TRANSMISSION DESIGN FOR WARSHIPS OF THE ROYAL NAVY

G. B. Beale, M.Eng, C.Eng, M.I.Mech.E., R.N.E.S.* and Lieutenant-Commander B. J. McD. Gowans, R.N., C.Eng., M.I.Mech.E., (Associate Member)†

Mr. Beale Lt. Cdr. Gowans

This paper considers the problems associated with the transmissions of warships driven by gas turbines. It discusses the two systems of reversing, through a reversing gearbox or by controllable pitch propellers and reviews the previous experience in the Fleet and research and development of components in transmission systems. The initial teething problems in H.M.S. *Exmouth*, the first all gas turbine warship, are discussed and the paper concludes by describing the design of the transmission for the Type 42, the all gas turbine destroyer now being built.

INTRODUCTION

It is the policy for all future major surface warships in the Royal Navy to use gas turbine propulsion. This policy has been decided upon in order to achieve greater availability, a better weapon payload, simplicity in operation and a reduction in manpower afloat.

The gas turbines are marinized versions of well proven aero engines. These engines are not reversible.

The transmission, gearing shafting and ancilliaries, have to convert the engine characteristics to those of the propeller in the best possible manner, provide the reversing capability, fit the ship layout, meet the requirement of the ship command and be reliable and maintainable.

This paper, from the section within the Ship Department of the Ministry of Defence responsible for propulsion transmission systems, describes how the various components of the system are chosen.

TRANSMISSION REQUIREMENTS

In order to carry out his task, the designer must bear a number of considerations in mind. Some of these are listed below, not in any order of importance since this may vary depending on the particular design.

Operator Requirements—speeds (maximum, slow, cruising), acceleration, stopping distance.
Military requirements—noise, habitability, shock.
Characteristics—of prime movers, hull, propulsors.
Costs—development, initial and through costing.
Timescales—for development, testing and production.
Reliability—effect of breakdowns, time between refit.
Maintenance—requirements and accessibility.
Manning—the manpower to operate and maintain.
Training—specialist operator and refitting knowledge.
Logistics—requirements for spares, etc.
Standardization—similarity throughout the Fleet.

In order to establish the system parameters a simulation is carried out of the machinery driving the hull in all the operating modes and various failure states. From this study the design requirements for each component in the system may be stated, and used as a basis of equipment selection.

* Naval Ship Production Overseer, East Midlands.
† Deputy Head Transmission Section, D.G. Ships, Bath.

SYSTEM SELECTION

With unidirectional gas turbines the major decision to be made is on the best way of achieving astern power, with a c.p. propeller or a reversing gearbox. These are compared under a number of headings below. Alternatives, such as electrical systems, are not discussed since, at this time, they have not been developed sufficiently for use in warships.

Operational

Controllable pitch propellers in naval ships change from ahead to astern pitch by rotating the blades through zero pitch. This means that there can be no overlapping of the blades and hence blade pressures are higher than for the equivalent f.p. propeller. The hub diameter is larger than that of a f.p. propeller and the blade roots need to have thicker sections in order to take the bending moment while blending into a swivelling circular palm. The net result of these limitations is that a c.p. propeller has an inferior form compared with a f.p. propeller, leading to lower efficiency with the possibility of cavitation.

To manoeuvre with a c.p. propeller, engine power and pitch can be matched to meet the operator requirements by suitable controls. Controllable pitch propellers have a design pitch and operation off this pitch reduces efficiency and further promotes cavitation. It is, therefore, desirable to operate the propeller as a f.p. propeller over the whole ship speed range for normal operation and use the variable pitch facility only for reversing. This type of operation is made possible with a free power turbine which at low ship speeds can operate below its unloaded idling speed to match the full pitch characteristics of the propeller. For rapid manoeuvring, however, better response can be achieved by varying propeller pitch with changing engine power. Fig. 1 shows an example of each of these operating modes.

The Royal Navy has 25 reversing gearboxes at sea in major warships. These have been satisfactory in service, although they are complicated to operate and require user skill and technique; under very difficult conditions maloperation could occur.

Reversing is achieved by a separate manoeuvring train

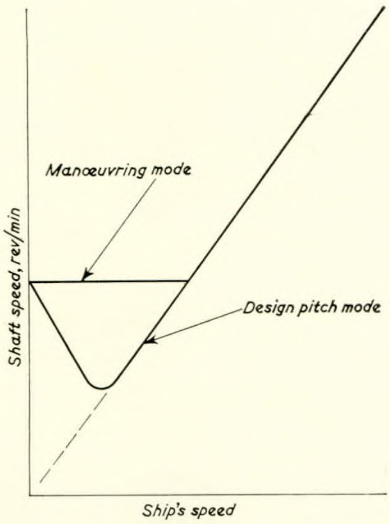


FIG. 1—*Typical speed rev/min characteristic for a c.p. propeller installation showing alternative operating modes*

incorporating fluid couplings. In order to limit the size of the couplings and the oil quantity the power transmitted through the coupling must be limited.

A new design is being developed which incorporates in line fluid couplings for manoeuvring and reversing and lock up clutches in parallel which can be arranged to short out the couplings. The fluid couplings will only be used at lower powers and less heat generated by coupling slip has to be dissipated. At higher powers the couplings are emptied and the drive is taken through the clutch.

In this design, the fluid coupling oil could have its own circuit and coolers so that sharp oil temperature transients will not affect gear lubrication. Furthermore, there will be no unloaded gears during any operation of the gearbox. Simplification of the design, attention to detail at the design stage and comprehensive interlocks will prevent damage by maloperation.

Manoeuvring

Although computer simulations have been carried out without two ships identical, except for their reversing mechanisms, it is very difficult to compare c.p. propellers and reversing gearbox installations on manoeuvrability and stopping distances. Ships with either system are, however, satisfactory.

The capability for stopping a c.p. propeller ship depends upon matching of the engine torque build up with the ability of the propeller to transmit the thrust into the water. Much depends on the response of the blades and hydraulic system and the loads that can be accepted in the hub. A problem peculiar to c.p. propellers is that when the blades go to zero pitch they form a disc which can effectively blanket an in line rudder.

With reversing gearboxes, the stopping distance is a function of the torque that can be transmitted through the fluid couplings when the two halves of the coupling are rotating in opposite directions. It is possible to use the natural braking of the ship so that correct choice of the phasing of the application of the couplings has a marked effect on the coupling size.

Size

As a straight criterion, size is not always valid, for the aim is to make the best use of the space available.

Controllable pitch propeller systems incorporate smaller gearboxes but require separate hydraulic pumps, coolers and tanks.

Reversing gearboxes require extra oil for couplings and this too may involve extra pumps, coolers and tanks. These gearboxes are generally larger than those for c.p. propellers as shown in Fig. 2. Occasionally in a multi-engine installation this difference in size is very small, since the positioning of the engines often dictates the gearbox size.

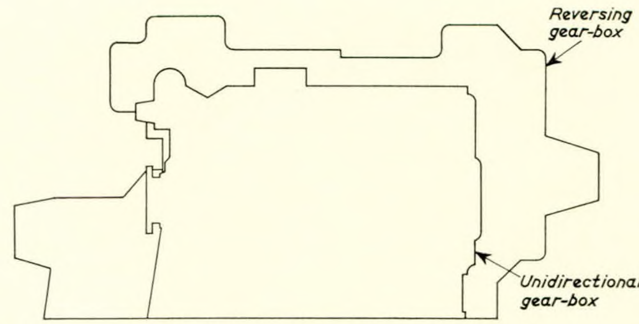


FIG. 2—*Illustration of size difference for a destroyer cogog gearbox design*

Maintenance

For c.p. propeller installations the gearbox is simple and unidirectional. It is never stalled under heavy torque and there is no risk of slow speed scuffing. The hydraulic system has to be sealed where the oil passes into the shaft. If the hub requires maintenance the ship has to be docked.

The reversing gearbox contains more rotating parts and extra complexity from high speed clutches and couplings.

It is not possible to suggest which is the more inherently reliable, but it is to be expected that either, properly designed and produced, will perform satisfactorily.

Cost

Through cost studies have been made for both systems, taking into account capital cost, transmission and propulsive efficiency, refit and breakdown costs. These studies show that, for warships, there is little to choose between either system.

Training, logistics, standardization

These constrain the designer from making a purely technical choice. If the Fleet is to consist of a mixture of c.p. propeller ships and ships with reversing gearboxes, then there is a penalty in extra training for ships' staffs, extra spares holdings and lack of standardization.

The Choice

There are a large number of points to be taken into account before committing a class of ship to one or other design of transmission system. Each case must be judged on its merit and by systematically examining each of the requirements, the logical solution appears.

COMPONENT SELECTION

The selection of components depends upon the requirement for each within the system, experience with similar designs in service and the results of research and development. These are in fact all tied very closely. The impetus for research and development stems basically from the projected needs of the future coupled with the experiences with and limitations of machinery in service.

The organization of the Ministry of Defence is such that a specialist section is involved in the problems of the running fleet as well as new design work and research and development. With a large number of ships in service with a variety of prime movers and transmissions, it is possible that any one—guided missile destroyer, *Leander* class frigate or even coastal minesweeper, may provide information or reveal a problem potentially relevant to new designs. Provided operating conditions in terms of the basic engineering properties, stress, pressure etc., are used, there is no danger in reading across from one ship design to another.

Research and development is carried out for two reasons—to solve problems in the Fleet and to provide the design rules and equipments for the future. For the former much is necessarily rapid and *ad hoc*. For the latter, work is to a more defined plan in order to provide sufficient confidence to design whatever type of transmission is required when it is needed.

The number of options in terms of prime mover combinations and propulsors makes it economically impossible to cover the field in depth. In an attempt to overcome this the research and development is split into two categories. The first of these is common research and development, work which will be of benefit whatever type of machinery is to be used. In the case of specialized items of research and development, which may only have a possible use, spending is carefully restricted until certain decision points have been reached. During this restrictive period much of the thinking and planning of the project is done so that when an affirmative decision is taken the work can be pursued quickly and efficiently. If the decision is not to proceed, then the amount of wasted effort is kept to a minimum. In practice it does not always work so neatly since the decisions are rarely timed to suit the transmission designer.

In many industries where components are small and cheap, a 'suck it and see' approach to development may well be the most economical way of deriving the best product. In marine engineering, testing and development of equipments at full scale is expensive. Economy of testing can be achieved by breaking the work down into three phases, the theoretical analysis, small scale testing (not always essential) and full scale testing. With the analytical tools now available it is possible to gain a good theoretical understanding of most components and systems at a reasonably low cost. The results of this analysis does the initial sorting to reduce the number of physical tests to be conducted, and then defines the parameters for the tests.

Although a comprehensive research, development and trials programme has been drawn up, it is only possible here to highlight some of the more important, and, it is hoped, more interesting projects and the service experience on which they are based.

Propellers

The hydrodynamic design of propellers is conducted by the Admiralty Experiment Works based on tunnel and tank testing. The propellers are cast in a manganese aluminium bronze alloy and finished to a high degree of accuracy.

The achievement of the necessary accuracy is expensive and time consuming as it is at present done by means of hand chipping for the larger propellers or by copy milling for the smaller sizes. A profiling machine of adequate versatility to meet the Navy's needs would be very expensive and could cause a production bottleneck. Instead, a development programme is underway to increase the accuracy of casting so that only light hand polishing will be necessary. Techniques for predicting casting distortions are being developed so that moulds can be biased. The methods of prediction are a fusion of the latest analytical techniques with measured results taken over a large number of years.

Controllable Pitch Propellers

There are few c.p. propeller ships in the Royal Navy due to the preponderance of steam ships with reversible turbines. Controllable pitch propellers are fitted to some Diesel frigates, survey ships and tugs and H.M.S. *Exmouth*, the first all gas turbine ship. The propeller in H.M.S. *Exmouth* is the first highly loaded one in service and is dealt with separately. In the Diesel ships the c.p. propeller has permitted better b.m.e.p./speed matching, the pitch being variable from full ahead to astern through zero pitch. Problems have been experienced in the accurate setting of pitch, particularly important being the zero thrust position required for ship stopped condition with the shaft rotating. Accurate calibration with temperature compensation is required to eliminate this error.

The c.p. propeller type favoured is that with the hydraulic piston in the hub since this does not require load carrying cranks or rods down the tail shaft. The hub diameter is generally smaller and there is less mechanical problem than having the piston in the main shaft.

Reduction in hub diameter would lead to increased efficiency and lower blade loading if more of the disc area can be used for blades. The mechanical components of the hub are being critically examined to see where size can be reduced and the feasibility of increasing hydraulic operating pressures without causing sealing problems is being studied.

Shafting

The design rules used by the Royal Navy consider steady state bending and shear stress, separately, and take account of reductions of both diameter and fatigue limit by corrosion; these have proved adequate for present designs. More valid design rules are being formulated to take account of all the stresses in the shafting and study is in hand to produce such a criterion based on principal stresses. Some work is in progress to establish the actual conditions of loading, taking into account the effect of wake pattern, eccentric thrust, bearing weardown, vibration and the manoeuvring transients.

The greater weight of a c.p. propeller imposes increased bending loads on the main shaft. With warship configuration with long outboard shafts supported by 'A' brackets the high alternating bending stress can be in an area where the material fatigue limit is reduced by corrosion effects, so that the design must accommodate the combined effects of torsion and bending in an area liable to corrosion.

Experience with oil lubricated stern tube and rubber 'A' bracket bearings has not been satisfactory. Leakage into the oil has caused premature failure of the former and destruction of the latter has occurred if the shaft is not periodically turned when in dock. The present policy is to use sea water lubricated asbestos reinforced phenolic resin bearings. These give satisfactory service if designed to the correct hydrodynamic criteria, encompassing length/diameter ratio, projected area pressure and rubbing velocity. In cases where the shaft is deflected to a steep angle, the bearing is bored to match.

Face type stern seals are now invariably used. An emergency seal is, however, incorporated to allow maintenance without docking. Research is being conducted to produce better mating materials, such as ceramics, to increase overhaul life of seals in service. A longer term study into the fundamentals of sealing is being conducted in order to meet the sealing requirements of the future. Final testing of seals is being carried out in a Ministry of Defence establishment.

Main thrust blocks have been integral with the gearbox in recent designs in order to conserve space. This construction poses problems of access, maintenance and the transfer of thrust through the gearcase structure. Designs are being formulated for separately mounted thrust blocks incorporating facilities for absorbing propeller excited noise and axial vibrations. Such units could be mounted well aft where the ship's lines are finer, so permitting better gearbox mounting such that ship dynamic movements do not effect the meshing of the gear elements.

Inboard shafting has to be supported and watertight integrity at bulkheads must be assured. A simple combination of plummer block and bulkhead gland is being developed but the use of such a component is not always possible, since the position of the shaft components is dictated by the shaft deflected line and the vibration characteristics.

Checks are carried out to determine the acceptability of any shafting design. Using Clapeyron's equation and influence coefficients the optimum shaft line, to achieve satisfactory bearing loadings, and bending moment and bending stress curves are produced. A typical set is shown in Fig. 3. Shaft axial, torsional and transverse natural frequencies are also checked.

Hydraulics

There has been an increasing use of hydraulics in transmission systems in the last few years. Development work on hydraulic components and systems is carried out on Ministry of Defence sponsored rigs and work is in progress to improve reliability, efficiency and system silencing.

The use of similar components in all the hydraulic systems, developed for the Royal Navy requirements but applicable commercially, simplifies logistics and operator training. They require standards of cleanliness and maintenance which were, until recently, rare in marine engineering. However, the concurrent introduction of other sophisticated equipments and controls has brought about the necessary change of attitude and training.

For c.p. propeller hydraulic systems, the aim is to produce a compact power pack using axial piston pumps and flow

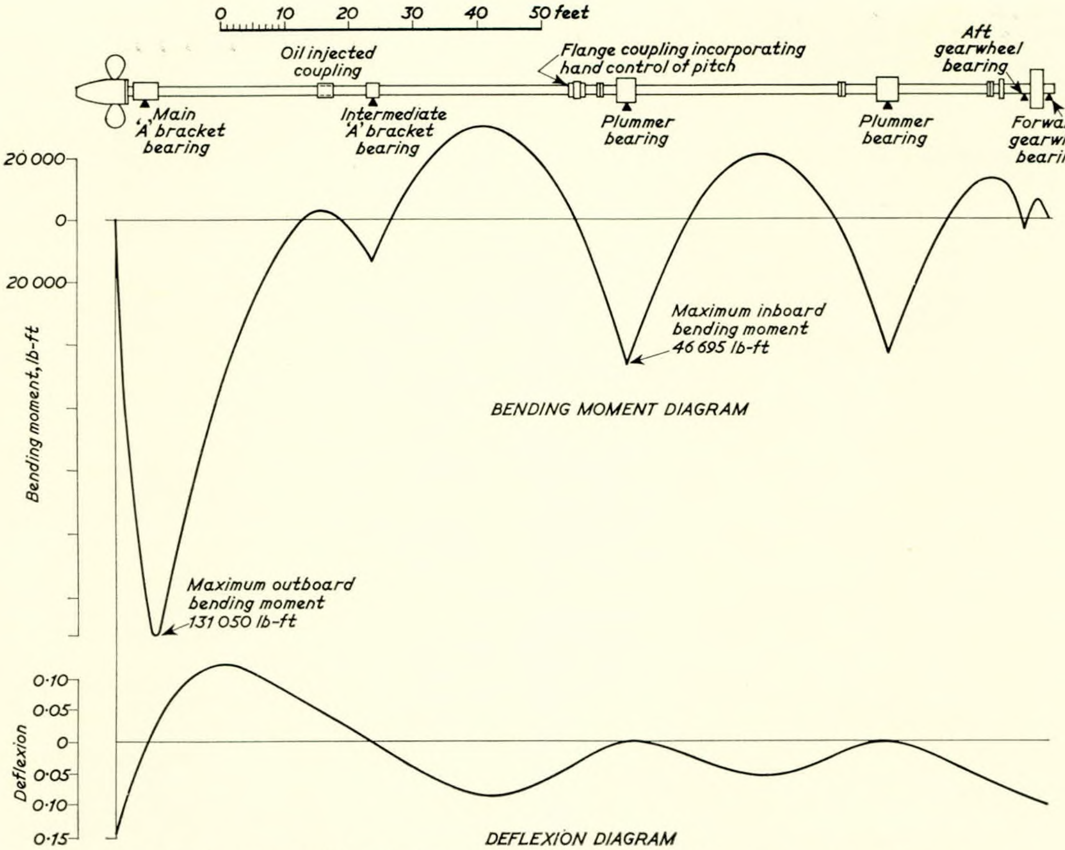


FIG. 3—*Recent shafting design showing deflexion and bending moment diagrams*

controlled systems since this has the greatest potential for increased pressure and efficiency. This will permit the engineering of pumps, header tanks, coolers, system valves and piping into a unit small enough to be integral with the main gearing. This will ease the machinery layout, by reducing the amount of piping and drive shafts, simplify local control, fault finding and feedback arrangements, and permit proper repair by replacement by flushed and tested units.

Gearing and ancillaries

The Ministry of Defence has maintained a full scale shore transmission test facility of a *County* class destroyer gearbox driven by two *G6* gas turbines. It has been possible to conduct on this rig a number of independent proving trials to back up the testing done at manufacturers' works and to reproduce the correct environment for developing and uprating in-service components.

