THE NON-MAGNETIC DELTIC ENGINE RELIABILITY IMPROVEMENT PROGRAMME

BY

M. F. Clover, M.A., C.Eng., M.I.Mech.E., *Chief Engineer*

AND

J. N. Ramsden, C.Eng., M.R.Ae.S., *Director of Design*
(*Paxman Diesels, Ltd.*)

ABSTRACT

The non-magnetic Deltic engine used in the Hunt Class of MCMV has posed unusual problems in respect of engine design. The diagnosis and solution of these problems is described and statistical analysis used to demonstrate the improvement in reliability to date and to predict the reliability levels in the future as these improvements flow into service via the overhaul programme.

Introduction

This article is an invited companion article to 'Bent Bathtubs' in this issue. 'Bent Bathtubs' describes the application of 'hazard functions' to a service situation. The service situation chosen was the propulsion engine of the Hunt Class minesweepers, the Paxman Deltic 59K engine. These engines are maintained by unit replacement, so that they are in that respect a suitable subject for the type of analysis described. However, as also is evident from the article, the engines, with their integral clutches and gearboxes, are somewhat more complicated than electrical light bulbs, with more modes of failure. Moreover during the period during which engines have been in service, it has been possible to do investigations and introduce modifications into later engines and into rebuilt engines, thereby progressively improving engine reliability.

The Deltic Engine in the Hunt Class

Better known as 'Napier' engines, the Deltic engines originally entered service in 1950. Following company mergers the Deltic engine design team moved to Paxman at Colchester in 1970, so that Paxman has had a direct interest in the engine for half of its commercial life.

The basic architecture of the Deltic is as shown in Fig. 1. Opposed pistons in a delta formation of cylinders apply power to three crankshafts via 'blade and fork' connecting rod assemblies. A gearwheel on the end of each crankshaft feeds power into the gear train within the phasing gearcase.

There are, of course, no cylinder heads. The engine operates on the two-stroke principle, supercharged air being blown into each cylinder through ports at one end and exhaust gases escaping through another set of ports at the opposite end of the cylinder. Not surprisingly, the engine has a high inherent balance and runs smoothly. Following in the aircraft tradition of the Napier Company, virtually all castings in the Deltic are of aluminium alloy. Bearing in mind the engine has its own reduction gearbox, the overall installed weight in high-speed patrol craft gave unrivalled power-to-weight performance. The basic high non-magnetic content at 50% made the engine attractive for mine counter-measures use and the engine found its first application in the Ton Class.

For the HUNT Class the decision was taken to push the non-magnetic principle as far as possible. Not suprisingly, the task of using the necessarily unusual materials has had its problems.

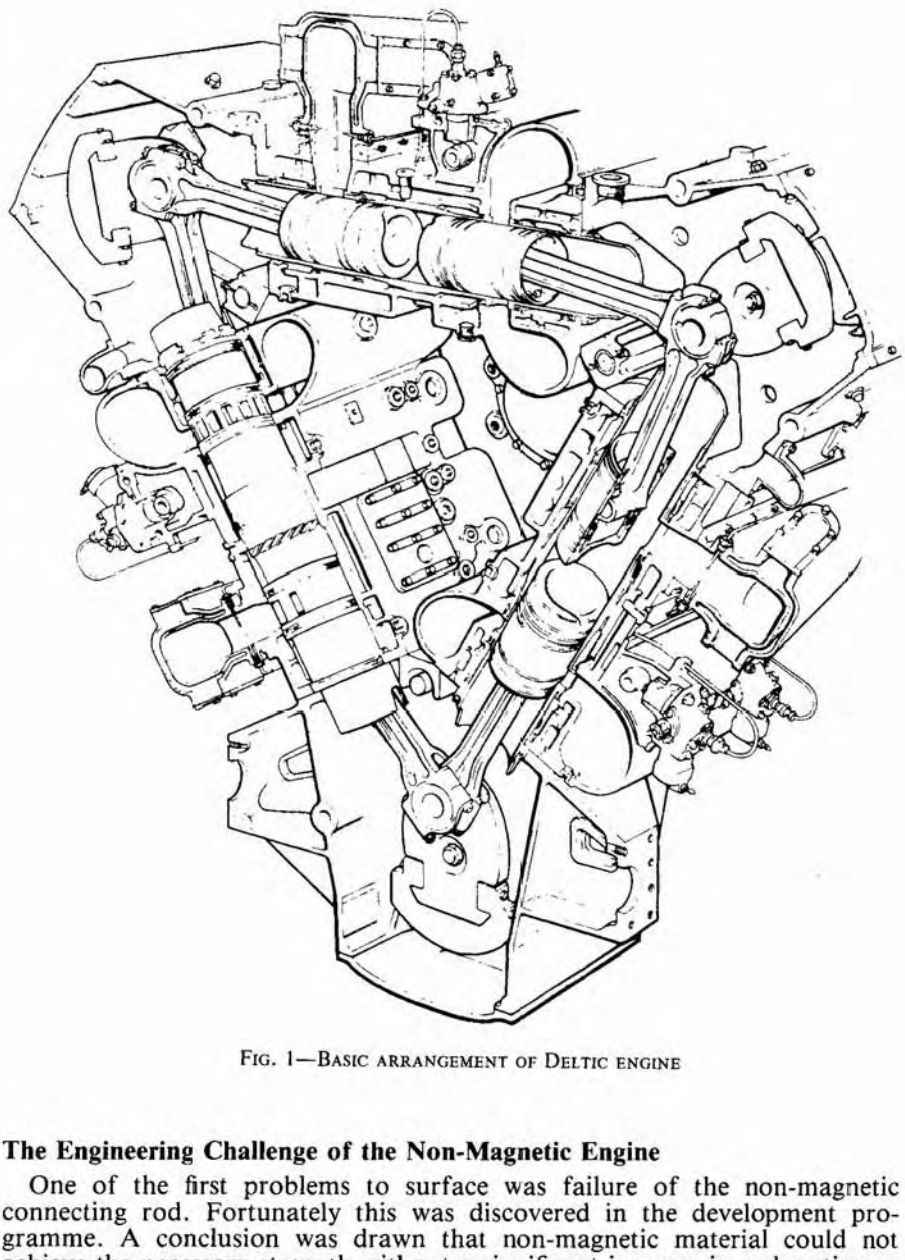

FIG. 1—BASIC ARRANGEMENT OF DELTIC ENGINE

The Engineering Challenge of the Non-Magnetic Engine

One of the first problems to surface was failure of the non-magnetic connecting rod. Fortunately this was discovered in the development programme. A conclusion was drawn that non-magnetic material could not achieve the necessary strength without a significant increase in rod section; a change was, therefore, made back to the normal material.

On entering service, new problems became apparent. The Pareto chart in FIG. 2 classifies the number and types of problems in the period to 1988.

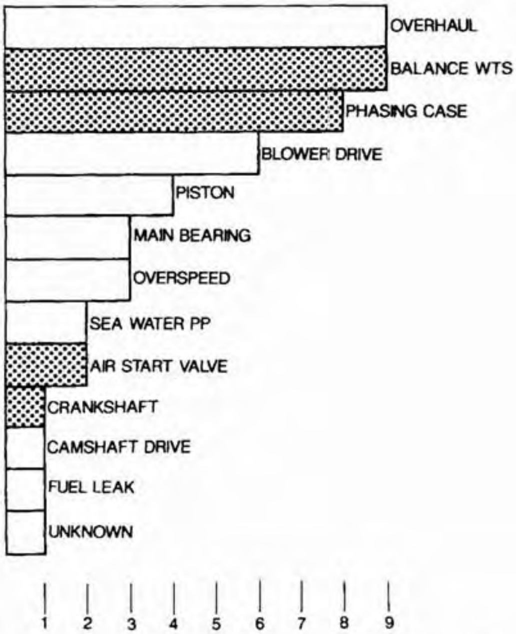

FIG. 2—PARETO PLOT OF DELTIC 59K ENGINES TAKEN OUT OF SERVICE, SHOWING TYPES AND NUMBERS OF PROBLEMS

Pistons

The first problem experienced was the piston problem. The first two piston seizures were attributed to high liner surface temperatures due, in turn, to the low thermal conductivity of the non-magnetic material of the liner. The material was changed back to normal specification. Although two further failures occurred with the normal liner, this was attributed to faulty bedding-in of the piston rings. The running-in procedure has been changed and the situation now seems well under control.

Crankshaft

Failure of the non-magnetic crankshaft also occurred early in the engine's history (1981). During development it had been found necessary to increase boost levels to increase engine manoeuverability (59K version), thereby raising cylinder pressures. This led to crankshaft failure. A slight retardation of fuel injection timing combined with an increase in crankshaft fillet radii cleared the problem.

Air Start Valves

Not all problems were caused by non-magnetic materials. Air start valve failures were found to be due to a circumstance where the starting air could open the valve with the engine running. This was overcome by changing the air supply system off the engine.

Gear Tooth Scuffing

A further problem was scuffing of gear teeth in the phasing case. These failures were due to insufficient tip relief on some gears. In loaded gear trains it is usual to specify tip relief of gears back from the true involute form to compensate for tooth distortion under load, and to give a small lead-in to promote the formation of an oil film. In this instance, although following established practice, it was possible to obtain gear combinations, within the drawing tolerance limits, which were not satisfactory. Insufficient tip relief leads to surface scuffing, surface wear and increasing vibration. Ultimately a tooth breaks or a bearing fails. An increase in nominal tip relief has reduced vibration levels and monitoring of phasing case vibration and introscope inspection indicates that this change has been successful.

