Some Interesting Ship and Machinery Defects, Their Investigation and Cure

A SYMPOSIUM OF SHORT PAPERS

1. EXCESSIVE CONSUMPTION OF LUBRICATING OIL ON DOXFORD ENGINE

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This incident relates to a new vessel which was one of the first to be fitted for the burning of heavy oil. On the maiden voyage, heavy blow past developed due to stuck power rings on the pistons. Time was taken at an intermediate port to correct this position, which was found to be due to the need for an increased axial ring clearance as compared with the builder's standard for engines designed to run only on Diesel oil.

When the vessel proceeded on her voyage it was found that there was a heavy loss of lubricating oil, this rising to an average of 14 gallons per day per cylinder (670 mm. bore). This was accompanied by a heavy discharge of oil from the scraper box drains, and on one occasion as much as 15 gallons was collected from the scraper drainpot of one cylinder in a period of 24 hours.

On completion of the voyage, examination revealed heavy carbon deposits in the scraper boxes and these were dismantled and cleaned out, new scraper rings being fitted. The net result of this refit was no improvement whatever in oil consumption.

Various further investigations were carried out to determine the source of loss and particular attention was naturally given to the inherent spring in the scraper rings, further replacements of which were carried out, all to no avail. Throughout this time it was noted that some cylinders were consistently worse than others as measured by the amount of oil collecting in the scraper box pots.

A scraper box on one of the worst cylinders was removed for detailed examination but it appeared to conform with the standard drawing in all respects. It was replaced, however, by a spare scraper box, which had been machined at the same time as the working boxes. Conditions in this cylinder regained normality immediately.

It was at this time that the builders learnt of a somewhat similar experience in the motor industry, in which grossly excessive oil consumption on new engines had been found to be attributable to conical landings of the scraper ring grooves, the coned effect being quite minute but sufficient to lead to the generation of an oil wedge against the working face of the the scraper ring.

With this knowledge available, the scraper box which had been replaced, and which appeared by normal examination to be satisfactory, was then returned to the manufacturers and subjected to stringent gauging of the scraper ring landings. It was in fact found that the grooves were of conical formation to the extent of 0·002 inch over the radial width of the landing, as shown in Fig. 1. The direction of the coning was such as to provide for the generation of an oil wedge during the scraping stroke.

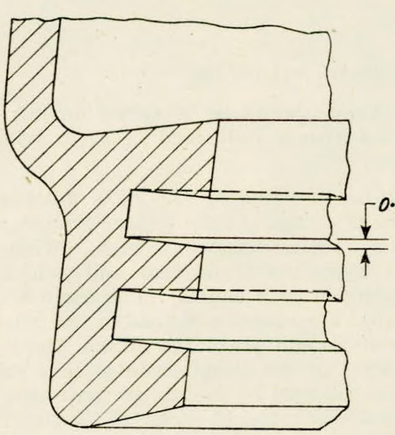

FIG. 1—*Section of scraper box grooves showing conical lands*

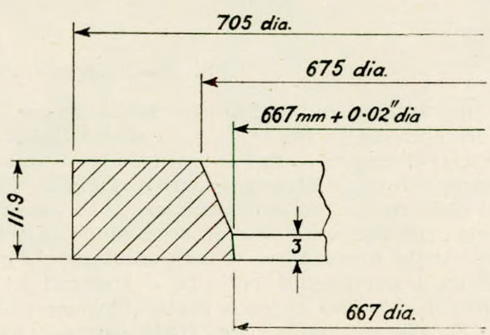

FIG. 2—*Section of special scraper ring*

Another scraper box machined at the same settings on the same machine was similarly gauged and found true.

It was concluded that the coning of the working scraper boxes had been brought about by overheating of the boxes when the original blow past of the power rings took place. The effect of such heating would be to bellmouth the scraper box, which,

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being of cast iron, would not return to its initial condition on cooling.

The condition tends to be a stable one so far as the rings are concerned because the oil wedge that is generated precludes wear on the up stroke and provides oil on the surface of the skirt for lubrication on the down stroke.

To save changing all the remaining scraper boxes for so microscopic a deformation, special scraper rings were provided of the form shown in Fig. 2. These counteracted the coned formation of the grooves and are giving satisfaction in service.

2. PROBLEMS OF A GEARED DIESEL PROPULSION SYSTEM

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The desire behind the presentation of these notes is to warn others to tread warily in the complexities of geared Diesel machinery. It is hoped that the lessons learned through the installation described will lead only to the advancement and expansion of a system of propulsion which has much to commend it where operational circumstances are suitable. The pilot vessel under review, now in service in the Middle East, has these dimensions:—

Length, b.p.	147ft. 6in.
Breadth	33ft. 0in.
Depth	16ft. 0in.

The shipbuilders' engineering department fitted the machinery on board, laid down the operational scheme, and requested the engine builders, the makers of the detuning couplings, and the gearbox manufacturers to co-operate in developing a somewhat unique installation. The subsequent difficulties were unusual and the solution to them rather protracted, but nevertheless successful.

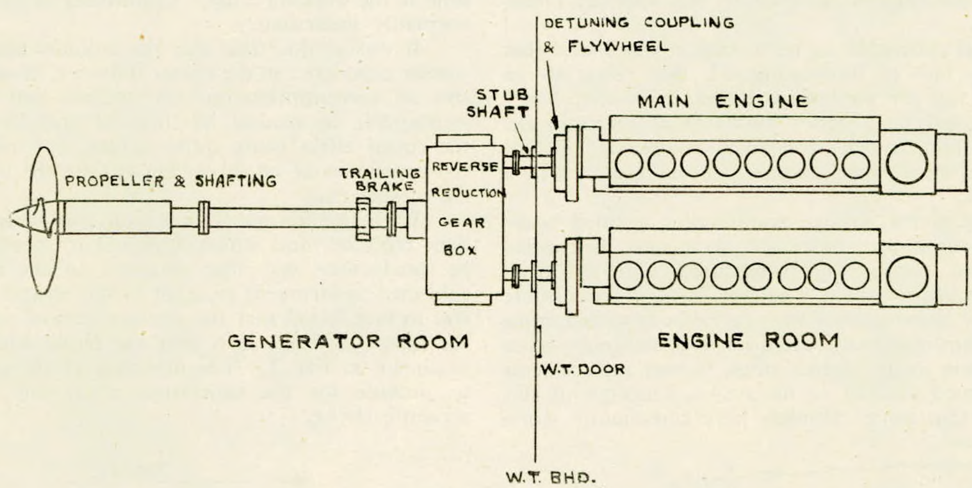

FIG. 3—*Diagrammatic arrangement of geared Diesel installation*

DESCRIPTION OF MACHINERY INSTALLATION

The arrangement in Fig. 3 shows a geared Diesel installation in which two engines, either independently or jointly, drive a single screw through a reverse-reduction gearbox. Each unidirectional main engine has eight cylinders, is of the two-stroke single-acting type and is capable of developing 1,400 b.h.p. at 300 r.p.m. At the forward end of each crankshaft, a hydraulic damper of the pumping-chamber type is arranged and at the aft end there is attached to each engine flywheel a detuning coupling. Separating engine room from generator room is a watertight bulkhead through which two stub shafts pass, each connecting an engine to the reverse-reduction gearbox, in which are fitted the oil-operated manœuvring clutches and constant mesh gearing. In the ahead gear train, the reduction ratio is 1·82:1 and the propeller speed at full power just under 165 r.p.m. The reduction ratio in the astern gearing is 2·62:1 and the single collar thrust block is incorporated at the aft end of the welded mild steel gearbox. One length of intermediate shafting connects gearbox output shaft to tailshaft, to which is fitted a four-bladed solid bronze propeller, 11 feet in diameter. At the forward end of the intermediate shaft, a hand-operated trailing brake is arranged.

Control gear is fitted for manœuvring the vessel from the navigating bridge either with one or two engines running and is coupled to the gearbox control cock. The ship may also be manœuvred from the engine room starting platform by disconnecting the bridge control gear in the engine room. Under normal service conditions, one engine meets all requirements, of which frequent manœuvring is an important feature. The second engine is thus a standby unit or is brought into service for high speeds. On one engine a ship speed in the region of eight knots was contemplated and substantially exceeded on trial. With two engines driving, the ship is very adequately powered to meet two other specified requirements, first, that a speed of 14·5 knots should be sustained on fair weather service and, second, that a full speed of 15·25 knots should be obtained on trial.

ENGINE GOVERNING AND GEAR RUMBLE

During the first trials it soon became evident that the main engine governing was unacceptably weak. With two engines running it was impracticable to obtain and maintain a uniform balance of loading in each engine. The trend was always for one engine to shed a proportion of load to the other. With one engine in operation and set at an idling speed around 130 r.p.m., engagement of the clutch resulted in a major drop in speed which was followed by momentary reduction of the gearbox oil pressure and consequent release of the clutch, whereupon the engine speed increased, to be followed by a rise in gearbox oil pressure and re-engagement of the clutch. Thus an unstable surging condition was created, primarily caused by weak governing and accentuated by the fact that the gearbox oil

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pumps were mechanically driven from the input shafts. Later trials demonstrated that these two adverse factors had been successfully eliminated by:—

(1) The fitting of a hydraulic governor of the Pickering Isochronous type to each main engine in addition to the overspeed governor, and

(2) the withdrawal of the oil pumps from the gearbox and the discarding of the mechanical drive. Each pump was arranged on a common bedplate for independent electric motor drive, one pump taking the entire load and the other being for emergency.

Facilities for testing the gearbox under load were not available at the manufacturers' works. On the engine builders' test bed the largest available brake was fitted to one input shaft, and to the other, one engine was coupled, thus a twin engine test through the gearbox was not carried out prior to installation of the equipment on the vessel. Slight gear rumble was noticed on the test bed and persisted on trial within the range 180/210 r.p.m. This feature was most noticeable in the single engine condition and, as a precaution, a bar against continuous operation within this speed range was applied.

The maximum single engine speed attained was 234 r.p.m. and at 175 r.p.m. a ship speed of 10·6 knots was recorded. For the bridge control settings it was decided to arrange three notches for manœuvring between idling speed and 175 r.p.m. and a fourth at 225 r.p.m. This permitted all normal single engine service to be conducted below the zone of gear rumble, and was believed to offer one higher speed capable of being reached and maintained without detriment to the gearbox. Additional bridge control settings were allocated for twin engine operation at engine speeds up to 300 r.p.m. An important feature of these settings was the introduction of a time delay of 10 seconds for the travel of the gearbox control cock between disengaging an ahead clutch and engaging an astern clutch, or *vice versa,* at idling speed. Of this period it was estimated that about seven seconds would be the actual pause in neutral available to diminish the shock effect on the clutches when reversing.

Torsiograph records were taken on trial by two parties and no unsatisfactory features were reported. Engine records were taken from the forward end of the crankshafts and another instrument was set up at the intermediate shaft just aft of the gearbox.

CLUTCH SLIP AND ENGINE DAMPING

An uneventful passage to the Middle East was made on one engine only. The port and starboard engines were used alternately at an engine speed of 175 r.p.m. except for two periods of about twelve hours each when the voyage log showed a maximum engine speed of 225 r.p.m. On reaching the destination and before acceptance by the owners, a full power trial of twelve hours' duration was satisfactorily completed.

After about seven months in service, clutch slip was reported in the starboard ahead clutch at 215 r.p.m. on one of the very few occasions when either engine had run up beyond 175 r.p.m. As this was the first indication of clutch slip, single engine operation was continued but restricted to a maximum of 175 r.p.m. and the vessel remained in service pending investigation. Tests were made both before and after the arrival of a representative of the gearbox manufacturers, These demonstrated that with single engine operation the starboard clutches, ahead and astern, slipped at 215 r.p.m., the port astern clutch slipped at 180 and picked up again at 215 r.p.m. and the port ahead clutch did not slip up to 230 r.p.m. No slip was experienced with twin engines driving.

The gearbox was then dismantled for examination. The starboard astern inner clutch members were found to be cracked and were replaced. The state of all other clutch members necessitated remedial attention. Inspection of the teeth of the gearing showed these to be in first rate condition.

Concurrently the main engine dampers were examined. The condition of the starboard damper was unsatisfactory in two respects. Evidence existed from which it could be deduced that seizure had probably occurred and that the damper had operated with restricted effectiveness. The port damper had suffered some scoring between the swinging faces, as in the starboard damper, but of much less severity. The oil pressure to the port damper had always been higher than that to the starboard. On the enginebuilders' instructions, adjustments were effected to these units prior to reassembly.

On completion of this work, the vessel re-entered service with an upper speed limit of 160 r.p.m. when operating on one engine only. This further restriction was introduced at the request of the gearbox manufacturers after tests demonstrated that the remedial work on the engine dampers had not effected the desired improvement and that clutch slip could be induced easily in single engine operation. The evidence now pointed to torsional vibration problems hitherto latent. Accordingly, Lloyd's Register of Shipping—with whom the vessel was not classed—were invited to investigate the installation, based on the information available, but omitting, meantime, the necessity to take further records from the ship.

A RECOMMENDATION AND THE RESULT

In due course the Engineering Research Department of Lloyd's Register issued a comprehensive report. Among the findings it was concluded that the engine dampers were not as efficient as could be reasonably expected and that efforts should be made to secure improved results with the existing design of damper before considering a change of type.

The engine builders had no hesitation in agreeing to supply two modified dampers of similar type and these were installed fully one year after the original report of clutch slip. During the intervening period the vessel remained on regular service and gave no trouble when operating within the single engine speed restriction of 160 r.p.m., and with an extension of three seconds in the time delay for the travel of the gearbox control cock to a total of thirteen seconds. The assistance of Lloyd's Register of Shipping was again solicited to take torsiograph records over the complete range of engine speeds with the new dampers in place. A further report recorded that with single engine operation, clutch slip and gear tooth separation occurred at 190 r.p.m. and were spasmodically present at greater speeds. The total torque (transmission and vibratory) at which slip and tooth separation occurred was approximately 152 per cent of the full load torque. With twin engine operation, clutch slip only was observed when the total torque was of the order of 146 per cent of full load torque. The report also included two recommendations and, based on these, the conditions governing the installation became:—

1. *Single Engine Operation*
   - (*a*) From idling speed to 180 r.p.m. without restriction.
   - (*b*) At 180 r.p.m. a ship speed of 10·9 knots was secured on trial, an adequate margin over the speed specified to be in the region of 8 knots.
   - (*c*) Above 180 r.p.m. a ban is imposed without loss of efficiency in service as the speed range is not required.
   - (*d*) Vibration stresses in the shafting are within limits which would be acceptable to Lloyd's Register of Shipping if the installation were classified by that Society.

2. *Twin Engine Operation*
   - (*a*) From idling speed to 270 r.p.m. and from 290 r.p.m. to full speed without restriction.
   - (*b*) Within the range 270/290 r.p.m. the engines should not be run continuously.
   - (*c*) The ship speed corresponding to the zone of restriction is 14·75/15·25 knots but the specified fair weather service speed, 14·5 knots, is clear of restriction, as is the full speed obtained on trial, 15·6 knots at 306 engine r.p.m.
   - (*d*) Vibration stresses in the shafting are within limits which would be acceptable to Lloyd's Register of Shipping if the installation were classified by that Society.

Lloyd's Register's observations were made with an oil pressure of 75lb. per sq. in. at the gearbox control cock, the maximum service pressure recommended by the gearbox makers. After these tests were completed the owners experimented with two higher oil pressures, 85 and 95lb. per sq. in. The results from the latter are illuminating.

	Engine	Direction	Maximum r.p.m.	Clutch condition
Single engine	Port	Ahead	220	No slip
	Port	Astern	215	Slip at 215 r.p.m.
	Starboard	Ahead	220	No slip
	Starboard	Astern	215	Slip at 215 r.p.m.
Twin engines	Both	Ahead	295	No slip
	Both	Astern	—	Not recorded

CONCLUSION

There is little doubt that the condition of the engine dampers, particularly the starboard unit, deteriorated during the early service of the vessel. Their main duty was to damp the 3-node eighth order critical vibration which peaked at 212 r.p.m. in single engine operation and at 209 r.p.m. with twin engines. Reduced damping effect imposed increased vibratory torque in the gearbox clutches. The makers stated that the drive through the gearing could be treated as solid, as the clutches would not slip unless an overload of 50 per cent were applied. The transmission torque in single engine operation up to 175 r.p.m. was well within the capacity of the clutches but it was found in practice that deterioration took place and adversely affected their capacity for overload. To this feature these four causes contributed: (*a*) the shock effect of frequent reversals when manœuvring; (*b*) the necessity for the clutches to overcome the inertia of gearing masses designed for twin engine propulsion; (*c*) the fact that the astern clutches were designed to transmit a lower torque than the ahead clutches; and (*d*) the reduced effect of the engine dampers when these were not in good condition. The insertion of hydraulic or electric couplings between engines and gearbox would, of course, materially reduce, if not altogether obviate, this clutch condition.

The detuning couplings were believed to be satisfactory in accomplishing their function within the torsional system and, provided the engines ran with reasonable regularity, there would be no fourth order or other 2-node order vibration of consequence. In the design stage the engine builders were asked to modify the firing order of the engines but this request could not be conceded. It was ascertained later that the 2-node fourth and sixth order vibrations together had a substantial effect in single and twin engine operation which could have been removed by the substitution of hydraulic or electric couplings for the detuning couplings. Unfortunately, this proved quite impossible because of space limitations.

Three propositions were explored with a view to removing the partial restriction in twin engine drive at 270/290 r.p.m., thus, (*a*) increasing the stiffness of the detuning couplings; (*b*) fitting viscous dampers of greatly increased mass; and (*c*) adjustment of the propeller. All were discarded for sound reasons and the vessel continued in service under an extended guarantee which terminated satisfactorily some time ago.

3. TEMPORARY REPAIR OF A CRACKED DIESEL ENGINE CRANKSHAFT

A. R. GATEWOOD, S.B. (Member)*

Although several methods of carrying out this type of repair were described in last year's symposium, it was felt that the description of this particular repair might be of interest since it can be carried out with materials and equipment which are now available in practically every part of the world.

During the war a loaded oil tanker fitted with a 2,800 b.h.p., 4 cylinder, opposed piston, Diesel engine, suffered a fractured crankshaft in way of the No. 4 centre pin and web.

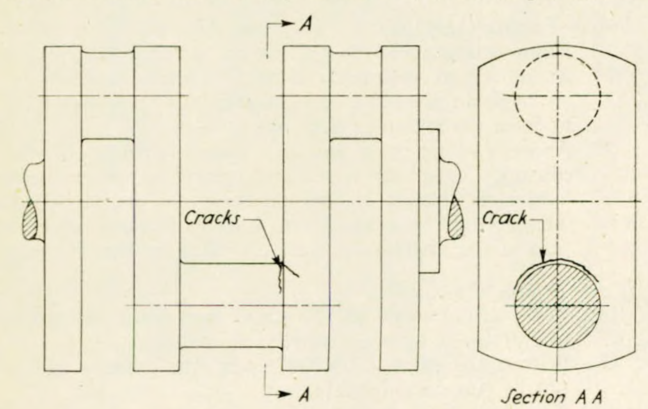

FIG. 4—*Cracked main engine crankshaft*

Due to the emergency then existing and because no spare section of crankshaft was immediately available, it was essential that temporary repairs be effected as quickly as possible so that the vessel could continue on her voyage.

The crankshaft was cracked at the forward end of the centre crank pin (Fig. 4). The fracture was approximately 18 inches long and extended around the pin for approximately one-third of its circumference. The crack also extended into the integrally forged crank web.

The first step in the repair was to vee out the crack, both in way of the crank pin and the web, to a depth of about three-quarters of an inch. The crank section was then preheated to between 200 and 300 deg. F. and the fracture welded. The welds were peened after each layer of weld metal was deposited.

The crank web, in way of the fracture, was then fitted with three-quarter inch doublers on both sides and ends (Fig. 5, Plate 1). These were welded to the web and the welds were peened as the welding progressed.

In the meantime two girders were being fabricated which were to be fitted on each side across all four crank webs. As can be seen (Fig. 6, Plate 1), these girders can be built up out of practically any type of material which is available. In this case each girder consisted of a 32 inches × 60 inches plate, 1 inch thick, with the toes of three "T" sections welded to the plate. The tees were half an 18-inch "I" beam section, 70lb. × 8¾ inches flange. These two prefabricated girders were stress relieved prior to installation.

The crank section was again preheated and the two girders welded to each side across the four webs (Fig. 7, Plate 1). All welding on the centre webs was completed prior to welding to the end webs. The 1-in. girder plates adjacent to the webs were slotted longitudinally along the toe of the "T" sections so that a substantial plug weld connexion could be made to each web.

