The Trials and Operation of the Gas Turbine Ship *Auris*

R. M. DUGGAN, M.A. (Associate Member)* and A. T. O. HOWELL (Associate Member)†

The paper is a record of the extensive trials carried out on the first British ocean-going gas turbine merchant ship.

It is divided into three parts and deals with the basin trials, sea trials and commercial service.

The machinery was first erected in the ship and all test bed running of this prototype machinery was carried out during the basin trials. In this period many new problems and delays were encountered but were successfully overcome. Some details are given of these difficulties. The sea trials were designed to ensure that the ship was safe to go to sea and were very successful.

After entering commercial service the ship crossed the Atlantic to Curaçao then traded in North West Europe for six months giving a wide variety of climatic conditions and a vigorous testing of the manœuvrability of the machinery under winter conditions in a large variety of ports.

The failure of some compressor blades while the ship was in service is discussed and some reasons are given for the withdrawal of the ship from commercial service.

INTRODUCTION

The development of a new marine engine is usually a slow process over many years and invariably involves a large number of engines and many different manufacturers.

The three *Auris* installations took place over ten years and, although this may appear to be a long time, when she entered commercial service in 1959, she was propelled by a completely new engine, which could compete well with a steam turbine plant of similar power.

This unit was virtually designed in the period 1953-54 and there is no doubt that it is only one stage in the eventual development of marine gas turbine propulsion. Any new development takes a long time but it was unfortunate that for quite extraneous reasons the *Auris* did not re-enter commercial service until August 1959.

The co-operation between the shipyard, engine builders and designers was a remarkable achievement. The engine was never tested ashore and ran successfully from the moment of the first start. The troubles experienced show that the *Auris* had more than her fair share of ill luck, as many of them were of the type that could occur to any ship and were not related to the new main engine.

The rather late decision to install the single 5,500 b.h.p. gas turbine engine in the *Auris* meant that certain design features had to be accepted which would otherwise have been avoided. Examples of this are the left handed propeller, the distance between shaft centre lines and the retention of a Scotch cylindrical boiler. Although the decision was taken that the *Auris* unit would be considered experimental, as against the commercial proposition in *Hemisinus*, an attempt was made to reduce the capital cost as much as possible. A number of the old auxiliaries and valves were thus incorporated into the new machinery. This proved to be a mistake, because the auxiliary plant was unnecessarily complicated and the overhaul trouble experienced and restriction on design far outweighed the hoped for saving in cost.

A gas turbine plant itself only requires fuel oil, lubricating oil and circulating water (with an intercooled cycle only). The full advantage of this fact was perhaps lost in the *Auris* plant, but even so the machinery proved very flexible. A careful check was kept on all machinery weights and this clearly illustrates the unnecessary weight of the auxiliaries and associated pipework.

The decision was taken early that the ship's engineers would operate the engine from the start. The engine builders and designers of course had ultimate control and acted as advisors. The engineers were not specially selected and none of those who eventually sailed with the ship had ever seen a large gas turbine previous to joining the ship. Special training was not possible, but a series of lectures was given to them by each of the engine designers and builders on his own subject, each lecture being recorded. The value of this was limited as of course no one had the experience that eventually became established practice.

A paper describing the testing of a new design of machinery must inherently include a number of troubles which occurred and it is hoped that, whilst these are included here to give a true record, they will not give the impression of a chronology of failure.

There were three distinct phases of tests which the paper will describe. Firstly basin trials, followed by sea trials and finally operational service.

HISTORY

The early history of the *Auris* is well known and her start with four Diesel engines and later conversion to part gas turbine and Diesel electric propulsion has been adequately described elsewhere[(1)].

The success of the first gas turbine is unquestioned, for it operated for 20,510 hours of which 6,649 hours were on residual fuel oil. What is perhaps not so well known, is that the four-stroke Diesel engines produced almost every known defect and trouble including seven crankcase explosions, many piston seizures, excessive liner and ring wear, vibration trouble, endless bearing failures and control difficulties. On several occasions this gas turbine was responsible for the safe return of the ship and her cargo.

* Research Engineer, Shell Tankers Ltd., London.

† Permanent Chief Engineer, s.s. *Zaphon*, Shell Tankers Ltd., London.

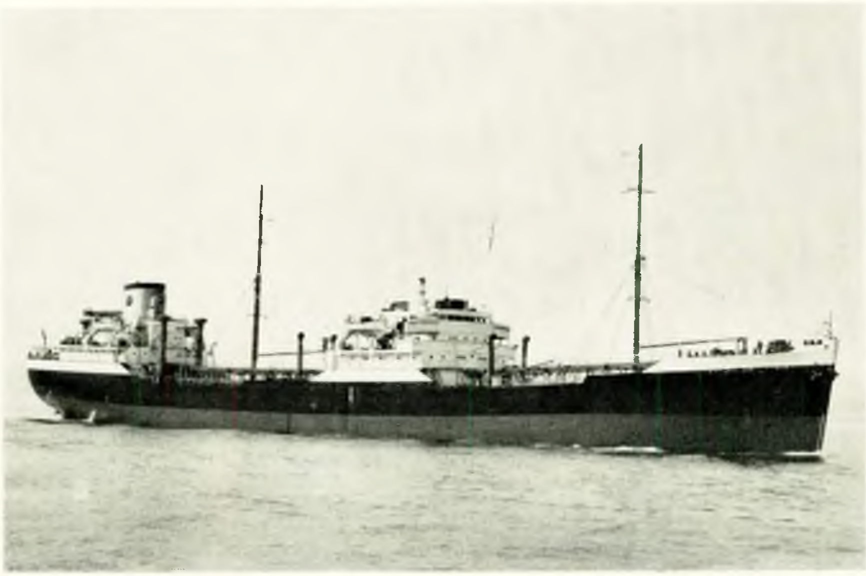


FIG. 1—*Ship profile—g.t.s.* Auris

After the original gas turbine machinery had been in operation for about a year an order was placed for the installation of larger twin propulsion units for a new 18,000 ton d.w. tanker *Hemisinus*. This ship was to be a commercial unit and would thus have to be competitive with a large class of sister ships with geared steam propulsion.

Drawing work was in an advanced stage in 1955 when it was decided to cancel this project and install one of the gas turbine units in *Auris*. Work had actually started on the turbines and compressors and this was later to have some important repercussions on the *Auris*.

The reasons for this rather late, but fundamental decision are briefly as follows:

1) control of the electric propulsion system was proving difficult, necessitating lower efficiency geared induction motors;
2) this would result in higher initial and operating costs and complexity;
3) by adopting geared transmission it was hoped to eliminate at least one intermediate stage in the development of marine gas turbine propulsion;
4) as mentioned above the three remaining Diesel engines installed in *Auris* were in 1955 proving even more difficult to maintain than in previous years. Although, the ship was only seven years old she required some new machinery.
5) the installation of a single gas turbine unit in *Auris* could be considered experimental as against the intended commercial proposition in *Hemisinus*.

The ship arrived at Birkenhead in October 1956, and the entire engine room was emptied in the following two months. The structure was then modified to receive the new machinery, which necessitated the removal of the screen bulkhead, lengthening of the forward end by two frame spaces, a new all welded bulkhead and strengthening and raising of the upper portion to carry the weight of the heat exchanger. What had been a pre-war profile, now became a ship with a pleasant appearance (Fig. 1).

Production of the main engine was unfortunately retarded by manufacturing difficulties and labour disputes so that installation was not completed until July 1958.

As no shop testing was carried out, a carefully drawn up testing programme was devised so that every item had to be proved successfully before the ship could be risked at sea or accepted for service. Subsequent results showed the value of this programme although at times there were lengthy interruptions. For instance the failure of the astern converter in August 1958, stopped all tests until February 1959; this type of delay also produced maintenance difficulties, particularly with electrical equipment, the starting turbine and the main thrust block.

The ship left for sea trials in April 1959, but once again labour disputes held up her entry into service until August 1959. The successful operation in service was limited to six months, for the ship is now laid up in the River Blackwater for commercial rather than technical reasons. It is hoped that in history it will be appreciated that the *Auris* hull is virtually pre-1939 design. During the three years in which the conversion took place, the size of tankers leapt from 28,000 to 65,000 tons d.w. or more, and even an 18,000 ton ship was considered small.

Description of Machinery

The new machinery arrangement has been adequately described elsewhere[2] and as the purpose of this paper is to discuss the tests and operational service, only a brief description of the machinery is included for convenience.

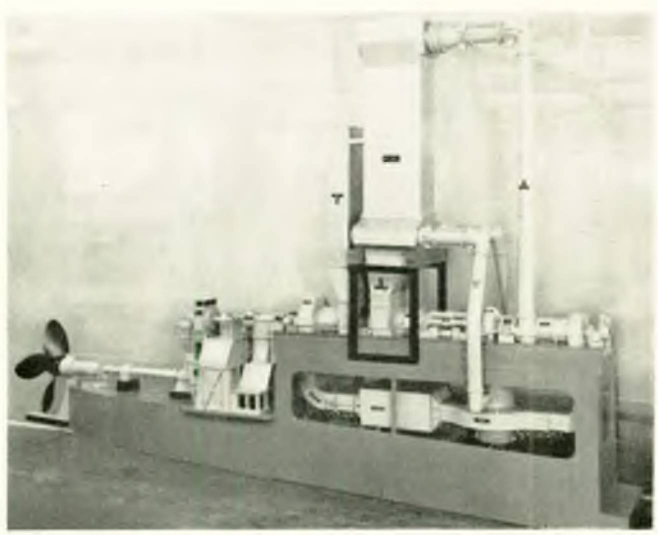


FIG. 2—*Scale model*

Fig. 2 shows a photograph of a scale model of the main engine and transmission.

The single left hand propeller is of the same diameter as the original, but with increased pitch and developed area and is fitted in the same aperture. It is attached to the original propeller shaft and intermediate shaft, but the existing thrust block was modified for the increased load.

The 32 : 1 double reduction double helical gearbox is of conventional design except for the primary line which incorporates three hydraulic components and a friction clutch. One hydraulic coupling transmitted power ahead, one (known as the astern converter) astern and the third drove a 550 kW alternator for cargo pumping.

A small steam turbine of 450 maximum horse power was also attached to the gearbox in case of a complete failure of the main engine at sea; saturated steam was supplied from a retained auxiliary Scotch boiler.

The gas turbine is a two shaft open cycle unit with inter-cooling between the H.P. and L.P. compressors and a vertical tubular heat exchanger preheating the high pressure air before entry to the single vertically mounted combustion chamber. A waste heat boiler is installed in the funnel. The five stage H.P. turbine drives the 16 stage H.P. compressor and the eight stage L.P. turbine drives the 12 stage L.P. compressor and propeller. Air is drawn normally from the atmosphere through a spray chamber at boat deck level. By a system of doors the engineer on watch has a choice of intake from the leeward side or from the engine room.

The air intake duct forms a straight through silencer to reduce compressor blade noise.

The refractory lined combustion chamber although attached directly to the H.P. turbine is counterbalanced to prevent any load being imposed on the turbine casing. By firing vertically upwards the risk of fractured refractory entering the turbine blading is considerably reduced, but the necessary space had to be cut out of the ship's double bottom tanks.

The long narrow design of the engine, whilst inconvenient to the layout draughtsman, enabled considerable saving to be made in duct losses, particularly between the turbines.

The four rotary components were mounted on two frames, but the heat exchanger, forced circulation waste heat boiler, and intercooler were directly mounted on the ship's structure.

Many new and even now advanced techniques were incorporated into the mechanical design to eliminate such troubles as casing distortion, sideways thrust on keys and expansion. Both working and spare H.P. turbines and the H.P. compressor had hinged casings for ease of examination.

A built in additive system was incorporated to combat ash deposition and a complete spare H.P. turbine with different blade materials was carried, to enable fuel testing to be carried out with the maximum benefit.

The engine room auxiliaries were all 440 volt 60 cycle a.c. and supplied by a 200 kW steam turbo-alternator but the remainder of the ship retained the original 110 volt d.c. system supplied through germanium rectifiers. Lighting of the engine room was of a high level and entirely fluorescent.

The stand-by to the steam turbo-alternator was originally intended to be a 125 kW gas turbo-alternator unit but this failed to give its rated power on the test bed ashore. Its place was taken by a rather massive 200 kW Diesel alternator unit.

The ventilation proved vastly superior to the original installation as the four 22,000 c.f.m. fans were required to prevent any reduction in engine room pressure when the gas turbine was taking air from the machinery space.

Considerable re-arrangement of the bunkers was necessary due to the lubricating oil sump tank, the combustion chamber recess, and the extension of the centre of the engine room through the original cross bunker. With the new arrangement the ship has a steaming range of 10,000 miles.

Instrumentation.

To test a prototype engine ashore not only requires adequate space for a long period, but also extensive instrumentation, some of which must be repeated on the ship installation. Time for the installation in the ship is reduced, and erection ashore is easier and cleaner, but it is a costly method.

It was decided that the *Auris* machinery would be erected in the ship and that test bed instrumentation would be permanently fitted.

Fig. 3 is a diagrammatic arrangement of the points of measurement.

Very careful consideration was given to accuracy, type, mounting and installation of instruments. Every care was taken to adopt the instrument manufacturers' recommendations and this policy was well rewarded in practice. At each point of measurement in the gas cycle, one pressure and one temperature was required. At certain positions it was necessary to fit two instruments, either because of the size of the duct, a typical example being the L.P. turbine outlet, or due to the importance of the position such as the crossover between the turbines. All gas turbine instruments were located near the control panel and all gearbox instruments near to, but not on, the gearbox.

For temperature measurement each point was fitted with 6in. dial mercury in steel thermometers of one manufacture. Experience on the first gas turbine had shown that even when the unsheathed bulbs were exposed to the vanadium compounds at 650 deg. C. (1,202 deg. F.), they were satisfactory with quick response. All temperature instruments were calibrated on the centigrade scale for:

a) the engine performance was checked and the basic design was in centigrade units.
b) instruments have a more open scale and the divisions are more satisfactory.
c) a wider range of instruments are readily available.

To assist the ship's engineers initially the dials were coloured red above the normal limits. Anticipated difficulty of engineers appreciating centrigrade temperatures proved quite unfounded.

For turbine bearing temperatures, edgewise mercury in steel instruments was used and mounted so that any abnormal reading was immediately visible by glancing at the panel. The H.P. turbine bearing instruments also contained an alarm contact, which proved to be of weak design.

All capilliary tubes were clipped along cable trays with rubber between the tube and tray. The instruments themselves were resiliently mounted in the panel.

The main gearbox temperature instrument was an existing multi-point thermocouple unit used on the previous alternator installation. In spite of resilient mounting this gave considerable trouble.

The number of temperature instruments basically required for testing would clearly tend to confuse a watch keeping engineer. A six point recorder was therefore installed for help in the basic running of the set.

The temperatures recorded were:

1) H.P. turbine inlet.
2) H.P. turbine inlet flange metal.
3) H.P. turbine outlet.
4) L.P. turbine outlet.
5) H.P. turbine/combustion chamber weld.
6) L.P. turbine outlet, H.P. compressor outlet or engine room temperature as required.

This instrument proved most useful for starting, warming up and general operation. It was sufficiently sensitive to record the effect of bad steering and shallow water.

Pressure gauges were also of one manufacture and resiliently mounted with individual shut off needle valves. All piping was of steel and fitted with water traps where necessary. Fuel pressure gauges were originally fitted with snifters to prevent damage when the fuel was tripped, but these proved unsatisfactory and they were removed.

Differential pressure gauges were tried, but these proved too sensitive. Six foot mercury and water manometers were therefore installed in parallel, and provided care was taken it was found possible to use these at sea under normal conditions. Mercury and water traps were fitted to each gauge, and each

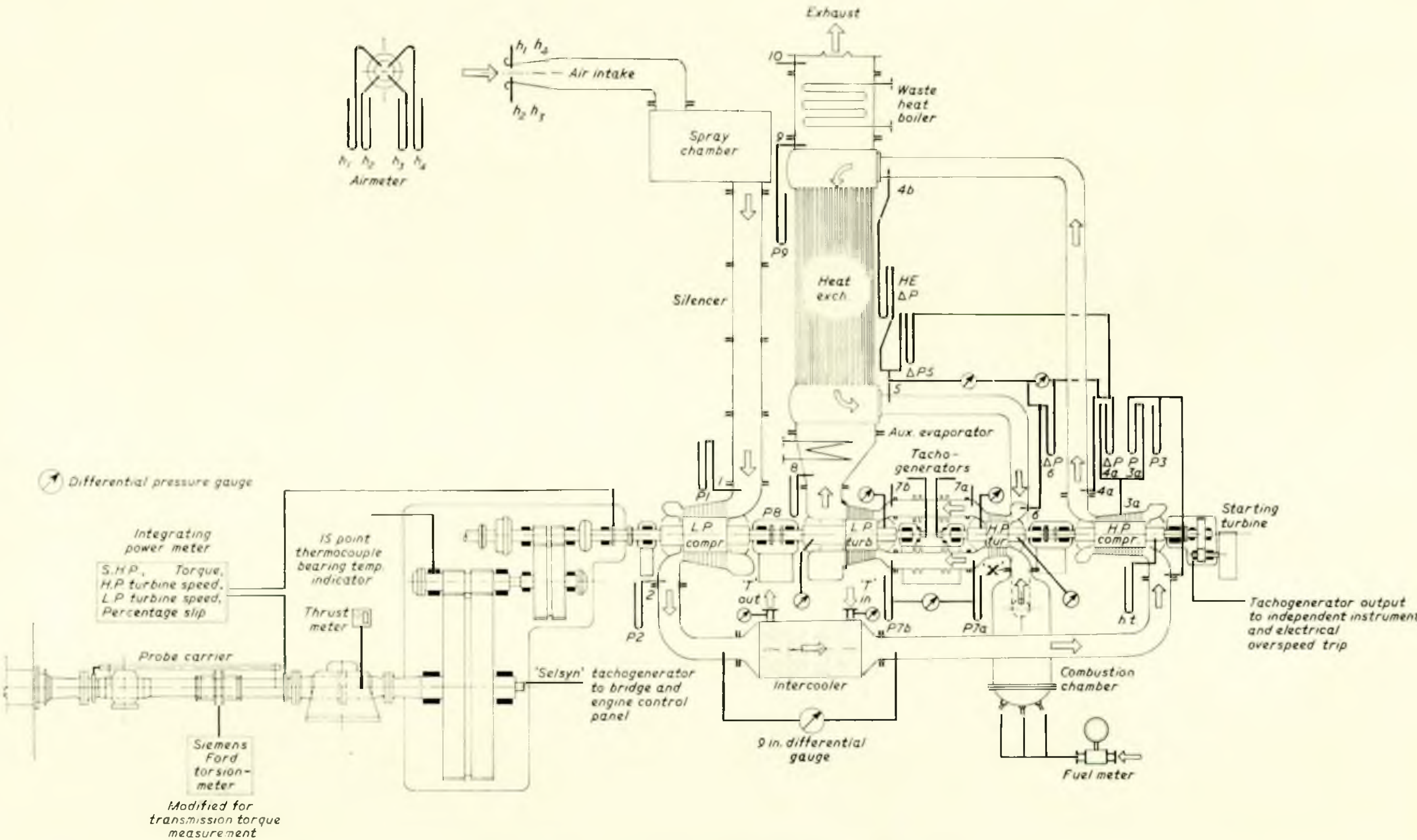


FIG. 3—*Diagrammatic arrangement of measuring points*

Note:

1. Gold Slide Fortin Barometer fitted on bridge.
2. Gas Turbine Tachogenerator outputs led to:
   a) Independent meter
   b) Power meter
   c) Electrical overspeed trip
3. Alcohol in glass thermometers fitted at measuring points T_1 and T_3.
4. 6in. Mercury in steel dial thermometers fitted at T_2, T_3, T_4, T_5, T_6, T_{7b}, T_8, T_{8b}, T_9, T_{10}, intercooler water inlet and outlet lubricating oil to gas turbine bearings. Instruments calibrated for difference in vertical head between bulb and dial where necessary.
5. 5 Point Honeywell Brown recorder connected to chromel alumel thermocouples at T_6, T_{7b}, T_8, "X" and H.P. turbine flange.
6. Edgewise pattern distance reading thermometers fitted to all gas turbine bearings.
7. Two pyrometer indicators used for measuring T_6 port and T_6 starboard. Average of two thermocouples indicated by each instrument.
8. Temperature alarms fitted to both H.P. turbine bearings, L.P. turbine inlet bearing and T_9.

length of pipe was carefully pressure tested to eliminate the smallest leak; a point of vital importance when using a differential manometer. This seemingly minor job took six men three days.

For power measurement the Siemens Ford torsionmeter was refitted, and in addition a Pametrada Integrating Power meter measuring over a 12ft. 10$\frac{1}{16}$in. length of shaft. This latter instrument was virtually a prototype, but proved most useful because it also accurately measured H.P. turbine speed, L.P. turbine speed, percentage slip between quill shaft and pinion, shaft horse power and torque.

For fuel measurement a calibrated V.A.F. type AHD high pressure displacement meter was fitted between the heaters and burners. In addition every fuel tank was twice checked against a calibrated water meter and pneumacator gauges.

Air mass flow was measured by a calibrated Venturi meter mounted horizontally on the boat deck. The two intake doors were blanked off and the engine room intake doors sealed, so that air could only be drawn through the Venturi into the spray separation chamber.

Another prototype instrument which was to prove invaluable was the capacitance clearance monitor (Fig. 4). An earlier variety had been tried in 1953-54 on the first H.P. gas turbine. This second unit contained a number of improvements, and enabled turbine and compressor blade radial clearances to be read at any instant. The 36 monitors were fitted top, bottom and horizontal at the inlet and outlet of each of the four main components, and also in the centre of the H.P. compressor. The vast quantities of condensed fresh water which were produced by the intercooler in service occasionally upset the H.P. compressor probes, and rain interfered with the L.P. compressor inlet probes.

The original installation of the special cable for this instrument was difficult owing to the need to have a means of disconnecting the cable to a turbine or compressor casing, when it was opened. The accuracy of this instrument later became unquestioned and a check that was carried out will be described in the appropriate place.

FIG. 4—*Capacitance clearance monitor*

The only instruments which completely failed in their duty were the three calibrated orifice flow gauges in the lubricating oil supply to the gas turbine, gearbox ring main, and quill shaft. The instruments were however most useful for removing entrained air from the respective systems.

The remaining instruments were of the conventional types such as Selsyn r.p.m. indicators and alarms.

For special indicator lights intermittent flashing switches proved most successful.

Close examination of blading in all components was carried out as on the original turbine by means of an introscope. In the new installation however, special and more extensive provision was made for this purpose.

Owing to the complexity and importance of the instrumentation a special instrument electrical control panel was installed.

The trouble taken with the instrumentation was rewarded by the wealth of accurate data obtained and the immediate determination of an unusual situation.

PART I

BASIN TRIALS

Testing of the main propulsion unit of the g.t.s. *Auris* commenced on the 10th July 1958, when the unit was operated for nearly three hours. The purpose of this run was to establish that the engine would start, and also to find out what peak starting temperature would be reached. To minimize this effect the centre burner only was used, and the H.P. turbine was run at 3,480 r.p.m. to enable the gas tightness of casings and ducting and pressure gauge piping to be checked. In addition information was required about thermal expansion and blade radial clearances during starting, running and after shut-down.

The turbine did in fact start successfully the first time and the temperature recorder showed a short peak of 700 deg. C. at the H.P. turbine inlet. After a few starts the technique improved and 700 deg. C. was later considered a high peak. During this first start the electronic clearance monitor worked satisfactorily and indicated a decrease in the H.P. turbine radial clearance from 0·070in. to 0·030in. during the starting peak period. This reduction gradually restored to normal as the casing warmed up and the inlet temperature steadied at 295 deg. C.

The value of this instrument in the early stages was immense. When the turbine was first operated above 600 deg. C. the monitor suddenly showed a violent decrease in radial clearance on one side. The cause was quickly found to be insufficient clearance for expansion in way of a heat shield. Without this information serious blade damage could have resulted.

After the initial run, testing followed in progressive stages of first mechanically proving the particular item, and then performance testing in case the information was lost by a fault or breakdown at a later stage. This practice was well rewarded, and for convenience some of the results are enumerated below under separate headings. Clearly it was not always possible or convenient to carry out one test at a time and some of the results were obtained simultaneously.

MECHANICAL TESTING OF MAIN GAS TURBINE UNIT.

After all the known defects experienced on the first run had been rectified, the unit was restarted and the H.P. turbine speed allowed to settle at 3,000 r.p.m. "No load" performance readings were taken at this condition and then at progressively increasing speeds of 500 r.p.m. intervals. To ensure accurate performance readings the turbine was always run at each speed for at least two hours, and then all gauges read simultaneously; usually a set being obtained in less than ten minutes. This practice was continued on all basin, sea, and operational performance readings.

It was during this first series of performance readings that the first unexpected mechanical difficulty arose. All the clearance monitor plugs were at this period being carefully scrutinized, and it was found that the top outlet end clearance of the L.P. compressor, the coldest unit, was steadily decreasing with each increase in speed. By the time the H.P. turbine had reached 4,500 r.p.m. and the L.P. turbine 2,737 r.p.m. this radial clearance had decreased from 0·040in. to 0·019in.; the bottom having increased 0·010in. with both turbines still well below their maximum speed. Closer investigation was required.

The first and most common reaction was to suspect the instrumentation. The monitor readings had all been checked (when cold) against a direct feeler gauge reading between probe and blade, and the reading with both shafts barring round had closely tallied. Now, as a further check, a dummy white metal tipped plug was inserted at the top outlet end, with an initial cold clearance of 0·010in. and the gas turbine was run up to an L.P. turbine speed of 2,730 r.p.m. The plug was then replaced with its electronic counterpart which again recorded 0·019in. clearance. Measurement of the white metal tipped plug disclosed that 0·009in. of white metal had been removed, thus indicating a total loss of clearance of 0·019in. or within 0·002in. of the electronic plug reading.

A further example of the accuracy of the monitor was later established when the L.P. compressor inlet end plugs showed a decrease of 0·004in. on their normal reading. In this case atmospheric dirt had built up on the blade tip.

The accuracy of the instrument was now established, but the cause of this change of clearance on a relatively thick cold casing proved elusive. The rapidity with which the clearances decreased, with turbine speed increases, discounted the cause of the trouble being due to thermal expansions, for the monitor observer could detect a change before the engineer had removed his hand from the fuel lever.

One suggested theory was that the loss of radial clearance could be attributed to a certain amount of restraint being imposed on the L.P. compressor outlet flange by the action of the outlet ducting and bellows under air pressure loading.

The results of tightening the bellows tie bars did not support this belief, nevertheless the L.P. compressor "make-up" piece was removed and 0·25in. machined off the flange face, thus removing any restraint being imposed by the ducting on the L.P. compressor outlet flange. No noticeable benefit was derived from this action.

External measurements using optical sighting gear on the L.P. compressor inlet casing did, however, confirm a tendency for this relatively light fabricated casing to "stretch" in the region of 0·030in. when the unit was running due to air pressure loading on the outlet casing. An axial stiffening plate, $\frac{3}{4}$in. thick, was welded across the inlet ducting, $\frac{1}{16}$in. draw being allowed. As a further precaution the casing was lifted 0·005in. relative to the rotor and two additional dowels were fitted to the vertical joint at the barrel outlet end. Unfortunately these modifications did not eliminate the clearance troubles, although the 0·005in. increase in the initial top clearance allowed "no load" performance readings and further trials to continue at higher speeds.

Subsequent static tests on the L.P. compressor outlet casing, using screw jacks to simulate pressure loading conditions, resulted in a centre plate and two stays being welded in the outlet casing. As a result of these modifications the compressor casing became much more stable and only comparatively slight changes in radial blade clearances were encountered. L.P. compressor clearances never gave any trouble in subsequent tests.

During the early stages of the trials, oil seepage was observed from the L.P turbine inlet bearing at the higher turbine speeds. Apart from the loss there was a risk of it igniting off the local hot surfaces.

After several examinations closer investigation by the designers indicated that the large diameter of the oil thrower, as fitted to this bearing, could create a pumping action, thus allowing oil to be drawn out of the bearing. The subsequent modifications were that the oil thrower was machined to a smaller diameter, three grooves were machined into the shaft in way of a new "eyebrow" of reduced radius, three anti-whirl internal baffles and an oil deflector plate were fitted.

These modifications were carried out and at subsequent trials no trace of oil seepage was observed from this bearing.

STARTING TURBINE

Experience from the first gas turbine installation with direct current electric motor starting, with the additional technical problems of the new a.c. system, resulted in the choice of steam turbine starting.

A well known make and proved design of turbine was chosen, but before the *Auris* left for sea trials, three turbine rotors had been used for what was outwardly a simple job.

During the later stages of design the mechanical drive was modified to incorporate an SSS clutch so that the starting turbine could be re-engaged without waiting for the high pressure compressor to stop rotating. This proved entirely successful from the first start and was later to prove invaluable.

The steam starting turbine was supplied with saturated steam from the Scotch boiler, exhausting into the ship's auxiliary condenser with a maximum available vacuum at the condenser of 12in. Hg. Both the steam and exhaust lines of the turbine were long and tortuous due once again to the general layout of the converted engine room. Careful attention was always paid to draining these lines before bringing the starting turbine into operation, but complete elimination of water was always uncertain, both during the running of the turbine and for the long periods it was not in use.

On commencement of one of the series of basin trials the H.P. turbine failed to accelerate on disengagement of the starting turbine at 2,100 r.p.m. The turbine was then re-engaged, but during the run up (the nineteenth occasion the turbine had been used) an unusually harsh noise was heard from the vicinity of the starting gearbox. An immediate examination showed the shaft to be 0·004in. out of truth, but it was reassembled and run up to 2,600 r.p.m. (13,000 r.p.m. was normal maximum for this unit) and then again examined. The turbine shaft was now found to be bent 0·038in. and a small fracture was observed at the radius adjacent to the turbine wheel. Examination later undertaken by a metallurgist (not connected with the turbine manufacturers) disclosed that two cracks had been formed. These had originated in surface corrosion marks which were on a part of the shaft in the steam space. One spread into the radius to a radial depth of $\frac{3}{16}$in, and the second one, which was approximately diametrically opposite, spread into the radius over half the full cross sectional area of the shaft (Fig. 5). The shaft failure was considered to be due to vibratory fatigue effects of an intermittent nature, such as would occur when the shaft passed through a critical speed. The stress was clearly concentrated at the particular fillet where the failure occurred, and there was little doubt that the trouble was aggravated by the long shut-down of the machinery during the winter of 1958-59.

FIG. 5—*Fatigue failure, starting turbine shaft*

The spare turbine shaft, rotor bearings and packing were then fitted and the alignment carefully checked. During a trial run after this work had been completed, it was noticed that the turbine became "rough" at 1,200 r.p.m. (compressor speed). At a later date vibration records were made and their analysis proved that a critical existed in the starting turbine between compressor speeds of 1,200 and 1,500 r.p.m.

This second turbine shaft and rotor was later replaced by a shaft of increased tensile strength, the steel used being nickel/chrome/molybdenum 50/55 tons/sq. in., u.t.s., and heat treated for hydrogen embrittlement effects after chromium plating.

Subsequent examination of the second shaft showed it to be 0·004in. out of truth, and so this shaft was returned to the manufacturers for straightening, and then retained on the ship as a spare.

An additional water trap was fitted in the steam supply line, but no further trouble was experienced with this unit on sea trials or in service.

TRANSMISSION

Before the ship could be taken to sea it was essential to prove that the hydraulic components could not only transmit sufficient power in both directions, but also that each unit could withstand the arduous manœuvring conditions. At first no one knew what these conditions would be.

The original testing programme appeared straight forward on paper, but in practice new problems arose as each test proceeded. Testing soon fell into a pattern of a mechanical test, examination, and modification. To examine the transmission components, the entire top line assembly had to be removed from the gearbox, and dismantled ashore taking about two days and a night.

Naturally tests on the gas turbine proceeded simultaneously and the delays incurred by the transmission were often most useful for other modifications (such as the blow-off valves) to be carried out. On two occasions the gas turbine was tested without the gearbox being connected and this provided some very useful but controversial information which will be discussed later.

In general the difficulties with the hydraulic components were due to:

a) overheating;
b) mechanical damage;
c) design of seals.

Overheating was partly due to the failure of oil seals and partly due to windage. Mechanical damage was due to the complete failure of the astern converter, and also the difficulty of aligning the quill shaft inside the pinion with a practical bearing oil clearance and a workable oil seal.

The problems of designing the oil seals were immense, and any failure immediately resulted in a rapid increase in temperature.

First Series

At a very early stage of the "no load" gas turbine tests it was noticed that the ahead hydraulic coupling and alternator coupling covers were overheating, the temperature of the housing inside the covers being 90 deg. C. (195 deg. F.).

On commencing the initial "load" runs, with the astern converter fitted, an immediate further rise in the external casing temperature occurred and reached 136 deg. C. (277 deg. F.). Examination through the inspection covers showed that oil was leaking along the quill shaft, and also the ahead coupling nozzles were churning oil in its sump. The leakage of the seals was conclusively proved later when the alternator started running while attempting to engage the friction clutch.

Seven attempts were made at this stage to engage the clutch with the ship's stem against the dock wall. The H.P. turbine, L.P. turbine, and propeller speeds were 4,800, 2,300 and 68·5 r.p.m. respectively. The first two attempts were quite successful with the engagement being completed in two or three seconds. The third and fourth engagements took approximately 30 seconds for the slip to become zero, and at the last three attempts the clutch failed to engage.

The first complete examination of the hydraulic components then took place and there was abundant evidence of overheating, for the primary ahead wheel was covered in a black lacquer (known to occur above 150 deg. C.), a bronze seal had fused, and thick carbon deposits were found where the quill shaft had been subject to severe local heating. The white metal seals themselves were in a poor state and were of course renewed with some improvements and additions.

One feature observed on this examination was that the astern converter contained some small cracks in both rotating units in the blade brazing, but their significance was not appreciated at this time.

A multitude of modifications quickly followed and these were aimed at better sealing, reduced churning by making new casings, and various means of dissipating heat, such as finned casing and cooling oil sprays.

Second Series

On August 28th 1958, the second series of mechanical proving runs commenced with the express intention of mechanically proving the astern converter.

The gas turbine unit was started and allowed to settle to normal idling conditions of temperature and speed. After filling the astern converter, the relative H.P. turbine, L.P. turbine, and propeller speeds were 4,100, 1,800, and 39 r.p.m. respectively. Over a five hour period the propeller speed was then gradually increased to 69 r.p.m. astern with the usual oil supply pressure of 31lb./sq. in. and a converter drain temperature of 78 deg. C. A dramatic change in note of the gas turbine unit then occurred, accompanied by a rapid increase in L.P. turbine speed and a complete loss of transmitted power. The turbine speed was controlled by the quick action of a test engineer reducing fuel, but as the propeller shaft stopped, an ominous rumbling noise could be heard from the forward end of the gearbox in the vicinity of the astern converter casing. The trial was abandoned and the turbine shut down for what was to prove to be six months.

