped the fracture of the crankcase, but he did mention that due to pressure relieved, the flames and heat might cause death and injury to those in charge.

Mr. Pounder and Mr. Jackson had given them interesting particulars in the verbal discussion of what caused the explosion in some cases.

His own experience on land installations had been that piston blowby and hot lubricating oil were the most frequent cause of crankcase explosions. When bad blowby occurred the hot air went down past the piston skirt into the crankcase, making an explosive mixture of hot air and oil.

The piston skirt ran dry and hot, and seizure due to piston skirt expansion caused a very hot spot. Under these circumstances the smoke detector could not operate with accuracy.

If the lubricating oil were reasonably cool, it would damp down the hot spot; he had known of bad seizures and no explosion.

He was not surprised by the number of crankcase explosions that had caused relief valves to be fitted, seeing that no attention appeared to be given to the rise in crankcase temperature. The higher the temperature of the lubricating oil, the greater the risk of a hot spot.

On opening crankcase inspection doors after running an engine under what were considered normal conditions, there was a cloud of mist so dense that one could not see inside the crank chamber and the result of touching a big end bearing was a burnt hand. Forty years ago when he was working on Diesel engines, they would not have dared to continue running an engine under these conditions without finding the hot spot. They did not hear of trouble in those days due to running with cool crank chambers. In fact they touched the crank chamber doors to detect hot big end bearings.

It must be agreed that with the higher sulphur content, fuel liner wear was decreased if cylinders were kept at high temperatures, and it was advisable to prevent the presence of any moisture in the crankcase.

In his opinion a cold supply of circulating water for the oil cooling system, with efficient coolers, and a hot supply for cylinder cooling was required, and the extra cost would be justified by better running conditions.

The engine salesman and lubricating oil suppliers had their own particular reasons for claiming that working conditions were improved by high crankcase temperatures.

Manufacturers need not supply larger lubricating oil pumps and oil coolers or a separate water cooling system to keep the working parts cool.

The lubricating oil salesman claimed that the doped lubricating oil his firm supplied would be satisfactory at high temperatures, would not deteriorate, and would remove carbon and keep it covered in oil and prevent wear of surfaces; also a lot of detergent oil was used—because it could not be satisfactorily cleaned.

His company had proved to its satisfaction that if pistons received proper attention and a liberal supply of good clean oil of the correct viscosity, cooled to the right temperature, were circulated through the engine system, the result was a quiet running engine free of bearing troubles; and there should be no ignitable vapour or hot spot which increased the possibility of a crankcase explosion.

Mr. R. W. Cromarty (Vice-President) thought that many superintendents reading the author's paper would be able to identify some of the cases he described. He knew he could, and, in view of this, wondered why all too many engine builders still buried their heads in the sand like ostriches rather than in shame.

They too must know about these cases and the costly damage that arose from them. That they continued to recur not only supported the author's contention that their importance was not appreciated but also that nothing effective was being done to remedy them.

The Institute and the author were to be congratulated for a paper so full of interest and common sense. It was to be hoped that the lessons arising from the investigations of the cases given in the paper would be taken to heart.

It was quite apparent that the failures reported under "Welding" were the result of poor fitting of the parts which were attached by fillet welds. In his opinion Lloyd's Register should never permit fillet welding for loaded parts such as were indicated by Figs. 15 and 16—only full depth welding could make a satisfactory connexion.

So many failures arose from defective material that it would seem evident that the method of production or manufacture of materials and their subsequent inspections or tests were at fault. But it still went on. Almost daily engine builders asked superintendents to accept porous castings and the like without any sort of guarantee. This, of course, was to be preferred to machining castings under size, presumably to get rid of all obvious defects, without advising anyone, and have them fail in service.

The failure of coupling bolts as shown in Fig. 24 might be lessened if the bolt holes were machined or ground to their final size rather than "finished" by a hand operated rimer. In this connexion, his company had for many years fitted only taper bolts in the intermediate shaft couplings, and these had never given any trouble. But both engine builders and classification societies did not like them for some reason or other. Perhaps the author could let them know if taper coupling bolts were more prone to failure than the parallel type.

Fractures of the type shown in Fig. 28 were still all too common. Every bolt should have a substantial radius at the bolt head and every bolt hole should have the corners radiused or chamfered. In addition, all bolts should be proved that the faces of the nuts and bolt heads bore squarely on the face of the coupling or whatever they were holding together.

Even today it was still the practice to use 1-in. bolts in $1\frac{1}{8}$-in. holes to secure the guides to the columns of a well known engine. In fact, until quite recently only relatively small dowel pins located the guides.

Failure by engine builders to profit by the knowledge given so freely by the author to correct and improve the reliability of their machinery could only be due to failure to appreciate in these days of full employment its ultimate reaction, or an obstinate refusal to admit their errors and mend their ways.

Engine builders might consider offering suitable appointments to active retired superintendent engineers and surveyors, not necessarily full time, to "free-lance", and with authority to correct at all levels the errors and omissions that had resulted in the cases so well presented by the author, and for which he deserved their thanks.

Mr. J. F. R. Ellison (Member) wished to comment on the reference made by the author on page 380 to the importance of efficient locking arrangements for threaded components.

The following account of a serious smash-up was an instance of the inefficiency of the castle nut and split pin type of locking device, which, in his opinion, should not be allowed in any moving part of a reciprocating prime mover.

Although at least two such incidents, to his knowledge, occurred some years ago, no account had been published, and the risk of such an occurrence should be fully realized.

On certain two-stroke Diesel engines the piston crown was connected to the rod by studs fitted with castle nuts and split pins or locking wire between every two studs. Repair facilities being as they were today, the nuts were sometimes over or under-tightened, with the result that a stud might fatigue and fracture. The broken portion was retained in the skirt and in a very short time peened the whole surface of skirt and piston rod, work hardening the metal in the process. The noise of a broken stud bouncing about inside a piston skirt in the crankcase could not be heard and might go unnoticed for a considerable period, with the result that the skirt might eventually fracture at the top end where it changed section and thinned down to a larger internal diameter to slide on the piston rod flange and crown. The sequence of events was then as follows: The work hardening effect on the remainder of the skirt, or slight inaccuracy in the machining of the flange attaching it to the piston rod at the bottom, caused the top end of the skirt to spring out of concentricity as soon as the guiding effect of the portion surrounding the piston crown was lost due to the fracture. At bottom dead centre the fractured edge of the skirt came below the scraper box and on the next upward stroke fouled the box and the end of the liner and smashed them to pieces. The resistance offered to the rising piston was sufficient to stretch and fracture the studs holding the guide slipper to the crosshead, and the whole line of piston, crosshead and connecting rod was emptied out on to the plates, smashing everything in their path, whilst the guide slipper fell into the crankcase. (See Figs. 33, 34 and 35.)