Drive shafts and Couplings

In the *Leander* class frigates, the fine tooth flexible couplings between steam turbine and gearing have experienced severe fretting caused by misalignment. The misalignment is a result of differential growths, static deflexions of the ship structure under different conditions of payload, and dynamic movement of the hull being transmitted to the gearbox and turbine seatings. Palliatives have been strict alignment procedures and also the use of nitrided and sulphinused material combinations and oil retainment collars in the couplings. As a result of satisfactory service of membrane flexible couplings in *Tribal* class frigates and *County* class destroyers, a trial is to be carried out in 1970 with this type of coupling in a *Leander*.

In the *Tribal* class frigate gearing power is transmitted through a long jack shaft from the steam turbine. It has been found that only by accurate alignment and careful balance can the level of vibration, due to the slenderness of the shaft and the disposition of the masses on it, be kept within acceptable limits. This has indicated the need for accurate location and matchmaking of all parts during assembly, for all components and assemblies to be rigorously balanced to a fine tolerance and for accurate alignment of the axes of centres of gravity. It is nevertheless necessary to check that the vibration is below a level indicating a danger of mechanical damage. To do this, accelerometers are attached to bearing caps and an analyser can indicate the plane of correction such that metal can be removed from torque tube flanges to achieve smooth running. A design of coupling flange incorporating balance correction weights is being developed.

In cases where a large engine to gearbox separation is necessary and a conventional torque tube could whirl, a supercritical shaft has been designed and evaluated. In this case the shaft is allowed to run through its first critical at low input speeds, being restrained in damped bearings while it does so.

Clutches

Self-synchronizing clutches are used for the *County* class destroyer and the *Tribal* class frigate gas manoeuvring drives. These clutches incorporate an automatic disengagement and a lock-in facility to permit various machinery states.

Damage has been caused by inadvertant selection of engagement of the clutch when the input is running faster than the output, resulting in broken pawl and ratchet ring noses. Such damage has occurred with operators inexperienced with this complex gearbox and no interlocks can be incorporated to prevent it. The policy is to simplify the clutch so that the failure regime is not present. Thus, when the automatic disengagement is not required in a unidirectional transmission it will not be fitted. Simplification of the clutch and its controls can also be achieved by removal of the lock, which is only required when there is sufficient reverse torque over a long enough period of time to disengage the clutch against its dashpot.

Fluid Couplings

The fluid couplings for the *County* class destroyer and *Tribal* class frigate designs have given satisfactory service and provided precise and simple manoeuvrability. In the Type 82 destroyer the two *G6* gas turbines of the *County* class are replaced by a single *Olympus* engine such that twice the power must pass through a single manoeuvring train. The couplings used in previous designs were uprated by a development programme which included decreasing the coupling back pressure to get an increased oil flow through at 100 per cent slip, improving the runner and impeller attachments to permit higher stresses in the limited space, improving production techniques to remove stress raisers, and checking the natural frequencies and modal shapes of the blades and carrying out modifications accordingly.

Brakes

Brakes are required for assisting clutch engagement on certain reversing transmissions and for arresting the continuously rotating shafting of gas turbine c.p. propeller transmissions to prevent wires entangling the screws when berthing, and for other emergencies.

The energy to be dissipated in stopping a shaft system, gearing, propeller and entrained water, and power turbine is such that in order to keep the brake to a suitable size, early single disc designs could only be used with the ship stopped in the water. A design study is in progress to develop an aircraft type multidisc brake using inorganic pad materials having a larger heat sink, this will have greater energy dissipation capacity for the same space and should be suitable for use when under way.

Lubricating Oil Systems

The *Whitby* and early *Leander* class gearboxes were plagued with commissioning problems due to dirt being left in the system during assembly. Modifications to the gearcase for the later *Leander* class allowed all internal piping to be straightened and blind ends removed. Proper flushing procedures and thorough inspection has reduced the number of initial and post refit bearing failures.

Since it is only possible to have small drain tanks in warships, the effect of lubricating oil aeration is noticeable. Steps are now taken in the system design to reduce aeration at source where possible, and to improve drain tank layout to give the best flow conditions to allow the air to be removed by baffles and perforated plates, as an example shown in Fig. 4. This is part of a major study into all aspects of lubricating oil system design which is in progress.

Since the scuffing experienced in the early *Whitby* class frigates, OEP69 (extreme pressure oil) has been used in all later classes. However, material changes and improved standards of production have made the use of this oil less necessary. As a result of shipboard trials the Navy Department Fuels and Lubrication Advisory Committee are now assessing alternative oils to OEP69 for use in present and future ships.

Gearing

There have been no recent material problems with gears in the Fleet. Since the introduction of carburized and hardened pinions and through hardened wheels, the *Leander* class have been free from tooth deterioration. The *County* class destroyers and *Tribal* class frigates' reversing gearboxes, which contain mainly carburized gear elements (with nitrided and induction hardened elements in certain cases) working at K loadings of 500 ahead and 820 when running astern, have also been completely satisfactory. This is adequate proof that these materials are suitable for present designs and the research in progress will permit even higher loadings with the utmost reliability in the future.

It is recognized that, due to the dynamic conditions felt by

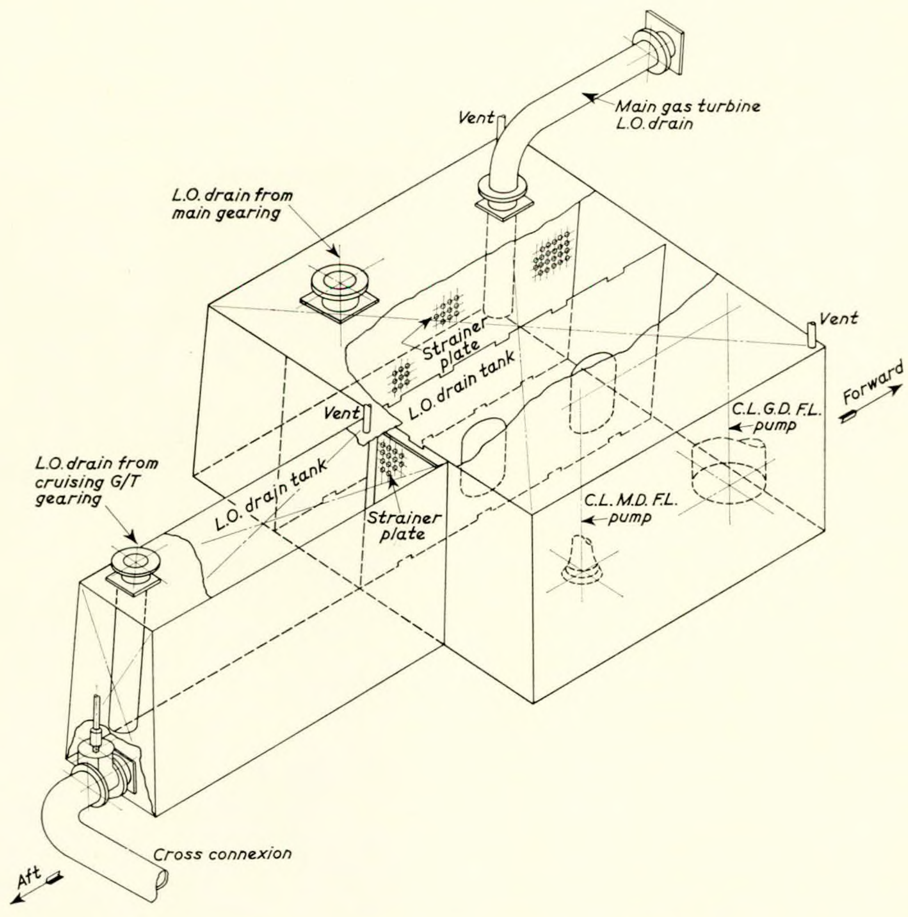


FIG. 4—*Lubricating oil tank design for Type 42*

the gears in a ship at sea, localized loading results so that in effect the gears are subjected to a much higher K value than the nominal design figure. A number of steps have been taken to minimize this:

i) gearcases for the later *Leander* frigates have been stiffened by the use of a double wall box construction;

ii) the correct tip, root and end relief is applied as a result of full scale back to back loading trials;

iii) seatings have been improved. The *Leanders* have three area support, one area around the thrust block and smaller areas at the forward end. The Type 82 destroyer has gearboxes supported by a hydraulic constant position mounting system developed by Y. ARD. Investigations are in progress to permit the use of anticlastic bearings, as used in bridges, as three point supports.

The object of all this is to divorce the gearing from its environment and remove those externally applied loads which upset the meshing, so that the factor of ignorance applied to full scale load trial results may be reduced.

In addition fundamental research is being carried out by the Navy and Vickers Gearing Research Association (NAVGRA) where the three principal suppliers of naval gearing co-operate to improve naval gear performance and quality.

Navgra

Since 1945 NAVGRA (then AVGRA) has been in the forefront of research into new materials, surface hardening and better manufacturing standards. In 1967 a new programme was

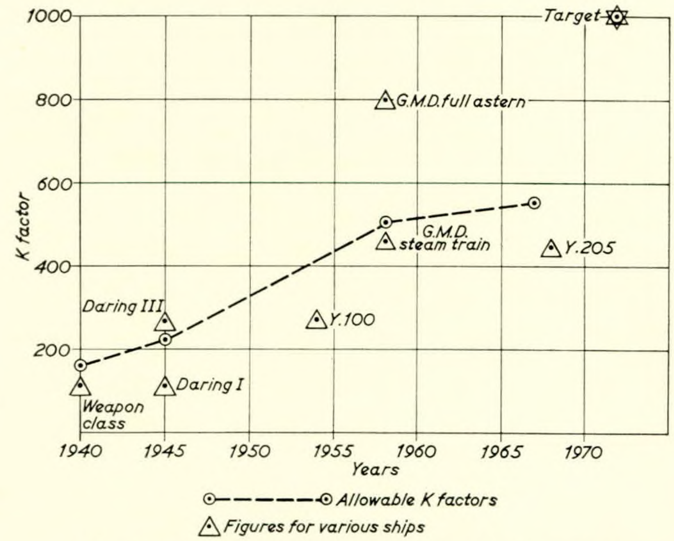


FIG. 5—*K loading in service and projected where*

$$K = \frac{load/inch\ face\ width}{pinion\ diameter}\left(\frac{1 + gear\ ratio}{gear\ ratio}\right)$$

(K is a comparative measure of the surface stress)

started to double the present limits of gear loading to 1000 K by 1972 in order that there may be more flexibility in design, smaller, lighter and cheaper gearboxes and better operating conditions.

The organization of NAVGRA was simplified so that under the council, which formulates the policy, is a Technical Committee charged with attaining the new aims. Responsible to this committee are the Design Committee, which co-ordinates all the work on material research, dynamics, structures etc., and the Production Committee which is required to initiate advances in production techniques.

The Design and Production Committees define areas of ignorance or required research and set up temporary panels of specialists from the firms and Ministry of Defence laboratories to detail the necessary work. The panels have a defined objective and a finite life and generate the work before being disbanded. The NAVGRA staff check the progress of the research, which is

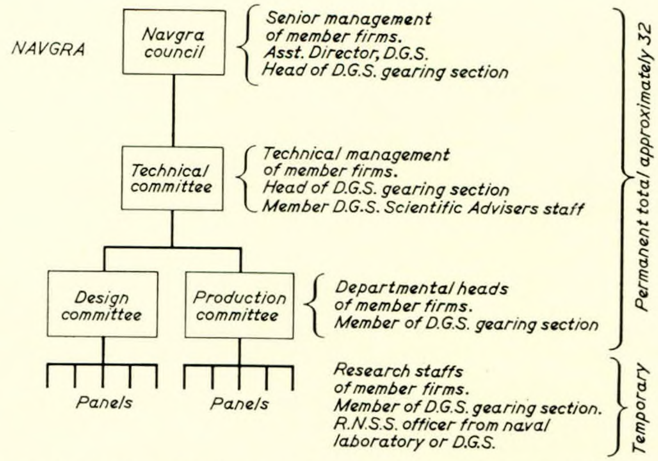


FIG. 6—*New committee structure of the Navy and Vickers Gearing Research Association*

conducted in the firms' laboratories, using up to date management techniques. Network schedules are used to ensure valid priorities and correct usage of finance and facilities.

As results of research are obtained, the Design and Production Committees integrate the work and formulate new design rules.

Research is at present conducted in the following fields:

a) *Materials*

A comprehensive surface and bending fatigue testing programme is underway to find the effects of variables such as heat treatment (times and temperatures), grain size, composition etc. on material properties. Methods of predicting residual stresses are also being investigated.

b) *Gear Dynamics*

Investigations are in hand based on earlier work at Cambridge and Sheffield Universities to provide a theoretical and

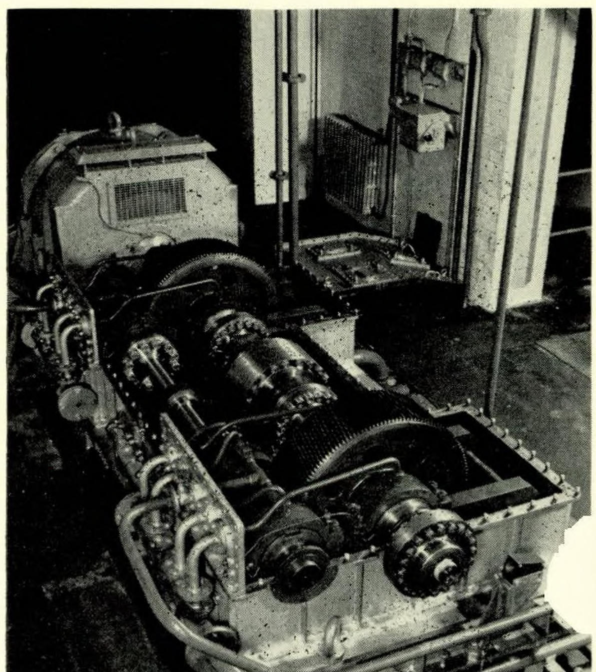


FIG. 7—*18 in centres gearload and noise test rig*

quantative understanding of the dynamics of vibration excitation, transmission and emission. This is being handled by advanced finite element computer techniques and rig tests will be used to verify the results.

c) *Structures*

In order to ensure that gears mesh at all times as they are designed to do without local overloading, a computer design process is being established. This takes account of ship dynamics and sets up the best structure for holding the elements without redundant material.

d) *Lubrication*

This research investigates the mechanism of lubrication as teeth slide over one another under the low and high speed regimes. In this way the fundamental reasons for scuffing may be established and verification by disc machines and full scale testing will be undertaken.

e) *Bearings*

As permissible gear loadings are increased, the journals get disproportionately large compared to the pinions under the present rules of projected area pressure. Research is in hand to develop a new design criterion based on minimum film thickness, taking into account dynamic conditions under load and temperature, in order that bearing loadings may be increased.

f) *Production*

It is necessary for the production to reproduce faithfully the designer's requirements. This is achieved by continuous dialogue between the Design and Production Committees.

The kinematics of the machine tools and the processes are being studied in order to reduce the inherent inaccuracies, decrease vibrations and provide better surface finish. New metrology equipment is being developed for tooth dimensions and transmission error.

Research has been initiated to consider the feasibility of producing gears by a more fundamental method than cutting. This would lead to improved load carrying and fatigue properties and should result in reduced costs.

New materials such as carbon fibres will be considered and the optimization of present production methods are under review in order that gears of the necessary accuracy may be produced at a competitive price.

NAVGRA gives the Royal Navy a stake in advanced industrial research and development, a most effective way of achieving progress. It has the incomparable resources of the member firms on which to call in both research and production. By 1972 NAVGRA will have established a load capacity equivalent to 1000 K. Each firm will thus be able to produce gears of the correct dynamic configuration required for the Fleet of the future.

Epicyclic Gearing

The NAVGRA research is applicable to both parallel shafted and epicyclic gearing. Epicyclic gearing shows to advantage particularly in the single engine per shaft configuration. However, in a multi engine installation the advantages of its small size cannot always be utilized.

The Brave class fast patrol boats have two epicyclic stages in series which use the well established Stoeckicht principle. These boats have been in service now for ten years under very severe conditions, and although there have been a number of transmission problems, the epicyclic stages have given no trouble.

The recent innovation of the flexible planet pin has permitted an increase in the number of mesh points and hence a reduction in gearbox size. This has recreated interest in epicyclic gearing within the Royal Navy.

The principle is being tested first on two auxiliary gearboxes in service at sea, a 40 hp electric driven compressor with high starting inertia and a 700 hp total energy gas turbine alternator where the gearbox space is very limited.

A design study is in progress for a main propulsion gearbox which may be back to back tested in order to confirm the criteria for this type of design.

The overall programme

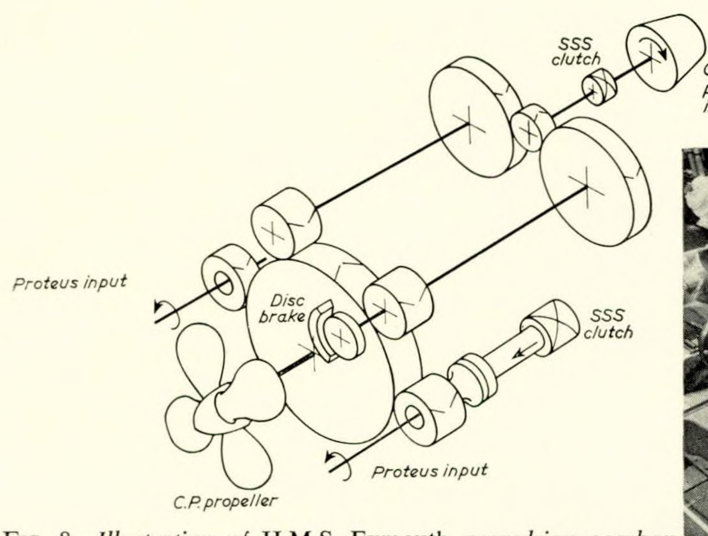
It has been necessary to deal only briefly with selected components in the transmission in the sections above. It has not been possible in the space allotted to describe the integrated programme necessary for all research and development or to indicate the relative importance of each item, the interdependence or the timescale.

Two recent designs, that of H.M.S. *Exmouth* the all gas turbine conversion and for Type 42 destroyer, now being built at Barrow-in-Furness, are discussed below to show how the results of the work outlined above are put into practice.

H.M.S. EXMOUTH

In 1965 the Navy Board took the decision to convert H.M.S. *Exmouth* into the first all gas turbine powered major warship in the Royal Navy, where the gas turbines were to be marinized versions of aero engines. The ship was not to be simply a trials ship but was to be able to take her place amongst the British Fleet in order to give operator experience with the gas turbines, the associated transmission, a large c.p. propeller and control system.