Balance Weights

In 1985/86 the first failures of crankshaft balance weights occurred. The method of attaching the weight to the crankshaft web and the general fracture line is shown in FIG. 3. During inspection of the failed engines some bolts were found loose. A rapid check through all engines in service showed further bolts had slackened off and some were discovered broken. All were re-tightened and that seemed to stabilize the situation. An engineering investigation was put in hand.

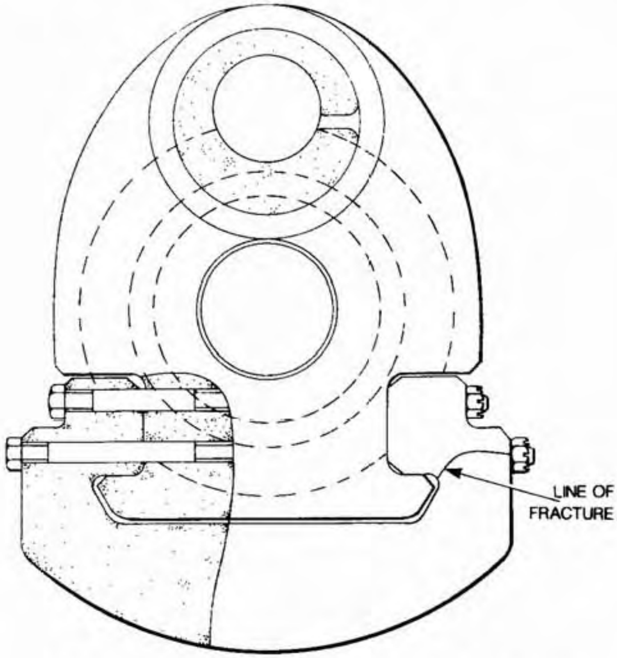

FIG. 3—DELTIC CRANKSHAFT COUNTERBALANCE WEIGHT

Metallurgical examination of the failed weights showed larger than expected grain sizes in the Hidurel (copper-based) alloy and a crystalline fracture face which did not seem associated with fatigue. Nevertheless fatigue tests on large grain and small grain specimens were carried out. These showed a reduction in strength of the material with increasing grain size. Furthermore, they showed that the material could suffer fatigue and yet display a crystalline fracture surface. FIG. 4 shows the dramatic effect of grain size on fatigue life.

Careful inspection of crankshafts showed that in some cases the holes drilled through the webs for the balance weight holes were slightly cranked; the holes had been drilled from each side and they met in the middle. This was a problem which had not been experienced with the magnetic crankshaft.

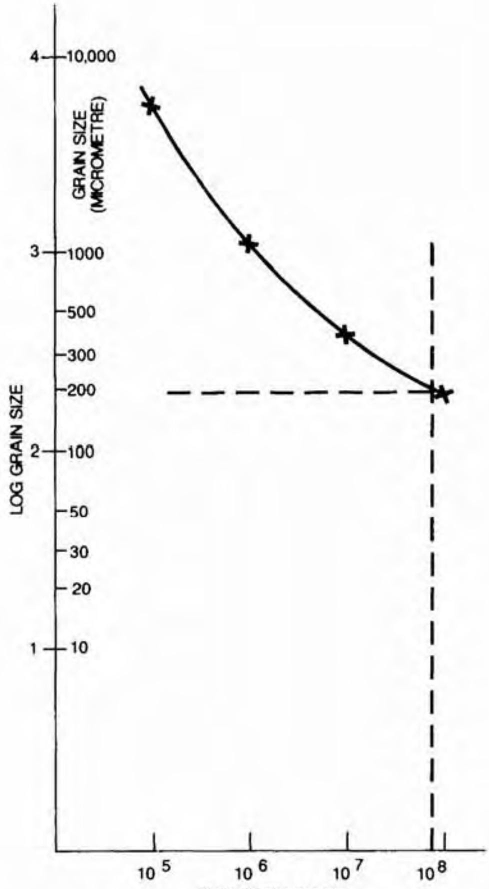

FIG. 4—EFFECT OF GRAIN SIZE ON FRACTURE LIFE OF HIDUREL 5 (MEAN LOAD 20 MN/m^2; ALTERNATING STRESS ± 75 MN/m^2)

The non-magnetic material was extremely tough and it was felt this factor had led to difficulties in controlling drill direction. The standard bolt was already a tight fit, the cranked holes made it even tighter. In this situation the nut pull had not been sufficient to draw the bolt head hard down onto the balance weight. This helped to explain the slack, and in some cases broken, bolts found in service.

Whilst it was becoming clear how weights could have reduced strength and bolts could become loose, there was no clear picture of the nature of the cyclic forces which had caused the fractures. A crankshaft was fitted with a balance weight having strain-gauges at the fracture site and run in an engine. Various degrees of balance weight fit were also tested to see if it influenced stress levels. Test results showed the presence of a vibratory stress linked to firing load. Miniature accelerometers on the weight showed that the weight was undergoing a flapping motion triggered by crankpin and web bending. It was shown that the dynamic stress levels could also be influenced by the degree of balance weight fit. The pieces in the puzzle started to make sense.

The target grain size for reliable operation was established and forging and heat treatment specifications changed to achieve better than target figures. All crankshafts coming in for overhaul are now re-machined to achieve well-fitting bolts and weights. New, fine-grained, balance weights are fitted using bolts which have been driven-in hard before tightening the nut. We expect no futher problems in this area.

Operating Procedure

The main bearing failures were traced to oil contamination and, in one instance, to the oil supply being cut off. The overspeed failures were caused by operator error. Changes in operating procedures should prevent recurrence.

Blower Drive Shaft

A current problem which reflects the tribological difficulties associated with the use of non-magnetic materials is the fretting and, sometimes, consequential failure of the Inconel blower drive shafts. Two of these take power off the gear train in the phasing case and transmit it the length of the engine to the blower gear train. The fretting itself occurs in the splined end connections.

Service trials of various solutions are currently in hand. These cover bronze-plating, electroless nickel plating and friction-welded steel ends. Of these probably the latter will offer the best chance of success.

Quantification of Current and Future Reliability

'Bent Bathtub' has already dealt in some detail with the reliability statistics of the HUNT Class 59K Deltic engine. Nevertheless, it may be useful to show here the overall progress made on the engine. FIG. 5 shows a Duane plot of the 59K in the period from 1981. The Duane method has been described in the *Journal*[1]. The plot shows a steady improvement in Mean Time Between Failures (MTBF), other than for 1988. Clearly the major improvements in engine design now going into service will not have an effect for a while.

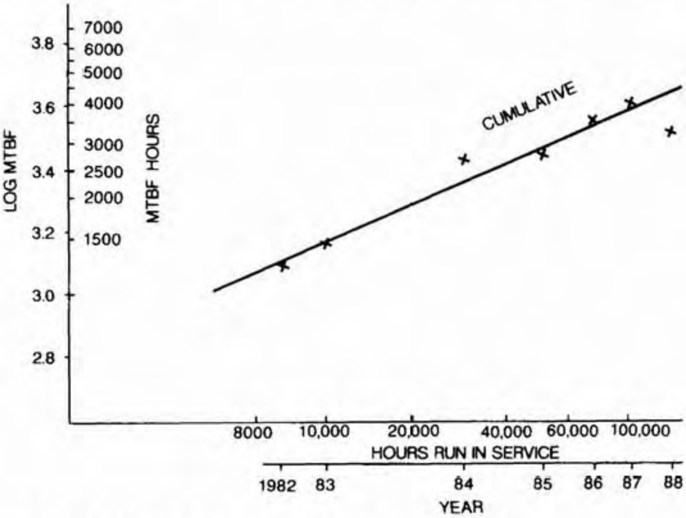

FIG. 5—DELTIC 59K ENGINES: DUANE PLOT OF FAILURES IN SERVICE

FIG. 6 sets out to demonstrate the eventual beneficial effects of various modifications on MTBF. In this figure MTBF of the average engine has been plotted against engine hours from new or rebuild, based on service experience to date.

Plot A in FIG. 6 shows the position taking all failures into account. Accumulative failures at any time up to about 2500 hours run are at a rate of 4000 hours MTBF, above which the MTBF falls rapidly. On this analysis, had it not been possible to improve the engines, a reduction in the overhaul life from 4000 to 3000 hours would have had to have been considered in order to maintain reliability.

Plot B shows the effect of assuming the crankshaft, air start valve, phasing gear and balance weight modifications are fully effective. In the analysis failures attributed to these four causes have been eliminated. The general level of MTBF is doubled, but the unsatisfactory state above 3000 hours run remains.

Reference back to the Pareto plot shows the blower drive as the third major attributed cause of engine failure. In plot C all blower drive failures, as well as the four other types of failure, have been eliminated from the MTBF calculations, as if the problem had also been cured. The effect is to lift the MTBF at 4000 hrs service to 10,000 hrs. At this point a case could be made for extending the engine overhaul life to, say, 5000 hours.