The measures taken to avoid a recurrence of this accident were as follows: 1) Studs were secured in the piston crown by

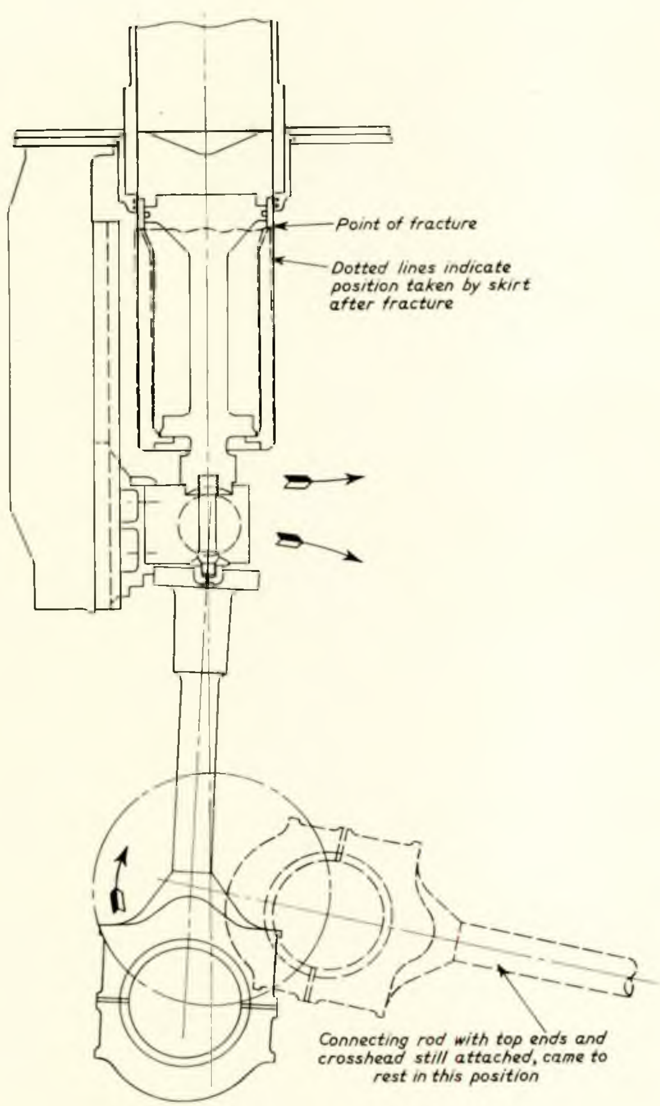



FIG. 33—*Sketch illustrating sequence of events after piston skirt had fractured*

grub screws inserted from the periphery of the crown; this method was considered superior to the use of collar studs. 2) The studs were provided with an extension of smaller diameter screwed left-hand thread for a locking nut, and a tab locking washer inserted between both nuts. 3) The change of section of the skirt wall was made more gradual.

MR. W. F. JACOBS (expanding his verbal comments) remarked that in this particular paper the aim of the author appeared to be to show some machinery defects with the idea of finding out why they occurred, thereby learning how future defects could be avoided—either in design or method of operation. He proposed to take a few of the cases in the paper and give his ideas of how the defects occurred.

With regard to Fig. 1, one could hardly think the whole oil system was not cleaned out before erection, but it was possible that some of the items, pipes or fittings, were not fitted with blind flanges the whole time between cleaning out and fitting into place—and it was on the site where foreign matter made its way into the system. Once any oil, fuel or lubricating lines were cleaned out, they should be blanked until the final fitting in place was complete.

Coming to Fig. 2, what was hard to understand was that the temperature at the bearing oil pocket (assumed) was about 275 deg. C. If so, it seemed very high indeed, and indicated that steps should be taken at once to lower the temperature.

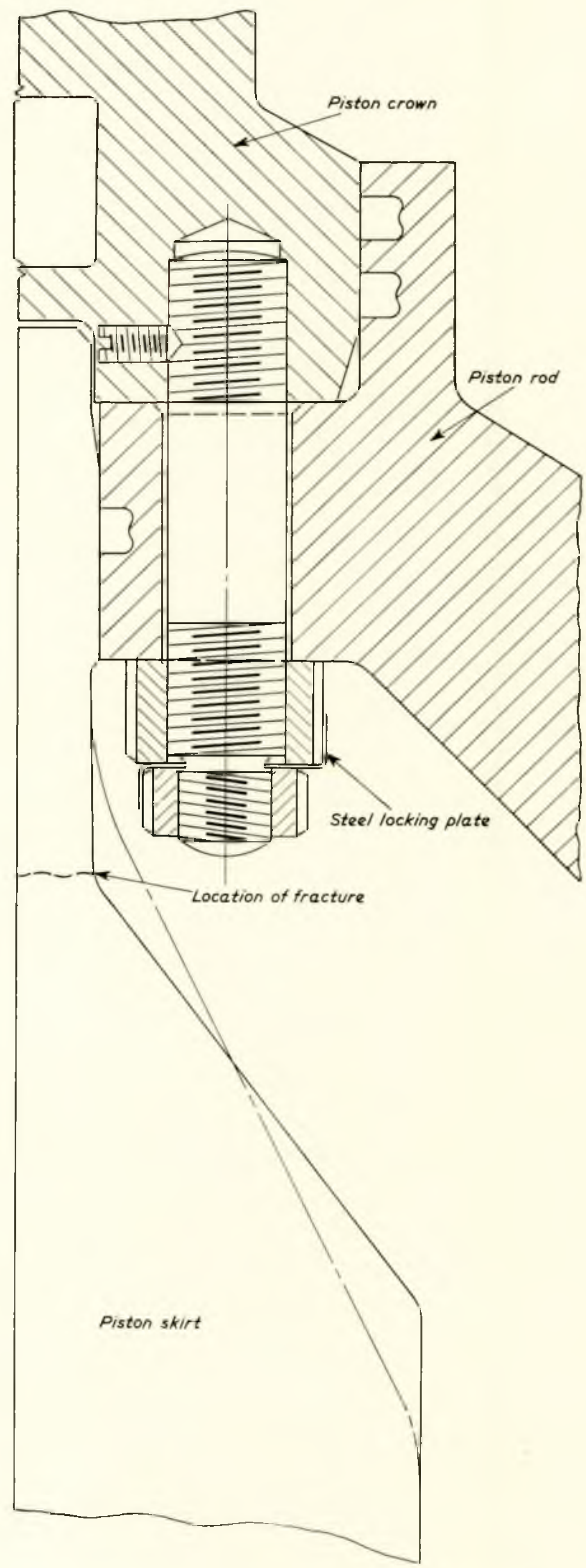



FIG. 34—*Sketch showing modifications to piston crown studs 2nd top of piston skirt*