These cannot be seen in the illustrations but were made by welding through the space between the flanges of the "T" sections. Suitable end and tripping brackets were then welded

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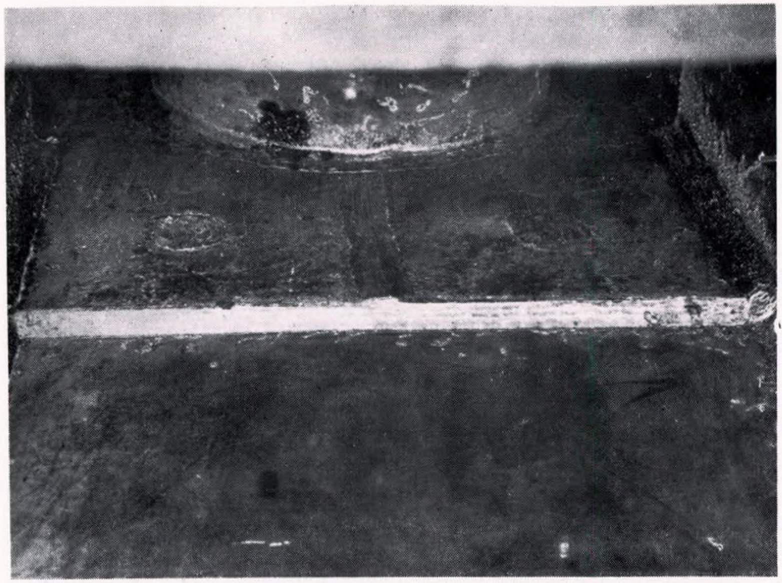

FIG. 5—*Doubler plates fitted*

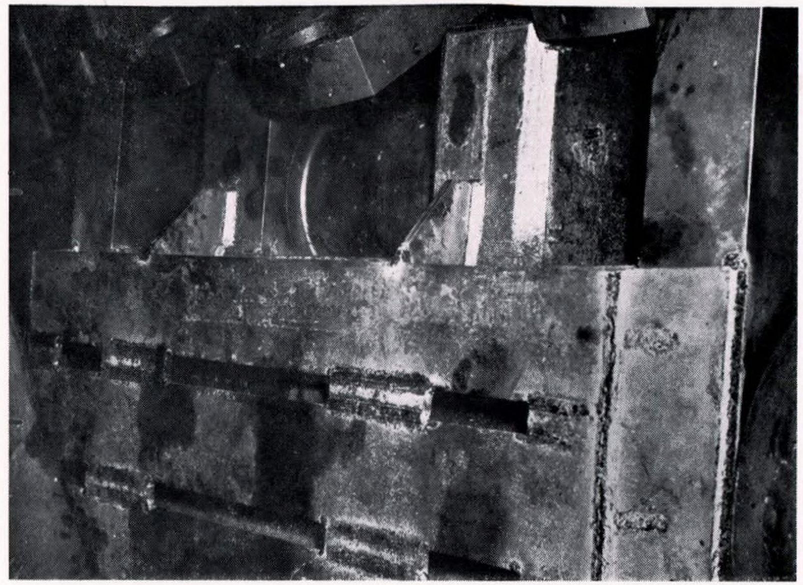

FIG. 6—*Prefabricated girders fitted across crankwebs*

FIG. 7—*Prefabricated girders fitted across crankwebs*

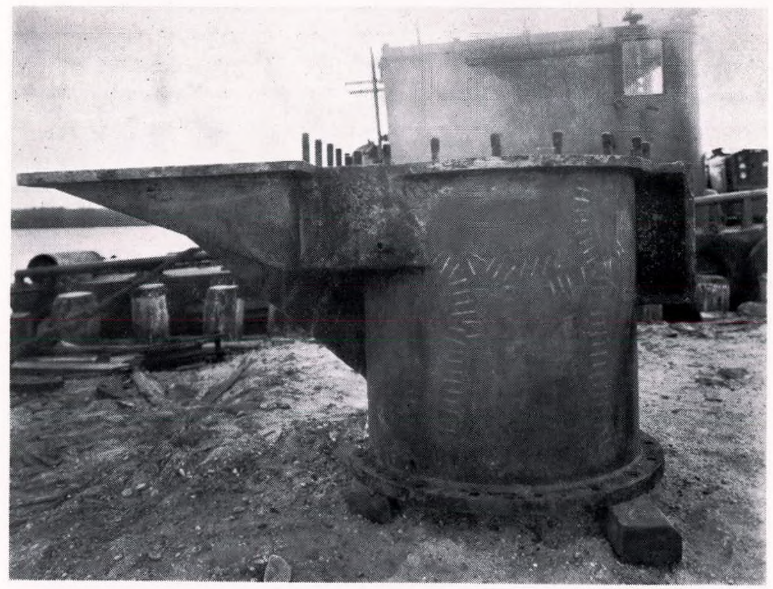

FIG. 12—*Metallock repaired air pump cylinder, removed from ship after seven years*

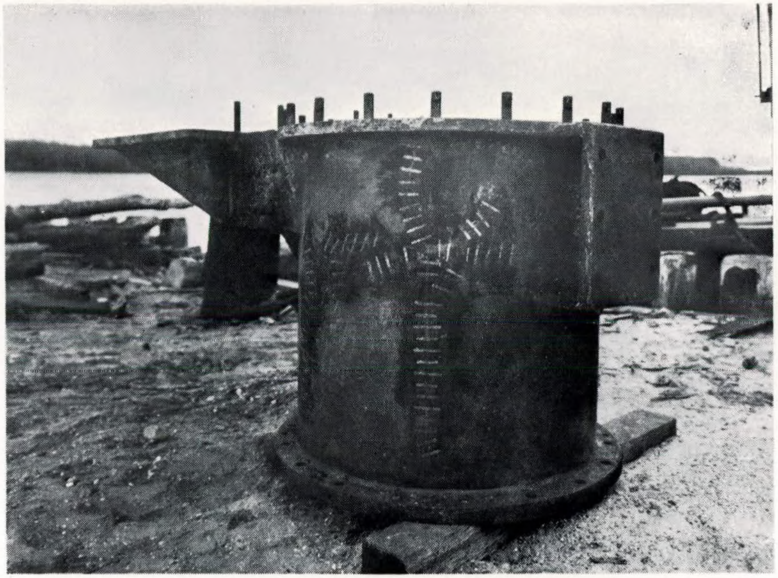

FIG. 13—*Metallock repaired air pump cylinder removed from ship after seven years*

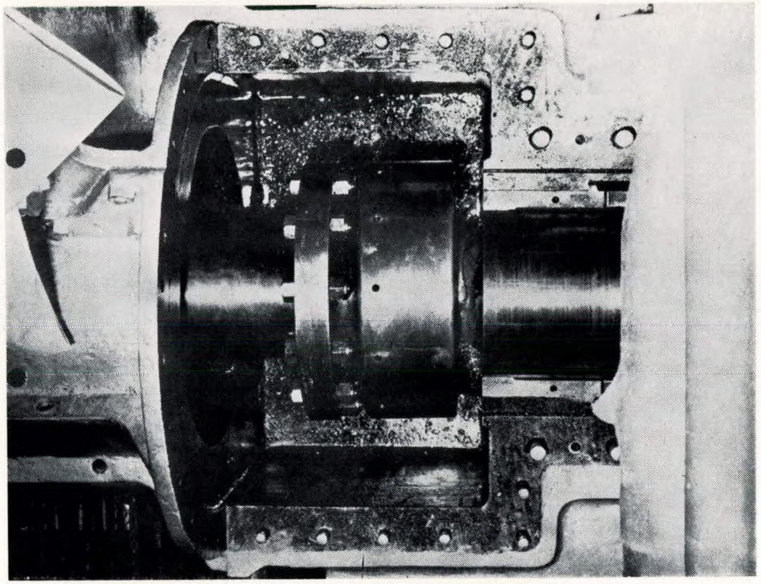

FIG. 14—*Fractured low pressure turbine rotor shaft*

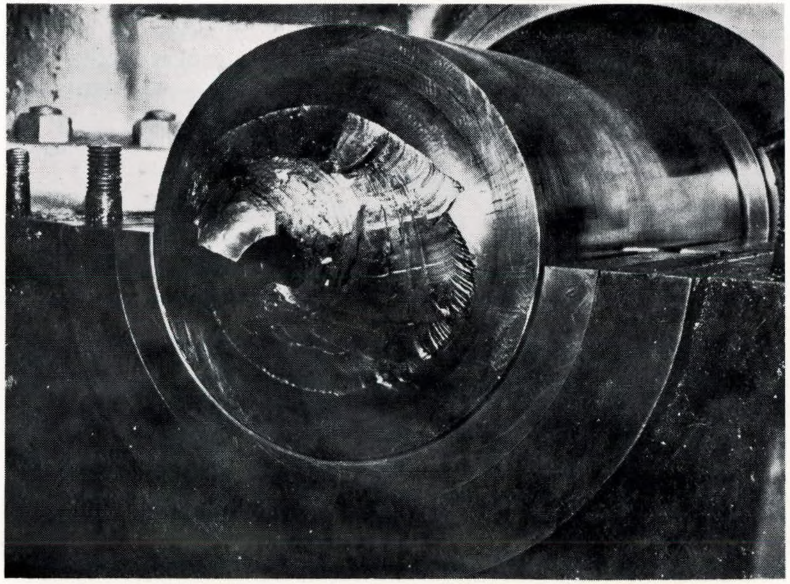

FIG. 15—*Fracture on turbine end*

FIG. 16—*Fracture on coupling end*

to the webs. The centre and side connecting rods, of course, were swung to one side and secured with clamps.

The ship was held up for two weeks while the repairs were being made and then continued in service without any further troubles for three months before a new section of shafting was installed.

4. REPAIRS TO THE FRACTURED CAST IRON CYLINDER OF A BEAM AIR PUMP

A. R. GATEWOOD, S.B. (Member)

The vessel in this case was a Great Lakes bulk carrier built in 1901 and powered with a 1,480 i.h.p. triple expansion steam reciprocating engine having a beam driven air pump (Fig. 8).

As is well known, vessels on the Great Lakes operate only during a part of each year, so it is quite serious to have a ship tied up during the operating season.

In this case a valve guard stud had broken off and cracked the air pump cylinder wall in several places and a triangular-shaped section, about $3\frac{1}{2}$ inches × 5 inches, was broken out completely. Because of the age of the engine, no replacement cylinder was available and inquiry disclosed that a new casting could not be obtained until after the close of the season so it

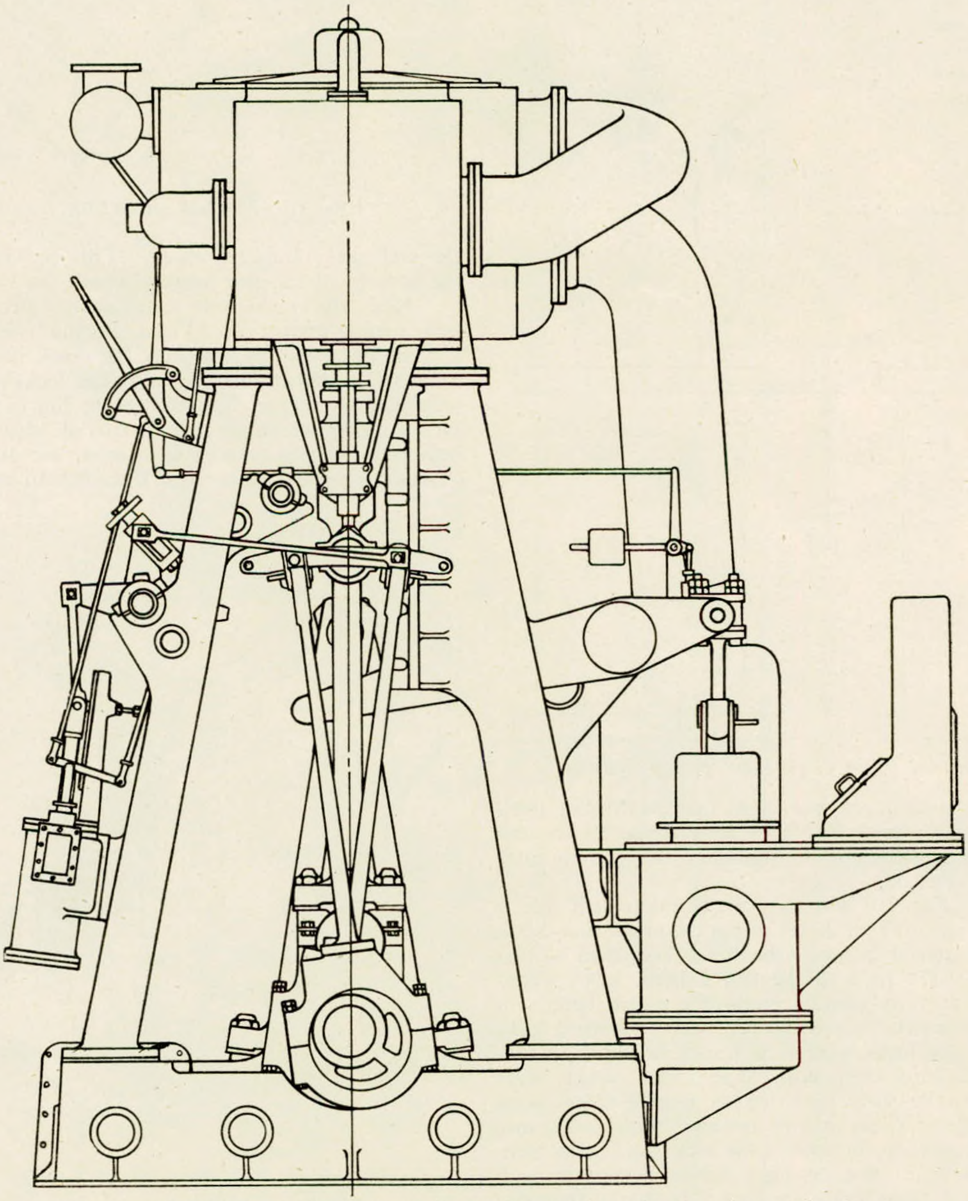

FIG. 8—*Arrangement of steam reciprocating engine showing beam air pump*

was decided to attempt to carry out a repair by the use of a method known as "Metalock".

During the war, considerable experience had been obtained in repairing iron castings by this method and undoubtedly many are familiar with the process. For the benefit of those who are not, it might be said that if done properly it is a relatively easy and efficient means for repairing such cast iron parts as cracked Diesel engine cylinder heads, cylinder liners, bearing pedestals, bedplates, etc. The description of this repair, however, has been chosen because up to that time the American Bureau had had no experience of attempting actually to put the parts back together again and have the pump functioning satisfactorily.

On the starboard after side of the pump, in addition to the triangular piece "A" which was broken out completely, there was a total of 49 inches of crack extending from within 1 inch of the top flange to within 4 inches of the bottom flange (Fig. 9). On the other side of the cylinder there was an additional 23 inches of crack. It was also found that the cylinder was sprung out of shape.

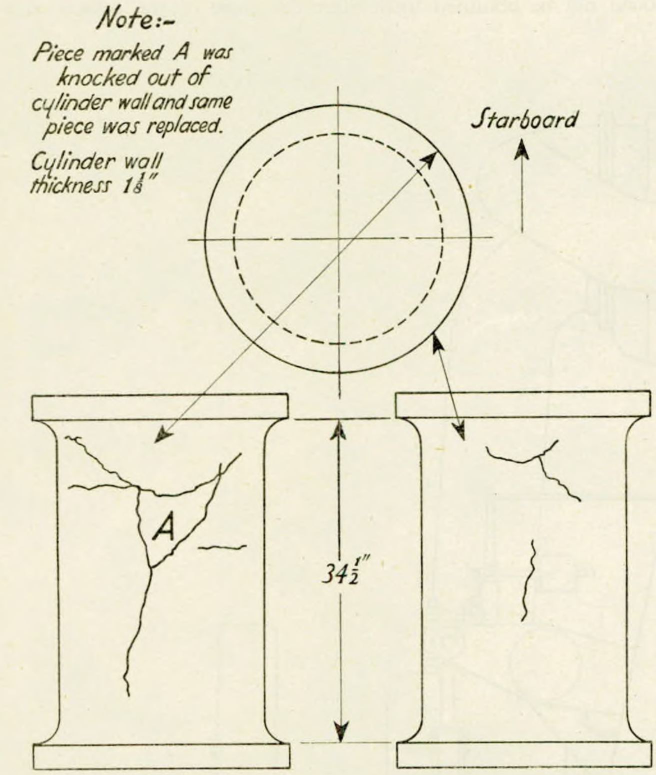

FIG. 9—*Fractured main engine air pump cylinder*

First, the triangular piece was fitted into its former position and the cylinder was pulled back into shape by the use of strong-backs. The strong-backs remained in position until the repair was completed.

The next step (Fig. 10) was to drill, through drill jigs, a series of accurately spaced ¼-in. holes across the crack to a depth of ¾ inch. The material between the holes was then slotted out to a width of 0·175 inch to the full ¾ inch depth. Into these slots, placed at 1-in. centres along the entire length of the crack, were then inserted successive layers of preformed locks made of Invar. These locks were each ¼ inch in depth so that three were inserted into each slot. The locks, which were slightly larger than the slots prepared to receive them, were individually applied by peening into position with pneumatic tools in order to mushroom or spread the locks and force them against the casting wall, thus giving a bearing point at each of the knob-like projections along the lock. It should be noted that the slots were only prepared to a depth of ¾ inch, although

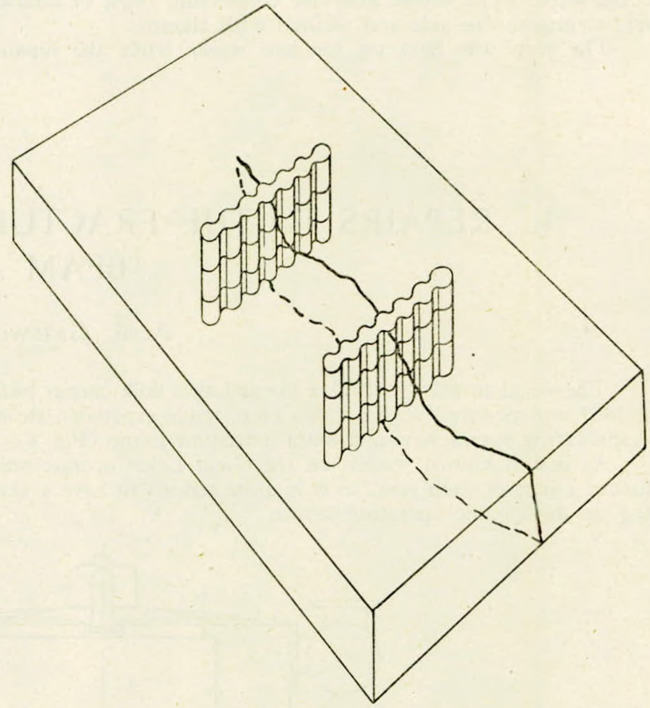

FIG. 10—*Method of fitting transverse locks*

the wall was 1⅛ inches thick. This provided parent metal at the bottom of the slot against which the locks could be upset.

Next, the cracks were sealed against pressure by filling with new metal, known as "Metal Lacing" (Fig. 11). This was accomplished by drilling into the crack itself a series of holes tangent to one another between the locks. The first of these were slightly etched into the centre lug of the locks and then tapped, after which threaded material, also Invar, was screwed into each hole. Successive holes were then drilled and tapped, all overlapping each other, and threaded Invar inserted until the

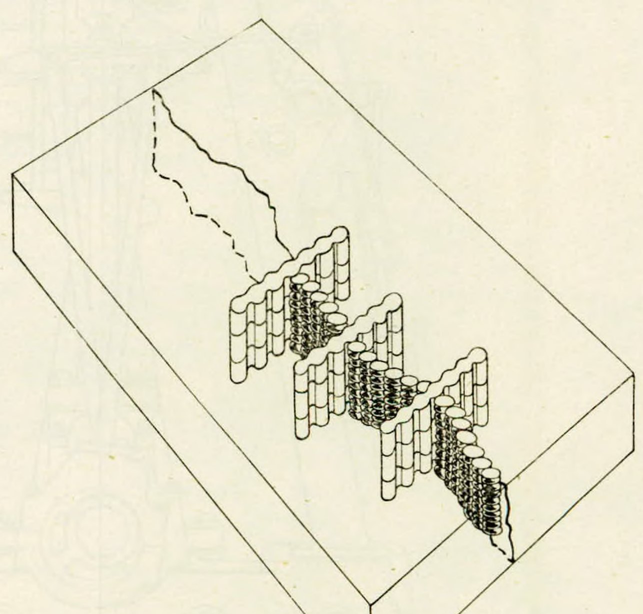

FIG. 11—*Method of fitting "metal lacing"*

entire crack was filled. The lacing, which was the same depth as the locks, was then cold worked by peening in the same manner as the locks.

The purpose served by the lacing operation was first to pre-stress the locks in order to prevent any reversal of stress. Second, to seal against pressure, either outwardly or inwardly. And third, the threaded walls of the drilled holes and the thread on the inserted material were intended to prevent movement of the casting in any direction.

In this case the locks and lacing were applied only from the outside of the cylinder. The cylinder walls had been scored, so the interior of the cylinder was metal sprayed to fill the scores and cover the cracks.

The entire repair (Figs. 12, Plate 1 and Fig. 13, Plate 2) was carried out over a weekend and the vessel returned to service. The repair was examined from time to time and after six years it was still in good condition and the air pump was giving satisfactory service.

5. TEMPORARY REPAIR OF A FRACTURED LOW PRESSURE TURBINE ROTOR

A. R. GATEWOOD, S.B. (Member)

This vessel is a twin screw, transatlantic passenger liner built in 1939, which saw very arduous duties as a troop transport during the war. It was stated that when the vessel was leaving the dock at Southampton, on opening steam to the starboard unit the turbines would not operate. Examination revealed that the stub portion of the low pressure turbine rotor shaft aft of the after bearing, to which the male claw of the flexible coupling is secured, had completely fractured in such a manner as to cause a jamming between the rotor thrust and the after end of the flexible coupling (Figs. 14, 15 and 16, Plate 2).

Several methods of procedure were considered but since they involved either a long delay in returning the ship to service or a considerable reduction in power and speed, it was finally decided to attempt a temporary repair (Fig. 17).

The remainder of the stub end was removed by facing off to the machined face outside of the original stub piece. A new stub piece was forged with a 1-in. thick flange. The flange was recessed so as to fit over the end of the rotor journal for the purpose of alignment. This, of course, necessitated machining back the end of the bearing shells about $\frac{3}{4}$ inch to allow the piece to fit over the journal.

When the stub piece was set up in place, four set bolts were fitted and tapped into the end of the rotor and four dowels were fitted to permit the stub piece to be fitted again later. The stub piece was then taken off and the flanges and recessed hub turned, so that the flange would be the same diameter as the journal.

The stub piece was then replaced on the end of the rotor and secured with the dowels and set bolts. Twelve holes were drilled through the flange and into the end of the journal for $2\frac{1}{2}$ inches. These holes were tapped with a $\frac{7}{8}$ inch gas tap and hollow steel ferrules were screwed tightly into both the stub piece and the end of the rotor. Three-quarter inch diameter steel dowels were then driven home into each ferrule and the ferrules riveted over.

As the final step, a high tensile steel ring $1\frac{5}{8}$ inches $\times$ 1 inch was shrunk over the flange and the end of the journal (Fig. 18).

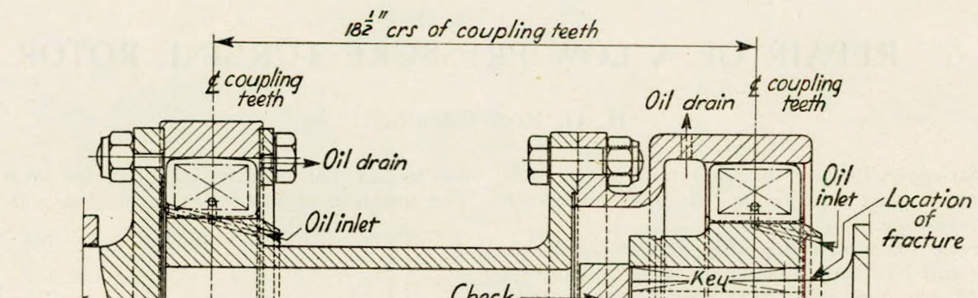

FIG. 17—*Original arrangement of flexible coupling to low pressure turbine*

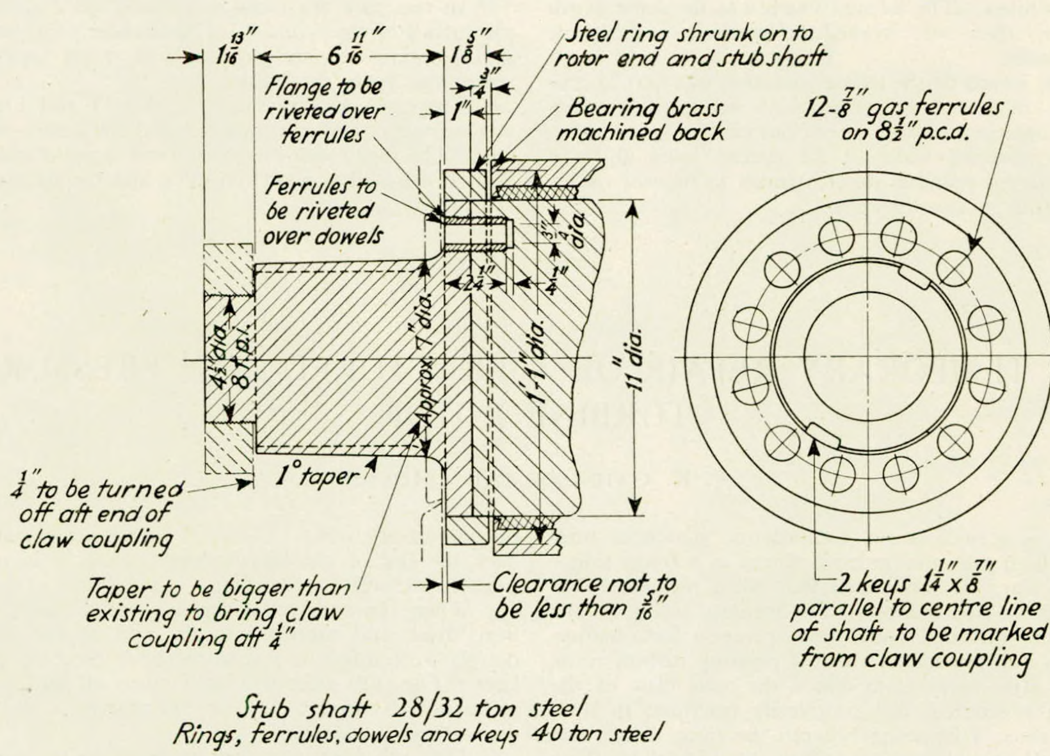

FIG. 18—*Temporary repair to low pressure turbine rotor*

Upon completion, a surface gauge check showed that the stub piece was out less than 1½ mils (0·0015 inch).