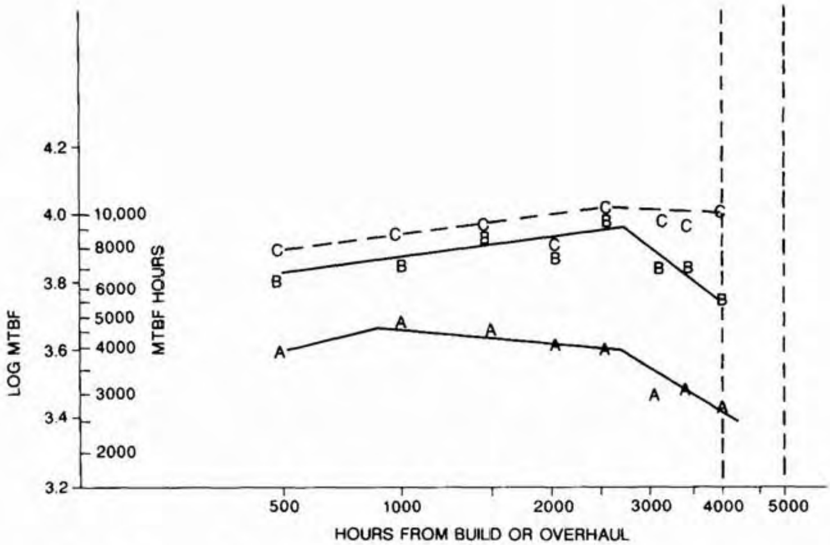

FIG. 6—DELTIC 59K ENGINES: MEAN TIME BETWEEN FAILURE V. ENGINE HOURS

In 1988 a higher proportion of engines reached 3000 hours for the first time. Examination of the failure statistics shows that, as expected, the main contributor to engine removal was blower drive failure.

Conclusions

The use of non-magnetic materials in the HUNT Class Deltic engines has presented its designers with a set of unusual problems to solve. The basic causes of these problems have all been identified and cures established, although in the case of the blower drive a certain amount of proof running remains to be done before final selection of the way forward.

With the complete conversion of all engines to the latest standard of build the prospect of halving failure rate in engines having up to 3000 hours of service looks good. Introduction of an effective blower drive modification should eventually quadruple the historic reliability at the overhaul point and enable that point itself to be extended to 5000 hours. That said, we must also be realistic and recognize that as larger numbers of the new standard of engine are afforded the opportunity of running in the 3000 hours plus region one or two new problems may appear.

Finally, in the HUNT Class Deltic engine non-magnetic material substitution has been introduced to an extent that makes the engine and gearbox combination as non-magnetic as has been achieved anywhere. This is the paramount advantage of this somewhat complex design. Whilst the current mean time between failure and overhaul life may appear modest by normal medium speed diesel engine standards, the HUNT Class Deltic engine has already set a challenging standard for its specialist class.

Reference

1. Hill, R. C. F.: Do a Duane; *Journal of Naval Engineering*, vol. 23 no. 2, Dec. 1976 pp. 245–251.

EPICYCLIC GEARS

BY

D. E. YATES, B.SC., C.ENG., M.I.MECH.E., M.I.MAR.E.
(*Technical Manager, NEI-Allen Ltd. - Allen Gears*)

This article is an edited version of a paper presented at the Institute of Marine Engineers' Centenary Year Conference, 'Past, Present and Future Engineering in the Royal Navy', held at the Royal Naval Engineering College, Manadon, from 6th to 8th September 1989. It is printed here with the ready agreement of the Institute.

ABSTRACT

The basic principles of the design, operation and performance of marine propulsion epicyclic gears are discussed in relation to typical applications. The importance of noise and vibration is recognized and reference is made to a development programme using a large power circulating test rig. Factors determining the configuration and design details of multiple ratio gears are surveyed and examples are given. Specific applications of epicyclic gearing in relation to possible future naval requirements are included. The unique features of this type of gear acting as a differential to divide the power between the propellers of a contra-rotating propulsion system are reviewed.

Introduction

Marine propulsion epicyclic gears have been developed over the years[1,2] from their use as primary reductions in gas turbine powered fast patrol boats, through their use in steam turbine propulsion representing a cumulative power of more than 12 000 MW to the present day where they are employed for high speed yachts and naval hovercraft. Epicyclic gears are used extensively in industrial applications ranging from gas and steam turbine driven alternators to the recently installed water turbines, where gears, comparable in size with those required by final drive naval propulsion, have been developed.

It was at the end of the last World War that the Admiralty became aware of the high-powered epicyclic gears which were being designed and developed by Dr Stoeckicht of Munich and tribute must be made to the British naval officers who recognized the potential of this type of gearing and encouraged and promoted its development in the U.K.

It is interesting to note that the development of main propulsion surface-hardened gears for the British Navy—H.M.S. *Diana* in 1946 was the first to be so fitted—was complementary to the evolution of epicyclic gears where the conformal profile of the annulus mesh allows a through-hardened material to be used for this component whilst full advantage of the higher load capacity of the surface-hardened sun and planets is taken at the sun mesh.

Spur gears were used in the early epicyclic designs since the absence of axial thrust from the tooth forces resulted in uniform and symmetrical loading on the planet bearings. The generation of high tooth contact noise and vibration with these gears soon led to the adoption of double helical teeth for all high speed applications. The double helical gears with their larger effective facewidth and lower stresses resulted in a reduction in gear diameter for a given power and speed.

Since those early days, this type of gearing has evolved into the compact, lightweight reliable units of today. The success of this development may be judged by FIG. 1 which shows the cumulative power of epicyclic gears supplied by NEI-Allen Ltd - Allen Gears. The total power of nearly 22 000 MW includes 13 000 MW of gears for marine propulsion which range from the small 2600 kW units of the early naval fast patrol boats to the 25 000 kW steam turbine main propulsion gears of VLCCs and fast container vessels.

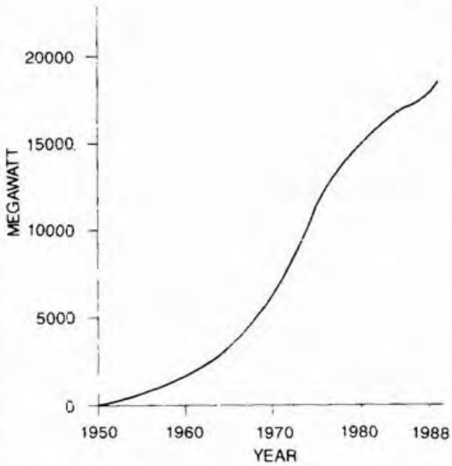

FIG. 1—CUMULATIVE POWER OF EPICYCLIC GEARS SUPPLIED BY ALLEN GEARS

Design Principles

An epicyclic gear consists essentially of three elements, the input, the output and the torque reaction. In a parallel axis epicyclic gear, the planet wheels mesh simultaneously with the sun and annulus. The planet wheels are supported on bearings located in the planet carrier. Any of these three elements—the sun, planet carrier or annulus may be input, output or torque reaction and permutations on these result in the three types of epicyclic gear—the planetary where the stationary annulus is the torque reaction, the star where the carrier is the torque reaction, and the solar where the sun is the stationary torque reaction element.

The combination of elements, each of which can rotate, enables the gear to be used as a differential in which rotation is added to or subtracted from the reaction element to change the speed of the input or output. It is this feature that makes epicyclics particularly suitable for variable ratio or reversing gears.

The epicyclic gear employed as a differential can also be used to divide the power between two outputs. This can give arrangement advantages in multi-stage gearing and be used for dividing the power between two outputs such as a contra-rotating propeller system[3,4].

The multiple torque paths provided by the planet wheels enable the size of an epicyclic unit to be significantly smaller than the equivalent parallel gear.

The load sharing of a three-planet epicyclic gear, like the three-legged stool, is kinematically balanced if one of the elements is allowed to float and take up a position determined by the tooth contacts. Thus, if the sunwheel is allowed to float within its backlash, the load sharing will be balanced no matter what tooth or planet positioning errors occur.

It is an advantage to use more than three planets where the ratio permits because of the reduction in gear size resulting from the increased number of transmission paths. Load sharing between four or more planets depends on the planet positioning and gear tooth errors, and the torsional flexibility of the gear system. The flexibility must be such that the deflection at the teeth is greater than the combined errors at the tooth meshes. Several methods of obtaining the flexibility are employed and the author's company uses a flexible annulus which is designed to deflect radially under the action of the radial tooth forces.

The success of the method of load sharing depends on the response of the flexible component to the errors under operational conditions. The effectiveness of the flexible annulus method has been thoroughly evaluated on a development test rig at NEI-Allen Ltd - Allen Gears, using carefully calibrated strain gauges positioned in the tooth roots and on the body of the annulus. The results of the tests have indicated that the maximum variation in load between any of the meshes of either helix of a typical five-planet gear is no more than 6%.

The operating conditions of most high-speed epicyclic gears necessitate the use of white metal lined hydrodynamic bearings where the white metal is deposited on the spindle to avoid fatigue loading. In contrast to parallel gears, the output bearings of an epicyclic support only the weight of the associated components and are not subject to tooth loads. The low speed torque is transmitted through the planet bearings where the higher speed results in better hydrodynamic bearing performance. In other words, the higher speed of the loaded planet bearings and the lower load of the output bearings on a epicyclic gear operate under better hydrodynamic conditions compared with the parallel shaft bull wheel bearings which are subject to the combination of high load at low speed particularly under manoeuvring conditions. The balance of the tooth forces on the sunwheel and its light weight obviates the need for bearings to support it and results in a much improved efficiency. It is usual to measure more than 1% better overall efficiency compared with the equivalent parallel gear.