The subsequent examination soon disclosed the complete failure of the brazing attaching the vanes to the cast steel torous of the primary wheel. The secondary wheel was still intact, but several of its vanes were bent or tearing away at the brazed attachment to the outer torous (Fig. 6). In addition

FIG. 6—*Failure of hydraulic converter*

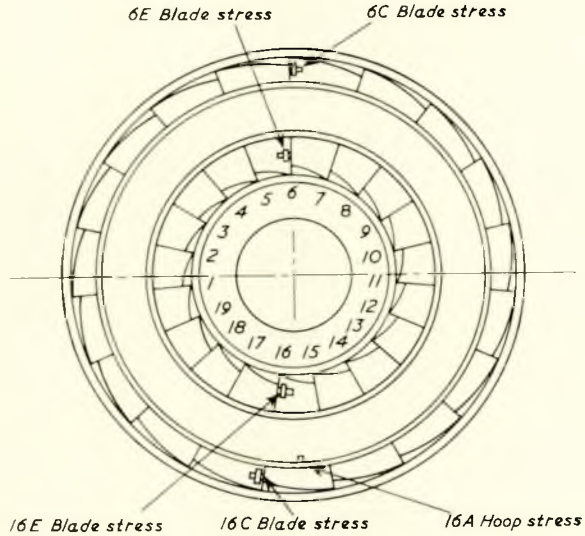


FIG. 7—Auris *astern converter primary wheel overspeed tests*

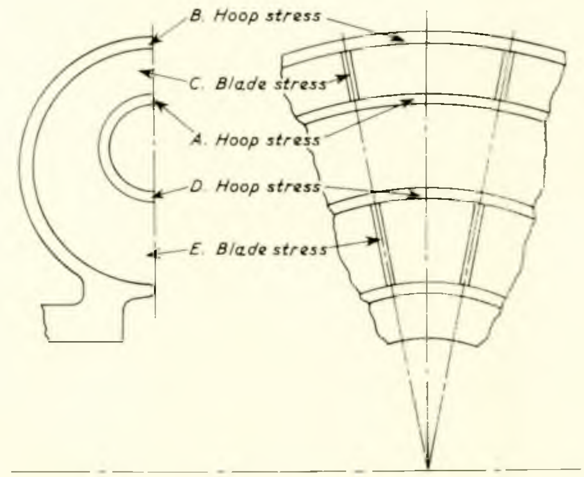


FIG. 8—Auris *astern converter secondary wheel overspeed tests*

	Speed r.p.m.	Stresses Tons/sq. in.				
		16C	6C	16E	6E	16A
PRIMARY WHEEL	1,020	—0·03	—0·04	0·11	0·10	0·31
	2,116	0·06	0·07	0·37	0·20	1·45
	3,014	0·06	0·12	0·68	0·43	2·89
	3,584	0·10	0·15	0·96	0·64	4·32
	4,824	0·45	0·17	1·48	0·95	7·65

GAUGE GROUP NO. 1. (COMPENSATING GAUGES ON HUB.)

	Speed r.p.m.	Stresses Tons/sq. in.				
		A	B	C	D	E
SECONDARY WHEEL	1,060	0·55	0·43	—0·16	0·20	0·18
	2,014	2·06	1·50	—0·41	0·64	0·45
	3,068	5·05	3·23	—0·79	1·53	0·88
	3,580	7·08	4·63	—0·80	2·03	1·25

GAUGE GROUP NO. 2. (COMPENSATING GAUGES ADJACENT TO ACTIVE GAUGES)

	Speed r.p.m.	Stresses Tons/sq. in.				
		A	B	C	D	E
SECONDARY WHEEL	1,032	0·24	0·21	—0·07	0	—
	2,030	1·20	1·07	0·05	0·20	—
	2,970	2·58	2·20	0·18	0·49	—
	3,630	4·12	3·42	0·52	1·19	—
	4,842	7·72	6·29	1·19	1·59	—

serious damage had been sustained by the stationary reversing cascade.

Failure of the converter was due not only to the lack of strength in the brazing itself, but also because inaccessibility precluded some of the periphery of a blade being brazed.

The total time the converter had been engaged was 16 hours during a gas turbine running time of 166¼ hours.

During the ensuing months the astern converter was completely reconstructed utilizing the original outer torous but with 40 per cent thicker blades than the original design.

Difficulties of reconstruction were aggravated by the necessity of having to weld the new blades in an already completely finished machined torous. Fabricated welding jigs, repeated heat treatment and infinite care finally resulted in a very much stronger converter that withstood several overspeed tests (see Figs. 7 and 8).

Whilst this work was in progress a booster pump had been fitted in the ship in an endeavour to improve the astern converter efficiency and also for improved clutch engagement.

Third Series

The third series of tests commenced on the 24th February 1959, to mechanically prove the transmission. The engagement of the ahead coupling at progressively increasing L.P. turbine speeds was repeated as better instrumentation had been installed to measure the transient torque. The result was encouraging for even at the highest speed of engagement (3,000 r.p.m.) the recorder trace showed no transient comparable with normal torque loading; the only feature of interest on the record being an oscillation during the filling time of the coupling of approximately six cycles per second of an amplitude of about one-twentieth full load torque.

During this third series of basin trials another example of the frustrating delays occurred. The propeller always had to be stopped to enable other ships to be moved in the basin. On this occasion after a two hour wait the propeller was re-engaged normally, but steady conditions were not obtained on the turbine, and immediate investigation disclosed that the main thrust block ahead pads had severely wiped and the thrust collar in way of the ahead pads scored. No satisfactory explanation could be found although the following theories were expounded:

a) the thrust meter had been inadvertently left in an operational condition while the shaft was stationary thus losing the "oil wedge" on restarting.

b) hull vibration due to the shallow water during previous tests might have damaged the pads. Evidence of scuffing was certainly found at the bottom of the thrust collar.

c) the lubricating oil to the bearing was temporarily suspended due to mal-operation of a hand controlled valve in the supply pipe.

The ahead thrust pads were renewed, the thrust collar machined ashore, and re-assembly in the ship completed within three days.

Tests recommenced to complete the ahead coupling series and also to test the benefit of the booster pump. It was now found difficult to keep steady conditions on the turbine as the propeller speed fluctuated between 69 and 79 r.p.m. due to the shallowness of the water and an oscillating movement of the after part of the ship relative to the quay (Fig. 9 shows part of the Honeywell Brown recorder).

The third series of basin trials also came to a dramatic end. The booster pump was first tried with a modest astern converter engagement speed of 2,250 r.p.m. The shaft accelerated normally and the propeller speed settled at 34 r.p.m. but, immediately, the ahead coupling casing temperature started to rise and reached 130 deg. C. (265 deg. F.) in eight minutes. The converter was emptied and the casing temperature allowed to cool to 49 deg. C. (120 deg. F.) before re-engaging the converter, this time without the use of the booster pump. The propeller speed was gradually increased over a four hour period to 74 r.p.m. with a power output of 1,740 s.h.p. On this occasion the ahead casing temperature increased to 99 deg. C. (192 deg. F.) but remained steady for an hour.

The test was then repeated with a higher speed of engagement (2,350 r.p.m.) but 12 minutes later a minor explosion occurred within the gearcase and oil vapour and smoke poured out from various ventilators and oil seals. At the same time there was a rapid increase in the ahead coupling oil drain temperature, 126 deg. C. (258 deg. F.), and casing temperature.

Inspection of the primary transmission line showed little damage to the hydraulic components or bearings but the excessive clearances of the latter had clearly caused mechanical damage to various seals.

Considerable attention was then paid to the physical alignment of the transmission components, and details of the remedial action are given in reference[3], pages 440-441. After each modification a dynamic balancing check in the full and empty condition was always made but this was now becoming even more difficult due to the quantity of metal which had been removed from the torous of the astern converter.

Fourth Series

Temperature instruments with alarms were now fitted to the ahead and astern casings. The final series of tests was fortunately successful. First the ahead coupling was operated for 11 hours with a propeller speed of 70 r.p.m., a slip of 7·97 per cent and shaft power of 1,100. No undue rise in casing temperature occurred. The astern converter was then engaged at various L.P. turbine speeds from 2,250 to 3,000 r.p.m. concluding with a successful steady run at 42 r.p.m.

Once again the booster pump was tried. The astern converter was filled and after attaining steady conditions, the supply pressure was gradually increased to 85lb./sq. in. The result was interesting, for instead of the expected increase in converter efficiency, the slip actually increased at the higher pressure from an L.P. turbine/primary pinion speed ratio of 63·1 per cent to 60·1 per cent indicating that no advantage could be gained by using high pressures to the converter.

A number of astern to ahead and ahead to astern straight through movements were then carried out successfully at various L.P. turbine speeds up to 2,400 r.p.m., and then the series was terminated with a prolonged fifteen hour astern run, during which the ahead coupling casing steadied at 71 deg. C. (160 deg. F.).

The astern converter was used for one further period at 40 r.p.m. to enable the gas turbine and fuel control equipment to be tested on high viscosity fuel for nine hours.

It was now considered that further runs in the basin with the ship's draught limited to 17ft. would serve no useful purpose, and due to the excessive hull vibration damage might ensue.

MANŒUVRING

During the early stages of the transmission proving trials in 1958, it became obvious that the couplings could not be disengaged at even average L.P. turbine speeds. The immediate rapid increase in speed was due to the heat inertia of the unit and could easily cause the L.P. turbine overspeed trip to operate.

Two methods were initially adopted to counteract this overspeeding:

a) after reducing fuel, the hydraulic coupling was successively engaged and dis-engaged until the H.P. turbine inlet temperature had fallen to about idling conditions;

b) the coupling was left engaged until the L.P. turbine was dragged down to such an extent that it would not overspeed, on the coupling being emptied.

Neither was a practical solution for operating a ship. The first required 10 to 12 successive operations before the propeller could be stopped and the second resulted in a high L.P. exhaust temperature. Both lacked the response to the telegraph order.

An alternative solution was to fill the ahead coupling and the astern converter simultaneously. Pametrada rig tests showed that with an L.P. turbine speed of 2,000 r.p.m., twice the propeller law torque could be absorbed from the turbine, whilst keeping well within the full load quill shaft and primary pinion gear tooth stress. It will be appreciated that at this point the hydraulic components were giving so much trouble that this solution was not then particularly favourable. In addition failure of the quill shaft could prove disastrous to the turbine. At a much later stage this procedure proved very satisfactory for dead slow ahead operation, as the torque from the ahead coupling was slightly greater than that of the astern converter.

The gas turbine designers approached the problem with a view to safeguarding the turbine under all conditions. It was decided to fit two 4in. blow-off valves to the H.P. compressor discharge duct. By arranging these to be pneumatically operated they could be linked to the overspeed trips. Their function was threefold:

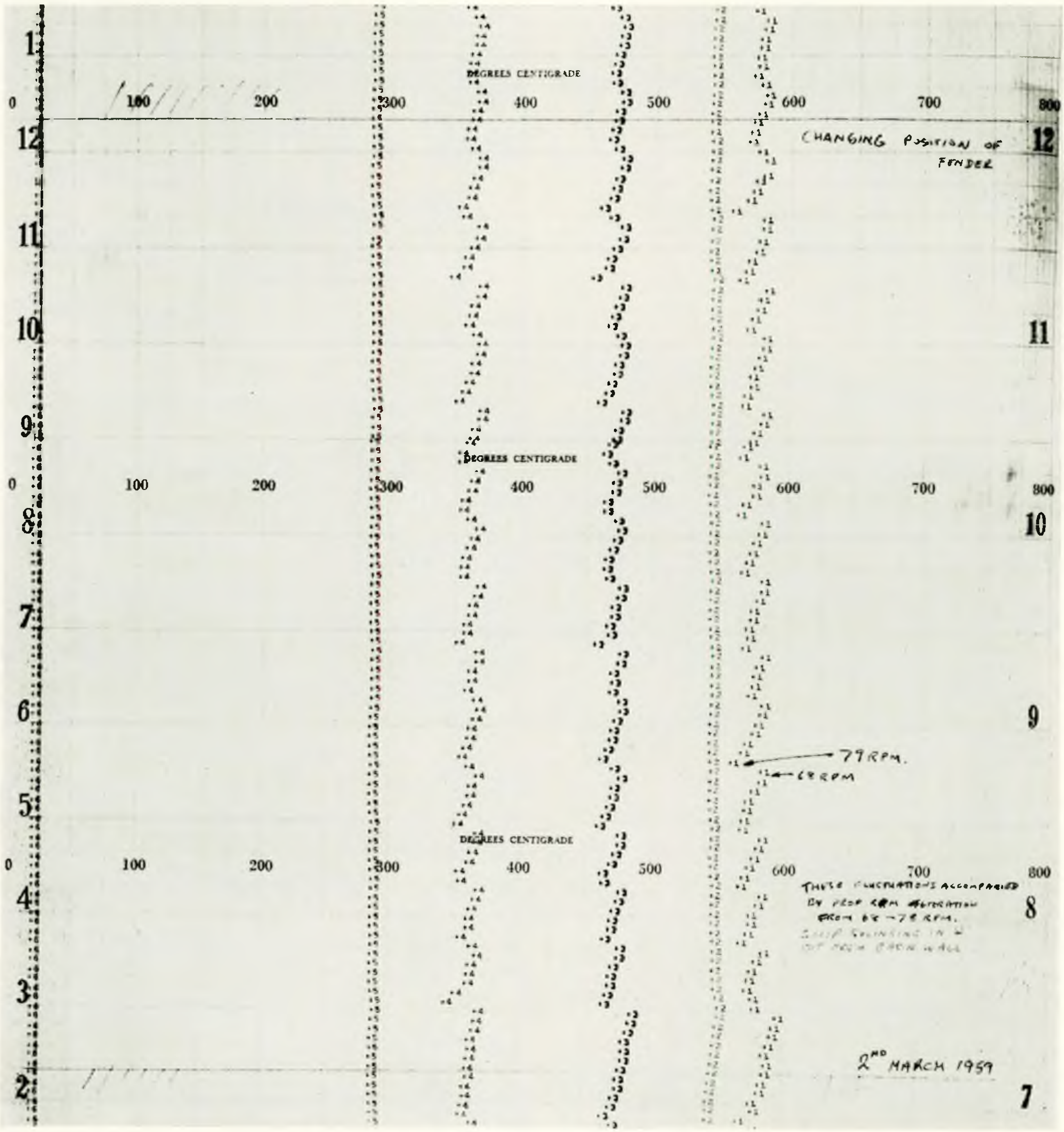


FIG. 9—*Honeywell Brown recorder*

a) prevention of overspeed while manœuvring, either by loss of transmitted power through lack of lubricating oil or by mechanical failure in transmission;
b) warming up the turbine without using the propeller;
c) as a safety device if accidental flooding of fuel in the combustion chamber occurred.

The failure of the astern converter in August 1958, enabled the design and manufacture of these valves to proceed, but the problems were considerable as the air passed through the valves at very high velocity, exact remote control at different openings was required, and being a safety device there might be no second chance.

The reconstruction of the astern converter was completed before the new valves arrived at Birkenhead, so two 8in. hand operated lubricated cocks were used temporarily to test the effect of bleeding air from the cycle. It was found that at various propeller speeds, both ahead and astern, it was possible to limit the L.P. turbine speed to any predetermined value on draining the hydraulic coupling. In addition, it was now possible to preheat the turbine to any desired value before engaging the propeller, thus preventing the rapid rise previously encountered.

These encouraging results also changed the speed at which manœuvring could be conducted, for the power dissipated in the blow-off valves was now immediately available to the propeller. This in turn made the transient torque tests mentioned in the third series of transmission tests essential. It was these tests that finally fixed the ahead coupling engagement speed at 3,000 r.p.m.

After a few initial control system difficulties had been rectified, the pneumatic valves (Fig. 10) operated very well. One hand-controlled lubricated cock was retained for additional adjustment and "stand-by" purposes, but it remained an uncertain liability due to the lubrication being blown out with seizure of the plug and operating mechanism, from time to time.

The air passing out of the valves to atmosphere made a very loud noise and caused some embarrassment on the initial sea trial. This difficulty was overcome by fitting two silencers in series, their total weight being 1·71 tons.

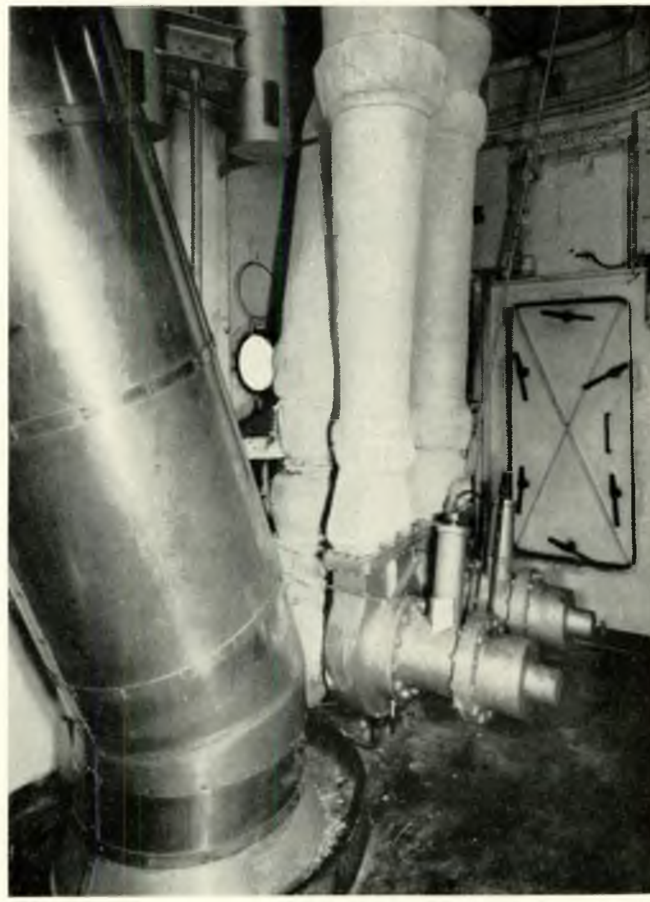


Fig. 10—*Blow-off valves*

PART II

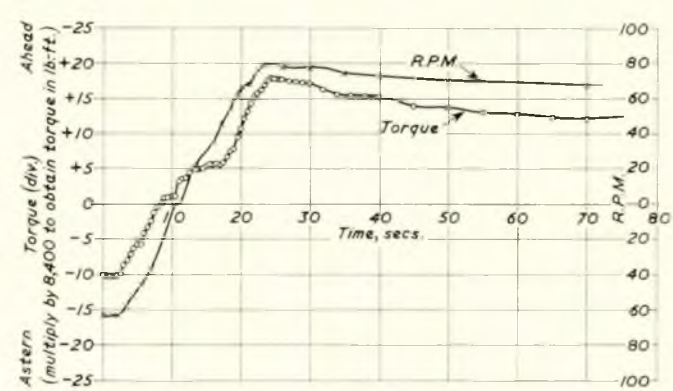
PRELIMINARY SEA TRIAL

During the latter part of the mechanical proving trial in the basin, the tests were often interrupted by restrictions due to the tidal level in the river and other ship movements within the basin. In addition it had been found difficult to maintain steady conditions with only a few feet of water beneath the keel.

The preliminary sea trial was therefore designed to be of 24 hours duration in Liverpool Bay to establish that the ship could manœuvre safely, and to determine the limiting engagement speeds for operational purposes. Although the ship left the basin completely under control of tugs, it was soon established that the manœuvring capabilities were all that had been expected.

To test the manœuvrability, the telegraph was operated from the bridge in the exact sequence of a berthing operation taken from an old movement book. These movements were successfully accomplished with full use being made of the blow-off valves to maintain the L.P. turbine speed and H.P. turbine inlet temperature. The maximum rate of manœuvring on this test was 28 orders in 25 minutes and a total of 46 separate movements in 72 minutes.

Tests were again made to determine the magnitude of the maximum transient torques in the quill shaft during engagement of either the ahead or astern couplings, with astern or ahead way on the ship and also with the ship stationary. The maximum L.P. turbine speed of engagement was 3,000 r.p.m., and the tests were carried out in the light and loaded ship condition. Analysis of this series of tests confirmed that



Prop. speeds 60 r.p.m. (nominal) astern to 72 r.p.m. (nominal) ahead
Ahead converter engaged at L.P. turbine speed of 3,000 r.p.m.
Tests carried out 4.4.59. 03.40 hrs. g.m.t.
Draughts 27ft. 3in. aft. 24ft. 9in. fwd.

Fig. 11—*Transient torques (preliminary sea trials)*

under these conditions the transient torques would never exceed full load torque (see Fig. 11).

A test was then made to determine the minimum ahead propeller speed with a limitation of 420 deg. C. on the L.P. turbine outlet temperature. By reducing fuel and using the blow-off valves an L.P. turbine speed of 1,750 r.p.m. was reached, corresponding to about 50 r.p.m. on the propeller. By filling the astern converter as well as the ahead coupling, the propeller speed was reduced to 42 r.p.m. When the ship was in service it was normally possible to reduce the speed to 35 r.p.m., for an improved technique was devised for accelerating the L.P. turbine after such a movement.

There was of course no difficulty in obtaining dead slow astern owing to the higher percentage slip in the astern converter. Tests on the friction clutch had to be postponed due to its pilot control equipment jamming.

The remainder of this preliminary sea trial was occupied with a routine seven hour duration run, ballasting and checking the H.P. turbine trip speed. This latter operation could only be carried out at sea.

During these higher power runs it was noticed that a high frequency vibration occurred in the waste heat boiler. With an L.P. turbine speed exceeding 3,000 r.p.m. the entire main evaporator tube assembly vibrated violently.

On re-entering the shipbuilder's basin, an error of judgement, in no way connected with the machinery, resulted in the ship's stem being badly damaged by the dock wall. The resulting delay during repairs enabled "no load" runs on the turbine to be carried out and an analysis made of the waste heat boiler vibration. This clearly showed that, above L.P. turbine speeds of 3,000 r.p.m., a standing wave in the gas existed between the two banks of main evaporator tubes. Although this was only the second example experienced by the boiler manufacturers, the trouble was easily overcome by fitting a vertical baffle plate between the tube banks.

The enforced delay also enabled the H.P. turbine last stage blading to be examined and the radial clearance to be measured. The blading was in perfect condition and only a small reduction in the blade clearance was measured.

MAIN SEA TRIAL

Extensive sea trials in St. George's Channel and off the West Coast of Scotland commenced on the 20th April 1959, to prove the mechanical reliability of the main and auxiliary machinery. In addition it was the express intention to obtain the maximum amount of gas turbine performance data, for the "as new" condition of the machinery would never again be available.

Considerable discussion had taken place, during the preceding years, whether the comparatively low efficiency of the astern converter would jeopardize the ship when trying to stop quickly. Two series of trials were devised to clarify this. The first series was conducted at a draught of 20ft. and the second at 27ft. 6in. In both cases the engine condition, time and stopping distance were recorded as accurately as possible. The ship was taken to a particular area off the Isle of Man and the stopping distance measured by Decca Navigation. The results are shown in Table I and clearly indicate the exceptional ability of the ship to stop.

The comparative data for other ships was not then readily available so, three ships of various classes in Shell Tanker's fleet were asked to make a similar test from an ahead speed of 66 per cent full r.p.m. to 40 per cent of full ahead r.p.m. astern. Due to the widely scattered areas of trading, these ships could not supply Decca stopping distances but the results may be of interest and are shown in Table II.

During the first series of "stopping" trials, transient and propeller torque records were again taken and a typical example is shown in Fig. 12.

A friction clutch trial was next carried out with complete success. The clutch was first engaged with a falling L.P. turbine speed at 2,000 r.p.m. thus minimizing the ahead coupling slip and engagement torque. The propeller speed was then gradually increased up to 113·75 r.p.m. during which time the torsional vibrations were recorded and a critical established between 78 and 87 r.p.m. The clutch was used on this occasion for six hours at 113·75 r.p.m.

Although the astern converter had by this time operated for long periods during the basin trials one further duration run astern was required. It will be recalled that the working propeller was left handed to suit the machinery already designed for *Hemisinus*. The spare propeller however was the original manganese bronze right handed one, for reasons of economy, and also because a cast iron propeller could not be designed for the existing aperture. Thus if the spare propeller had to be used, the astern converter would then have to drive the ship ahead.

A four hour duration run astern, with the rudder amidships, gave an exact turning circle of 1,770ft. at a propeller speed of 80 r.p.m., shaft horsepower of 1,910, and the ship's speed 5·31 knots. In view of the difficulties encountered, during basin trials, with overheating of the ahead coupling casing at these conditions, it was interesting to discover that after the initial rise the temperature remained steady at 74 deg. C. (165 deg. F.).

During this astern trial, the opportunity was once again taken to prove the ineffectiveness of the booster pump, under its most favourable conditions, for increasing the efficiency of

TABLE I.—STOPPING TRIAL

LIGHT SHIP CONDITION.	DATE 20th APRIL, 1959.	DRAUGHT 20 ft.—0 in.	
Propeller speed ahead	Propeller speed astern	Time to stop ship	Distance in feet
85 r.p.m.	full	5 min. 21 sec.	4,104
60 r.p.m.	full	4 min. 4 sec.	1,490
40 r.p.m.	full	3 min. 17·75 sec.	1,337
112 r.p.m.	full	6 min. 15 sec.	5,168
112 r.p.m.	60 r.p.m.	7 min. 34·75 sec.	5,472
112 r.p.m.	50 r.p.m.	8 min. 41·5 sec.	5,806
LOADED CONDITIONS.	DATE 5th MAY, 1959.	DRAUGHT 27 ft.—6 in.	
Propeller speed ahead	Propeller speed astern	Time to stop ship	Distance in feet
107 r.p.m.	full	6 min. 30 sec.	4,985
108 r.p.m.	60 r.p.m.	9 min. 30 sec.	6,688
107 r.p.m.	50 r.p.m.	9 min. 52·4 sec.	5,776
85 r.p.m.	full	5 min. 26·5 sec.	2,797
60 r.p.m.	full	3 min. 41 sec.	1,520
40 r.p.m.	full	2 min. 55 sec.	1,003

TABLE II.

Ship (deadweight)		Full ahead r.p.m.	r.p.m. During manoeuvre		Time to stop ship		$\triangle S^2$ s.h.p.
			Ahead	Astern	Trial sec.	Calculated sec.	
s.s. *HAMINELLA*	(18.193)	105	70	45	600	625	743
s.s. *HYGROMIA*	(18,011)	105	70	45	315	625	743
t.e.s. *HYLIX*	(17,892)	105	70	45	545	643	789
m.s. *ABIDA*	(18,090)	115	80	50	620	570	680
m.s. *AMASTRA*	(18,037)	115	80	60	645	435	680
m.s. *CRANIA*	(12,985)	108	70	45	475	671	778
m.s. *CINULIA*	(12,985)	108	70	45	250	671	778
m.s. *CAMITIA*	(12,985)	108	70	45	840	671	778
s.s. *ZAPHON*	(38,390)	108	70	45	679	750	865
s.s. *ZENATIA*	(38,443)	108	70	45	840	750	865
s.s. *VIBEX*	(31,956)	109	70	45	660	725	831
s.s. *VOLVATELLA*	(32,170)	109	70	45	885	725	831
m.s. *DROMUS*	(12,079)	120	80	50	467	625	742
m.s. *DORCASIA*	(12,100)	120	80	50	659	625	742
g.t.s. *AURIS*	(11,987)	120	85	70	326	365	715

$\triangle$ = Displacement
S = Full speed
s.h.p. = Power required

the astern converter. No apparent improvement in the L.P. turbine/primary pinion speed ratio was observed on this ocasion, although a further trial at a later date gave the following result:

L.P. turbine speed, r.p.m.	3,000	3,500	3,840
Pump pressure		Speed ratio	
60lb./sq. in.	0·709	0·713	0·712
50lb./sq. in.	0·709	0·711	0·710
40lb./sq. in.	0·709	0·709	0·708
30lb./sq. in.	0·709	0·704	0·702
25lb./sq. in.	—	—	0·698

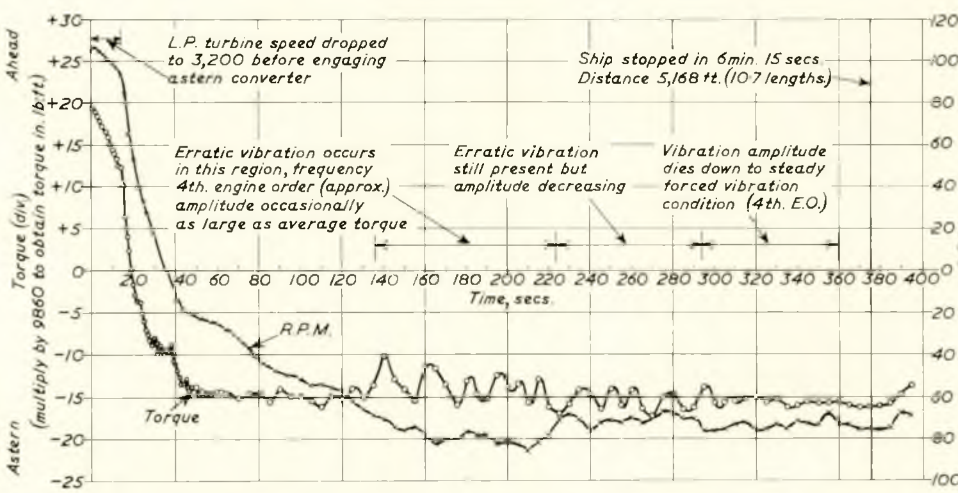
Thus even at the highest L.P. turbine speed there was only a very small gain in the speed ratio, making the operation of the booster pump, with its added complications for manœuvring, worthless.

On completion of this astern trial, the ship proceeded to an anchorage in the Clyde, where both compressors were Teepol and water washed while various inspections were made. Some concern was felt about the after pinion bearing, which had been operating with a return oil temperature of 92 deg. C. (198 deg. F.) during the run on the ahead clutch at 115 r.p.m. These fears were however unfounded for the bearing was in excellent condition.

The trials continued with a duration run of 947 miles during which time performance readings were taken in both "loaded" and "light" ship condition. During the latter part of these trials the weather deteriorated to rough seas and wind force 7, but no difficulties were experienced, except that the after pinion bearing again reached a temperature of 91 deg. C. At the termination of this trial an additional oil inlet to this bearing was brought into use.

The "full power" trial is invariably one of spectacular interest and not always of much practical value. This trial was of singular importance, for by this time much discussion centred round the transmission losses.

The trial was conducted over a distance of 100 miles with an average L.P. turbine speed of 3,800 r.p.m. and driving the propeller through the friction clutch at 118·7 r.p.m. It will be seen from Table III that the mean shaft horse power was 4,840 which, although completely satisfactory for the normal operation of the ship, raised numerous questions concerning "missing power". It was also found in practice that



Prop. speeds 113 r.p.m. ahead to 80 r.p.m. (nominal) astern
Astern converter engaged at L.P. turbine speed of 3,200 r.p.m.
Test carried out 20.4.49 21.52 hrs. g.m.t.
Draught 20ft. mean.

FIG. 12—*Transient torques (sea trials)*

TABLE III.—PERFORMANCE TEST RESULTS

		Full power Run (6 hrs.)	Boiler fuel run	Southampton to Curaçao	Curaçao to England	Curaçao to England
Test No.		27	29	45E	52C	52E
Date		28.4.59	30.4.59	6.9.59	7.10.59	13.10.59
Time (hours)		15.15	2.15	10.25	9.45	9.45
Ambient	Ta deg. C.	9	7·5	26	28	18·9
	Pa, In. Hg.	29·415	30·04	30·01	29·95	29·95
L.P. compressor	T. in, deg. C.	10	9·5	25·7	28·3	20
	T. out, deg. C.	94·2	84	90·5	99·8	97·1
	P. in, lb./sq. in. abs.	14·27	14·63	14·63	14·575	14·56
	P. out, lb./sq. in. abs.	33·05	30·83	27·50	28·90	31·06
	Pressure ratio	2·3157	2·1081	1·880	1·982	2·134
	Efficiency, per cent.	90·40	89·35	90·38	90·13	91·12
	Calculated h.p.	4,545	3,509	2,874	3,291	3,858
Intercooler	Thermal ratio	90·13	89·40	94·49	92·79	89·77
H.P. compressor	T. in, deg. C.	16·33	16·5	26·25	30·4	26
	T. out, deg. C.	130·5	121	137	144	135
	P. in, lb./sq. in. abs.	32·22	30·19	27·01	28·26	30·37
	P out, lb./sq. in. abs.	87·06	74·65	71·59	75·10	81·14
	Pressure ratio	2·7024	2·4726	2·651	2·658	2·672
	Efficiency, per cent.	82·55	81·13	86·01	85·25	88·17
	Calculated h.p.	6,113	4,875	4,684	5,158	5,403
Heat exchanger	Thermal ratio	68·22	67·88	69·59	69·61	69·66
H.P. turbine	T. in, deg. C.*	630	595	646	638	603
	T. out, deg. C.	518	486	536	529	498
	P. in, lb./sq. in. abs.	84·21	72·10	69·38	72·69	78·54
	P. out, lb./sq. in. abs.	47·39	41·46	39·73	41·32	44·36
	Pressure ratio	1·7771	1·7390	1·746	1·759	1·770
	Efficiency, per cent.	87·69	88·11	86·44	87·73	86·53
	Calculated h.p.	6,338	5,280	4,908	5,223	5,462
L.P. turbine	T. in, deg. C.	518	486	536	529	498
	T. out, deg. C.	326	328	374	362	330
	P. in, lb./sq. in. abs.	46·65	41·07	39·43	41·12	43·92
	P. out, lb./sq. in. abs.	14·89	15·07	15·03	15·04	15·07
	Pressure ratio	3·1342	2·7256	2·623	2·735	2·914
	Efficiency, per cent.	94·61	90·91	91·10	91·11	89·81
	Calculated h.p.	10,783	7,641	7,215	7,997	8,732
Speeds	L.P. line, r.p.m.	3,807	3,480	3,351	3,540	3,670
	H.P. line, r.p.m.	5,639	5,429	5,582	5,640	5,632
	Propeller shaft, r.p.m.	119	107·2	104·6	103·6	107·85
	Drive	Clutch+ Coupling	Clutch+ Coupling	Clutch+ Coupling	Coupling only	Coupling only
Horse powers	Propeller shaft by Pametrada	4,680	3,510	3,590	3,434	3,780
	Propeller shaft by Siemens	5,000	3,480	3,826	3,276	3,580
	Auxiliary horse power	0	0	0	158	146
	Mean propeller s.h.p.	4,840	3,495	3,708	3,513	3,826
	Calculated turbine power	6,238	4,132	4,431	4,706	4,874
	Correction from G586 (Test 36)	340	295	270	258	298
	Mean missing power transmission losses	1,058	342	453	935	750
Air flow (L.P.C. inlet) lb./sec.		85·28	74·05	66·98	72·02	78·74
Fuel consumption, lb./sec.		0·748	0·654	0·624	0·639	0·698
Specific fuel consumption (Propulsion only):—						
lb./hr./mean propeller s.h.p.		0·556	0·674	0·606	0·655	0·657
lb./hr./calculated turbine power		0·432	0·570	0·507	0·489	0·516
Overall efficiency based on:—						
Mean propellor s.h.p., per cent.		25·03	20·98	22·87	21·27	21·20
Calculated turbine power, per cent.		32·26	24·81	27·33	28·49	27·01

*NOT USED in calculating h.p. turbine efficiency.
Calculations based on variable specific heat with temperature.

NOTE
Lower calorific value (Gas Oil 10,150 C.h.u./lb.)
(Boiler fuel 10,000 C.h.u./lb.)