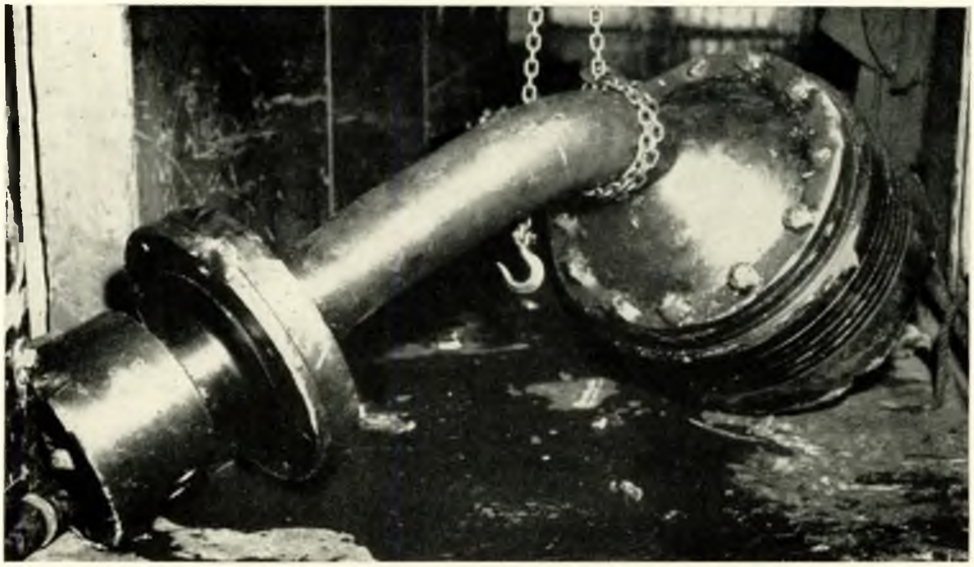


FIG. 35—*Damaged piston 2nd rod*

In dealing with high pressure, high superheat turbines, perhaps they might be given the highest allowable temperature at the bearing thermometer pocket. Many of them would be alarmed at a temperature of 200 deg. F. at a bearing pocket.

The lesson to be learned from Fig. 3 was that it was advisable, for all high speed revolving systems, to have the criticals calculated—a job for the computers.

Coming to the breakdown shown in Fig. 4, of a turbo-generator on trial, from the information given it would seem that owing to the failure of some locking device, a nut securing a thrust collar slacked back and allowed a quickly revolving part to come into contact with a fixed member, causing vibration, shaft bending and severe damage.

If the thrust washer in question was for taking the normal thrust of the turbine, it might be considered advisable to alter the design so that the usual thrust was taken up by an integral collar and not one held by a nut. If the vibration was noticed before a dangerous overspeeding occurred, one wondered why the machine was not stopped as soon as the vibration was noticed. Why did the collar securing nut slack back? A tag washer of the correct size and material, properly fitted and turned on to the nut was most reliable. If safety measures were designed but not properly fitted, the cause of the failure was human error. If it was absolutely necessary to have a loose collar or similar fitting secured by a nut on a shaft rapidly revolving system, it would be advisable to have a fine thread on the shaft for the nut and that so "handed" (right or left as the case might be) that the normal tendency, due to inertia, oil, washing, etc., would be to tighten the nut.

The slipped gear wheel rim shown in Fig. 7 seemed to have been caused by the cast iron centre being made of an unsuitable grade of cast iron. For this job a fairly tough grade of iron, hard cylinder or nickel cast iron, seemed the correct grade to use, and the cost would not be very different from that used.

In regard to welding, it would be as well if as much thought were given to repairs carried out by welding as was given to new work.

With regard to crankshaft failures, had "rolling" the fillet between pin or journal been tried, as in motor car crankshafts? He realized that there was a great deal of difference between a shaft of 3-in. diameter and one nearer 30in. The Motor Industry Research Association claimed that it had lessened crankshaft failures where it had been tried.

In the cases mentioned on page 389 dealing with the breaking of four studs which secured a piston cooling assembly (and presumably in several vessels of the same class), as this occurred in several vessels with the same design of engines, one might look for a defect in design—even if it were only of the kind in which the correct hardening of the nuts was difficult, such as the nuts being pocketed requiring special spanners, or having to use a spanner and having no room to harden up properly; otherwise, if the design were right, these nuts might have been hardened up by one and the same fitter who had a weak spanner arm. No doubt, the matter having become known, frequent hardening of the nuts was now carried out, with or without head office orders, so no further trouble had occurred.

In the case of the cracked furnaces due to oil in the boilers, might they assume that the vessel was fitted with reciprocating engines using superheated steam, and that generous charges of cylinder oil were used to save wear during running in? It seemed that the amount of oil used was far too much for the size of the job and that frequent inspection of the filters was omitted. It seemed that clear instruction should be supplied in the case of staffs with no previous experience of these jobs. Among the engineers of his acquaintance, getting oil into the boilers was a crime akin to manslaughter in their eyes, if not other people's, for it led to furnace crowns down and possible explosions.

Fig. 31 showed corrosion due to acidic oil in the crank-case, this being due to leaking piston and scraper rings. No doubt these rings should have been rectified, but they did not know the conditions of service, or the time away from a home port. However, the corroded purifiers should have indicated the trouble and much less corrosion would have resulted if the defective parts had been renewed. In these days, if the vessel were away from home ports, renewals for purifiers could have been flown out. Also, washing the oil would have resulted in the acid content being much less and the purifiers also would have got rid of much of the water content of the oil. If the vessel touched home ports it seemed that advantage had not been taken of the oil companies' services in examining and reporting on samples of oil drawn from the sump in use. It was noted in the photographs of the corroded surfaces that on the journal there was much less corrosion in the way of the oil groove in the bearing than on the surfaces in contact with the white metal. It would be interesting to know if the vessel had spent much of her time in port and if the crank-shaft were allowed to remain unmoved for the greater part of the time. When vessels were laid up, it had been a standard precaution to turn the engines a little now and again to prevent corrosion of the working surfaces.