Multi-stage Gears

Large overall ratios require multiple reduction gears. There are several factors which govern the choice of the number and division of the ratios. The advantages of minimum overall space and weight which result from a larger number of stages each with a smaller ratio, must be balanced against the higher efficiency and smaller number of parts with fewer stages. The reduction in weight and size of a gear is reflected in the reduction in cost and is illustrated in FIG. 2 which shows the relative cost of single and multiple reduction gears on the basis of overall ratio. It is also seen that the change-over points in the number of stages varies with the low speed torque and reflects the lower relative cost of larger sized gears in contrast to smaller units.

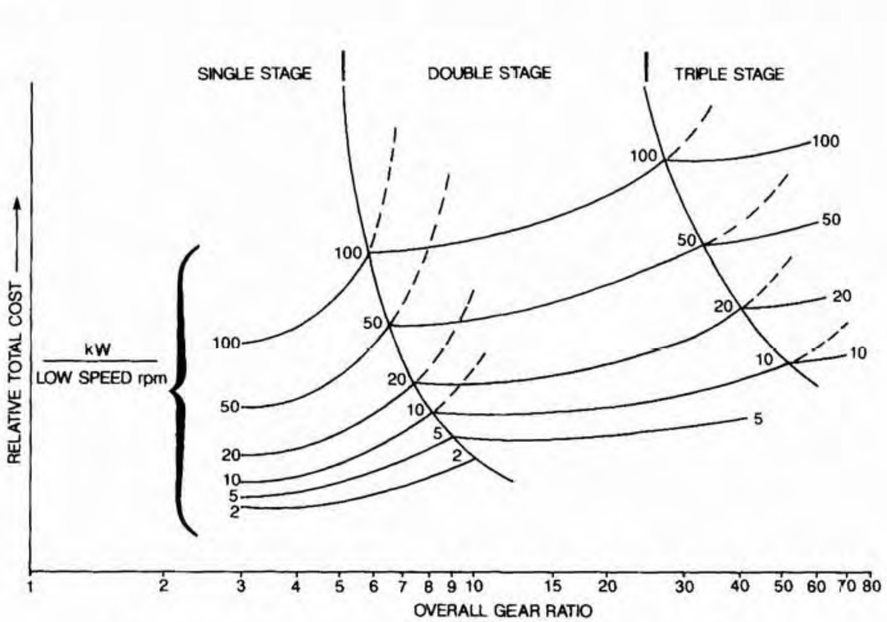

FIG. 2—THE ECONOMIC SELECTION OF MULTIPLE STAGE EPICYCLIC GEARS

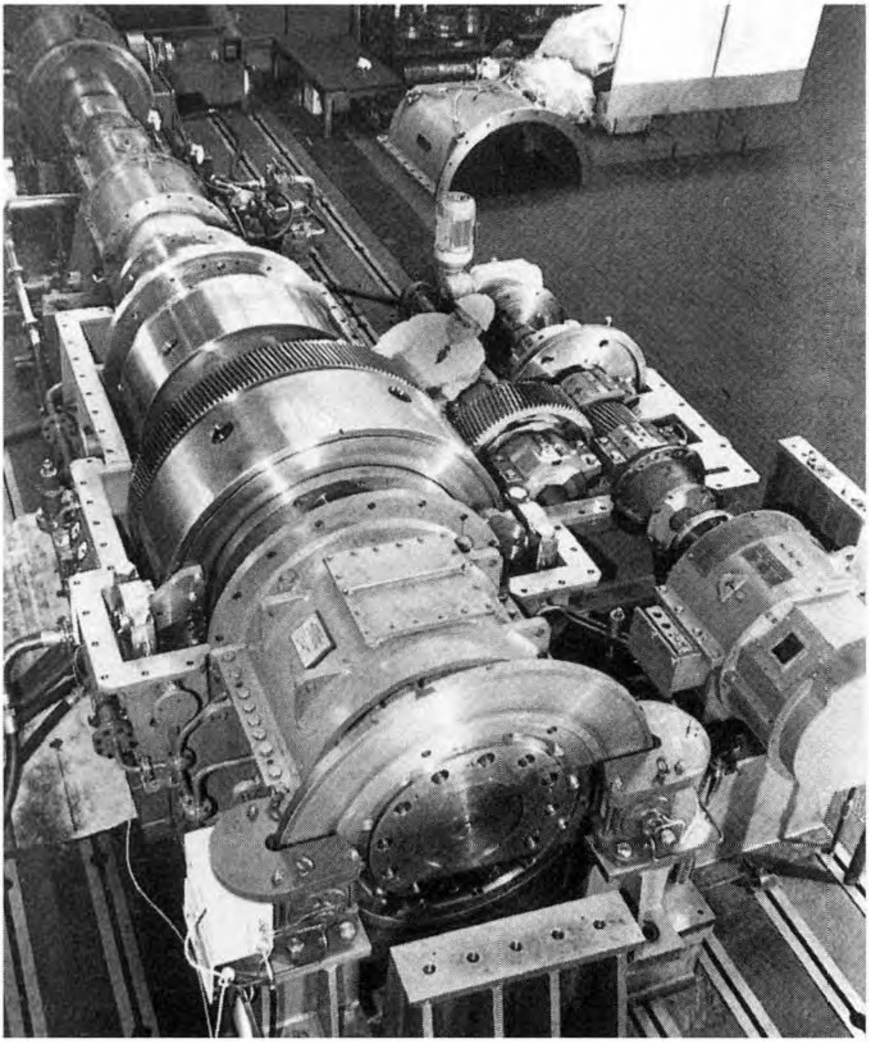

FIG. 3—TRIPLE STAGE EPICYCLIC GEAR FOR 'IRON CARPENTARIA': 8500 kW, 6556/112 REV/MIN

The small size of reduction gear which can be achieved with a multi-stage epicyclic is demonstrated by FIG. 3 which shows the reduction gears for the BHP bulk carrier *Iron Carpentaria* whose gas turbine driven triple stage epicyclic gear has an overall ratio of 58.53[5]. These gears included an auxiliary alternator drive which could be used as a motor to power the propeller shaft in the event of a breakdown of the main drive. The small size of the final stage epicyclic may be judged by comparison with the integral thrust block adjacent to it.

In arrangements where two or more prime movers are employed, parallel gears generally serve to combine the inputs and allow the best disposition of the prime movers relative to the propeller shaft within the engine room layout. The minimum overall weight and space is achieved when epicyclic gears are used for the low speed reduction of a multi-stage gear. This was recognized by Pamatrada in 1965 when the 'Paraplan' epicyclic gear was being proposed for multi-cylinder steam turbine applications[6]. In contrast, the highly successful Stal Laval A.P. machinery used a final drive parallel gear to obtain a single plane arrangement[4]. It is the compromise between the optimum engine room layout and minimum overall weight and space which determines the best gear arrangement.

An excellent example of a multi-stage gear, where the engine room layout has required high speed primary epicyclics and a low speed parallel shaft combining gear, is the fast yacht *Shergar*. FIG. 4 shows this jet pump gear powered by two Allison 570 engines. The gearcase, of fabricated steel, supports the overhung engines and houses the primary epicyclics, each connected through an SSS clutch to the final drive parallel gears.

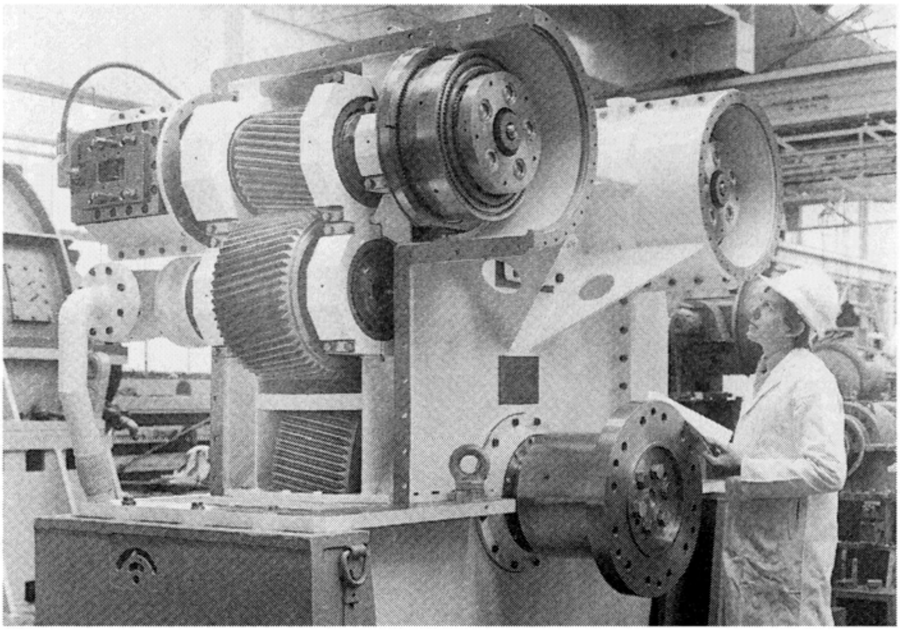

FIG. 4—JET PUMP PROPULSION GEAR FOR 'SHERGAR': TWO 5210 kW, 11500/1648 REV/MIN

FIG. 5 shows the combined epicyclic and parallel shaft gear for a high speed launch which uses an AVCO gas turbine driven jet pump for boost operation. The engine room configuration has required the use of a pinion, idler and wheel to achieve the required centre distance between the primary epicyclic gear and the final drive to the jet pump.