Thorough dock and sea trials were then carried out, during which there was considerable manœuvring and running astern. Examination of the coupling and stub piece through a portable plate, which the ship's staff had fitted in the casing, showed that the repair had proved quite satisfactory and the vessel subsequently made five Atlantic crossings before permanent repairs were carried out.

6. REPAIR OF A LOW PRESSURE TURBINE ROTOR

H. G. ROSS (Member)*

Twin screw passenger steamer of 5,000 tons gross, built in 1935; fitted with Parsons type single reduction turbines—three turbines per set—total s.h.p. 6,000.

The usual service on which this ship was engaged was between Melbourne and Northern Tasmanian ports, but during the 1939-45 war she was used as a troop transport in tropical waters. In normal service the ship steams at high speeds but, as a result of war conditions, there were long periods of low speeds in convoy.

During a periodical survey of the engines, it was found that the l.p. rotor of the port engine was very badly eroded at the exhaust end and the erosion diminished towards the inlet end.

A contributing factor was thought to be the prolonged steaming at low speeds during certain periods of the war. The erosion was so extensive that the roots of the blades were almost exposed on the exhaust side of the blades and the metal in the rotor drum was so affected at the after end that pieces of it broke off when subjected to very light blows.

In order that the ship should steam efficiently again, it was essential that the blades be fitted properly but the erosion was so bad that a normal repair of the rotor was impracticable. The question of a new rotor forging was considered but, as

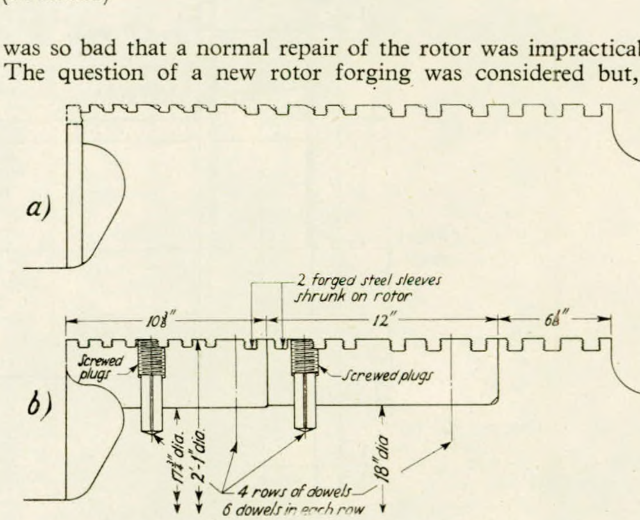

FIG. 19—*(a) Section through l.p. rotor, showing erosion of grooves; original outline shown by dotted lines. (b) Section through l.p. rotor, showing repairs effected*

* Works Manager, Cockatoo Docks and Engineering Co. Pty., Ltd., Sydney, N.S.W., Australia.

this would have involved some months' delay, a method of repair was investigated and, on being approved by Lloyd's Register of Shipping, was carried out.

This repair involved turning down the existing rotor drum to solid metal to take two forged steel sleeves which were shrunk on to the rotor and secured by four rows of dowels. The sleeves were then turned externally to the size of the rotor, the grooves were recut and complete reblading carried out. Details of the repair and reblading, which took about four weeks to complete, are shown in Fig. 19.

The ship has continued in service for some time with this repair and did not lose any operating time, as the work was carried out during an ordinary survey period.

A further order has been received to carry out the same repair to the starboard l.p. turbine of the vessel, indicating that the method has proved satisfactory to all concerned.

7. THERMAL STRAIGHTENING OF TURBINE ROTORS

H. G. Yates, M.A.*

Various methods of straightening a bent rotor have been used from time to time, including the method of peening and the direct heavy-handed method of the hydraulic press. The former is slow and very limited in its action, while the latter runs the risk of producing a correcting bend in the wrong place as well as requiring heavy equipment. The thermal method has come to stay. As it acts by applying exactly the same process as that which originally introduced the bend, it will be well to consider first how the latter is caused.

Consider a turbine rotor running at speed in its casing, and imagine that one of the glands is brought into contact with the rotor by some distortion process. The rotor is almost certain to be initially out of truth by a few tenths of a mil. As soon as contact occurs, heat is developed by the high-speed rubbing. If the amount of interference between shaft and gland is less than the local eccentricity, the rotor will be heated at one side only. If the rub is heavy, it will at least be heavier at one side than at the other—a fraction of a mil is sufficient to make one side hotter than the other.

Immediately, the metal in the rubbing zone begins to expand in all directions. The tangential expansion has little effect; the radial expansion slightly increases the severity of the rub and the disparity between its effect at high and low sides; but the axial expansion is much more serious. Consider a short length of shaft in the region of the rub, defined by two transverse planes, say, one inch apart axially. The metal fibres at the heated (or most heated) part of the periphery, initially an inch long, becomes slightly longer and force the short cylinder to take on a wedge shape. The expansion is, of course, at first confined to the metal very near the surface, and consequently points on the defining transverse planes cease to lie in the same plane and stresses are set up in an axial direction, some tensile and some compressive, with an average value of zero and a zero resultant bending moment. Hence, the wedge angle is less than that calculated by dividing the increase in length of the outer fibre by the local shaft diameter, but some local change in straightness of the shaft axis is produced.

If the shaft were free to bow, the part in the region of the rub would move laterally by an amount dependent on its distance from the two bearings, the effect being greatest if the rub occurs at mid-span. Furthermore, it is important to note that the side of the rotor which was high initially tries to move out still further, since it is here that the rubbing commenced. The shaft, however, is *not* free to bow since it is in contact with the gland; hence, the rub increases in severity. The heating becomes greater, and the bowing forces greater again, in a vicious circle.

By this time the gland is carrying heavy radial loads as the bowed rotor whirls round within it, partly due to centrifugal forces on the rotor mass and partly to the forces produced by the bowing action and restrained by the gland. The latter are sometimes sufficient to bring about contact all round the gland even if it were initially a lateral movement of the latter which caused the first contact. Naturally, the process is accompanied by heavy vibration and noise. The bow may well reach a stage at which it would amount to a sixteenth of an inch or more if the rotor were free, but it is kept to 20 or 30 mils by the restraining action of the gland.

This restraining action adds to the heavy compressive *stresses* in an axial direction, which act on the heated metal near the crown of the bow. This metal is already softened by the local heat, and consequently yields. The yielding action tends to prevent the bow from increasing, but at the same time metal slightly further in is becoming heated by conduction and in its turn contributes to the bowing; it is not known whether the process would reach a natural limit, since the machinery is usually shut down within a few seconds after the rub commences, on account of the heavy vibration.

After shutting down, the shaft begins to cool as the local heat flows away by conduction to the body of the shaft and to the gland with which it is still in contact. The bow gradually lessens. However, when the shaft has equalized in temperature it will be found bowed *in the reverse direction.* This is due to the permanent contraction produced by plastic yielding in the hot metal which formed the crown of the bow while rubbing continued; this side of the shaft is slightly shorter than the other and so the shaft takes on a reverse bow. Hence the apparent paradox that the shaft in its final state is bowed but shows score-marks diametrically *opposite* to the high side of the bow.

The thermal straightening process acts by subjecting the metal *near the crown of the final bow* to powerful and rapid heating. In the first instance, this will make the bow worse; but if the heating is sufficiently rapid, the metal in the heated zone will expand until it becomes plastically crushed, owing to the high radial temperature gradient. The cool body of the shaft provides the resisting moment to counter that due to the local expansion, and the reduced strength of the hot region facilitates the crushing action. On subsequent cooling, the bow will be found to have diminished if the heating has been sufficient.

Before undertaking the straightening process, the shaft must be carefully "clocked" for out-of-truth at a sufficient number of axial stations to establish the position of the bend with certainty; generally at least six stations, including two in the heavily scored region, are desirable. This is most conveniently done in a lathe, although it may be done with the shaft resting in its own bearings if necessary. It is wise to take axial clock readings also at one or two vital positions, such as the face of the rotor drum or disc nearest the bend and the face of the thrust collar; in doing this, take care that the shaft does not move bodily in an axial direction, as it would do if located by a thrust collar which was itself out of truth. The clock readings must now be plotted on a sectional

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drawing of the rotor to a suitable scale, as shown typically in Fig. 20. It will be found in nine cases out of ten that the bend is near the neck of the main gland, and is concentrated into a few inches' length of shaft; the bend is usually in one plane, as shown by the fact that all the largest readings occur at one angular setting of the rotor, though small departures from this may arise near the points of support due to some initial eccentricity. Check that the axial readings are compatible with the bow as deduced from the radial readings, and take care not to get confused by mis-reading of a 0-50-0 micrometer scale (on one occasion in the author's experience this nearly happened).

Medium carbon steels normally used for steam turbine rotors, with or without a small molybdenum content, are unlikely to give trouble. Two austenitic steels of types often used in gas turbines have also been successfully dealt with.

there be signs of burning. The flame should be *slightly* "weaved" over an area about 1 to $1\frac{1}{2}$ inches in diameter, but *not* over a large area—speed of heating a small patch is the important factor. If the shaft carries gland collars there will be a tendency to soften or burn at the sharp corners, but a small amount of this is generally unavoidable. Attempts to protect adjacent areas by shielding with asbestos sheet wrapped round the shaft have not been successful, and free exposure of these areas to the atmosphere is better as it permits cooling. If possible do not allow the flame cone to approach the adjacent disc or rotor body nearer than about one inch.

One's first attempt at the heating process usually fails through excessive caution. It is essential to approach the job with firmness and decision, aiming at raising a two-inch diameter area to a full cherry red in about 20 to 30 seconds. Prolonging the heating beyond 40 seconds after the first sign

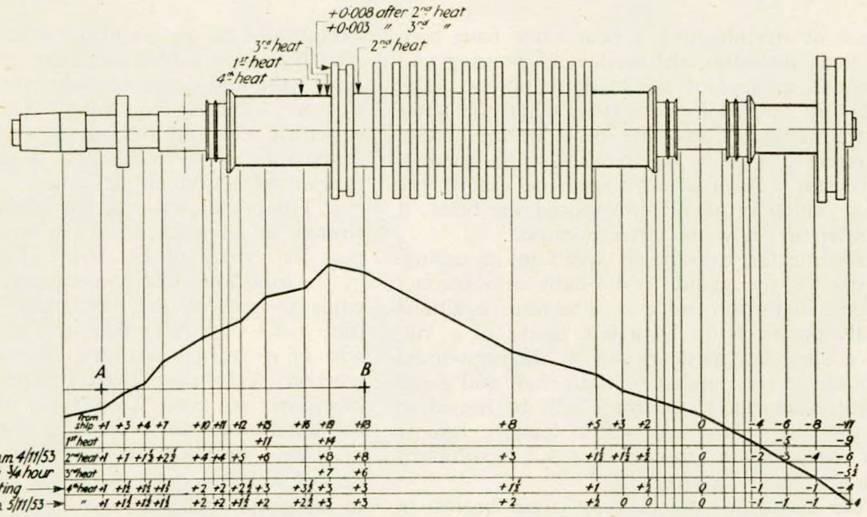


FIG. 20—*Graph of clock gauge readings taken in lathe. After fourth heat, high spot at A was 90 degrees from original plane of bend. Between A and B the plane of maximum bend gradually returned to the original position*

Our experience with alloy steels of the nickel-chrome-molybdenum type has not yet been sufficient to indicate whether the straightening process should receive special care (e.g. controlled cooling) in order to avoid cracking, but such steels are rarely used in turbines for merchant marine propulsion and never in British practice to the author's knowledge.

Carefully clean the area to be heated with fine emery paper, in particular removing any trace of copper which might cause embrittlement by intergranular penetration. If the shaft carries gland fins of yellow metal they must be removed from the zone to be heated. Turn the shaft until the crown of the bend is uppermost, and arrange a micrometer clock in a convenient position. If the rotor is being dealt with in a lathe, the clock is best placed underneath at the hollow of the bend, but, if in its own bearings, locate the clock above the rotor some nine inches away axially from the zone to be heated. If in a lathe, slack back the tailstock centre slightly to permit expansion, supporting the rotor near that end on a steady with the top pad lifted clear.

For heating, an oxygen-propane torch has been found most suitable (flame temperature lower than oxy-acetylene and less risk of carburizing), fitted with a multi-jet heating nozzle which will give a hot zone of about one inch in diameter. The gas pressures should not be so high as to produce a widely-diverging flame and should be adjusted to give a flame length approximately equal to the diameter of shaft to be heated. The burner valves should be adjusted to give a neutral flame. The end of the nozzle should be held about an inch from the surface of the shaft but this distance must be increased should

of red heat is rarely worth while. An assistant can usefully call out times and clock readings, though the actual rise of the shaft on heating is not much guide—the *rate* of rise appears to be more important.

Experience has shown conclusively that if two heats are applied to the same spot, little or no advantage is gained by the second. It is better to move two or three inches either axially or peripherally; in fact, if the bend is very heavy it will pay to begin a little to one side (laterally) of the high spot, and apply the next heat symmetrically on the other side.

The rotor must be allowed to cool to within nearly room temperature and sensibly uniform all round, before determining the gain obtained. A jet of cool dry air may be used after the metal has blackened in order to save time. Even by accelerating the rate of cooling in this way it will be at least half an hour before a check can be made and a further heat applied, and no more than about four heats should be attempted in as many hours. When the bend appears to have been reduced by about 80 per cent, allow to cool for six or eight hours and clock carefully at the axial stations originally employed. If in a lathe, rotate the shaft after each heat to check whether the micrometer has shifted. After heating near a disc of a gashed rotor, ascertain whether the rim of the disc has been pulled axially in the axial plane containing the heated area; a local "wriggle" of 5 or 10 mils sometimes occurs, and should be corrected by a heat near the disc fillet on the opposite side. When the rotor gets near straightness, attention should be concentrated on the clock readings at the journals and the face of the thrust collar. If the latter shows a variation of more than 1 mil, or either journal shows a similar difference between its two ends, then these parts must be trued up subsequently by machining—in an emergency twice these figures may be permitted, or even three times if the turbine is restricted to three-quarter speed in this condition. It is, of course, desirable to balance the rotor dynamically but in a real emergency the author would confidently risk reassembly of the turbine if all clock readings are below 3 mils (in relation to line of journals), assuming no previous history of eccentricity. These figures relate to a turbine designed speed of about 4,000 r.p.m., and should be smaller for higher speeds and *vice versa.*

It is sometimes recommended that the rotor should be subjected to a thermal stability test after straightening; in this it is heated in a gas or electric heating box to a temperature 50 deg. F. higher than the working temperature, meanwhile rotating and checking truth by micrometers acting through pushrods. When circumstances permit this to be done conveniently, it does no doubt provide a useful measure of reassurance, but most of the rotor-straightening operations with which the author and his colleagues have been associated have had no such test carried out, and in no case has any trouble subsequently arisen. Finally, examine the rotor carefully for cracks near the regions heated, and if in doubt polish the surface and apply magnetic or other crack-detecting gear. If it is considered necessary to do any subsequent machining on the parts which have been locally heated, it is likely that some relaxation of stress will occur, causing a further slight bend, and the rotor must be checked for truth again. Needless to say, the thermal straightening process is not limited to turbine rotors, and it has been successfully applied to line shafting.

8. HIGH PRESSURE TURBINE CYLINDER CASTING REPAIR

W. H. FALCONER*

Hair cracks were observed during a routine inspection of the turbine; they were pop marked and the turbine was returned to service while preparations were made for its comp'ete removal for repair by arc welding. The cracks did not extend during the voyage. Fig. 23 (Plate 3) illustrates a typical example of a crack. At the end of the voyage the complete turbine was 0·026, manganese 0·54, molybdenum 0·55, having a tensile strength of 32 tons per sq. in. Fig. 21 outlines the original casting defects rectified and reported by the foundry during manufacture. However, Fig. 22 shows the larger defects observed before the repair. The discrepancy between the reported and the actual defects should be noted. During the

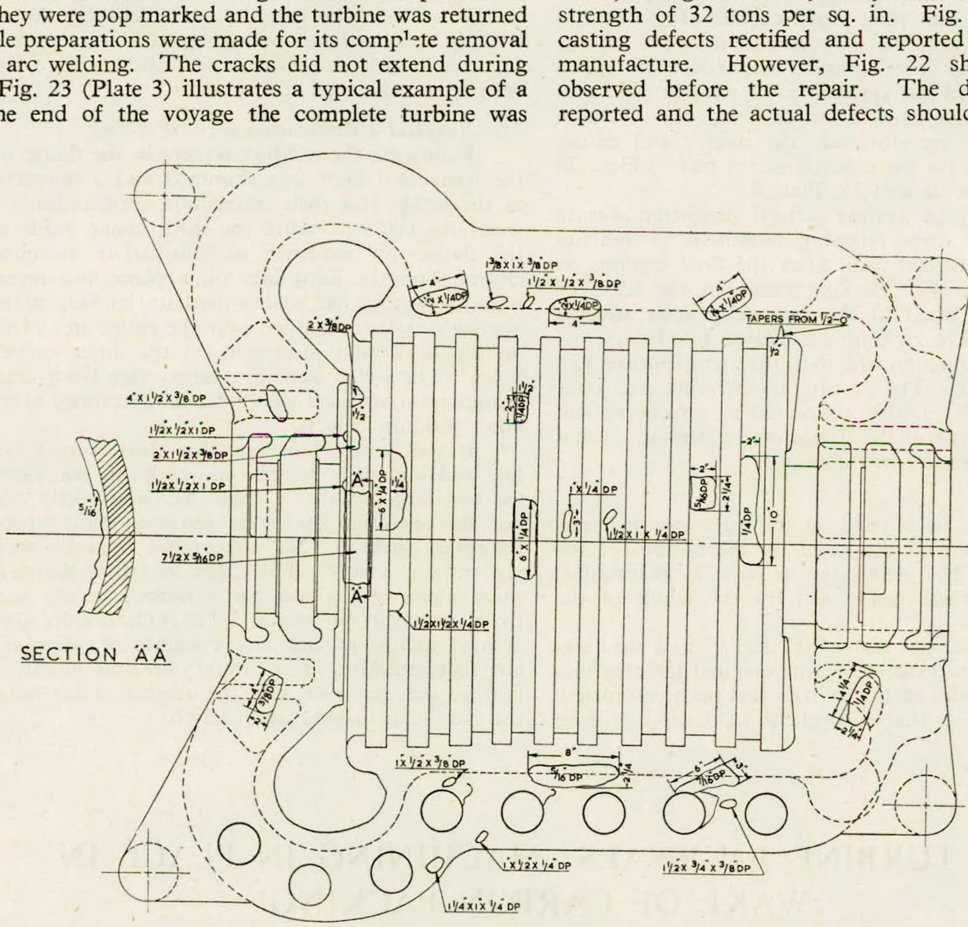


FIG. 21—*Sketch showing defects reported by founders*

removed to the workshop and the vessel sailed on schedule on her i.p. and l.p. turbines only.

The cylinder material is to the Admiralty Specification for molybdenum steel castings and has the following chemical analysis: carbon 0·18, silicon 0·35, sulphur 0·021, phosphorus

* Assistant Superintendent Engineer, Alfred Holt and Company.

general examination of the casting, prior to the repair, sand inclusions were found in the exhaust belt, as shown in Fig. 31 (Plate 4). It should be noted that the casting had not been pickled during manufacture but this is further evidence of the need of such treatment.

It should be noted that the more extensive cracks on the

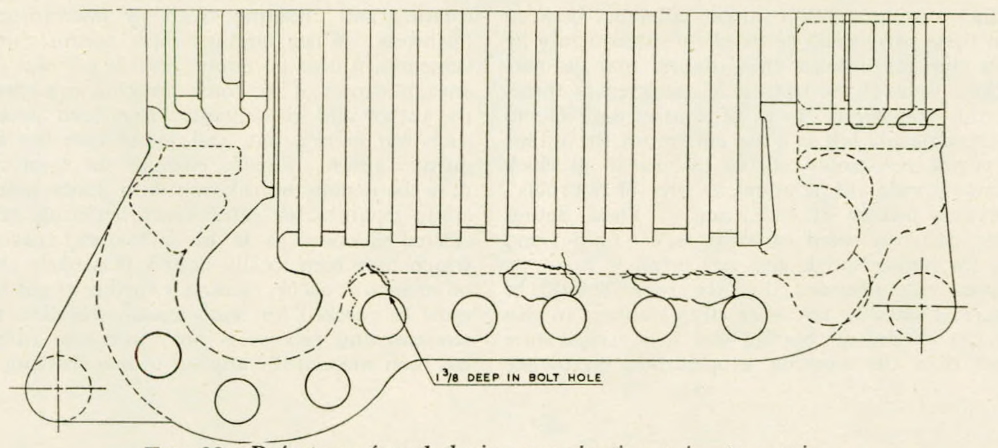


FIG. 22—*Defects as found during examination prior to repair*

joint were the result of, and ran along, the boundary of previous welding operations and their removal has resulted in three large cavities, as shown in Fig. 24 (Plate 3), between the bolt holes and the bore of the cylinder. During the machining out, one or two blow holes were exposed at a depth of 1½ inches from the surface of the joint. They showed clearly that slag inclusions were the cause of the cavities. A small exploratory cavity (Fig. 26, Plate 3), and three small cavities (Figs. 24 and 25, Plate 3), due to the removal of contraction cracks on the edge of the joint face and bore, complete the record of repairs required on this face. All the external radii were ground and polished, thus revealing five contraction cavities, four of which required attention, viz., one alongside the inlet gland casing and three on the surface on the exhaust steam belt. (Figs. 26 and 27, Plate 3, and Figs. 28 and 29, Plate 4.)

The casting was braced against general distortion during welding and subsequent stress relieving treatment by bracing bars which were not disturbed until after the final cooling, as shown in Fig. 30 (Plate 4). Backing strips in the form of ¼-inch steel tubes were fitted to the four bolt holes affected by the major cavities. The casting was heated to 450 deg. C. in a furnace and soaked to ensure that this temperature had been attained throughout. The casting temperature was controlled by a thermocouple suitably connected to a recorder and inserted into a suitable hole in the flange of the casting so that the actual casting temperature was obtained.

Welding

"Tensivec" electrodes were used on the main cavities until a complete covering had been deposited on the bottom of the cavity in each case. "E.H." electrodes, as large as practicable, were used to complete these welds and for the whole of the minor welds.