TABLE IV.—WASTE HEAT BOILER
Readings taken during full power.
Date: 28th April, 1959

Time of reading	12–15	13–15	14–15	15–15	16–15	17–15
Propeller r.p.m.	118·9	118·2	118·3	119·0	119·3	117·0
Shaft horse power	4,900	4,950	4,940	5,000	4,920	4,560
Gas below heat exchanger, deg. C.	313	315	319	320	324	323
Gas above heat exchanger, deg. C.	179	176	179	178	185	182
Gas after waste heat boiler, deg. C.	176	176	176	176	179	174
Waste heat boiler drum, lb./sq. in.	80	100	110	120	105	60
Waste heat boiler superheater, deg. C.	171	171	171	171	171	171
Forced circulation pump diff. lb./sq. in.	20	20	25	25·5	25·5	26·5

Note—All steam was passed through condenser until 16-45 hours when the boiler was opened up to the turbo-alternator.

little advantage could be gained for the addition of 1,000 s.h.p. above 109 r.p.m., the ship being of an old form and the propeller designed for an abnormal condition.

During the latter part of this full power trial, steaming tests were carried out on the waste heat boiler. The procedure adopted was to transfer all the electrical load to the Diesel alternator, and the steam turbo-alternator was shut down. The waste heat boiler was then paralleled with the Scotch boiler before closing the isolating valve to the latter, thus allowing the waste heat boiler to carry the steam load of the feed pumps and essential services. The turbo-alternator was then restarted and the 118 kW electrical load gradually transferred from the Diesel alternator.

Whilst maintaining this load, the steam pressure fell slowly to 78lb./sq. in., at which stage the test had to be abandoned following a bridge request for reduction of power. The principle readings during this test are shown in Table IV.

The approximate boiler evaporation at 78lb./sq. in. under the conditions at 17·15 hours was found to be 2,540lb./hr. and later tests at 50lb./sq. in. steam pressure showed an evaporation of approximately 3,540lb./hr. This was below the specified 6,000lb./hr., but it could be entirely accounted for by the low gas inlet temperature to the main evaporator, partly due to the low ambient temperature (the designed temperature being 25 deg. C.), and partly due to the cleanliness of the heat exchanger.

To gain experience in manœuvring, some trials were carried out by all the ship's engineers obeying telegraph movements. A total of 194 movements were carried out in 368 minutes; full use being made of the blow-off valves to control the L.P. turbine speed and the H.P. turbine inlet gas temperature. No difficulty was experienced in manœuvring, although acceleration from the lower speed ranges to higher ranges were comparatively slow whilst a coupling was engaged. This was easily overcome by momentarily disengaging the coupling and allowing the L.P. turbine speed to rise rapidly and again engaging the coupling at the predetermined maximum L.P. turbine speed of 3,000 r.p.m.

Reduction of speed could quickly be accomplished by reducing fuel and using the blow-off valves. The astern converter was also used as a brake whilst the ahead coupling was engaged. This system of manœuvring was the method adopted while the ship was in service.

On completion of these manœuvring trials, the gas turbine unit was idled with the centre burner only in use whilst high viscosity fuel of 1,500 sec. (Redwood I at 100 deg. F.) was recirculated through the main burner system. The main burners were then brought into use, and after some preliminary manœuvring, a twenty-four hour endurance trial using high viscosity fuel was carried out at an average propeller speed (with the friction clutch engaged) at 109·5 r.p.m. and s.h.p. 3,588. A fuel temperature of 115 deg. C. was maintained, and a specific fuel consumption of 0·674lb./s.h.p./hr. was measured.

To ascertain the capabilities of the gas turbine using the centre burner only in the event of any defect in the main fuel control system, an ahead run was carried out using the centre burner only. In this instance an orifice of 0·091in. diameter was used and a burner inlet fuel pressure of 740lb./sq. in.

A propeller speed of 60/65 r.p.m. was attained with the engine developing 780 s.h.p., the vessel at the time of this trial being at "loaded" ship conditions.

In the event of a complete breakdown of the main propulsion unit, the emergency steam turbine could be connected to the primary gearing by a removable shaft and pinion. This was tested with a five hour run in the sheltered conditions of the Clyde estuary. Steam was supplied from the Scotch boiler, all other non-essential steam services were shut down, and the electrical load taken by the Diesel alternator. The average propeller speed was 36 r.p.m. with a maximum measured shaft horse power of 153 (see Table V). Under these comparatively slow speed conditions there was thus about 300 h.p. "missing" in the gearbox.

Trials over the measured mile at Arran under various conditions of speed were carried out at a mean draught of 20ft. and 27ft. 6in. which are the normal "light" and "loaded"

TABLE V.—EMERGENCY TURBINE TRIAL
Date: 2nd May, 1959

Time of taking reading	12–30	13–30	14–30	15–30	16–30
R.p.m.	35	35	36	36	36
Torsionmeter reading	6·5	6·3	6·5	6	6
Shaft horse power	147	147	153	150	150
Boiler steam, lb./sq. in.	175	180	160	155	150
Turbine steam, lb./sq. in.	165	162	145	145	145
Turbine vacuum, in. Hg.	12	13·5	7	7	6
Gland steam, lb./sq. in.	0·5	0·5	4	2	1
De-mounable pinion sprayer, lb./sq. in.	10	10	10	10	10
Emergency gear-box sprayer, lb./sq. in.	10	10	10	10·5	10·5
Primary pinion forward bearing, deg. C.	40	43	45	44	44
Primary pinion aft bearing, deg. C.	45	46	47	46	46
Primary wheel forward bearing, deg. C.	36	37	39	39	39
Primary wheel aft bearing,deg. C.	35	36	37	39	39
De-mounable pinion for'd bearing, deg. C.	35	36	37	37	37
De-mounable pinion aft bearing, deg. C.	36	37	38	38	38
Primary pinion thrust bearing, deg. C.	38	38	39	39	39

ship conditions. The friction clutch was engaged for the first eight runs in each series, and "coupling only" for the final two runs owing to the transmission operating in the friction clutch critical range between propeller speeds of 78-87 r.p.m. The tabulated results are shown in Table VI.

On completion of these extensive "sea trials", but before the ship returned to Birkenhead, the main plant was shut down and the combustion chamber examined for the first time since commencement of basin trials. The plant had at this stage operated for 787 hours and approximately 400 tons of fuel had been burnt in the combustion chamber. With the exception of one brick which was replaced due to pitting and spalling, all refractory bricks and the cast refractory in the combustion chamber head were in good condition. The swirlers and draught tubes were also in excellent condition and the mixing zone and conical shield showed the beneficial effect of a special arrangement of cooling air. It will be recalled that these items required considerable attention during the development and service of the original gas turbine.

In an effort to reduce aeration of the lubricating oil to the main unit which had been observed throughout the trials, the gravity tank internal supply line was considerably shortened. Little or no benefit was derived from this modification, and the problem eventually defeated all attempted remedies while the ship was in service.

SUMMARY OF SEA TRIALS AND SUBSEQUENT MODIFICATIONS

During sea trials the main propulsion unit was operated for a total running time of 336 hours, including 232·6 hours during which time one or another of the transmission couplings was engaged. Approximately 2,182 miles were steamed in an ahead direction and a total of 517 separate engine movements were carried out in the course of manœuvring.

The gas turbine machinery operated extremely well under all conditions of power and weather. Cleaning of the compressor units during shut-down periods was affected by spraying a solution of Teepol and water into the units, followed by water washing whilst they were running at low speed under the control of the starting turbine. This was quite successful, but any fall-off of compressor efficiency due to atmospheric pollution and rain whilst the units were in operation was quickly recovered by introducing ground coconuts of 20/40 mesh into the air stream to these units.

The fuel control system which in itself was a prototype operated trouble free at all times both on fuel and gas oil. A design weakness in the burner lances caused a fatigue fracture in a burner lance tip during the measured mile runs, but the resulting modifications proved entirely satisfactory during the next 4,000 hours operation.

Despite the severe testing of the primary transmission line during the trials no mechanical defect arose. The efficiency of the astern converter, calculated as being 38·2 per cent was well below the specified 60 per cent, and no advantage was derived from the use of the booster pump. The emergency steam turbine was only intended for the low propulsion power in the event of a complete breakdown of the main unit, but the developed s.h.p. of 150 during the five hour trial with this turbine, in favourable weather conditions, fell short of expectations.

Although the waste heat boiler steam output was below its specified evaporation per hour, it was considered that tropical conditions, fouling of the heat exchanger and any deterioration in gas turbine performance would be favourable to the boiler.

Vibration of the auxiliary Scotch boiler and boiler platform also occurred at L.P. turbine speeds above 3,000 r.p.m. and as a result of hull vibration tests, carried out during the latter part of the sea trials, extensive stiffening of this platform was undertaken. This work proved completely unsuccessful, so the final approach was to give some flexibility to pipework to avoid fracture.

The ship was once again involved in labour disputes and her final acceptance was three months ahead.

A twenty-four hour sea trial was undertaken to prove the effects of some further modifications to the primary pinion bearing, during which time a six hour full speed run was carried out with the friction clutch engaged, at an L.P. turbine speed of 3,840 r.p.m. and s.h.p. 4,930. The bearing drain temperature was 81 deg. C. with a gravity tank lubricating oil supply to the bearing, reducing to 76 deg. C. with a pressurized lubricating oil supply of 15lb./sq. in. and to 74 deg. C. with a supply pressure of 18lb./sq. in. The oil inlet temperature was 36 deg. C.

Analysis, by the gas turbine designers, of the performance readings taken during sea trials disclosed a large discrepancy between their calculated gas horsepower output and the shaft horse power as measured by the torsionmeters. Their indications were that most of the power loss was within the transmission. The gas horsepower output, as on three previous gas turbines, was estimated from the measured mass flow of air at the inlet to the L.P. compressor and the air and gas temperatures at the inlet and outlet of the compressors and turbines. The usual estimate for gland leakage was of course made, but in this ship instrumentation had also been provided. The method of calculating the gas power output, whilst sound in theory, was open to criticism, as it assumed that the mass flow to the L.P. compressor remained, with the exception of gland leakage, throughout the unit. Unknown leakage could of course occur at such places as the heat exchanger and ducting flanges. To support this criticism, estimated horsepower losses in the transmission (based on lubricating oil flow and oil temperature rise), were in the region of 416 at a L.P. turbine speed of 3,780 r.p.m. compared to an estimated loss of approximately 800-1,000 horse power based on the gas turbine designer's calculations.

To determine the power losses within the gas turbine unit, two further "no-load" performance trials were carried

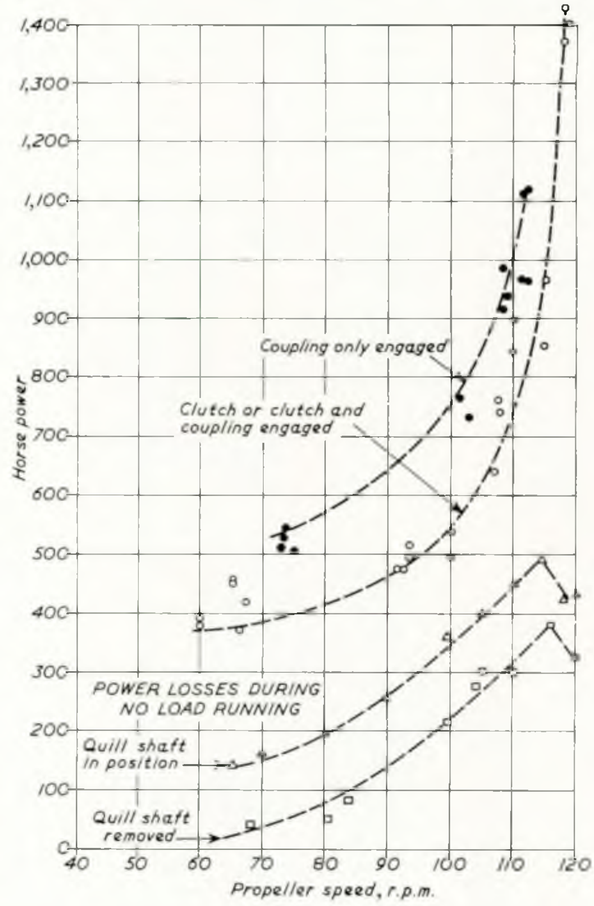


FIG. 13—*Difference between calculated coupling horsepower and measured shaft horsepower during sea trials*

				S
Group	Run number	Direction	Time on 2 miles mins. secs.	Speed ove ground (knots)
1	1	N	8–23·2	14·31
	2	S	8–32·5	14·05
2	3	N	8–51·2	13·55
	4	S	9–01·3	13·30
3	5	N	9–45·7	12·29
	6	S	9–23·9	12·77
4	7	N	10–19·1	11·64
	8	S	9–50·8	12·19
5	9*	N	5–48·9	10·32
	10*	S	5–26·4	11·03

*Runs 9 and 10 over one mile only.

1	1	N	8–32·5	14·05
	2	S	7–54·5	15·17
2	3	N	9–38·5	13·13
	4	S	8–15·3	14·54
3	5	N	9–55·4	12·09
	6	S	8–42·7	13·79
4	7	N	10–40·5	11·24
	8	S	9–01·7	13·29
5	9	N	11–47·1	10·20
	10	S	9–35·3	12·52

out during the enforced delay. The first trial was with the quill shaft disconnected and the second trial with the quill shaft connected, the unit being allowed to settle out to steady conditions of temperature and speed at various L.P. turbine speeds up to 3,840 r.p.m. Four hours were allowed to elapse after each increase of speed before performance readings were taken, the plotted results of these readings are illustrated in Fig. 13. If these results are accepted, then it is reasonable to assume similar losses under "load" conditions.

During these trials several H.P. ducting flanges were exposed and the blow-off valves were checked for air leakage, but no air leaks were detected. At a later date all heat exchanger tubes were pressure tested but again no defects were observed, so the assumption, in calculating gas horsepower output, that the mass flow through the unit is as measured at the inlet must be believed. The question of horsepower losses remained unanswered, with all parties concerned convinced in their own interpretation of these losses. The final solution of measuring the turbine output was put in hand, but unfortunately never carried out due to the ship's withdrawal from commercial service.

As had been previously agreed with Lloyd's Register, the primary transmission line was once again removed from the ship and completely dismantled for examination. This disclosed that the white metal seal in way of the forward end of the primary pinion, had badly wiped and the leaded nickel/bronze seal, contained in the astern converter, had rubbed on several occasions, causing slight grooving to an approximate depth of 0·008in. in the quill shaft. No satisfactory explanation could be given for the failure of the pinion seal, although it was suggested that some unbalancing effect could be taking place during the friction clutch engagement. It was interesting that the failure of this seal occurred without any apparent defect becoming noticeable in the operation of the transmission during sea trials. This confirmed the effectiveness of the leak off arrangements within the primary pinion preventing oil flow along the quill shaft to the ahead coupling, whilst operating on the astern converter. Once again both these seals were renewed, the primary pinion seal being renewed in leaded nickel/bronze and given a $\frac{1}{16}$-in. radial clearance. The friction clutch, ahead coupling and astern converter wheels were in good condition, and the balance of these components in both wet and dry conditions was found correct. Both couplings were crack detected without any defects being noted. The quill shaft was placed in the lathe and again found true.

The after quill shaft bearing was now found to be badly scored and this bearing was remetalled. Shotblast material was found embedded in the quill shaft thrust bearing pads with the resultant scoring of the thrust collar (Fig. 14). These thrust pads were renewed and the thrust collar machined.

The primary pinion forward journal bearing metal had wiped, so this bearing was remetalled and the effective length of the bearing metal modified, and greater radial clearance provided at the oil entry in a similar fashion to the after bearing.

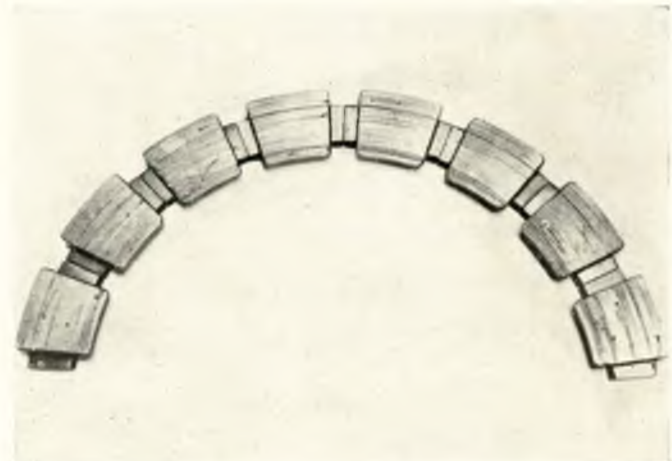


FIG. 14—*Thrust pads*

The spare H.P. turbine unit supplied to the ship was now fitted to ensure that no difficulties would be encountered in service if the occasion arose.

Subsequent examination of the original H.P. turbine, which had been in service for a total running time of 949·3 hours, found the blading to be in good condition, free of deposits. The first row of rotor blades had a rough surface from high temperature effects. Rotor radial blade clearances were in the region of 0·064in. minimum.

The compressor and turbine bearings were each exposed for examination, and with the exception of the H.P. compressor outlet bearing all were in good condition. The H.P. compressor outlet bearing was scored and so this was remetalled.

On completion of all outstanding work, the main unit was started, and the new H.P. turbine systematically heated over a period of fourteen hours to a gas inlet temperature of 600 deg. C., use being made this time of the blow-off valves to increase the inlet temperature from the normal idling temperature of 300 deg. C.

The only defect observed during this H.P. turbine proving trial was the high bearing metal temperature of the H.P. turbine outlet bearing, which rose to 72 deg. C. with an oil drain temperature of 51 deg. C. Examination of this disclosed that the journal was bearing at the ends only, so it was rebedded.

This completed the trials of the main propulsion unit during which time the unit had operated for a total of 973 hours 34 minutes, including over 518 hours with one or other of the transmission components engaged. Since the reconstruction of the astern converter, 942 separate engine movements were undertaken during manœuvring.

PART III

Auris IN SERVICE

The ship sailed from Birkenhead on the 20th August 1959, bound for Southampton where a number of demonstration runs were to be undertaken. Three hours after "full away" conditions had been attained it was noticed that the lubricating oil pressure at the gas turbine bearings and gearcase had fallen to 4·5lb./sq. in. and investigations showed an air lock had occurred in the "full flow" run downs from the 3,000 gallon capacity lubricating oil gravity supply tank (Fig. 15).

The gas turbine and quill shaft bearing lubricating oil main was supplied by three run down lines, and the gearcase lubricating oil main supplied by two run downs. In the event of a lubricating oil pump failure the gas turbine and quill shaft were thus supplied for 30 minutes and the gearcase for ten minutes. With the exception of the top take-off, or full flow lines, all run downs were fitted with a combined sight glass and flow indicator. It was the increased flow rate through these indicators which showed that an air lock had occurred within the top take-off lines. The main propulsion unit was shut down and the gravity tank allowed to empty in order to free these air locks. Whilst the gravity tank was being slowly replenished, air was purged from all available points in the lubricating oil systems and eventually the normal supply pressure of 9-10lb./sq. in. at the units was attained.

The aeration of the lubricating oil was a problem which was never overcome, despite the fitting of continuous air purging lines to the lubricating oil systems and the lubricating oil coolers, and resulted in four stoppages at sea while the ship was engaged on commercial trading. The cause of this aeration could be attributed to the basic design of the lubricating oil system, and in particular to the drain tank, the capacity of which allowed an insufficient depth of lubricating oil to be maintained at and around the pump suction. This was one of the inherent problems of the conversion, as the centre line of the shafts fixed the available space beneath the gearcase.

On continuing the voyage to Southampton, instrument readings indicated an unduly high propeller torque in relation to measured horsepower which could not be accounted for within the ship. One explanation was that the hull or propeller had become foul in the two months period the ship was lying in the fitting-out berth since docking after sea trials. In order to verify the condition of the hull and propeller, the ship was diverted to Falmouth where divers confirmed that the propeller was encrusted on all blades with shell type organisms to a depth of ¾-in. at the leading edge of the blades, but the hull plating was comparatively free of marine growth. By adjusting the fresh water ballast carried during the voyage, the ship was trimmed to an after draught of 12ft.,

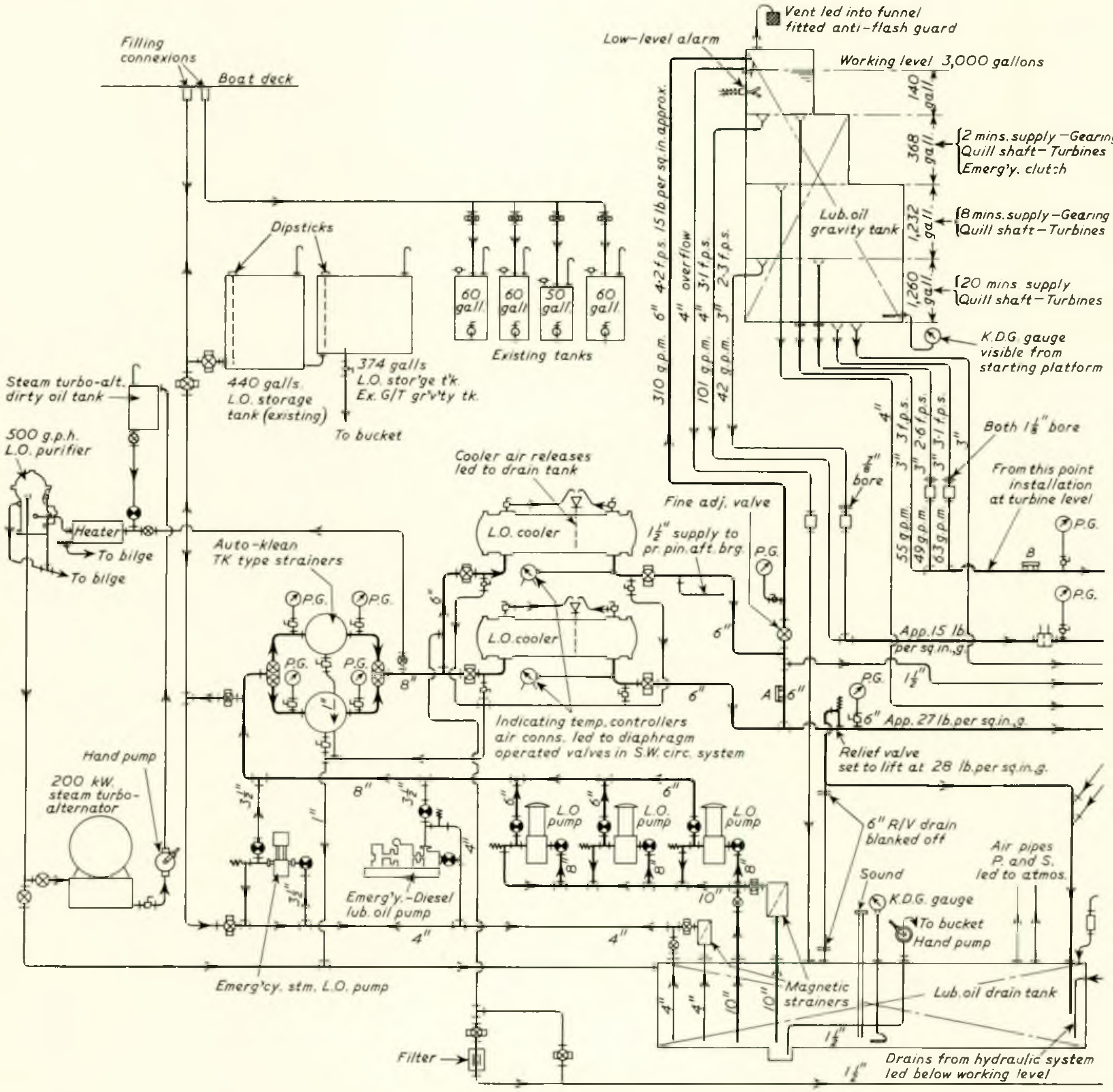


FIG. 15—*Lubricating oil diagram*

and the propeller blades cleaned with scrapers and acid in less than six hours. The ballast was re-adjusted and the voyage to Southampton completed, during which time performance readings indicated a recovery of approximately 600 horsepower by the cleaning of the propeller.

Despite the extensive stiffening of the Scotch boiler platform, this unit continued to vibrate at normal service speed, and the vibrations were later shown to be in the order of four per propeller revolution of ⅛in- amplitude. After the demonstration runs were successfully carried out at Southampton, the main unit was shut down to facilitate further minor modifications being made on the auxiliary boiler, for it was clear that vibration could not be prevented.

The ship left Southampton on the 2nd September 1959, and, after clearing the piloted channel west of the Needles, the friction clutch was engaged with the assistance of the booster pump at a decreasing L.P. turbine speed of 2,000 r.p.m. Upon engagement an excessive noise was heard at the forward end of the gearcase and the clutch was immediately disengaged. A subsequent clutch engagement a few minutes later was quite normal, and the L.P. turbine was accelerated to 3,500 r.p.m. The voyage continued under these conditions of clutch and coupling in operation without any mechanical defects being encountered. Slight reductions in ouput power became necessary to maintain the gas inlet temperature to the H.P. and L.P. turbines within the permissible maximum temperature of 650 deg. C. and 550 deg. C. respectively, due to the increasing ambient temperature.

After five days it was decided to try utilizing the cargo pump alternator for all electrical requirements. In order to

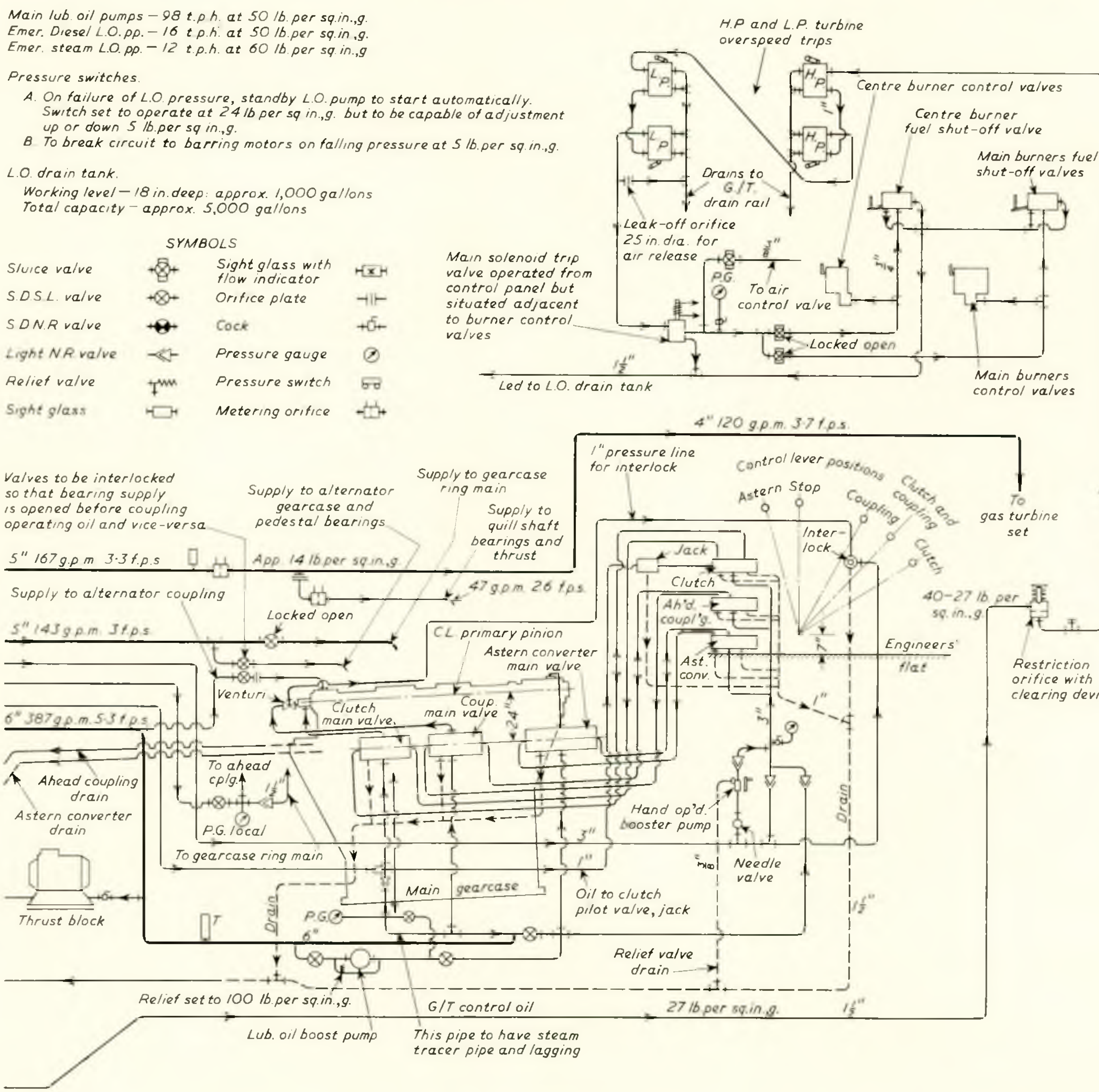


FIG. 15—*Lubricating oil diagram*

minimize the torque on the alternator during engagement, the L.P. turbine speed was reduced to 2,000 r.p.m. prior to flooding the alternator coupling. During the deceleration the friction clutch again slipped and was immediately disengaged. The re-engagement of the clutch was later carried out normally with the aid of the booster pump, and at a falling L.P. turbine speed of 2,000 r.p.m., but whilst the L.P. turbine was accelerating through the friction clutch critical (2,500-2,785 r.p.m. the clutch again slipped and was disconnected. Another attempt to engage the friction clutch was successfully carried out and the unit steadied to normal running conditions. After transferring the 120 kW load from the steam turbo-alternator to the cargo pump alternator, it was found that the temperature limitations of the H.P. and L.P. turbines, under much warmer ambient conditions, did not permit a high enough turbine speed to maintain sufficient cycles for satisfactory operation of the electrical auxiliaries. It was then decided to disengage the friction clutch and operate on the ahead coupling only which, with its inherent slip of about six per cent enabled a higher L.P. turbine speed to be maintained. This not only resulted in lower H.P. and L.P. turbine inlet temperatures due to increase mass flow, but the cycles of the cargo pump alternator were sufficiently high (57-58 c.p.s.) to maintain satisfactory output of all the electric auxiliaries. A further unforeseen advantage in utilizing the cargo pump alternator in place of the steam turbo-alternator, was the reduction in steam requirements. This enabled the firing of the auxiliary boiler to be suspended, adequate steam being generated by the waste heat boiler for normal daily steam requirements, including the operating of the steam soot blowers and the circulation of the auxiliary boiler through its hydrokineter. The pressure was maintained on this boiler in case of emergency

Advantage was taken of the prolonged stay in Curaçao (due to cargo difficulties) to inspect various components of the main engine. Every item was in excellent condition, except that the main evaporator tubes of the waste heat boiler were free to vibrate $\frac{3}{32}$-in. and in some instances the tube spacers had cut into the tubes.

Before leaving Curaçao both compressors were cleaned with kerosene, Teepol and fresh water in the normal manner.

During the return voyage from Curaçao, an interesting series of tests were carried out with the primary object of comparing the performance of the main propulsion unit whilst operating on the coupling or clutch, and with or without the cargo pumping alternator engaged.

The friction clutch was successfully engaged and performance readings taken when the unit had settled out to steady conditions at the maximum permissible H.P. turbine gas inlet temperature. The friction clutch was then disengaged, and as expected the L.P. turbine speed increased and the propeller speed slightly decreased. After the unit had settled out at these new conditions, further performance readings were taken and it was observed that all the gas temperatures had fallen by various amounts. Previously the whole plant had operated on the temperature limits, but it was now operating at a satisfactory condition.

The cargo pump alternator was then engaged and the auxiliary electrical load transferred from the steam turbo-alternator. The fuel rate was set as close as possible to that of the previous tests, and the plant again allowed to settle to steady conditions before performance readings were taken.

Analysis of the principal readings taken during these tests, are shown in Table VII, and indicate that for practical operation of the ship at propeller speeds up to 112 r.p.m., it was advantageous to operate on the ahead coupling with the cargo pump alternator engaged.

During the latter part of the return voyage, lower ambient temperatures permitted higher turbine speeds with reduced gas inlet temperatures. It was hoped that this opportunity would enable the performance of the plant to be determined with the friction clutch engaged and with the cargo pump alternator supplying all auxiliary power, but once again the friction clutch slipped.

TABLE VII.—COMPARATIVE TEST OF ALTERNATIVE METHODS OF TRANSMISSION

	TEMPERATURE			SPEEDS			s.h.p.	Electrical load kW.	Gas turbine fuel flow lb./hr.	Specific fuel consumption gas turbine lb./hr.	Approximate auxiliary boiler fuel consumption lb./hr.	Over all specific fuel consumption lb./s.h.p./hr.
	H.P. turbine in	L.P. turbine in	L.P. turbine out	H.P. turbine	L.P. turbine	Propeller						
Clutch engaged	651 deg. C.	545 deg. C.	371 deg. C.	5,660	3,433	107·3	3,829	122	2,475	0·646	260	0·685
Coupling only	622 deg. C.	510 deg. C.	341 deg. C.	5,620	3,596	105·2	3,553	122	2,460	0·692	260	0·732
Coupling and alternator	645 deg. C.	535 deg. C.	357 deg. C.	5,690	3,632	106·2	3,637	122	2,510	0·690	nil	0·661

Its initial engagement was quite satisfactory with the booster pump in use. The L.P. turbine was then accelerated slowly to 3,650 r.p.m. which had been found to be the minimum at which the alternator could operate. As paralleling was taking place, the friction clutch began to slip, the meter showing two per cent, so the test was abandoned.

At the time of slipping, the clutch was supplied with lubricating oil at 32lb./sq. in. pressure, the booster pump having been dispensed with after attaining relatively steady conditions. The shaft horse power was approximately 4,500 at a propeller speed of 114 r.p.m., and it appeared as if insufficient margin in the clutch torque capacity had been allowed in its design to warrant its further use, and therefore no further operation of the clutch was ever attempted. The total time the ship had operated with the clutch engaged on commercial trading was 134 hours, and a total of 31 engagements had been undertaken since commencement of trials.

During the outward and homeward bound voyages the performance of the turbines, compressors and heat exchangers was checked daily. Typical results may be seen in Table III. It soon became apparent that the compressor efficiency declined due to the salt atmosphere, heavy rain having the effect of improving the L.P. compressor efficiency, and reducing the H.P. compressor efficiency. The introduction of ground coconut shells into the air streams of both compressors on these occasions produced immediate results in a reduction of gas inlet temperatures and increased L.P. turbine and propeller speeds (Fig. 16).