In conclusion, it appeared that during the last decade or so, two new additional factors had entered into the design and operation of marine machinery. These were vibration and kinetic stresses, both due to higher revolutions. All engineers should give these factors due consideration. Bolt and nut, etc., locking methods must be reliable and efficient—many which satisfied at low speeds did not do so with high speeds. Split pins were under suspicion, and frequently were more a comfort than a reliable locking device. They were only satisfactory if of sufficient size, a good fit in the bolt and nut slot, and the nut was really hardened up, and not to a convenient nut slot—there being a tendency for the tightener to tighten up to the convenient slot only. Split pins through a hole in the bolt only and on top of a plain nut were not worth the bother of fitting to high speed machinery. Tag washers and locking plates (the bolts of which were themselves well locked) were very satisfactory.

There should be no attempt by non-engineering personnel, no matter what position they might hold, to try to hurry up the departure of a vessel before the engineers were satisfied that all was ready. All the engineers he had met made it a point of personal honour to get the ship away as soon as it was advisable to do so.

It must be remembered that all these lessons must be learned afresh by every engineer as he became more experienced. Nothing was inherited, however much many things might seem second nature to some of the old experienced hands.

DR. INGVAR JUNG and DR. GUNNAR OHLSSON wrote that, although the De Laval Ljungström Turbine Company now only manufactured gears of an all-welded design, they had had some experience with shrunk-on gear rims in the past and would like to comment upon the author's information on slipping gear rims.

The author mentioned that a shrunk-on gear rim with an interference fit of 1/1200 of the diameter had slipped, although the "theoretical factor of safety" was over 10. This factor of safety was probably computed on the assumption that the rim slipped circumferentially like a solid body due to the tangential gear force; that was to say, that all points on the rim would start to slip simultaneously and slip an equal amount. The tangential force would then be equal to

$$P = \mu p d b \qquad (1)$$

where $\mu$ was the coefficient of friction and $p$ the shrink pressure between gear rim and wheel, $d$ the diameter and $b$ the axial width of the rim.

However, it appeared that part of the rim near the tangential tooth force might start to slip when the tangential force was considerably smaller than given by equation (1). When the force moved along the periphery, other parts of the rim slipped. Thus the rim slipped gradually. This might be easier to understand if, instead of the steel rim, one considered a rubber band stretched around the wheel and tried to move this in the peripheral direction.

Professor F. Odquist of the Royal Institute of Technology in Stockholm had given a theory for this kind of slip, based on a simple model (Fig. 36).

The tooth force $T$ might be resolved into a tangential

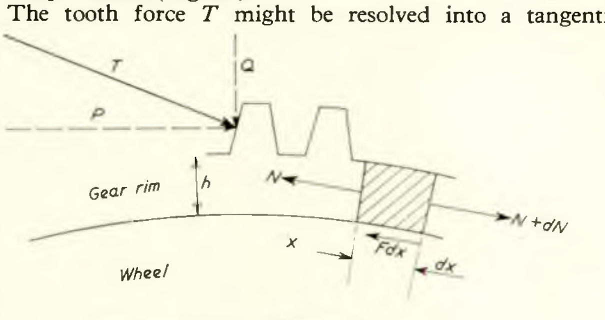



FIG. 36

component $P$ and a radial component $Q$. An investigation showed that these forces had a fairly small influence on the pressure $p$ between gear rim and wheel for common gear dimensions. Therefore this pressure might be taken as constant and computed from the interference fit and the elastic properties of the rim and wheel. The peripheral force $P$ is resisted by a friction force $F$ lb./in. of the periphery. $F$ usually varied but could not exceed a certain value $\mu\, p\, b$, when the friction force was fully developed. If no slipping occurred, the rim might only be deformed elastically and

$$F = Ku \qquad (2)$$

where $K$ was a constant and $n$ was the tangential displacement of the middle of the rim.

A force balance for the element $dx$ in the tangential direction (Fig. 36) gave

$$F = \frac{dN}{dx} \qquad (3)$$

From Hooke's law the normal force was

$$N = Ebh\frac{du}{dx} \qquad (4)$$

where $h$ was the radial width of the rim and $E$ Young's modulus of elasticity.

If at a certain distance $x \leq l_0$ from the force $P$, *no slipping* occurred, equations (2), (3) and (4) gave

$$\frac{d^2u}{dx^2} - n^2u = 0 \qquad (5)$$

where

$$n^2 = \frac{K}{Ebh}$$

If *slipping* occurred within a certain region $x \leq l_0$, then

$$F = upb = ku_0 \qquad (6)$$

where $u$ was a constant equal to the displacement at $x = l_0$. Equations (3), (4) and (6) gave

$$\frac{d^2u}{dx^2} - n^2u_0 = 0 \qquad (7)$$

The general solutions of the equations (5) and (7) were, respectively,

$$x \geq l_0: \qquad u = A \sinh nx + B \cosh nx \qquad (8)$$

$$o \leq x \leq l_0: \qquad u = n^2 \cdot u_0 \cdot \frac{x^2}{2} + Cx + D \qquad (9)$$

where $A$, $B$, $C$ and $D$ are constants, which might be determined from the boundary conditions.

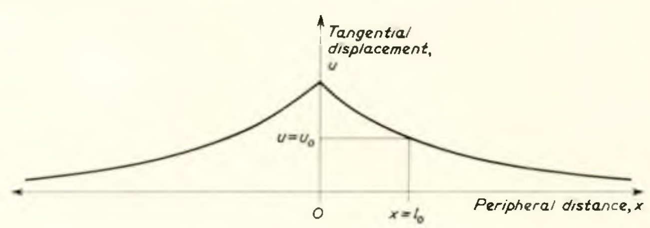



FIG. 37—*Rim without dowel pins*

Fig. 37 showed the result for partial slipping of a shrunk-on rim without dowel pins. Maximum slipping occurred where $x = l_0$, that was, where the force $P$ was applied. The displacement then decreased to $u_0$ at $x = l_0$. For $x$ greater

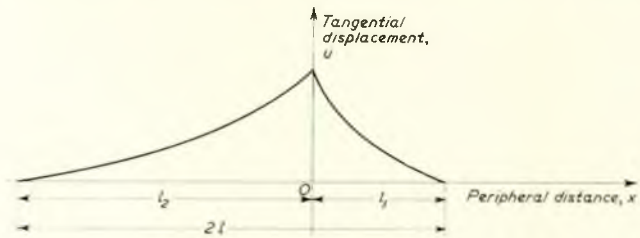



FIG. 38—*Rim with dowel pins at distance 2e*

than $l_0$, only elastic displacements without slipping occurred.