Noise and Vibration

In practice, there is little difference between the noise and vibration levels of double helical epicyclic gears and similar parallel gears of the same duty and accuracy. The generation of vibration by the larger number of mesh points of a epicyclic gear is offset by the attenuation of the vibration at the coupling and bolted joints in its transmission path to the gearcase walls and feet. Reduction in transmitted vibration can be achieved by flexibly mounting the gears. The co-axial input and output make epicyclic gears particularly suitable for this purpose since the reaction from the gear is nearly a pure torque, and would not result in any significant translation deflections of the

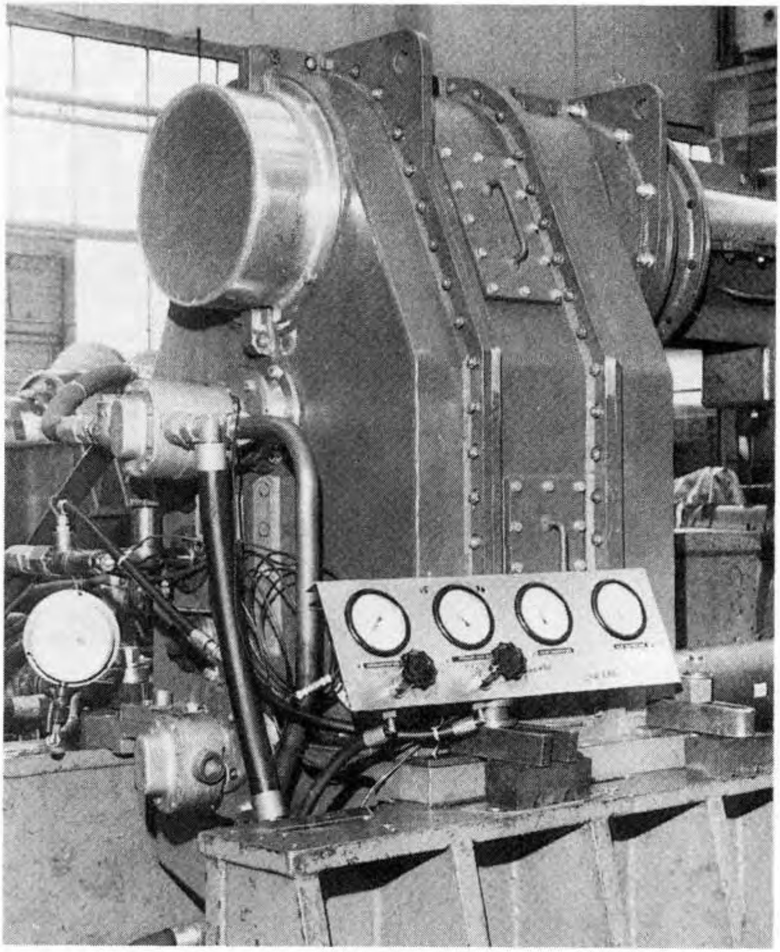

FIG. 5—COMBINED EPICYCLIC AND PARALLEL SHAFT GEAR FOR A JET PUMP PROPELLED FAST LAUNCH: 3500 kW, 15400/1664 REV/MIN

gear components which might affect alignment. Flexible couplings between the gear and the input and output shafts would be required to prevent transmission of vibration along these paths.

In order to obtain a more accurate picture of the noise and vibration signature of typical industrial epicyclics, the comprehensive measurement of a star gear, designed to transmit 18 000 kW with a speed ratio of 5400 to 1800 rev/min, has been carried out in collaboration with the MOD using the power circulating development test rig already mentioned. This power and input speed corresponds to the probable maximum power of the Rolls-Royce Spey engine.

A description of the test rig and its associated instrumentation, which has been used for the fundamental and applied epicyclic research development since 1975, has already been published[7,8]. FIG. 6 shows the test rig in operation. In order to distinguish between the vibration at tooth contact frequency, the two gears were made with different pitches and numbers of planets but employed identical reduction ratios to enable them to be tested by the power circulation method.

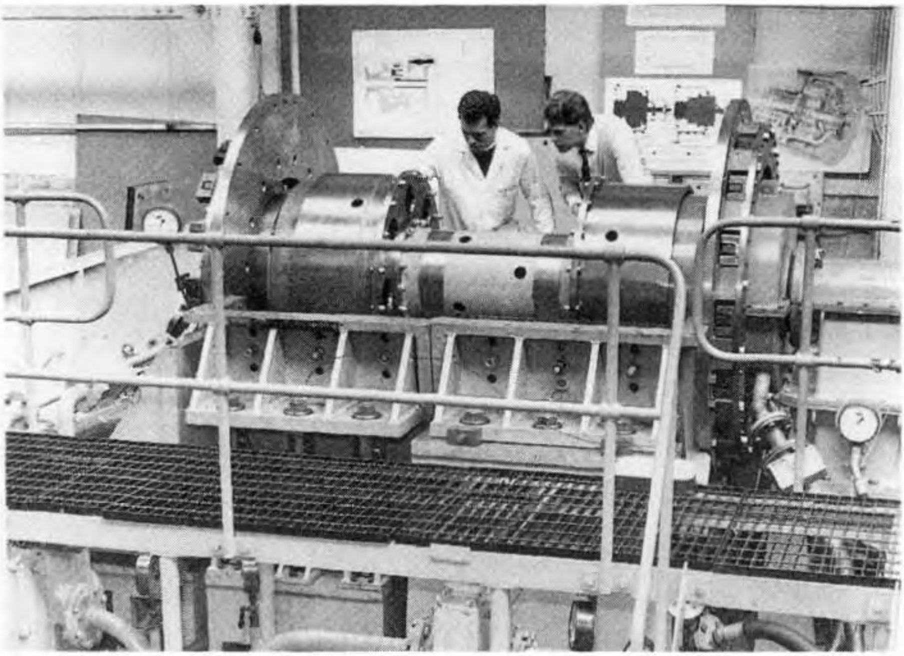

FIG. 6—EPICYCLIC DEVELOPMENT TEST RIG

The gears were designed to enable flexible supports to be fitted between the torque reaction gear components and the gearcase, and tests were carried out to measure the effectiveness of these. The vibration levels measured at the gearcase feet without flexible supports were well within the accepted criteria for precision gears. A reduction of some 15 dB in vibration velocity resulted when the flexible torque reaction supports were fitted.

Further development to minimize the generation of vibration is currently being carried out.

Epicyclic Gears for Reversing

The early BRAVE Class of fast patrol boats, powered by Rolls-Royce Proteus engines, employed final drive reversing epicyclic gears in addition to the primaries. Epicyclic gears do offer a number of advantages as a means of reversing and have been reviewed in an earlier paper[3].

Combinations of two epicyclic gears can be arranged so that shaft reversal is achieved by applying a stationary brake to the torque reaction member of either gear. The lightest and most economic combination is the star/planetary, planetary/solar arrangement used in the BRAVE Class vessels. This arrangement ensures that both gears are loaded for each direction of rotation, thus avoiding tooth separation and gear hammer during the reversal. None of the gear components come to rest during the transition and consequently are not subject to static friction which can contribute to propeller shaft stall.

The energy dissipation during a reversal of one of the BRAVE Class vessels was relatively small and permitted the use of oil-cooled drum brakes. Investigations into the reversing of larger displacement vessels indicated much larger energy dissipation and would necessitate the use of air-cooled dry friction brakes mounted outside the gearbox.

Gas Turbine Power Generation

The use of epicyclic gears for gas turbine power generation is advantageous since the coaxial input and output results in a more compact overall arrangement. This is particularly true when the epicyclic gear is flange-mounted directly on to the alternator thus obviating the need for low speed couplings.

The torque reaction of the gear is taken directly by the alternator housing and enables the bed plate to be made of a shorter and lighter construction. Such an arrangement is used for the Centrax Allison powered alternators and is shown in FIG. 7. The gearbox incorporates auxiliary drives for lubricating oil and fuel pumps as well as the gas turbine starter. The integration of these within the gearbox has resulted in a significant reduction in the overall length of the plant.