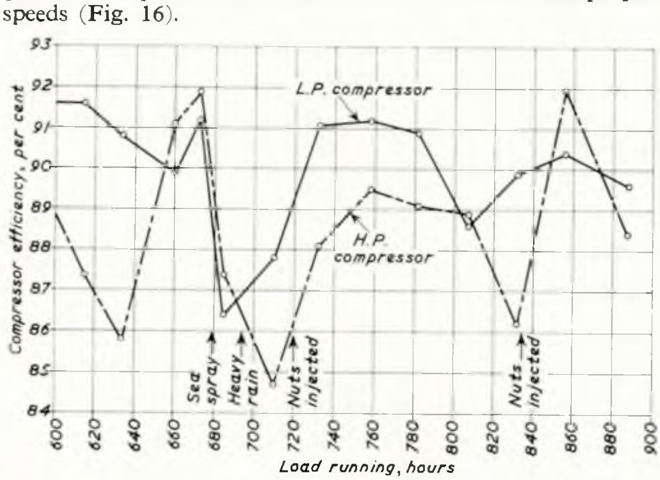


FIG. 16—*H.P. and L.P. compressor performance—Southampton to Curaçao*

Whilst operating in high humidity it was found that careful attention had to be paid to the intercooler drains as large quantities of fresh water quickly accumulated. It became a standard practice to keep the drain valves at the H.P. compressor side of the intercooler in a partially open position in rainy weather to prevent the compressor deposits being moved down the compressor. This air leakage was obviously a loss from the cycle, but could be overcome with a suitable trap arrangement with the additional possibility of a fresh water supply.

The statistics of the outward and homeward voyages were as tabulated below:

	Southampton to Curaçao	Curaçao to Thameshaven
Distance steamed (pilot to pilot) ...	4,068 miles	4,182 miles
Distance steamed (berth to berth) ...	4,094 miles	4,256·5 miles
Engine miles on passage (pilot to pilot)	4,366 miles	4,612 miles
Steaming time (pilot to pilot) ...	310·9 hours	326·7 hours
Steaming time (berth to berth) ...	314·8 hours	334·4 hours
Average propeller slip	6·8 per cent	9·3 per cent
Average shaft horsepower	3,545	3,771
Average propeller revolutions ...	106·6 r.p.m.	107·1 r.p.m.
Average speed (pilot to pilot) ...	13·08 knots	12·8 knots
Total propeller revolutions ...	1,988,670	2,099,930
Average air temperature	27·5 deg. C.	26·1 deg. C.
Fuel oil consumed (auxiliary boiler)	25·1 tons	11·4 tons
Gas oil consumed (gas turbine) ...	324·8 tons	366·9 tons
Specific fuel consumption (gas turbine)	0·660lb./s.h.p./hr.	0·667lb./s.h.p./hr.
Specific fuel consumption all purposes	*0·711lb./s.h.p./hr.	0·688lb./s.h.p./hr.

* *Note:*—Tank cleaning operations were carried out during outward bound voyage.

After the successful conclusion of the maiden voyage, the ship was engaged on commercial trading between ports in Great Britain and N.W. Europe, necessitating the operation of the gas turbine machinery under various ambient and extreme weather conditions. One of the first ports visited was Svolvaer, in Northern Norway, whose narrow entrance and confined manœuvring space inside the harbour is well known. The ship behaved remarkably well and this was commented upon by the local authorities.

Due to the ship trading on the coast in winter, sometimes in conditions of bad visibility, it was considered advisable to operate the plant in a condition where telegraph movements could be instantly applied. The cargo pump alternator was therefore never utilized for the auxiliary electrical requirements of the ship whilst coasting, although it was used extensively in port for cargo and ballast requirements, and for the auxiliary load when the steam turbo-alternator was shut down for minor adjustments and routine examinations.

During the next few weeks the machinery ran continuously except for one unexpected emergency.

The *Auris* was being manœuvred in extremely heavy seas off Trevose Head while taking part in a search for a distressed ship. The main engine suddenly stopped due to the loss of fuel oil pressure. The fault was quickly traced to the solenoid control valve mechanism which was promptly wedged with wood. This example however, illustrates the danger of electrical/mechanical devices being fitted with either insufficient robustness or lack of effective nut locking. The engine had by this time operated for 2,700 hours.

On the 29th November 1959, some 357 hours later, whilst the ship was being navigated into Pernis, the gas turbine unit was once again shut down following loss of lubricating oil pressure at the bearings, due to aeration of the lubricating oil systems. While the L.P. unit was coming to rest a noise was heard in the vicinity of the L.P. compressor which seemed to indicate a loss of radial blade clearances. Subsequent checks on these clearances with the clearance monitor did not support this belief, and when normal lubricating oil pressure had been restored, the main unit was re-started and the ship berthed without anything untoward happening. The gas turbine driven alternator was then utilized to discharge cargo tank ballast, the L.P. turbine speed being maintained at 3,500 r.p.m. When the cargo pump was no longer required, the alternator coupling was drained, but attempts to decrease the L.P. turbine speed resulted in violent surging of the L.P. compressor which could not be eliminated even by opening the L.P. blow-off valve. The gas turbine was shut down and the L.P. unit finally came to rest after numerous violent surges. Examination of the compressor blading from inside the inlet casing, and by means of the introscope in stage 10 clearance monitor probe holes, indicated that stage 12 rotor and/or stator blades had been severely bent. Work was then started to open up the compressor casing.

When the top half of the compressor casing had been lifted it was found that two rotor blades, almost diametrically opposite, in stage 12 had fractured about 1 cm. from the root platform and considerable damage had occurred to stage 12 stators which was at first thought to be due to the failure of the two rotor blades (Fig. 17). Further examination however disclosed nine stator blades to be missing and their fractures at the root radius were identical to those of the rotor blades (Fig. 18). Later after crack testing fluids had been used, four more cracked blades were found in stage 12 stator, and one of these was easily removed by hand. Very little physical damage had occurred to this particular blade but its fracture was again identical to the other stator blades. All the fractures were due to fatigue, the fractures having started at the leading edge and continued for about three-quarters of the chord length. In the case of the stator blades it appeared as if the blades had gradually twisted more axial, rubbed on the rotor, finally being hit by a rotor blade about three-eighths of an inch from the leading edge, the blades being clearly marked where this had occurred. Unfortunately stage 12 rotor blades had damaged stage 11 stator blades and the trailing edge of stage 11 rotor blades had all been "nicked" near where the blades joined their root platforms, in fact where any fatigue failure would take place.

To facilitate the search for the missing blades an access hole was cut in the crossover ducting between the L.P. compressor outlet and the intercooler, and all the missing stator blades were found by the inlet side of the intercooler. One rotor blade, which was bent nearly double and had metal missing, was found about 18in. from the intercooler, but despite an extensive search involving the removal of all the intercooler and split ducting doors the second rotor blade was never found. A quantity of small pieces of blade material was recovered, but only sufficient to account for the metal missing from the recovered rotor blade.

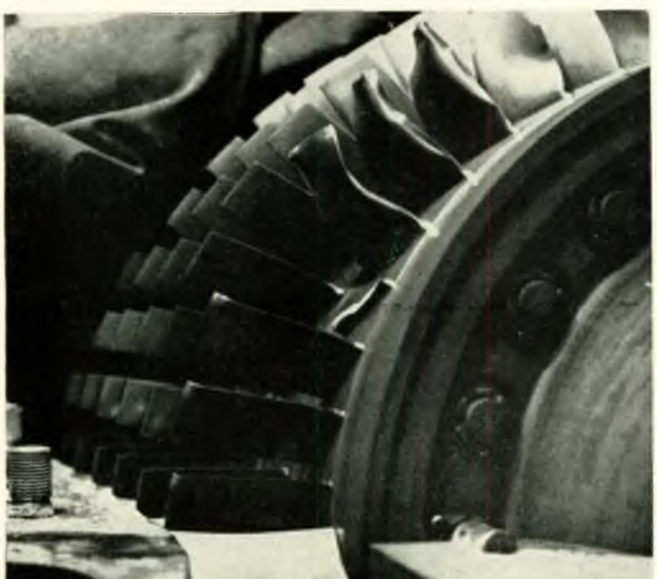


FIG. 17—*Failure of L.P. compressor blades rotor*

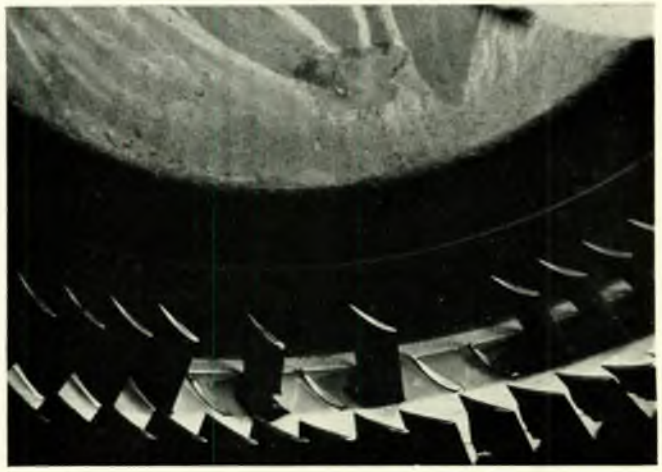


FIG. 18—*Failure of L.P. compressor blades stator*

The rotor was removed from the ship, and after crack testing of stages 11 and 12, these blades were machined off in place by cutting through the blades near to their root platform. The two stub ends of the missing blades were removed by hacksaw and together with all the recovered blades were retained for further examination. Stages 11 and 12 stator blades were removed and the blade profile cut off from the blade roots by milling cutters, the blade roots and spacers being replaced in the casing. This was not an easy job because the material of the blades was stainless FDP.

All remaining rotor and stator blades were crack tested without any further defects being located and after thorough cleaning of all blades and rebalancing of the rotor, reassembly of the compressor was carried out.

It was later confirmed by manufacturers that the blade failures were definitely due to fatigue, the fracture of the stator blades commencing before those of the rotor blades, but complete failure occurred more rapidly in the rotor blades than in the stator. Tests on blades, from the original production batches, found stage 12 rotor blades to have a resonant frequency which coincided with seven vibrations per revolution at about 3,200 r.p.m., stage 11 rotor blades were just outside the working range, but stage 12 stator blades had a resonant vibration which could not be satisfactorily explained as it was neither due to bending or torsion. Further investigation suggested that stage 10 rotor blades might have a resonant vibration at 3,350 r.p.m., and a suspected resonant in stage 10 stator blades at 3,450 r.p.m., and it was considered advisable to avoid operating the L.P. unit between the speeds 3,250-3,550 r.p.m. A second harmonic vibration was revealed in stages 11 and 12 of the order of 2,700 cycles/sec. and small amplitude, in the running range.

At this stage the possible causes of the blade failures were considered to be either the second harmonic vibrations, in which case it was satisfactory to continue running with the banned speed range, or rotating stall connected with the diffuser, in which case it was still safe to operate the unit, as there was now a converging passage before the diffuser, due to the removal of stages 11 and 12.

A third possible cause was that failure was due to rotating stall connected with the fundamental frequency, but this was considered most unlikely as the designed blading had very low stress values and even with the vibrations they should not have failed. One other peculiarity was that the unit had operated for over 3,050 hours before any blade failure occurred which must have involved a fantastic number of reversals with frequencies of the order that had been revealed.

The removal of the L.P. compressor rotor necessitated the opening up of the inlet and outlet end bearings, and it was found that the bearing metal and also the journals were badly scored, presumably through dirt within the lubricating oil system. Examination of the L.P. turbine thrust and outlet end bearings disclosed slight scoring on the thrust collar also on the thrust and surge pads, but the thrust pads were all taking the load evenly. This information was of value in view of previous experiences in 1955

Whilst this work on the L.P. unit was being carried out, opportunity was taken to examine the other components of the gas turbine unit.

The combustion chamber was opened and all the refractories were examined and found in excellent condition with only a few minor surface cracks. Whilst the combustion chamber was open, the H.P. turbine inlet scroll and first stage blading was examined and, as expected, a quantity of grey scale was cracking off the blading and casing. The first stage rotor radial blade clearances were found to be 0·083in. at the bottom, 0·052in. at the casing horizontal joint and 0·062in. slightly above and below the joint, the nominal clearance being 0·080in.

The H.P. compressor blading was found in a dirty condition, and some scoring was seen on the leading back face of the rotor blades; probably a result of condensed water passing through the unit with the air stream. The first stage blade tip clearances were 0·031in. at the top, 0·025in. at the bottom and 0·036in. at the horizontal joint.

The epoxy coal tar paint which had been used on the inside surface of the split trunking had not proved very satisfactory, several large areas of it having peeled off. This was possibly due to the quantity of condensed fresh water which as already mentioned, under certain atmospheric conditions is blown through this ducting.

The waste heat boiler evaporators and the gas turbine heat exchanger were water washed, and the waste heat boiler pressure tested to 180lb./sq. in. One tube in the main evaporator unit was found to be leaking, and this was plugged and the boiler re-tested.

After completion of all repairs, carried out in less than 11 days, and prior to starting the gas turbine unit, both compressor units and intercooler were Teepol and water washed.

At this point no one could be quite certain how the engine would operate. The peak temperature during the start of the main unit was exceptionally low (470 deg. C.) due to its cleanliness, and both units accelerated quite normally to idling conditions of speed and temperatures. During the following twelve hours the speed of the L.P. turbine was slowly increased and at 3,100 r.p.m. a violent surge was experienced which was immediately cured by opening the L.P. blow-off valve a quarter of a turn. The H.P. turbine gas inlet temperature was raised to 600 deg. C. and the ship proceeded on a twenty four hour sea trial. After ballasting to a mean draft of 22ft., the propeller shaft speed was increased to 107 r.p.m., and the machinery allowed to settle out to steady conditions for performance readings to be taken. A manœuvring trial was carried out including several direct reversals of the propeller. All movements were completely satisfactorily and the ship proceeded back to the repair berth, during which time the cargo pump alternator was again utilized for discharging ballast.

The performance of the gas turbine unit from these trials, was exceptionally satisfactory and no apparent changes in its operation were noted as a result of the removal of stages 11 and 12 rotor and stator blades of the L.P. compressor unit. The tendency for the compressor to surge during idling or light load conditions necessitated the L.P. blow-off valve being partially open during manœuvring or idling periods, also when the cargo pump alternator was in use.

Due to the fact that the L.P. compressor was now working very close to the surge point during no load, or light load conditions, the cleanliness of the unit became of paramount importance. This was aggravated by the fact that the ship resumed commercial trading between N.W. European ports so that the gas turbine unit was operated for relatively long periods in detrimental atmospheric conditions normally associated with loading or discharge ports. Owing to the nature of the cargoes carried by the ship, port regulations, and the comparatively short stay in one port, the engine was only shut down on two occasions in the three months following these repairs. This clearly limited the occasions on which the compressors could be water washed, so the introduction of ground coconut shell into the air streams to the compressors became the normal method of cleaning these units whenever required. This was usually determined by the daily checks which were continued on the gas turbine performance.

During the period of commercial trading, the ship operated in ambients ranging from minus 10 deg. C. to plus 33 deg. C., the gas temperatures throughout the unit varying according to the ambient temperature for any given horsepower. The instrumentation of the unit was such that any fluctuation in propeller shaft torque, such as would occur in shallow waters, or through erratic steering of the ship, was quickly observed by a fluctuation of gas temperatures, the principal temperatures being continuously recorded electronically every three minutes.

As the period of commercial trading progressed, a gradual increase in measured propeller shaft torque for normal propeller service speed indicated that a deterioration in the condition of the hull due to marine growth or roughening was taking place.

It was significant that after a voyage to Sweden which entailed the ship steaming for approximately ten miles through pack ice, to and from the discharge berth, (Fig. 19) a marked decrease in the propeller shaft torque was noted (Fig. 20). Immediately prior to the voyage to Sweden, approximately 16·4 per cent extra horsepower was developed to maintain a service speed of 108 propeller r.p.m., compared to the power developed during the performance trials. After steaming through ice this had dropped to approximately 9·1 per cent.

FIG. 19—*Pack ice in Sweden*

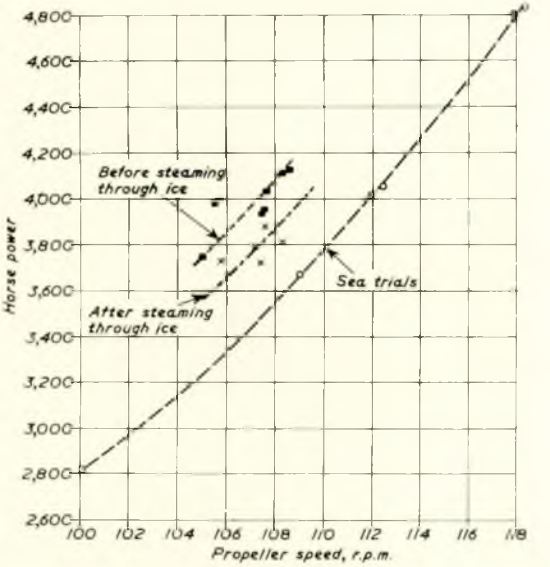


FIG. 20—*Fall off in ship performance due to hull deterioration*

When dry-docked a few weeks later it was seen that a fairly large area of the hull had been stripped of all paint, scale and marine growth, as effectively as if it had been shot-blasted.

Early in March 1960, the ship was unfortunately delayed whilst repairs were carried out. The hull plating had been damaged whilst the ship was being loaded in the Orkneys in adverse weather conditions. The opportunity was however taken to investigate further the cause of the L.P. compressor blade failures, and to effect miscellaneous repairs and surveys normally associated with the annual refit.

To facilitate the fitting of strain gauges to certain stator blades of the L.P. compressor, the top half casing of this unit was lifted and three stator blades from stages 2, 5 and 10 were removed from the top half casing. The second and third blade from these stages were selected to carry the strain gauges, and a $\frac{1}{16}$in. × $\frac{1}{16}$in. slot was machined across their root face to accommodate the leads from the crystal, a $\frac{1}{8}$-in. diameter hole being drilled through the casing in way of these slots. The nine blades which had been removed, together with all accessible stator blades in the top half casing, were crack tested without any defects becoming apparent. The strain gauge crystals were then cemented onto the six selected blades, and all blades and spacers replaced and secured. To determine the air flow to and within the diffuser, two holes were drilled through the casing in order to insert hot wire anemometers. Whilst this work was in progress, various stator blades were excited with special equipment by the blade manufacturers and the fundamental frequencies measured. This was found to be in the order of 370 cycles/sec. Prior to replacing the top half casing all rotor blades were cleaned and crack tested. All stator blades in the top half casing and the stator blades in the lower half casing, which were accessible were similarly treated. Whilst this was being done the ninth stator blade from the starboard horizontal joint of stage 10 in the lower half of the casing was found to be missing, and a subsequent search of the ducting to the intercooler found the blade to be lying close to the horizontal bellows. Examination of the blade revealed the fracture to be identical in appearance to those seen at Rotterdam in November. The discovery of this missing blade virtually sealed the fate of the ship.

To establish the cause of these failures, extensive electronic instrumentation for vibration analysis was installed, and on completion of all outstanding work, the gas turbine unit was started up and a series of vibration tests were carried out over a range of L.P. turbine speeds from 2,000 to 3,800 r.p.m.

The tests were fairly conclusive and indicated that the diffuser design was the basic cause of the failures, the fundamental blade frequency of 370 cycles/sec. becoming resonant and also rotating in the diffuser. The blades were found to be slightly overloaded, the 10 stages working to the same pressure ratio as twelve stages previously and the diffuser effect was imposed on top of this overloading despite the lengthening of the diffuser by virtue of the converging passage formed by the removal of stages 11 and 12.

From the strain gauges fitted in stages 2, 5 and 10 it appeared that the amplitude of the vibration decreased through the compressor towards the inlet stages, the amplitude being reduced to approximately 1/12 by use of the L.P. blow-off valve.

Other work and inspections undertaken during the hull repairs included the survey of the H.P. turbine. The combustion chamber head was lowered and the H.P. turbine first stage blading and inlet scroll examined by climbing up the inlet tube to the turbine. The blading was in good condition and the tip clearances were 0·087in. at the top, 0·084in. at the bottom, and 0·063in. and 0·060in. above and below the starboard horizontal joint respectively, indicating that although some distortion of the casing had taken place, the new design was vastly superior to the original turbine installed in 1951. Several heat cracks were observed in the inlet scroll nose casting, but were not considered to be of a serious nature. The three crossover pipes between the turbines were removed to examine the outlet bearing and journal which was found, in common with the inlet bearing and journal, to be in excellent condition. The removal of the crossover pipes enabled the last stage blading to be examined, which again was found in good condition with a tip clearance of 0·060in. at the top and 0·075in. at the port horizontal joint.

The combustion chamber refractories were examined and found in a comparatively good condition and suitable for further service, as was the cast refractory in the combustion chamber head, the mixing zone, and conical heat shield.

With the removal of the crossover lines between the turbines, the first stage blading of the L.P. turbine could also be inspected, and was seen to be in good condition and light brown in colour. Examination of the exhaust casing of the turbine disclosed that the L.P. exhaust bellows heat shield was extensively damaged on the forward side, the securing studs having sheared, and the plate ripped into three sections due to vibration, the pieces having fallen onto the exhaust guide vanes of the turbine. A new section of heat shield was fashioned, stiffening being provided by doubling the lower edge of the plate, and the section welded in place.

The waste heat boiler evaporator tubes and heat exchanger were water washed and modifications undertaken to prevent the vibration of the waste heat boiler tubes. The comb type supports fitted prior to the ship's departure from the shipyard and which had proved completely ineffective were removed, and a $1\frac{1}{2}$in. × $\frac{1}{4}$in. bar was welded across and to each tube, the bars being supported by lugs welded to the casing and allowance made for thermal expansion. The tube which had previously been found ruptured was not replaced, but several grooves which had been worn into the tubes by the comb support were dressed up by weld metal.

Shortly after the ship had commenced commercial trading, discussions were resumed with all interested parties on the question of horse power output of the gas turbine unit and the gearbox. It was finally decided that a Van Milligan Optical Torque Meter, the accuracy of which was agreed to by the parties concerned, would be fitted between the L.P. turbine and the gearbox in place of the gear type flexible coupling, so that the output power of the L.P. turbine could be accurately measured, and the power losses within the gearbox ascertained. Although the instrument was not available at the time of the docking, but in order to expedite its fitting when it was delivered, modifications were carried out to the L.P. compressor outlet bearing and flexible coupling casing.

Unfortunately, in view of the L.P. compressor blade failures it was not considered advisable to continue operating the ship on commercial trade until such time as it became practical to warrant the expenditure involved in correcting the L.P. compressor diffuser defects. The manufacturers were confident these could be corrected by a comparatively simple modification involving the fitting of an air splitter to the existing diffuser and replacing stages 10, 11 and 12. Stage 10 required replacing owing to the latest blade failure making the blades in that stage suspect of metal fatigue. Thus the question of the gearbox efficiency once again remained unresolved, and the effectiveness of the latest modifications to the waste heat boiler, as regards vibrations of the main evaporator tubes under normal service conditions, undetermined.

The ship proceeded under her own power from the Tyne, where the refit had been carried out, at 13·9 knots to a lay-up anchorage in the Blackwater, where upon arrival all the main and auxiliary machinery was shut down and steps undertaken to preserve it.

Since it had been installed in the ship, the gas turbine unit had operated for a total of 5,238 hours, including 2,991 hours during which time one or other of the hydraulic couplings had been engaged. In the course of commercial trading, 39 ports were visited involving a steaming distance between pilots of 21,145 miles and approximately 2,500 miles of harbour steaming, during which time 3,965 separate engine movements were successfully carried out. This included 1,216 engagements of either the ahead or astern couplings whilst the propeller shaft was stationary, and 173 direct reversals of the propeller shaft.

DISCUSSION AND CONCLUSIONS

The behaviour of the machinery well justified the early expectations, and as an experimental unit it has completely justified its inception. Lack of time and falling freight rates prevented its commercial acceptance.

In 1955 no one foresaw that 12,000 ton tankers would be commercially obsolete five years later, or appreciated that labour disputes and manufacturing difficulties would absorb three years for "installation".

In retrospect the decision to fit a single set of machinery in *Auris* as against twin units in *Hemisinus* was technically correct, although the advent of blow-off valves now makes synchronous electric propulsion an interesting proposition. In practice the hull did not justify the machinery; much valuable time was lost, a repair is never as good as new, and the layout was unnecessarily complicated by the use of existing auxiliaries.

The fact that the *Auris* eventually went to sea as a commercial proposition is interesting and there is no doubt that the machinery designed in 1953 could compete with the latest design of comparable size steam turbine machinery in 1960. All purpose fuel figures quoted without prejudice on the *Auris* (see Table III) are as accurate as instruments permit. The same fuel meter has since been used on two identical 18,000 ton tankers each about a year old, with an all purpose fuel rate of (*Aluco*) 0·64lb./s.h.p./hr. no evaporators and (*Arianta*) 0·73lb./s.h.p./hr. and two evaporators.

In the case of the two steam turbine ships they were operating slightly below their service powers.

The *Auris* was burning gas oil during the period under review, but there was every confidence that the ship could burn fuel oil successfully, because in this particular installation, unlike the majority of gas turbine plant, special attention was given to the design of the turbine which incorporated several novel features specifically to minimize the effects of fuel oil ash. It was expected that an initial roughening of the turbine blade surface might cause a fall off of about 2 per cent on the turbine performance, but otherwise the performance could be maintained with any fuel oil. Extensive tests to prove this had been planned.

The gearbox losses remain an important and unsolved factor. Tests for measuring this were also planned but unfortunately never carried out. If the losses were in fact 1,000 s.h.p. then the gas turbine did even better than expected. If on the other hand the gearbox losses were "only 400 s.h.p." then some other explanation is required for Fig. 13.

The *Auris* machinery is a prototype, and it is a remarkable achievement on the part of each of the three organizations involved that it was installed and ran in a ship, without any shore testing. The initial cost appeared high compared to normal commercial ships' machinery, often well tried and simply tailored to suit the ship in question. Compared to prototype gas turbine engines in the aircraft industry the cost was a fraction.

The hydraulic machinery well justified its inclusion as a prototype. Lessons learnt from the *Auris* have already been incorporated into new designs and rig tested by Pametrada, showing better efficiency and cheaper initial cost. The authors are convinced that the ship manœuvred as well as any ship afloat, but a new design of friction clutch must be proved on full scale tests at sea to justify hydraulic transmission.

The failure of the compressor blades was a great disappointment but not insuperable. Only time could prove the efficiency of any modification. Although this failure advanced the date of laying-up the ship, it was not the direct cause. By this time it was intended that the ship would only operate for 12 months.

This type of failure is probably at present only of passing interest to the shipping industry, but it may be of far more vital interest and consequence to other users of axial compressors, such as aircraft and industrial machines.

Designs for the next stage of gas turbine machinery show that instead of just breaking even with steam machinery of comparative power, they would be distinctly ahead even with conservative temperatures. Unfortunately the capital cost of gas turbines on limited production is considerably higher than comparable conventional machinery. It can not yet be foreseen that improved materials, design, or manufacture will change this differential. The aim should be no steam, no circulating water and simplicity. Fears that specialists and specially trained ship's engineers would be required were quite unfounded.

The authors consider that the *Auris* is only the second stage in the development of marine gas turbines. Steam turbines have their distortion, boilers and evaporator troubles; Diesel engines have cylinder liners and lubricating oil bills. Both are on their limits of design and the gas turbine is just beginning.

ACKNOWLEDGEMENT

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Discussion

CAPTAIN H. FARQUHAR-ATKINS, D.S.O., D.S.C., R.N. (Member) said that he felt unworthy of the privilege of opening the discussion as his connexion with gas turbines had ended eight years ago. However, he would speak his mind and there would be plenty of time for the experts to contradict him afterwards.

They would miss those two great enthusiasts, Mr. Lamb and Mr. Forsling, who played such great parts in *Auris*.

Would Mr. Duggan and Mr. Howell forgive him when he said that the greatest value of the paper was that it illustrated so well the weaknesses of the gas turbine for ship propulsion? No doubt these were not insuperable, but were people in the shipping industry prepared to pay the price, and, if so, could they get their money back? The same questions plagued them even more in trying to fit nuclear power into ships.

This paper was most welcome and timely, as the Institute looked forward to Dr. Richards' paper* on "High Temperature Gas Cooled Reactors for Marine Propulsion". Dr. Richards proposed the use of steam turbines but if such a reactor could be used with a gas turbine in a closed circuit, and the gas temperature was 1,200 deg. F. or above, better thermal efficiencies would be obtainable than with the best marine steam turbines, using the present limits of temperature. However, the steam temperature was kept down to limit the vanadium attack on the superheaters and to avoid birds nesting in them, as well as to avoid the increased use of austenitic steel and other expensive high temperature materials. With a reactor, vanadium and gas passage fouling would no longer be a worry, so the temperature limits of both steam and gas cooled reactor systems would depend solely on fuel canning and the structural materials and how they stood up to steam or gas at high temperature, and (for parts inside the shield) under irradiation as well. Incidentally, the greatest triumph of the first *Auris* gas turbine was the burning of residual fuel oil. Would the brick-lined combustion chamber have survived the attack of vanadium and sodium?

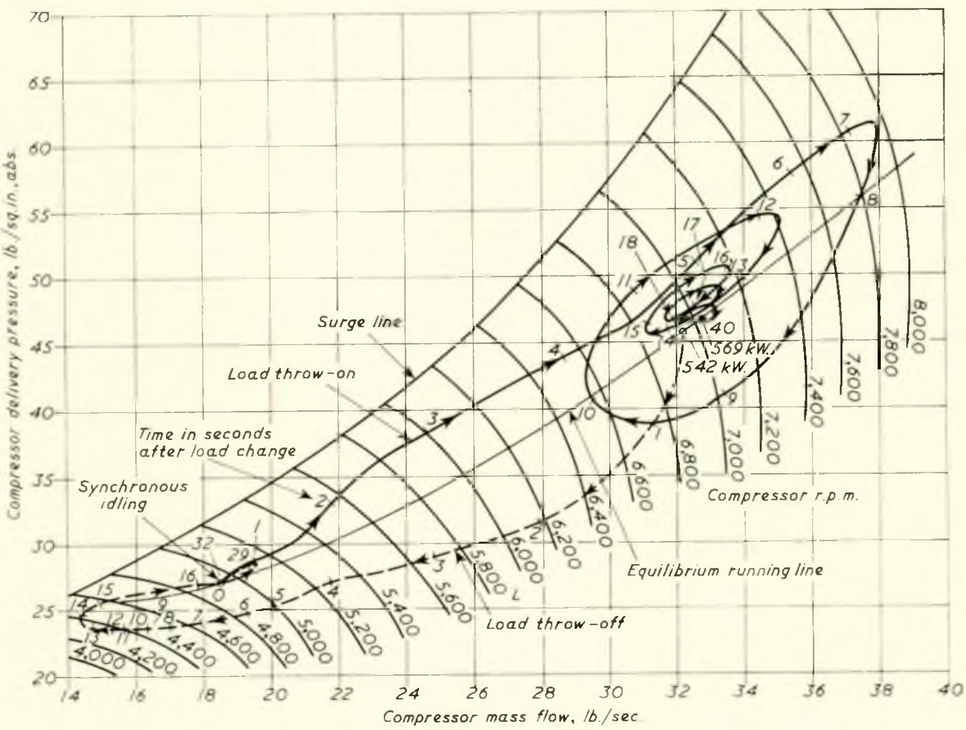


FIG. 21—*Allen 1 MW gas turbo-alternator*

Supposing the closed-cycle gas turbine did give better economy, theoretically, than steam, how would it work in practice? Its main disadvantages were the use of axial compressors, the need for gas tightness, and unreliable astern power.

Auris was made half a century later than *Turbinia*, partly through lack of high temperature materials, but also because it was much more difficult to compress gas than to pump

* Published in J. Jnt. P. Nuc. Mar. Prop., Vol. 6, No. 1, April 1962.

water. It was such a pity that an axial compressor looked rather like a Parsons turbine, and one tended to assume that it was as well behaved. Far from it. It was an over-bred, temperamental racehorse, finicky about its food and the very air it breathed, and needed long, expensive training compared with a stolid, dependable Clydesdale.

To remind those who were not gas turbine experts of the characteristics of an axial compressor, some results from an excellent design, for the Allen 1MW gas turbo-alternator were shown in Fig. 21. Compressor delivery pressure was plotted against compressor mass flow at various compressor speeds. The equilibrium running line for the clean compressor with air at its suction at normal temperature and pressure and the surge line, could be seen. The full line showed the running line for a load throw-on from idling to half-power, and the dotted line a load throw-off from half-power to idling. Full loads stop on or off would cause surge. In the tropics, where the air to the compressor suction was hotter and less dense, or when the compressor got dirty and its efficiency dropped, as in Fig. 16 of the paper, the running lines were pushed up to the left, nearer the surge line, and very likely the throw-on and throw-off lines for half-power would cross the surge line. When this happened the compressor might stumble a bit, blow off and jog on again, like the old grey mare, or the whole gas turbine, after some shattering pulsations, might grind to a halt. The starting motor or engine would have to be engaged, which usually meant waiting until the compressor stopped, and the whole starting cycle performed. The delay might be inconvenient at sea. In *Auris* the blow-off valves seemed to have prevented surge until after the blading failures. Had the compressors not been washed so often—or coconuts not proved effective scrubbers—manœuvring would have become "dicey". The analogy in a steamship was to fit only one feed pump which was liable to stop at any sudden change of power.

In a closed cycle, the suction pressure could be varied by letting more gas in or out of the circuit, but a sudden change of power might catch the compressor in a vulnerable state and push it over its surge line. The compressor in a closed cycle should not have to cope with salt or water but oil would probably reach its blades from the bearings in time, reduce its efficiency and push its running line nearer to surge.

In H.M.S. *Ashanti,* which was just doing her completion trials, the watchkeeper on the gas turbo-alternator could tell if he was going to have fried chips for supper as the maximum output dropped 50 kW (10 per cent) due to the cooking oil on the compressor blades.

If carbon dioxide was the coolant and was in contact with graphite, mass transfer of graphite to the compressor blades and heat exchangers would take place.

The effect of fission products was difficult to estimate but some would probably be sticky and harmful. No part of the primary circuit could be cleaned while running, as radio-active coconuts would add to the disposal problems and water might increase the gas pressures and the reactivity of the reactor unduly and cause corrosion.

Compressors were, of course, subject to static or rotating stalls in any of their rows of stator or rotor blades as well as to surging. The blades themselves might also vibrate in various modes if the speed of the compressor gave a resonant frequency and then fail by fatigue. This had been well demonstrated in *Auris,* as described on page 110. The fact that blade failure happened just after 3,000 hours running was interesting, although he realized that the first aircraft compressor ran for very much longer. At Farnborough in 1961 several gas turbines bore proud notices to say, "This engine has completed 3,000 hours without overhaul". In the aviation world this was great news, but did they think much of a steam turbine running for four months? Alterations to *Auris*' L.P. compressor blading would pobably only transfer stall to another row of blades which might fail in turn. Even if the engines of *Auris* and *John Sergeant* were considered satisfactory it was still a big step from them to the large closed-cycle gas turbine of three of four times the power that would be needed in a nuclear ship. To get reliability this development could not be done on the cheap as *Auris* was, neither could the marine industry afford the vast sums spent, largely provided by the Government, on developing gas turbines for aircraft. Far more could be done at sea by spending money on improving the design of their steam turbines than by dabbling with high temperature gas turbines. Gas turbine blades were the result of thousands of man-years of cascade, wind tunnel and running tests. Marine steam turbine blades had hardly altered since the days of Sir Charles Parsons. The steam turbine industry in this country had only trifling research facilities compared with those available for gas turbines. Why should not the National Gas Turbine Establishment apply its techniques to the improvement of steam turbines? Gas turbine engineers, in their search for high efficiency by increasing temperature, had introduced blade cooling and the cooling of stressed casings and pipes. Steam turbine designers had hardly had to bother with such ideas, but if higher temperatures were really thought worth while, perhaps they should.