All welds were carried out above 300 deg. C. and the casting reheated where necessary to regain the original temperature. Where the bottom and sides of the cavities had been completely covered with weld metal so that subsequent weld deposits were put down on a foundation of weld metal only, and where there was no undue constraint on the weld, some relaxation on the minimum welding temperature was remitted and welding continued to 200 deg. C. Each weld deposit was carefully descaled and the slag removed from the cavities. Each successive layer of weld metal was, after deslagging, peened whilst cooling to spread the deposit and reduce thermal stresses. On completion of the welding, the cylinder was recharged for stress-relieving without intermediate cooling for six hours at 600-650 deg. C., followed by furnace cooling to 100 deg. C. Originally, it had been finally annealed at 950 deg. C.

Metallurgical Examination after Welding

Following the welding repairs in the flange of the cylinder, the horizontal joint was skimmed and a magnetic test applied to the welds and their immediate surroundings. Radiographs were also taken covering the three major welds on one side of the flange for evidence of sub-surface unsoundness. After cleaning up the joint face on a plane and redrilling the bolt holes to remove the bushes used for backing plates, a thorough magnetic test was applied over the entire area of the welds, both on the horizontal joint and on the inner surface of the bolt holes. The welds were completely free from cracks and visual examination revealed no evidence of trapped slag, gas holes or other welding defects.

It was noted, however, that there were a number of very fine and obviously superficial cracks in the joint face between the two largest welds. These had apparently existed throughout the operation and were rendered more visible against the improved surface. No importance need be attached to these very minor defects. The entire length of flange containing the three major welds was radiographed, using gamma-radiation from a Cobalt 60 source. The radiographs show no sharply defined inclusions nor any evidence of internal cracking and are representative of a sound welding repair. It might be pointed out, however, that the original welds were X-rayed and the same conclusions were drawn.

9. TURBINE JOURNALS: MACHINING IN PLACE IN WAKE OF CARBON PACKING*

E. BYRNE, B.E.M.

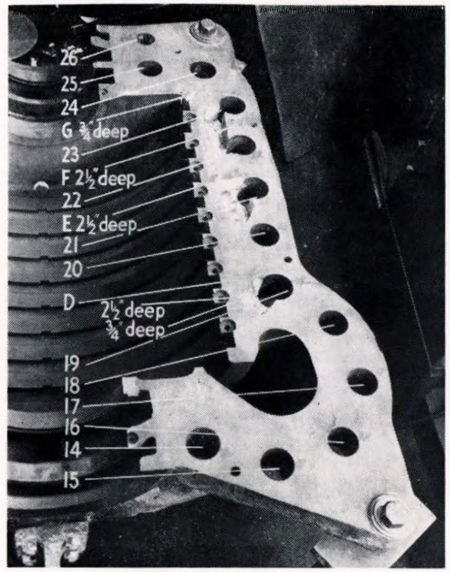
INTRODUCTION

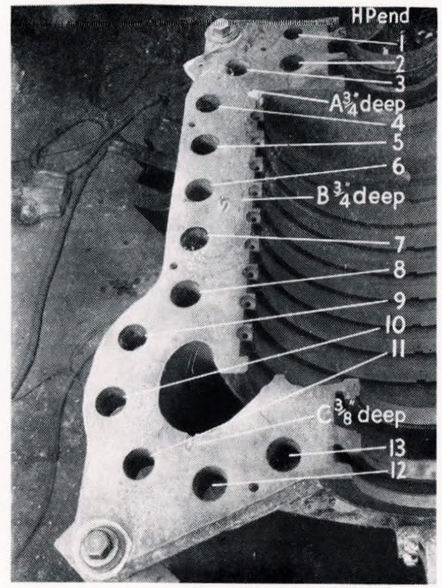
In 1948 the h.p. turbine rotor spindles in H.M.S. *Kempenfelt* in the wake of the carbon packing had become worn and badly grooved. In order to save time and the expense of removing the rotors from the ship, a special machine tool device was developed in H.M. Dockyard, Simonstown, to true up the spindles in place.

The apparatus was designed by, manufactured and operated under the personal supervision of the author, then acting fore-

* This article is published by permission of the Admiralty but the responsibility for any statement of facts or opinions expressed rests solely with the author.

FIG. 23—*View of defect in horizontal face of top half h.p. cylinder at bolt hole No. 8* (Fig. 25)





FIGS. 24 and 25—*Views of defects chipped out for welding in top half h.p. cylinder. The dimensions given indicate depth of holes*

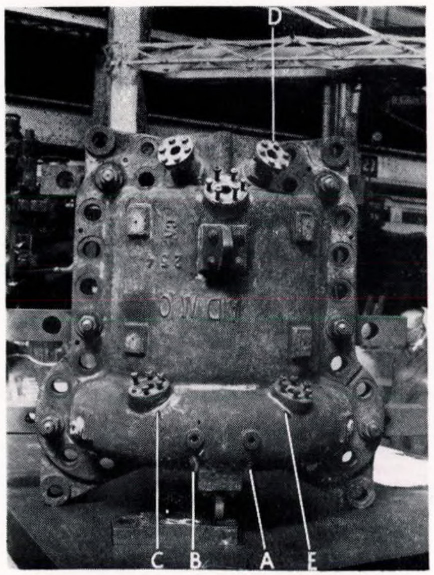


FIG. 26 *(left)—Top half of h.p. cylinder*

FIG. 27 *(right)—Top half of h.p. cylinder*

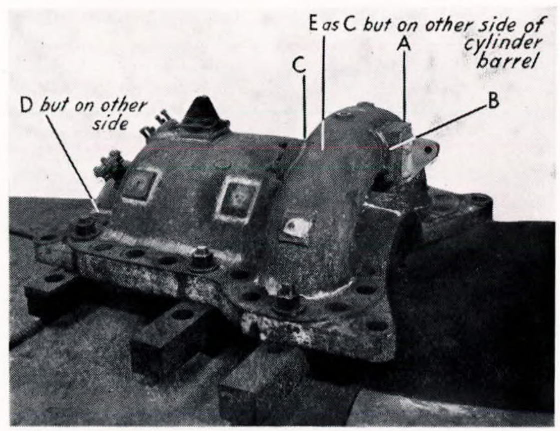


FIG. 28—*Top half of h.p. cylinder: view of defect "D"*

FIG. 29—*Top half of h.p. cylinder: view of defect "D" chipped out for welding*

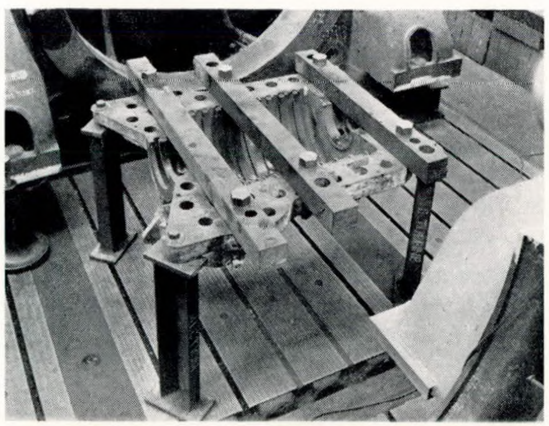


FIG. 30—*View of top half of h.p. cylinder strapped for welding*

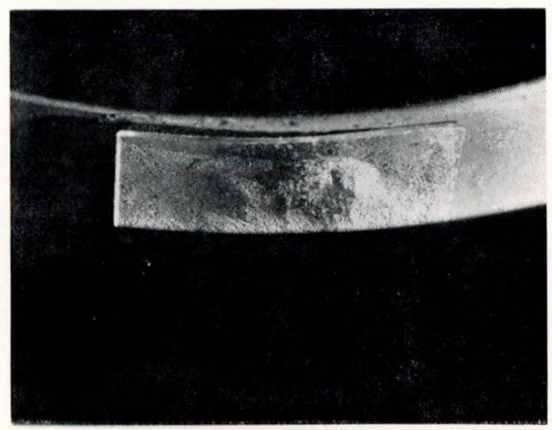


FIG. 31—*View of sand patch in steam belt of top half of h.p. cylinder*

man of the engineering branch at the dockyard, in June 1948, and this article is a description of the device and the method of operation.

PRELIMINARY WORK

Apart from the necessary first step of removing the upper half of the carbon packing box and the carbon segments themselves to enable the shaft surfaces to be inspected, the only other preliminary work required was to: —

(*i*) Remove the end cover at the forward end of the turbine casing so as to expose the end of the spindle.

(*ii*) Break the flexible coupling at the after end of the turbine so as to enable the rotor to be rotated independently.

DESCRIPTION OF THE DEVICE

The arrangement consists of two principal parts, viz., the driving mechanism, and the truing tool and holder.

Driving Mechanism

Motive power is provided by a low-geared pneumatic drilling machine which drives through a single train of two lathe gear-wheels, one (the driver) of 25 teeth and the other (the driven) of 75 teeth, both 3 inches wide.

The "driven" wheel is keyed on to an extension shaft which in turn is screwed on to the existing stud fitted in the end of the rotor spindle and normally intended for use in any axial adjustment of the turbine rotor. It is important to note that this arrangement means that the turbine must be driven only in the "clockwise" direction as otherwise the wheel and its shaft would be unscrewed from the stud.

The "driver" wheel and pneumatic drill motor are supported as convenient in suitable brackets and bearings, which are readily fabricated from miscellaneous parts and secured to any convenient fixed points at the turbine end or its vicinity.

The Truing Tool and Holder

The truing tool and holder consists of two major components, the holder block and the sliding tool which slides within it (Figs. 32 and 33).

The holder block is of steel, machined all over, and forms the fixed part of a "slide rest" in which the sliding portion is the tool itself.

The block, when the device is in use, is mounted on that landing face at the end of the turbine casing on which the gland box is normally secured, and is held in place by two

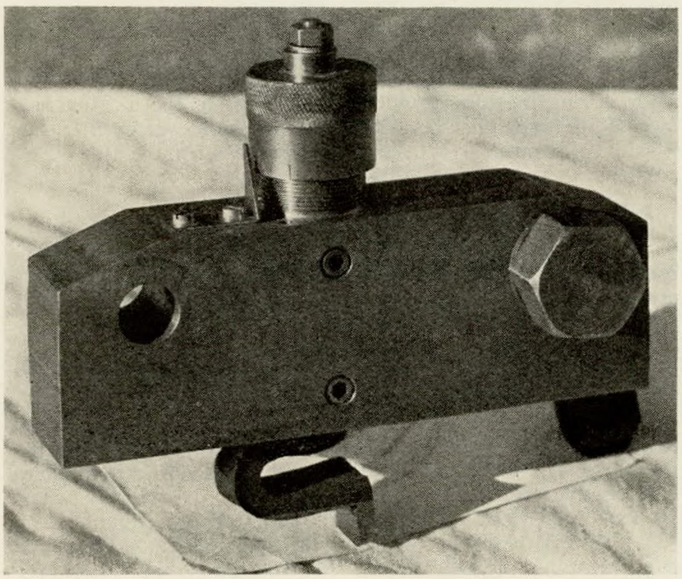


FIG. 32—*General view of tool holder and tool after assembly*

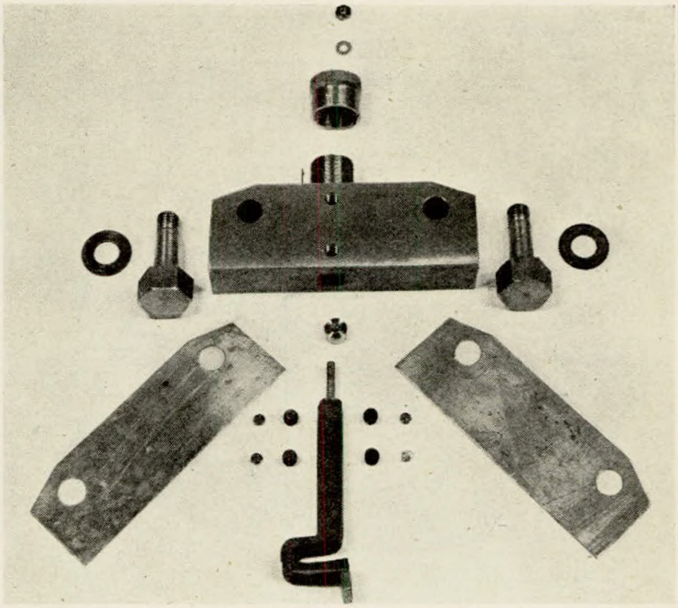


FIG. 33—*Separate parts before assembly*

special bolts which are inserted into two of the gland-box securing bolt holes (Fig. 34). These special bolts are given the minimum clearance in the bolt holes in the holder block.

The block is machined during manufacture to a thickness which allows for parallel liners of suitable thickness to be sandwiched between the block's inner face and the landing face on the turbine casing; this permits adjustment of the block (and tool) in the fore and aft direction so as to locate the tool relative to the portion of rotor spindle to be machined.

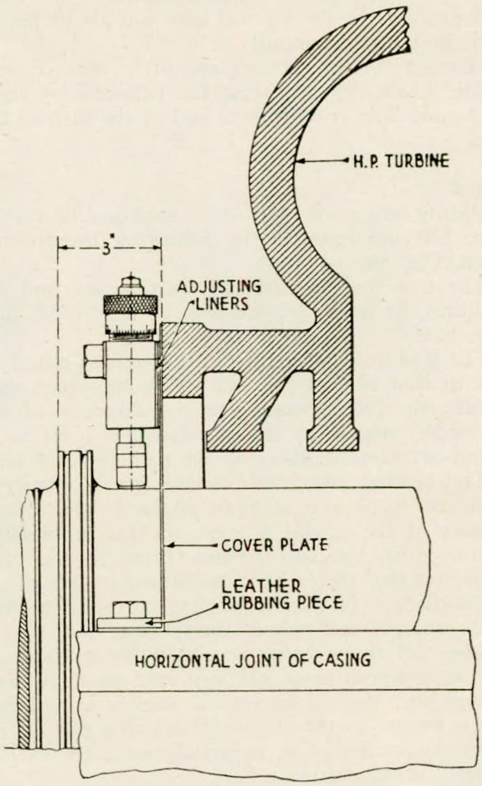


FIG. 34

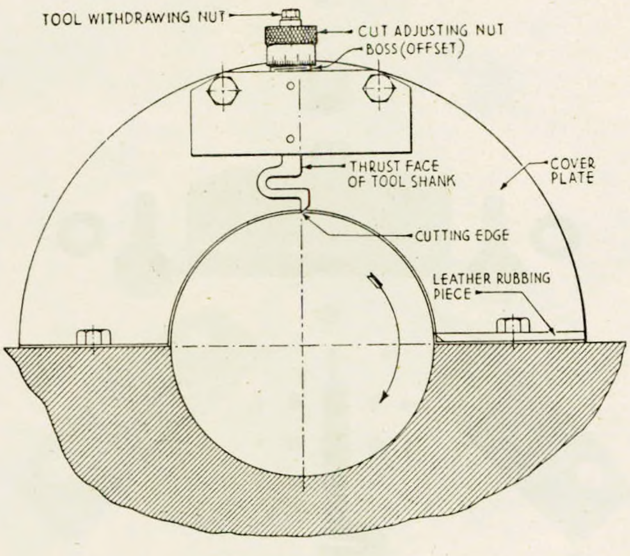


FIG. 35

These arrangements also ensure that the long faces of the holder block are in the true plain, both vertical and lateral, normal to the plane of the turbine rotor shaft axis. The block is provided on its upper surface with a round screwed boss, suitably offset from the fore and aft centre line.

Through this boss is drifted a square hole of accurate dimensions in which the forward and after internal faces are parallel to the long faces of the block and the two side internal faces normal thereto. The tool itself slides in this internal square hole.

The boss and square hole are so positioned that, with the block secured in place, the side internal face remote from the direction of rotation (that is, the face on which the tool thrust is taken) is precisely in the vertical fore and aft plane in which lies the axis of the rotor spindle (Fig. 35).

Two sets of "Allen" screws are fitted, one on each side of the holder block, for adjusting the tightness of the sliding tool in its square hole at whichever end of the turbine the block is mounted.

Sliding Tool

The sliding tool is of high-speed steel and of the ordinary spring type but incorporates the following important details in its design (Fig. 36): —

(*a*) The tool shank is ground truly square and is a neat sliding fit in the square hole in the holder block and its boss.

(*b*) The leading face of the cutting edge is exactly aligned with that side of the tool shank on which the thrust falls (the face remote from the direction of the rotor spindle) which, as already stated, is itself in the fore and aft vertical plane of the turbine rotor shaft axis.

(*c*) The cutting edge itself is also ground truly parallel in one plane and at right angles in the other to the sides of the square hole in the holder block; this is done with the tool already fitted in the block and ensures that the finished machined surface of the rotor spindle will be parallel and in correct alignment with the unmachined part of the spindle.

(*d*) The tool shank being too hard for drilling and tapping, a round brass nut is brazed on to its upper end and into this is screwed a steel stud; a small steel nut known as the "tool-withdrawing nut" is mounted on this stud for use in withdrawing the tool through the "cut-adjusting nut".

(*e*) The cut-adjusting nut, which operates on the screwed boss, is knurled externally and also graduated to show "cuts" of 1/1,000 inch or less; movement is indicated on a small vertical knife edge mounted on the upper surface of the holder block adjacent to the cut-adjusting nut.

(*f*) The width of the tool cutting edge is made about $\frac{1}{16}$ inch wider than the designed width of the carbon packing so as to ensure that the whole of the defective surface of the rotor journal is covered, and it is well radiused at the corners.

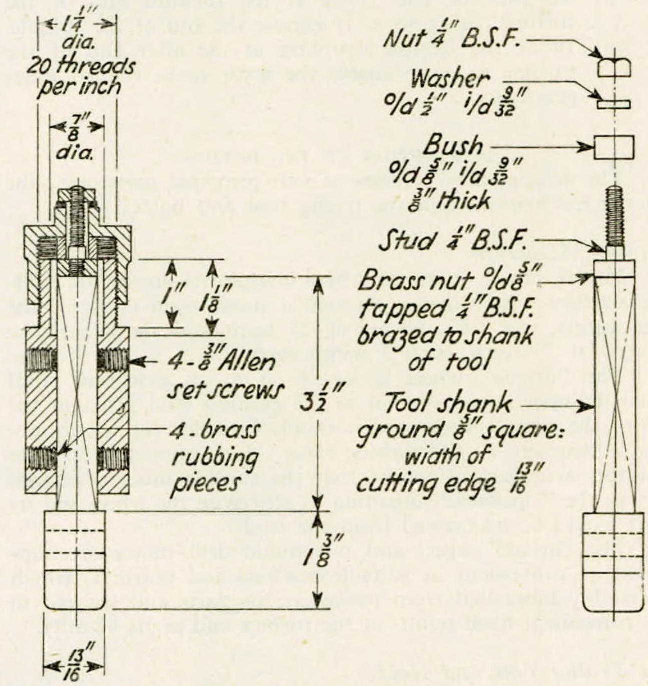


FIG. 36—*Details of tool assembly*

Special Feature of Design

It will be seen that the design is such that the line of contact of the tool cutting edge with the revolving rotor journal lies entirely along the highest points of the journal surface.

The cutting edge of the tool being in the same plane as the thrust face of the shank allows the tool to spring back under load or to return to its normal cutting position as the load reduces without any possibility of the tool "digging in"; it also allows for any lateral movement of the rotor itself in the horn clearances of its bearings.

Thus the design permits full freedom of both tool and rotor during machining without detriment to the finished surface of the machined part of the journal and yet ensures that it will be parallel and in true alignment with the untouched portion.

OPERATION

Forward and after journals are both machined with the rotor revolving in the "clockwise" direction. During the whole machining operation the turbine forced lubrication system is functioning with supply from the electric driven forced lubrication pump fitted. In this way true running conditions are maintained during machining with the rotor free from constraint and supported in its own bearings.

It will be found necessary to assist the starting of the rotor by hand lever on the teeth of "driven" wheel; thereafter the pneumatic drive, using about 10lb. per sq. in. air pressure, will revolve the rotor steadily at a speed of about 7 r.p.m. About 5 to 6 complete revolutions will suffice for each cut.

The cut is applied in steps of 1/1,000 inch by means of the "cut-adjusting nut", after first slackening the "tool-withdrawing nut" as necessary.

The tool is withdrawn when desired by tightening on the

TABLE I.—DATA GAINED WHEN MACHINING JOURNALS AT CARBON PACKING SPACES

To rig up drive and disconnect coupling		R.P.M.	Diameter of journal original, in.	Finished diameter, in.	Reduction in diameter, in.	Cutting time, min.	Remarks
Starboard forward	4 hours	7	8.5	8.493	0.007	7	—
Starboard after		7	,,	8.475	0.025	55	Very hard skin and irregular wear
Port forward	4 hours	7	,,	8.471	0.029	45	Very hard skin and irregular wear
Port after		7	,,	8.437	0.063	25	Very deep grooves

NOTE.—Tool maintained its cutting edge throughout the whole operation in each case.

tool-withdrawing nut after first screwing back the cut-adjusting nut as necessary to permit withdrawal. Each cut of 1/1,000 inch should be allowed to run itself out completely before a fresh cut is taken.

The cutting lubricant used up to the final cut is tallow; at the final cut, i.e. when the whole surface has been cleaned up, "Solvac" and tallow should be used as lubricants. This will be found to produce a highly polished finished surface.

Special precautions are necessary to ensure that all "turnings" are collected and removed as they leave the tool. These can take the form of temporary closing plates with leather rubbing strips to shut off the lower half of the gland box and the annular space at rotor end. In addition, the use of a strong magnet is required.

The majority of turnings become embedded in the tallow during machining and are easily collected and removed. Removal and search of the lower half of the gland box subsequent to machining in the operation described revealed that no fragments had escaped collection and removal from above the rubbing strips.

Three men are adequate for the operation; one to adjust the tool cut and spread tallow on the journal; a second to remove the cuttings by magnet and apply Solvac as necessary; and a third to control the pneumatic drive and assist generally.

Table I shows actual results obtained during the operation carried out in H.M.S. *Kempenfelt*.

10. AN IMPROVIZED BALANCING OPERATION†

Commander(E) H. E. C. HIMS, O.B.E., R.N.(ret.), M.I.Mech.E.*

The machine concerned in the defect to be described was a 500 kW. turbo generator with a spigoted flanged joint at the end of the rotor spindle to secure a stub shaft for the purpose of driving the governor gear and water pump assembly and carrying the overspeed trip gear. This stub shaft is shown in the accompanying sketch (Fig. 37).

The story starts with a fracture of the stub shaft just before the author joined the ship. The fracture occurred in the plane A—A, fortunately leaving the overspeed trip gear attached to the rotor, otherwise the speed governor would have "known" nothing but the fact that the machine was slowing down and would have supplied more steam as a consequence, almost certainly up to the limit where the rotor and/or its blading gave up the ghost through centrifugal force.

Although, from the shape of the stub shaft, it was obviously more likely for it to fail in the plane A—A than in a position where the overspeed trip gear would have been detached from the rotor, the latter position for a future failure existed as a possibility and considerable importance was attached to the determination of the cause of the original fracture.

After the failure, a spare stub shaft had been fitted, satisfactorily, so far as could be seen, except that there was a harsh vibration, of which the source appeared to be somewhere at the same end of the machine as the stub shaft.