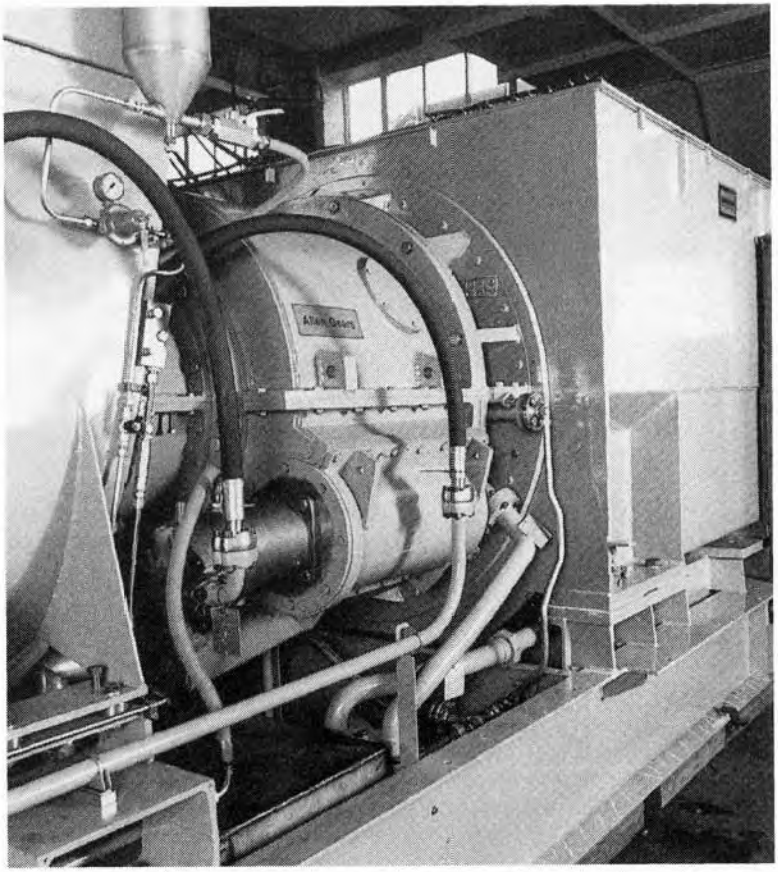

FIG. 7—ALTERNATOR-MOUNTED EPICYCLIC GEAR: 6000 kW, 11500/1500 REV/MIN

A recently developed compound epicyclic gear which transmits 2300 kW from a turbine speed of 27 288 rev/min to an alternator speed of 1500 rev/min is illustrated in FIG. 8. The high pitch line speed of the sun/planet mesh necessitates the use of double helical gears whilst spur teeth are used for the lower speed planet/annulus mesh.

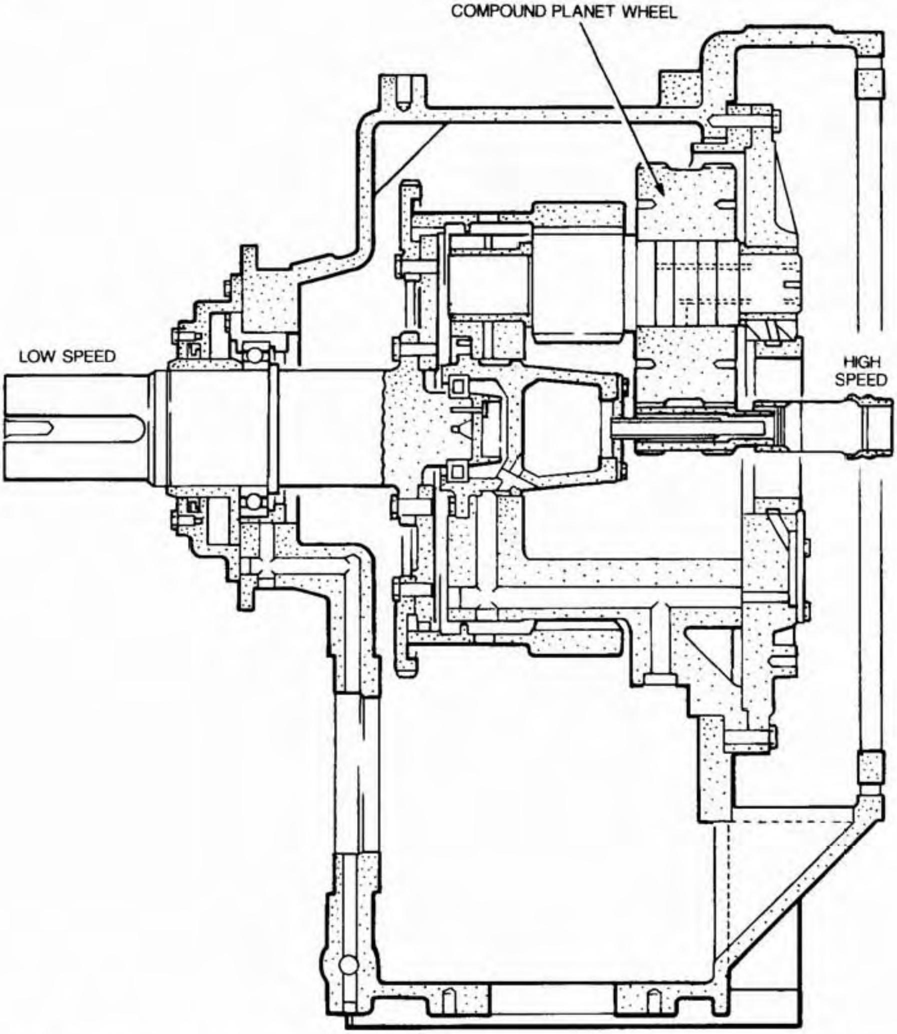

FIG. 8—COMPOUND EPICYCLIC GEAR FOR POWER GENERATION: 2300 kW, 27288/1500 REV/MIN

Gears for Navies

The use of epicyclic gears by the Royal Navy, hitherto, has been confined mainly to fast patrol boats where the advantages of their light weight and compactness are obvious.

Epicyclic gears, fitted as primaries to the Rolls-Royce marine Proteus engine, were introduced in 1958. Since then, some 250 units, which in the final version transmitted 3360 kW with a speed reduction of 11 600 to 5240 rev/min have been supplied to 13 navies. There have been no reported service failures of any of these units.

FIG. 10 shows one of the Royal Swedish Navy SPICA III vessels powered by Allison 570 engines with similar gearing which, as in the Proteus, was integrated with the engine and resulted in a very compact arrangement.

The prototype gears for the Spanish Navy hovercraft made by Chaconsa employs a transmission system which includes epicyclic gears (FIG. 9). Each port and starboard set of machinery is powered by an Avco Lycoming TF25 engine which drives through a primary epicyclic gear and divides the load between the parallel shaft gear driven lift fan and the bevel driven propulsion fan. An emergency cross drive shaft can be used to connect both gears in the event of an engine failure. The epicyclic gears, together with the use of fabricated aluminium gearcases, have combined to produce this lightweight transmission system.

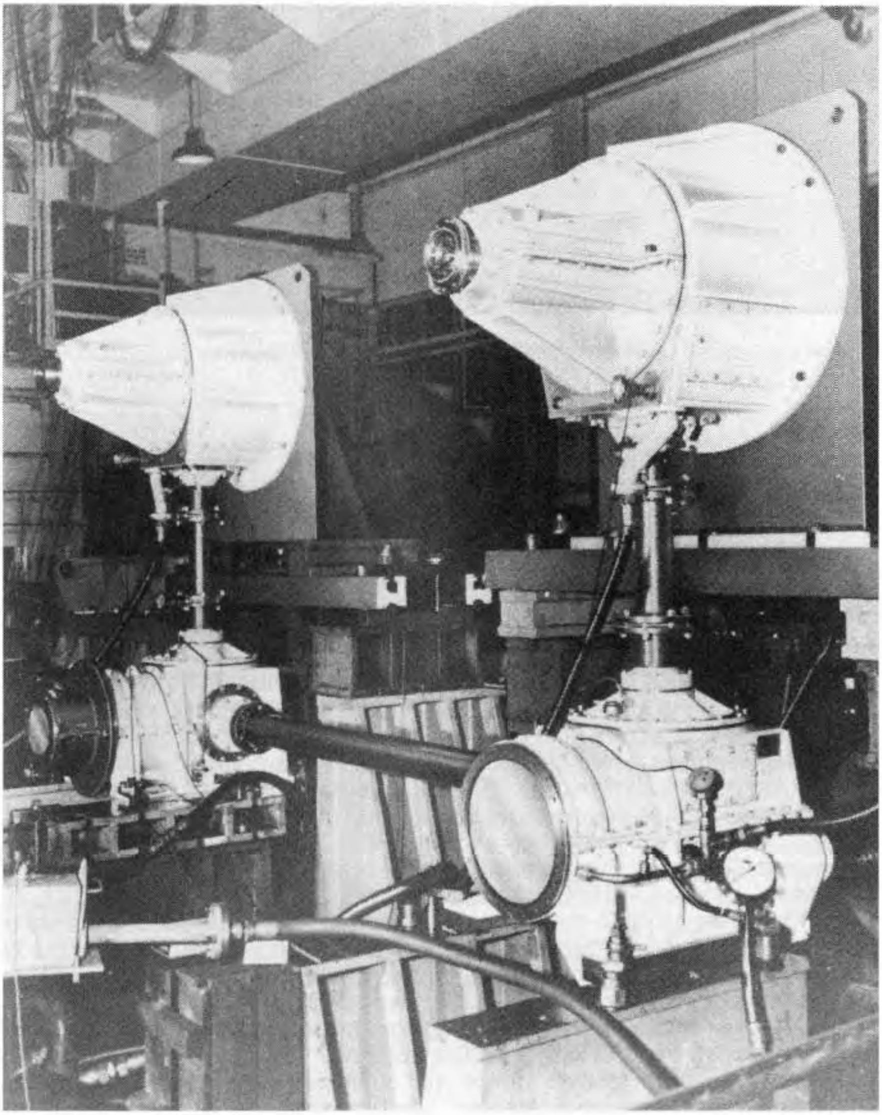

FIG. 9—THE PORT AND STARBOARD GEARS SUPPLIED FOR THE SPANISH HOVERCRAFT CONSTRUCTED BY CHACONSA

FIG. 10—ROYAL SWEDISH NAVY 'SPICA III'

Epicyclic Gears for Future Naval Applications

The choice of a gas turbine primary epicyclic for a CODAG or CODOG arrangement is obvious and would follow the experience existing with the high speed yachts. The scheme given in FIG. 11 employs a single diesel and gas turbine for each shaft. The conventional friction clutch connecting the diesel to the gearbox would be of the multi-plate oil cooled type and would be housed within the gear. An SSS clutch between the primary epicyclic and main gear would provide the gas turbine disconnection feature and reversing would be carried out with a controllable pitch propeller or a reversing diesel.