He had little doubt that the mysterious loss of power in *Auris* was due to air leakage, probably in the air compressors, horizontal joints and/or heat exchangers, and suggested that the aeration of the lubricating oil might have been due to air or gas from the compressors or the turbines finding its way somehow into the bearings. It might prove possible for a while to make a gas turbine system leaktight enough to use helium, but tremendous development would be needed to make a system with eight or more shaft glands and a very hot heat exchanger which would remain tight enough in a ship during years of service.

In steamships the astern turbine gave probably the most reliable astern power possible. The only time he could remember failing to provide astern power entering harbour was after his ship had nudged a sandbank off the Palisados going into Kingston, Jamaica; all condensers filled with sand and they lost all vacuum.

In *Auris* the sad story of the hydraulic couplings gave no confidence, and *John Sergeant's* convertible pitch propeller lasted only a few months before the boss failed in fatigue, which meant a month in dock.

In spite of the pioneer work on *Auris,* the development of a reliable marine gas turbine for a nuclear ship would cost about as much as the development of the reactor. To be good enough for a ship a gas cooled reactor must prove itself superior, when used with a heat exchanger to raise steam, to reactors using other coolants, whether they used steam in a direct or an indirect cycle.

The recent rejection of the tenders for reactors for a ship had been anticipated for so long that the blow was deadened. Nevertheless, $2\frac{1}{2}$ years had been lost and between all the firms, close on £1 million, and hundreds of man-years of their best designers, had been wasted.

There was a glimmer of hope in the Research and Development programme announced by Mr. Marples, provided that the A.E.A. placed contracts at once with industry.

There were distinguished proponents of the gas turbine present who might dispute his conviction that it would not displace the steam turbine for the main propulsion of large ships. The habitat of the gas turbine was the sky, where it was unrivalled. It was a bird, not a fish.

Thanks were due to the authors for their most frank and lucid paper, as well as sympathy for the fact that economics had laid the ship up and stopped the fascinating work.

FREGATTENKAPITAEN F. MOELLER (German Navy) began by expressing his thanks for the cordial invitation to the meeting and his pleasure in having this opportunity to discuss the extensive trials and numerous operations of the first British ocean-going gas turbine merchant ship *Auris.*

Through the friendly offices of German Shell a group of interested officers of the German Navy were granted permission to visit *Auris* on the occasion of a temporary stay at Hamburg. At that time (the end of 1959) the basin and the sea trials were completed. As a practical man he knew how many difficulties must be overcome before such an absolutely new plant, with inexperienced engineers and crew, could be operated with the necessary security for ship and crew to do duty in commercial service. All such conditions and requirements for a ship must be fulfilled if the new engine plant were to be introduced successfully. They were: 1) maximum reliability; 2) minimum first cost; 3) minimum consumption of fuel; 4) minimum maintenance and operating personnel cost; 5) maximum manœuvrability; 6) minimum weight and volume; 7) last but not least, the engine must have simplicity of arrangement, and be simple, easy and foolproof to handle.

Gas turbine powered aircraft and naval ships were a necessity in military service; most of the claims made were realized and had important implications also in merchant service. The increase in the number of marine gas turbine applications had been encouraging. Steam turbines, with their distortions, boiler and superheater troubles, the Diesel engines with their cylinder liner and lubricating difficulties, had reached their limits of technical development but the gas turbine, including in some special cases the free piston gas turbine, was just beginning.

The development trend in engines was illustrated by the drop in specific fuel consumption. It was superfluous to mention this fact. The limit today was 250 g./h.p. hr.

Naval and commercial experiences illustrated the reliability of the gas turbines. On her trials the *John Sergeant,* the first large merchant ship under any flag to be propelled solely by a gas turbine and fitted with a controllable pitch propeller, had, with her 6,000 h.p., a top speed of over 18 knots. She completed several trouble-free round trip Atlantic crossings.

The simplicity and the automatic controls partially incorporated in the gas turbine had essentially lowered operating costs.

The gas turbine could be started by a starter motor or turbine from a cold condition and deliver full power within a relatively short period. The quick and reliable cold weather starting abilities of the gas turbine were extremely important for naval and merchant ships, for auxiliaries, for pumps, de-icers, fog generators, air supplies, electric power, tanker cargo pump motors, life boats and so on.

To prove the lower maintenance cost, the U.S.A. reconstructed four Liberty ships; *Benjamin Crew,* with a steam turbine, using 16 persons in the engine room crew; *Thomas Nelson,* with a Diesel engine, using 14; *John Sergeant,* with a gas turbine, using 12; and *William Patterson,* with a free piston gas turbine, using 12 persons in the engine room crew. Trials with these vessels took place over a period of three years to find out the most efficient type, and it seemed that the gas turbine would be the winner.

The possibility of installing even large turbines on an integral base that could contain all accessories, and the fact that their plants required only small quantities of cooling water, were further advantages. A single cycle gas turbine might occupy about one-fifth of the volume and have one-tenth of the weight of a normal marine Diesel engine.

For instance, the gas turbine plant, turbine, gear and regenerator in the *John Sergeant* had a specific weight of about 45lb./s.h.p.; another (and new) plant of 7,500 h.p. only 2·5 kg/PSe, the German frigate *Köln* 5·4 kg/PSe, and the newest project, of 20,000 s.h.p. for destroyers, only a specific weight of 0·65 kg/PSe.

The time required to strip, inspect and overhaul gas turbines was only a fraction of that necessary compared with a steam or a Diesel engine plant and the normal time for overhauls would be round about 2,000 to 4,000 hours.

With regard to manœuvrability, during the voyage from Hamburg to Bremerhaven by *Auris* he personally had had occasion to run the engine and to see the two shaft open cycle unit with intercooling between the H.P. and L.P. compressors with the heat exchanger preheating the high pressure air before entry to the single vertically mounted combustion chamber. There were no difficulties for him during the one hour manœuvring time. The control panel gave a wonderful survey, and each abnormal reading was immediately visible by glancing at the panel. It was a point of paramount importance.

The lubricating oil consumption for a gas turbine could be negligible compared with the free piston turbine ship *William Patterson* using 80 gal./day, or with the Diesel ship *Thomas Nelson* using 40 gal./day.

He was absolutely sure that a lot of trouble could have been avoided during the trial of *Auris* if astern power could have been obtained by use of a controllable pitch propeller, like that on the *John Sergeant.* It was a great disadvantage that the original components of the present plant had to be used.

Another important point seemed to be that in *John Sergeant* there had been the possibility further to reduce the fuel cost by consuming specially prepared residual fuels. But when visiting her he had heard that they had used Diesel oil when crossing the ocean.

If any of those present were surprised that he did not mention the excellent work on the *Auris* by the research engineer, Mr. Duggan, and the chief engineer, Mr. Howell, this was only because there was nothing to be added. Their work, in one word, was perfect.

He now wished to say something of the frigate *Köln,* which possessed the largest gas turbine plant installed in a ship. From the technical and tactical point of view this plant was interesting because the high speed of above 30 knots would be produced by a combined Diesel and gas turbine plant. She was a twin-screw ship. On each propeller shaft two Diesels were working, each of 3,000 h.p., and an open circle single-shaft gas turbine of 12,000 h.p., with a combustion chamber lying horizontally above it. For manœuvring and cruising speed of round about 20 knots only the four Diesel engines were in operation. In consequence of the low consumption of gas oil of only 170 g./h.p. hr. the ship had a great steaming range. The Diesels worked on a fluid coupling and a two-stage planetary gear on the propeller shaft with a controllable pitch propeller constructed for 18,000 h.p. If higher speed was necessary, the gas turbine could be combined with the Diesels over the planetary reduction gear of the two-stage gear without interruption. The effect was the possibility of accelerating the ship to top speed in a short time. The plant of the frigate could be supervised by the control station on the bridge, the main engineer's control station, and in the vicinity of the motors with respect to the turbine itself.

The forward and astern speed of the ship could be regulated by the controllable pitch propeller from the bridge for the four Diesel engines only, for the complete plant from the main engineer's control station, and in an emergency from the shaft tunnel. Revolutions and pitch could be handled singly or combined by a so-called "combinator". All levers and wheels on the control panel were foolproof, so that neither mistakes by personnel nor overloading of the Diesel engines were possible. In case of overstepping the torque, the revolutions, or the inlet and outlet gas temperature of 750 deg. C. and 450 deg. C. respectively, the pitch would be reduced automatically as far as necessary.

In conclusion he wished to mention the 60,000 h.p. twin-screw plant of the British County Class. This destroyer possessed a conventional steam plant with an astern turbine for cruising speed, and a booster plant for top speed, with an automatically over-running clutch.

These examples proved, he suggested, that the era of the gas turbine was only just beginning.

The CHAIRMAN (Mr. B. P. Ingamells, C.B.E.) expressed the gratitude of the meeting to Captain Moeller for coming specially from Germany to give his contribution to the discussion.

MR. T. E. ADAMS, B.Sc.(Eng.) said that he wished to congratulate the authors on their paper. He could not think of a single detail on the trials of the *Auris* that had escaped them. Great credit was due to Shell for venturing on and paying for the development of gas turbine machinery for marine propulsion. He asked to be allowed to pay tribute to the late Mr. B. E. G. Forsling, who was his "Chief" during the design of both *Auris* I and II.

The main design criteria of a practical marine gas turbine were:

1) Manœuvring capability.
2) Reliability.
3) Efficiency.

The reason for this order was that unless the ship could manœuvre the question of reliability did not arise and similarly unless the machinery was reliable the question of efficiency barely warranted consideration. With *Auris* I the gas turbine was electrically disconnected from the propeller during manœuvring.

As the authors had stated, the connexion between the two gas turbines and the propeller on the proposed installation for the *Hemisinus* was a.c. electric. This type of drive was also considered for the single engine *Auris* scheme but the prime importance of manœuvring meant that the drive system was complicated by some sort of braking device so that manœuvring could be carried out easily and quickly. The braking system considered was electrical but unfortunately the dissipation of the enormous amount of heat generated during braking was a difficult problem, and it made the drive both complicated and messy.

The present hydraulic coupling system with mechanical drive was eventually decided upon for *Auris* II. With this system the hydraulic couplings were to be used as a braking device for the gas turbine during manœuvring.

During the early runs on basin trials, it was found that manœuvring was so hopeless that the ship was not fit to go to sea.

Fortunately, his company had up their sleeves another system of braking the gas turbine, which would enable the ship to manœuvre without regularly dropping its anchor.

Electrical power failure from the grid while testing *Auris* I at Rugby in 1950, had already suggested that the best safety device that could be fitted to that machine would be a blow-off valve at the outlet of the H.P. compressor, which could either be hand controlled or connected to the trips. This, it was thought, would cater for such faults as the sudden loss of electrical load due, for example, to excitation failure or the propeller falling off.

However, the original safety devices fitted on *Auris* I were shown to cope, and although sketches of the blow-off valve were made it was never fitted.

With the possibility of such a valve at the back of their minds, and the trials indicating that drastic measures would have to be taken to enable the ship to manœuvre at all, an immediate decision was taken to fit such a valve to the H.P. compressor outlet duct of *Auris* II. There were many other incentives to use such a valve, for example, the fact that all manœuvring would occur with the combustion chamber and H.P. turbine at very nearly constant full load temperature. It was impossible even to guess how much the maintenance of constant temperature during manœuvring had contributed to the reliability of the combustion chamber and the H.P. turbine, especially as these were made of high expansion austenitic steel. This H.P. blow-off valve could, of course, be used with electric transmission and would now make this system normal and similar to those which had been used in the past with steam turbines.

Because of fuel costs, it was always understood that *Auris* II would burn heavy oil, and at that time, about eight years ago, both Shell and B.T.H. agreed that whatever the safeguards, there was very little hope of running the set on ordinary commercially available heavy fuel at a turbine inlet temperature much above 650 deg. C.

There was, however, an excellent chance of doing so with temperatures below 650 deg. C., provided certain safeguards were taken. In any case whatever happened with this low temperature the turbine blading would remain and not vanish down the exhaust as liquid vanadate compounds, which is just what could happen with the high temperature set.

As was well known, the output and, to a much lesser extent, the efficiency of a gas turbine set depended upon the turbine inlet temperature—the higher the temperature the higher the output and efficiency.

One fact, namely the fixing of the maximum turbine inlet temperature to 650 deg. C., virtually settled the whole of the design of the *Auris* II gas turbine.

The relatively low temperature limitation had the following effects:

1) It made the components bulky in order to give the required power and efficiency.
2) A complicated cycle incorporating intercooler and heat exchanger became necessary.
3) The components had to be arranged in such a manner as to give the minimum pressure losses.
4) The split horizontal joint became essential on the turbines, for easy cleaning of the blading.
5) The aerodynamic design of the H.P. turbine blading was affected, in order to obtain minimum deposition with minimum loss of efficiency.
6) The spare H.P. turbine casing with its self-contained rotor had to be added alongside the main set.
7) It settled the additive and abrasive system to be used to prevent or remove blade deposition.

With selected heavy fuels containing reasonably low amounts of vanadium and sodium, it was probable that *Auris* II would have just scraped home to success. Did the authors still think that a high temperature gas turbine set was not a practical proposition?

The calculations made from the trials results showed a theoretical power output from the gas turbine when the set was disconnected from the gearbox. This obvious discrepancy, which was shown by the lower curve in Fig. 13, indicated that something was wrong somewhere. It was not known what it was. It was believed that the overall consumption figures were reliable. The correlation between the calculated output and the measured electrical output of both *Auris* I and the Nairobi South sets had been excellent, even at no load. The trials results showed the set roughly up to expectations and the voyage performance figures were almost the same as those for the trials. Had the authors similar comparisons for other ships?

With regard to the gearbox, he wished to make one point about which there was some confusion. If the slip on the fluid coupling was 6 per cent, and the efficiency of the gearbox with the clutch, i.e. without slip, was 90 per cent, then the drop in gearbox efficiency due to slip alone when using the fluid coupling was not 6 per cent but $6 \times 0{\cdot}9$, i.e. $5{\cdot}4$ per cent.

It was of interest that the method of solving the troubles mentioned by the authors had differed in each case. The oil leak problem was eventually solved, he was sorry to say, by the blanket method, i.e. by doing all at once everything that could be thought of to stop oil leakage.

The problem of the L.P. compressor clearances was solved by doing in turn various tests, until the right answer was found.

The problem of the failure of the pintle valve in a main burner was solved by inspection of the fracture and related parts of the burner. This he had always thought was the most serious and disconcerting failure that had happened up to then. It had always been possible, under any conditions that could be foreseen, to shut down the gas turbine by the flick of a knob. It was not difficult, however, to imagine circumstances where it was more dangerous to the ship to shut down the engines than to let the engine keep on running, despite the fact that it might be in serious trouble.

The detection of fouling of the propeller was a convincing demonstration of the value of good instrumentation on board ship. It also showed that it was wise to believe the instrument readings, despite all the qualitative arguments which were generally produced to try and disprove the unpalatable deductions.

It was noticeable that the main emphasis with regard to ship propulsion machinery was placed on the specific fuel consumption. Would the authors give some idea of how the fuel consumption of *Auris* II compared with the Diesel electric version of *Auris* I. Would they also indicate in which variant they would prefer to put to sea, and if they did not consider it an unfair question which one would they prefer to have if they were shipowners?

The thermal efficiency of a marine Diesel engine was about 40 per cent; that of a steam turbine about the 25 per cent mark. The effective thermal efficiency of the steam turbine was less than this as it was, of necessity, associated with either gear or electric transmission losses. If the authors agreed with the emphasis that was usually given to specific fuel consumption, why had turbine machinery been ordered in the past and why was it still being ordered today? Could the authors give the advantages of turbine machinery which over-rode the better fuel consumption of Diesel machinery.

Auris II had still to prove itself but it was some slight satisfaction, even if luck had sometimes been on their side, that the gas turbine had at all times after its erection, run and given power every time it had been called on to do so, and he concluded by suggesting that one of the reasons for the success of *Auris* II was the willing co-operation of the many people from the many firms that were concerned in the venture.

MR. A. FOWLER (Member), after congratulating the authors on their immensely interesting paper, said that he would confine his remarks to the intercooler designed and built by the company with which he was associated. Contributing some 26 per cent to the thermal efficiency of the cycle, this unit was perhaps not without interest, and it appeared to have given little if any trouble since it was installed.

It was not of conventional design and, of necessity, differed in several ways from those inter and pre-coolers they had already built and had since built for other gas turbine installations.

Early in the paper the authors had referred to the unnecessary weight of auxiliaries, and these remarks might possibly be applied to the intercooler were it not for one or two major factors influencing the design in the initial stages.

Experience with the first gas turbine fitted in *Auris* indicated that it was possible for quantities of salt spray to be drawn in through the air intake and with a machine fitted with an intercooler some fouling of the cooling surface could be anticipated. To cater for this condition a fouling margin of 25 per cent on cooling surface was provided, plain tubes were used and access doors were provided to enable the surface to be washed down. The fact that the designed thermal ratio of 84 per cent was considerably exceeded in service, as shown in the paper, might suggest that the fouling margin was pessimistic. Had a design of intercooler using secondary surface of the ribbon-wound tube type been acceptable, a reduction in matrix volume of at least 20 per cent could have been achieved with a considerable saving in weight.

Another factor influencing the design was that the space available for the intercooler below the L.P. turbine precluded the use of a cylindrical unit and it was necessary to adopt a rectangular form requiring considerable stiffening for the pressures involved, greatly adding to its weight.

Space restrictions also necessitated a fixed tube plate design having no provision for the thermal expansion of the tube stack, it being considered that a tube stress of 3 tons/sq. in. and a tube "pull out" load of 530lb. at the gas inlet end was acceptable.

In the light of their operational experience would the authors consider that the plain tube design of intercooler was justified? Also, would the authors comment on their reference to vast quantities of condensed fresh water produced in the intercooler, as under suitable conditions of high ambient temperature and high relative humidity he would have expected appreciable condensation, but the vast quantities referred to were a little puzzling.

MR. B. OPELT, B.Sc., congratulated the authors on a very accurate report on the life of the *Auris* with her new machinery.

During the whole of the trials and service life the performance of the engine had been carefully watched by the designers, and it was possible, due to co-operation between all the people concerned, and the availability of their electronic computer, to obtain performance results almost immediately a test point was taken.

The testing had been necessarily confined to the no-load and prop-law lines. Even though a lot of trouble had been taken with instrumentation, some of the readings had proved very difficult to take. However, the majority had been very reliable.

One of the difficult readings was the H.P. turbine inlet temperature, affected as it was by flame radiation, stratification, and soot deposition. This reading was, therefore, used only for comparison. The performance of the H.P. turbine had been evaluated from H.P. compressor temperatures, the H.P. turbine outlet temperature, and, of course, the static pressures. On this basis it was possible to estimate the error in inlet temperature, and it was interesting to note that it decreased from +15 deg. C. at the beginning of testing to −3 deg. C. at the end of service. This reduction in the error was perhaps due to some sooting up of the thermocouples.

The difficulty of obtaining very reliable fuel measurement had been recognized from the beginning, and four methods had been used; 1) weigh tank; 2) dips in the fuel tank; 3) pneumacator in the fuel tank; 4) V.A.F. displacement meter (calibrated by the makers). Of these the first proved impossible to use, and in most cases the performance calculations were based on the mean of pneumacator and V.A.F. meter readings. In all cases where agreement between readings was unsatisfactory or the error in H.P. turbine inlet temperature was inconsisent, the test was not used.

So far as Table III in the paper was concerned, it was worth pointing out that it contained individual tests as opposed to statistical or other mean values. The first and second columns showed the runs of special interest, since the only occasion on which the engine had been up to design speed was on full power run, and it had only been tested (for 24 hours) on boiler fuel. Unfortunately, during the full power run the sea had some swell and so manometer readings had been affected. The other tests in Table III were some examples out of the large number of tests made during the voyage to Curaco and back.

He now wished to mention how the performance of the ship had actually compared with the design figures. To obtain a more representative picture of the set all the results of tests made during the service life of the ship were plotted and curves extrapolated to design conditions. The following isentropic percentage efficiencies were thus obtaind:

L.P.C.	H.P.C.	H.P.T.	L.P.T.
90·5	87·3	87·5	91·8

On the whole these were very good compared with the expected values of:

88	88	88	90

The flow coefficients as defined by $K = M\left[\frac{Ti}{Pi^2 - Pe^2}\right]^{\frac{1}{2}}$

from service tests were 52·5 and 35·5 for L.P. and H.P. turbines respectively, as compared with design estimates of 52·8 and 33·0.

The performance of the heat exchanger had been good but that could be explained by the additional area.

A combustion chamber efficiency of 98 per cent was aimed at in view of the experience with the *Auris* installation No. 1 and the Nairobi South turbines. This, it was believed, had been attained. However, the expression for the combustion chamber efficiency involved two factors of doubtful certainty—the combustion chamber outlet temperature and the mean value of the specific heat, which might not be quite representative for the large temperature range involved.

At the design stage a gas turbine thermal efficiency of 27·5 per cent was hoped for. Since the measured efficiency was affected by errors in fuel readings it was difficult to obtain very reliable figures for short tests and it was thought that an overall figure for the whole voyage to Curaçao and back might be more significant. The figure thus obtained for the overall thermal efficiency was 26·85 per cent, which was equivalent to a specific fuel consumption of 0·518lb./h.p. hr. These figures would be improved if credit were given for the power output from the waste heat boiler and further if the set were run nearer its design speed and horsepower.

In conclusion, it could be said that on the whole the design goal had been reached and in some cases surpassed. Bearing that in mind, as well as the fact that the set would have been equally successful in its final form with an electric drive, could the authors comment on whether electrical or mechanical transmission would have been better. From the experience of *Auris* II what did the authors now think of the original scheme?

MR. R. F. DARLING said that one of his colleagues was waiting to discuss the hydraulic transmission in more detail. In that connexion he himself would merely like to say that, according to the worst interpretation that could be put on the figures shown in the paper, this gearbox was supposed to have dissipated 1,400 horsepower, i.e. 400 horsepower which was the calculated loss, plus 1,000 horsepower which was "missing". He appealed to the marine engineers in the audience —1,400 horsepower, and the only mention of anything hot in the gearbox was that one bearing ran a little hot during the sea trial. Did this make sense? He would say no more.

The authors stated at the end of the paper that present types of propulsion, steam turbine and Diesel, had their troubles and were at the limits of their design. He would say nothing about Diesels but he contested the statement about the steam turbine. Since the war the fuel consumption of marine steam turbines had been reduced by nearly 30 per cent, and the weight and size by over 50 per cent. Those figures were based on machinery for something like 600 ships, which his organization had designed and put to sea in that period, and even over the last five years there had been quite distinct and recognizable improvements. Developments which were now in hand—some of them quite small, some fairly major, some which would come to fruition and some which would not—were bound to have a further effect in years to come and he could assure the authors that when they made comparisons with steam turbines they were shooting at a moving target.

They had made it quite clear in their conclusions that they believed in gas turbines for marine propulsion and they talked about more up to date designs, but they gave no details. Everyone present would, he was sure, be interested to have a full description of the sort of gas turbine the authors would recommend—the arrangement of components, the maximum temperatures, the efficiencies they expected to get, the type of transmission they would advise, and so on.

Their own belief at Pametrada, after running one gas turbine for a long period and subsequently looking at the possible design of numerous others, was that the uncooled gas turbine—the kind they were talking about—was not worth persevering with for marine propulsion. Some years ago they had designed a set which they still regarded as the best that could be put forward. It had a fuel consumption of about 0·46lb./s.h.p. hr. with quite modest temperatures, but it utilized three turbines on separate shafts, an intercooler, a heat exchanger and two combustion chambers. On mature consideration they did not think it would have been worth the cost of development, and the owner in question was very wise to cancel his order, as he subsequently did.

Coming to the liquid-cooled gas turbine, of which again they had had experience, the matter was entirely different. They had run two rotors at a gas temperature of 1,200 deg. C. (2,200 deg. F.) and neither had given any mechanical trouble. It could be shown that a complete gas turbine unit based on those design figures would have a fuel consumption of about 0·38lb./s.h.p. hr., assuming component efficiencies rather poorer than quoted here. It would be far and away smaller and lighter than any other form of power plant that could be put forward. It was the sort of thing which could justify large scale development costs, and they felt very strongly that if gas turbines were to make their mark in merchant ships (saying nothing about naval requirements, because they were entirely different) they would have to show a really big advantage over existing Diesels and steam turbines and liquid cooling was really the only hope of doing that. What did the authors think about this?

He could not conclude without expressing his admiration for the authors. No one could read this paper without realizing what a worry this job had been to them. He had been associated with it to some extent himself, particularly as regards the repairs to the hydraulic transmission when it broke down. He had developed a few grey hairs and he had no doubt the authors did likewise.

Their perseverance and their bravery (in the sense of the owners continuing to run the ship) and the fact that other organizations had put in no small amount of their own money to help with the developments, reflected the greatest credit on the organizations and the personnel concerned—and on none more than on the two authors.

MR. A. LOGAN, O.B.E. (Member), said that as many of those present were wondering why the *Auris* had been withdrawn from service, perhaps this was the opportune time to clarify the position.

It was, he thought, true to say that the *Auris,* technically, although not economically, came up to their expectations. A number of difficulties were experienced, but not more than could reasonably be expected in so novel a propulsion unit. The compressor trouble described by the authors, though seemingly serious initially, was found by tests to be amenable to correction by straight forward modification.

The overall fuel rate was not significantly worse than design expectancy, and Chief Engineer Howell could vouch that the general reliability of the plant and the manœuvring qualities of the ship were as good as anything that could reasonably be wished.

He made these points because he did not wish it to be thought that the ship was taken out of service due to dissatisfaction with the machinery as designed, built and installed. When the project was first conceived by his late colleague, Mr. John Lamb, and throughout most of the development stages, it was hoped that it would prove that the gas turbine had a future of attractive economy compared with its conventional counterparts. In the later stages of the development of the project, however, it became clear that this hope was not to be realized, and, in fact, the project was continued because only by completing and operating the ship could there be any recompense in terms of valuable knowledge and in the development of the ship's many novel features.

There was no doubt that the installation of the machinery, the running trials and the months of operation, proved valuable in numerous ways to the builders, the gas turbine designers and to Pametrada. There were still issues to be resolved, such as accurate measurement of turbine output and the question of turbine blade fatigue failure, but it was felt that these issues were of more direct interest to industry rather than an operating shipping company, and they had to face the fact that the continued running of the *Auris* was an expense from which as a company they could expect little return; in fact, they were compelled at this juncture seriously to consider cutting their losses, particularly as they as a company were long in ships of the *Auris* size.

Nevertheless, they were reluctant to withdraw the ship from service whilst her technical usefulness was still unexpended. Industry—i.e. builders, designers and research bodies —were asked if they would be prepared to share certain costs if they felt that continued operation of the ship was in the general industry's interest. Regrettably, support from industry was not forthcoming, therefore, distressing as the position was to his colleagues, Mr. Duggan and others, the Company had no alternative but to take the ship out of service.

It would be agreed, he felt sure, that the paper was an excellent historical record of the first British ocean-going gas turbine ship.

Who could say what the future had in store? Were they ahead of time? Could it be that, with the age of automation approaching, this form of prime mover would find its place when simplicity in operation, together with reduced maintenance, forced the shipowner to leave the conventional steam and Diesel plants that were at sea today?

MR. F. R. HARRIS said that the strain gauge tests on certain stages of the L.P. compressor fixed blading gave two important results: 1) vibration, in stages 5 and 10, was, at all speeds, at the blade fundamental frequency; the amplitude increased with speed; 2) the vibration amplitude was markedly reduced (to one-tenth of previous value) by opening the blow-off valve at L.P. compressor outlet.

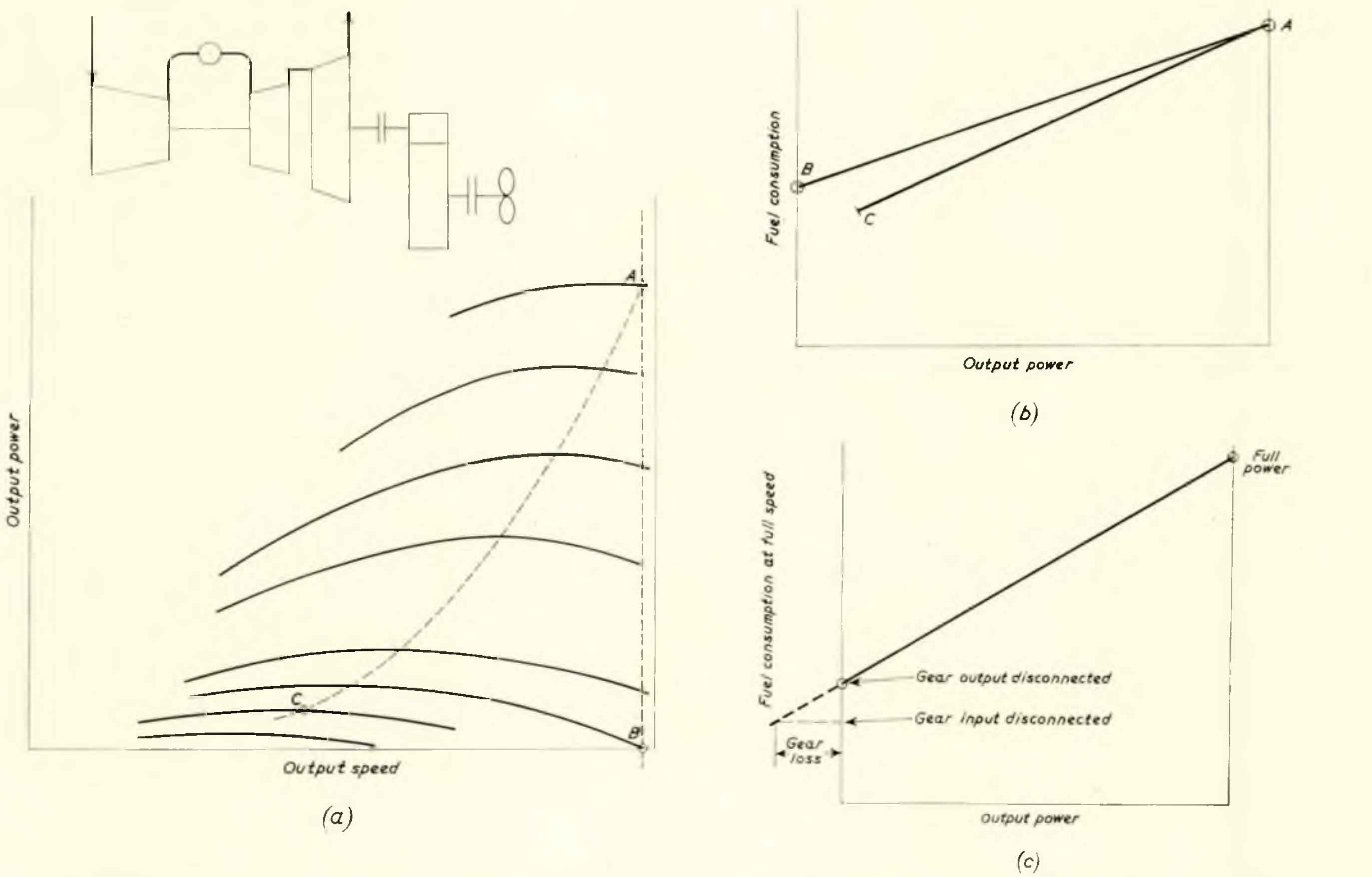
The deductions from these tests were that the vibrations were aerodynamic in origin, rather than mechanical. Such vibrations could be caused either by flutter or by something in the nature of rotating stall, since both of these commonly induced vibration at the fundamental frequency. The stiffness of the blades was such that flutter would not be expected. Additionally, it would be unusual if flutter vibrations were severely reduced by blow-off at compressor outlet. The conclusion was therefore that some action similar to rotating stall was taking place. Rotating stall did not normally occur at compressor speeds above about 70 per cent of design full speed, and was usually far worse at the inlet end of the compressor than at the outlet end; the *Auris* phenomena showed vibration worse at the higher speeds, and worse at the outlet end of the compressor, so that the usual form of rotating stall was not present.

It was considered that the combination of the toroidal diffuser with straightening vanes at outlet, and the high tangential velocity of the air at entry to the diffuser, contributed to the pulsations experienced, and that an acceptable solution could well have been to fit straightening vanes at the inlet to the diffuser instead of at the outlet. The possibility existed of wrong air incidence angles at the existing diffuser straightening vanes, which, under the quasi-stable conditions of diffuser flow, might well give rise to rotating areas of stalled and unstalled flow; the way in which such variations at the outlet of a diffusing passage could cause vibration and failure in blading some way upstream had already been described, in reference to a previous naval gas turbine.*

A feature contributing to the blade failure was the reduced fatigue strength of the blade material in conditions of a marine corrosive atmosphere; the fatigue strength at 10^8 cycles was about 65 per cent of the dry fatigue limit.

The authors had made no comment on the detailed economics of the gas turbine for merchant ship propulsion. The *Auris* gas turbine was a large and heavy machine, with low blade speeds, but, as was apparent from Fig. 16 and

* Trewby, G. F. A. 1954. "British Naval Gas Turbines". Trans. I.Mar.E., Vol. 66, p. 125.



(a) Variation of output power with output speed
AB—operation at constant output speed
AC—operation with propeller where power varies as cube of speed

(b) Variation of fuel consumption with output power
(c) Derivation of reduction gear loss by fuel consumption measurements, all at the same L.P. turbine speed

FIG. 22—*Gear losses for a simple two-shaft gas turbine power plant*

Table III, the component efficiencies were extremely high. It would be possible to build a smaller and cheaper gas turbine, with lower component efficiencies, and a higher specific fuel consumption than *Auris*; it would be interesting to have some cost figures on which to base an assessment of the suitability of such a gas turbine for merchant ship propulsion.

Additionally, the permanent use of a distillate fuel, instead of residual fuel, could lead to a still smaller and cheaper gas turbine, with a specific fuel consumption (of the more expensive fuel) the same as, or maybe a little better than, that achieved in *Auris*. Assuming that such a gas turbine could be built, for the same rating, for half the cost, and with an overall length of 75 per cent of the *Auris* machine, could the authors give an opinion as to the specific fuel consumption (distillate fuel) necessary to justify the installation?

The emergency propulsion unit was a steam turbine. In the *John Sergeant* installation the designers had stated* that, in a future ship, they would advocate for emergency propulsion an electric motor, driven from the ship's auxiliary supply, instead of a steam turbine. The authors' comments on the respective merits of the systems would be of value.

The problem of gearing efficiency was always with the designers of gas turbine plant; it was worse where there was no opportunity of tests on shore, and still worse with a gear which contained hydraulic components. The simple method of measuring energy dissipated in the oil cooling system did not always give a reliable figure for gear loss, and was affected by radiation from the gearbox. Some details might be of interest of a method which appeared to give at least a rough idea of gearing losses and would have been capable of use in *Auris*. It had been used to determine gear losses in a somewhat complicated reduction gear driven by a simple two-shaft gas turbine; the output characteristics of such a machine were shown in Fig. 22, and it was evident that the L.P. turbine, which was mechanically independent of the H.P. turbine and compressor, could run at full speed over the whole power range, from zero to full power. In a normal installation, where the cube law related power output to speed, the relationship between fuel consumption and power output was close to a straight line. For operation at constant output speed, the relationship between fuel consumption and power output was also nearly a (different) straight line, although such operation could be achieved only on a shore test bed, unless special facilities, such as controllable-pitch propellers, existed.