This paper is intended to describe how a transmission system



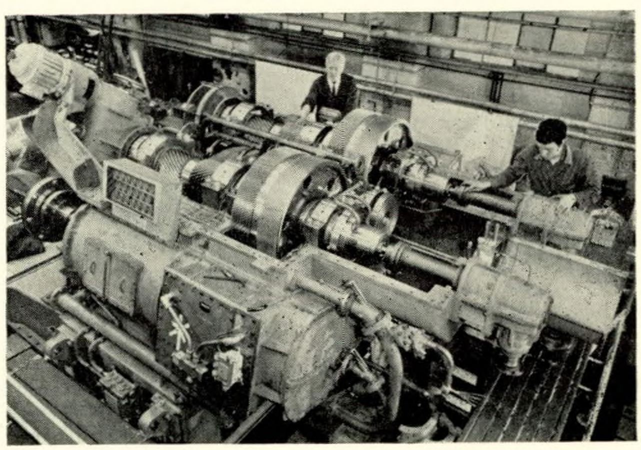


FIG. 8—*Illustration of* H.M.S. Exmouth *propulsion gearbox under construction and schematic layout of gear train*

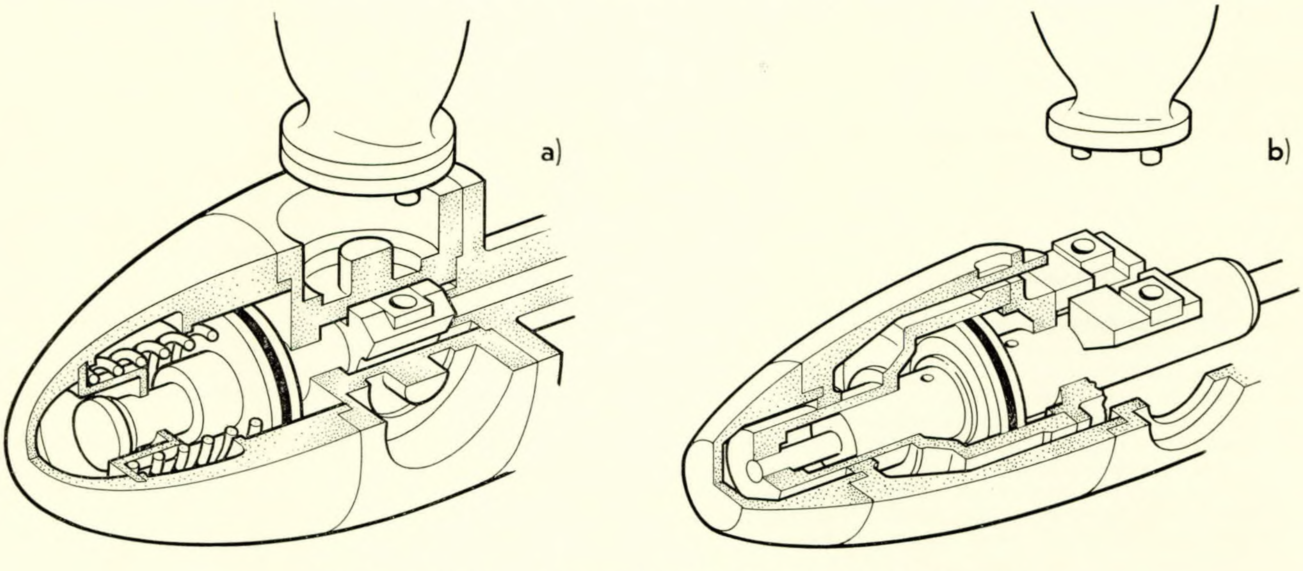


FIG. 9—*a) single acting c.p. propeller hub for* H.M.S. Exmouth
b) double acting c.p. propeller hub—Type 42

is chosen to fit a ship design. However, in many cases some constraint is laid on the designers and H.M.S. *Exmouth* is a case in point. The ship was originally built with steam turbine machinery on a single shaft. In order to meet a short time scale and keep to a suitable cost the conversion required that the new machinery should not take any greater machinery volume. The gas turbines are unidirectional and as reversing gearboxes are larger than the simpler unidirectional gearbox, a c.p. propeller was chosen.

The machinery is a single shaft COGOG (combined gas or gas) arrangement. Two marinized *Proteus* engines may be used singly or together as cruise and low power engines and for high power operation an *Olympus* engine is used.

Manoeuvring may be performed in any of the engines states, the transmission system permits changes from one to another without constraining the ship command.

With a short time scale it was considered that a conversion of a gearbox in service would be more acceptable than the development of a new design. The *Leander* class gearbox (a variant of the original gearbox fitted to H.M.S. *Exmouth*) was suitable for adaptation.

The input pinion is driven by the *Olympus* engine through a torque tube and a self synchronizing (SSS) clutch. The two *Proteus* engines drive through integral epicyclic gearboxes *via* torque tubes and SSS clutches to pinions which mesh with the main wheel as shown in Fig. 8.

A transmission brake is fitted to a secondary pinion line at the after end of the gearbox. This brake has a single disc of chromium plated copper and two air operated caliper units each with two pairs of organic pads. It stops the transmission and free power turbine rotor in under 15 s with engine at idling, propeller at zero thrust pitch and no way on the ship.

Experience has shown that complete action reliance cannot be placed on the supply of electrical power and so lubricating oil pumps driven from the transmission are a requirement. H.M.S. *Exmouth* has a gear driven positive rotary lubricating oil pump and two electrically driven pumps of the same type. These pumps may be connected either in parallel or in series. When in series the discharge of the motor driven pumps feeds the suction of the shaft driven pump to keep the total flow approximately constant at varying shaft speeds; the motor driven pumps are unloaded at high powers.

The controllable pitch propeller is the commercial single acting piston type shown in Fig. 9a. A double acting actuator is preferred for naval service but the time available for the conversion of this ship was shorter than that required to develop a new design.

The propeller hydraulic system is powered by two electrically driven screw pumps and one gear driven screw pump. The system is basically a constant pressure type operating at 500 lb f/in^2 with valve control and an open circuit. The supply and return oil to the hub is fed into the shaft *via* an oil distribution box at the forward end of the gearbox. This position for the oil entry is chosen since it permits input on the smaller diameter and therefore reduces the sealing problem.

The engine throttle demand is programmed with the propeller pitch. The pitch is increased with engine power up to 2000 bhp when the propeller reaches design pitch and it remains set for further power increases.

The shafting has to accommodate the larger overhung weight of the c.p. propeller and it has to have a sufficient bore to take the hydraulic fluid tubes. The shafting was redesigned to bending and shear stress criteria and the ship converted to take the increased size. The layout, however, has conventional plummer blocks and bulkhead glands, a water lubricated 'A' bracket bearing and a face type stern seal.

After conversion H.M.S. *Exmouth* did a long series of trials from Arctic to Mediterranean conditions. The object was to gain experience with the *Olympus* engine but it produced some worthwhile feedback from the transmission as well. The most important points are recorded below since they have affected subsequent design work.

Controllable Pitch Propeller and System

To prevent the possibility of loss of engines due to overspeeding, the control system is arranged so that the throttle demand is slaved to, and always follows, pitch demand on rising power.

With decreasing power, pitch and engine demands were simultaneously reduced. During early trials it was found that the system was not entirely satisfactory. Because of the effects of the hydrodynamic forces on the propeller blades on pitch reduction from maximum, the pitch angle reduced at a faster rate than that demanded by the controls, initially, and consequently the engine speed could rise to a level causing operation of the overspeed trip.

A modification to the control system was effected which cured the trouble. This comprised a low signal selection logical system which ensured that engine power was reduced in advance of pitch.

The c.p. propeller hydraulic oil temperature was low, particularly at start up since much lower engine room temperatures are experienced in gas turbine ships than in steam turbine ships. Since the hydraulic pump was initially "on line" started, this resulted in severe pressure surges which resulted in damaged pump seals. A period of running at low discharge pressure, as a temporary palliative, indicated that the system was working too close to the minimum possible pressure for satisfactory service; it was in fact possible to stall the system and this was partially responsible for the pitch reduction noted above.

As a result of the pressure surges and a sluggish relief valve the hydraulic system pressure peaked at over 1000 lb f/in^2. This caused stretching of some of the bolts in the hub, loosening of the piston nut and some damage to the seals in the distribution box. While this allowed some leakage from the hub and a knock on changing pitch it did not affect the operation of the mechanism.

The accuracy of the feedback mechanism presented problems in H.M.S. *Exmouth*. It appeared that the effect of ambient temperature and oil temperature changes after manoeuvring could alter the pitch by up to 1½ degrees. This was particularly noticeable when berthing and, while temperature compensation would have been possible, the late inclusion of extra complexity would not have been welcomed. The ability to apply the transmission brake enables the captain to prevent his ship creeping and can alleviate the problem. However, with experience the ship's staff can overcome the problem by anticipation and careful adjustment of the controls. This cannot be considered the complete solution. The simplest solution results from accurate specification of the setting up procedure so that, knowing the thermal effects from a number of tests, a bias may be applied to design zero thrust position.

Gearing and ancillaries

During ship trials the gears were deliberately over torqued by 15 per cent for 30 min. Subsequent inspections have shown that there have been no ill effects. This was the culminating act of a very creditable performance as the gearbox was designed, manufactured, delivered and installed in less than one year. This accolade is due to all the suppliers of the equipments in H.M.S. *Exmouth*.

It is considered essential that all gearboxes are given a spin test after assembly. Instrumenting the high speed bearings and certain casing positions can indicate malfitting, eccentricities, component unbalance etc. As a result of the short timescale this spin test had to be done with some components not in place, so it was not until the complete line was spun under turbine power in the ship that heavy vibration was discovered. This was traced to swash on a distance piece and was quickly corrected. This was however adequately convincing for probe tubes to be built into the gearcase to allow easier access to the high speed bearing for vibration checking. This was found to be invaluable at a later stage.

The original clutches were of the automatic disconnecting, lockable type. The automatic disconnexion was retained since barring of the power turbine is necessary. Had the clutch remained in the one way over running condition on an engine being barred the clutches would have engaged if the shaft was stopped. This would cause burn out of the barring motors on overtorque. The simplicity of removing the lock was recognized however, the ship was instrumented for trials and the results of these showed that the torque did become negative but the value was low. The spare main clutch was modified to remove the lock and provide alternative methods of producing the inputs to the machinery control and surveillance system. This was fitted and subjected to searching trials and appeared to be satisfactory although at sea there were indications of slight axial movement, under crash astern manoeuvres. The clutch was later stripped and was found to be in excellent condition.

Damage to the cruising engine epicyclic gearbox was traced to reverse torque such that the planet wheels could move across their pins blocking their bearing lubricating oil supply. This gearbox was designed for the engine in its aero role and had to be modified to accommodate the reverse torques.

Finally there has been bearing fatigue on the main gearing input line from the *Olympus* which has highlighted a phenomenon not previously experienced. During the trials H.M.S. *Exmouth* conducted pitch optimization experiments when she ran on reduced pitch and high shaft speed; such a condition, though less severe, also occurs when manoeuvring. This low torque state can effect the concentricity of the self synchronizing clutch, as this is basically an involute toothed gear coupling, and the running position of the pinion in its bearings. If there is additional unbalance, serious vibration can result. The bearing load is dependent on torque as well. Thus in the reduced pitch condition

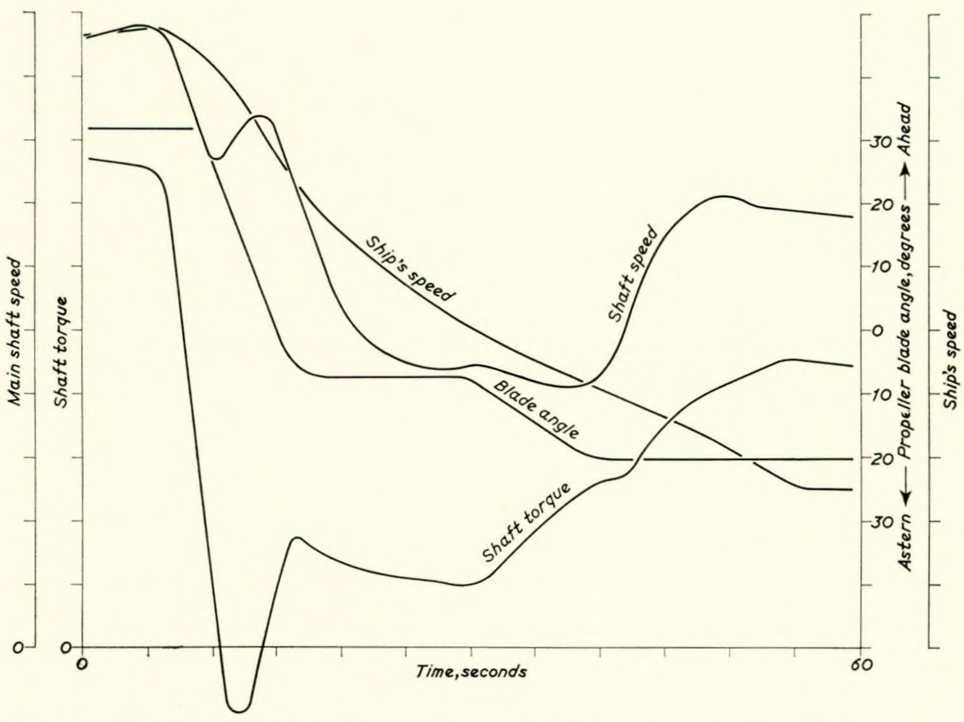


FIG. 10—*Manoeuvring trace from* H.M.S. Exmouth *during full power ahead to astern*

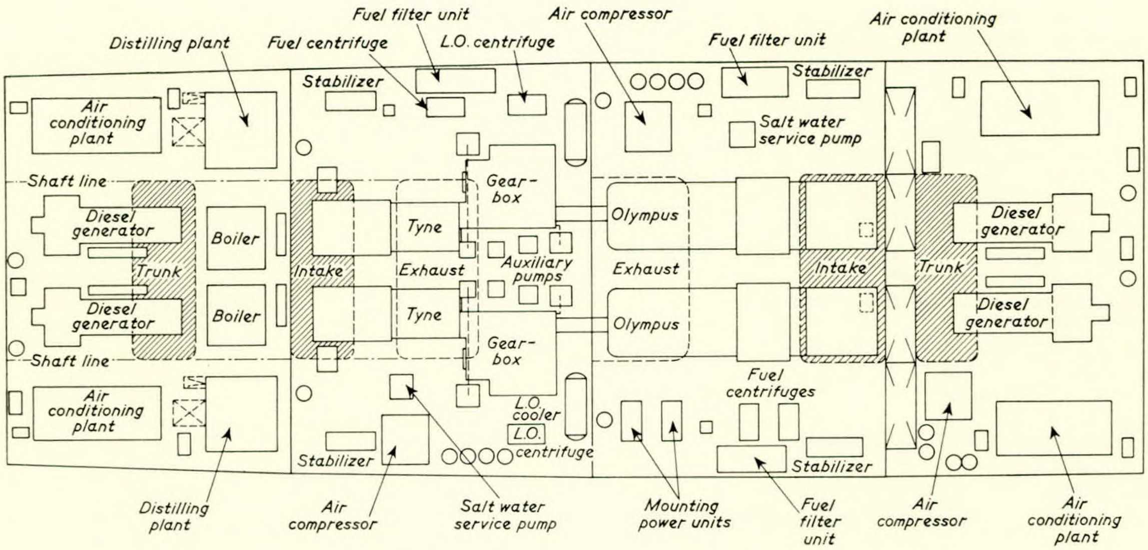


FIG. 11—*Type 42 machinery layout*

the bearings may experience a high alternating load, superimposed on a low steady load and fatigue conditions can exist. Vibration monitoring has been used to check acceptable conditions and it has been found that in H.M.S. *Exmouth*, providing the alignment of the axes of the centres of gravity of all high speed assemblies is satisfactory, in-place balancing has not been necessary.

H.M.S. *Exmouth* may appear to have been plagued with problems, and this is to some extent true. Nevertheless, the object of the ships conversion was to find out the problems in order that a change in propulsion policy could be implemented. These problems have been found and corrected and the lessons learned in H.M.S. *Exmouth* have been incorporated in Type 42 destroyer.

THE TYPE 42 DESTROYER

In November 1968 the Royal Navy ordered the first of a new class of guided missile destroyer designated the Type 42. The requirement was for an all gas turbine ship of high speed to provide air defence of the Fleet with secondary requirements in the anti-submarine and surface gunnery roles. The ship was to be an effective unit in war yet still be capable of the many peace time roles the Navy is called upon to carry out.

The technical requirements were for a machinery system to achieve maximum speed rapidly and to have a good endurance, while providing a significant saving in weight, space and technical manpower. Equipment was to be, where possible, removable for overhaul.

The initial feasibility studies showed that the required performance could be met using two shafts with one *Olympus* per shaft, but that a cruise engine would also be necessary. The marinized *Tyne* was selected and the engines are arranged to run separately i.e. COGOG. The mode of operation is that the *Olympus* is used for high powers and that the *Tyne* can be connected instead without constraint to the command. The machinery is controlled for cruising from a machinery control room, although some control is also available on the bridge. Necessary surveillance of systems is provided to produce a reduction in the watch-keeping task.

The two possible solutions of the reversing problem were considered: a reversing gearbox with f.p. propeller or a unidirectional transmission with a reversible pitch propulsor, and detailed preliminary studies were carried out with two of the schemes. In the ship design space was at a premium, particularly if the overhaul by replacement philosophy was to be adhered to, and the decision was taken to design the transmission with a unidirectional gearbox and a reversible pitch propeller. The transmission has therefore to solve the outstanding problems in the best possible manner so that the operator is not inhibited in any way.

The next section explains how this was done and describes the machinery selected for one shaft set.

The Propeller

At the time Type 42 destroyer was being designed the manufacturer of some 90 per cent of the Naval propellers produced a new double acting piston layout and this is shown in Fig. 9b. This design has a small hub diameter, which reduces the efficiency loss, and simplified internal design. Furthermore it subjects the blade palm pins to pure torque giving lighter loads in the palm bearings and simplifying the sealing arrangements.

Although completely new, this design was accepted for the Type 42 since it was based on sound engineering principles and a considerable background of experience in this field. Smaller versions of the design are being evaluated at sea in Ministry of Defence ocean tugs.

The piston is operated by hydraulic pressure passed down the shaft in tubes. The inlet to the shaft tubes is through a transfer box at the forward end of the main gear wheel shaft as in H.M.S. *Exmouth*. The selection of which tube is pressurized, and hence in which direction the blade moves, is achieved in the external hydraulic circuit. There is no volume change in the hub during operation so that the seals will not experience pressure variation.

The hub is continuously pressurized from a header tank in the machinery spaces to prevent water ingress. Water contamination of the main circuit, however, is unlikely since the main hub casting is separated from the main hydraulic circuit. Hub sampling is possible by overpressurizing the system which lifts the relief valve in the hub and this allows oil to pass back inboard for inspection.

A hand-operated pitch locking device is fitted and access to this is through the first inboard flange. The pitch locking is achieved by manually winding one tube in the shaft which moves a nut to lock the piston in the full ahead position.

The blades have been designed by the Admiralty Experiment Works. They will be cast in manganese aluminium bronze, profile machined and hand polished to a high accuracy. The maximum principal stress has been limited to 6 ton/in² to take account of the very rigorous operating conditions. The blades are to be attached by bolts to the hub and the whole hub will be faired off to give a good hydrodynamic shape.

The Hydraulic System

The propeller pitch hydraulic system is flow controlled with a variable displacement pump supplying fluid power to the actuator *via* a closed/boosted circuit.