The use of multiple reduction gears in an arrangement similar to the Type 23 Frigate is shown in FIG. 12. This is based on the Rolls-Royce Spey engine developing a probable uprated power of 18 000 kW. The saving in weight would amount to some 15–20%; the actual saving with an existing Spey engine would be greater. The low weight of the individual stages permits their easy removal for inspection and servicing. The complete low speed stage of the gear shown in FIG. 12 would weigh some 8.5 tonnes and permits its easy removal for inspection and servicing.

An interesting variant from the above could be the use of an all-electric drive using the Type 23 direct drive motor in conjunction with a 18 000 kW, 2650 rev/min geared electric motor and SSS clutch for high power operation. The gas turbine generators located at or near deck level would save the space required by the intake and exhaust ducts of a direct drive engine positioned low in the vessel. FIG. 13 shows such a gearbox.

The concept of a geared motor for merchant marine propulsion was investigated more than ten years ago[3]. At that time a 20% reduction in cost would have resulted from the use of a 6700 kW geared motor driving a propeller at 115 rev/min.

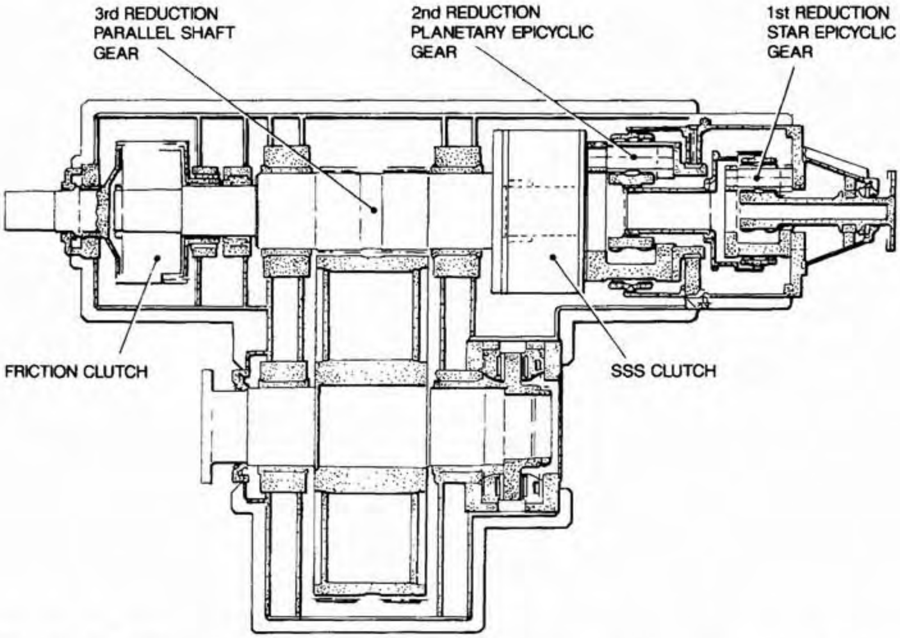

FIG. 11—CODAG OR CODOG GEARBOX INCORPORATING EPICYCLIC GEARS FOR THE GAS TURBINE PRIMARIES: 4500 kW, 520 REV/MIN AND 18000 kW, 5400/180 REV/MIN

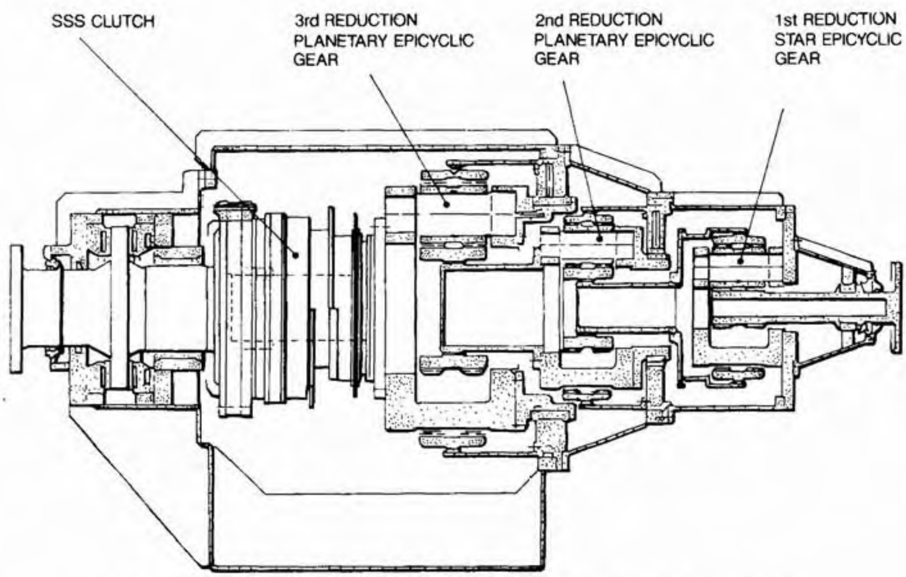

FIG. 12—CODLAG EPICYCLIC GEARBOX: 18000 kW, 5400/180 REV/MIN

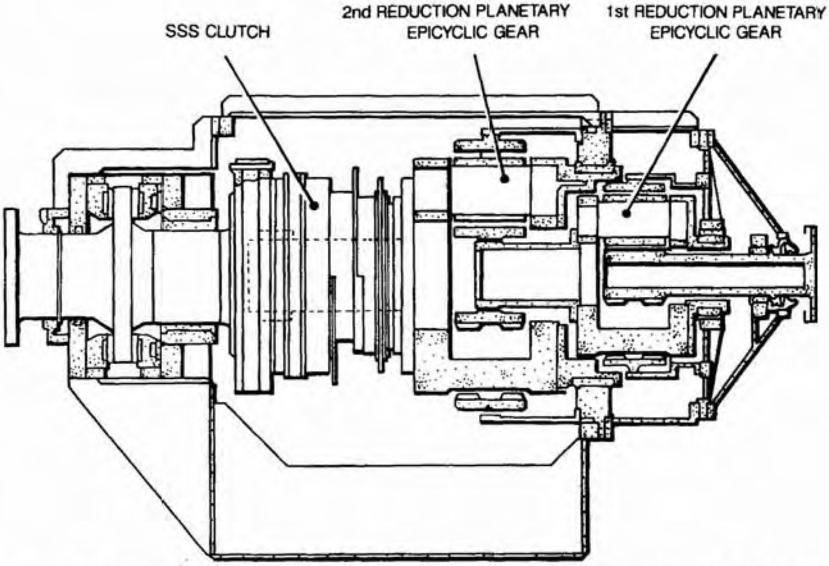

FIG. 13—GEARED ELECTRIC PROPULSION MOTOR EPICYCLIC GEARBOX: 18000 kW, 2650/180 REV/MIN

The use of epicyclic gears combined with a direct drive gas turbine and a reversing hydraulic coupling of the Franco Tosi type powering a single propeller as illustrated in FIG. 14 suggests that a standardized propulsion module, possibly raft-mounted for minimizing noise, could be developed[9]. Variations in gas turbine speed and power and propeller speed would not change the basic layout of this arrangement. The concept of providing the same developed module in vessels for different applications could be logistically attractive. This scheme may be particularly suitable for a combined cycle gas turbine with its good part load economy.

The basic concept of a combined epicyclic and parallel shaft gearbox incorporating reversing hydraulic couplings and driven by a diesel engine can offer significant weight, space and arrangement advantages.

Contra-rotating propellers for improving efficiency and reducing the water-generated noise have been under consideration for a long time[4]. The use of an epicyclic gear as a differential to balance the powers of the two propellers was one of the schemes investigated and it offered a very attractive solution enabling a large reduction ratio to be achieved with a small sized gear. The two propellers would be connected to the planet carrier and annulus respectively whilst the sunwheel would be the input.

Since, with this arrangement, there is no reaction torque, the input and outputs must balance and the planet carrier torque must equal the sum of the sun and annulus torques. Early analysis of arrangements where the annulus was connected to the larger forward propeller and the carrier was connected to the after propeller indicated a distribution of power and speed which would result in a poor overall efficiency[10,11]. A better distribution can be achieved, however, by connecting the carrier to the forward and the annulus to the after propeller. The resulting power and speed distribution would then be similar to that used on the recently reported *Toyofuji No. 5* where there was a speed ratio of 185/139 between after and forward propellers[12]. The improvement in efficiency in this case was measured to be some 16%. An arrangement which would give this power distribution with a simple epicyclic gear is shown in FIG. 15.