If, on shore, the output at constant full speed were varied from zero to rated output, the graph of fuel consumption against output could be drawn. If now the gas turbine were disconnected from the gear, and the output shaft again run to full speed, the fuel consumption would be less because there were now no gear losses; the actual gear loss could be found by extrapolation backward to the new fuel consumption. This check could be made at a series of output speeds.

Such a method assumed that gear losses varied with speed and not with load. This was an approximation and was probably good enough for many purposes. The method would require modification where power was transmitted through hydraulic components.

This method could be applied to installed machinery (including *Auris*) by measuring, at full output turbine speed,

Fuel consumption F_1 at full output, and output power P.

Fuel consumption F_2 at zero gear output (with gear output shaft disconnected).

Fuel consumption F_3 at zero gear input (with gear input shaft disconnected).

Then, on the assumption that the fuel consumption/power output relationship was linear, the gear loss L at full speed was

$$L = \frac{F_2 - F_3}{F_1 - F_3} P$$

Such an estimate of power loss, although not precise, was probably nearer the mark than one based on net turbine power calculations; these depended on accurate measurements of air mass flow and two sets of temperature differences, as well as there being no leaks, and such measurements were notoriously difficult to make with any precision.

Dr. T. W. F. Brown, *C.B.E.* (Member),† said that the authors had given a very full account of the teething troubles met with on this prototype installation and of the work which was necessary to achieve satisfactory results. The Shell Company were to be congratulated on their lead in installing this machinery and on the work carried out in association with the machinery designers and manufacturers to make it a satisfactory operational plant. They also deserved great credit for operating the machinery in *Auris* during this period when the 12,000-d.w.t. tanker was too small for economic operation.

The authors made clear that the gas turbine was originally designed to operate with electrical transmission, and it was unfortunate that this led to a choice of cycle which was not the most suitable for marine propulsion with direct transmission. The coupling of a power gas turbine to an L.P. compressor was considered to be fundamentally wrong in a direct-coupled main-propulsion gas turbine owing to the difficulties associated with low power running and manœuvring. Pametrada had consistently favoured the use of a cycle having an independent power turbine to give the necessary flexibility and rapid response during manœuvring, and also the use of reheat in conjunction with a higher pressure ratio to achieve a higher efficiency and reduced size and weight. The paper brought out the difficulty experienced in running at low speeds and in making rapid power changes with the turbine as origin-

* McLean, Tangerini and Van Cott. "Report on 9,000 hr. Operation of Marine Propulsion Gas Turbine in the *John Sergeant*." A.S.M.E. paper 60-GTP-5.

† Contribution read by Mr. Wilkinson (Pametrada) in Dr. Brown's absence.

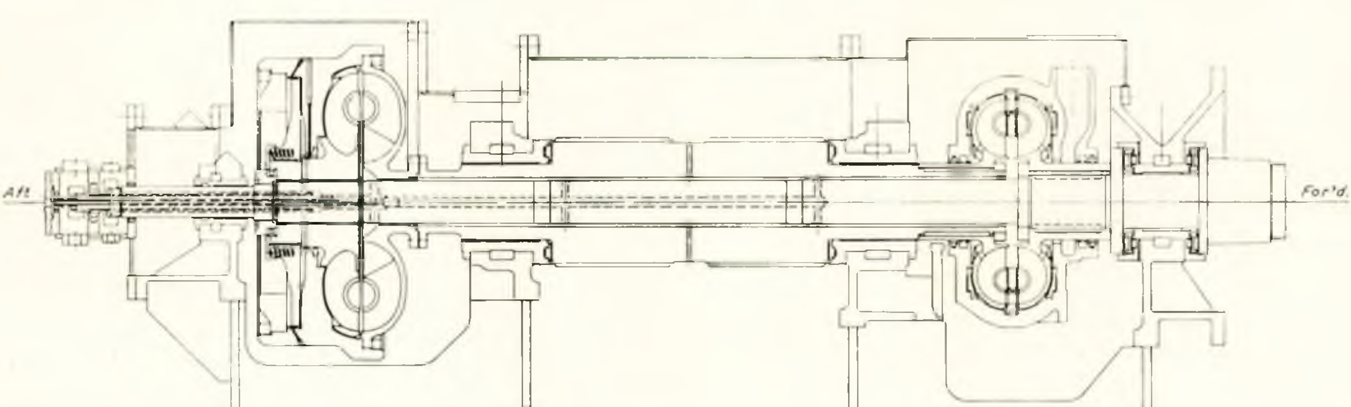


Fig. 23—*Section through Pametrada hydraulic transmission applied to parallel shaft gears*

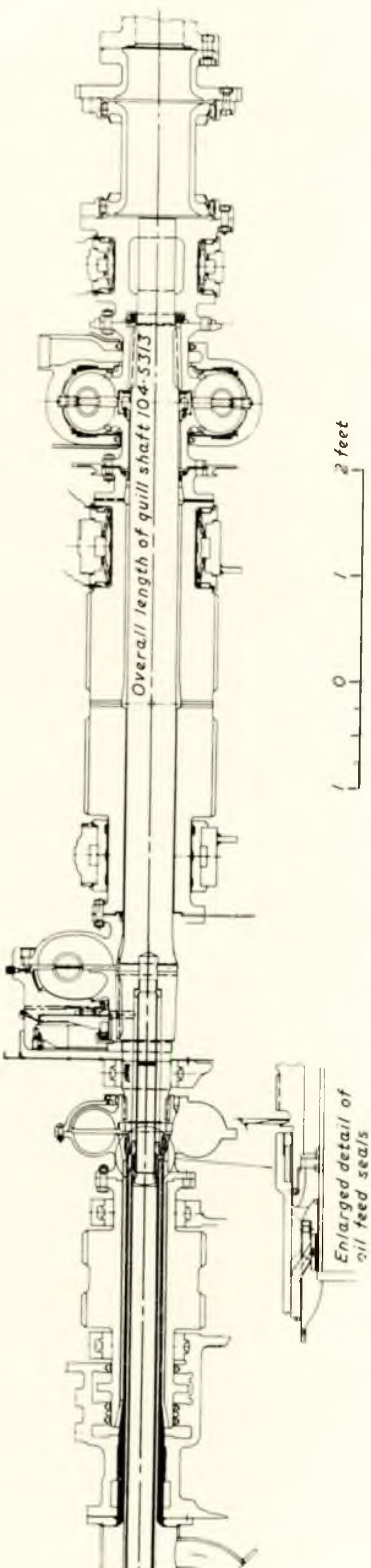


Fig. 24—*Arrangement of primary quill shaft and hydraulic transmission*

ally installed, which necessitated the fitting of blow-off valves. These were apparently very effective, although "dead slow" could still only be obtained by loading the ahead and astern couplings against one another, and the acceleration from this condition was low. The loss of efficiency when blowing off H.P. air was perhaps not of very great significance in a merchant ship, although it would be unacceptable at cruising power in a naval vessel. The need for large silencers when blowing off quantities of high-pressure air was fully appreciated!

It was thought that the authors were being optimistic in concluding that the *Auris* machinery could compete in fuel consumption with the latest type of steam turbine machinery. The *Aluco* and *Arianta* cited in the paper were two of a class of 18,000-d.w.t. tankers, and on trials several ships in this class gave an overall specific fuel consumption well under 0·600lb./s.h.p. hr. compared with about 0·66lb./s.h.p. hr. for the *Auris*. Moreover, these ships had machinery which was designed in 1955, and which had a very simple feed system. A much lower fuel consumption than this could be obtained from steam turbine machinery, and consumptions of around 0·50lb./s.h.p. hr., all purposes, were now being consistently returned from a number of tanker installations—admittedly developing 16,000 to 18,000 s.h.p.

The authors had dealt very fairly with the hydraulic transmission, and it was gratifying to have their conclusions to the effect that the transmission justified its inclusion as a prototype, and that the manœuvring of the ship was very satisfactory. One of the difficulties with this installation was that the arrangement of the transmission was too complicated for a first installation, owing to the presence of the cargo pump alternator with a separate gearcase, fitted at the aft end of the primary-pinion line.

Fig. 23 showed the preferred arrangement of the hydraulic transmission, and Fig. 24 showed the arrangement in *Auris*. The bearing at the aft end of the quill shaft had had to be carried from the alternator gearbox rather than from the main gearcase structure, and the oil supply to the clutch and ahead coupling had to pass through the alternator pinion and coupling, through a long tubular assembly having fine-clearance seals at the end. Lining up the quill shaft bearings and seals was in fact a nightmare, and it was not altogether surprising that there was a certain amount of trouble due to rubbing of seals. Pametrada accepted this arrangement because they were anxious to have the opportunity of getting a hydraulic transmission to sea, but it is an arrangement which introduced difficulties not inherent in the system.

The original designs of astern converter wheels having twenty and nineteen vanes in the input and output wheels respectively, gave poor accessibility for brazing, and this was the underlying cause of the failure of these wheels early in the trials, although the builders did a creditable job under practical difficulties. The outer torus forgings were undamaged and were used in the replacement wheels, in which the new vanes were attached by welding. Overspeed tests on the replacement wheels were suggested by Pametrada to prove the wheels before re-assembly and these tests were carried out at the Research Station and were witnessed and accepted by Shell representatives and Lloyd's surveyors. The wheels were overspeeded by 1,000 r.p.m., of which the first 300 r.p.m. corresponded to the additional loading experienced in service due to the wheels being full of oil, and the remaining 700 r.p.m. corresponded to a true overload.

The strain gauge measurements showed that the maximum working stress at service r.p.m. in the primary wheel was 5·17 tons/sq. in. and in the secondary wheel 4·61 tons/sq. in., except in one region where the thickness of the inner torus had been reduced during balancing, where the stress was 8·14 tons/sq. in. The material was 0·4 per cent carbon steel with a u.t.s. of more than 30 tons/sq. in.

Subsequently experiments showed that the efficiency of the converter could actually be increased by fitting a smaller number of thicker streamlined vanes. In the present-day design the two wheels had eight and seven vanes respectively. Accessibility for welding was therefore extremely good and the wheels were incomparably more robust, as well as being easier to manufacture and therefore cheaper. It was hoped ultimately to cast the wheels in 60 ton steel in order to give a still further increase in strength at reduced cost (to permit higher rim speeds). Some experience of cast astern converter wheels had already been gained.

The relatively low efficiency of the *Auris* astern converter was a disappointment, since efficiencies of 65 per cent had previously been measured on an 8-in. model converter. The chief cause of the trouble proved to be that in scaling up from model scale to full scale the losses due to internal leakage and also to the withdrawal of oil for cooling increased appreciably. The seals had now been redesigned to reduce internal leakage, and cooling oil was taken from oil which had leaked past the seals, so that the two losses were no longer additive. A full scale converter incorporating these changes and the reduced number of vanes already referred to had been tested and gave an efficiency of slightly over 65 per cent at full power. This higher efficiency could be expected to improve manœuvring characteristics still further in future installations.

The performance of the clutch had caused a good deal of discussion, but it was really a very simple matter. The friction surface area in the clutch was insufficient to give an adequate margin against slipping, having been based on a coefficient of friction (supplied by the manufacturer) which was later found in experiments to be too high. The margin was further reduced by the fact that the full centrifugal oil pressure was not built up behind the clutch plate owing to the oil being led into this space at the inner diameter and subsequently not reaching the full speed of rotation. The clutch was therefore not able to cope with increased loads due to the torsional vibration which occurred over a certain speed range. After slipping, some glazing would occur, further reducing the coefficient of friction, there naturally being a considerable difference between the static and dynamic coefficient of friction.

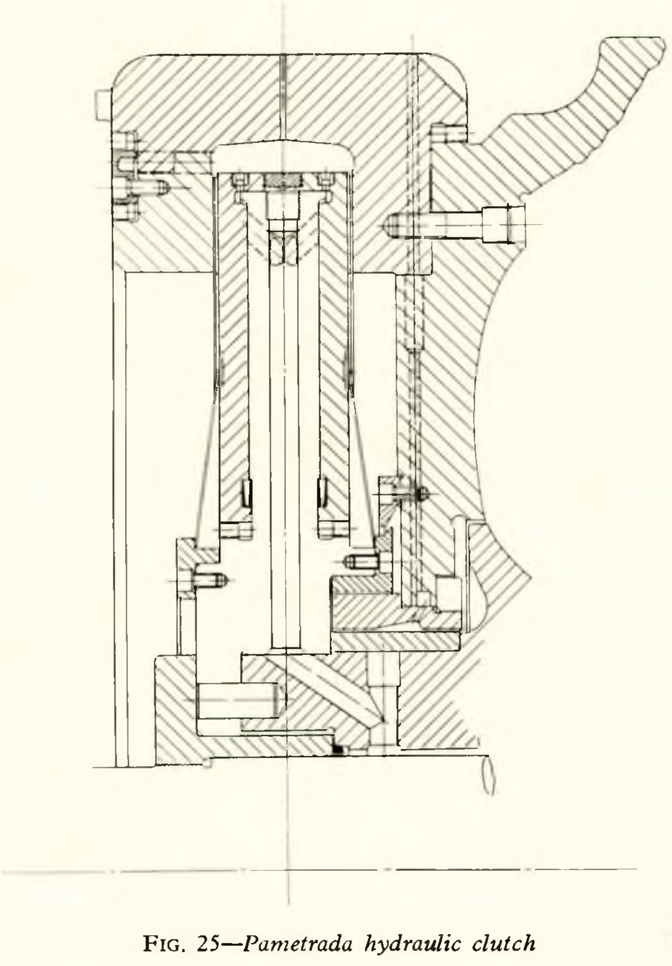


FIG. 25—*Pametrada hydraulic clutch*

Fig. 25 showed the latest design of clutch, which was at present under construction for testing in the near future. This had two clutch plates with sufficient surface area to give a factor of 2·5 over the service load, based on the dynamic coefficient of friction. Also the oil was led into the pressure spaces behind the clutch plates at the outer diameter, where it already had the full peripheral velocity. Minor improvements in construction, and in the sealing arrangements, had also been made, and there was every reason to be sure that its performance would be satisfactory. Since it was being tried out on the full scale at Pametrada Research Station, this should ensure confidence in the future.

Finally, he wished to make a brief comment on what might be called the mystery of the missing power, epitomized in Fig. 13. The authors' comments on this question were a model of impartiality, and it was not proposed to go over all the arguments again. A point which stood out, however, was that the loss curves of Fig. 13, in addition to being extremely high, had a most unusual form if viewed as gearbox losses.

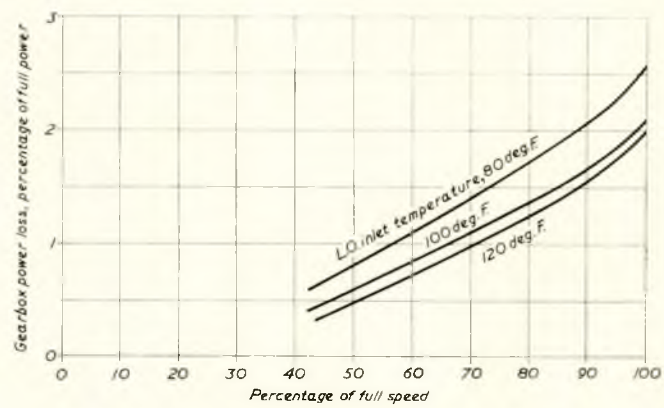


FIG. 26—*Typical losses in double reduction gearing*

Fig. 26 showed curves of measured power loss as a percentage of full power in a double reduction gearbox, obtained during back to back gearing trials at Pametrada. These were typical of normal gearbox losses, and should be compared with the curve marked "clutch engaged" in Fig. 13. The latter curve became almost horizontal at lower speeds, implying that the resisting torque increased as the speed was reduced, while at the upper end of the speed range the loss varied approximately as the ninth power of the speed. Losses in gears and bearings could not have this form, and it was difficult to visualize a loss mechanism within the hydraulic elements which could exhibit such characteristics.

On the other hand, one did not need to reiterate the known difficulties of estimating gas turbine output from measurements of gas pressure and temperature, particularly when facilitites for obtaining true mass mean temperatures and pressures were not available. It was well nigh impossible to position a temperature measuring instrument so that it gave the weighted true temperature in relation to mass flow. Deductions made from such readings could therefore be wildly misleading.

In conclusion he wished to thank the authors for their most informative and interesting paper, which could not fail to be of value in the future of marine propulsion, not only with gas turbine drive but also with uni-directional steam turbines able, therefore, to utilize higher inlet conditions, particularly higher temperatures.

Mr. F. G. Holmes congratulated the authors on preparing and presenting a very interesting paper which constituted a complete and unbiased description of this unique and successful machinery installation, which as they had said was an experimental unit.

He was proud to say that he was a member of the team of engineers who worked on this project and wished to endorse the authors' remarks regarding the very high degree of co-operation which existed between the parties concerned.

Most of the difficulties which beset them had been fully described both in the paper and elsewhere, but he wished to enlarge upon some of the manufacturing troubles, which unfortunately retarded the production. Great difficulty had been experienced in obtaining satisfactory castings in alloy steels; in the case of the L.P. turbine cylinders they had had to receive three sets of castings in order to produce one cylinder and in the case of the compressor cylinders, which were of stainless iron material, there were areas of porous metal, which were found during grooving operations, and these had to be repaired by the suppliers. The decision to erect the machinery on board ship without prior shop testing meant that all modifications necessary as a result of trials had either to be made on board or the parts removed from the ship and machined ashore, this latter operation being a great consumer of time. For example, the oil thrower on the L.P. turbine rotor, referred to by the authors, was machined with the rotor lying on its casing in order to save time and removal from the ship, the power being supplied by the barring gear.

From the details already given by the authors it would be seen that many things happened to the installation which should not have occurred, but it should be remembered, however, that as far as they were concerned the hydraulic transmission was a prototype set with many first-off features and there was practically no previous information to which reference could be made. He was, however, quite confident that a satisfactory hydraulic transmission could now be built for any purpose required and looked forward to the tests on the re-designed friction clutch, which were scheduled to take place in the near future. The original transmission system with an ahead and astern coupling was difficult enough, but when the alternator drive was added, things became really complicated. The hollow quill shaft was extended to carry the new coupling together with the extra oil passage required, but the forging for the first quill shaft which had passed inspection at the makers was found to have been bent on its journey to the shipyard and finally it was found necessary drastically to reduce the length of the quill shaft and redesign the oil inlets, to the transmission.

He thought it could be said that if thev had had the wealth of information which was now available, at the onset, they could have improved the final results, but the fact that the gas turbines and hydraulic transmission ran so successfully without any previous shop test or erection, was in itself a remarkable achievement, but they all regretted that this had given rise to the controversial question of gearbox losses and fuel consumption; these would have to remain unsolved until such times as the *Auris* returned to service.

His company believed that there was still a future for the gas turbine in the marine world. They felt that the capital costs could be drastically reduced and if renewed trials and further research were carried out, that a specific consumption could be achieved which would encourage further support for this type of machinery installation.

It should be noted that the maximum h.p. of the emergency steam turbine should be 300 and not 450 as quoted by the authors.

Mr. A. Cooper expressed his appreciation at being invited to contribute to the discussion. His company was concerned with the commercial development of the integrating power meter, designed by Pametrada and mentioned on page 93 of the paper. It was developed in order to avoid the limitations imposed by slip-ring methods such as were used in certain torsionmeters and thus to increase the accuracy.

The principle of the power meter would be seen by reference to the diagram (Fig. 27), which showed a more recent version. Two armatures C and D, which were steel rings, were fitted round the shaft and contained for example, ten mounted segments on brass blocks. As the shaft rotated, the armature segments passed the electromagnetic pick-ups A and B and each produced a pulse at the moment of passing.

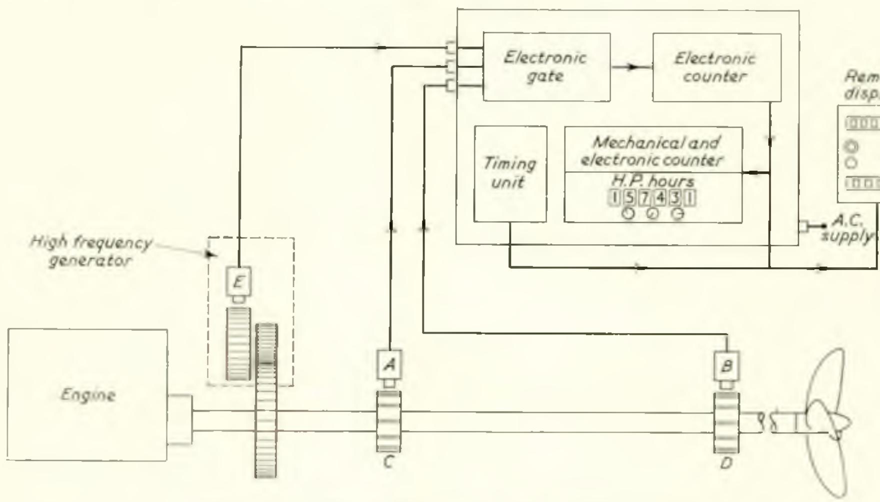


Fig. 27—*Block diagram of integrating power meter*

When torque was applied, the shaft twisted and one pulse lagged behind the other by an amount proportional to the torque. Preferably the rings should be positioned so that the length of the shaft between would twist about 2 deg. under full load.

A high frequency generator E, which consisted of a toothed wheel driven by the vessel's engine or shaft at about 1,000 r.p.m., produced an a.c. signal the frequency of which was proportional to the speed of the shaft. This signal was fed to an electronic gate, which was "opened" by a pulse from pick-up A and "closed" by a pulse from pick-up B. The cycles of the a.c. signal which passed through the gate were fed to frequency dividers, the output of which operated a mechanical counter.

Since the interval between the "open" and "close" pulses was proportional to torque, the counter reading increased at a rate proportional to the work done. The reading on the counter therefore represented the total power delivered, i.e. the shaft horsepower hours.

Provided that the torque/twist characteristic of the shaft was known, the accuracy of the meter could be as good as 0·25 per cent.

Remote displays could be provided as shown giving r.p.m. or s.h.p. over predetermined periods. The versatility of the instrument was shown by the fact that the *Auris* installation measured H.P. and L.P. turbine speed, percentage slip, s.h.p. and torque. It had now been transferred to the m.v. *Amoria*.

Another successful installation had been that on the *Antenor*. Two commercial instruments, which were in continuous use, had been provided in the *Oriana*, and others were in hand.

The company believed that instrumentation of this kind was of growing importance and that their method of measurement would help in the development of more efficient engines and hulls.

Mr. A. Lakinski said that the British shipping industry was sometimes criticized for a tendency to follow a well trodden path and a reluctance to try out new ideas. The case of the *Auris* was one instance where such criticism was unjustified.

Looking back at the early tests with this machinery it was regrettable that a complete investigation of blade vibrations was not undertaken at the time. These tests were omitted because the blading was similar to that used on the first *Auris* gas turbine, which operated quite well for 20,000 hours, because the matching of the compressors and turbines was good and because the bending stresses in the blading were very conservative. The bending stresses in stage 12 of the L.P. compressor blades, both rotor and stator, which ultimately failed in fatigue, were under 1 ton/sq. in. Indeed, one wondered whether the steady bending stress had any practical significance as far as blade fatigue was concerned. There was no trouble with surge, except possibly at dead slow ahead, when some signs of incipient surge could be detected in the L.P. compressor.

Blades of selected stages of the gas turbine were tested in a static rig for their natural frequencies, shortly after manufacture, but, as it happened, little useful information could have been gained from these tests. The blades ultimately failed in the fundametal transverse mode of vibration, the natural frequency being 370 c/s for stage 12 stator. It was of interest to note that the rotor blades passed the stator blades with this frequency when the propeller shaft rotated at 15 r.p.m.

The blade vibration tests carried out just before the ship was laid up established the following facts: the blades vibrated always at the fundamental transverse frequency; the amplitude was increasing from the inlet to the outlet of the compressor and was generally increasing with speed; at a constant speed the amplitude was quite small in the normal operating range of the machine, but increased rapidly as the compressor was approaching surge.

The fuel used during tests and in service was gas oil, but a 27-hr. run was made on high viscosity fuel oil. No difficulties were encountered, and since the turbine blading was generously designed there was every hope that the turbine would run successfully in service on heavy fuel oil. It might be of interest to mention that the Nairobi South gas turbine ran successfully for about 5,000 hours on commercially available Middle Eastern furnace oil before cleaning. Subsequently the turbine ran for 1,300 hours with only a slight deterioration in performance. Inspection at the end of that time showed only a slight deposit on the turbine blades. The combustion chamber of this turbine was lined with bricks.

The *Auris* turbine was controlled manually. Since a large proportion of the power developed by the L.P. turbine was absorbed by the L.P. compressor, there was no need for quick acting automatic controls, apart from the safety devices such as over-speed trips. Perhaps the authors could indicate if any difficulties were encountered on that score in bad weather conditions.

The turbine was started by a steam turbine supplied with steam generated in a Scotch boiler. This, of course, was not the only solution of the starting problem. A small air turbine or air motor, using compressed air stored in receivers, was feasible. Possibly a self-contained lightweight steam generating unit using a flash boiler, recently developed for starting aircraft turbines, could be adapted for heavier sets.

Mr. D. G. Penry, B.Sc.(Eng.) said that the paper was both factual and comprehensive and brought back memories of the interesting, if arduous, times spent on board the *Auris*.

Comparing the gas turbine cycles of the original *Auris* engine and the present engine, it could be said that the splitting of the compressor and the inclusion of the intercooler between the two had increased the overall efficiency by approximately 5 points. Splitting the compressor had also meant that the transmission line had a permanent load on it (this being true even with the blow-off valves open), which had improved the manœuvring characteristics of the set considerably. The returns, therefore, amply justified the present cycle, although it meant extra weight and complexity.

Concerning the testing itself he wished to make one or two observations. Firstly, Pametrada must be complimented on their integrating meter for measuring power. Although a prototype, this piece of equipment proved to be very accurate and extremely reliable. Only one minor fault occurred on it throughout the trials. It was, in fact, a very useful piece of equipment, for not only did it give the shaft power and torque but also gave accurate figures for the speeds of the H.P. and L.P. lines. Secondly, the question of air flow measurement and leakage had been mentioned. The air flow to the L.P. compressor was measured by a bell-mouth Venturi meter built to design drawings kindly supplied by the Rolls-Royce Co. Ltd. This particular Venturi was first used on the gas turbines supplied to the Nairobi Power Station and was at that time accurately calibrated by A.E.I. by means of a pitot traversed across the throat of the Venturi. Their calibrations tied in extremely well with the calibration supplied by Rolls-Royce Ltd. The design of the Venturi was within the recommendation given in B.S.1042.

As was pointed out in the paper, any leaks at the duct flanges were checked for and found to be nil. The known bleed-offs from the compressor to the turbine glands were carefully estimated and allowed for in calculating the gas turbine performance. As a check of the estimation, the largest of the bleeds, i.e. that from the seventh stage of the H.P. compressor to the H.P. turbine outlet gland and the L.P. turbine inlet gland, was measured by means of an orifice in the line, a sufficient straight length of duct being available to do this. The agreement between the estimated and the measured flow was extremely good. He suggested that this justified their values of the estimated bleeds to the other turbine glands. It was worth noting that the weight of fuel added in the combustion chamber was also allowed for in the performance calculations.

The *Auris* installation appeared to be somewhat cumbersome and almost filled the engine room. What were the comparative weights of and space occupied by alternative machinery for the same power output: i.e. how did conventional steam turbine machinery (including boilers) or Diesel engines compare with the *Auris* installation? Could the authors state the lengths to which they had gone to obtain reliable fuel flow meters?

Mr. P. Draper (Associate) wrote to say that he would like to draw attention to a vital component of the gas turbine, the combustion chamber, which had received little mention in the paper and none in the discussion.

He recalled that in early gas turbine development, during and after the war, the item on which least knowledge was available was the very high intensity combustion system. In fact, this proved difficult in the case of the aero engine; even with an easy fuel such as kerosene. Similarly, when heavy gas turbines were being designed from scratch, it was considered that there was information available concerning turbine, compressor and heat exchange techniques, but none on the sort of combustion necessary for using residual fuel oils.

Since the paper was an interesting and factual catalogue of the major and minor troubles encountered by the engineers, lack of mention of the combustion chamber was satisfying to the teams of designers who produced it, namely the Shell team under the late Mr. Isaac Lubbock and the constructor's group under the late Mr. B. Forsling.

It appeared that one refractory brick had been replaced and brief reference was made to the interesting multi-burner head. A great deal of careful thought, calculation and design was devoted to the system and advantage was taken of the experience with the two smaller refractory lined combustion chambers from the same stables, employed with such success in the earlier *Auris* gas turbine.

It was appreciated that the ship went out of commission before much operation was made on heavy residual fuel oil, but from earlier experience and from that on the Nairobi Plant, it was not anticipated that any difficulty would have been encountered in burning such fuels in the combustion chamber.

He complimented the engineers on their untiring enthusiasm in this pioneer work, in the face of numerous setbacks, and with them he regretted the necessity for withdrawal of the ship by the owners, who had already spent so much money and effort to introduce this form of ship propulsion. It appeared clear from the prototype work that further attempts could result in marine propulsion units which would be superior to the now well developed alternatives of steam turbine and large Diesel engine.

It was to be hoped that it would not be too long before another enterprising shipowner entered the arena.

Commander E. B. Good, R.N., considered that the Shell Tanker Company were to be congratulated on their courage in sponsoring the trials of the first British ocean-going gas turbine merchant ship. In advancing marine engineering there was still no substitute for the ultimate practical experience, which could only be gained during operating conditions at sea, and it was refreshing to find a British company prepared to back their judgement and bear the whole cost of this important experiment. In contrast, the Americans with their counterpart to the *Auris,* the *John Sergeant,* had had the benefit of their first ship being sponsored by the U.S. Maritime Commission.

Comparing these two "firsts", one was struck by the overall similarity in experience. In each case the gas turbine gave comparatively long hours of trouble-free operation while major outages in the ship's operation were caused by defects in the transmission, this despite the fact that in *Auris* hydraulic transmission was used and in *John Sergeant,* a variable pitch propeller. These experiences gave a strong hint as to the direction in which further developments should be made.

Although hydraulic transmission had a relatively long history the design adopted appeared to have been originated without necessarily utilizing established components and existing manufacturing knowledge. It was not without significance that the method of gas turbine manœuvring employed by the Admiralty in their classes of gas turbine boost warships had operated remarkably successfully and had used the existing manufacturing experience of a firm engaged in producing fluid couplings commercially.

The decision to provide extensive shipboard instrumentation was to be applauded and must clearly have enhanced the value of the experience gained. In particular, more details of the capacitance clearance monitor would be invaluable.

The problem of "missing power" had been read with much interest and recent "shore trials" experience with prototype naval gas turbine machinery confirmed how teasing this search could be and would be until such time as a reliable high speed torsionmeter could be produced. It would seem, however, as though the windage losses of the hydraulic transmission line were probably underestimated in the original gearing efficiency calculations.

It was agreed that the cause of aeration could be attributed to the design of the lubrication oil system, but it was not clear why a low oil depth in the drain tank should be a major cause, unless there was vortex formation at the pump suctions. If this was, in fact, the case then the trouble was not aeration but simply lack of oil!

Some of the causes for aeration in the system as designed were:

i) The use of hydraulic couplings which, if not effectively designed, can give rise to a high degree of aeration.
ii) Free standing pumps with relatively high suction lifts, which greatly aggravate the effect of entrained and dissolved air in the oil.
iii) The use of magnetic strainers in the suction pipe which increases the suction lift.
iv) The use of pump relief valves which return direct to their pump suction instead of to the drain tank, thereby minimizing the possibility of air being released.
v) The use of a gravity tank which forms the first reasonable opportunity for air to escape—the air purging valves fitted at coolers and filters are not nearly as effective as the tank. In the latter case the air rises immediately to the surface where the top take-off lines are located and which were those reported air locked.

Correspondence

Mr. M. C. Jourdain (Member) in his contribution wrote that being used to attending sea-trials, he fully appreciated the care taken in the fitting of the instrumentation on the *Auris* and the accuracy of the results obtained by the authors.

His attention had been drawn to a note in Fig. 3: "Siemens Ford torsionmeter modified for transmission torque measurement". He would like to know more about the instrument and the nature of this modification.

Fig. 12, on a crash-stop trial, was very similar to diagrams obtained on larger steam ships.

He had paid special attention to Table VI, referring to measured mile trials. The correlation between torque and thrust measurements was so good that it showed a systematic difference between the Northward and the Southward runs, very small in the loaded condition, except for the last group, but substantial in the light condition for all runs. The reason for that was probably a loss of efficiency of the propeller in a following sea. The r.p.m. measurements were also fair and it seemed that the least accuracy showed in connexion with the speed, perhaps because the tidal stream was not perfectly constant, as was usually assumed when each group consisted of two runs. Some times weather instability was also a cause for scatter of the plot.

From the voyage data it appeared that the apparent power-wake was much less than on the trials; it was a pity that the lack of thrust measurements on voyage prevented a cross-check of this fact.

It was worth noting the gain of 600 h.p. due to the cleaning of the propeller and also the effect of an eventual ice-blasting of the hull roughness.

Dr. J. J. McMullen, B.S., M.S., Dr. Ing. (Member), wrote that the authors were to be congratulated on their excellent presentation of the background and facts in connexion with the installation of the gas turbine unit in the *Auris*. Such complete presentations were necessary in order to permit overall evaluation and to disseminate the information throughout the industry.

In reviewing the paper, it was obvious that the *Auris* suffered from the same problems that most marine development programmes encountered; namely they were burdened from the very beginning by a series of compromises which were imposed by either financial restrictions and/or conservatism. However, in addition to these normal handicaps, it appeared that the *Auris* suffered severely from long delays in manufacture and supply of equipment. It would seem that the lesson to be learned was the fact that marine equipment could no longer be designed and manufactured on a special basis. Admittedly, in the years past, the marine field had tended to lead the industrial field in the development of power equipment and industry had taken over these developments from the marine field. However, more recently, it had become apparent that developments in both the industrial and aircraft fields could readily be adapted to marine purposes. In addition to reducing overall costs, this adaptation of equipment and machinery from other industries also ensured greater standardization and availability of parts and spares. It was obvious that the *Auris* would have been processed much more readily if such equipment had been adapted to marine requirements from either the industrial or aircraft fields for this specific purpose.