Various opinions as to the state of the machine with the old stub shaft indicated that this vibration had always been present and gave cause for consideration of fatigue as the reason

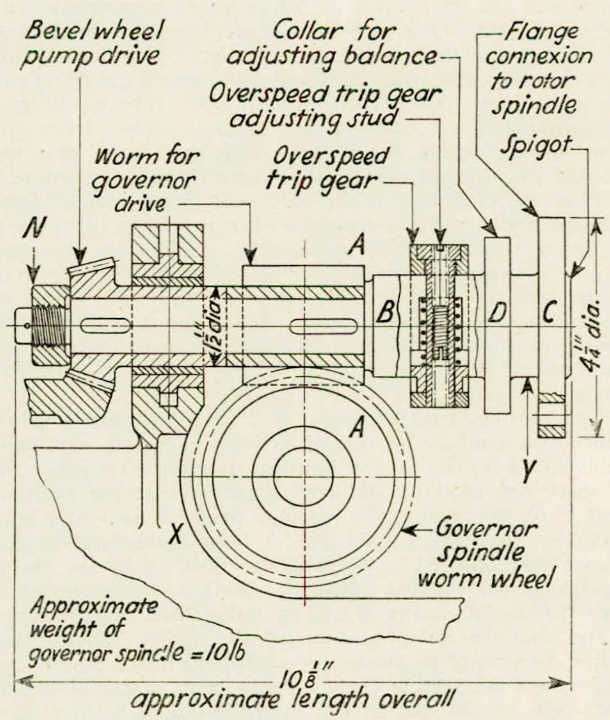


FIG. 37—*Arrangement of stub shaft*

* Test Bed Superintendent, Parsons and Marine Engineering Turbine Research and Development Association, Wallsend-on-Tyne.

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for the fracture, with the possibility of a similar fracture at some time in the future.

As a first step, circumstances being favourable, the rotor, complete with stub shaft, was removed and checked in a dynamic balancing machine, the result showing that the assembly was apparently in balance. Actually, as will be seen later, the balancing machine was not suitable for the job, but further investigations were necessary before this statement could be proved. Following this check, the machine ran with the same vibration as before.

The next step was to secure the attendance of representatives of two Admiralty departments and one from the makers of the machine. After a day's investigation the only conclusion reached was a half hearted one to the effect that the structure of the seating for the machine was unsuitable. This was not agreed by ship's officers, as a similar machine in the ship, on a similar structure, was running with no vibration at all and the flimsiest structures are at least unlikely to vibrate without some disturbing force. Following this visit and bearing in mind the future programme of the ship, it was decided that the efforts of the ship's staff must be directed towards finding a solution for the problem.

The first difficulty was in the determination of the frequency of the vibration. This sounds simple, but in practice it was impossible to guess whether it corresponded to the rotor spindle revolutions of 7,000 per minute or to the speed of the governor shaft, driven off the stub shaft at a much lower speed. The first assumption made was that this shaft was responsible and running it in a lathe at its normal running speed showed a certain amount of unbalance, which was cured by removing metal from one weight until it balanced the other exactly on a scale, having due regard to the position of its centre of gravity. The hopes aroused by this discovery were false, however; there was no change in the vibration when the corrected governor shaft was replaced in the machine and it was clear that its speed was not sufficient to make its unbalance felt.

The next step was to determine accurately the frequency of the vibration, there being as many guesses available as there were individuals observing. To achieve this a telephone receiver was strapped with insulating tape to a valve spindle on the machine and the vibration recorded as a very approximate sine curve on the tube of a cathode ray oscillograph, the distance in inches between its peaks being measured. To obtain a time base, a further curve was produced by using a quartz crystal in the circuit and this curve measured also. The results gave the frequency of the vibration as within less than 5 per cent of the running speed of the rotor and this 5 per cent was attributed to errors in observation (correctly, as it transpired).

That the fault lay in the stub shaft itself was a logical conclusion and proof was obtained when it became necessary to alter the trip speed of the machine by moving the adjusting stud in the centre of the hollow bolt through the trip ring. Moving one way increased the vibration and by moving the other way it was possible to eliminate 50 per cent of it. The adjusting stud was replaced in its correct position and the stub shaft removed from the machine for an attempt at correcting the balance.

An improvised method was used, taking advantage of the presence of a machine for testing aircraft generators, capable of variable speed between a few hundred and 10,000 r.p.m. The stub shaft was mounted in narrow bearings at the positions X and Y on the sketch, the bearings being secured to a rigid bedplate on the testing machine. A large diametrical bearing clearance was allowed, approximately 0·040 inch, the theory being that when running freely the shaft would rotate about an axis through its centre of gravity rather than its geometrical axis and that the radial position of the out of balance mass could be determined by advancing scribers to touch the surfaces B and C on the sketch, the markings being obtained on the "light" side, i.e. the side remote from the out of balance mass.

A brass pulley was substituted for the nut "N" and the drive from the machine taken through a belt made of lamp wick, to be slipped off on reaching a sufficiently high speed and so allow the shaft to run freely while slowing down, with only the constraint of the two loose bearings. The required scriber marks were obtained during the slowing down process. This arrangement proved satisfactory. The out of balance position was indicated clearly and the necessary correction made by filing a flat on the collar "D" on the sketch. The process was continued until the scriber markings suddenly moved through 180 degrees, indicating that the correction had been overdone.

By this time, a certain amount of wear and tear had shown itself in the bearings and the assembly generally and so it was decided to stop. With the stub shaft replaced on the rotor spindle the vibration had been reduced by at least 70 per cent. Shortly afterwards, two new stub shafts arrived from the makers and, with one fitted to the generator, the vibration disappeared completely.

An attempt was made to complete the balance of the original shaft by mounting the bearings at X and Y in such a way that they could be made free to vibrate laterally against helical springs and also turn about an axis on the vertical diameter; in other words, to produce a dynamic balancing machine with only slight departures from a conventional type to fit this particular shaft. The principle aimed at was to run the shaft at a speed well above the estimated natural frequency of vibration of the shaft and bearing assemblies, remove the drive (by slipping off the belt) and making scriber markings to determine the radial position of the out of balance mass when the natural frequency was reached, i.e. when the disturbing force was in resonance with the spring system. The out of balance mass was assumed to lie in the plane through the middle of the trip ring, and by a series of calculations assuming various unbalanced masses at various radii, the conclusion was reached that for any unbalance that could reasonably be present, resonance would occur between 2,000 and 5,000 r.p.m., which proved to be correct.

This project had to be abandoned eventually through shortage of time and for other reasons, but an interesting point arose. The stub shaft was so light and the balancing machine so clumsy that the former slowed down too quickly when the belt was slipped off and came to rest before resonance could develop. This difficulty was overcome by placing the axis of the balancing machine in line with the driving spindle and using a piece of rubber tubing as the driving mechanism. The advantage was that for all practical purposes the tubing could only transmit torque, and thus left the shaft free to vibrate laterally while being driven.

With this drive and the variable speed machine it was possible to find the exact speed at which resonance occurred and maintain it indefinitely while obtaining the necessary scriber markings, but this was as far as the experiment could be taken.

Conclusions

Balancing machines exist where indication of the extent and position of unbalanced masses are given by electrical means, magnet and coil systems causing the vibration set up by the unbalanced masses to generate currents, which are amplified to give the necessary indications. These machines are claimed to be much more sensitive than the purely mechanical type and certainly possess a considerable advantage in that it is not necessary to run at resonant speed. Whether such a machine would be capable of indicating the unbalance described in the foregoing, with the stub shaft attached to the rotor, is not known to the author, however.

The following conclusions apply to the use of the purely mechanical types of balancing machine, which, so far as is known, are the only types available in service establishments to date: —

(*a*) The machine in which the original check on the balance of the rotor and stub shaft had been carried out was unsuitable for this particular assembly. The natural frequency of its spring assemblies was chosen to suit the mass of the rotor, which imposed a very definite limit for safety reasons. At any speed within the range of this machine, therefore, the disturbing force due to the small out of balance mass present in the stub shaft was insufficient to have any effect on the springs. It is to be noted here that the disturbing force varies as $(\text{r.p.m.})^2$.

(*b*) When a rotating part consists of a large mass joined to a small mass, which construction may be expected when the small mass has a complicated structure, it is unlikely that a satisfactory dynamic balancing operation can be carried out on a single machine. In such a case, therefore, the two parts should be separated and each checked for balance on a machine suitable for its own mass. It is obvious, of course, that close attention is necessary to the fit of the spigot or other means provided to ensure that the two are eventually joined together truly axially.

11. SOME FURTHER VIBRATION PROBLEMS RELATING TO SHIPS AND THEIR ASSOCIATED MACHINERY

T. W. BUNYAN, B.Sc. (Member)* and B. HILDREW, M.Sc. (Associate Member)†

The examples given have all been the subject of investigation by the Engineering Research Department of Lloyd's Register of Shipping. The first three concern crankshaft failures of large main propulsion Diesel engines, the symptoms being in each case predominantly axial vibration. In the next section of the paper, reference is made to two cases where somewhat unorthodox means were employed for dealing with primary forces and couples in reciprocating machinery. The first example involved the fitting of a balance weight to the intermediate shafting in order to eliminate very heavy hull vibration which was produced by primary unbalanced forces in the main engine, the second was the attachment of balance weights by welding to achieve a reduction in the primary couple.

Case 1. Tanker: 466 feet × 59 feet × 36 feet; 8-cylinder, 4-stroke cycle single-acting main engine, 108 r.p.m.

It was reported that the main engine crankshaft had fractured through No. 7 crankpin some three years prior to the investigation. The cause of the failure had never been satisfactorily determined and when subsequently a failure of the thrust block occurred it was decided by the owners that the problem should be investigated. From the evidence available, axial vibration was suspected and a vibration test was made on board during the voyage home for repairs. The thrust block was found to be held together by strong backs and was secured to the adjacent deep frames by chains and bottle screws; in addition, some local patching had been made in way of the fractures in the casing. Records were obtained of the axial vibratory movements at the thrust shaft coupling and at the free end of the crankshaft. At the same time records were also obtained of the torsional vibratory movements at the free end of the crankshaft, which confirmed the calculated value of 590 c.p.m. for the one-noded mode of torsional vibration for the dynamic system.

The records of the axial vibratory movements gave values of 0·200 inch at the free end of the crankshaft and approximately 0·100 inch at the thrust at 103 r.p.m. at a frequency of 412 c.p.m., i.e. 4th order. At this critical the noise emitted from the crankcase was suggestive of the sides of the crankwebs hammering against the bearing cheeks. A crankcase examination revealed that the coupling bolt points had in fact heavily gouged out the ribs and flanges of the adjacent transverse bearing girder. A static compression test was then made on the crankshaft, as shaft in Fig. 38, to obtain a reliable value for the axial stiffness. Electric resistance strain gauges were affixed to Nos. 7 and 8 crankpins and deflexion gauges placed in position in each of the crank throws and at the forward and after ends of the crankshaft. It was remarkable to observe the accuracy with which readings could be repeated for various loadings and the return to zero of all gauges when the load was removed.

Recommendations were made for cropping of the propeller tips with a view to raising slightly the service r.p.m.

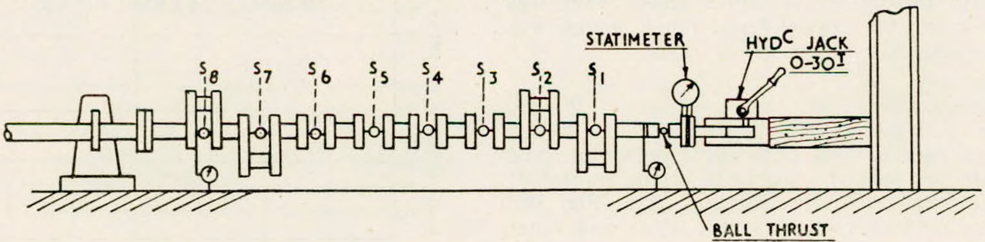


FIG. 38(a)

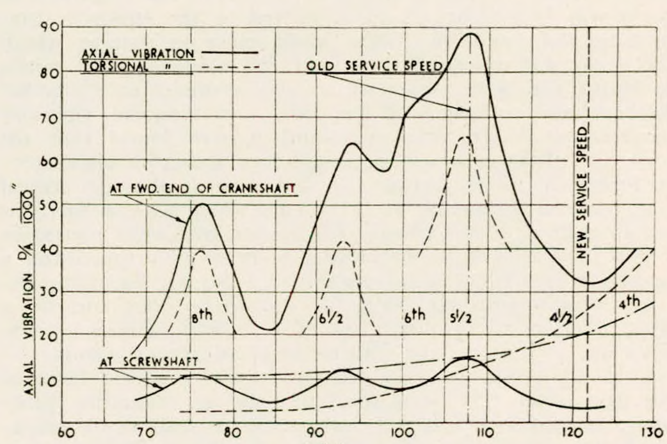


FIG. 38(b)

* Principal Surveyor, Engineering Research Department, Lloyd's Register of Shipping.

† Surveyor for Research, Engineering Research Department, Lloyd's Register of Shipping.

which at 108 r.p.m. coincided with the $5\frac{1}{2}$-order torsional critical. The new thrust block was finally fitted and rigidly secured and a further trial was carried out in home waters during the vessel's outward voyage. The axial vibratory movements at the thrust were reduced to negligible proportions and the engine operated smoothly at the new service speed of 122 r.p.m. (Fig. 38(b)). The bending stresses imposed on Nos. 7 and 8 crankpins were then deduced from the recorded axial vibratory amplitudes at the critical, which was found now to occur at 108 r.p.m. instead of 103 r.p.m. These were estimated to be approximately ± 2,000lb. per sq. in., which would not directly account for the failure. It was felt, however, that the critical should be avoided in any case and recommendations were made accordingly.

It is interesting to note that the dynamic thrust load applied to the thrust block at axial resonance of the crankshaft amounted to ± 21 tons which would be superimposed on the steady value of 31·5 tons.

Case 2. Tanker: 465 feet × 60 feet × 35 feet; 6-cylinder, 2-stroke cycle double-acting main engine

This vessel was reported to have had repeated trouble due to the failure of the geared drive to the fuel pumps and brackets supporting the layshaft. An axial vibration critical was suspected as this would provide the necessary axial vibratory reactions to the gearing which it was considered would cause the failure of the brackets supporting the idling wheel in the fuel pump drive. New brackets were made and fitted to accurately remachined and bedded landings on the engine columns. Oversize fitted bolts and dowels were used to secure and locate the bracket to the columns. On completion, records were obtained of the axial vibratory movements at the forward end of the crankshaft, and it was found that there existed a strong axial critical coincident with and excited by a major torsional vibration critical, which was strong enough to produce hammering of the thrust collar on the ahead thrust pads at the critical, suggesting that in this case the dynamic thrust considerably exceeded the steady thrust. Some frequency raising of the axial critical was produced by stiffening up the thrust seating, which also produced a reduction in the vibration amplitude. A barred speed range was rigidly imposed to cover the critical speed range, which was fortunately situated well clear of the service r.p.m.

Case 3. General cargo carrier: 452 feet × 63 feet × 38 feet; 6-cylinder, 2-stroke cycle single-acting oil engine, 111 r.p.m.

A vessel was reported to have experienced failure of some of the coupling bolts securing the couplings of the crankshaft bobbin piece. A photograph of the coupling faces (Fig. 39), which indicated considerable frettage at the inner and outer extremities of the couplings, suggested working between the coupling faces. Calculations indicated that the axial natural frequency of the crankshaft with the thrust as a semi-nodal point was 430 c.p.m. which, if excited by the 4th-order harmonics, would give a critical speed close to that at which the vessel was operating. A flight was made at short notice to board the sister vessel which was approaching Colombo. Records were obtained of the axial and torsional vibratory movements of the crankshaft and it was found that the torsional vibration was negligible, which indicated satisfactory performance of the detuner. A heavy axial vibration critical was, however, occurring at 111 r.p.m., which was, in fact, the running speed of the vessel. The results are shown plotted in Fig. 40. This critical was found to be sharply tuned and a reduction of 3 r.p.m. was sufficient to avoid it. Recommendations were accordingly made for this to be done and for a new propeller to be fitted having a slightly increased pitch. This has now been done with entirely satisfactory results.

In the meantime, the heavily fretted coupling faces of the first vessel were being machined and an interesting problem presented itself. It was not intended to remove the crankshaft but to reface these couplings in place. The couplings of the bobbin piece, which could be removed from the engine

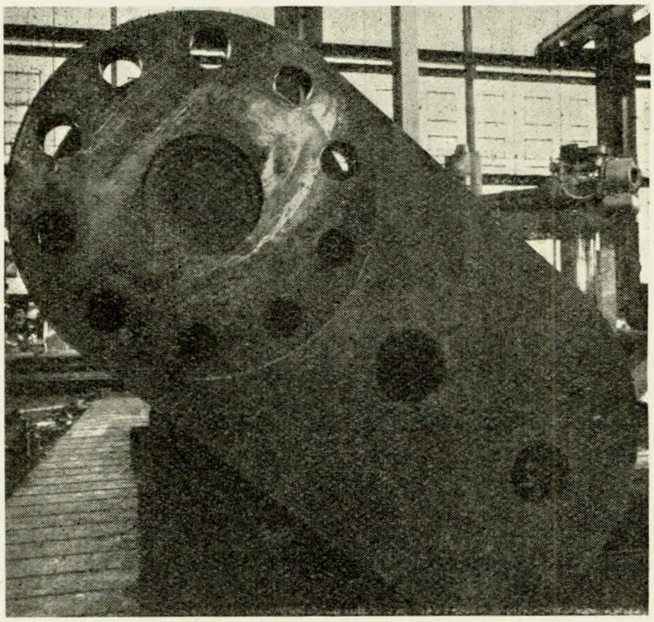


FIG. 39—*Showing frettage pattern on coupling face*

by merely dismantling the adjacent connecting rods, would be resurfaced in the lathe while the abutting crankshaft coupling faces would require hand work to be carried out *in situ*. The problem was how to ensure that when the four coupling faces were put up to each other there would not be excessive malalignment between the faces. The following arrangement was adopted and is shown in Fig. 41.

A $1\frac{1}{2}$-in. diameter reference hole was drilled right through both couplings before unbolting. The hole was lapped true

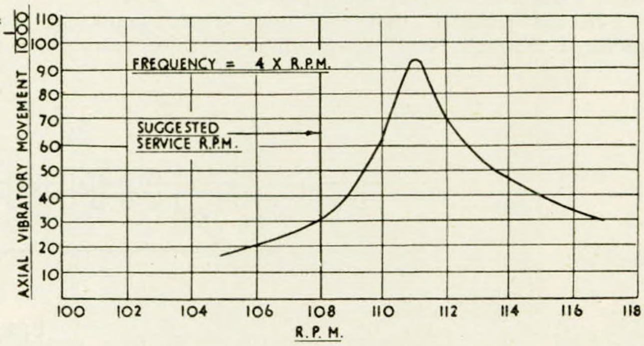


FIG. 40

and round. The bobbin piece was then withdrawn and machined. A ground bar size and size was inserted in one of the reference holes in the bobbin piece (Fig. 41(a)). The radius arm with two bearing bushes having a good fit on the bar and carrying a dial indicator at the other extremity (Fig. 41(b)) was threaded on to the bar and located by spring pressure against a registering collar secured to the bar. The arm was then run round the outermost plane surface of the coupling face and the indicator readings at each of the numbered bolt holes were noted. These are shown plotted in Fig. 41(c) as the full line. The bar was then inserted into the reference hole in the corresponding mating crankshaft coupling. The results obtained are shown dotted in Fig. 41(c). Small reference patches filed in way of the corresponding numbered holes in the crankshaft coupling would give a datum plane true to the machined surface of the bobbin piece coupling. It should be remembered that in this case directions are reversed, i.e. a high spot shown on the bobbin piece coupling would

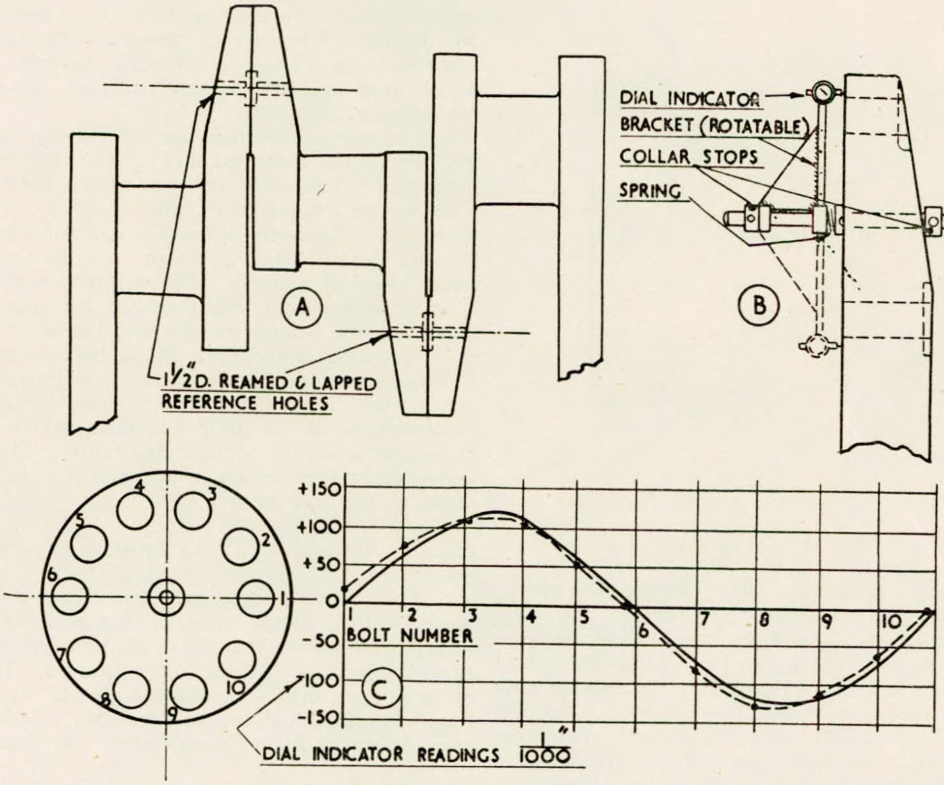


FIG. 41

mate with a low spot of the same amount on the crankshaft coupling because the dial indicator on the radius arm would be on the opposite side of the plane representing the abutment of the two mating couplings. The dotted curve shown has been plotted with signs reversed.

The arrangement worked admirably and saved the anxiety of having trial and error sessions in order to get the required accuracy of alignment of the coupling faces. It will be appreciated, of course, that it makes no difference to the accuracy of the method were the reference holes drilled at any angle to the coupling faces. The only prerequisite is that the holes should be continuous and true through both mating couplings. The lack of squareness of the reference holes with the coupling faces merely produces a sine curve instead of a straight line, which would result from a reference hole drilled perfectly square to the coupling faces.

Recommendations were also made that the abutment of both coupling faces should be limited to an annular fitting strip of 4½ inches on either side of the pitch circle, and that the very hard fit previously employed for the coupling bolts should be reduced as it was felt that the effective elasticity of the bolts had been seriously reduced.