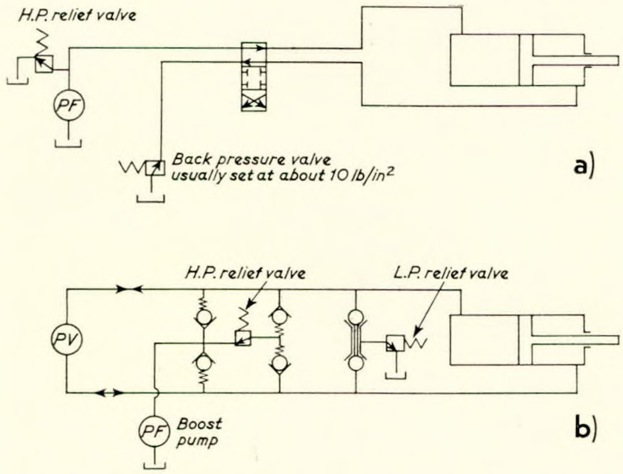


FIG. 12—*a) open circuit c.p. propeller system b) closed circuit c.p. propeller system*

This system was chosen as it has the following advantages:

1) the system pressure is only that demanded to perform a particular operation. During the majority of the time the system is only holding the pitch against the hydrodynamic forces and the pressure is likely to be of the order of 200 lb f/in^2.
With the majority of running at low pressure the system reliability will be high and the cooling requirements will be low;
2) The effect of the hydrodynamic forces on rapid reduction of pitch can be accommodated better than with an open circuit. Fig. 12a shows a simplified open circuit. It is clear that if the regenerative forces are high enough the actuator piston will be accelerated and if the pump flow is not adequate the demanded rate of change of pitch will not be achieved.

Fig. 12b shows a simplified version of the closed boosted flow controlled circuit being fitted to the Type 42. Under negative power conditions it will be seen that the low pressure leg of the circuit automatically switches to become the high pressure leg and *vice versa*. Provided that the relief valve is set high enough then the amount of fluid in the circuit (neglecting leakage) and its rate of flow, and hence the rate of pitch change, is determined solely by the pump speed and stroke and is independent of external forces.

The complete hydraulic circuit is shown in Fig. 13. Two main pumps are fitted, one gear driven, the other electric driven for start up and low shaft rev/min running. Boost pumps are driven off the main pumps. Each boost pump takes suction from the header tank and feeds the main pump *via* a filter and combined valve block, relieved oil returning to the header tank *via* the main pump casing and an oil cooler. Within the header tank return oil and supply oil are separated by a fine mesh gauze which extracts entrained air as oil passes through it.

The oil used is OM 33, a naval patternized hydraulic oil, but provision is made for running on OEP 69, the gearbox lubricant, in emergency. Due to the higher viscosity of the EP oil some system complication was necessary such as the fitting of a separate valve block for each pump. System safety and isolation is achieved by having automatic shut off valves in each pump circuit at the transfer box, held open by boost pressure.

Some leakage from the working circuit is inevitable, since the seals on the transfer box are more reliable with no back pressure and their drains are led back to a bottom ready use tank. Make up to the header tank is achieved by an auxiliary pump controlled by a float switch. Fluid cleanliness is assured by supplying oil to the ready use tank from storage tanks through a centrifuge.

Normal cruising will be done on the gear driven pump. The electrically driven pump is automatically cut in on loss of boost pressure and may also be started from the machinery control room when high pitch changing rates are required.

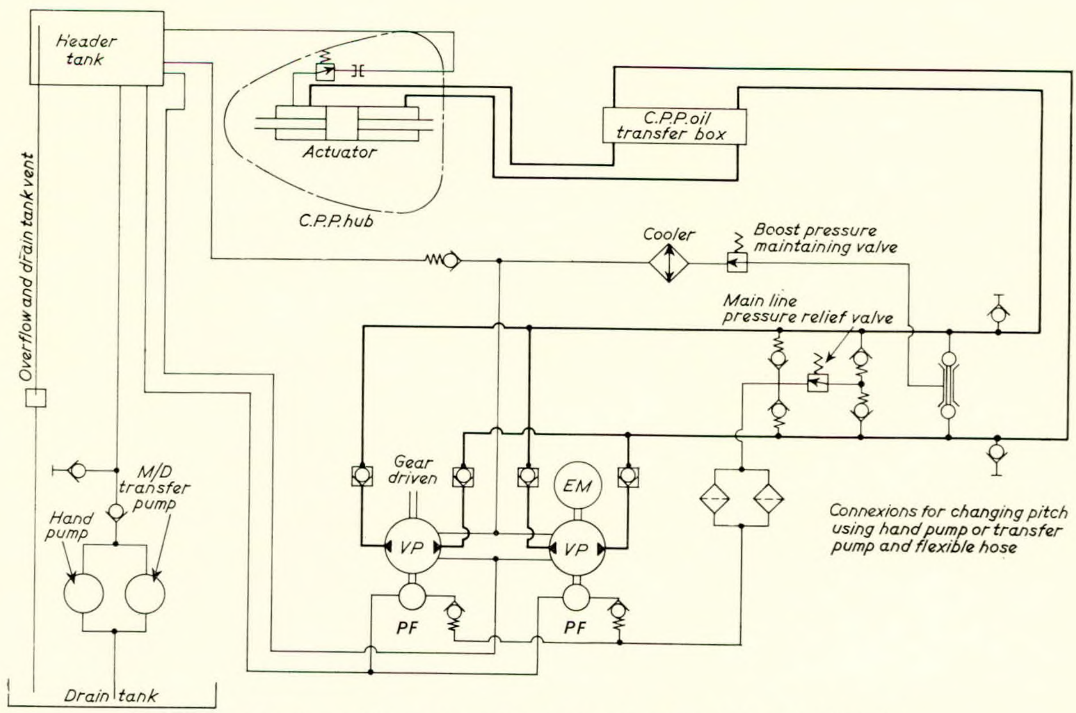


FIG. 13—*Type 42 c.p. propeller hydraulic oil system (simplified)*

The pumps are controllable swash axial piston type, commercial variants of a National Engineering Laboratory design. This type of unit has been in service with the Navy for a number of years, although in this size only as a motor. Since a hydraulic motor experiences variations of speed and pressure at full swash more arduous than its equivalent pump, this unit was selected subject to rig testing under controlled conditions in the necessary orientation.

A hand pump is provided for emergency pumping across of the hub. The swash plates are also manually operable on failure of the control system.

It is not intended to describe in detail the control system, sufficient to say the c.p. control function is integrated into the engine control system. A desired pitch output is achieved from single lever control of the complete engine and propeller system. This desired value is compared with the actual value which is fed back up the shaft by a compensated axial movement of the two hydraulic tubes. The error is used to actuate a stepper motor on the swash plate control.

The Shafting

The Type 42 shafting is shown in Fig. 3. Forged from 34/38 tons f/in^2 UTS steel, it is dimensioned to shear stress limits of 13 000 and 11 500 lb f/in^2 and bending stress limits of 5000 and 3000 lb f/in^2 for inboard and outboard shafts respectively, including stress concentration factors.

The shaft is held in two inboard pivoting pad plummer bearings and spherical glands are fitted at bulkheads.

Outboard the shaft is held in water lubricated bearings in two 'A' brackets. No stern tube bearing is required due to the proximity of the aftermost plummer bearing.

The bearing positions were established for the ship layout but each span was checked for vibration. Bearing heights were chosen from consideration of influence coefficients and shaft deflexion.

The water lubricated bearings are designed to the hydrodynamic criteria which keeps the length to diameter ratio of the bearing below 2·5, the ratio of full speed surface velocity (ft/s) to projected area pressure above five and the projected area pressure below 80 lb f/in^2. In the case of the Type 42 the shaft dimensions in way of the bearings were adequate to meet these requirements. Shrunk on gunmetal liners act as the mating surface with the bearings. The 'A' bracket bearing is bored to the shaft deflected line.

The shaft is in four sections, two inboard intermediates, one outboard intermediate and a tail shaft. Inboard connexions are by solid forged flanges and bolts. The aft intermediate to tail shaft connexion is outboard and an oil injected coupling is used. This coupling replaces two conventional loose couplings to make fitting easier and is of such a shape that it will have minimum effect on the wake pattern. The outboard shafting is protected by a resin bandage. The c.p. propeller is bolted to a flange on the tail shaft.

The stern seal is a face type seal and incorporates an emergency inflatable seal. This emergency seal is also suitable for limited running as a "get-you-home" device.

Shaft vibration characteristics have been checked for torsional and axial modes and no critical resonances are expected. Nevertheless a resonance changer device is to be fitted to the thrust block for the first of class.

The Gearing

The gearing design has to solve the problems associated with the position and the speed of the prime movers. The top speeds of the prime movers are widely different, the *Tyne* at 13 500 rev/min and the *Olympus* at 5600 rev/min. Rather than a single complex gearbox it was decided to have a main gearbox and a separate primary gearbox associated only with the *Tyne* engine which brings its speed down to a level acceptable to the main gearing.

The Primary Gearbox

It was decided to place the gearbox on the same mounting frame as the *Tyne* power unit as a module, the bed plate being mounted flexibly from the ship's structure. The *Tyne* free power turbine drives into the gearbox *via* a torque tube and flexible couplings. The output is taken to the main gearbox *via* a torque tube and membrane couplings which have to accommodate misalignment between the flexibly mounted engine module and the rigidly mounted main gearbox.

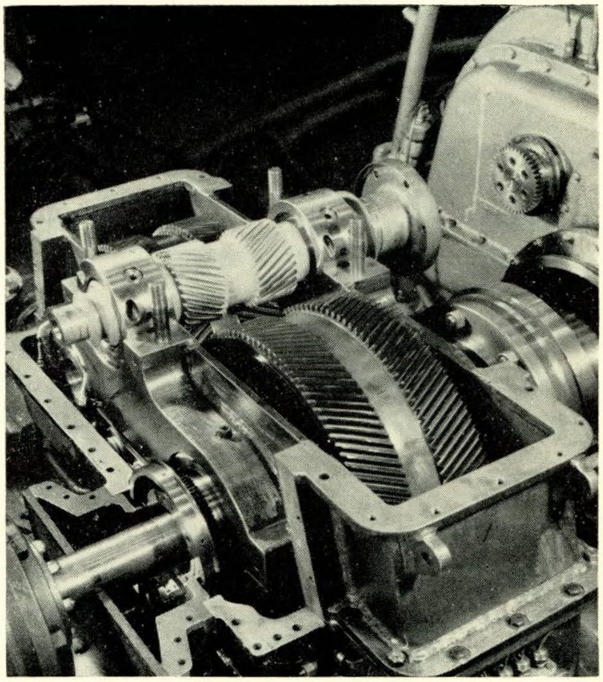


FIG. 14—*Tyne gearbox*

Unlike the *Olympus*, the *Tyne* power turbine is not handed so that an alteration of the sense of rotation for port and starboard main shafts has to be achieved in the primary gearbox. This is done by placing an idler in the train for one primary gearbox without altering the disposition of the input and output axes.

The gearing is arranged as a single reduction train with the pinion meshing directly with the wheel for the starboard set and *via* the idler for the port set although they appear the same externally. The gears are double helical, basically of involute form, hobbed and ground to a standard better than BS 1807 Class A1. The gearing is shown in Fig. 14.

The pitch line velocities of these gears are outside normal naval practice being of the order of 23 000 ft/min. Considerable thought was therefore given to the dynamic conditions. This resulted in special design criteria for the gear case (to avoid oil churning and impact effects), the bearings and the appropriate tip, root and end reliefs. The reliefs are applied to the pinion in the starboard box and to the idler in the port box.

The gearcase is of single wall welded steel construction, designed for maximum rigidity. The bearings are of medium thickness white metal held in a steel backing ring.

Certain module auxiliaries are driven from an auxilliary spur tooth train, these include speed signal and tacho generators and a barring motor for the power turbine. The barring motor in this case is an air operated diaphragm type where a pecker engages with a toothed wheel. This device can be stalled without damage, cannot be oversped and is very simple with a short axial length. The provision for stalling is required since in normal operation the self synchronizing clutch can overrun in one direction only, it will therefore engage when the main gearing is stationary and the *Tyne* engine is barred in the forward direction.

Lubrication is supplied through a single connexion with a fine control valve from the main forced lubrication system to rails. A single rail supplies all bearings of the same speed, and a rail also supplies the "into mesh" sprays. A large bore drain is provided to prevent oil level build up.

Main Gearbox

In order to fit the inbut shaft positions the gears have been "rolled" around the main wheel as shown in Fig. 15. The gear designers have put both inputs into the same pinion. In this case, having matched the input speeds by the gear ratio in the primary gearbox, the main gearbox is not affected by which prime mover is driving and none of the gears rotate unloaded.

The gearing is a double reduction, dual tandem, locked train arrangement. The common input primary pinion meshes with two primary wheels located either side of it. Each primary wheel is connected to a secondary pinion through a quill shaft, and each secondary pinion meshes with the main wheel. The primary train is located aft of the secondary train.

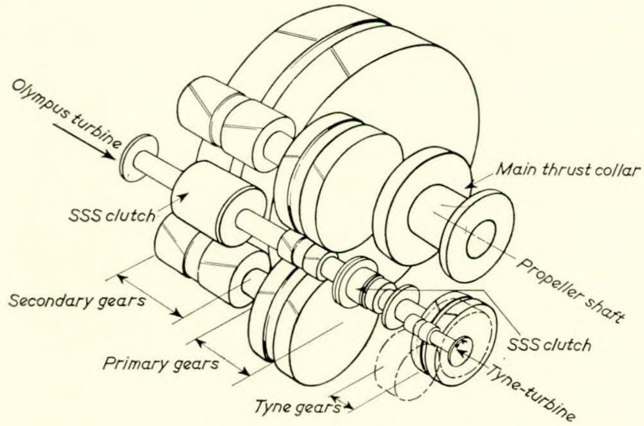


FIG. 15—*Rotating elements of Type 42 main gearbox*

The gears are double helical, all gears being finished to a standard better than BS 1807 Class A1. The pinions are solid En 36a carburized, hardened and finish ground. The primary wheel rims are En 40b nitrided, but, since this is a low distortion method of hardening, they are not ground. The rims are welded to side plates which in turn are welded to the shafts. The main wheel rim is hobbed of through hardened En 30b and is bolted to side plates which are bolted to flanges on the shaft.

Tip, root and end relief will be applied to the pinions. Helix correction is not required since at this level of loading the axial movement and subsequent redistribution of load from one helix to another is adequate. Had the gears been single helical there would have been a problem of torsional wind-up of the primary pinions since, if helix correction was applied to suit the *Olympus* input, it would have been the wrong way round for the *Tyne* input.

The rotating elements are held in medium wall white metal bearings, each carried in a separate steel sleeve and made in halves. To permit alignment modification, provision is made to adjust the position of one bearing per mesh. This facility is not used during manufacture but is available if required during refit or maintenance operation. The primary pinion is offset from the line of the wheel centres in order to stabilize the pinion in its bearings.

The quill shafts between primary wheels and secondary pinions pass through each, coupled rigidly at the forward end to the secondary pinions and *via* a fine tooth coupling to the primary wheels. The end of the quill shafts are used to drive various auxiliaries.

Torque tubes with sealed ends and non-lubricated flexible membrane couplings take the drive from the *Olympus* engine and the *Tyne* module to the gearbox. The permitted angular misalignment of these couplings of 20 minutes of arc continuously and one degree transient is adequate for the dynamic deflexions of the input line.

The thrust block is integral with the gearbox, and by having the primaries aft and placing the thrust block below them, a saving in gearbox length is possible. The thrust block is a conventional tilting pad type with drowned lubrication and ahead and astern hydraulic thrust measuring equipment is fitted.

The gearcase is of welded steel construction to give maximum rigidity with minimum weight. It is split at the plane of the main shaft and the plane of the quill shafts, the major part being of double wall box construction. Light alloy inspection plates are fitted above each mesh and clutch. All plates are designed to avoid critical resonances in the running range. Three area support is provided for good isolation, with the main group of bolts

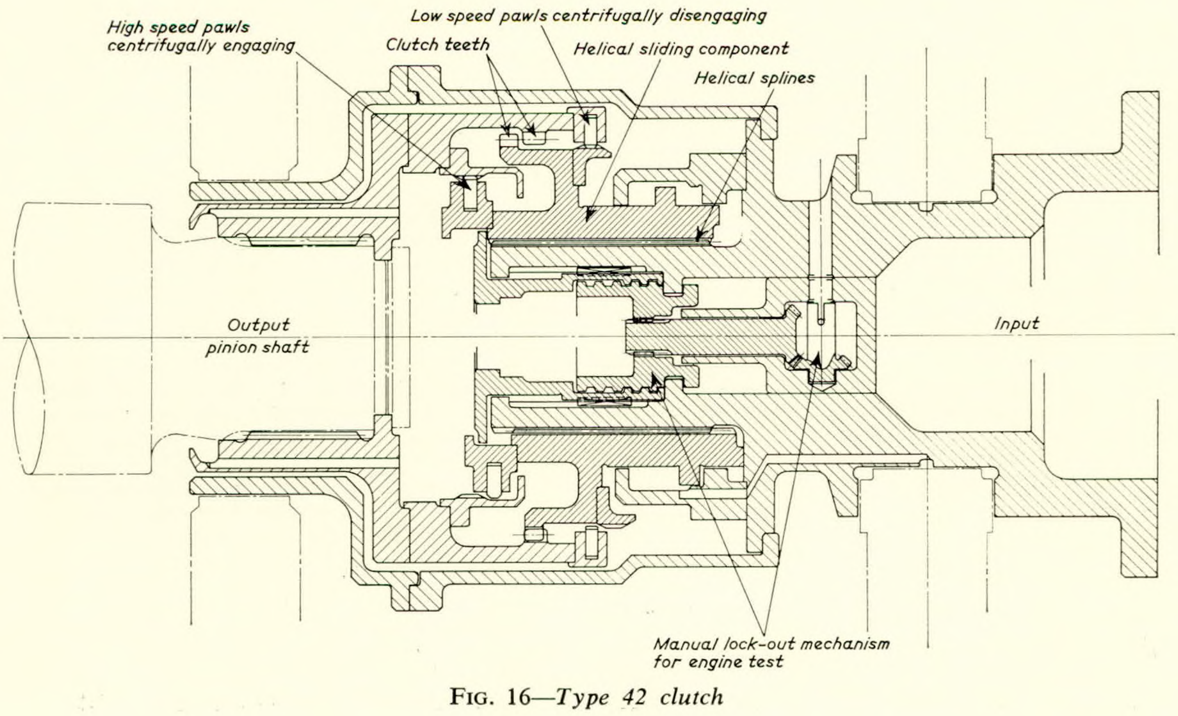


FIG. 16—*Type 42 clutch*

around the thrust block and areas either side in the plane of the main wheel. The ring main piping is inside the gearcase where possible and provision is made for visual inspection. A rail supplies each set of bearings at the same speed and a separate rail supplies each spray. Tuning is possible by fine control minimum closing valves.

In order to permit the engines to be connected or changed over without constraint to the ship's operation, self synchronizing clutches are fitted in each input line. The principle of these has been explained elsewhere, but in normal operation these clutches when disengaged remain in a one way overrunning condition, since with unidirectional shafts there is no requirement for automatic disconnexion. Two rows of pawls are fitted, one high speed set on the input side which centrifugally engage and a low speed set on the output side which centrifugally disengage. Both sets of pawls are inert when the clutch output is at high speed and the input is stopped or at low speed. Engagement is achieved by the low speed set of pawls to start the shaft from rest and by the high speed set at synchronizm when the shaft is rotating under the action of the other engine. A manual method of complete disconnexion is available to permit engine test without rotating the gearing.

In order to simplify the clutch and its controls, it has been designed for, but not with, a lock. It is difficult to predict precisely the amount of reverse torque likely in a new design of two shafted ship, particularly during turns. Evidence from H.M.S. *Exmouth* and other applications indicates that with a properly designed dashpot disengagement is not likely with short transients. The Type 42 is being fitted initially without lock but for the first of class provision is made to accommodate a lock mechanism which can be added simply should trials indicate that this is necessary. An advantage of omitting the lock is that the line vibration will be reduced as the clutch locking sleeve, not being a torque loaded member, is not self-centring such that it could run out of true within the manufacturing tolerances of its sliding clearances.