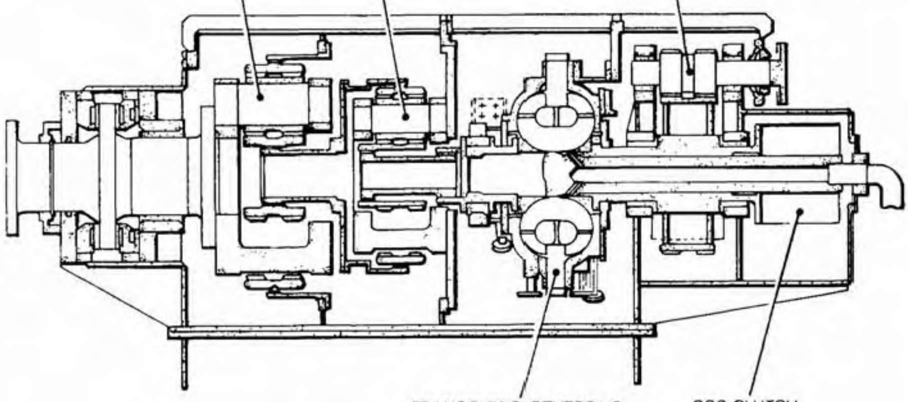

FIG. 14—EPICYCLIC GEARBOX EMPLOYING FRANCO TOSI REVERSING CONVERTER-COUPLING: 18000 kW, 5400/180 REV/MIN

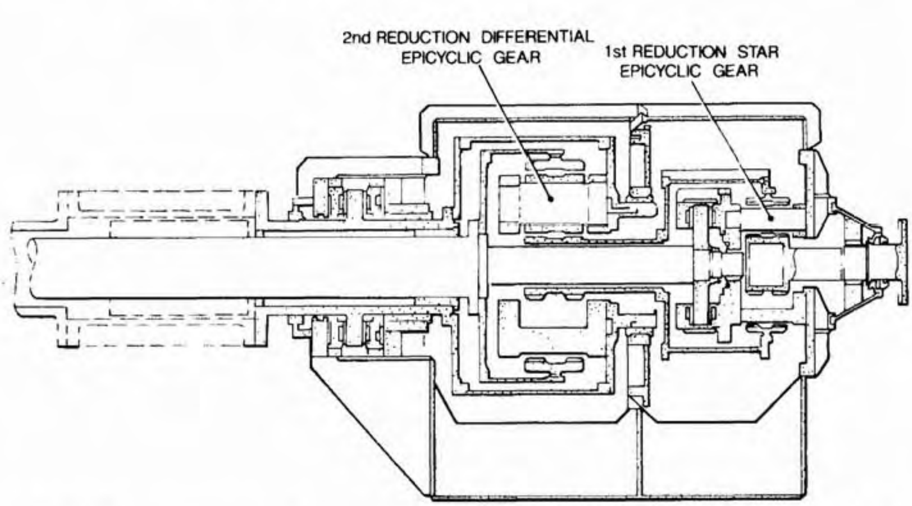

FIG. 15—CONTRA-ROTATING PROPELLER GEARBOX EMPLOYING AN EPICYCLIC GEAR: 18000 kW, 2650/206/155 REV/MIN

Concluding Remarks

Epicyclics present navies with a number of ways in which their compact and lightweight construction can be used to significant advantage. The use of an epicyclic gear acting as a differential is one of the most economical ways of dividing the power of a contra-rotating propeller scheme.

Epicyclics for primary reductions of steam and gas turbine marine propulsion have been very well proven. Their use for medium speed diesel propulsion is also commonplace. The dearth in the building of large ships over the last 10 to 15 years and the adoption of cathedral diesel engines in merchant vessels in the interests of economy have slowed down the development of large final stage epicyclic gears for ships' propulsion.

The development of large gears in a comparable application—that of water power generation—now presents the marine industry with large proven epicyclic gears with torque capacities at present far larger than those likely to be required for naval propulsion. FIG. 16 shows the low speed stage of the water turbine gear designed and constructed for the Southern Electric International for their Murray Lock and Dam water turbine installation at Little Rock, U.S.A. With a nominal power of 23 600 kW and an input speed of 45.3 rev/min, it has an annulus pitch circular diameter of 2.112 m.

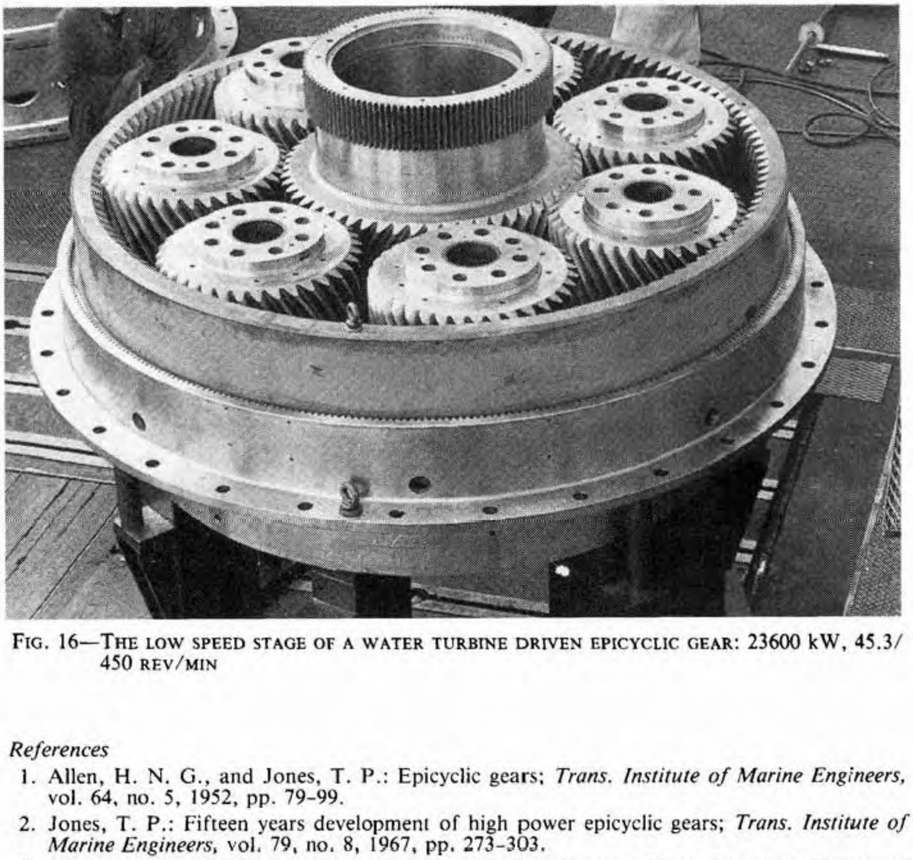

FIG. 16—THE LOW SPEED STAGE OF A WATER TURBINE DRIVEN EPICYCLIC GEAR: 23600 kW, 45.3/450 REV/MIN

References

1. Allen, H. N. G., and Jones, T. P.: Epicyclic gears; *Trans. Institute of Marine Engineers*, vol. 64, no. 5, 1952, pp. 79-99.
2. Jones, T. P.: Fifteen years development of high power epicyclic gears; *Trans. Institute of Marine Engineers*, vol. 79, no. 8, 1967, pp. 273-303.
3. Yates, D. E.: Epicyclic gearing for gas turbine main propulsion; *American Society of Mechanical Engineers 21st Gas Turbine Conference, New Orleans, 1976*, Paper no. 76-GT-24.
4. Jung, I. K. E.: Steam turbine machinery; *Trans. Institute of Marine Engineers*, vol. 81, no. 5, 1969, pp. 137-150.
5. Shannon, G. H.: Breakthrough in gas turbine propulsion gearing; *APE Engineering*, no. 21, March 1976.
6. Coats, R.: Pamatrada standard turbines. Present position and future outlook; *Trans. Institute of Marine Engineers*, vol. 77, no. 10, 1965, pp. 327-343.
7. Cole, G. F. T., Lack, G. J., and Warin, J. W.: Epicyclic gear development; *APE Engineering*, no. 23, Dec. 1976.
8. Warin, J. W.: The development of a telemetry system for a gear test rig; *APE Engineering*, no. 27, March 1978.

9. Clements, H. A., and Fortunato, E.: An advance in reversing transmissions for ship propulsion; *American Society of Mechanical Engineers 27th Gas Turbine Conference, Wembley, 1982,* Paper no. 82-GT-313.

10. Shannon, J. F., and Young, I. T.: Marine turbine propulsion gearing—the last decade; *Proc. Institution of Mechanical Engineers,* vol. 184, part 30, paper 2.

11. Shannon, J. F.: *Marine gearing;* London, Marine Media Management, 1977.

12. Nakamura, S., Ohta, T., Yonekura, K., Sasajima, T., and Saki, K.: World's first contra rotating propeller system successfully fitted to a merchant ship; *The Motor Ship 11th International Marine Propulsion Conference, March 1989.*