Moreover, in considering the installation in the *Auris*, it had to be evaluated on the basis of the year 1953. At that time, gas turbine designers tended to base their designs on steam turbine technology, both from the standpoint of weight and arrangement of the individual components. Since 1953, the trend had gone through two successive steps; namely, the fifty-fifty approach representing a midpoint between steam and aircraft gas turbine technology, such as represented by the gas turbine unit installed in the g.t.s. *John Sergeant*, and now completely to the aircraft concept. In view of the millions of hours of experience being obtained from aircraft gas turbine operation and the trend towards ever increasing inlet temperatures and longer life, it was apparent that these units would soon form the basis for an overall marine propulsion plant. It was his opinion that installations such as the *Auris* and the *John Sergeant* would never be repeated. However this did not mean that they were unsuccessful, but merely that they were steps towards the ultimate goal which should be obtained in the near future by the installation of an aircraft gas turbine unit, adapted to marine requirements, to the propulsion system of a merchant ship. It was hoped, however, that the installation would be based on a completely new ship and design and not handicapped by a conversion such as in the past. There was no question but that a new ship, properly designed with the concept of utilizing an aircraft type gas turbine unit, could result in higher outputs and lower space and weight than those of steam plants. It was recognized that the fuel problem still remained, but the reduced initial cost, the reduced cost of maintenance and repairs and reduced operating costs would more than offset the increase in fuel oil costs represented by the difference between Bunker C and Diesel fuel. As a matter of fact, the elimination of one or two operators would more than offset such a difference.

Mr. A. Park wrote that his first reaction on reading the paper was recognition of its honesty of record and purpose. The authors had missed nothing.

His comments were based on shop floor experience, having been connected with the *Auris* project for maintenance and repairs since its earliest days, also with the building and erection, on board, of the machinery, the trials of which the authors had so ably described.

Mr. Duggan had mentioned the remarkable achievement of the way in which the three principals had co-operated in this project but he thought that a great deal of the credit for this must go to Mr. Duggan as the co-ordinator in chief.

It seemed a great tragedy, that after bringing this project to a successful conclusion (technically it had been a great success) that the country's shipowners had not carried the banner further. If the present stage of development had been reached ten years earlier, would it have made a difference? Or was the industry now waiting for atomic energy to power ships in the future, while for the present foreign Diesel engines held sway.

Mr. Duggan had mentioned ill-luck in the building, but with regard to capital cost, the circumstances of the *Auris* project could not have been worse in respect to labour costs. Repair rates, plus allowances for the last cargo, plus erecting the engines on board, all added up to make an expensive job more expensive. New designs, applied under proper conditions, could bring about a reduction of anything up to 50 per cent in direct labour costs.

The time required to do the job was lengthy by ordinary standards, but it could be seen in its proper perspective, if the problems to be overcome in a prototype of this size, were considered; it should be remembered that the plant to manufacture the design was not especially laid down, but that the work was carried out in a job shop, the technique being formulated as the work progressed.

This was not the time to comment in detail of the trials and tribulations through which the *Auris* passed but, as the paper had emphasized, the marine industry could pioneer and bring to a successful conclusion, new designs and given the chance could put the gas turbine in the place once held by the steam turbine.

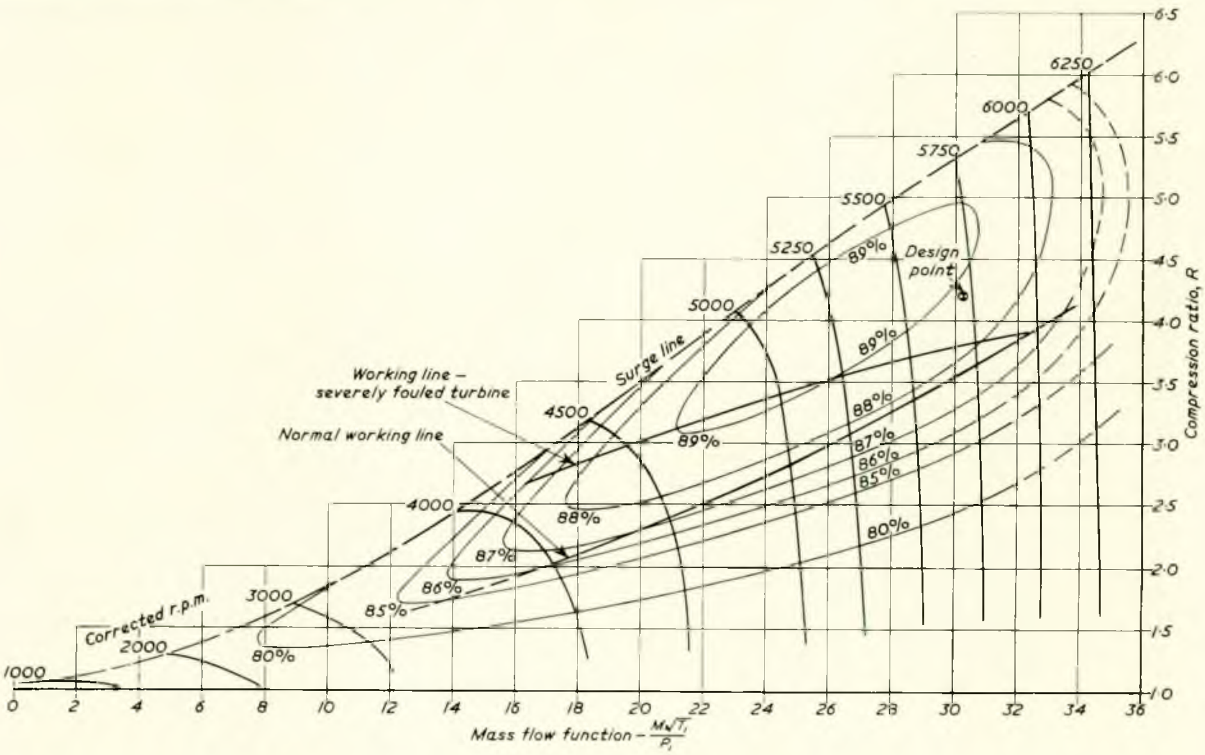
Thanks were due to the Shell organization for taking this project so far on their own.

Authors' Reply

The authors expressed their appreciation of the wide interest taken in the discussion of the paper. This clearly illustrated the wide variety of problems that were still encountered with marine machinery.

Whilst Captain Atkins suggested that the paper illustrated the weaknesses of the gas turbine for ship propulsion the authors had endeavoured to show that whilst a number of problems were encountered with the transmission and auxiliaries, only three problems were encountered with the main turbine. Two of these, the L.P. compressor clearances and L.P. turbine oil seal, were solved before the ship left for sea trials. The remaining problem of L.P. compressor blade failures could be cured but had not been proved in service. With an experimental engine knowledge had to be paid for, with a commercial unit the owner expected a financial return.

The lining of the second *Auris* combustion chamber was 80 per cent alumina refractory, which was used with great success for many thousands of hours on the first combustion



R = Compression ratio based on static pressure rise from intake to outlet flange.

M = Mass flow – lb./sec.

T_1 = Intake temperature — deg. K.

P_1 = Intake pressure – lb./sq. in.

Mass flow at design inlet conditions $= \frac{M\sqrt{T_1}}{P_1} \times 0{\cdot}858$lb./sec.

$$\eta t = \frac{\left[R^{\frac{\delta-1}{\delta}} - 1\right]}{\frac{\Delta t}{T_1}}$$

Where: Δt = Temperature rise

$\delta = 1{\cdot}404$

FIG. 28—*Auris marine gas turbine axial compressor characteristics Isentropic efficiency (ηt) shown as contours on characteristics*

chambers. The authors had no fear for these from sodium and vanadium attack and the only difficulty experienced in the past had been due to swelling and chemical additive attack. The life was expected to be two years and a complete change could be made in two hours.

In general closed-cycle gas turbines had a greater number of heavy components and were more complicated. The great value of gas turbines in the future would be simplicity. With a closed cycle the high temperature problems were transferred from a rotating unit to a heat exchanger. Gas tightness of casings was required with any machinery and the authors had bitter experiences of superheated H.P. steam casings leaking.

It had been stated in the paper how well the compressors behaved. Table III illustrated perhaps more clearly the high performance achieved and only axial compressor and gas turbine designers could fully appreciate the achievement of putting in series and matching with turbines those two compressors, without any test bed trials. The compressors with load on the engine were far from the surge line, unlike the case illustrated, and it was never necessary to wait for compressors to stop, prior to engaging the starting turbine which controlled surge, as an SSS clutch was incorporated.

The original *Auris* compressor, whose blading was basically designed in 1932, ran without surging or blade failure for over 20,510 hours. Fig. 28 showed the compressor characteristic for the first *Auris* gas turbine. Imposed on this was the design point and the normal working line. In addition there was the actual working line which was plotted during a high viscosity test run with the most difficult fuel ever tried. It would be seen from this that even under the worst conditions the compressor was well within the surge line.

Captain Atkins was reminded that the H.P. blow-off valves were fitted to dissipate energy while manœuvring and had nothing to do with surge. The L.P. blow-off valve was only used after the blade failure.

The authors also considered it unwise to compare a prototype with production engines. The number of engines produced by the aircraft industry before achieving a service period of 3,000 hours probably ran into thousands. Early compressors had many failures, and steam turbines were still having failures. The entire design and testing of the H.P. and L.P. axial compressor blades for the last *Auris* project was carried out with the following expenditure of effort. Design engineer's time—20 man weeks, draughtsmen—50 man weeks and testing—6 man weeks. Clearly this was a fraction of the time envisaged by Captain Atkins.

The development of steam turbines to a comparable stage had occupied a vast number of men and many firms. The authors also felt it was relevant to mention, in view of Captain Atkins' remarks on cooling of stressed casings, that steam turbine designers appeared to be in difficulties with double casing H.P. turbines, which when compared with the *Auris* gas turbine were relatively cool.

With regard to air leakage it had been mentioned on page 105 of the paper that H.P. ducting flanges, blow-off valves, and all heat exchanger tubes were tested after the sea trials had been completed. As a result of these tests the authors, manufacturers and designers were convinced that no leakage occurred from casing or ducting joints.

The bearing design of this particular engine made it impossible for compressed air or gas to leak into the lubricating oil system.

Very little reliable data was available for loss of power with astern steam turbines; particularly with leaking manœuvring valves. There were many steam turbine ships with time and temperature limits for running astern. Bent rotors and distorted casings due to astern turbines were once comparatively common.

The authors were very pleased to see Fregatten-Kapitaen Moeller and particularly to hear of the progress made in Germany with gas turbines. The suggestion of integral packaged units occupying one-fifth of the space of a normal Diesel engine was topical. With the advent of fully automated ships this type of unit would be an essential development.

An investigation was carried out in 1954 to see whether a variable pitch propeller could be incorporated into the *Auris* project. Whilst reliability was then still an unknown factor for variable pitch propellers absorbing 5,000 s.h.p., the cost was prohibitive. Apart from the essential modifications to the stern frame and rudder, the most acceptable tender for the propeller with pneumatic control equipment was £37,585. This did not of course include the cost of a gearbox (£23,000).

The developed hydraulic transmission without gearbox on the other hand was estimated at £10,000 in 1954. It was also interesting to note that another quotation for a variable pitch propeller gave an estimated weight for propeller, shafting and control gear at 42 tons.

Mr. Adams had referred to the maximum turbine inlet temperature of 650 deg. C. fixing the whole design of the *Auris* II gas turbine. In particular he suggested that it was responsible for complicating the cycle. The authors pointed out that this was true in 1953 but the next design in 1960 had a much simpler cycle with a considerably improved performance and yet the same inlet temperature limitation. Presumably any reduction of duct losses would still be of importance in a high temperature unit, and of course blading design might have to deal with even worse deposits of a different composition at a higher temperature. The spare H.P. turbine was installed not because of the temperature limitation but for the following reasons:

a) Safety of the ship. The emergency steam turbine was not a satisfactory solution. A spare H.P. turbine was at least a sound proposition.
b) This unit was expected to give most trouble and had the longest delivery of spare parts.
c) Fuel experiments were planned and experience with *Auris* I had already illustrated the big effects of different materials.

TABLE VIII.—SPECIFIC FUEL RATE LB./S.H.P. HR. (ALL PURPOSES)

Ship	Designed power s.h.p.	Fuel Rate: Designed	Fuel Rate: On trials when new	Fuel Rate: Voyage rate for first year of service	Fuel Rate: Repeat trial after period of service post docking
Steam Ship A	7,500	0·565	0·583	0·6814	0·647
Steam Ship B	7,500	0·565	0·597	0·6478	0·629
Steam Ship C	7,500	0·602	0·629	—	—
Steam Ship D	7,500	0·602	0·656	0·6750	0·764
Steam Turbo-Electric Ship E	7,500	0·744	0·793	—	—
Steam Ship F	13,000	0·527	0·543	No	—
Steam Ship G	13,000	0·527	0·516	Torsion Meter	—
Steam Ship H	13,000	0·527	0·539	fitted	0·623
Steam Ship J	13,000	0·527	0·589	0·569	0·563
Motor Ship K	8,000	0·364	0·361	—	0·3899

The authors did not consider that a high temperature gas turbine was suitable for marine propulsion when marine fuel oil had to be lifted at any port in the world. The differential price in Las Palmas between gas oil and marine fuel oil in 1953 was £5 12s. 6d. per ton. The differential had to date changed very little. Even with the new sources of crude supply, the choice still clearly had to lie between a high temperature set on gas oil or a low temperature set on marine fuel oil. Even if the ship was on a specialized trade this choice eventually had to be faced. If space and weight were all important it was feasible that the more compact high temperature unit would be chosen. Some cargo ships were virtually floating warehouses with a minimum engine room space and engines for occasional voyages between weeks in port. With the long life of tankers, few shut-downs with little weight or space restriction made the low temperature unit the best choice.

The correlation between design, trial and voyage performance had for years been difficult and masked by the lack of reliable instruments on ships. Few ships had had such full treatment as the *Auris* II plant. In general four pieces of information were watched by ship operators, namely: —ship's speed, fuel consumption, shaft revolutions and in fewer cases shaft horsepower. Of these, ship's speed, fuel consumption and power were extremely difficult to measure accurately. Table VIII was incorporated to show how much scatter was encountered on all purpose specific fuel rate on new ships of the same class and also to illustrate how the *Auris* trial and voyage results compared with "conventional" plants of similar power.

The Diesel electric plant as originally fitted in *Auris* was surprisingly economical on fuel (marine Diesel oil). The original trial figures were not now available but each engine had a specific rate of 0·354lb./b.h.p. hr. at service conditions. There were of course losses with the multiple unit electric transmission and the ship never operated at its designed speed of 12 knots. The average of light and loaded voyages was slightly over 10 knots. At this condition the Diesel engines could steam the two Scotch boilers satisfactorily with exhaust heat and the fuel consumed was only 10 tons/day (0·584lb./s.h.p. hr.). This of course tended to give a completely false impression for, at one period, 80 gallons of lubricating oil were consumed per day and the heat from an exciter motor materially assisted in raising the engine room temperature to 140 deg. F. in the tropics. This was partially rectified by fitting four large fans.

The advantages and disadvantages between the Diesel electric plant and the final gas turbine plant could be briefly summarized as follows:

Advantages of the Diesel Electric Plant

a) It was very quick starting.
b) Running maintenance could be carried out as the ship steamed satisfactorily on three of the four Diesel engines. A complete stoppage of the ship only occurred twice in the authors' experience.
c) The machinery weight was 590 tons in 1948 as compared with 839 tons in 1959. (Maximum shaft horsepower was increased from 3,750 to 4,950.)
d) Astern power was the same as the ahead power.
e) Part load fuel consumption was good, although this is not normally required in a tanker.

Advantages of the Gas Turbine/Hydraulic Plant

a) Practically no maintenance, fewer auxiliaries required, only two men required on watch (for union rules).
b) High speed of manœuvring.
c) Excellent cargo pumping capabilities.
d) Centralized control.
e) Excellent engine room ventilation, no fumes, clean, little vibration.

It should be emphasized that whilst the concept of the multiple Diesel electric plant was good there were few engines of the right size available in 1948. The misbehaviour of the four Diesels was exceptional and a more modern version with electric cargo pumping might be attractive. The big disadvantage of the Diesel electric plant, in the authors' view, was the vast number of parts such as valves, piston rings, bearings, fuel pumps, turbo-blowers, etc., that were required. An indication of the work involved could be illustrated by the fact that 96 pistons were withdrawn, changed or re-ringed in one three-month period and an extra engineer had to be carried.

Although experiments were carried out with one Diesel engine burning fuel oil in 1950 it was not a practical possibility for the whole plant.

From the point of view of a shipowner, preference must clearly go to the gas turbine version which was capable of long periods of service without maintenance, coupled from the point of view of tanker owners, with good pumping facilities.

Mr. Adams asked why steam turbine machinery had been ordered in the past and why it was still being ordered. Before 1939 there was no question that, for powers above 5,000 s.h.p., the turbine reigned supreme. It was the authors' view that whilst this marginal power had probably increased now to 8,000 s.h.p. it was only true for tankers. These ships required long periods of unbroken service and also considerable steam plant for pumping and cargo heating. It was clearly a considerable capital levy on a Diesel (or gas turbine plant) to have to install a large steam plant. For cargo ships the position was different; "floating warehouses" had plenty of time to carry out the very large amount of maintenance needed on a Diesel plant. Passengers were not amenable to vibration and this had to influence the choice for passenger ships on all but special short haul routes. With special applications quick starting was important.

It was clearly not sufficient for the fuel meter on the motor ship to rotate at half the speed of the steam ship of comparable power. The lubricating oil consumption of the *Auris* gas turbine plant could hardly be measured; that of a steam turbine plant was often less than 2 gal./day. A comparable motor ship used 7,000 gallons of lubricating oil in the first six months of its life. This was equivalent to about 39 days steaming per year at service power if it had been fuel oil on a cost basis. The authors' views could be summarized thus:

A steam turbine plant is more flexible in arrangement, it is cleaner, requires less maintenance, is almost vibration free, and is capable of a very big power range for little relative increase in size. Boilers and evaporators are troublesome, Diesel engines are designed to the limit, and are derated for service conditions. They have many reciprocating parts. As mentioned in the last paragraph of the paper both types of engine are on their limits of practical design and only relatively small improvements can be gained for heavy expenditure involved, such as the use of special materials required for advanced steam conditions.

The authors wished to enlarge on Mr. Opelt's remarks concerning Table III. The specific fuel consumptions in this Table were of course for propulsion only.

Typical "all purpose" rates were shown in Table VII and if required the auxiliary boiler consumption of 260lb./hr. could be added to the figures in Table III.

The full power and Southampton to Curaçao columns were with the friction clutch engaged; both Curaçao to England columns were with ahead coupling engaged.

The advent of the blow-off valves would certainly have made the use of electric propulsion much easier. The original scheme with two units was attractive as an experimental plant where tests could have been conducted on one unit. The authors were not in favour of the geared induction scheme. The transmission efficiency would not have been better than the hydraulic system. A single synchronous electric scheme would have been attractive and cargo pumping would have been simplified but the authors believed it would be much easier to lose excitation than the lubricating oil supply to the hydraulic couplings and hence the load on the L.P. turbine. In addition electrical control equipment was much more difficult to maintain than the simpler robust hydraulic components.

This type of problem did not arise until four or five years of service.

A further consideration was that shipbuilders did not make electrical equipment but in the future there was no doubt that smaller compact packaged units would be attractive. Electric transmission was ideal for this type of application, and probably for the all electric, no steam, gas turbine ship.

Mr. Fowler had drawn attention to the weight of the intercooler, whilst the authors considered this item as main machinery it would of course, have been beneficial to reduce its weight of $16\frac{3}{4}$ tons. No difficulty was experienced with the plain tube design but the access doors were very unmanageable. In service several opportunities occurred when the tube banks could be examined on the air side. Apart from the first few rows it was very clean and no difficulty was experienced when washing it out. The fact that the intercooler operated trouble free for all the trials and service periods fully justified its design, in the authors' opinion. To give an indication of the large quantity of fresh water discharged from the intercooler forward drains it might be of interest that the valves were 2-in. bore, and the discharge pipes often ran full bore for several hours. The water was proved fresh as it only just clouded a silver nitrate solution. It was believed that a wave of water built up at the forward end of the intercooler. There was little doubt that this condensation helped to keep the intercooler clean.

Mr. Darling was evidently confused by the "missing power". Referring to Fig. 13 the bottom curve "Quill shaft removed" indicated a loss of nearly 400 h.p. at 116 r.p.m. As the gearbox was disconnected this loss was clearly within the gas turbine system, calculation, or instrumentation; in fact it was unaccounted for by the engine designers. The 1,400 h.p. mentioned by Mr. Darling was the total difference between calculated and measured power. It had thus to be implied that the difference of 1,000 h.p. was within the transmission, for the system of calculation for the bottom curve was also applied to the other curves. Now of this 1,000 h.p. the gear transmission designers calculated a loss of 350 to 400 h.p., and as mentioned on page 104 a test using oil flow and temperature rise (a notoriously unreliable method) gave 416 h.p. The windage losses for the ahead coupling were estimated at 4 h.p., the astern converter at 5 h.p. and the alternator coupling at $7\frac{1}{2}$ h.p. During one overspeed test the losses were measured on one ahead coupling wheel at 70 kW. power in excess of estimated losses.

The authors had mentioned on page 95 how the coupling casing had to be finned, and cooling oil sprays had to be fitted to dissipate heat.

It was found most unwise to run with the sprayers turned off and in spite of this the gearbox was always referred to as "the hot end" by the engineers.

When comparing fuel consumption of marine steam

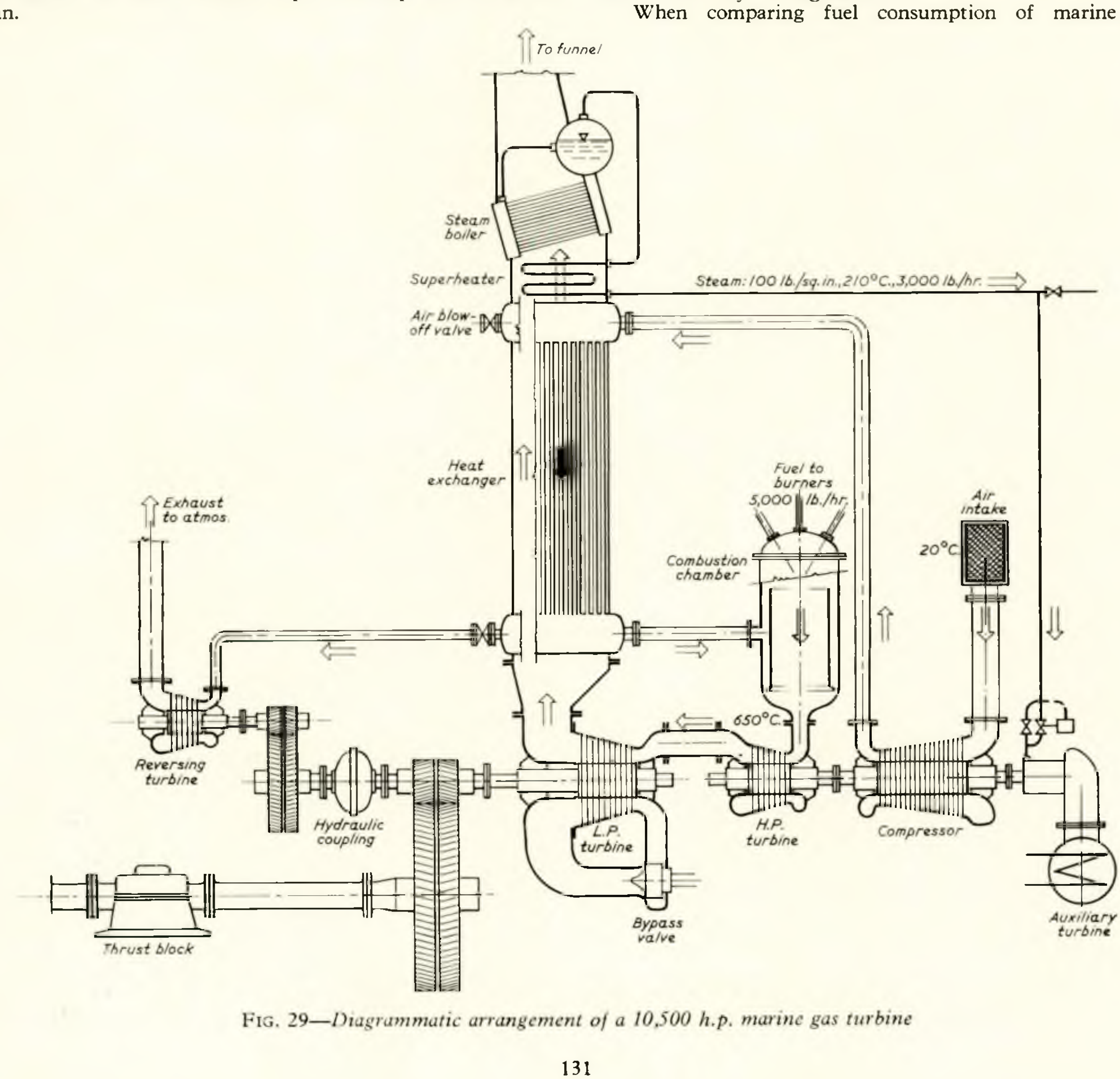


FIG. 29—*Diagrammatic arrangement of a 10,500 h.p. marine gas turbine*

turbines great care was required. Firstly the overall picture had to be considered, and secondly the size of turbine had to be given consideration. It would be interesting to know if Mr. Darling's figures relating to 600 ships were all less than 8,000 s.h.p. Table VIII showed the type of fuel rate achieved by ships on trial, and in service, and the ships A and B were presumably of the latest design of double casing turbine produced by the Association which Mr. Darling represented, when they entered into service two years previously. Boiler manufacturers had been striving for many years to gain another 1 per cent but clearly they were on limits. Raising the steam temperature had been tried but the expense of special materials, and also the extra cost of working them had lead to a trend in actually reducing the temperature to within practical limits.

The authors agreed that a complicated cycle gas turbine, or steam turbine, was not a practical proposition even though the fuel rate was attractive. In this respect the investigation carried out two years previously for the third stage of development might be of interest. In this case simplicity was of paramount importance but steam was still required as the investigation was for a tanker plant.

The diagram (Fig. 29) showed that only two turbines and one compressor were required for the ahead running and an air turbine of 40 per cent ahead power was used for astern movement. The transmission utilized one hydraulic coupling (or other suitable drive) which was only in use for astern running. The bypass valve design was very well proved in the first *Auris* plant.

Brief particulars of the designed performance were as follows:

Ambient temperature	20 deg. C. (68 deg. F.)
Turbine inlet temperature ...	650 deg. C. (1,200 deg. F.)
Total fuel consumption (based on a net (lower) heat value of 18,000 B.t.u./lb.)	5,000lb./hr.
Output at L.P. turbine coupling	10,500 h.p.
Output at ship auxiliary alternator terminals	300 kW.
Steam for heating	3,000lb./hr.
Steam pressure	100lb./sq. in. gauge
Steam temperature	310 deg. C. (410 deg. F.)

Raising the temperature to 675 deg. C. (1,250 deg. F.) increased the fuel consumption by about 5 per cent and the net output by about 10 per cent.

It would be seen that the turbine inlet temperature was still very modest yet the specific fuel rate of 0·49lb./s.h.p. hr. was better than one of the latest steam plants of twice the power. By raising the inlet temperature to 675 deg. C. the specific fuel rate would be about 0·47lb./s.h.p. hr.

The auxiliary steam turbine was also used for starting and the high pressure gas turbine casing and rotors would be cooled thus enabling the cheaper ferritic materials to be employed, bigger fluctuations of gas temperature to be accepted and quicker starting. It was hoped that this example would show that gas turbine design was still just beginning, and also that an extremely simple plant would not only have a much better fuel rate than a steam plant of comparable size, but it would also be very close to the very complicated gas turbine plant proposed by Mr. Darling.

Looking to the future with fully automatic machinery there was little doubt that gas turbines were ideal, whereas the multiplicity of a steam cycle was very difficult and costly.

Liquid cooled gas turbines were very attractive for a low fuel rate, but fuel cost was only one consideration. The authors were full of admiration for Pametrada for carrying out advanced work of this nature for it was doubtful whether a commercial concern could take such a large step. If liquid cooling meant complication and the necessity of extra engineers or additional maintenance, a very strong case had to be made with the fuel saved. It was assumed that although the gas temperature was 1,200 deg. C. the metal surface temperature would be below 650 deg. C. and therefore deposition and corrosion problems would be no worse than at present. It was relevant to mention that unpublished work at Thornton Research Centre showed that with slightly reducing conditions on the fuel rich side of stoichiometric, the corrosion rate was very substantially reduced even with metal temperatures of 800 deg. C. These conditions might of course raise other problems but in the future this could have a big influence on gas turbine design.

The authors were indebted to Mr. Harris for his remarks concerning the L.P. compressor blade failure and the explanation of possible conditions in the diffuser.

The figure of 65 per cent reduction in fatigue strength due to marine corrosive atmosphere was interesting but not confirmed by the steel makers.

The authors had purposely omitted comments on the economics of the gas turbine plant. Owing to the experimental nature of the *Auris* installation it would clearly bear little comparison with production models.

There were several basic economical considerations apart from overall factors like the size of ship for the required trade. The cost of operating the engine in pence per s.h.p. hr. could be plotted as a straight line relationship with capital cost (above or below the standard) and the slope was equal to the capital charges percentage. If a gas turbine plant could lie below this line then it was a sound economical proposition. Naturally the fuel price and fuel differential varied with different operators and trades and also the capital charge percentage. Some reasons had been given earlier why the authors believed that steam turbine plant was still being ordered, although Diesel specific fuel rates were so advantageous. A gas turbine had to show a clear advantage over steam or Diesel. The gas turbine fuel rate had obviously to be better than the steam turbine fuel rate but not necessarily as good as for a Diesel engine. However where the gas turbine fuel rate was not as good as that of Diesel its maintenance had to be perfect. In the authors' opinion it had to burn marine fuel oil, for now the majority of oil burning ships at sea were operating on fuel oil.

The projected gas turbine plant mentioned in reply to Mr. Darling was compared on a strictly fair basis of one of 12 ships against a conventional steam ship of similar size. It had been shown that the fuel rate was excellent, simplicity was achieved, it was intended to burn fuel oil, its weight was satisfactory, yet it was not acceptable. Even allowing for considerably reduced maintenance costs over several years the extra capital cost, estimated at 25·4 per cent of the conventional plant, made it a completely uneconomical proposition.

A test, using the cargo pump alternator as an emergency propulsion motor driven by the auxiliary steam and Diesel alternators, was planned on the *Auris*. It was believed that this would be superior to the emergency steam turbine installation. In general the authors would favour whichever system had the most plant available without additional complication or heavy capital cost. For instance, with a tanker containing a large boiler but small auxiliary alternator, they would favour an emergency turbine. If there was no boiler but adequate auxiliary power then clearly an electric motor would be advantageous.

The suggested method of measuring gear losses was most interesting. It would have been a useful check if the output shaft had been disconnected.

Dr. Brown had presumably compared the voyage fuel rate of *Auris* (Table VII) with builders' trial figures for the *Aluco* class of ship. It was hoped that Table VIII and subsequent remarks would illustrate the authors' views on this subject. It was, of course unwise to compare large plants with small. Dr. Brown's remarks concerning the friction clutch not being able to cope with the increased loads, due to torsional vibration, helped to emphasize the authors' views that however carefully tests were carried out ashore or under laboratory conditions, the final proving had to be full scale, at sea.

The authors felt that he had perhaps laid too much emphasis on the complications arising from the decision to fit the cargo pumping alternator; this was later fully justified and enabled excellent cargo discharge rates to be achieved. The seals associated with that unit did in fact only give anxiety once, whereas the leakage between the astern converter and ahead coupling were a continuous problem. The quill shaft to the fixed tube was one sealing problem, but the quill shaft to pinion, which could both move within their independent bearings, was vastly more complicated, and failure could be disastrous.

Even if the suggestion that it was wrong to have a compressor on the power turbine was accepted, it was interesting to recall that the *Auris* II cycle did in fact work very well and could be manœuvred down to 35 r.p.m. (29 per cent of ahead speed), without resorting to filling both couplings. The use of the single centre burner for slow speed running was also very satisfactory. A free power turbine meant that safety devices had to operate with 100 per cent reliability and effect.

Dr. Brown's remarks concerning the difficulty of estimating gas turbine output from temperature and mass flow measurement could, of course, be applied to trying to measure gearbox losses by temperature rise and oil flow. The multitude of drain returns on the *Auris* gearbox made it even more difficult than trying to measure a single duct temperature!

The correction in maximum power for the emergency steam turbine mentioned by Mr. Holmes was presumably due to the steam consumed by the forced draught fan and feed pump, for on test the Scotch boiler produced slightly over its full rated evaporation of 12,000lb./hr. of steam.

In reply to Mr. Lakinski the authors confirmed that no difficulty was ever encountered with controlling the main engine in bad weather; apart from the failure of the solenoid valve mentioned on page 109. With the original gas turbine plant and a free power turbine "racing" resulted in the L.P. turbine speed change of 1,200 to 3,000 r.p.m. ten to twelve times a minute. With the second installation this effect was so small that to the authors' knowledge it was never recorded.

Both Mr. Lakinski and Mr. Penry had indicated that the two compressor cycle was fully justified for ease of control and improved performance. This, it would be noted was in direct contrast with Dr. Brown's second paragraph. The authors agreed that the double compressor was fully justified for the *Hemisinus* in 1953 for at that time the bypass valve on the *Auris* I plant had not run 20,000 hours and the blow-off valves did not even exist on paper. It was also justified for reducing the complexity of automatic controls, but now with the experience gained, the desire for simplicity and reduction in capital cost the free power turbine was attractive.

The authors wanted to emphasize that the *Auris* II plant gave the impression of filling the engine room, due to:

a) The power being more than that for which the ship was designed.
b) Extensive use of existing auxiliaries and a tendency to "double up" due to the experimental nature of the plant.
c) Test bed instrumentation.

The weight of 839 tons for the *Auris* II machinery quoted in reply to Mr. Adams was as accurate as could possibly be measured. It included auxiliaries, pipework, shafting and wrought iron work.

The equivalent weight for an 8,000 s.h.p. steam plant by the same builders was 960 tons. A comparison between the same power steam and Diesel (British) engines showed the latter to be 120 tons heavier. The same steam plant compared to a continental Diesel plant showed the latter to be 163 tons heavier. The actual weight of a 7,000 h.p. Diesel engine was 640 tons. This could be compared with the *Auris* gas turbine of 249 tons.

The weight comparison was of course much more complicated than this statement. For instance as the fuel meter only went round at three-fifths the speed with the Diesel plant, the saving in weight of bunkers for similar steaming range was considerable (about 728 tons on a 37 day voyage). Again the increased lubricating oil storage weight had to be deducted from the Diesel's credit (54 tons).

Space occupied by machinery is normally fixed by regulation and penalties between 9 per cent and 13 per cent of the gross tonnage (space) of the ship. What was more important was the shape of the space and the shape of the engine. Clearly a Diesel engine was just one large mass, a steam turbine, (particularly with electric transmission) was much more flexible and in between was the gas turbine which was closely tied to ducting requirements.

The *Auris* engine and gearbox were slightly longer and much narrower than the equivalent Diesel engine. The height could hardly be compared as the heat exchanger also acted as an exhaust duct.

Difficulty in measuring accurate fuel flow was first encountered during the seagoing experiments on *Auris* I. Tank soundings either by dips or specially calibrated gauges were poor. In 1953 a specially made volumetric cylinder with piston displacement was used. This gave trouble due to "O" rings swelling on the timing device. For *Auris* II not only were all fuel tanks calibrated with water but the pneumacator gauges were very carefully checked.