A low speed shaft turning gear is provided to turn the gearing and shafting, this drives into one quill shaft. On the other quill shaft the brake is mounted. The torque requirement in the Type 42 is higher than that of H.M.S. *Exmouth* so, as there is a limit to the brake disc diameter due to the proximity of the *Olympus* torque tube, three pairs of calipers are fitted around a chrome plated copper disc of the same size as in H.M.S. *Exmouth.* The control system builds the brake torque up slowly preventing pad burn out when first applied at the highest rubbing velocity. The brake can dissipate 3×10^6 ft/lb of energy, stopping the shaft in less than 15 s with a torque build up time of 4 s.

It is a requirement of the Type 42 to be capable of full propulsion machinery operation in the event of loss of electrical supply to propulsion auxiliaries. A main lubricating oil pump, c.p. propeller hydraulic pump and a sea water circulating pump are driven from quill shaft ends through shafts and bevel drive gearboxes. Electrical driven standby pumps are also fitted for when the shaft is stopped or at low rev/min.

There is a problem in sizing gear driven pumps to cover the speed range of the ship. In the case of the lubricating oil pumps the gear driven one, which is positive displacement, has been sized so that at low rev/min an electrically driven centrifugal pump is matched to run with it to give adequate output. In the event of electrical failure at low rev/min an emergency air motor will automatically cut in to drive the centrifugal pump until the electrical supply is re-established. If at high shaft rev/min it is found that the surplus flow from the gear driven pump, when discharged through the relief valve, causes excessive aeration, then a hydrostatic variable speed drive may be fitted.

The c.p. propeller pump and sea water circulating pumps are not so critical. The effect on pitch changing times of the gear driven c.p. propeller pump running alone at lower powers is imperceptible since with variable delivery pumps it is permissible to have them oversized. The sea water pump has the ship's salt water main in reserve.

The complete gearbox will be tested at light load up to full speed after manufacture, the vibration level being checked to ensure all components are correctly assembled. Vibration checks will also be taken when the complete line is assembled in the ship, driving on gas turbines with the propeller in zero thrust condition and also during sea trials. A provisional vibration acceptance level of 0·03 in/s (RMS) has been set but this may well be modified due to the interactions of the stiffnesses of the seatings which can have a large effect on the vibration response.

The lubricating oil system has two pumps in parallel supplying E.P. oil through a cooler and filters. The cooler is single pass with the water in the tubes and a thermostatic oil-side bypass. There are three filters in parallel, each with a felt cartridge with a nominal filtration rating of 35 μin. The supply then splits to the main and *Tyne* gearboxes and the *Olympus* power turbine, the *Olympus* gas generator and *Tyne* engine using a synthetic oil from separate systems. Separate drains are led back to the drain tank (shown in Fig. 4). The drain tank is "L" shaped, around the c.p. tank which also has to be positioned in this area to fit the pump position and the gearbox off take drives. The returns from the main gearbox, the *Olympus* power turbine and the system reliefs are led into the large volume and the *Tyne* gearbox into the wing. The flows pass through air restraining baffles before they combine and pass through another baffle to the pump suctions. Air vents are placed by each baffle. Despite the limitation of tank sizing, giving less than three minutes dwell time, the flow pattern which is set up is expected to reduce the air carryover to the pump suctions.

Each component will be flushed and sealed before delivery to the ship yard. The completed system will be flushed, using a portable pump and filter unit, in sections to give the maximum oil velocity, while bypassing the major components by flexible hose. The system has been designed to have no dead ends during flushing.

A de-humidifier is fitted on the main gearcase to prevent corrosion. It takes suction from the vent pipe and passes de-humidified air into the base of each gearbox. Controlled by a humidistat, it will be in use during the ship building stage and at refits. While the ship is in service the de-humidifier will be in use whenever the lubricating oil pumps are stopped.

General

The design of the Type 42 transmission system has attempted to solve in the best way the problems associated with a warship propelled by gas turbines. Although much new thinking has been necessary the selection of each equipment and sub-system has been based on a knowledge of what its requirements are, and what previous experience there has been in service or in the research and development field. Many new philosophies have emerged, some as a direct result of the trials in H.M.S. *Exmouth.* It has been necessary to consider a complete system, taking into account the interaction of each component or sub-system on the whole.

CONCLUDING REMARKS

Although this paper is written about warships it is hoped that the content will be of interest to those engaged in the commercial field confronted with similar studies. It is regretted that, due to the allocation of space, only superficial treatment of some aspects has been possible.

ACKNOWLEDGMENTS

The authors wish to state that although Ministry of Defence permission has been granted to give this paper, the opinions are those of the authors and do not necessarily constitute Naval policy.

The authors wish to acknowledge the influence of the previous heads of their section and to thank the various members of the staff of the Director General Ships and the following for their assistance in the production of this paper and over the last four years: W. H. Allens, Boulton Paul Aircraft, British Ship Research Association, Crane Packing, David Brown Gear Industries, Dowty Group, Dunlop Aviation Division, English Electric/AEI, Flexibox, Fluidrive, Hawker Siddeley Dynamics, Navy and Vickers Gearing Research Association, Rolls Royce, SSS Gears, Stone Manganese Marine, Vickers and Y.ARD.

REFERENCES

1) WEAVING, P. D. V., and SAMPSON, W. H. 1963. "Progress and Development in Naval Propulsion Gears", 1946–1962. *Trans. I.Mar.E.,* Vol. 75, p. 73.
2) RIMMER, R. F., and LIDDELL, R. E. 1969. "Design of Lubricating Oil Systems for Marine Propulsion Gears". Imas Conference, Section 4, p. 25.
3) CARR, J. and MARTYN, D. K. 1969. "Design Aspects of Propeller Shafting". Imas Conference, Section 15, p. 35.
4) GOODWIN, A. J. H., IRVINE, J. H. and FORREST, J. 1968. "The Practical Application of Computers in Marine Engineering". *Trans. I.Mar.E.,* Vol. 80, p. 209.
5) KLAASSEN, H. and ARNOLDUS, W. 1964. "Actuating Forces in Controllable Pitch Propellers". *Trans. I.Mar.E.,* Vol. 76, p. 173.
6) BOWERS, N. K. 1966. "Gas Turbines in the Royal Navy". *Jnl. Naval Engineering,* Vol. 16, No. 2, p. 194.
7) CHAPMAN, E. R. 1969. "Type 42 Destroyer", *Jnl. Naval Engineering,* Vol. 18, No. 2.
8) CRUMP, J. R. 1966. "H.M.S. *Exmouth* Conversion". *Jnl. Naval Engineering,* Vol. 16, No. 3.
9) TREWBY, G. F. A. 1962. "Marine Gas Turbines in the Royal Navy". *I.Mar.E.* International Conference. Vol. 74, p. 347.
10) DUNLOP, J. M. and GOOD, E. B. 1963. Machinery Installations of Guided Missile Destroyers and General Purpose Frigates. *Trans. I.Mar.E.,* Vol. 75, p. 11.

Discussion

CAPTAIN R. M. INCHES, R.N. (Member) said he was not opening the discussion because he was an expert in transmission design. Probably his closest connexion with it was that he worked for some years across the passage from the Inspector of the Transmission Design Section in the Ship Department. Power transmission problems occurred widely in engineering and thus the authors had a great deal of ground to cover.

Their decision to be selective was in his view entirely justified and he also felt that on the whole their selection had been good. He would have liked to have heard more about work on the noise reduction problem, both as regards propeller noise with its concomitant threat of cavitation erosion, and gearing noise with its adverse effect on engine room personnel and habitability generally. There was no mention of research work on lubricants, but he felt sure it was being carried out under NAVGRA or other auspices. Would the authors comment on those two aspects?

Membrane flexible couplings were mentioned, and one gained the impression that they were attractive because they were less likely to give trouble than the fine tooth type. It might be that the name itself was an indication of how those couplings worked. Would the authors enlarge on this?

Finally, he noted a reference to the damaging effect of dirt and the need for special procedures to ensure getting rid of it. This need was mentioned so frequently nowadays that he suggested it was time that those concerned declared the elimination of all unwanted matter from machinery and its associated systems normal procedure, and called it a special procedure when one did not mind if dirt did get in.

MR. W. M. BROWN, B.SC. (Associate Member) said the problems associated with main propulsion transmission systems in both warships and merchant vessels had occupied a great deal of attention in recent years due to their undoubtedly complex and troublesome nature. A paper ranging over the many considerations involved would be a valuable addition to the Transactions of the Institute. It was of great personal interest to him since his company designed and installed complete transmission systems and additionally manufactured the majority of the components involved.

The authors had shown what advances had been made in recent years and also indicated that further advances could be expected as a result of research now in progress.

He had been privileged to have been closely associated with parts of the work described, but found this somewhat inhibiting when attempting to offer critical comment. He could only plead more recent personal knowledge and experience as justification for any comments which appeared to conflict with the authors' views and for the different application solutions he proposed to offer.

He intended to confine his remarks to system design and main propulsion gearing. The two could not be separated because the gearbox was virtually the nut between the nut-cracker and it was becoming ever more important not only for the gear designer to understand the environment in which his equipment had to operate, but to consider its design, manufacture, installation and servicing as an integral part of a perfectly matched transmission system.

Dealing with system design, under that heading of their paper the authors had confined their remarks to a comparison of the application of controllable pitch and fixed pitch propellers with unidirectional prime movers and had considered other system design factors under their discussion of component choice. It was possible to adopt a wider interpretation and classify the complete system under two distinct sub-sections, i.e.:

a) ship propulsion dynamics;
b) mechanical design of system.

Ship propulsion dynamic studies were conducted to determine the most appropriate type of transmission system for a particular ship type and could compare the performance of c.p. propeller systems with reversing transmissions employing f.p. propellers. Such studies required a consistent set of ship and propulsion plant characteristics. The equations governing the response of the ship and propulsion plant were complex nonlinear differential equations for which advanced digital computer techniques were becoming available to obtain realistic numerical solutions. The results could be expressed as curves which indicated the performance required of the machinery in order to achieve a required ship operational performance. For powers in excess of say 20 000 hp such studies tentatively indicated the use of f.p. propellers and certain American organizations had accordingly committed themselves to reversing transmissions. G.T.S. *Callahan* was a notable example. Could the authors say if they employed such techniques and comment on their experience of comparing computer simulations with measured results? Furthermore, while they had quite rightly adopted a non-committal generalized position regarding c.p. and f.p. propellers Mr. Brown welcomed their views on installations requiring propulsive powers in excess of 20 000 shp.

Having selected the type of transmission for a particular ship the purely mechanical system design could be undertaken. The basic problem was to design a system capable of transmitting the torque from the prime mover output flange to the propulsor and be capable of carrying the propulsor generated thrust to the thrust block. Other requirements were the generation of acceptable torsional and axial vibration characteristics between the two ends of the system, with adequate support so bearings were not overloaded nor unacceptable transverse vibrations generated. At the same time the system required to possess sufficient transverse flexibility to accept the most extreme hull movements.

The basic problem of torque carrying capacity and thrust carrying capability did not in general present a problem but the other aspects were worthy of comment.

Several well established methods existed for determining the axial and torsional vibration characteristics of geared shaft systems. The main difficulties in calculating the response characteristics were not the numerical solution but in assessing the principle elements in the vibrating system particularly in the case of axial vibrations. The torsional system consisted of all the rotating parts and was well defined, but when axial vibrations were considered the system included non-rotating parts such as the hull structure, seating stiffness, thrust block stiffness and its position. For merchant vessels a reliable empirical equation had been established for estimating seating stiffness at the design stage. What method was used in warship design and what was its potential accuracy?

The stiffness of the well known thrust blocks produced by the Vickers Group had also been measured, and consequently, when separated from the gearing reliable stiffness data was available at the design stage. When the thrust block was integral with the gearing it was difficult to provide an axially stiff structure due to the discontinuity in way of the main wheel. Had the stiffness values the authors obtained with the integral thrust designs inspired their remarks about changing to separate thrust blocks?

Mr. Brown was also interested to note the authors were considering positioning the thrust blocks well aft. Some 12 years ago his company positioned the main thrust block 80 ft aft of the main gearing in a large passenger liner in order to generate acceptable axial vibration characteristics. Studies conducted since then had indicated that there was an optimum position for it in the shaft line, but to determine this position each case required careful consideration because it was dependent on other system parameters. Consequently the authors could find it difficult to realize their aim in all applications.

All vibrations required an exciting mechanism, and in the case of geared turbine installations the propeller predominated, setting up exciting forces in torsional, axial and transverse modes. The magnitude of such forces depended on many factors and a great deal of work was being undertaken to determine accurate data. This aspect of transmission design was so important that the value of the paper would be considerably enhanced if the authors could have given some indication of any work in this area with which they had been associated.

So far as the transverse flexibility of warship systems was concerned, the authors had shown an interesting deflexion diagram in Fig. 3 and while this clearly indicated the necessity of slope boring the stern tube Mr. Brown wondered if the authors had given consideration to positioning the axis of individual tunnel bearings along the deflexion lines to reduce end loading. Furthermore, Fig. 3 seemed to indicate a deflexion of 0·03 in across the main gear wheel, and their comments on the likely consequences at the gear mesh would be appreciated.

With regard to gearing it was probably true to say that British Naval gears were the most highly loaded marine main propulsion gears in the world and it was particularly gratifying that the authors had sufficient service experience to demonstrate that these levels were adequate for current designs. It could, perhaps, have been more clearly emphasized that the 820 K application persisted for very small periods of time. However, although no mention was made in the paper, a new class of gear was nearing prototype testing at Vickers, Barrow Engineering Works with continuous running loadings in excess of 500 K and would, in fact, be the highest loaded second reduction gear unit in the fleet. It was also emphasized that such levels of loading were extremely reliable, having a factor between full scale test rigs and service of the order of 2·5. With the system analysis techniques which were now becoming available, environmental factors would become clearer, and it would be possible to reduce service factors with confidence.

Mr. Brown agreed with the authors' implication that the success of naval gears since World War II could be undoubtedly attributed to the work of the Navy and Vickers Gearing Research Association. They had also indicated a target of 1000 K for service gears by 1972. This aim, although simply stated, encompassed many considerations on which the authors had briefly touched. So far as the gears themselves were concerned, Mr. Brown desired to offer some amplification. To determine the loading limitations of a pair of gears it was necessary to consider simultaneously, three criteria:

i) the bending fatigue limit of the tooth;
ii) the Hertzian stress limit of the tooth;
iii) the load carrying capability of the oil film between the tooth flanks.

In Fig. 17 these three considerations were brought together in a single diagram based on the leading dimensions of the second reduction gears of the first all-British surface hardened gear design (YEAD I) built by Vickers some years ago. The lower loading diagram was based on current design limits while the higher levels referred to the so-called 1000 K targets. From this

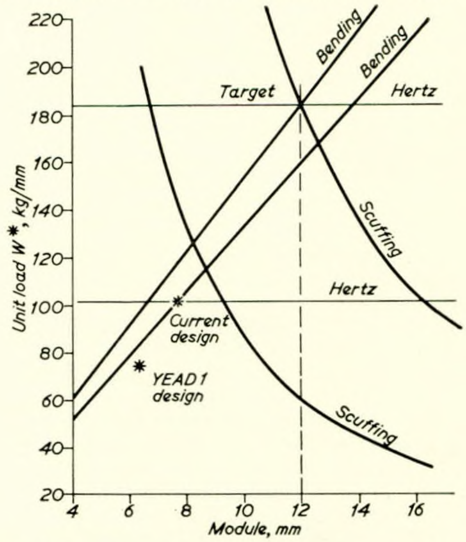


FIG. 17—*Gear performance diagram based on Yead 1 secondary reduction gears—Pinion P.C.D. 11·805 in—Wheel P.C.D. 83·159 in*

diagram the following observations were immediately apparent:

1) with current design levels, scuffing did not present a problem;
2) to achieve optimum surface load carrying capacity the selection of the right pitch was a pre-requisite;
3) increasing Hertzian stress levels to 1000 K could be achieved without the use of excessive pitches and a reasonable maximum limit had been set at 12 mm module (2 dp);
4) within these constraints it was clear that research must be directed not only to improve Hertzian load carrying capacity, but also to improve bending fatigue properties and the associated lubrication problem. This illustration helped to put the research programme outlined by the authors in better perspective and so far as the YEAD I design was concerned the following power improvements would result without any increase in size:

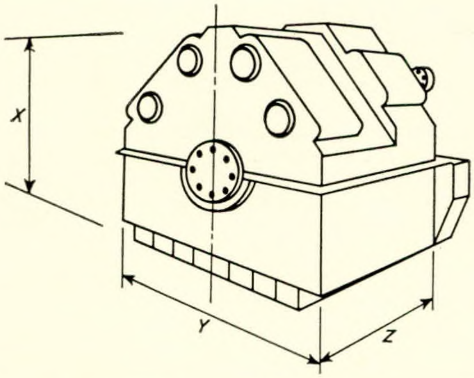
Original design	30 000 hp
Current design rules	39 000 hp
1000 K target	71 000 hp

Such improvements would require improved production techniques to which the authors had made reference. However, as a gear manufacturer Mr. Brown thought he must say that the authors' remarks about improved methods and competitive prices were surely directed to the new production techniques which could be required, and he hoped they would agree that current designs were provided at competitive prices and the quality attained was already far in excess of what was required by merchant marine applications.

Mention had been made of flexible pin epicyclic gears. Hicks* described the principle of that system to the Institute in 1967 while Mr. Brown† showed to the Institute last year the

* JONES, T. P. August, 1967. "Fifteen Years' Development of High Powered Epicyclic Gears." *Trans. I.Mar.E.*, Vol. 79, p. 294. Contribution by Hicks, R. J.

† JUNG, I. K. E. May, 1969. "Steam Turbine Machinery." *Trans. I.Mar.E.*, Vol. 81, p. 151. Contribution by Brown, W. M.



Design	X in	Y in	Z in	Volume of envelope of rotating elements
A Continental design specification	103·9	109·5	71·4	100 per cent
B Vickers specification to Royal Navy standards	82·5	81·8	66·3	55 per cent
C Vickers specification for parallel shaft primary and epicyclic secondary gears	61·2	84·0	50·0	32 per cent

FIG. 18—*Foreign warship design comparison based on conventional twin input locked train gear configuration*

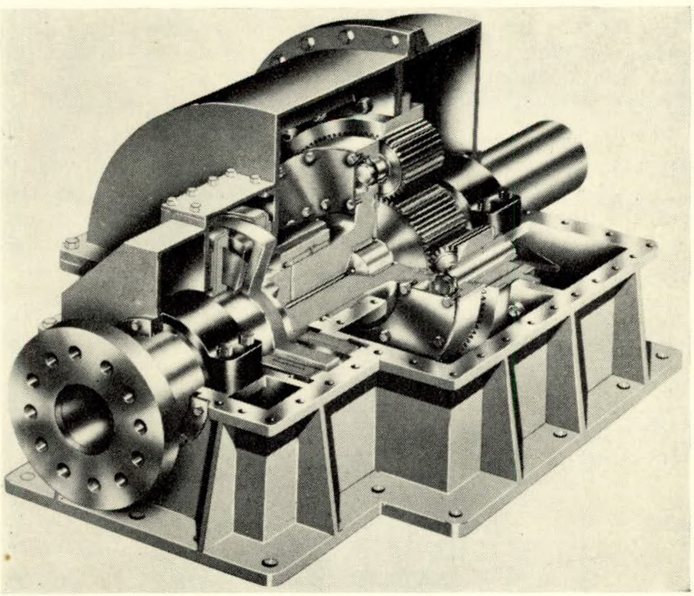


FIG. 19—*Vickers standard marine epicyclic gear size MD 900*

gains which could accrue from the adoption of the principle in a 30 000 hp steam turbine tanker installation. He had been involved with some application work for foreign warship designs and Fig. 18 indicated the gains made possible in comparison with conventional locked train gear arrangements.