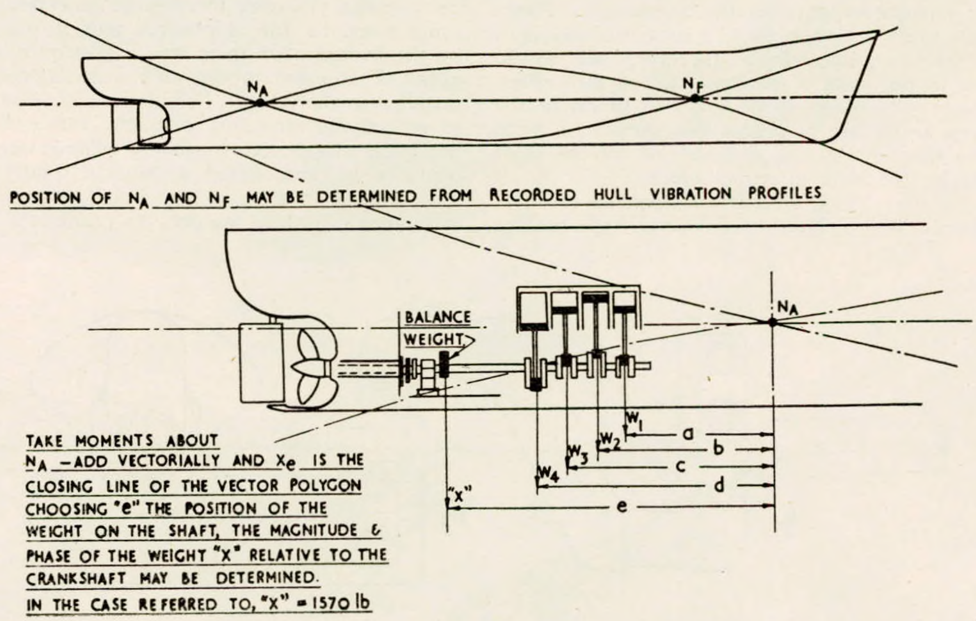


FIG. 42

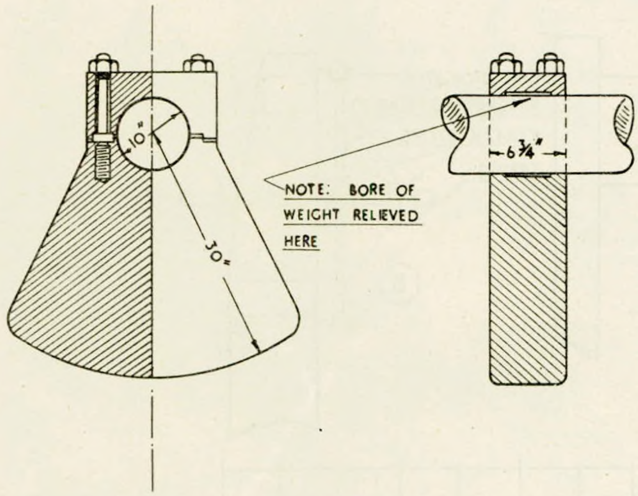


FIG. 43

The predominating theme in each of the foregoing examples has been one of axial vibration which to date has not reached such proportions as could be regarded as serious. The actual crankshaft failures which can be attributed directly to axial vibration are few indeed. In each case the remedy has been that of avoiding the axial critical for continuous operation, a simple matter, particularly as owing to little damping existing with normal axial resonance the criticals are sharply tuned. Propeller-excited axial resonances which have been met with occasionally can be considerably reduced by ensuring adequate propeller aperture clearances, which greatly reduce the exciting forces.

Case 4. 4-cylinder steam engine tanker

Excessive hull vibration was experienced on this vessel at full speed during her loaded trials. Records taken on board indicated that the hull was vibrating in the 2-node vertical mode and that the amplitudes were indeed large.

The cause of the vibration was readily appreciated to be primary unbalance in the engine which at full speed was exciting the hull at the 2-node natural frequency of 116 r.p.m.

The ideal solution would have been to balance the engine by the addition of balance weights on the crankshaft. However, owing to limitations of space, such a proposal was both inconvenient and expensive, especially as the crankshaft would have to be removed to put such a recommendation into effect.

A balance weight was therefore designed to fit on to the intermediate shafting as far aft as practicable, consonant with reasonable access to the stern gland, and support of the shaft and balance weight by the after plummer block.

The existing plummer block, which was designed to withstand vertical load only, was replaced by a Michell type bearing which would operate satisfactorily when subjected to the 360 degrees radial forces generated by the balance weight. The running clearance in this bearing was adjusted to 0·012 inch and the seating stiffened to prevent any local buckling or deflexion.

The method of determining the size of the weight was not new and is indicated in Fig. 42. The balance weight was not keyed to the shaft but was of the form shown (Fig. 43). The surface of the shaft and the bore of the weight were thoroughly degreased chemically and suitable clearance allowed between the keep and the weight in order that a good clamping stress could be assured. The weight was totally cased in and a small access door was provided for examining the witness marks on the balance weight and the shaft.

In the unlikely event of the balance weight moving due to extraneous causes such as the propeller striking a submerged object, the immediate consequence would probably be a reappearance of the hull vibration, which should draw the engineers' attention to this possibility. The arrangement has given satisfactory service to date and is to be applied to another vessel in the near future.

Case 5. Balance of a twin cylinder 12in. × 9in. refrigerating compressor

The occasion sometimes arises that due to the structure of the seatings or the presence of a hull critical, it is necessary to balance forces or couples in reciprocating machinery in order to avoid local or general hull vibration. In the design stage of an engine, this is normally a simple matter but on a completed installation it can be somewhat awkward. The previous example was a happy compromise that functioned admirably.

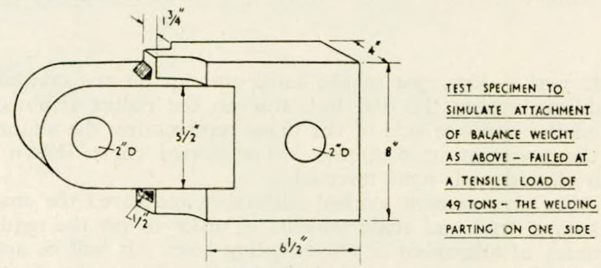


FIG. 44(b)

A problem was recently presented to the department where quite strong vibratory forces were generated in some compressor units intended for shipboard, and it was rightly considered by the makers that there was a possibility of trouble with the joints of attached piping and other ancillary equipment when installed in the vessel. Calculations showed that it was possible to reduce the offending primary couple in the compressor to negligible proportions by fitting balance weights of a size which could be accommodated within the totally enclosed crankcase. The difficulty arose when consideration was given to the attachment of these weights to crankwebs. Welding was the

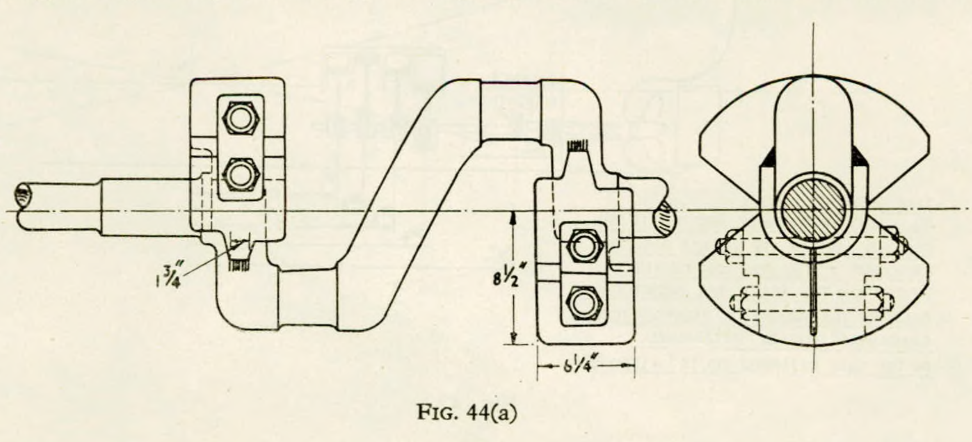


FIG. 44(a)

only convenient and practical attachment. The first unit dealt with had weights of the form shown in Fig. 44(a) attached by a fillet of welding 1¾ inches long by ½ inch throat section. The running tests on the balanced engine indicated that the vibration was virtually absent. Finally, in order to convince all parties concerned as to the safety of the attachment, a full size model (Fig. 44(b)) web and balance weight were welded together, using identical technique and electrodes, and tested to destruction in a tensile testing machine. The results gave a factor of safety of approximately 20 on the failing load. It will be noted that the design of the weights has been chosen to give a minimum locked up welding stress in the assembly and a minimum amount of weld has been employed to avoid distortion of the shafts, which were finish-machined. In this case, an approved welder and electrodes were used and the finished product was carefully crack tested.

ACKNOWLEDGEMENT

The authors wish to acknowledge with thanks the spontaneous approval for publication which was given by all parties for whom the investigations were originally carried out. They also wish to thank Dr. S. F. Dorey, C.B.E., F.R.S., and the Committee of Lloyd's Register of Shipping for their willing permission to publish this paper.

Discussion and Authors' Replies

MR. L. BAKER, D.S.C. (Vice-President), said that the symposium marked a very great step forward in the publication of information on a very important side of shipping—repairs. This year, as Mr. Sampson had pointed out, there were papers from America and Australia, and the Institute was deeply indebted to the authors who had taken so much trouble in sending them. He hoped that in due course they would find the contribution of the Institute as useful as their own.

He did not intend to make a detailed contribution on each paper, but there were one or two points in relation to some of them to which he thought he should draw attention.

In Paper No. 1, Mr. Davis had described a very ingenious repair on scraper boxes. He himself had no doubt at all that it was very satisfactory to the engine builder; but he was surprised that it was satisfactory to the shipowner. He would hate the idea of special packing rings in a ship which were different from those on all the other ships of the same class.

Mr. Gatewood's contribution in Paper No. 4 was an extraordinarily good demonstration of what a powerful tool "Metalock" was. This would no doubt be familiar to many people. His own company had used it extensively over the last five or six years for repairing cylinder heads and liners of Diesel engines, for jackets and frames, and for similar types of application. Had such a tool been available during the war, the consumption of cylinder blocks for small compressors and the like, which smashed with unfailing regularity, would have been much reduced.

Paper No. 5 by Mr. Gatewood had struck two chords in his mind. The photographs given in Figs. 15 and 16 in particular reminded him very much of an unusual occurrence in a cruiser in which he was serving at the beginning of the war. There were four turbo generators, but they should be nameless. They were unfortunate enough—or perhaps he should say fortunate enough from the point of view of the turbo generators—to be bombed. Just afterwards, it was discovered that two of the turbo generators had failures in exactly the same location as, and exactly similar to, those referred to by Mr. Gatewood. They were able to run on the remaining two from Scapa Flow to Glasgow. They then found that the two other generators had not quite the same failure but one due to the same cause. In the second case, the failures were fractures of the actual claws that broke off. The moral of this interesting story, as of many of Mr. Bunyan's contributions, if he might say so, was that one should not neglect the initial warning.

The two turbo generators which failed first in the way illustrated had a previous history of fretting of the couplings and the keys, and this was dealt with by the customary repair of skimming the shaft down, making a new coupling, and keying it on. Nobody thought of looking to see what was the cause. After some fifteen years' total service, the cause revealed itself in the form of a torsional oscillation which ultimately caused the failure of the two couplings on the forward generators. Had this been realized at the time of the first repair, the second would never have been necessary.

It would be noted that in Fig. 17 there were the words "Check nut". He himself had sent in a contribution which, Mr. Sampson said, would ultimately be published, referring to these check nuts, and to their slackening off. Why they should slack off was a complete mystery, but they did slack off and they rotated sufficiently to jack the turbine shaft forward and to ruin the thrust which was, he thought, unusual. While the cure was relatively simple—to put in adequate pins to stop it—the cause was still uncertain.

Mr. Yates's paper (No. 7) on thermal straightening, was extremely valuable. He sincerely hoped that it would not be necessary in the case of his own ships, but should it be necessary, the way to do it was on record.

He might perhaps mention in this connexion the need for ensuring stability of the rotor before it was put into service. Whilst it was fairly common practice now to ensure thermal stability, he was not at all certain that it was always adequately done, or that the reason for it was always adequately appreciated by the builders. It might be of interest that in twenty years, during which Metro-Vick had been doing this, they had at last come across a rotor which would not stabilize itself; in other words, it was a rogue and it just could not be used.

There was not much for him to say about Mr. Falconer's contribution on the defects in a high pressure turbine cylinder casting, except that the morals were there for those who wished to draw them. The casting was not a good one, but very few castings were! The defects should have been quite acceptable had they been adequately prepared. They were not adequately prepared. Consequently, defects occurred at a later stage, resulting from the repairs. He would further draw particular attention to the photograph of the sand which was found in the casting (Fig. 31). The line marking the end of the sand pocket which had been in service for about four years could be seen towards the right-hand side. Failure to remove the sand was one more example of failure to appreciate the importance of good workmanship, not only in the foundry but also on the part of the engine builders.

Finally, he would like to point the moral, if he might, of

Mr. Bunyan's contribution; and it applied indirectly to Mr. Low's paper (No. 2) also. He thought the answer was this: when you have unusual circumstances outside the range of your experience, for heaven's sake ask other people's advice!

MR. W. MCCLIMONT, B.Sc. (Member) referred to Paper No. 1 by Mr. Davis, concerning experience with Doxford scraper rings and scraper boxes. He said he was not entirely happy about the simplicity of the diagnosis. It was concluded that the coning of the lands of the scraper boxes had been brought about by overheating of the boxes when the original blow past of the power rings took place. He could not see that such overheating could occur directly. During the expansion stroke, cylinder gases escaping past the power rings would pass into the scavenge belt and the production of sufficient heat at the scraper boxes due to the heat contained in the blow past gases was hard to accept. Though he had not seen the job in question, he would suggest that it was more likely to be due to a scavenge belt fire, possibly set off by blow past due to sticking rings. The incidence of such a fire would also be made more likely by build-up of unburnt residuals in the scavenge belt associated with the use of heavy oil. The trouble might also be aggravated by over-oiling of the cylinders. The significance of the case outlined here could, therefore, be extended to include scavenge belt fires.

Another possible solution might be of interest—a method of modifying the rings other than the one suggested by the author. What the author did was to put a taper on the 3 mm. wide land of the ring to counteract the coning of the ring grooves; the same effect could be achieved by using scraper rings of an "L" section. One had the original ring in the form shown in Fig. 45(a), and the ring was modified by cutting away a ring of metal (Fig. 45(b)).

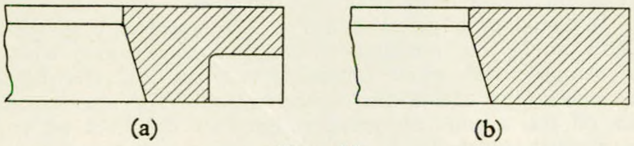


FIG. 45

This modification to the ring must be carried out after the ring was completely finished; otherwise, if finished on the inside diameter after being made into an "L" section, its oil controlling qualities disappeared.

The modification to the ring caused it to become conical in shape; that is, it had the same effect as the modification which Mr. Davis showed. It was suggested that it could be carried out in an emergency more readily than the scheme shown in the paper.

It might be worth experimenting with the cutting away of rings in this manner on all engines, because such a scheme had the advantage that it ran in very quickly and retained its oil-controlling qualities.

Like Mr. Baker, he felt that it was not the wisest of palliatives to introduce something non-standard. Possibly the modification he himself had suggested would not be quite so troublesomely non-standard as the one which had been brought in.

It was doubtful whether the real function of these rings was always appreciated. They should not be regarded as "glands". He rather liked the American terminology "oil cutting rings", and suggested that the lower ring in particular should be regarded in this way. It should have the lightest pressure consistent with the desired economy and should present a definite cutting edge to the film of oil on the piston rather than be bedded in over its full length of contact with the piston.

MR. J. MCAFEE (Member) hoped the authors of these papers would agree that the interest of their subjects lay not so much in the defects themselves as in the way they were overcome. After all, every motorist talked about excessive oil consumption; fractures in castings dated from before the time of the practical man who first thought of fitting a cement box, and cracked or vibrating shafts were part of a normal day to at least one of the contributors. The emphasis, then, was on *how* the job was done.

Mr. Gatewood gave an example of the use of "Metalock" for repairing an air pump cylinder. During the seven or so years since this system, of American origin, was introduced in Britain, it had on many occasions proved its advantages for the repair of iron castings in circumstances where renewal would have entailed long and costly delay. It was particularly useful where the fracture had been caused by an "accident", but where material or design was at fault "Metalock", or indeed any other form of repair, could at best only defer the day when renewal became necessary. It was suggested, therefore, that Mr. Gatewood's reference to cracked Diesel engine cylinder heads and liners might require some qualification.

He was interested in Mr. Yates's remarks on the straightening of turbine rotors by thermal means, since only a few days ago he took some part in a discussion between an engine builder and a superintendent following one of these operations on a new ship. The engine builder was confident that a first class job had been done but the superintendent was thinking ruefully of his brand new machinery and, in spite of the restored smooth running, did not appear to be entirely contented. Well, of course, some people were never satisfied, but still, some of his sympathies went towards the superintendent. Bent rotors were far too common these days and he wished Mr. Yates had told them not only how to repair them but how to stop them from getting that way at all.

The lucid description of what happened to the metal of a rotor shaft during heating and bending interested him as he found there some support for an explanation which had occurred to him in connexion with another kind of machinery defect having nothing whatever to do with turbine rotors.

It might be remembered that one of the contributors to the similar symposium a year ago described an unusual form of cracking in the steam drum of a watertube boiler. That was discovered in 1946 when he himself was occupied with a sister vessel where the same trouble came to light. These incidents were brought to mind about a couple of months ago when exactly similar cracking was seen, this time in water drums and in boilers of a different type.

These cracks all had common features: they occurred in the plain undrilled part of the drums, they ran circumferentially and not axially, varying from an inch or so to about eighteen inches in length and of varying depth, and in all cases the surface of the drums in way was either exposed to the furnace flame by design or apparently had become so through faulty brickwork. Subsequent tests had indicated that the drum material was not at fault and his own view was that the trouble was a purely thermal effect which in origin was analogous to what happened in the heating of a turbine rotor. The part of the drum exposed to flame reached a higher temperature than the opposite side and tried to expand but was restricted by the remainder of the drum, although this probably tended to bow. The hottest metal would be near the surface and with the large temperature gradient across the comparatively thick wall it would be well above the point where plastic flow commenced. Due to the tendency to expand, and the restriction, this metal would actually yield compressively. The result was that when the boiler was shut down and the drum cooled, with uniform temperature throughout due to conduction, then the previously overheated metal surface which was now shorter than before, was placed in tension.

This would probably not be serious if the process were not repeated at the end of every steaming period. The cracks which had been noted disappeared in some instances after light grinding, but in one case the fracture extended through the thickness of the drum. This was discovered not under steam but during survey, the reason being, no doubt, that when the boiler was in operation the cracked side was under strong compression and leakage could not take place. The considerable tensile stress present, however, when cold, was shown by the sudden rupturing of the metal when a suspected drum was subjected to hydraulic test and hammered.

He was sure that they would hear more of these defects in due course, but in the meantime he should apologise to Mr. Yates for using his contribution to the symposium as an excuse for talking about something else.

The CHAIRMAN said he now proposed to call upon some of the authors to reply.

MR. J. DOWIE, B.Sc., replying to questions on Paper No. 1, informed Mr. Baker that the owners were very satisfied with the modification carried out. Had they wanted to replace the defective scraper boxes they could have done so, but having had a good deal of worry and trouble for some months the owners were very pleased to have a solution. He took it that Mr. Baker was an owner.

MR. BAKER said he wished he were.

MR. DOWIE said Mr. McClimont had raised a difficult point. It was found that some rings, when bedded to the skirts, showed openings up to six-thousandths of an inch. The scraping edge was important from the point of view of "bedding-in". They had actually tried to work to openings of two-thousandths of an inch, bedding ring to skirt. With a vertical clearance of six-thousandths to which they now nominally worked, he doubted very much whether the form of ring proposed by Mr. McClimont would be satisfactory. He asked whether he was right in assuming that Mr. McClimont did not like the tendency to have a sharper edge.

MR. MCCLIMONT said he wanted to achieve an edge effect without making it excessive.

MR. DOWIE said that the lubricating oil consumption had become normal and that it continued to be normal. There was no tendency for the scraping edge to become ineffective and the oil wedge to develop on the upstroke.

MR. MCCLIMONT said he did not doubt the efficiency of the modification. He was suggesting an alternative way of achieving the same effect.

MR. DOWIE said he was not clear as to the advantage of the ring suggested. It could possibly provide an alternative, but the suggestion might be inadvisable from the point of view of the possibility of its allowing the ring to pivot, having a small bearing surface on the upper side and on the back of the ring. He might add that the six-thousandths vertical clearance had in some instances been increased owing to the gumming-up that developed in relation to the power rings. Mr. McClimont did not seem to be convinced that the gumming-up was really due to blow past. Again, he could only say that to his knowledge the chief engineer of the ship had never reported scavenge fires and the chief engineer had been very apt at making lengthy reports.

Previous practice had been to adopt vertical clearances of two-thousandths of an inch on the power rings, but it was now realized that this clearance was inadequate and four-thousandths vertical clearance was being adopted. The gumming-up was due to blow past and the small vertical clearances on the power rings might have caused a further tendency to gumming.

MR. MCCLIMONT said he should perhaps have added that it seemed to him that any blow past during the expansion stroke would pass into the scavenge belt where its temperature would be lowered by the scavenge air and that it was unlikely that appreciable quantities of hot gases would reach the scraper box.

MR. DOWIE hoped that it would be appreciated—though it was not shown on the diagram—that the scraper box extended up and was bolted to the entablature. If the blow past could get to the scavenge belt the box would heat up.

MR. D. KROEGER (Member) said Mr. Gatewood had sent some slides which would perhaps give a better impression of the turbine and coupling ends than Figs. 15 and 16. These might clarify the position. (*Two slides shown.*)

MR. BAKER agreed that they were very good.

MR. KROEGER (continuing) said Mr. McAfee was correct in saying that "Metalock" was not normally used where the design was defective. It was mainly used on fractures due to accident. It could, however, be used in connexion with the porosity of castings. One reason why it was thought that this particular case might be of interest was that it was one of the few cases where a piece of the casting was completely fractured out in a triangular shape. Actually the same piece was replaced.

"Metalock" could be used should porosity be found in a casting. After some years of service, a piece could be completely removed and a new section "metalocked" into place. This had been done in a number of cases.

He did not see any possibility, except perhaps in very rare cases, of trying to correct design with "Metalock". They had used it for a number of years. They had repaired high pressure cylinders on Liberty ship engines with it. Only recently, in Liverpool, they did a repair on an engine bedplate by "Metalock". Perhaps unfortunately, some owners were not completely sure of it, and one often found that a doubling plate was also installed over the top. But this was no fault of "Metalock". Most people would rather be on the safe side until "Metalock" was completely proven to them personally. If there were any other questions, they would be passed to Mr. Gatewood in New York.

MR. YATES said it was very apt that Mr. Baker should refer to the thermal stability test, but he would suggest that the reference was general and not specific only to rotors which had been straightened. Twelve rotors had been straightened by this process during the past eighteen months. In ten cases there were no convenient facilities for doing thermal stability tests, and additionally those ten cases did not include the first in the series. There had so far been no sign of regression in spite of the thermal stability test having been omitted.

In two other cases, the test was carried out but did not show any measurable change in the final condition of the rotor.