Fig. 38 showed the case of a shrunk-on ring with dowel pins at a distance $2 \cdot l$. The force $P$ was applied at $x = 0$ at a distance $l_1$ from one pin and $l_2$ from the other. In this case the dowel pins, which might be axial or radial, were assumed to be completely stiff, so that no displacement $u$ could occur at the pins. When the pin was completely stiff, it had to resist the whole force $P$, when $P$ was at the pin.

This, however, was a limiting case which did not occur in actual gears. Depending on the stiffness at the dowel pin, the pin only had to resist part of the total force $P$.

If only the rim were considered to be elastic and the wheel were assumed to be completely stiff, the theory gave as a result that slipping might occur at a tangential force which was many times smaller than the force given by equation (1). If the wheel were also elastic, the maximum force for no slipping to occur, was larger.

Both theories predicted that the tangential force might be increased if the interference fit, and thus the pressure $p$ between gear rim and wheel, were increased. However, in this case the hoop stress might increase and cause the rim to fail due to fatigue at the tooth root because of stress concentration at that point.

It was therefore concluded that:

1) Gradual slipping would occur between a shrunk-on rim and the wheel at a much smaller tangential tooth force than given by equation (1). It was thus not safe to use this equation.
2) If dowels were used and the spacing was great compared with rim thickness, a safe rule would be to dimension these pins so that each axial pair of them could resist the tangential tooth force.

Mr. P. Pluys (Member) thought that no doubt everyone—and shipowners in particular—would have found Mr. Pemberton's paper most interesting.

Reference was made by the author to ultrasonic testing. He would like to describe one or two of the many applications his company usually made of that non-destructive control method precisely with a view to overcoming the recurrence of some severe breakdowns experienced in the past with typical engine parts.

He thought they would all agree that the interpretation of ultrasonic tests was often difficult. They therefore never discarded any engine part found to be doubtful by that method and always tried to have the results confirmed by more accurate means such as magnaflux.

All they expected from ultrasonic tests was a *warning*, and were confident that this method would show *important* cracks—in other words, dangerous failures which could lead to a breakdown in a short time—provided that the instrument was always used by the same operator dealing with similar engine parts.

Despite the fact that double-acting Diesel engines were obsolete—hundreds of them were still in service—everybody knew how difficult it was to make a good and safe connexion between piston rod and crosshead. After three spectacular accidents where rods had failed when the engine was running, it became essential to increase the safety margin as they had about 100 of those rods continuously in service. They first discarded the double-nut connexion and gave the preference to the solution with a conical seat at the top and only one tightening nut. Afterwards they decided to make the ultrasonic testing of all rods each time such ships were in a home port, *i.e.* every six to ten weeks. Such a job, which was now done regularly, required the disconnexion of the oil pan under the crosshead to reach the end of the rod with the ultrasonic testing probe. Moreover, to make sure that the pre-tightening was maintained they fitted a system to check the remaining pre-strain on the spot—especially with a one-nut system. According to their statistics this had undoubtedly considerably improved the safety of that vital part. Indeed they found that a magnaflux inspection every twelve months was not nearly safe enough. Ultrasonic testing was also used on all tie rods, eccentric rods, connecting rod bolts, and generally speaking to all main parts subjected to alternate stresses. In one year two major accidents had been avoided by its use.

It was also customary to have the tailshaft of their ships ultrasonic tested at every drydocking. Although this meant disconnecting the propeller nut hood they considered it was worth while doing so. He would like to have Mr. Pemberton's personal opinion on ultrasonic testing when used in such a way.

Finally, a typical example which he thought might be useful to describe briefly concerned the use of magnaglo and/or ultrasonic testing for gears. His company still had a few Victory ships in operation and after fourteen years one of them had gears showing teeth with rather extensive wear. Careful visual examinations were regularly made in the United States by really specialized people and favourable reports were systematically issued. Nevertheless, two teeth broke in service a few days after such an examination. Fatigue cracks which had most probably developed some time before were evident and proved that visual control—however carefully carried out—was not foolproof. To avoid the recurrence of such a case, which might be of serious consequence, they decided on a general magnaglo and ultrasonic testing of all gears, tooth by tooth. Such a job took about two to three days on a Victory ship, and he was wondering whether that sort of test was not the best for classification society surveyors and superintendents, when, dealing with gears with worn teeth, they had to decide how long they could be operated safely.

Mr. N. A. Vlassopulos (Associate Member) considered that this was the type of technical paper which, unfortunately, was published only too rarely, and served to accentuate some of the risks involved in ship operation. It was interesting to reflect that similar papers, about seven years ago, contained the word "unusual" in the title, and one could not help wondering if the author of this paper was tempted to precede the title by the word "modern" or "contemporary".

Certainly, some of the hull and machinery defects, or failures, of the past decade would make interesting reading, published in the form of a marine engineering textbook. Such a publication, even if depressing, would be invaluable to all engaged in naval architecture and marine engineering. Perhaps an authority such as Mr. J. H. Milton of the Engineering Investigation Department of Lloyd's Register could be encouraged into obliging them.

After reading this interesting paper one was left with the impression that the more spectacular, or unusual, the failure the more simple appeared to be the explanation. This appeared, particularly, to be so in the case of the turbine rotor shaft, where it seemed fantastic that such a reduction in cross-section could have been avoided by having an oil "leak off" pipe of greater bore, and changing the oil.

Were they to conclude that in the race for technological advancement the more simple, and fundamental, engineering principles were being overlooked? Or was it, in actual fact, a case of certain precautionary steps being taken to minimize the possibility of a repetition without having found the initial cause?

#### Author's Reply

The author, Mr. H. N. Pemberton is at present visiting the United States and Canada and is re-presenting his paper to the Sections there. He will make his written reply to the Discussion when he returns to the United Kingdom in November, and this will be published in a later issue of the Transactions.

# INSTITUTE ACTIVITIES

**Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 29th March 1960.**

An Ordinary Meeting was held by the Institute on Tuesday, 29th March 1960 at 5.30 p.m., when a paper entitled "Marine Machinery Failures" by H. N. Pemberton (Member of Council) was presented and discussed. Mr. W. R. Harvey (Vice-Chairman of Council) was in the Chair and 195 members and visitors were present. Ten speakers took part in the discussion that followed.

A vote of thanks to the author, proposed by the Chairman, was accorded by acclamation. The meeting ended at 7.45 p.m.