Additionally a standard range of marine epicyclic gears had been designed mainly for medium speed Diesel engine applications. A gear from this range was shown in Fig. 19. This design could be provided with or without integral thrust, and since the discontinuity required by a main wheel had been largely eliminated the axial stiffness was expected to be of the same order as standard thrust blocks. This made possible the conception of positioning a standard thrust/gear module well aft in the transmission line with a medium speed and therefore lighter intermediate shaft line to a simple first reduction gear.

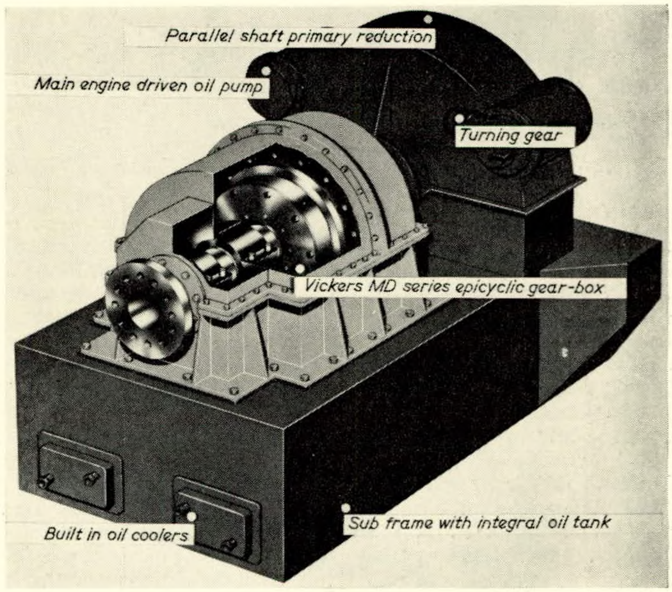


FIG. 20—*Vickers standard final stage planetary gear combined with parallel shaft primary gears for conventional cross-compound steam turbine installation*

The authors made the point that with multi-engine installations the advantages of flexible pin epicyclic gears could not be fully utilized. Since one of the major aims of the designers of high powered marine gears was to achieve a maximum number of mesh points in the final reduction train where torques and tooth loads were high Mr. Brown did not subscribe to this view and some typical examples would serve to illustrate this

In Fig. 20 it was shown that it was relatively simple to employ a standard epicyclic module as the final reduction of a conventional cross-compound steam turbine arrangement.

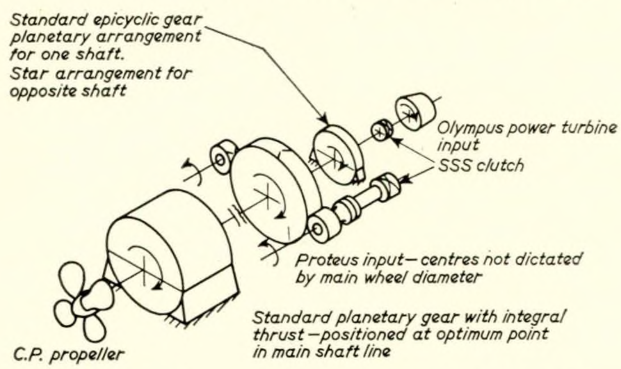


FIG. 21—*Schematic arrangement of alternative solution for* H.M.S. Exmouth *main propulsion system*

Fig. 21 indicated how the layout for H.M.S. *Exmouth*, shown in Fig. 8, could be modified to incorporate a standard final reduction epicyclic module, while Fig. 22 showed a similar

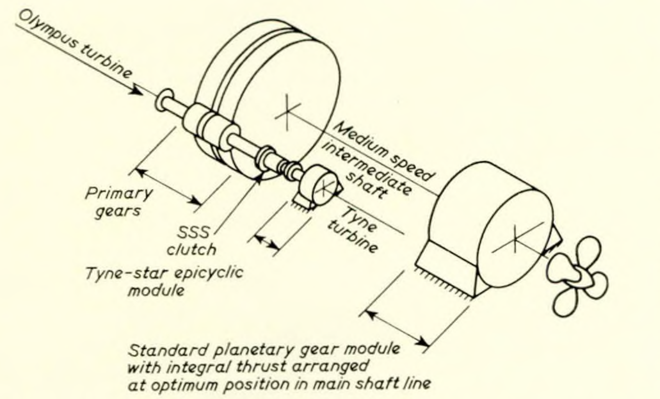


FIG. 22—*Schematic arrangement of alternative solution for Type 42 main propulsion system*

modification to be possible with the Type 42 arrangement illustrated in Fig. 15. It was not Mr. Brown's intention to argue the advantages which would result from the adoption of those solutions. This would be inappropriate since when conceived the designs shown in Figs 8 and 15 were undoubtedly the best available and were eminently suitable for their purpose. Mr. Brown's object in giving these illustrations was simply to show how final reduction epicyclic gears derived as standard modules could be employed with the majority of multi-engine installations.

In conclusion it was clear that the diversity of factors involved in the design of a transmission system required the closest co-operation between the marine transmission engineer and naval architect. It followed that advanced shipbuilding companies such as his own, who were at the same time experienced in power transmission system design and additionally were in the forefront of component design, development and manufacture were ideally placed to offer optimized transmission systems and also provide a service to ensure that their individual equipments could be integrated in the best possible manner into other systems.

MR. P. H. ELLIS said that as one who had been concerned with the gears in the Brave Class fast patrol boats it was gratifying to read that the epicyclic gear trains had given complete satisfaction. It was perhaps worthwhile to mention that the two epicyclic gear trains used in the Brave Class boats used five and eight planet wheels respectively. He felt sure the authors would not wish to give the impression that Stoeckicht gearing was seriously restricted in the number of tooth contacts which could be usefully employed. By and large that figure was determined by the geometry of the gear ratio, and that was particularly true for ratios greater than about 2·5:1.

Fig. 23 indicated the theoretical possibility and practical use of numbers of planet wheels according to the gear ratio. It would be noted that there was little difference between the geometrically theoretical maximum and the number actually used, particularly for ratios of 2·5:1 and higher.

Were the authors able to give details of the powers and speeds of the design study for epicyclic main propulsion gears which they said was in progress? Mr. Ellis felt sure they did not intend to infer that the use of epicyclic gearing for large main propulsion application was not already established, as, since 1965 epicyclic gearing had been used in the steam turbine propulsion machinery of large merchant ships. Already 125 gear trains were in service, totalling one and a half million horsepower and installed in five and a half million tons of shipping. By the end of this year there would be at least 250 trains of gears in service, totalling three million horsepower, and installed in nine and a half million tons of shipping. Already individual trains transmitting up to 20 000 hp were operating. All those gears were for the primary trains in double reduction sets and the primary and secondary trains in triple reduction sets. It was only a question of time before such gears were in service as the final train of large propulsion machinery.

It was probably true to say that virtually all the experience in the world in high power epicyclic gears of 10 000 hp and above was with gears based on the well established Stoeckicht principles, and equally true that by far the greater proportion of that experience lay in this country.

On the more general aspect of gear performance, could the authors enlarge upon the means by which the target of 1000 K illustrated in Fig. 5 would be achieved? For example, the materials to be used, manufacturing techniques, type of lubricating oil and degree of filtration, and whether that target related to peak power or continuous power operation?

Finally, there was no doubt that the authors' paper was of considerable interest to those engaged in the commercial field and possibly there was not always as much difference between commercial and naval applications as was sometimes thought. For example, system cleanliness was certainly a common problem, and it was noteworthy that special care would be taken to ensure cleanliness of the lubricating oil system for the Type 42 machinery. Had the authors any views on why generally it was so difficult to get that important but often neglected detail to be given sufficient serious thought by all concerned with installation and operation?

MR. R. W. JAKEMAN, M.Sc. (Associate Member) commented on propeller shafting, and in particular that shown in Fig. 3 for the Type 42 destroyer. In the absence of complete data on that shaft system, one could only make a qualitative assess-

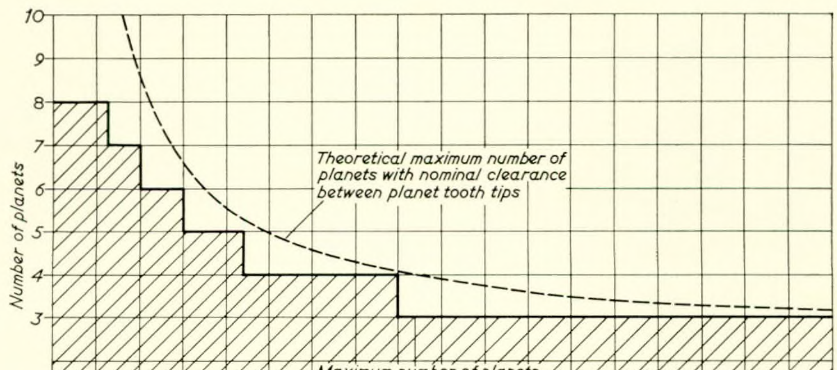


FIG. 23—*Variation of numbers of planets employed in epicyclic gears according to gear ratio*

ment of Fig. 3, but despite that limitation there did seem to be features worthy of discussion. It would appear advantageous to lower the aftermost plummer bearing to some extent in order to reduce the load on that bearing and the shaft bending moment in way of it, and, more important, to increase the load on the intermediate 'A' bracket bearing, which appeared to be very lightly loaded. The importance of avoiding lightly loaded bearings had been indicated by cases where such bearings had effectively lost contact with the shaft due to causes such as inaccurate alignment, bearing weardown or hull distortion. In such cases the unsupported shaft span was roughly doubled, and that might lead to shaft whirling within the limit of the unloaded bearing's clearance. It was possible, however, that the intermediate 'A' bracket bearing was generally not lightly loaded due to the fact that Fig. 3 showed the static conditions only. This led on to the more general comments he wished to make.

Mr. Jakeman had been involved with some work on propeller shaft analysis over three years ago at which time there seemed to be a general awareness of the limitations of a static analysis, not only in respect of the significance of dynamic effects, but also of that due to hull distortion which even rendered some degree of inaccuracy in the static analysis itself. There was very little quantitative information available at that time with which one could try and allow for those factors, and from certain comments in the paper, it seemed that little, if any, progress had been made since then.

One of the cases he worked on was a ferry having a c.p. propeller and a long—275 in—stem tube requiring three bearings to prevent tailshaft whirling. On that it was possible to obtain a figure for the maximum moment applied to the propeller due to the offset centre of thrust at steady state maximum power conditions. No information was available on the transverse thrust applied to the propeller due to the offset centre of torque, or on the cyclic variation of those quantities. In view of that it was decided to apply the very crude approach of applying the maximum propeller moment to a static analysis in order to get at least a rough idea of the significance of that factor. The results were somewhat startling. The shaft slope at the propeller was reversed, the upward angular departure from the straight line datum being almost equal to the downward angle in the true static case. The aftermost sterntube bearing load was reduced by 36 per cent and the centre sterntube bearing load increased by 428 per cent. Although those figures were undoubtedly excessive in view of their origin, they clearly indicated the significance of assymetric propeller thrust. That also raised the question of whether or not slope boring the aftermost tailshaft bearing to suit the static shaft deflexion curve was worthwhile. However, with the exception of bearings having a very poor axial load distribution, bearing wear should only occur at very low shaft speeds when hydrodynamic support conditions could not be attained. In that operating region the assumption of static conditions for the shaft analysis should be a very reasonable approximation, and slope boring the aftermost tailshaft bearing to suit the static deflexion curve was therefore desirable in order to attain the best axial load distribution at low shaft speed, and thereby minimize the speed at which hydrodynamic conditions failed, and to ensure uniform bearing wear when they did.

Finally, the authors had not mentioned the problem of propeller shaft alignment, and it would be interesting to hear their views on that. At the time of Mr. Jakeman's experience in that field, the gap and sag method was much favoured. Optical methods seemed to suffer from "bending" of the optical datum due to variation in ambient temperature along the hull.

COMMANDER A. J. H. GOODWIN, O.B.E., R.N. (Member) said that one of the most interesting things in the paper which gave such a clear account of the "state of the art" at present in transmission design was the promise that it held out for the future. That was true both in gearing and shafting design.

On the subject of gearing design, Fig. 5 gave a slightly misleading impression with its suggestion of a steadily rising K factor. In fact, in the early 1950s the introduction of hardened and ground gearing under AVGRA auspices brought a steep change in K values from around 200 to 500. The target at that time was 600 K but YEAD I trials established the safety of 500 K and that had remained the design limit since. The 820 figure mentioned in the paper was merely a lucky accident. Since then, naval gearing design had consolidated that position and concentrated on absorbing the complications and control problems that arose from multi-engine installations and non-reversing engines. The paper indicated how successful that had been. Production methods had now improved to the extent that the authors' target of 1000 K should well be possible of achievement.

When 500 K was established it was realized that to prevent gear-boxes becoming bearing boxes, it was necessary to make a similar step change in permissible bearing loadings, and taking advantage of I.C. engine experience, those were increased from about 100 to 500. The authors mentioned that increases were now again in mind and perhaps they could divulge what their new target was for bearing loading.

Shafting in the Royal Navy had been remarkably trouble free over a number of years, and the permissible stress limits that the authors quoted for Y205 were also those fixed in the early 1950s. Much more must have been learnt about fatigue and corrosion fatigue since then, so that it was gratifying to note that a fresh look was being given to the shafting design rules. However, the use of the phrase principal stresses in the paper seemed inappropriate where an alternating stress was imposed on a steady stress. That used to be the criterion many many years ago before the present limits were set—the two stresses were added together in textbook fashion as static stresses, and then treated as though they were alternating stresses. No doubt the authors would avoid that over-simplification.

Another feature of interest in shafting design was the effect of the surrounding water on whirling of the outboard shaft. The likely effects seemed to be an increase of mass due to entrained water causing a reduction in critical speed, and considerable damping of the amplitude of whirl. Some recent experience suggested that the latter was by far the more important and that damping was so effective that the amplitudes at whirling were quite acceptable, and moreover that significant amplitudes occurred at very much lower speeds than the critical. It was to be hoped that the authors would include model tests of the water effects, which might lead one to conclude that the whirling in outboard shafts could be ignored, and hence do away with 'P' brackets, with beneficial effects on ship resistance.

MR. A. ILLINGWORTH asked whether the authors had considered anything other than the c.p. propellers and the f.p. propellers. He was thinking particularly in terms of the shroud propeller design which had been employed in gas turbine propulsion machinery. It appeared to have a considerable advantage in that it could be operated at much higher speeds than the standard propeller, hence a saving in the size of gear-box, and although this was not a reversible gear-box, advantage could presumably be taken of the higher output speed to compensate for the increase in size which would be necessary with the brake system in a reversing gear-box.

It was interesting to note that there were some problems which had already shown themselves in service with cleanliness of the hydraulic system of the c.p. propeller. That was one factor which must be looked at very carefully in the application. There had recently been a case of ship trials having to be postponed because of dirt in such a system.

It would be interesting to hear some enlargement on the methods of de-aeration of the oil, which was mentioned in the paper, with a strainer. It was known that there were some problems in getting rid of the air in the oil, in particular in a marine oil system which was, of necessity, very small.

In considering the c.p. propeller it would be interesting to know whether there was any serious consideration given to the possibility during manoeuvring that the crash stern operation of the propeller would cause overspeed of the turbine machinery, due to the ship drawing the propeller.

The authors mentioned the unloading of the bearings causing movement of the shafting system. Had any thought been given to the use of multiload point bearings? It could possibly influence the alignment of the shafting system.

MR. H. A. CLEMENTS said he would limit his remarks, in the main, to additional comments with regard to the synchronous self-shifting clutches.

Mention was made of the *Olympus* drive pinion bearing fatigue experience in H.M.S. *Exmouth*, which occurred after the abnormal condition, of light load, and very high speed operation with the propeller in fine pitch.

Due to the particular machinery installation arrangement possible within an existing ship, the clutch had to be mounted on the gas turbine side of the pinion, and the overall length had to be kept as short as possible. For that reason, the clutch was mounted on an input shaft which was supported at its aft end by an internal bearing overhung from the pinion shaft. That internal bearing was not subjected to relative rotation when the clutch was engaged. When the clutch was engaged and transmitted the high torque normal with high speed operation, the bearing journal was held central, within the bearing clearance, by the powerful centring effect of the clutch teeth. However, at light load the clutch input shaft could move off centre within the bearing clearance, and cause a rotating load on the adjacent pinion bearing.

As locked train gearing was used, the pinion shaft bearing loads were comparatively low, even under normal conditions, but that load was further reduced with the propeller in reduced pitch, so the additional rotating load from the clutch input shaft could cause instability of the pinion bearing.

In Type 42 gearing, the *Olympus* and *Tyne* clutches were mounted at each end of the same pinion and, in that case, the internal bearing was not practicable or desirable. Unlike the *Exmouth* installation, no pinion shaft end was available in which to feed oil from the normal gearing lubricating oil system to lubricate an internal bearing. For that reason, the design was developed as shown in Fig. 16.

Attached to the clutch input shaft was a drum which surrounded the clutch output components, and permitted the input shaft to be supported at the pinion shaft end by a normal gearcase mounted bearing. That meant that the bearing lubrication and accessibility was improved, and any much reduced transverse forces from the clutch input shaft did not react on the pinion shaft. Therefore, the instability condition could not arise, even under transient conditions or abnormal sustained conditions.

The synchronous self-shifting clutches used in the *County* and *Tribal* class vessels incorporated a dashpot in order to cushion clutch movement into engagement only. That was necessary to cater for propeller shaft retardations occurring at the same time as the turbine was accelerating to engage the clutch at synchronizm. That dashpot mechanism was proved to be completely effective. After clutch engagement, the clutch was automatically locked into engagement as was essential to permit astern torque to be transmitted from the gas turbine reversing gear.

The clutch, without a locking control, used in H.M.S. *Exmouth* and mentioned in the paper, incorporated a double-acting dashpot, that is, it was effective to slow down the rate of clutch movement towards disengagement as well as engagement. That meant that a negative torque at the clutch had to be sustained for a brief period in order fully to disengage the clutch.

In a c.r.p propeller installation a small negative torque, reversal usually occurred during a crash astern manoeuvre and, during that period, the clutch would commence to shift towards disengagement. However, the amount of travel depended on the magnitude and duration of the negative torque. As mentioned in the paper, in the case of H.M.S. *Exmouth*, the clutch did not fully disengage. However, should the clutch disengage, it was always arranged that the gas turbine fuel control was automatically set at a lower setting to provide a lower acceleration of the power turbine, so the clutch re-engaged at an acceptable rate immediately the clutch input and output shafts passed through synchronizm. That permitted acceleration rate took into consideration the maximum retardation rate of the propeller shaft likely to occur at the same time. The Type 42 clutches would, of course, incorporate a double-acting dashpot.

The paper emphasized the importance of avoiding unloaded high speed idling gears in a combined machinery installation. In view of the Royal Navy's considerable experience in that field, going back to the use of cruising turbines in *Whitby* class machinery, it would be helpful to learn if measurements had been taken to prove that the noise level of such idling gearing was undesirable.