In the opinion of the research association with which he was connected, rotors undoubtedly required the thermal stability test. He would not spend long on it; it was perhaps a little irrelevant. But when this test was first publicized, it was intended to correct a state of affairs which sometimes arose in the final machining of a rotor. It was said that this machining sometimes introduced into the outer skin of the rotor some local changes of properties believed to be variations in the thermal conductivity of the metal near the skin. The result was that when the rotor was heated in any fashion—either in a heating box or in the turbine—it heated unevenly, simply because the heat could not penetrate all parts of the rotor at a uniform rate. Very little, if any, evidence had been found that that was a frequent occurrence in turbine rotors, but definite evidence had been found that rotor forgings were sometimes not thermally stable. In these cases it was found that no matter how often the thermal stability test was carried out—the test comprising heating to a temperature somewhat above the running temperature and perhaps a little higher if there had been appreciable bowing—the rotor still distorted on fresh heating after cooling. Mr. Baker had stated that Metropolitan-Vickers and Co., Ltd., had had one case in twenty-one years. His own association had found three in the last two years, and only that morning he had heard of five others experienced by another firm.

In some cases it was possible to use the rotor by subjecting it to a complete reheat-treatment. In some cases it simply had to be rejected. It was, therefore, becoming standard practice among marine builders to do this test on the rough forging, after rough machining, in order to detect a faulty rotor as quickly as possible before too much time and money had been spent on machining and blading it.

Another speaker had mentioned a recent case in which a rotor was straightened and the builders were quite satisfied, while the owner had some doubt. He did not blame him if it was his first experience of the process. It *was* somewhat frightening. But there was every hope that after twelve months he would be able to say that it seemed to be working out all right.

The question whether bending could be stopped was, he thought, receiving something like 20 per cent of all the effort that was being devoted to the development of steam turbines. So far, there were one or two clues. It was thought to be an advantage if the gland fins were on the rotor working against flat inserts in the cylinder. In a recent case it was found that there was insufficient freedom for the cylinder to expand and the cylinder hogged and bent the rotor in spite of the fact that the gland was of this type. But the bend was a small one. In addition, the material of the strips was harder than it need have been. When the fins were made in some non-ferrous material, they usually had to be something stronger than soft brass, because they had to withstand appreciable pressure difference, and the working stresses in the fins were not negligible. But if such fins were integral with the rotor and, therefore, of steel, it was possible to use really soft inserts, and it was intended to experiment with some other materials in the hope of finding one which would take a rub without bending the rotor.

The cracking of boiler drums due to local contact with the flame sounded as though it might be a similar process, though it was a little difficult to understand how the crack extended right through the drum. With local heating on one part of the drum, there would be a steep temperature gradient through the wall and, therefore, the outer members eventually ended up in tension and the inner ones in compression. But as the crack penetrated nearer and nearer to the inside wall, it must be affected by other parts of the drum away from the zone of flame impingement which produced the tensile stress and that was much less likely to cause a large stress as the drum was free to bend and even out stress.

He would like to ask a question himself. It related to Mr. Falconer's paper, No. 8, on the repair of a high pressure turbine cylinder casting. He had not enquired, but looking at the sketch he thought it was a cylinder with which he was connected in a design done in about 1947 or 1948. He wondered whether Mr. Falconer thought it might be an advantage to saw-cut through the outer part of the flange into at least a few of the bolt holes. It struck him that the kind of stress which could be put into cylinder flange when the turbine was worked up in power must be similar to the stress which occurred when cooling off after casting. In other words, the same effect as initially caused some cracks in these regions would operate during service and tend to extend the cracks. Designs of this type had been modified in a number of cases, certain of the bolt holes being saw-cut through so that the metal of the outer (and cooler) part of the flange did not restrain the part inside. One had, of course, to leave certain holes for fitted bolts; but normally the number of fitted bolts used was, or used to be until a few years ago, very many more than was really necessary to register the top half of the cylinder with the bottom. He would be interested to know whether Mr. Falconer had considered this additional item.

MR. W. H. FALCONER said he had not considered the possibility of introducing saw-cuts, because it was believed that the cracks were due to an inherent fault in the original casting. As would be seen from the photograph, the cracks ran round original welds in the metal, and it was thought that if the old welding were removed and the casting cleaned out, that would meet the case.

Considering Mr. Yates's suggestion further after the meeting, he understood that saw-cuts were sometimes introduced to make the flanges more flexible. As the casing did not show any signs of distortion or leakage, the object of the repair was purely the removal of the defective metal and its replacement by welding with the minimum amount of distortion. This was successfully accomplished and could be claimed to be a completely satisfactory repair.

The discussion was then resumed.

MR. BRYAN TAYLOR, B.Sc.(Eng.) (Member), said he proposed to confine his remarks to Paper No. 11 and to refer in particular to the examples of longitudinal vibration of shafting. In this paper, as in most of the others, the emphasis was, of course, on the solution to the problem, but he himself wanted to refer, in the first place, to the cause of the trouble, namely, the origin of the exciting forces which caused axial vibration of propeller shafting.

On page 70, when referring to the effect of propeller aperture clearances on the magnitude of the exciting forces, the authors stated that propeller-excited axial resonances had been met with occasionally. While he understood that all vibratory axial forces originated at the propeller, he enquired whether he had interpreted this statement correctly in assuming that longitudinal vibrations caused by variations in the wake distribution were the exception rather than the rule.

It seemed that there were two ways in which axial exciting forces could originate. Firstly, torsional vibrations could be converted by the propeller into axial vibratory forces, as in Case 2. Here the authors said (page 86) "it was found that there existed a strong axial critical coincident with and excited by a major torsional vibration critical". The second way in which axial forces could be set up in the shafting was by the variation in thrust due to irregularities in the wake distribution. When this occurred the frequency of the applied forces would be equal to the number of blades on the propeller multiplied by the r.p.m.

In Cases 1 and 3 he noted that the axial criticals coincided with the blade frequency (assuming the propellers to be four-bladed) and it seemed that in both these ships it was possible that the vibrations were due to wake variation. He invited the authors' comments on this point, however.

There were one or two other points of interest in connexion with Case 1. Firstly, he noted that after cropping the propeller tips the natural frequency of axial vibration increased from 412 to 432 v.p.m. whereas the torsional critical speed remained unchanged at 108 r.p.m. (assuming the curves given in Fig. 38(b) referred to the conditions after alteration of the propeller). He would have expected the change in inertia of the propeller and entrained water would have affected the torsional critical also, and he asked the authors whether they could explain this. The second query in connexion with this investigation concerned the stresses in the crankpin. He asked the authors whether the figure quoted of $\pm$ 2,000lb. per sq. in. was measured on the plain part of the pin or whether it represented the maximum stress, taking into account the stress concentration effect of the fillet between the pin and web. Also, did they consider that torsional stresses could have been a contributory factor in the failure reported?

Although he had not made a close study of the subject it seemed to him that the natural frequency of axial vibration must be affected to a considerable extent by the stiffness of the thrust block. He wondered whether the use of hydraulic cylinders to support the thrust shoe, as in a thrustmeter, would alter the stiffness of the thrust block and so result in a change in the natural frequency of axial vibration. If so, this might possibly be a cheaper way of overcoming the trouble than fitting a new propeller or even altering the existing one.

The last point to which he wished to refer was the fit of coupling bolts, mentioned by the authors on page 87. He agreed that the practice of making these bolts a hard fit over the whole of their length reduced the effective elasticity. He had often wondered why the diameter of the coupling bolt was not reduced over, say, half of its length, so that it would fit only over a section at each side of the junction of the two couplings. In that way the elasticity of the bolt would be increased and so tend to reduce the possibility of fatigue failure from fluctuating axial stresses in the bolt arising from misalignment of the shafting.

MR. BUNYAN said he had hoped he would not be drawn into a lengthy discussion on the causes of axial vibration, which was a vast subject. In the examples quoted in the paper, the vibration was propeller-excited, i.e. four times shaft r.p.m. It had been found from other work that the severity of the excitation was increased where propeller to sternframe aperture clearances were small; in some cases this could approach full load thrust. It could be seen, therefore, that large forces were available in certain cases for causing axial vibration. It had also been found that axial vibration criticals of crankshafts could also be excited by gas load. In the cases referred to in the paper, the mode of vibration was coupled with the dynamic impedance of the thrust seating. In the first case the critical referred to in the paper as occurring at 103 r.p.m. was recorded with the damaged thrust block temporarily repaired, and it would be seen that the securing of the thrust block raised this value to 108 r.p.m.

Mr. Taylor's idea of using a modified form of thrustmeter might be practicable.

In regard to Mr. Taylor's remarks about the ± 2,000lb. per sq. in. bending stress recorded by strain gauges, this value did in fact include the effect of the stress concentration of the fillet, the gauges being applied directly in the fillet.

He was grateful to Mr. Taylor for raising the matter of coupling bolts, as it gave him a further opportunity to express his views on the subject. Mr. Taylor was probably better informed because he had been personally associated with a research into the fatigue strength of large bolts in direct reversed stress. It appears to be difficult for some engineers to accept what might be regarded as a revision of established ideas on what constituted a well fitted bolt. Particularly in the case of coupling bolts which were liable to bending caused by malalignment, it was essential that the full elastic strain of the bolt be utilized in the hardening-up process. Where bolts were driven in hard, it was doubtful whether the hardening-up of the nut produced any appreciable pre-stress, and under these conditions, when subjected to axial load or bending, there was a real danger of fatigue failure at the first thread or of slackening off of the nut.

MR. L. C. LEIGH (Member) said he did not consider the suggestion to introduce flexibility at the thrust block to be satisfactory. If there was a 50 per cent torque variation due to wake difference, there would surely be a corresponding thrust difference, since the propeller would convert this change from torque to thrust. With such large variations, the propeller must be held rigidly by the thrust block or it would be dancing about. If any attempt was made to eliminate it, there must be a corresponding difference in the attitude of the propeller to the water. The axial movement would not be in thousandths of an inch but much greater since they were dealing with a large thrust variation.

Again, since the crankshaft must be located, a flexible connexion would be required between thrust block and crankshaft and this would introduce other complications and fretting between sliding surfaces.

MR. BAKER, referring to Mr. McAfee's comments, said that "Metalock" had, to his certain knowledge, been used for cylinder heads and liners for five-and-a-half years, and he had every reason to believe it was used before that. It had been entirely successful in dealing with thermal cracks, and the only case of failure of which he was personally aware was a cylinder head on a 740 supercharged four-stroke engine which was "metalocked" due to cracking and which ran for two years before it started to leak again. The failure through leaking was exactly the same as if it had cracked in the first place. The normal operation did not endanger the engine at all. He had no hesitation in using it in any part of the engine at all where there was no reciprocating motion. He would not put a ring to run over a "metalock" in a Diesel engine but in any other part he would have no hesitation whatsoever.

He was glad to hear from Mr. Yates that 20 per cent of effort was going on the design of glands, because he felt sure that was right and proper. Glands, as normally designed, were responsible for considerable loss of efficiency. They caused even more inefficiency if they did not stand up to the job they were supposed to do. It might interest Mr. Yates to know that, during the trials of a 950° F. turbine there was a touch on an h.p. turbine and sparks were seen. They ran for half-an-hour on the turning gear when the rotor was perfectly all right and so were the glands.

MR. F. J. WELCH (Member) said that what struck him about the present symposium as compared with the last one was the very high standard of engineering skill that was displayed. In fact, it was the work of experts rather than of the average "general practitioner" engineer and it made him feel that he had gone to sea in a very much cruder age when that famous artist of pre-war days entertained them with his delightful sketches which were much nearer the truth then than now. He referred, of course, to Heath Robinson.

For instance, one would not, in those days, have looked for a mere two-thousandths of an inch to explain the oil leakage in Paper No. 1.

The repair of the crankshaft in Paper No. 3 showed how completely welding had made such a piece of first-aid possible. This would have been quite impossible thirty years ago, as would have been the repair described in Paper No. 4. The tools available at sea, and even in some of the repair shops in those days were not very far removed from the old, hand forged flat drills.

In Paper No. 5 he was very interested in the method of attaching the two surfaces by the screwed ferrules with internal dowels. No doubt many people had put that under their hat for further use! He himself had used screw dowels successfully in a steering-gear shaft repair. It was interesting that in an 11-knot ship in an Atlantic gale the repair had to stand up to the maximum stresses, because the seas hit the rudder just as hard, even though the ship was stopped. It was the effect of the seas on the rudder and not the ship's speed that caused the damage.

Again, the straightening process was a job for the expert when it came to turbine rotors, but some people might be interested to try it in a small way on something less complex. This was a very good guide book indeed for dealing with straightforward cases.

He had intended to ask some questions about axial vibration, but most of them had been answered already. In Paper No. 11, he was astonished at the amplitude of the recorded figures of 0·1 and 0·2 inch at the aft and fore end of the crankshaft respectively. Surely they must have been very obvious to the people who were running the ship. He could not imagine himself seeing any part of a crankshaft moving that amount and not being slightly perturbed by it. Insufficient clearances in the propeller aperture were mentioned as one of the major causes. He would be interested to know whether that was a matter that concerned only the shipbuilder. Would it not be comparatively simple in the design stage to eliminate this defect almost completely? A matter of 12 inches ought to do all that was necessary. It seemed to him that one might plug the idea a little harder—to use a borrowed expression—to get the shipbuilders to give the propeller room. He took it that one would not get this trouble as readily in a twin-screw ship, and that might be the answer. The nearer one got the single-screw ship aperture to that of the twin-screw ship the more likely one was to remove one of the major causes of trouble.

He congratulated the authors on Case 3 in Paper No. 11. No one had mentioned it, but he had quite enjoyed reading about the use of the bobbin piece. It showed a remarkable command of English that, without looking at the sketch, one knew exactly what was being described. When he saw the sketch and the sine curve, he thought it was a very clever piece of description.

There was one minor point on the safety angle. Fig. 43 showed a balance weight that he imagined few sea-going engineers would like to see revolving at 116 r.p.m. in close proximity to the ship's side. It was thankfully noted that it was hidden from view in what, it was to be hoped, was a substantial casing. He took it that the object was partly to lessen the shock in case anything did come unstuck! The test on the welded attachment in Fig. 44(b) was of interest and the factor of safety no doubt looked after the effect of fatigue.

MR. BUNYAN thanked Mr. Welch for raising another issue to which he himself had given considerable attention during the past eighteen months. Propeller aperture clearances were an important factor from a consideration of the effect of axial, torsional and hull vibration, which had been borne out by work which had been done on actual ships and on models. With a tight aperture it was possible to produce quite severe hammering in the gearing of geared turbine jobs, and there had been cases where fatigue failures of tailshafts had been directly attributed to the same cause. He hoped that it would be possible before long to determine clearly the minimum aperture clearances which would ensure satisfactory operation.

Mr. Welch's comments on the balance weight were understandable, and the weight in question was fitted with a guard all round, more from a point of view of preventing damage to personnel but no doubt serving as a warning device in the case of displacement of the weight in service. He mentioned that the weight provided a neat solution and a complete cure for what had been a most severe hull vibration problem.

In reply to Mr. Welch's comments regarding the welding-on of balance weights, the question of fatigue failure was carefully considered, and it was felt that the factor of safety was such as to provide a minimum risk.

MR. DOWIE observed that Mr. Welch had tended to minimize the two-thousandth coning on the radial width to the scraper ring grooves. It was not a minor discrepancy. Two-thousandth tapering over the (19 mm.) landing was really 85-thousandths on the diameter at the locating face of the scraper box, which represented quite a substantial error.

MR. YATES said he would like to say a word in defence of his term "mil" which had been treated with such derision. He was, of course, brought up to speak of thousandths of an inch and in common parlance "thou's". That was all very well; but give it to a typist and she put a full stop after it, and it looked very funny. Give it to a printer, and he would not have it at all. He inserted "thousandths of an inch". The sentence, often long enough in the beginning, became even longer. It was twenty years since he had found "mil" in common use in United States laboratories, and its meaning seemed to be clear over there. In good time this country would have to accept whatever became standardized over there, and this might follow suit. He had never found any risk of its being confused with metric systems and he was surprised at Mr. Hildrew who had referred to a mil as a hundredth of a millimetre. He wondered whether this was a local use and not general usage on the Continent. He himself had never come across it. The word "mil" used in an English context was a completely logical abbreviation of "milli-inch", and did not logically signify a hundredth of anything.

He felt there was a tendency to become confused about torsional and axial vibration. He had noted that it was, in fact, as Mr. Bunyan had said, vibration in a coupled system, a system of more than one degree of freedom. But usually such cases could be treated as either torsional or axial vibration, each slightly modified by the presence of the other, but not to the extent of a major change in the frequency. The point he wanted to make here was that a point could be a node for the torsional motion and not a node for the axial motion. He had no doubt that the reverse could also occur.

To take a purely hypothetical case of a simple torsional vibration, suppose the thrust block was rigid, so that it must be a node for any axial vibration. The characteristic feature would be that there was a node somewhere in the shaft. It might or might not be in the thrust block, though there was usually one not very far from it. At that point, the torsional motion was if not zero at least very small. The further one went away from it the bigger the motion became, so that if the propeller was the exciter, the maximum motion was at the propeller.

If the thrust block was not quite rigid, the motion of the propeller caused torque associated with it. The original exciting torque might be only a small percentage—5 or 10 per cent. But at resonance the torque in the shaft might well be ten times as great, and the additional torque must reflect back in a thrust which would be transmitted along the shaft. It did not necessarily compress the shaft appreciably. In other words, the thrust block moved fore and aft, and the shaft went after it.

It would not be fair to call it an axial resonant vibration, though there was some axial motion. It was clear that in Case 1 of Paper No. 11, there was some approach to axial resonance, since the forward end had an amplitude of two-tenths whereas near the thrust block it was only one-tenth. This did not necessarily indicate that resonance was near, but merely that the operating point was somewhere on the flank of an axial resonance but not necessarily close to the resonant point. He hoped Mr. Bunyan would not mind his mentioning this, but he was only trying to put the same thing in different words, in order to help put over a somewhat complicated point. He did not think the thrustmeter would be very effective as a damper, because the movement which these thrustmeters would stand was, he gathered, extremely small. He had never used them, but he thought it was a matter of two or three thousandths of an inch, whereas the displacement of the thrust block under the steady working load was a good deal more than that —a good deal more than most engineers realized. He had never found a case where it was possible to calculate it. He worked with a curve in which one related thrust block stiffness to thrust shaft diameter; and with a reasonable tolerance above and below it the curve held remarkably well for a very wide range of shafts.

He was glad to hear that in the case of the 950° ships there was a case of a gland touching without damage to the rotor. He did not suppose it was possible to tell how severe the touch was but the fact that sparks were observed indicated that it was on the outside of the gland. Was Mr. Baker quite certain that it was a gland and not something else?

MR. BAKER said he was quite certain. In reply to a further question he said about a thousandth was taken off the tip of the strips. They were not spring-backed but were solidly maintained.

MR. YATES said it was encouraging that the gland would stand that much. Even then, one did not know the extent of the rub but it was some evidence of how much the gland approached the shaft. It was hoped that in perhaps twelve months' time a little more would be known quantitatively about the behaviour of rubbing glands. At the moment, they were beginning to run a rig in which there was a shaft, in this case stationary for convenience. The gland was rotated and the shaft was forced into contact with it and distortion was measured after a rub of known duration and known interference, and so on. In this way, it was hoped to learn more about the severity of gland rubs.

MR. C. A. SINCLAIR (Member) pointed out that nothing had been said in Paper No. 3 as to the cause of the original fracture. Information of this kind would constitute a valuable addition to the paper. Something might also be said about the order of deflexion before it occurred. A second disaster might be worse than the first if these other matters were not remedied, and perhaps the vessel might become a casualty at sea. He would seek help as to the cause of the original failure, therefore.

Further, he would seek information as to the benefits to be derived from the method of fastening with screw ferrules and dowels mentioned in Paper No. 5. He was thankful, too, for Commander Baker's help as to the cause, or possible cause of the fracture; for he felt that these matters were important. A good many young engineers were reading the TRANSACTIONS, and they might think of the repair only and not of the possible cause of the difficulty.

In the case described in Paper No. 5 (Fig. 19(b)) the number of dowels that had been fitted would suggest that there was little confidence in the shrunk fits. Why were so many dowels fitted? Was the same operation likely to be necessary again in the repair of another vessel?

With regard to Paper No. 2, he had always felt that the isochronous condition in a governor rendered it quite unusable but the author had fitted a hydraulic governor of the Pickering isochronous type. In what way was the principle of isochronism employed in this governor?

MR. KROEGER said he was unfortunately unable to state the cause of the fracture in the low pressure turbine rotor. This question could best be answered by Mr. Gatewood.

The repair was carried out excellently on the ship *America* by Harland and Wolff, Ltd., at Southampton.

MR. YATES said he was not familiar with the details of the Pickering governor, but there were others like it. They were called isochronous and for long-term variations were very nearly isochronous but not quite. They worked on a speed variation of the order of one-eighth per cent instead of the more normal four per cent or more, but for comparatively rapid speed changes were not isochronous, but had a temporary speed variation of more like the usual four or even eight or ten per cent. Isochronous was more of a trade name, therefore, than a precise technical term.

Correspondence

MR. K. V. TAYLOR, B.Sc.(Eng.) (Graduate) considered that the authors of Paper No. 11 had given a modest account of a number of cases of vibration trouble which had been investigated successfully by Lloyd's Register of Shipping, and whilst the cure was more important than the investigation, considerable effort and thought must have been expended before a cure could be developed. Each one of the cases referred to would be of considerable interest by itself, and it was a pity that details of the investigation had to be mentioned in the laconic phrase, "Records were obtained", etc. The fact that propeller-excited axial resonances could arise from several causes, such as insufficient clearance between blades and hull, cavitation, or slight differences in pitch of the blades, was readily appreciated. However, he would have thought that this vibration would have been confined mainly to the mainshaft and not transmitted through the thrust block to affect the crankshaft. Was it not possible that structural weakness of the block gave rise to the axial impulses being transmitted and that a more rigid block might have sufficed in the first instance?

In Case 4, it was stated that the balance weight might only move if caused by some unforeseen happening such as the striking of the propeller. In the event of this occurring, the balance weight would become "out of phase" with the arrangement shown in Fig. 42, and would this not result in far greater vibration than before? This, no doubt, would soon be remedied, but not before damage had taken place.

MR. LOW, in reply to Mr. Sinclair's question concerning a hydraulic governor of the Pickering isochronous type, wrote that the term "speed droop" was used to denote the change in speed that occurred when an engine was running under no-load condition as compared with the full-load condition. The speed droop mechanism in the governor was adjustable and the setting depended upon the type of operation desired.

Zero speed droop was the position in which the governor gave isochronous regulation and this condition was used in single-engine operation. In twin-engine operation, when it was desired to distribute the load changes uniformly, the speed droop should be adjusted at some value other than zero, usually the same on both governors. The maximum adjustment was normally about 5 per cent, but in certain governors might be as much as 8 per cent.

MR. A. R. GATEWOOD, in a written reply, referred to Mr. Sinclair's request for information concerning the cause of the original crankshaft fracture described in Paper No. 3. In this case it was attributed to misalignment of the crankshaft bearings and for the temporary repair the bearings were adjusted in accordance with the bridge gauge readings.

Mr. McAfee had suggested that the reference in Paper No. 4 to the repair of cracked Diesel engine cylinder heads and liners might require some qualification as he did not feel "Metalock" could be used to correct conditions where material or design was at fault. Mr. Kroeger had pointed out examples of how "Metalock" could be used in certain cases to correct a faulty casting. As regards faulty designs, particularly those where expansion and contraction of the part had not been properly allowed for, his organization had experiences similar to those pointed out by Mr. Baker in that cylinder heads and cylinder liners had given years of successful service after having been repaired by "Metalock". On one ship all of the cylinder heads in operation had been "Metalocked", while a complete complement of new heads had been on board for a number of years but had not been installed. It should be pointed out, however, that it was customary to call "Metalock" repairs temporary until such time as they had proved themselves in service and the owners were willing to have them written off as permanent.