**Autumn Golf Meeting at Hadley Wood Golf Club**

The Autumn Golf Meeting took place at the Hadley Wood Golf Club on Thursday, 22nd September 1960. In spite of the course playing long due to the spell of wet weather, 36 members enjoyed the morning and afternoon competitions.

In the morning the Stableford Competition was won by F. Sands (8) with a total of 37 points. Commander J. White (13) was second with a total of 36 points. An additional prize given by a member was awarded to Mr. C. J. Probett.

The Greensome Bogey Competition in the afternoon resulted in a win for Messrs. C. A. Larking and D. G. Welton, two up, Messrs. L. E. Smith and R. R. Strachan were second, two down.

Mr. Stewart Hogg, O.B.E., Chairman of the Social Events Committee, presented the prizes and thanked the Committee of the Hadley Wood Golf Club for their hospitality. He also thanked the contributors to the prize fund and the Secretary for the arrangements made for the meeting.

It was announced that the next meeting would be the Spring Meeting at Moor Park Golf Club, on the 17th May 1961, and that the Autumn Meeting 1961 would be held at the Berkshire Golf Club, on the 27th September.

**Section Meetings**

*Scottish Section*

The opening meeting of this Section took place on Wednesday, 12th October 1960, at 7.30 p.m. in the Weir Hall of the I.E.S., Glasgow.

Mr. J. W. Bull, Vice Chairman, presided in the unavoidable absence through illness of the Chairman, Mr. T. W. D. Abell.

After apologising for Mr. Abell's absence, the Vice Chairman welcomed all members and visitors to this opening meeting and then introduced Mr. James Brown who read Mr. Abell's paper entitled "Rationalization in the Marine Engineering Industry".

This controversial, constructive, and well-written paper was ably delivered by Mr. Brown before an audience of leading Scottish shipbuilding and marine engineering personalities and representatives of the Press.

Due to the absence of the author, it was decided not to have a discussion following the reading of the paper.

A vote of thanks to Mr. Abell for his valuable paper and to Mr. Brown for so ably presenting it, was very aptly proposed by Mr. F. R. Topping, B.Sc., Member of Committee.

The meeting terminated at 8.55 p.m., after which light refreshments were served.

*Singapore Section*

On Saturday, 6th August, members of the local section and the Joint Group, together with their guests, visited the St. James Power Station, Singapore.

There were 57 acceptances including guests and of this number about 45 attended. The attendance of Section members was disappointing.

Mr. M. Morris of the Free Piston Engine Co. Limited acted as host and guide, being assisted by his colleagues from his own company, the consultants and the power station staff.

St. James Station was originally a steam station commissioned in 1925. The present installation when complete will comprise forty-eight Free Piston gasifiers in banks of eight, each of which is connected directly to one of six 6,000 kW gas turbines. The total output of the station will be 36,000 kW when all units are operating; at present 18,000 kW are being generated.

The gasifiers were manufactured by the Free Piston Engine Co. Limited and are of the GS34 type each developing 1,250 g.h.p. The turbines were manufactured by Messrs. C. E. M., French licensees for Brown Boveri turbines. The installation was designed to fit into the existing St. James building with a minimum of structural changes; the design of the old building therefore had considerable influence on the layout of the plant. The gasifiers are arranged in six banks each of eight gasifiers on the same level as the turbine room floor. Below each gasifier is situated a surge chamber with a Venturi air inlet to damp pulsations in the inlet system. Two banks of eight gasifiers draw their air from a central channel which is further fitted with oil coated screen type filters to prevent ingress of grit from a nearby cement works.

The gasifiers are at the present burning Bunker C fuel of a viscosity of 800 secs. Red. 1 centrifuged and heated to give a viscosity of about 75 secs. for atomization. The gasifiers are started up and shut down on gas oil to prevent clogging of the fuel lines. The gas enters the turbines at about 45lb./sq. in. and 850 deg. F. exhausting to atmosphere at 400 deg. F. There is no waste heat recovery. The design of the air and gas ducting is so carried out that the noise level in the gasifier house is no greater than one would expect from a medium speed Diesel installation of about the same horse power.

The station has been designed to fulfil the functions of both peak and base load station and the novel features of the installation ensure that it will carry out both functions at a reasonably high efficiency.

The station was officially opened by Dr. Goh Keng Swee, the Minister for Finance on 2nd July. The fact that on completion at the end of this year St. James will be the largest station of its type in the world speaks well of the progressive outlook of the Singapore State.

*Sydney*

Fifty-one members and guests attended a meeting of the Sydney Section at Science House, Gloucester Street, Sydney, on 18th July 1960.

Captain G. I. D. Hutcheson, R.A.N. (Local Vice-President) presided over the meeting at which Captain A. M. Clift, R.A.N. gave an address on "The New Look in a Naval Dockyard".

Captain Parker, Commanders Elliott, McMahon and Pigott and Mr. Joselin took part in the discussion which followed.

A vote of thanks to Captain Clift was proposed by Mr. W. H. Gregory and carried with acclamation.

Supper and refreshments were served as usual.

A meeting of the Sydney Section was held on 16th September 1960 at Science House, Gloucester Street, Sydney.

Captain G. I. D. Hutcheson, R.A.N. (Local Vice-President) presided over the meeting which was attended by seventy-four members and guests.

"A Recent Visit to the United Kingdom and North America" was the subject of a talk by Captain R. G. Parker, O.B.E., and Captain Good, Commander McMahon, Messrs. McGillivray, Buls, Munro, Miller and the Local Vice-President contributed to the discussion which followed.

Mr. McGillivray proposed a vote of thanks to Captain Parker, which was greeted with acclaim.

The usual supper and refreshments were served at the end of the meeting.

*West of England Section*

A general meeting of the West of England Section was held at the University of Bristol on Monday, 10th October, at 7.30 p.m. Mr. W. John, M.B.E. (Chairman of the Section) presided and there was an audience of twenty-three, including the Local Vice President, Mr. D. W. Gelling.

The Chairman then introduced Mr. A. Bell, B.Sc. (Associate Member) who gave a very interesting lecture entitled "Corrosion and Fires in Marine Boiler Air Preheaters" which he illustrated with numerous lantern slides.

In the discussion that followed time would only allow for three speakers to take part and Mr. Bell dealt with their questions in a satisfactory manner.

A vote of thanks to the author, proposed by the Chairman, was accorded with acclamation.

The meeting ended at 9.15 p.m.