Correspondence

MR. J. NEUMANN (Associate Member) wrote that the authors had rightly referred to the previous difficulties in assessing the relative stopping performance of rival means of reversing. Fortunately, computing techniques had now been developed to tackle this problem, and an increasing confidence was being generated in the results obtained from these techniques. They relied on dynamic simulation of the whole ship/propeller/machinery system on a suitable computer*. The increasing confidence arose from a growing body of satisfactory correlation between actual trial results and the simulated results produced by the computer.

In the brief description of the Type 42 machinery, the authors referred to the use of controllable swash axial piston pumps in the propeller hydraulic system, and to the possible future use of hydrostatic drive for forced lubrication pump drive. The applications of this rapidly developing technology into main propulsion auxiliary systems would no doubt be watched with much interest. Satisfactory experience could well see a much wider application to the drive of other services.

The authors referred to the flushing of the main gearing before delivery to the shipyards, and to subsequent flushing of the whole system. The importance of these operations could not be over emphasized, particularly in these days of relatively complex designs with small clearances. To ensure a satisfactory flushing routine it was necessary to bear in mind the vital nature of this operation right from the beginnings of the "Paper" design, and to design the pipe runs and provide off-takes and connexions accordingly. Had the authors any evidence that the efforts made in this direction during the last few years had in fact borne fruit?

Reverting to the flushing of the gear-box before delivery, it would be of much interest to learn how much confidence the authors had that all swarf, dirt and other foreign bodies would in fact have been cleared out from the gear-box before delivery; also whether any special provisions were intended to dislodge contaminant which might be wedged in inaccessible spots outside the main line of flow.

* FORREST, J. and NEUMANN, J. "Design Development, Evaluation, Testing, Tuning, Commissioning and Training by Computer Simulation." *Imas Proceedings*, Section 1, p. 18.

Authors' Reply

Replying to the discussion, the authors thanked the contributors for their help in amplifying necessary points and for adding to the text from their own experiences.

Mr. Brown had suggested that for powers in excess of 20 000 shp, a fixed pitch propeller and reversing gear-box arrangement was the best solution for the gas turbine ship problem. At present the authors had no knowledge of any studies which clearly demonstrated this. They thought it could well be true that for certain types of ships this solution was acceptable, but it was very much in the balance for warships, as pointed out in the paper. Mr. Brown was correct in emphasizing the need for close co-operation between the transmission designer and the naval architect. Reference was made in the paper to the Admiralty Experiment Works at Haslar, which was staffed by naval architects and scientists and where the hydrodynamic forms of propellers and hulls were designed and subjected to model testing. It was A.E.W. which proved much of the basic data for the computer simulations which defined the transmissions and control systems requirements.

It was possible, as Mr. Illingworth suggested, to overspeed the power turbine during manoeuvring. From Fig. 10 it would be seen that early on in a manoeuvre, the shaft speed fell and then rose again. This was due to the feedback of power (note the negative shaft torque) with the propeller fined off. If pitch was taken off before the shaft speed had reduced then there was the danger of overspeeding the turbine. During the computer simulations the optimum lag between throttle closing and pitch reduction was deduced, so that for future designs overspeed was not envisaged. In a failure mode, for example, if pitch fined off when at full power and overspeed occurred, then the engine was protected by an overspeed trip. If there was no lock on the clutch the engine could then run down safely.

The authors thanked Mr. Ellis for his comments on the number of planets which could be practically employed in epicyclic arrangement. Whilst it was true that at higher gear ratios the number of mesh points was severely limited, at ratios below two there was a dramatic increase in the number of mesh points possible. Studies had shown that in certain cases a better power weight ratio could be achieved by combining two low ratio stages than would be the case with a single reduction.

The epicyclic gearing design study referred to in the paper was based on a typical modern frigate and considered both primary and secondary epicyclic stages in a double reduction.

Mr. Brown's suggestion that epicyclic gears could have been used in H.M.S. *Exmouth* and the Type 42 was quite valid. Epicyclic gears were not considered for H.M.S. *Exmouth* since she was to use a gear-box as close to a standard *Leander* box as possible due to the short time scale and installation limitations. The *Proteus* primary drive was, of course, epicyclic. Such gears were considered for the Type 42 design but were rejected in favour of parallel shaft gears for a number of reasons including:

i) the need to tie together the shaft ends with a reasonable distribution of gear loading;
ii) better accessibility;
iii) the lack of development time available.

Mr. Illingworth suggested that a high speed shrouded propeller might be useful to reduce the gearing problem. This certainly would reduce the amount of gearing, but the authors would expect such a high speed propulsor to be very noisy, and to suffer extreme cavitation erosion. Their experience with high shaft speeds on fast patrol boats had been that propulsor life was short indeed, although trials with stainless steel propellers were encouraging.

In answer to Mr. Brown's questions regarding stiffnesses of thrust blocks and seatings, the MOD(N) had no set method of calculating this at present as it depended very much on the unknowns involved in gearcase and seating construction. Measurements had shown that the combined stiffness of thrust block and seatings in a typical integral thrust block frigate application ranged from 2000 to 4000 ton/in. Axial vibration calculations were carried out over this range of stiffnesses. This situation was far from satisfactory. However, resulting from NAVGRA work on gearcase structures referred to in the paper, very soon there would be a satisfactory analytical prediction method. The accuracy of this method would depend upon the refinement of the model used, which in turn would depend upon the amount of money expended. Great accuracies were not, of course, called for since the natural frequencies were a function of the square root of the stiffness.

It was agreed that the siting of the thrust block aft, although ideal from a transmission point of view, might not be possible because of installation factors, one of the most obvious disadvantages being that it either had to have its own L.O. system or else pipes leading to and from the gearing, which presented action damage problems.

Work on propeller excitation forces was in hand, but due to the complicated nature of the problem it would be some time before meaningful results could be obtained. The work ranged from computer studies to model tests and full scale tests in a frigate.

Mr. Jakeman suggested that by lowering the aftermost plummer bearing in Fig. 3 the loading could be increased on the intermediate 'A' bracket bearing. The actual bearing reactions were shown in Table I. The intermediate 'A' bracket bearing load was deliberately kept low to enable a reduction in the size of the bearing scantlings and hence improve the hydrodynamic flow conditions around the after end of the hull, whilst at the same time the loading was not so low as to allow loss of support. In addition to this there was the point made by Commander Goodwin about the possible effects which immersion in sea water might have on whirling. His suggestion certainly merited close attention, since removal of the intermediate 'A' bracket would be a considerable advantage.

TABLE I

Bearing	Bearing reaction (kg)	(lb)
Main gearwheel forward bearing ...	2520	5550
Main gearwheel aft bearing	3160	6970
Forward plummer bearing	5770	12 670
Aft plummer bearing	6220	13 720
Intermediate 'A' bracket bearing ...	4320	9510
Main 'A' bracket bearing	19 200	43 290

Regarding Mr. Brown's question about tailshaft failures due to whirling, the authors had no such experience. The only recent failure was due to corrosion of a tailshaft at the position of highest bending stress.

The authors agreed with Commander Goodwin that to use the term principal stress could be misleading. It would be more accurate to refer to a summation of the alternating stresses. This would require analysis and need to be combined with the static stresses in order to predict a fatigue life. Such a method was proposed for the future.

The shafting shown in Fig. 3 was based on the existing design rules and did not include the dynamic effects which Mr. Jakeman suggested. The authors agreed with Mr. Jakeman about the limitations of the static analysis. The new design rules which were being developed would be based on a design study which would consider the dynamic effects. This point, could, perhaps, have been emphasized more in the paper.

The authors thanked Mr. Jakeman for his interesting account of some of his own experiences and fully endorsed his observations about lack of quantitative information available to the designer.

Lining up of shafting was carried out optically and as Mr. Jakeman said there were inaccuracies due to changes in ambient temperatures. No account was taken of this and since the bearing loadings were conservative, no trouble had been experienced. The gap and sag method was used in the alignment of the gearing to shafting. Since this was quite sensitive, correction factors related to temperatures were applied.

The effect of the ·03 in deflexion between the bearing of the main wheel referred to by Mr. Brown was not, in fact, quite correct. The gear-box was displaced in relation to the forward shaft flange but not as much as might be implied from Fig. 3. The effects on the mainwheel bearings of closing this gap were very small compared with gear load forces and the structural stiffness of the gearcase (see Table I).

Mr. Brown's remarks on the reductions of the rig to ship factor were valid. The factor of 2·5 was an environment factor. If gear-boxes were designed and mounted or isolated in the way referred to in the paper, there should be a considerable improvement in the environment and hence an increase in gear loading might be permitted. Quantifying and subsequently isolating the effects of propeller vibration, transient manoeuvring torques and seaway effects would not be easy, so that the factor would never be reduced to 1·0.

Commander Goodwin was quite correct when he said that in terms of practical application there was a step increase in K factor. However, the capability for increasing K factor had grown steadily, so that the dotted line in Fig. 5 could be regarded as a growth of confidence, based on research and development and experience at sea.

Mr. Ellis' questions as to how 1000K was achieved were briefly answered by Mr. Brown's remarks in the discussion which supplemented the authors' brief coverage on pages 6 and 7. A more detailed account of the NAVGRA programme was given by Lieutenant Commander B. J. McD. Gowans*. The K values aimed at were for continuous ratings.

Commander Goodwin quite rightly pointed out that an increase in K value meant an increase in bearing loadings if 'dumbell' shaped pinions were to be avoided. There were no load targets as such. The criterion for bearing design for the future was to be one of minimum oil film thickness with an overriding temperature limitation. Studies based on practical tests had shown that bearing design would not be a limitation except under certain instances; in those cases slight modifications to the gearing design would be necessary.

In reply to Mr. Illingworth's question on lobed bearings, these had been considerable for certain applications, such as the *Olympus* input line in H.M.S. *Exmouth* described in the paper, but had been discounted due to the complexity involved.

Captain Inches asked about the noise reduction of gears and propellers. In the case of gears, the source of noise and the transmission of noise through gear-box structures was being studied through the NAVGRA research programme. The aim was to make sure that the right sort of profile corrections were obtained to give minimum gear noise. This was only true for a given condition, since the corrections were designed for a particular deflexion, and this was associated with a particular torque, so it was necessary to select the power at which the minimum noise output was desired. Transmission of noise had been dealt with by NAVGRA, but it was a problem common to other machinery, so that within the Ministry of Defence (Navy) this whole subject was under review. For seatings, constant position mounts were mentioned in the paper, and there were various methods of hooding and cladding which were being looked into for gear-boxes and other types of machinery.

The noise reduction of propellers was a classified subject, but in general an effort was made to operate propellers at design pitch and to manufacture them accurately. With the c.p. propeller the aim was to get to the design pitch as soon as possible for what might be called a "quiet running position", although for rapid manoeuvring consideration would be given to changing the pitch to give a better ship response. This was indicated in Fig. 1.

Mr. Clements raised the question of noise from idling gears. The authors had no quantitative information to divulge, but idling gears were not popular, not only from the noise point of view but also because of the risk of hot bearings and this was borne out by experiences with a cruising train in the Y100 design. It would be noted that in the Type 42 design there were no idling gears.

Captain Inches raised the subject of lubricants and lubrication and of who was responsible for the research. Research into gearing lubricants was carried out by the oil companies and the Admiralty Oils Laboratory (A.O.L.). This work was sponsored by the Navy Department Fuels and Lubricants Advisory Committee (N.D.F.L.E.A.C.). Research into lubrication of gears, with particular reference to scuffing, was carried out by NAVGRA. However, it was clear that these two organizations needed to work closely together to prevent overlapping and to maintain a strong dialogue. This was done through a N.D.F.L.E.A.C. panel which included in its membership the NAVGRA research officer and a member of the D.G. Ships Transmission Section.

One of the problems under consideration by this panel at the moment was whether E.P. additive oils were necessary and if they were, what the specification should be so that it was available world wide. This was of significance not only for logistic reasons, but also when considering the export of warships. It was through this panel that the naval gearing firms informed the Oil Industry of the materials, finish, operation, etc. of future gear designs. The Oil Industry with this information then produced a suitable formulation. The oil was tested for acceptability by A.O.L. Obviously this was a most complex problem to solve, particularly in foreseeing the requirements of the future.

Mr. Illingworth asked for some enlargement on deaeration methods. These were covered well by Rimmer and Liddell [2]. This indicated that proper baffling allowed air to be separated from the air oil solution. Alternative methods under consideration included "fish tails" where the oil was returned to the surface of the tank in a smooth horizontal flow so that the chance of natural air separation was greatest.

Captain Inches pointed out that elimination of dirt should be the rule rather than the exception. This was a statement with which the authors fully concurred; and this sentiment was obviously shared by most of the other contributors.

How could we achieve high standards of cleanliness? The answer lay, of course, in educating industry of the need,

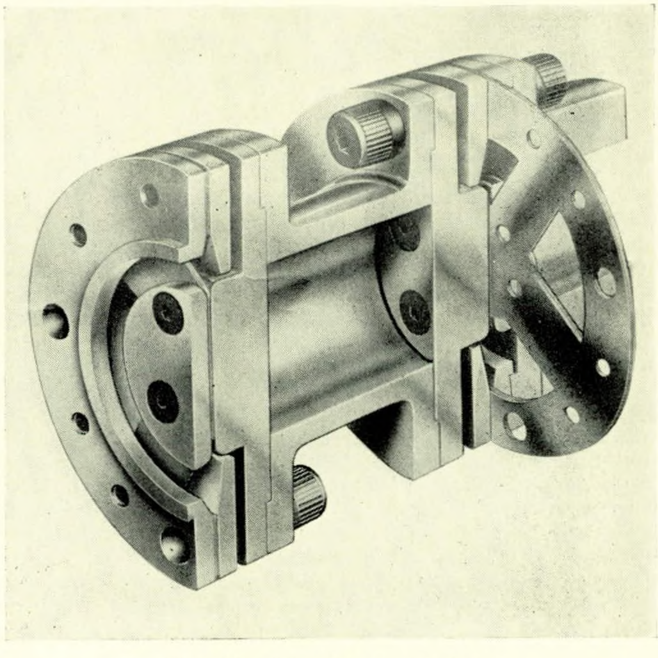


FIG. 24—*Membrane coupling*

* Gowans, Lieutenant-Commander B. J. McD., NAVGRA (Navy and Vickers Gearing Research Association), M.O.D. publication, Journal of Naval Engineering, Vol. 18, No. 2, p. 152.

and the sound economics of quality, of which cleanliness was one aspect. The Navy Department had gone a long way towards improving standards of both the equipment manufacturers and the shipbuilders. There was at the moment a tremendous campaign underway to improve quality, initiated by government departments and the nationalized industries. Since these organizations had immense purchasing power there was no doubt that higher standards would be achieved.

Mr. Neumann asked how to ensure that gear-boxes were thoroughly clean before delivery. This was achieved first of all at the design stage by ensuring that there were no dead-ends which could not be opened for inspection and no ledges inside the gearcase. After assembly, flushing was carried out at very high velocities. The box was then stripped and inspected in clean room conditions under close scrutiny. It was then reassembled, and sealed before delivery.

Mr. Neumann raised the question as to whether cleanliness had paid off. It was difficult to quote meaningful statistics, but the cleanliness standards had been improved to such an extent over the last ten years that only two major problems had occurred on initial commissioning in that period.

The MOD(N) had not experienced major problems with dirt in controllable pitch propeller hydraulic systems as implied by Mr. Illingworth. This was because a great deal of care and attention had been paid to rigorous flushing procedures and periodic monitoring. Economically this was wise, for the extra effort and expense before commissioning was amply repaid by lack of problems in service which, by their very nature, were difficult to rectify.

The authors agreed with Mr. Neumann that there was considerable scope in the future for the use of hydrostatic variable speed drives. The most obvious application was for auxiliary pumps driven from the main machinery whose output needed to be independent of mainshaft speed. The other advantage of this type of drive was its flexibility in that such an auxiliary pump could be located remote from the power source.

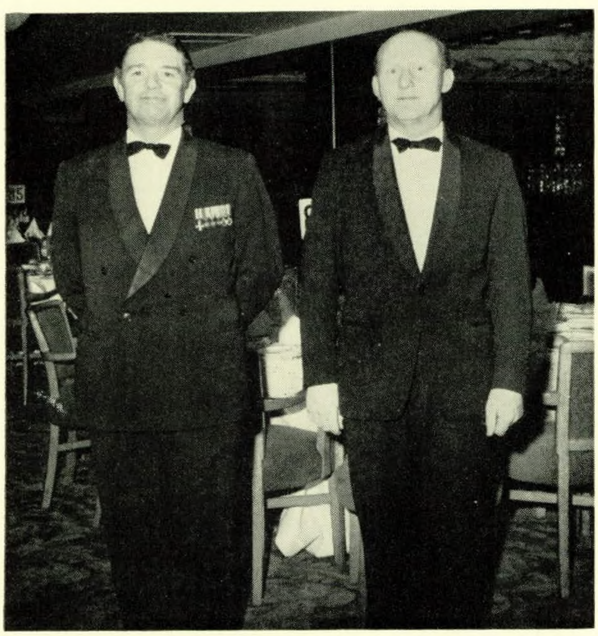
Captain Inches asked for an explanation of the diaphragm coupling. This coupling consisted of a number of thin stainless steel membranes held between two flanges. The membrane had spokes and the outer ring was connected to one side of the drive and the inner ring to the other. Torque could be transmitted in a misaligned state; the spoke deflected to accommodate angular and axial misalignments (see Fig. 24). Normally two banks of membranes were used separated by a torque tube. An emergency tooth drive was incorporated.

ANNUAL DINNER

The Sixty-seventh Annual Dinner of the Institute was held on Friday, 13th March 1970 at Grosvenor House, Park Lane, London, W.I., and was attended by 1390 members and guests.

The President, F. B. Bolton, Esq., M.C., was in the Chair. He was supported by the Chairman of Council, B. Hildrew, Esq., M.Sc.