Mr. Sinclair had asked for information as to the probable cause of the low pressure turbine rotor failure (Paper No. 5). In this case torsional calculations as well as measurements on the ship were made and it was not believed that torsional vibration could have been the primary cause of the failure, as sighted in a similar case by Mr. Baker. Evidence was found, however, of cocoa-coloured particles which were a characteristic of fretting corrosion and it was believed that the failure might have started in this manner and then progressed due to normal service loadings. In the permanent repair, the fitted part of the coupling, together with the keyways, were eliminated and the claws machined in the new forged piece.

A question was also raised concerning the benefits of screwed ferrules with internal dowels. This arrangement was used to simulate fitted coupling bolts as they were intended primarily to take only shear while the shrunk-on ring was to take the place of the usual coupling bolt heads and nuts to prevent the subjecting of the ferrules to any bending load in the event of slight misalignment. The number of dowels was not considered excessive because an attempt was being made to effect a repair which would be as near as possible to the strength of the original part.

MR. H. G. ROSS, in a written reply to Mr. Sinclair, said that the fitting of dowels was the usual practice in the construction of Parsons turbine rotors. Dowels were fitted because at the working speed of the rotor the shrink grip for design purposes was only regarded at 60 per cent. If another vessel required a similar repair, the same operation would be applied.

MR. BUNYAN, in reply to Mr. Taylor, wrote that in a paper of this sort, one had to be brief, and details of the measuring and recording techniques and the calculations involved had to be cut to a minimum.

On the matter of the transmission of axial vibration through the thrust, the amount of vibratory coupling between the intermediate shafting and crank shafting depended on the natural frequencies of both systems, and it could occur that vibration could be transmitted from one to the other through a practically rigid thrust block and seating. In the case quoted, it would be observed that the stiffening of the thrust block merely raised the critical r.p.m. from 103 to 108.

Regarding the balance weight shifting, it could be said that the automatic reaction of the engineer on watch would be to reduce the engine revolutions in the presence of heavy hull vibration. As to the question of resultant damage, Mr. Taylor was unduly pessimistic, as it was surprising how severely, and for how long, a ship's hull could be subjected to hull vibration without any signs of distress; which was, after all, understandable, as very large vibration amplitudes would be required to produce hull stresses of a magnitude comparable with those produced in a moderate seaway.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Institute on Tuesday, 9th February 1954

An Ordinary Meeting was held at the Institute on Tuesday, 9th February 1954, at 5.30 p.m. Mr. Stewart Hogg (Chairman of Council) was in the Chair and 106 members and visitors attended.

The symposium of eleven short papers listed below were briefly introduced by Mr. W. Sampson (Vice-President) and a discussion followed in which seven speakers took part. Messrs. J. Dowie (vice A. W. Davis), D. A. Kroeger (vice A. R. Gatewood), H. G. Yates, W. H. Falconer, T. W. Bunyan and B. Hildrew replied to questions and comments which referred to their respective papers.

1. "Excessive Consumption of Lubricating Oil on Doxford Engine", by A. W. Davis, B.Sc. (Member).
2. "Problems of a Geared Diesel Propulsion System", by D. W. Low (Member).
3. "Temporary Repair of a Cracked Diesel Engine Crankshaft", by A. R. Gatewood, S.B. (Member).
4. "Repairs to the Fractured Cast Iron Cylinder of a Beam Air Pump", by A. R. Gatewood, S.B. (Member).
5. "Temporary Repair of a Fractured Low Pressure Turbine Rotor", by A. R. Gatewood, S.B. (Member).
6. "Repair of a Low Pressure Turbine Rotor", by H. G. Ross (Member).
7. "Thermal Straightening of Turbine Rotors", by H. G. Yates, M.A.
8. "High Pressure Turbine Cylinder Casting Repair", by W. H. Falconer.
9. "Turbine Journals: Machining in Place in Wake of Carbon Packing", by E. Byrne, B.E.M.
10. "An Improvized Balancing Operation", by Commander(E) H. E. C. Hims, O.B.E., R.N.(ret.), M.I.Mech.E.
11. "Some Further Vibration Problems Relating to Ships and Their Associated Machinery", by T. W. Bunyan, B.Sc. (Member), and B. Hildrew, M.Sc. (Associate Member).

Local Sections

Merseyside and North Western

Two senior meetings have been held recently by the Merseyside and North Western Section in the Liverpool Engineering Society's Rooms, 24, Dale Street, Liverpool, of which the following reports have been received:—

On Monday, 1st March 1954, Mr. G. Keenan (Associate Member) read his extremely interesting paper entitled "Some Aspects of Mathematics in Relation to Everyday Practice in Marine Engineering", which showed how seemingly intractable problems could be readily solved by relatively simple mathematics. With his very wide experience the author was able to cite many instances of constantly recurring problems where the engineer's judgment must depend on a mathematical approach.

Mr. R. H. Grundy opened a discussion in which many members joined. The vote of thanks proposed by Mr. G. Pickering was passed with enthusiasm.

Mr. T. McLaren, B.Sc., was in the Chair and ninety-eight members, visitors and students attended.

On Monday, 5th April 1954, Mr. James Calderwood, M.Sc. (Vice-President), introduced the symposium "Some Interesting Ship and Machinery Defects, Their Investigation and Cure".

Mr. Calderwood described how the various short papers included in the symposium had been selected with a view to providing a wide range of topics. He then gave a short précis of the papers whose authors were unable to attend. Mr. Calderwood then introduced Messrs. W. H. Falconer, B. Hildrew, M.Sc. (Associate Member), and H. G. Yates, who each spoke briefly on their own particular subjects.

Mr. G. Pickering opened the discussion with a contribution which involved all three authors present. An interesting discussion then ensued.

Mr. L. Baker, D.S.C. (Vice-President), proposed a vote of thanks to Mr. Calderwood and the authors, which was carried with much applause.

Mr. T. McLaren, B.Sc., was in the Chair and 110 members, visitors and students attended.

Scottish

The last general meeting of the 1953-54 session was held at the Caledonian Hotel, Edinburgh, on 24th March 1954.

Mr. D. W. Low, O.B.E. (Chairman of the Section) conducted the proceedings and 144 members and visitors were present to hear an instructive and interesting talk on the Denny-Brown Ship Stabilizer by Sir William Wallace, C.B.E. (Vice-President). In this talk the author reviewed generally the history of active stabilization of ships, using slides and films, also demonstrations with models to illustrate various points.

The talk opened with illustrations of why ships roll, followed by a discussion on the problems associated with ships of various designs, principally those associated with the great variation of metacentric height, and the difficulties of stabilizing ships with a very large G.M. The brief historical survey of various stabilizing devices included the Frahm anti-rolling tank and the Sperry gyro type, with its advantages and disadvantages. Explanation was given of how the Denny-Brown stabilizer had eased the whole problem; whereas the gyro stabilizer must attain the stabilizing power from its own output, the Denny-Brown stabilizer obtains its stabilizing power from the main engines, as obviously the lift derived from the angle of fin is a product of drag and lift, but the advantages of the lift overcome the disadvantages of the drag. Admiralty trial had proved that assuming the same horsepower with the ship rolling unstabilized, the speed drops as compared with the ship running in calm water, and with this new stabilizer it was possible to maintain the same speed as when the ship was rolling freely, and at the same time provide complete comfort for the ship's passengers.

Sir William also demonstrated that, due to the great advantage derived from increased lift, guarantees could be given for roll damping of nearly three times the degrees which were guaranteed in the case of the gyro stabilizers fitted in the *Conti Di Savoia.*

After some discussion, a vote of thanks proposed by Mr. R. Hunter Miller was enthusiastically accorded.

This occasion included a visit to Brown Brothers' Works which afforded interest and instruction to some 120 members, many of whom had travelled from Glasgow by special buses.

South Wales

The last illustrated lecture of the 1953-54 winter session, entitled "A Review of Ship Propulsion Methods", was held on Monday, 15th March 1954, at the South Wales Institute of Engineers, Cardiff.

Mr. J. H. Evans, M.B.E. (Chairman of the South Wales Section), presided and he was supported by Mr. David Skae (Vice-President), Mr. J. E. Church (Member of the Council of the Institute), and Mr. W. Gracey (Honorary Secretary); sixty members and marine engineering students attended the meeting.

Mr. H. S. W. Jones (Vice-Chairman of the Section) read his paper in a masterly manner, the subject was of exceptional value and interest, was well received and was followed by a full discussion.

Mr. Church proposed a vote of thanks, which was seconded by Mr. Skae, and Mr. J. Wormald (Member) proposed a vote of thanks to the Chairman.

Sydney

The Annual General Meeting of the Sydney Section was held at Science House, Gloucester Street, Sydney, on Wednesday, 3rd March 1954, at 8.0 p.m. Mr. H. A. Garnett (Local Vice-President) was in the Chair and sixty-one members and guests were present.

The adoption of the Annual Report and Balance Sheet was proposed by the Honorary Secretary, seconded by the Honorary Treasurer and carried unanimously. The following members were elected as office bearers for 1954:—

Chairman: H. A. Garnett (Local Vice-President)
Honorary Secretary: Eng. Capt. G. I. D. Hutcheson, R.A.N.(ret.) (Member)
Honorary Treasurer: A. N. Grieves (Member)
Committee: W. G. C. Butcher, B. P. Fielden, H. W. Lees, J. Munro, J. C. Sundercombe, and H. P. Weymouth (Members).

A vote of thanks to the Honorary Secretary and Honorary Treasurer was proposed by Mr. H. W. Lees, seconded by Mr. W. G. C. Butcher, and carried with acclamation.

After the business meeting, Mr. K. Longes (Member) presented his paper entitled "Automatic Control Equipment for Application to Marine Boilers"; the paper was very fully illustrated by lantern slides. A spirited discussion followed, to which contributions were made by Messrs. W. G. C. Butcher, D. N. Findlay, A. V. Franklin, J. Munro, H. P. Weymouth, E. L. Buls, H. W. Lees, and G. E. Arundel. A vote of thanks to the author was proposed by Mr. F. J. Ward and carried by acclamation.

A most successful visit to the Peninsular and Oriental Steam Navigation Company's new ship *Arcadia,* was paid by members of the Sydney Section recently. About forty members were received by the Chief Engineer, Mr. J. Thomson, shown round in small groups by the engineer officers of the ship, and afterwards entertained very pleasantly. The visitors were most impressed by the ship and were most grateful for the excellent arrangements made by the company and the ship's officers.

West Midlands

The First Annual Dinner of the West Midlands Section was held at the Imperial Hotel, Birmingham, on Thursday, 18th March 1954, there being fifty-seven members and guests present.

The Chairman of the Section, Mr. H. E. Upton, O.B.E., was in the Chair, the principal guest being Sir Gilmour Jenkins, K.C.B., K.B.E., M.C. (President), who received all members and guests. Other prominent guests included Mr. Stewart Hogg (Chairman of Council), Captain(E) F. E. Lefroy, R.N., and Mr. J. Stuart Robinson, M.A. (Secretary of the Institute).

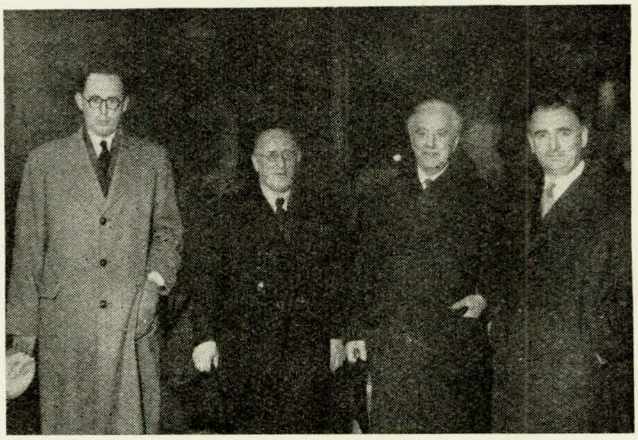
After an excellent dinner, the toast to the Queen was proposed by the Chairman. Mr. Upton then proposed "Our President", and Sir Gilmour Jenkins responded.

The Institute of Marine Engineers was proposed by Captain Lefroy, and Mr. Hogg responded.

Mr. G. A. Plummer (Vice-Chairman) remarked on the success of the Local Section in its first year.

Between the speeches, the company were entertained by George Hardeman, baritone, and Linda Vaughan, soprano.

The presence of the President and the other guests contributed much to the success of the evening, and the officials and members of the Section wished to express their appreciation.



Visit to Birmingham
Mr. J. Stuart Robinson (Secretary), Mr. Stewart Hogg (Chairman of Council), Sir Gilmour Jenkins (President), and Mr. H. E. Upton (Chairman of the West Midlands Section)

A General Meeting of the Section was held at The Imperial Hotel, Birmingham, at 7.0 p.m. on Thursday, 8th April 1954. Mr. G. A. Plummer (Vice-Chairman) was in the Chair, and eighty members and visitors were present.

Mr. H. E. Upton, O.B.E. (Member) presented his paper entitled "Heat Exchangers for Internal Combustion Engines". The paper dealt briefly with the adoption of fresh water cooling in oil engines, hence the need for coolers, and went on in some detail to analyse various systems employed, including the pressurized coolant system. The paper continued by describing various types of heat exchangers and dealt with design considerations. A description of the charge air cooler followed and the paper concluded with a description of automatic temperature control valves.

Seven members took part in the ensuing discussion. The Chairman thanked Mr. Upton for his excellent paper, and the meeting closed at 9.15 p.m.

Junior Section

Merseyside and North Western

Bolton

A meeting was held at Bolton Technical College on the afternoon of 27th February 1954, when Mr. R. S. Hogg gave a lecture on "The Launching of Ships".

The Chair was taken by Mr. A. J. Jenkinson, M.A. Mr. Hogg's audience included forty engineer cadets, whose interest in naval architecture in general and launching in particular was shown in the discussion which followed the lecture.

Liverpool

A Junior Lecture was held at Riversdale Technical College, Liverpool, on Wednesday, 3rd February 1954, when a lecture entitled "The Marine Diesel Engine" was given by Mr. A. G. Arnold (Member) to the students studying at the college under the new scheme for training marine engineers. Mr. A. R. Kinsman, Principal of the College, took the Chair, and the Institute was represented by Mr. G. Kenworthy-Neale (Associate Member).

A complete survey of Diesel development, illustrated by slides and an excellent film, was made by Mr. Arnold. Particular reference was made to indicator diagrams and their telltale characteristics, and to some details of the Archaouloff system of fuel injection.

Question time provided opportunity for a lively discussion, and the vote of thanks to Mr. Arnold proposed by Mr. K. Waters, Chairman of the Students' Union (Student Member), and seconded by Mr. R. Chadburn (Student Member), was passed with enthusiasm by the eighty members of the college and fourteen visitors.

South East Essex

A Junior Lecture was given at the South East Essex Technical College on Wednesday, 24th March 1954, when Mr. J. C. Grey, B.Sc., A.M.I.Mech.E., A.I.Lg., gave a lecture on "Gas Turbines". Mr. F. Heathcoat, M.Sc., Ph.D. (Principal of the College), was in the Chair, and there was an attendance of about forty, the majority being students enrolled at the College in the classes for the Higher National Certificate in Mechanical Engineering.

The lecture, in which these students appeared to be absorbingly interested, was followed by a discussion on a high level which was not so prolonged as the audience would have wished because of the lateness of the hour.

Mr. R. H. Jones (Member) represented the Institute at the meeting and gave particulars of the qualifications required for admittance to membership of the Institute in the student grades and the benefits offered to those admitted.

Student Lecture

85, The Minories

A meeting was held on Monday, 22nd March 1954, at 6.30 p.m., at the Institute when the Esso film entitled "The Basic Principles of Lubrication" was shown to over seventy junior members. The Chairman, Mr. F. D. Clark (Convener of the Junior Section Committee) introduced Mr. R. G. Sayer, B.Sc. (Eng.), who said that the film was divided into four parts:—

1. Factors affecting friction.
2. Boundary lubrication.
3. Extreme pressure lubricants.
4. Hypoid gear lubricants.

After the film had been shown, Mr. Sayer answered a number of questions on various aspects of lubrication. Following a vote of thanks, proposed by the Chairman and carried with acclamation, Mr. Sayer said that he had recently shown the same film to a well-known engineering society but was asked only three questions, none of which were up to the standard of any he had been asked on this occasion!

Membership Elections

Elected 5th April 1954

MEMBERS

Norman Blakelock
Stanley Richard Eastlake
Andrew Neill Edgar
Frank Leveson George, Cdr.(E), R.A.N.
Samuel St. John George Griffin
Arthur Thomas Hogan
Edwin Knewstubb
John Smith Gardner Mackay
Douglas Charles McLarty
Cecil Rupert Manasseh, Lieut.-Cdr.(E), R.N.
James Henderson Mortimer
Harold Martin Munro
Alan Norris
Joseph Pigdon
Frank William Roberts
John Rogerson
James Simpson Smith
Thomas George William Watson

ASSOCIATE MEMBERS

Jamil Ahmed, Lieut.(E), R.P.N.
Terence Devine

ASSOCIATES

Harry Rees Bates
Herbert Waltham Bennett
George Ernest Derek Bogle
Howard Campbell
Daniel Collings
Brian Donald Collins
William Craig
Edward Ronald D'Alessio
John Miles Duckworth
Alexander Haddow
Ernest Francis Henry Hemming, Lieut.(E), R.N.
Thomas Amos Lamb
Jack Leicester
Jeremiah John Lydon
Ibrahim Roger Maari
Harry Megson, Lieut.(E), R.N.
Maurice Patrick Morel
Sidney Offord
Gilbert Harold Vincent Page
Peter Stanley Powell
William Cameron Rouse
Antonio Marcelino Miguel Saldanha
Pinhas Ellis Sassoon
Raymond Stephen Shellard
William Robert Shorten
Richard Hartley Simpson
John S. Skelton
Demetrios Paul Spathis
Robert Stobie
William Gwyn Webbern
Colin Andrew Wedderburn
Arthur Edward Westbury
Osmond Wisbach

GRADUATES

Reuben Fraser McLean
Brooking Young

STUDENTS

Abdul Karim Maki Al-Kamil
John Russell Brown
Richard Howard Chadburn
William John Coppard
Montague Neil De La Harpe
Gerald John Oxenbury
William Smiles
Barrie Glenn Stead

PROBATIONER STUDENTS

Richard Vincent Clarke
Terence Berkeley Cook
Colin Hendry
John Laurence Hutchinson
Roger Nicholas Jackson
Alistair Johnston
Oliver Edmund Newman
Donald Lyon Dean Parry
Geoffrey Graham Sarjantson
Thomas Tarpey
Peter Roland Whitehead

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER

Samy Aly El-Rashidy, Lieut.-Cdr.(E), E.N.F.

TRANSFER FROM ASSOCIATE TO MEMBER

Neville Alfred Dawson
Indreswar Gogoi
Cyril George Nightingale
John Edward Radcliffe
Gilbert Kennedy Sanson
Charles Tye

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Albert Frenkel

TRANSFER FROM GRADUATE TO MEMBER
Ernest Sheppard Baker, Lieut.-Cdr.(E), R.C.N.
Robert Saint George Stephens, Lieut.-Cdr.(E), R.C.N.

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Ejaz Ahmad, Constr. Lieut., R.P.N.
John Ashby McDonald
Kenneth Victor Taylor, B.Sc.

TRANSFER FROM STUDENT TO ASSOCIATE
Reginald Charles Crouch, Jnr.

TRANSFER FROM STUDENT TO GRADUATE
Walter Murray Simpson

OBITUARY

JOHN PERCY HALL (Member 3449) was born in 1882. He was educated at Caius College, Cambridge, from which he graduated, being awarded an M.A. degree. The rest of his life was spent in the service of J. P. Hall and Sons, Ltd., pump manufacturers, the company founded by his father. For many years he was a partner and he had been a director since 1953, when the business was converted to a public company. He was a member of the Institution of Naval Architects and had been a Member of the Institute since 1918.

CHARLES MCDONALD (Associate 10158) was born in 1919. His apprenticeship was served with A. Hall and Company from 1936-41, when he joined the Lago Shipping Co., Ltd., as fourth engineer, serving with them for several years and being promoted second engineer in 1944. He obtained a First Class Ministry of Transport Steam Certificate in 1947. From May 1951 until his death in hospital on 2nd August 1952, he was second engineer with the Asiatic Steam Navigation Co., Ltd. Mr. McDonald was elected an Associate of the Institute in 1944.

MARK ROBINSON (Member 2504) was born in 1872. He served an apprenticeship of two years with the Wallsend Slipway and Engineering Co., Ltd., and three years with Harland and Wolff, Ltd., Belfast, and he continued in the service of the latter company from 1892-1900 as a draughtsman. He then worked as a draughtsman for Vickers, Son and Maxim for a year. He went to sea with Lamport and Holt and obtained a First Class Board of Trade Steam Certificate, sailing with the company for many years thereafter as second engineer. From 1930-39 he sailed on West African coasters as second and chief engineer and from 1939-47 he was employed by Cammell Laird and Co., Ltd., as a fitter. He died on 10th March 1954. He had been a Member of the Institute since 1911.

GANES SPINKS (Member 14004) died suddenly on 4th February 1954. He was born in 1906 at Methil, Fifeshire, and served an apprenticeship there with William Ogilvie and Co., Ltd., from 1921-26. He then spent two years at sea as fourth engineer with the Ben Line Steamers, and from 1928-30 he was third engineer with Strick Lines. From 1930-31 he was second engineer with M. Taylor and Company. In 1931 he obtained a First Class Board of Trade Steam Certificate, and a Motor Endorsement in 1952. In 1933 he served for a few months as senior fifth engineer with the Red Star Line before taking up a partnership in a ship repairing business in Buenos Aires, where he remained until 1941. From 1941-46 he served in the Royal Naval Reserve, first as Lieutenant(E) and later as Lieutenant Commander(E). After the war he was engaged as second engineer with the Maritime Shipping and Trading Co., Ltd., and then as chief engineer by Continental Lines, Ltd.; Mr. Spinks had just left this company in January 1954 and returned to Methil, where he was to go into partnership with his brother in a small engineering works, when he was taken ill and died within a few days.

Mr. Spinks was elected a Member of the Institute in 1952.

JAMES YOUNG (Member 5289) was born in 1884. He was educated at the John Neilson Institute, Paisley, and served an apprenticeship at the engineering works of Thomas Shanks and Co., Ltd., Johnstone, Renfrewshire. He went to sea from 1906-10, when he obtained a First Class Board of Trade Steam Certificate, and then sailed as second engineer, later as chief, in the *Galavale* until 1914. From 1915-17 he was second engineer of the Gow Harrison ship *Vermont,* when he joined the Australian Commonwealth Line, sailing as chief engineer of the *Austalcrag, Largs Bay, Boorara* and *Hobsons Bay,* between 1917 and 1930. In 1922 he was second engineer in the *Esperance Bay* on her maiden voyage and served in her for a year. From 1930 until his retirement in 1950, Mr. Young was chief engineer of the *Mataroa.* He died on 15th January 1954. He had been a Member of the Institute since 1925.