On Friday, 16th September 1960, a Dinner Dance was held at the Royal Hotel, Bristol, to mark the opening of the 1960/61 session.

This function took the place of the Annual Cocktail Party which has been held in passing years.

The 81 members and guests were received by the Section's Chairman, Mr. W. John, M.B.E., and Mrs. John, at a reception prior to the Dinner. The principal guests were the Chairman of Council, Mr. W. R. Harvey, and Mrs. Harvey, the Chairman of the South Wales Section, Mr. Richardson, and Mrs. Richardson, and also the Group Resident Superintendent of Atomic Power Projects at Hinkley Point, Mr. R. Wall and Mrs. Wall.

After the "Loyal Toast" the Chairman proposed the toast of "Guests and Ladies".

Mr. Harvey, replying on behalf of the Guests and Ladies, thanked the Chairman for inviting both Mrs. Harvey and himself and said that he brought with him the best wishes

*The West of England Section Dinner and Dance*

*(Photograph by R. C. Rollason, Ltd., Bristol.)*

*Left to right—Mr. W. R. Harvey (Chairman of Council) and Mrs. Harvey, Mr. W. John, M.B.E., (Chairman, West of England Section) and Mrs. John, Mr. Richardson (Chairman of the South Wales Section) and Mrs. Richardson, Mr. D. W. Gelling (Local Vice-President (Bristol)) and Mrs. Gelling*

of the Council in London for the continued success of the Section's activities. He thought that the ladies, in particular, would be pleased that the Dinner and Dance was to be an annual function in place of the previous social gatherings, and he and Mrs. Harvey hoped that they would be fortunate enough to be invited next year.

Dancing to the orchestra of Arthur Alexander followed the dinner, including a number of novelty dances, until hands were joined at 1.00 a.m. for Auld Lang Syne.

**Election of Members**

*Elected 17th October 1960*

MEMBERS

William David Adam
Reginald Thomas Bargewell, Eng. Lieut., D.S.M., R.N.
Sydney Francis William Blackall, Lt. Cdr., R.N.
Eric John Blackledge
Dennis Buckley
Charles Stuart Church
Stanley Harris
John Joseph Joyce
Ingvar Karl Einar Jung
Thomas McGhee
Harry L. Morgan
John William Charles Pye
Eric A. Roberts
Ernest Thexton
Philip James Walsh

ASSOCIATE MEMBERS

John Thomas Bramley
Thomas James Browne
Richard Henry Butt
Gordon William Calvey
Gordon Campbell
Santi Nath Chatterjee
George Goodwin Connor
Alexander Donaldson Cowie
Leo Allan James Coote
Hugh Crawford
Roland August De Guyper
Zbigniew Drewnicki
Herbert William Joseph Duarte, Sen. Cd. Eng., I.N.
Michael Charles Fox, Lieut., R.N.
James Konynenburg
Donald McIntosh
Parvez R. Mehta
John Milne
William James Mitchinson
Graham Linsley Gifford Moore, Lieut., R.A.N.
Michael Morrison
Gordon Munro
Hejmady Narayanan
George S. Panas
Bert Pearson
Sidney Peel
Frederick Bernard Price
Trevor Dresser Saul
Ashley Ignatius Stephenson
Srikantan Ramanathan Viswanathan
Hendrik Vogelzang
Herbert Basil Macleod Vose, B.E.(Sydney)
John William Warden
George Morrison Watson
William Wedderburn
Lawrence Kenneth Wong

ASSOCIATES

Mundakal Jacob Alexander, Cd. Eng., I.N.
Jai Chand Dhakarwal, Cd. Eng., I.N.
William Edgar Doole
Krishna Sadashiv Durge, Cd. Eng., I.N.
Giuseppe Tommaso Gonzales
William Keith Hillis
Ronald Ibbitson
Lodewyk Krygsman
George Charles Parritt
Bryant Paterson

William John Peart-Jackson
Gopendra Narayan Sen
Bhag Singh, Cd. Eng., I.N.
Kenneth Charles Thomson
Peter van Strien
Peter Ernest Walker, Cd. Eng., I.N.
John Arthur Wilde

GRADUATES
Gordon Hamilton Boyd
Gerard Maurice Holmes
Prabhat Kumar, Sub. Lieut., I.N.
Sharat Kumar, Sub. Lieut., I.N.
Roy James L'Enfant
Ramesh Kumar Mathur, Sub. Lieut., I.N.
Roy Mitchell
James Monaghan
Donald William O'Connor
John Craig Robertson
Harpreet Singh, Sub. Lieut., I.N.
Satish Kumar Singh, Sub. Lieut., I.N.
Surendra Singh, Sub. Lieut., I.N.
Ernest Sunny, Sub. Lieut., I.N.
James Syme

STUDENTS
Anthony John Barber
Patrick Thomas Booker
Kam Wing Cheung
David Chinery
David Anthony George
John Leslie Hall
Peter George Swift

PROBATIONER STUDENTS
Brian John Allsopp
Peter Graham Banks
Stephen Roger Bateman
Richard Benson
David Stanley Bird
John Michael Brewster
Thomas Joseph Williams Carey
James Thomas Casher
Kenneth Charnock
Rodney Stewart Clench
Anthony Peter Coleman-Wilson
Gerald Thomas Cordingley
John Ernest Coulthurst
Antony David Davies
Trevor Ivor John Davies
William Howard Davies
Vincent Clifford Grimshaw
David Edward Guppy
Lance Vincent Gwilliam
Ian Harper
Paul Heald
Peter John Houghton
John David Hudson
Anthony Francis Imaz
Howard Melville Johns
Colin Jones
Derek Robert Jones
Thomas James Lockyer
Harry Peter Loveday
David Anthony Lower
David William Manning
Rodney Mayor
Robert William Newrick
Victor Stephen Park
Anthony Peter Quinn
Michael John Robertson
Eric Jeremy Sample
Robert Ernest Smith
William Ralph Spencer
Brian Patrick Stewart
David George Sutton
Alexander Cameron Thomson
James Walsh
David John Whitby
Christopher David Whittaker

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER
Peter John Atkinson
Gordon Kenworthy-Neale

TRANSFER FROM ASSOCIATE TO MEMBER
Stanley Clarke, B.Sc(Queen's Ont.)
Walter Thomas Collison
Rashioor Natarajiar Kripa

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
Narayan R. Bhalinge
Thomas Leslie Skipp

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Donald James Beswick
Peter Broadway
Montague Neil De La Harpe
William Leaver Evans
Terence Charles Harris
John Arthur McDavitt
Henry Harvey Murdoch
Uday Shanker Sharma
Douglas Steen
Peter Frederick Wall

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER
Yan-Ming Yip

TRANSFER FROM STUDENT TO GRADUATE
Roy Robert Grenham
Michael John Payne, B.Sc.(Eng.) Durham
David James Williams

TRANSFER FROM PROBATIONER STUDENT TO GRADUATE
Keith Huntley

TRANSFER FROM PROBATIONER STUDENT TO STUDENT
Anthony Charles Clark
David Michael Fuller

It is regretted that an incorrect announcement appeared in the September "Election of Members".