A high pressure (700lb./sq. in.) piston type fuel meter was used on this installation (see page 93) and it had since successfully operated for long periods on heated fuel oil. Three other varieties of this type proved unsuccessful. Unfortunately most meters were volumetric and the accuracy desired necessitated a very good check on the fuel gravity. The authors had closely examined eleven different types of meter and tested seven. This investigation included piston, rotary, swash plate, meshing gears, orifice, electromagnetic, and sonic meters. Only one met the required accuracy of better than 0·5 per cent on heated fuel oil but this had only just completed 1,000 hours operation. It was hoped that a mass flow meter (at present experimental) of similar principle might prove to be the answer.

Mr. Draper could be assured that the combustion chamber was a splendid design, it operated trouble free from the start, and even during the atmospheric rig tests, after manufacture at Rugby, it never gave any trouble, or even required any modification. The triple burner head, pneumatically operated igniters, and even the carbon rod igniters themselves were excellent. The counter balance system, heat shields, brick lugs, sight windows and hinged cover all proved successful. The only fault which might have had serious consequences was the burner pintle failure mentioned on page 104.

In reply to Commander Good the following specification details of the capacitance clearance monitor might be helpful.

The clearance monitor was designed to measure the clearance between a fixed insulated probe and any earthed conducting part which moved or vibrated so as to produce a change in the capacity between itself and the probe. It could detect changes in capacity from 0·1 to 0·00005 of a Pf. throughout the frequency range 10 c/s to 25 Kc/s.

The monitor could be used in turbines or compressors to measure the radial and axial clearance between a probe in the casing and the moving blades. Radial and axial eccentricity could be studied in conjunction with an oscilloscope, and irregularities in the blade spacing could also be seen.

Alternatively, the monitor would measure the R.M.S. amplitude of a vibration without introducing any mechanical loading on the subject under examination. If an oscilloscope or wave analyser was used the vibration could be studied in detail.

Typical sensitivity:

Minimum detectable capacity change	5×10^{-5} Pf.
Maximum capacity change ...	0·1 Pf.
Maximum clearance	0·15in.
Minimum clearance	0·005in.

(Due to dielectric breakdown of the airgap)

Performance:

Probe voltage (through 20 Megohms)	250V.
Amplifier gain (voltage) ...	324

Frequency response ...	$\pm$1 per cent 10 c/s to 25 Kc/s
Meter sensitivity Range 1	6·25/μA/Volt
Range 2	25/μA/Volt
Range 3	100/μA/Volt
Output terminals. Upper from amplifier output at approximately 150V. d.c. through 22 k.ohm limiting resistor. Lower to earth.	
Limits	
Probe: Minimum insulation resistance ...	200 m.ohms
Maximum capacity is limited by sensitivity required.	
Cable:	
Maximum capacity, inner to outer screen	0·025/μF
Maximum capacity, inner screen is to conductor limited by sensitivity required.	
Minimum insulation resistances:	
Conductor to inner screen	50 m.ohms.
Conductor to outer screen	200 m.ohms.
Inner to outer screen ...	50 k.ohms.
Maximum length of flexible cable with P.V.C. intermediate insulation	20 yds.
Maximum length of "Pyrotenax" cable	100 yds.
Supplies:	
Voltage	200-250V. 50 c/s.
Wattage	85W.
H.T. voltage ...	250V.
Dimensions:	
Case	15in. × 9in. × 8in. deep
Overall ...	17¾in. × 9in. × 8⅝in. deep
Weight ...	30lb.: 13·7 Kg.

It had been mentioned on page 106 how the design of the drain tank was fixed by the shaft centre lines. The lubricating oil pumps were 90 tons/hr. capacity each and two were required with a hydraulic coupling in use. Although great care was taken with the design of the 10-in. suction at the base of the tank it was never established how much vortexing took place. The oil depth was increased by 5in. during the trials but this did not overcome the aeration. The internal structure of the tank was designed to give the maximum time for entrained air to separate between the returns and suction but clearly this was not good enough. The suggestions made by Commander Good were most interesting and probably all contributed to the trouble.

Mr. Jourdain had drawn attention to Fig. 3 and the note below Siemens Ford torsionmeter. Perhaps better wording would have been "Modified for Transient Torque Measurement".

The instrument was basically the normal 14½-in. unit but two magnetic pick-ups were mounted diametrically opposite and at 90 deg. displacement from the transformers. Four additional slip-rings transmitted the signal through the brush gear to an amplifier and thence to a moving pen recorder.

Referring to Table VI the speeds quoted were "as measured". The trial had been fully analysed by The British Shipbuilding Research Association and they did not of course assume that the tidal speed was constant.

Thrust measurements were observed during the voyage from Curaçao to England and the readings were 35·60 and 38·29 tons for tests 52C and 52E respectively. These were taken at the same time as the readings in Table III.

The authors agreed with Dr. McMullen that a marine gas turbine plant of the *John Sergeant* or *Auris* type would probably not be repeated. During this discussion a possible line of development had been mentioned with particular reference to simplicity. Whilst the casings and general structure might become lighter with increased use of cooling and improved design, the authors did not feel that multiple aircraft gas turbines were the answer. At present tankers were operating for 320 days a year at sea (7,680 hours) and seven days for repair. It was envisaged that a gas turbine plant would be used for pumping also. Even if aircraft turbines could treble their life it was hardly a year's service. For "floating warehouses" it was a possible solution, but in general the fewer, simple, yet robust units with the minimum personnel, or even automatic operation appeared to be the most likely to advance. In this respect the reference to eliminating one or two operators was most topical.

Mr. Park had quite rightly mentioned the additional wages imposed on the *Auris* conversion which would not have been incurred by the *Hemisinus* project. To illustrate the amount incurred the following additional figures might be of interest.

Repair rate for existing ship compared to new ship, +5 per cent.

Repair rate for "white oil" tanker above a clean cargo ship, +15 per cent.

If the last cargo carried by *Auris* in 1955 had been a black oil then the 15 per cent increase for a tanker would have been considerably less. These percentages were based on normal wages and not basic wages so that the reduction of 50 per cent in direct labour costs with a new design mentioned by Mr. Park was a realistic figure.

In conclusion it was clear from the contributions of both Dr. McMullen and Mr. Park that they confirmed the authors' opinion that a repair or conversion was never as good as new, and that the *Auris* had filled a gap in the history of marine engineering.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at The Memorial Building on Tuesday, 14th November 1961

An Ordinary Meeting was held by the Institute on Tuesday, 14th November 1961 at 5.30 p.m., when a paper entitled "Trials and Operation of the Gas Turbine Ship *Auris*" by R. M. Duggan, M.A. (Associate Member) and A. T. O. Howell (Associate Member) was presented by the authors and discussed.

Mr. B. P. Ingamells, C.B.E. (Chairman of Council) presided at the meeting which was attended by 130 members and guests.

In the discussion which followed thirteen speakers took part.

The Chairman proposed a vote of thanks to the authors which was greeted by prolonged acclamation.

The meeting ended at 8.0 p.m.

Minutes of Proceedings of the Meeting of the Joint Panel on Nuclear Marine Propulsion Held at the Memorial Building on Tuesday, 28th November 1961

A Meeting of the Joint Panel on Nuclear Marine Propulsion was held on Tuesday, 28th November 1961 at 5.30 p.m. when a paper entitled "High Temperature Reactors for Marine Propulsion" by Dr. J. E. Richards, B.Sc., was presented by the author and discussed.

Sir Victor Shepheard, K.C.B. presided at the meeting which was attended by sixty-eight members and guests.

Ten speakers took part in the discussion which followed.

A vote of thanks to the author proposed by the Chairman received acclamation.

The meeting ended at 7.45 p.m.

Section Meetings

Karachi

The newly constituted Committee for the Karachi Section for 1962 is as follows:

Local Vice-President: Capt. S. Z. Hasnain, P.N.
Committee: H. Abbas
A. S. Benjamin
J. Mansoor
Honorary Secretary and
Honorary Treasurer: Cdr. M. A. Ansari, P.N.

North East Coast

Junior Meeting

A junior meeting of the Section was held in the new theatre of the Marine and Technical College, South Shields, on Monday, 5th March 1962 at 4.0 p.m.

The meeting was opened by Dr. J. Hargreaves, M.A., Principal of the College and Mr. W. Embleton (Member of Committee) then took the Chair. A paper entitled "The Steam Reciprocating Engine" by G. Yellowley (Member), was presented by the author to an appreciative audience of 303, comprising junior members and marine engineering students. A number of junior members from Hartlepool and Sunderland were also present.

The paper was followed by a question period.

The Chairman called upon Mr. W. G. Adams (Probationer Student) to propose a vote of thanks to the author.

The meeting closed at 5.30 p.m.

Centenary Lecture

A general meeting of the Section was held on Thursday, 15th March 1962, in the new theatre of the South Shields Marine and Technical College in honour of the Centenary of the College.

The meeting was opened by the Principal, Dr. J. Hargreaves, M.A., who gave a brief talk on the history and development of the College before handing over the meeting to Mr. S. H. Dunlop (Chairman of the Section).

Following the usual business, the Chairman introduced the speaker, Mr. C. C. Pounder, President of the Institute.

The Centenary Lecture was a paper entitled "Problems Arising in Design" specially requested for the occasion and was delivered in such a way as to hold the attention of the large marine audience of 458. This was comprised of 100 Probationer Students, 100 Students and Graduates, 150 Corporate Members, 15 senior officers of the Institute, superintendents, shipowners and College Governors.

The College open period from 4-6 p.m. which was popular and well patronized was restricted to members and a few personal guests.

Northern Ireland Panel

A special meeting was held on Tuesday, 20th March 1962, at 6.45 p.m., in the College of Technology, Belfast, for the presentation by the President of his address. Mr. D. H. Alexander, O.B.E., F.C.G.I., M.Sc., Wh.Sch. (Local Vice-President), was in the Chair and about eighty members and visitors were present.

In introducing the President, Mr. Alexander spoke of the growth of the Institute since its formation in 1888 and drew attention to the fact that the Institute was at present expanding at a greater rate than at any time in its history. He reminded the audience that Mr. Pounder was the fourth President to come from Belfast, Lord Kelvin, Viscount Pirrie and Sir Frederick Rebbeck having preceded him as President in 1892, 1906 and 1931 respectively.

The President then gave his address "The Marine Engineer and the Common Life".

At the conclusion a vote of thanks was proposed by Mr. B. K. Batten, M.Sc. (Associate Member), Engineer Surveyor with Lloyd's Register of Shipping, which was carried with enthusiasm.

South Wales

Cardiff

A meeting of the Section was held in co-operation with the Institute of Welding, on Monday, 5th February 1962, at the South Wales Institute of Engineers, Park Place, Cardiff.

Mr. David Skae (Vice-President), Chairman of the Section, presided over the meeting which was attended by 28 members.

A lecture entitled "Welding in Small Ship Construction" was delivered by Mr. R. Du Cane and was copiously illustrated with slides and excellent colour film. That the lecture was of great interest to members of both Institutes, was evidenced by the discussions that followed.

A vote of thanks to the lecturer was proposed by Mr. Lewis, Institute of Welding.

Swansea

A meeting of the Section in co-operation with the Institute of Welding was held on Tuesday, 6th February 1962, in the showrooms of the Gas Company, Swansea. Mr. David Skae was again in the Chair and fifty-five members attended.

Mr. R. Du Cane repeated his lecture "Welding in Small Ship Construction" with again the same success.

A vote of thanks was proposed by the Chairman.

General Meeting

A meeting of the Section was held on Monday, 12th March 1962, at the South Wales Institute of Engineers, Park Place, Cardiff, when a lecture entitled "Spheroidal Graphite (Nodular) Cast Iron in Marine Engineering" was given by Mr. J. C. Robinson, of Messrs. Clarke Chapman Ltd.

The lecture, given on behalf of the British S. G. Producers Association, was well received by the thirty members present. Their appreciation of a most interesting lecture was expressed in the vote of thanks to Mr. Robinson, proposed by Mr. P. A. Ridyard (Associate Member).

The meeting concluded with a vote of thanks to the Chairman Mr. David Skae (Vice-President), Chairman of the Section which was proposed by Mr. W. S. Holness (Member).

Sydney

Annual Report

The Committee has pleasure in presenting the Thirteenth Annual Report of the proceedings of the Sydney Section, during the year 1961.

The Section has gained eight members during the year and the membership now stands at 174. In addition, enquiries regarding membership have been received from seven prospective members.

Members of the Section were honoured by the election on 25th April 1961, of their Chairman, Captain G. I. D. Hutcheson, C.B.E., B.E., R.A.N., as Vice-President for Australia.

Four general meetings were held during the year; also the Thirteenth Annual Meeting for students and apprentices, as follows:

22nd March	Annual General Meeting: "The R.A.N.'s First Fifty Years—Developments in Warships Design, Machinery and Equipment" by Rear Admiral K. McK. Urquhart, C.B.E.
24th May	"The West Sydney Cove Passenger Terminal" by E. Ian Griffin.
16th June	Students' Meeting: "The Maintenance of Doxford Diesel Engine" by W. Fyffe (Member).
29th June	"Some Interesting Features of *Canberra's* Machinery" by T. W. Bunyan, B.Sc. (Member).
27th September	"Screw Conveyor Equipment Installed in *North Esk* for the Handling of Bulk Grain" and "Special Cranes for Container Ships *South Esk* and *William Holyman*" by M. Langtree.

These talks were followed by an address by Mr. H. P. Weymouth, O.B.E. and other engineers on "Some Interesting Aspects of Ships Building for the Australian Trade".

The Annual Dinner was held at the Wentworth Hotel on Thursday, 2nd November 1961. This was a most successful function and was attended by sixty members and fifty-nine guests. It continues to be one of the most important annual gatherings of those associated with shipping in Sydney.

The constitution of the Committee during 1961 has been as follows:

Vice-President and Chairman: Capt. G. I. D. Hutcheson, C.B.E., B.E., R.A.N.
Committee: W. G. C. Butcher
W. F. Ellis
W. Fyffe
E. G. Hughes
W. T. Mathieson
Capt. R. G. Parker, O.B.E., R.A.N.
Honorary Secretary: N. A. Grieves
Honorary Treasurer: Lt. Cdr. J. W. Lamb, R.N.

The Council continues to take a very keen interest in the activities of the Section, and has always been most encouraging towards it, and we are most grateful for this interest.

The Committee is also grateful to all local Members for their continued support, as it is only through this that the Section can continue to function successfully.

Annual General Meeting

The Annual General Meeting of the Sydney Section was held on Wednesday, 28th March 1962.

The following have been elected members of the Committee of the Section for 1962:

Vice-President and Chairman: Capt. G. I. D. Hutcheson, C.B.E., B.E., R.A.N.
Committee: W. G. C. Butcher
E. S. Clarke
W. Fyffe
D. A. Gillies
W. T. Mathieson
Capt. R. G. Parker, O.B.E., R.A.N.
Honorary Secretary: N. A. Grieves
Honorary Treasurer: Lt. Cdr. J. W. Lamb, R.N.

West of England

A general meeting of the Section was held on Monday, 12th March 1962 at the University of Bristol, at 7.30 p.m.

Captain R. G. Raper, R.N. (Chairman of the Section) was in the Chair and the audience, which included Mr. D. W. Gelling (Local Vice-President), numbered twenty-nine.

A paper entitled "A Critical Survey of Marine Water Treatment Methods" by I. Raleigh, was presented by the author. The paper gave critical consideration to some of the more important aspects of marine boiler water treatment methods in coping with the problems of limiting, economically, the corrosion rate of the basic material, namely, steel, by water and the impurities, both solid and gaseous. The paper incorporated an account of an oxygen recorder being developed by the author together with a description of his Marine Ionostat. Members were particularly interested in the author's description and working details of the apparatus developed by his Company, for de-salting sea water to provide drinking water in ships' lifeboats.

Eight speakers took part in the discussion which followed, and all questions were ably dealt with by the author.

A vote of thanks to the author proposed by the Chairman received warm appreciation.

The meeting ended at 9.15 p.m.

West Midlands

A general meeting of the Section was held on Thursday, 22nd March 1962 at the Engineering Centre, Birmingham, at 7 p.m. when a lecture entitled "The Second Generation of Hydrostatic Power Transmission" was presented by Mr. D. Firth.

Mr. H. E. Upton, O.B.E. (Local Vice-President and Chairman of the Section) presided over the meeting which was attended by forty-five members and guests.

With the aid of slides, Mr. Firth fully described the applications of both hydraulic pumps and motors for marine and land systems. These systems showed considerable saving in volume and weight.

A very interesting discussion followed, during which a number of questions were put to the speaker, all of which he dealt with ably.

On behalf of the members and guests present, the Chairman thanked Mr. Firth for a most interesting lecture.

The meeting closed at approximately 9 p.m.

Election of Members

Elected on 12th March 1962

MEMBERS
John Gilbert Bancroft
Francis Xavier Cleary
Russell Hugh Crabbe
Harold C. Dear, Lieut. Cdr., C.D., R.C.N.
Walter Francis Dixon
Norman Taylor Greig
Bruno Keidans
George William Ovenden, Eng. Lieut. Cdr., R.N.
John Hugh Sherwood
Robert Graham Thomson
Ernesto Verdera Tomas
Harold Whincup
Claude Leslie Gordon Worn, B.Sc.(Eng.) (London)

COMPANION
Jack Howard Kirby

ASSOCIATE MEMBERS
George Aitken
Emmanuel Francis Aquilina
David Fraser Bell
John McConnell Bell
James Appleby Burt
John Craven Carden
Roy John Carnell
Douglas Crawford
John Allen Frederick Crook
Richard Errington
Barry Edward Fincham
Kamlesh Kumar Garga
Thomas Harris, Eng. Lieut., R.N.
Cecil Henry Hughes, B.Sc.(Eng.) (London)
Edwin Innes
Cyril Herbert Lamprecht
Raymond Lees
Francis James Edward Lockley
Bjorn Henrik Markussen, B.Sc.(Durham)
William Richardson
Alan George Smith
Chit Swe, Lieut., (E), B.N.
Edward James Tweed, Eng. Lieut., R.N.
Peter Thornton Wales
John Ward
Paul Reginald Whitehouse, B.Sc.(Birmingham)
Cyril George Wood
David Robert Woods

ASSOCIATES
Francis Peter Begbie
Henry Paddison Granlund
Frank Johnson
John MacDonald MacGregor
Stanley Nichols
Christopher Selwyn Priestly Rawson
Leonard Bremner Smith
Geoffrey Thomas Stanton
Edward Taylor, B.E.M.

GRADUATES
Bruce Rodney Allen
Allan Milburn Bowman, B.Sc.(Durham)
Francis Vincent Brett
Tom Rawlinson Campbell
Clive Carwardine-Palmer
Colin Taylor

STUDENTS
John Charles Barrett
Francis Joseph Cauley
Sarukalige Thusitha Ranjan de Silva
Bryan Peter Hermer
Graham John Hunt
Raymond Stanley Jeffree
Edward Ronald Morgan
Frederick David Petit
Victor Charles Riley

PROBATIONER STUDENTS
Stephen Barns
Harvey William Gant

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER
Edwin James Thomson Caie
Kenneth Alfred Moody Goodyear
Kenneth Gregan
Cyril William George Hawken
Derek Norman Moore
David Arthur Taylor
John West, B.Sc.

TRANSFERRED FROM ASSOCIATE TO MEMBER
Donald Birchon, B.Sc.(London)
Newburgh Henry Card, Lieut., R.N.
Dorian Davies
Peter Henry William Evans
Philip Ernest Wood
Frederick Bruce Youngman

TRANSFERRED FROM ASSOCIATE TO ASSOCIATE MEMBER
Kunnath Peter George, Sen. Cd. Eng., I.N.
John Hedley
Bruce Pragnell, Eng. Lieut., R.N.

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER
Kenneth Nichol Bexon
Bernard Thomas Boyce
John Leonard Carey
Michael Gordon Derham
Martin Herman Greeff
Mohamed Zahir Navaz, B.Sc.
Brian Denton Rolls
Bhopendra Singh, Lieut., I.N.

TRANSFERRED FROM GRADUATE TO ASSOCIATE
Derek Everett Newling Turner

TRANSFERRED FROM STUDENT TO GRADUATE
Maxwell Keith Bruce
Walter George Vernon Lugg
Michael Frank Roberts
Charles Ralph Willoughby

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE
Terence Charles Whitney Booth
Malcolm Richard Littler

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT
David Watling Freeman
Stephen Francis Green
Michael Frederick Hatt
Anthony Francis Imaz
John Robb Ritchie Linnen
David Munnoch Mitchell
Alec Mottram

Election of Members

Elected on 9th April 1962

MEMBERS
Abdul Rachman Atmosudirdjo
Leonard Owen Brough
William Louis Clifton
Frederick Victor Craig

Maurice George Grapes
Dr. Herbert Cecil Perera Gunewardene, L.R.C.P., M.R C.S.
Arthur Humphrey
Gordon Harrison King
Thomas Lawson
Charles Cecil Luter
James McGregor
Harry Segar McKay
Hugh McKean
David Mills Phillips
Edgar Phillips, Eng. Lieut., R.N.
Sabino Roccotelli, Magg.G.N.Ing.
William Eric Stanton
Richard Frank Wooden
Jack Worrall

ASSOCIATE MEMBERS
Surendra Agarwal
Ramchandra Motiram Bhivandker
George Louis Chalcraft, Eng. Lieut., R.N.Z.N.
Leslie Cowle
William Crowe
Asmus W. Feck
George Grant Gourlay
Alan Douglas Hern
John Alfred Horswell
James Jenkinson
Vasant Vithal Kantebet
Kallidaikurichi Ramakrishnan Lakshmanan
Karayi Lakshmanan
Maung Maung Lay
Nateson I.C. Lobo
Iain Alan St. Clair MacDougall
Eoghann Niall MacKinnon
Daniel Francis Xaxier Marks, Lieut, R.N.
Servulus J.B. Pereira
John Walker
Alec Webster
David Wilson

ASSOCIATES
Jack Frederick George Arman
Reginald Dennis Fielder, Capt., U.S.N.
Thomas Ian Kirk
James Martin
Robert J. Power
Kamalesh Sarkar
Roy Thomas Sheaves
Szeto Ying Hon
Tong Shiu-Wing

GRADUATES
Ian Brown Baxter
Jacobus Boone
Daniel Brophy
Kenneth Knowlton Gandy
James Munro Hepburn
Colin Hudson
Koh Jin Long
Arunashis Maitra
John George Short
Richard Irving Stevenette
Muhammad Sulaiman Syed
Perumal Chettiyar Thangaraj

STUDENTS
Rattan Singh Amrik
William 'Kodi S. Aniche
Peter Charles Armour
David Ronald Blackwood
Steven Roger Crocker
Dennis Grattan Dodd
Vinode Sobhraj Gajria
John Edward King
Arthur John William Largue
John Blair MacIntyre
Dalip Singh Mehta
Patrick Anene Onwubuya
Kenneth George Powell
Roger Brian Smedley
James Lawrence Barclay Smith
Christopher John Wragg

PROBATIONER STUDENTS
Cedric Gordon Eldred
John Lingham
David William Varcoe-Baylis

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER
Trevor Wallace D'Arcy-Evans
Arthur Reginald Eyles
Ravalnath Ananth Kamath
David Ferguson MacDonald
Frederick Ian Mason
Alan Swarbrick

TRANSFERRED FROM ASSOCIATE TO MEMBER
Robert Broadfoot Stitt MacAdam

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER
Norman John Baynham
Donald George Hickey
Charles Mackenzie
William Woodward Wright

TRANSFERRED FROM GRADUATE TO ASSOCIATE
Donald Sinclair Weir Martin

OBITUARY

DONALD JOHN ALLCOCK (Associate 13955) was born on 1st December 1920. Between 1937-41 he was an engineering student with the South Metropolitan Gas Company, and in 1941 joined the Royal Navy and was rated as an engine room artificer, fifth class. For the following years of the war, he was trained in Naval engineering practice aboard H.M. Ships *Vindictive, Edinburgh Castle, Beaufort* and *Warspite* and achieved the rating of third class E.R.A. in the last-named vessel. In April 1945 he was commissioned with the rank of Sub-Lieutenant (E) and was appointed to H.M.S. *Devron;* in September of the following year he was released from his service, and, after an interval, he enrolled for a course at Wandsworth Technical Institute, where he studied for the next three years.

In October 1946 Mr. Allcock was appointed first assistant to the mechanical engineer at the Rotherhithe works of the South Eastern Gas Board, and in September 1961 was transferred to a similar position at Dover Docks.

Mr. Allcock who became an Associate of the Institute in 1946, died on 26th February 1962.

VICTOR WALTER BACK (Member 9840) was born on 21st August 1893. He served an apprenticeship with the engineering firm of Pertwee and Back, from 1907 until 1912, for the last year as an improver. For six months during 1912 he was a draughtsman in the employ of the Entre Rios Railway Co., Argentina, and the same year returned to England and for the next twelve months worked as a fitter to the assistant tester of internal combustion engines, at Mann, Egerton and Co. Norwich. In 1913 Mr. Back was appointed works manager of Three Towns Engineering Works, Plymouth. During World War I he joined the Royal Naval Volunteer Reserve reaching the rank of Lieutenant.

After the war was over he returned to Messrs. Pertwee and Back as works manager and later director, a post he held until becoming managing director in 1932.

Mr. Back, who was elected J.P. for the County of Great Yarmouth, was chairman of Yarmouth Magistrates at the time of his death last March and a former member of the Town Council. He was elected an Associate Member in 1944 and became a full member of the Institute in 1945.

CECIL BERTRAM CHAMBERS (Member 10101) died on 25th December 1961 aged 65 years. Apprenticed to Earles Shipbuilding and Engineering Co. Ltd., Hull between 1911-17, he obtained a First Class Ministry of War Transport Steam Certificate in 1927 on leaving Elder Dempster and Co. Ltd., with whom he had served in various seagoing appointments since 1919. For the next eight years, he held the grades of third and second engineer and chief refrigerating engineer with Shaw Savill and Albion Co. Ltd., and then transferred to the Arctus Shipping Company Ltd.

In April 1936 Mr. Chambers joined Clan Line Steamers Ltd. and acquired further experience in refrigeration maintenance, until 1940 when he became chief engineer on the Australian line of Burns, Philp and Co. Ltd., Sydney, a post that he held till 1944, when he continued to serve as a chief engineer in a variety of ships under a variety of flags.

The last four years of his seagoing career were spent with the Euxine Shipping Co. Ltd., London. Mr. Chambers was elected to Membership of this Institute on 7th November 1944, and had also been a member of the Navigators and Engineer Officers' Union.

REUBEN CLARKE, O.B.E. (Member 12770) was born on 6th April 1897. In 1912 he joined Laurence Scott and Electromotors Ltd. of Norwich, as an apprentice, and thus began an association with the company which was to last 48 years.

After receiving his technical education at Norwich College, he joined H.M. Forces for the duration of the First World War and returned to Laurence Scott as an electrical draughtsman in 1918. In 1923 he continued his drawing office experience at the Willesden works of British Thomson-Houston Ltd. He then returned to Laurence Scott and Electromotors as a designer, and after gaining considerable experience in the design and manufacture of marine and industrial control gear he was appointed chief engineer to the switchgear department in 1939. During the Second World War, in addition to a great deal of development work on special control gear, he was responsible for the building and equipping of a branch factory where 90 per cent of the employees were unskilled women, an experiment which proved very successful.

Mr. Clarke joined the board as marine sales director in July 1945. He was awarded the O.B.E. in the New Year Honours List of 1951, and became a Member of the Institute in 1950. He resigned his directorship of Laurence Scott and Electromotors at the end of 1960 owing to ill health, and his death occurred on 23rd February 1962, after a prolonged illness.

THOMAS STEVENSON CRAGGS (Member 9027) died on 1st February 1962 aged 71 years. Apprenticed at the Neptune Engine Works of Swan, Hunter and Wigham Richardson Ltd., Wallsend-on-Tyne, between 1907-12, he continued to serve with this company and was closely associated with the building of this company's first Diesel engine. On completion of this prototype, which was fitted to m.v. *Arabis*, he proceeded to sea on behalf of the builders. Apparently he remained in this vessel until 1915 when she was sunk by enemy action, and Mr. Craggs was the only surviving officer. After this episode, he rejoined Swan Hunter, where he was engaged on further Diesel research until 1917, when he obtained a seagoing appointment with the Anglo-Saxon Petroleum Co. Ltd.

He remained with this company until 1934, advancing through the various grades until reaching chief engineer's rank. As he went to sea on a motor vessel comparatively early, Mr. Craggs was one of the first engineers to obtain a First Class Board of Trade Motor Certificate, after its introduction by the Board of Trade. After service with various other companies, he served with Scottish Tankers Ltd. until late 1940. His sea-career was finally terminated by a leg injury, which, sustained during an engine room accident, proved incurable and forced him to take up an appointment ashore. He accepted a post as manpower board inspector with the Ministry of Labour at Sunderland and continued in this capacity until the end of World War II.

When his service with the Ministry of Labour had finished, Mr. Craggs found that his health was such as to prevent him from continuing a business career, and he therefore retired at a comparatively early age. He was elected a Member of the Institute on 13th December 1939, and his son, Thomas, is an Associate of the Institute.

IDRIS DAVIES (Member 4007) was born on 29th March 1889 and died on 29th March 1962 at the age of 73. After serving an apprenticeship he joined the Merchant Navy as an engineer and served with various companies; for some years with the Blue Star Line Ltd. In 1920 he came ashore where he was employed as assistant manager with a number of South Wales shiprepairer firms and was, at one time, the engineer manager at the old Eastern dry docks, Newport, now owned by the Mountstuart Dry Docks Ltd.

Eight years later Mr. Davies left Wales to take up an appointment in London, with Watts, Watts and Co. Ltd., as assistant engineer superintendent; in January 1931, he succeeded the late Mr Vessey Lang as senior engineer superintendent of the company. In 1948 he became consultant engineer and during the latter part of his service to the company Mr. Davies was primarily concerned with the building of the *Wanstead* class of vessels. He retired from Messrs. Watts, Watts in October 1951 and shortly afterwards took up an appointment as marine and engineer superintendent with the Purvis Shipping Co. Ltd. Seven years later he became marine consultant to this company with which he remained, in this capacity, until his retirement a few weeks before his death.

He was elected a member of the Institute in 1920 and was also a Fellow of the Society of Consulting Marine Engineers and Ship Surveyors.

SYDNEY VICTOR FLECK (Member 13391). Born on 20th June 1897, he began his long association with Swan, Hunter and Wigham Richardson Ltd. as an engineer apprentice between 1913-17, and in the latter year served in the Royal Navy.

After his demobilization in 1919, he joined the Asiatic Steam Navigation Co. Ltd. and served as fifth, fourth and third engineer before coming ashore to study for his First Class Steam Certificate at South Shields Marine School in 1923. This he obtained in July of that year, and went on to study for his Extra First Class Engineer's Certificate, but unfortunately these further studies had to be discontinued. For the following months he was assigned to erecting and testing Diesel engines for Swan, Hunter and Wigham Richardson Ltd., and from 1924-40 he undertook service in Diesel-engined vessels as second, then chief engineer from 1937. He then acted as guarantee chief aboard a Doxford-engined ship from June 1939. He added a Motor Endorsement to his First Class Steam Certificate in 1926.

From May 1940 until August 1946, Mr. Fleck was engaged by Swan, Hunter in charge of the erection of corvette and frigate engines, and also of turbines and Diesel engines. He then left the company and returned to sea as chief engineer of a Burn-engined ship, and came ashore three years later. In April 1949 he took up an appointment as superintendent and surveyor with F. J. Trewent and Proctor Ltd. and remained with them until April 1957 when he went to work for Messrs. Coulouthros Ltd. based in London. In 1959 he was transferred to Goulandris Bros., also of London, and since then his work has been mostly abroad. He spent 13 months in Japan and his last assignment was in France, where he worked from August 1961 until the time of his death. He died suddenly in hospital at St. Nazaire on 24th January 1962, and his body was brought back to Newcastle upon Tyne, his home, where it was cremated on 10th February 1962.

He joined the Institute in 1951, and was also a member of the Marine Engineers Association. Mr. Fleck leaves a widow and five daughters.

JAMES STANLEY MARSHALL (Member 2760) died in Edinburgh on 9th March 1962. Beginning his career with an apprenticeship at Messrs. Amos and Smith, boiler makers of Hull, from 1903-1908, he thereafter acquired drawing office experience with the same firm, and then began his seagoing career with the Wilson Line, as a fourth engineer. He later sailed as second engineer with the Ellerman Wilson Line. He obtained an Extra First Class Board of Trade Steam Certificate in January 1914, and following employment in the drawing office of The Parsons Marine Steam Turbine Co. Ltd., he received a temporary commission as an Engineer Lieutenant in the Royal Navy, and was later promoted Engineer Lieutenant Commander in the Special Reserve, Royal Navy.

Mr. Marshall then took up industrial work and subsequently held staff appointments as maintenance engineer, chief engineer and engineer and works manager in the Midlands and the South of England. In 1926 he was appointed chief engineer of the North British Rubber Co. Ltd. of Edinburgh and Dumfries. He retired in 1954, and afterwards acted as a consultant to the company until the need of a major operation obliged him to conclude his business activities in 1956.

Mr. Marshall, who was born in 1887, enjoyed a long association with the Institute, having been elected a Member in 1913.

LESLIE BENJAMIN PERRY (Member 12182) was born on 4th November 1900, and served his apprenticeship in the marine workshops of the Great Eastern Railway at Harwich. After attending Colchester Technical College to the age of twenty, he joined the Furness Withy Line as fourth engineer, and in 1922 served as junior then senior (1925) engineer aboard London and North Eastern Railway cross-Channel vessels, based on Parkeston Quay at Harwich. He obtained his First Class Board of Trade Steam Certificate in 1928 and remained at sea in railway vessels till 1940, when he was commissioned Lieutenant Commander (E) in the Royal Naval Reserve, and was attached to the Admiralty Fleet salvage department, after service in the Mediterranean in H.M.S. *Malines*.

The war over, Mr. Perry returned for a further tour of duty on cross-Channel boats, until his appointment in 1947 as assistant to the Marine Superintendent Engineer at Parkeston Quay. In 1951 he was appointed assistant marine superintendent engineer, a post he held until his retirement, through ill-health, in 1961.

Mr. Perry's membership of the Institute dated from December 1948, upon his election as full Member.

JAMES ROWLAND TAIT (Member 3671) who was born on 25th August 1886, died last January. He began his marine engineering career as an apprentice at the Fairfield Shipbuilding and Engineering Co. Ltd., Glasgow, and afterwards went to sea for nine years, obtaining meanwhile, a First Class Engineer's Steam Certificate.

In 1919 he became assistant to J. E. Wimshurst, consulting engineer, and spent a year in consultant's work at Antwerp, after the First World War. Thereafter, for the next twenty years or so he was connected with Jacobs, Barringer and Garratt Ltd., and was appointed superintendent engineer of this consulting firm in 1940. For many years his firm had been associated with C. T. Bowring and Co. Ltd., shipowners of London and Liverpool, as their engineering consultants, and, in July 1949, Mr. Tait became a superintendent engineer with this company.

On his retirement, in December 1955, he directed his energies to horticultural interests, in which field he became well known, particularly for his success with peaches and apples. During the later years of his life, however, he was not able to be as active in this sphere as he would have wished.

Mr. Tait joined the Institute as a full Member on 3rd June 1919.