Under the heading ASSOCIATE MEMBERS the reference to John Millar should have read: John Millar, B.Eng.(Hons.) Liverpool.

# OBITUARY

Darrel Campbell De Lastic (Associate Member 17672) was born on 8th June 1928. He served his apprenticeship between 1946 and 1951 with the Garden Reach Workshops Ltd. of Calcutta. After completing his apprenticeship, he joined the Asiatic Steam Navigation Co. Ltd. as fifth engineer for one year. In 1953 he joined Kentships Ltd. serving as fourth, third and second engineer until in 1956 he left this company. He was elected a Graduate of the Institute and obtained his First Class Ministry of Transport Certificate in the same year. He worked in 1957 successively for the Eagle Oil and Shipping Co. Ltd. and from October of that year with the United Baltic Corporation Ltd. serving on m.v. *Baltic Trader*.

Mr. De Lastic was transferred to Associate Membership in 1958. The date of his death is unknown to the Institute.

Norman Fay (Member 9596) died on 4th July 1960 aged 71 years. He served his apprenticeship with the Liverpool Engineering and Condenser Co. Ltd. between 1905 and 1910. He went to sea with various companies including the Cunard Line for five years and during this period obtained his Second and First Class Board of Trade Certificates. In 1915 he joined the Royal Naval Reserve and served as engineer sub-lieutenant on H.M.S. *Caronia*. He was invalided out of the Navy in 1918 and for a year while convalescing worked as assistant in the metallurgical laboratory of the Cunard Steamship Co. Ltd. After this he branched out into his own business founding the firm of Fay's Motors in the Crosby Road, Liverpool. After the war Mr. Fay moved to Bournemouth where he worked as a consulting engineer until, forced to retire through ill health, he returned to Liverpool in April of this year. Mr. Fay was a keen supporter of the Southern Joint Branch of the Institute and the Royal Institution of Naval Architects at Southampton and was elected a Member of the Institute in 1943.

Charles John Maynard Flood (Member 6463) died on 27th July 1959 aged 67 years. Mr. Flood served a six year apprenticeship at H.M. Dockyard, Chatham which he concluded in 1914. He was educated at Gillingham High School, Kent and at H.M. Dockyard Upper School in Chatham. He also studied at Gillingham Technical Institute and the East London College. After two years as a draughtsman engaged on submarine work at Chatham Dockyard, in 1917 he became assistant in the Admiralty Engineering Laboratory working on submarine engineering research under Commander Hawkes and Sir Dugald Clark. In 1919 he moved to Wimbledon Technical College and remained there for over 20 years, becoming head of the mechanical engineering department in 1939.

Mr. Flood was a Whitworth Exhibitioner (1914), a Bachelor of Science, and had invented and patented the Flood Indicator Diagram Reproducer (Patent No. 286,575), which was marketed by Cussons of Manchester in 1930. In the same year Mr. Flood joined the Institute as a Member, and did much work on Institute committees over a considerable period of his life.

William Thomas Gwynne (Member 4142) died on a date unknown. He served his apprenticeship with the Cardiff Junction Dry Dock and Engineering Co. Ltd. and spent 16 years at sea, five as a chief engineer. In 1900 he obtained his First Class Board of Trade Certificate. He was for 22 years with the Mountstuart Dry Docks Co. Ltd. of Cardiff whom he had joined as head draughtsman in 1920. Mr. Thomas was elected a Member of the Institute in 1921 and a non-paying Member in 1951.

Ronald George Iliffe (Member 9566) died on 31st July 1960 aged 47 years. Apprenticed to Cammell Laird and Co. Ltd. of Birkenhead, between 1929 to 1934, he subsequently served this company as an apprentice draughtsman for one year before joining T. and J. Brocklebank Ltd., starting a long association with this firm. In 1940 he obtained his First Class Ministry of War Transport Steam Certificate and a Motor Endorsement thereto a year later. The same year he was appointed third engineer in s.s. *Machada* and then s.s. *Mahsud* and in 1945 second engineer to s.s. *Maihar*. On 1st September 1954 Mr. Iliffe was appointed chief engineer and sailed in this capacity on s.s. *Mathura*, s.s. *Maskeliya* and other vessels of the fleet. He served throughout the Second World War and followed in the footsteps of his father, who had also served a number of years for T. and J. Brocklebank Ltd. and, like his son, held the position of chief engineer.

Mr. Iliffe was transferred to Membership of the Institute in 1948, five and a half years after his election as an Associate.

Alfred Edgar Jones (Member 8549) was born on 15th February 1887 and died some time during this year. From 1903 he was apprenticed for five years with the Mersey Engineering Company, Liverpool. He held a First Class Board of Trade Certificate and was a seagoing engineer between 1908 and 1924, including in this time, four years' service in the Royal Naval Reserve. In 1926 he entered the employment of the Liverpool Screw Towing and Lighterage Co. Ltd. and in 1939 became marine superintendent of the company. He was elected to Membership of the Institute in 1937. For fourteen years until his retirement Mr. Jones was a senior engineer surveyor with Brookes Bell and Co. Ltd. of Liverpool.

Charles Victor Lewis (Member 2245) died on 10th August 1960 aged 70 years. He started his career as apprentice at the Thames Engineering Works Ltd., Greenwich and in 1918 joined Hammersmith Power Station where he was employed as engineer-in-charge. In 1924 he joined Thomas Firth and John Brown Ltd. of Sheffield as power station superintendent and in this year also obtained his First Class Board of Trade Certificate. In 1946 he was made power station superintendent and service engineer and continued in this post with Firth-Brown until his retirement in 1958. During this period Mr. Lewis exhibited a lively interest in public life and served on the local council at Dronfield, near Sheffield, for many years, being elected to the office of chairman for a term of two years.

Mr. Lewis enjoyed a long association with the Institute, being elected Graduate in 1909, Associate in 1912, Associate Member in 1915 and being finally transferred to Membership in 1935.