The official guests included: His Excellency El Marques de Santa Cruz, The Spanish Ambassador; His Excellency Baron Jean van den Bosch, The Belgian Ambassador; His Excellency A. P. Rajah, High Commissioner for Singapore; Sir Denis Blundell, K.B.E., High Commissioner for New Zealand; His Excellency Dr. M. V. Peiris, High Commissioner for Ceylon; Monsieur P. C. Witte, Minister Plenipotentiary (Economic Affairs), representing His Excellency the Netherlands Ambassador; Tore Bøgh, Esq., Counsellor of the Norwegian Embassy, representing His Excellency the Norwegian Ambassador; Rear Admiral B. Yashin, Soviet Naval Attaché, representing the Soviet Ambassador; Commodore N. P. Datta, Naval Advisor to the Indian High Commission; Captain E. Photiadis, R.H.N.F., representing His Excellency the Greek Ambassador; Monsieur Richard Gaechter, Officer in charge of Economic Affairs, representing His Excellency the Swiss Ambassador; Captain L. W. L. Argles, C.B.E., D.S.C., R.N., Merchant Navy College; Sir Max Brown, K.C.B., C.M.G., Second Permanent Secretary, Board of Trade; J. Calderwood, Esq., M.Sc., Honorary Member; The Reverend L. E. M. Claxton, M.C., M.A., Rector, St. Olave's, Hart Street, London, E.C.3; R. Cook, Esq., M.Sc., Honorary Treasurer; Commander J. E. Dumbrille, R.C.N., Canadian Defence Liaison Staff; Dr. D. F. Galloway, President, Institution of Mechanical Engineers; G. Geddes, Esq., F.C.M.S., President, Society of Consulting Marine Engineers and Ship Surveyors; B. Hildrew, Esq., M.Sc., Chairman of Council; Captain L. A. Hill, D.S.C., R.D., R.N.R., The Master, Honourable Company of Master Mariners; Stewart Hogg, Esq., O.B.E., Past Chairman, Social Events Committee; G. C. Howard, Joint Managing Secretary, Salvage Association; Dr. R. Hurst. G.M., Director of Research, British Ship Research Association; R. A. Huskisson, Esq., Chairman, British Shipping Federation; Sir Gilmour Jenkins, K.C.B., K.B.E., M.C., Past President W. A. C. Kendall, Esq., President, Association of Teachers in Technical Institutions; J. H. Kirby, Esq., Vice-President, Chamber of Shipping of the United Kingdom; A. Logan, Esq., O.B.E., Past President; The Right Honourable., The Lord Mancroft, K.B.E., T.D., Deputy Chairman, Cunard Line Limited; A. J. Marr, Esq., C.B.E., Chairman, British Ship Research Association; D. E. Maxwell, Esq., Director and Secretary, British Marine Equipment Council; W. S. Paulin, Esq., President, North East Coast Institution of Engineers and Shipbuilders; Sir David Pitbladow, K.C.B., C.V.O., Secretary, (Industry), Ministry of Technology; R. S. Punt, Esq., Chairman, National Association of Marine Enginebuilders; Vice-Admiral R. G. Raper, C.B., R.N.R., Director General, Ships, Ministry of Defence, (Navy); The Right Honourable Goronwy Roberts, M.P., Minister for Shipping and Civil Aviation; I. W. Robertson, Esq., Chairman, Social Events Committee; H. F. Schwarz, Esq., B.Sc., President, Institution of Electronic and Radio Engineers; The Right Honourable Viscount Simon, C.M.G., President, Royal Institution of Naval Architects; M.A., Sinclair Scott, Esq., C.B.E., President-elect; A. L. Stuchbery, Esq., O.B.E., President, Institution of Production, Engineers; George Sulzer, Esq., Sir William Swallow, Chairman, Shipbuilding Industry Board; Commodore E. C. Thorne, R.N.Z.N., New Zealand Defence Liaison Staff.



At the Annual Dinner held on Friday, 13th March 1970 at Grosvenor House, London, W.1: The President of the Institute, Mr. F. B. Bolton, M.C. (left), with Mr. B. Hildrew, M.Sc., Chairman of Council.

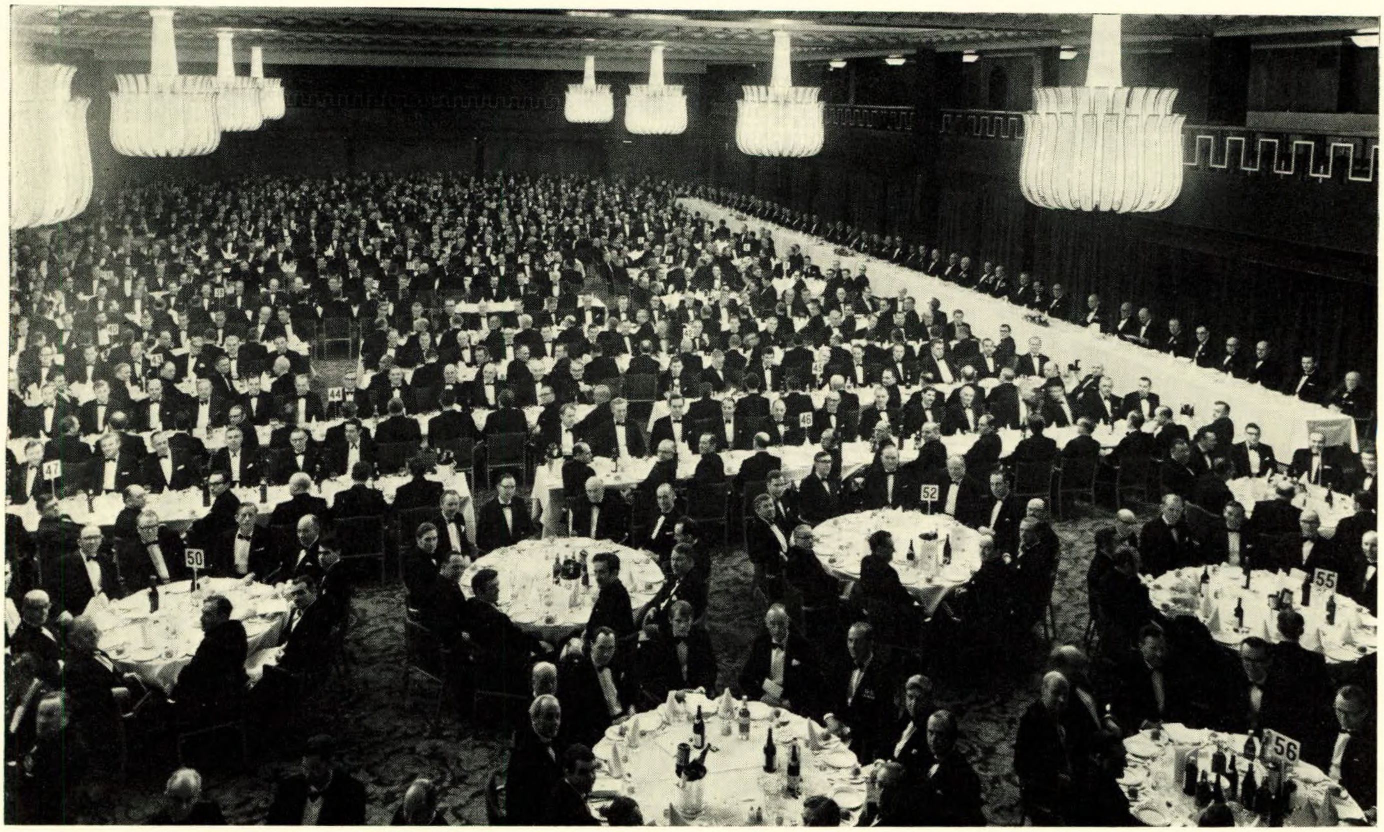
The Loyal Toasts having been duly honoured.

MR. GEORGE SULZER proposed the toast of The Royal and Merchant Navies of the British Commonwealth.

He said: It was to me a very great honour as well as a big surprise when, as a Swiss coming from a land-locked country, I was invited to propose the toast to the Royal and Merchant Navies of the British Commonwealth at tonight's Annual Dinner.

Looking around this hall, I must admit also to being very much impressed by the size and quality of my audience. I fully realize that I have a responsibility to try to live up to those ambitious standards of the Institute of which we are all well aware. I do not know who chose Friday the 13th for this dinner. *(Laughter)*

Last year two outstanding historical events took place: The first man set foot on the moon, and also for the first time in history two mine sweepers of the Royal Navy paid an official visit to a Swiss port. In April, 1969, H.M.S. *Flintham* and H.M.S. *Dittisham* sailed up the Rhine to Basle and it was a proud moment for my countrymen to watch them anchor in Swiss waters. *(Applause)* The Royal Navy has for generations been almost enviously admired by old and young in Switzerland as the incorporation of skill,



Annual Dinner 1970

valour and unique tradition. The ten officers and ratings and 30 naval cadets spent a few days visiting places of interest in Switzerland and were also given a cordial welcome in my home town. We hope that now that the ice is broken, further such contacts will follow, and although we shall probably never see an aircraft carrier, some day an advanced hovercraft may cross the Alps. (*Laughter*) The local wines and Kirsch which your sailors seemed to enjoy during their visit may be an encouragement for them to come again, and perhaps even some compensation for the abolition of their rum ration.

My country's relations with the merchant marines are, however, longer and closer, not so much because of the Swiss fleet which, I must remind you, does exist although it never enters a Swiss port, but because of its not unimportant contribution to marine engineering. Engineers all over the world speak the same language. This fact has made it possible for Swiss developments to share a part of the world market in propulsion machinery. It opened the door to the world for Swiss engineers and is also the reason for my having the honour and privilege of being a member of your Institute.

Unlike politicians, we—I mean engineers—prefer to speak about matters we know at least something about, and in my case this is marine propulsion. Instead of a silver spoon I was, speaking metaphorically, born rather with a Diesel engine in my mouth, and have served some time as an assistant engineer—in fact, as an apprentice—on board ship. "Join the Navy and see the world" is a tempting slogan for any young man, particularly a Swiss, and for me it was professionally a short, but unique, experience. Unfortunately, I did not see much of the world apart from the sea itself, because in every port immediately after stopping the engines, the below-decks gang got busy drawing pistons, and our recreation consisted in watching the waves when again at sea. This was, of course, many years ago, and things have changed since then. (*Laughter*) If necessary at all, I presume that soon such work will be done by pressing a button on the bridge or by a computer programme.

The pattern of marine propulsion is undergoing rapid changes in many ways. Whilst in the past the development of propulsion machinery such as steam engines, steam turbines and Diesels had a major influence on the size, speed and design of ships, today and in the future the determining factor in shipbuilding is more and more the overall economy of a transportation system. The needs of world trade set the pattern, and shipbuilders and marine engineers will have to comply with these requirements. There is a vigorous challenge to marine engineering and the beginning of rapid evolutionary and revolutionary developments. It calls for a complete revision of our thinking in many ways and basically new approaches to a new philosophy.

The total gross tonnage of the world's merchant fleet is today 220 million tons and is expected to double within the next ten years. There will certainly be room for many ships of conventional types, but the bulk is moving rapidly towards new sizes and concepts of vessels in the field of tankers, bulk carriers and a wide variety of different types of container ships. What will the effect of these trends mean for propulsion machinery?

The slow-speed direct-drive Diesel, referred to as cathedrals by the sophisticated-minded younger generation, will no doubt hold its own for many years due to the possibilities it offers with regard to reliability, increased outputs, the capacity to burn heavy fuels and its maximum efficiency as a thermal prime mover. Medium-speed Diesel installations have gained considerable importance for lower power and especially where the height of the engine room is limited by the ship's design. For high-powered ships, the substantial improvements realized with steam turbines will also assure this system's future. However, in the light of the modern tendencies, gas turbines in combination with highpower reversible gears, variable pitch propellers or water jet propulsion are gaining ground for special types of ships. Low weight and a concentration of power in small dimensions as well as maintenance facilities and immediate availability may easily offset a somewhat higher fuel consumption and higher fuel costs in an overall economic consideration. With time, metallurgical progress and blade cooling devices are also likely to allow the use of higher temperatures and thus improve the efficiency of the gas turbine and make its operation on cheaper fuels possible.

Nuclear power, I believe, is not an immediate competitor for the merchant navy of the next decade, but in view of the growing acceleration in technological developments it would be dangerous to rely on any so-called expert forecast in this respect.

Many excellent and comprehensive papers have been written recently on the subject I am only able to touch upon tonight, and I realize that I have not even mentioned remote control, computer control and many other new trends. The challenge is world-wide and the scope for ingenuity in marine engineering unlimited. Technical developments are, furthermore, like the ships on the sea, serving all nations, not bound to any national frontiers, and international co-operation is an essential prerequisite for progress.

The rationalization and concentration process in shipbuilding and marine engineering is taking place in all industrial countries and the thinking behind the Geddes Report has a universal validity. As a consequence, the growth of monolithic structures encourages a greater measure of central government intervention leading ultimately to control, and we must be prepared to face the challenge to private enterprise. I am thoroughly convinced that private enterprise cannot be replaced by any more efficient and stimulating system for the achievement of genuine progress *(Applause)* and we who are impelled by the nature of our profession to think on a world-wide scale, have a responsibility to ensure its continuation.

Much of what I have said in regard to new developments and new materials applies to warships as well as to merchant ships, but one factor which does not change is the quality and loyalty—and sometimes even the sacrifice—of those men who man the ships in good times and bad and without whom we should find ourselves exposed to dangers and a shortage of those things we need to maintain our living standards. In this respect the Royal and Merchant Navies of the British Commonwealth have an outstanding record and admirable tradition.

I now raise my glass and propose the toast to the Royal and Merchant Navies of the British Commonwealth.

The Right Honourable The Lord Mancroft, K.B.E., T.D., in reply and also proposing the toast of The Institute of Marine Engineers, said:

I am grateful to the Institute for the compliment they paid me in asking me to be amongst their guests this evening, and for having charged me with the responsibility, indeed, the almost overwhelming task of not only replying to the toast which has just been proposed with such expertise, fluency, charm and—if the Minister will permit me—ringing Tory tones by Mr. George Sulzer, but also of proposing the toast of your Institute, with which I have the honour to couple the name of your President. Gentlemen, this is too much for one man. I am beginning to feel some sympathy for the chap who found himself as the seventh husband of a film star upon his wedding night wondering, whilst he realized what he had to do, how on earth he was going to make it interesting. *(Laughter)*

For a landlubber like myself to have to reply on behalf of the Royal and Merchant Navies is a presumption, even *lese-majesté*. I never had the honour to serve with the Royal Navy, though I did, like many others present, once have the honour of serving under the Royal Navy, in June 1944, when many of us soldiers and a few airmen too ploughed up and down the lanes of Normandy under the encouraging noise of the 15-in and 16-in shells of *Warspite, Nelson* and *Rodney*, humming reassuringly over our heads.

You, Mr. Sulzer, referred to the age of change. We have had recently in this country a debate on naval and defence affairs, and anybody who listened to that debate in the House of Commons and—again begging the Minister's pardon—dares ever to use the expression "drunk as a Lord" . . . *(Laughter)*.

May I take one account from the Royal Naval debate, a speech by the Hon. Member for the Farnham Division of Surrey. He said:

> "We have been repeatedly told in this debate that the rôle of the Royal Navy is changing with bewildering speed. But has this not always been the case, for the role of the Navy can never be static. What is gratifying, however, is the speed and efficiency with which the Navy adapts itself to any new role, usually in the face of discreditable indifference from Parliament and the public."

That speech was made by the Honourable Member for Farnham, but not Mr. Maurice Macmillan, who is now the Honourable Member for Farnham in the debate on the Royal Navy on 4th December, 1926. That was my own father. *(Applause)*

> May I be permitted to switch to the House of Lords. "Once again"—says the speaker—"the rôle of the Navy is changing, a reflection of the changing and diminishing rôle of this country on the international stage. If, however, we still want a voice in the councils of the world, well and good. If not, scrap the Royal Navy and we shall neither have nor deserve a voice."

I agree with that, which is just as well, because it was the opening sentence of my own maiden speech in the House of Lords 26 years ago. Of course, times have changed and will continue to change for the Royal Navy and the Merchant Navy.

I can speak with a little more decisiveness about the Merchant Navy, for I have for four or five years been earning my living in their midst, though my interest goes back many years, to the time when I was a schoolboy at Winchester. A broad-minded house master enabled my friends and myself to bicycle down to Southampton Docks and go around on a half holiday visiting the ships tied up. To show that we had behaved ourselves, we were compelled to bring back a picture postcard of the ships we had visited, countersigned by the officer of the watch. The last time we visited Southampton we entertained to luncheon in the South Western Hotel a Miss Fifi de la Bonbon, or words to that effect, who was appearing in a non-stop striptease review at Southampton Empire—and twice nightly at that. Unfortunately, the leader of our mission, with remarkable lack of diplomatic tact, when he came to hand the postcards back to our house master accidentally failed to remove from the pack a signed postcard of Miss Fifi de la Bonbon clad in nothing but a bewitching smile and three small pieces of Elastoplast. It is for this reason that I referred to our visit as the last to Southampton. Our leader, whose lack of diplomatic skill I emphasized, ended up a very senior officer in the British Foreign Office.

When I visited the South Western Hotel four years ago it had been converted into the Cunard headquarters in Southampton. I went up to my room and they showed me the desk I was allowed to use. When I took out the rubbish left by my predecessor, I found a photograph taken from the roof of the hotel in, I suppose, about the middle of the 1920's. It showed Southampton Docks with over 22 big passenger liners tied up. Out of curiosity I went up to the roof and stood on the same spot at which the photograph had been taken. How many ships were there? Three—three where there had been over 20 in the middle of the 1920's. This is the change to which Mr. Sulzer referred. It is a change brought about largely by the jet. We read every day of the death of the passenger liner. I do not believe it, though it is difficult to contradict it when you realize that today, Friday, 13th March, you cannot cross from England to America as a passenger by sea. On this day there is not a single liner on that run—this in the 350th anniversary year of Mayflower's crossing.

Where are they? They are all cruising, all out in the changed world, the leisure market. It is very sad to see familiar old names going, but the passenger fleets of the world were rebuilt after the war at roughly the same time and they are roughly now coming of age. The change is further accentuated by the fact that as passenger ships have less and less need for speed the cargo ships are coming up to the 22, 23 and 24 knot range. The two Queens, bless their sweet hearts, war-winning weapons as they were, were nothing more or less than gigantic ferry boats. Now one is at Long Beach, California, and the other at Port Everglades, Florida, propping up the United States of America like a couple of gigantic bookends. *(Laughter)*

Although this change has taken place, something else is happening which is not always recognized. There are 19 big passenger ships on the stocks of the world at this moment, and seven more under discussion, in one or two of which I personally have a more than academic interest. The day of the passenger ship is not over. We are moving into the leisure world, into a world where the three-week, if not four-week, holiday is becoming a commonplace, and two holidays in the year are almost taken for granted in the Western world. We are moving into a world where people are eager to find more leisure. Businessmen are prepared to go out by air to do their deal and come back triumphant and relaxed by sea. We are moving into a world in which we must not forget 25 million of our fellow countrymen and women have never yet left this country for a holiday. Therefore, the day of the passenger ship is very far from over.

We are moving, though, as Mr. Sulzer told us, into a world of change, as, indeed, is the Royal Navy. Neither could move at all but for the expertise, skill and industry of you and your Institute, whose toast we are to drink. You have governed its affairs with commendable skill and energy, Mr. President, to which we pay our proper tribute. I pay no more than a tribute, because I have no engineering expertise. My knowledge of engineering can be put in a nutshell, still leaving ample room for the nut. It is a stupid thing ever to talk to experts about their own affairs. I do not, therefore, talk to anybody about engineering.

I did it once, in the early stages of the war, when I was turned on to lecture a squad of recruits on the Barr and Stroud Rangefinder Mark II. I knew all about the Barr and Stroud Rangefinder Mark II, because I had been given a lecture on it by our adjutant about 10 minutes before. I was going strong, drawing on a fertile imagination to fill in the all too evident gaps in my technical armoury when all of a sudden a little recruit at the back started to ask one of those series of questions which, unless stopped at once, will obviously end in tears. So I fell back on heavy sarcasm, commonly the defensive armament of all sergeant majors, as I then was. I said, "Now then, cocky, how long have you been in the Army?" He replied, "Three days, but I have been 16 years in Barr and Stroud."

I cursed my adjutant. Mark you, he was no engineer either. When I first joined the regiment we had horses. When we were mechanized he was sent for a mechanization course and he returned talking with immense authority about the speed of the engine, the ball-bearings, the governor and the rotor arm. We were all deeply impressed until one of us had a look inside his note book—a very fat note book—in which he had written only one thing on one page. This is what caused us to doubt his engineering efficiency. He wrote: "As the speed of the engine increases, so the governor's balls fly off in every direction."

Please forgive me for flying off in every direction. Let me control myself and return to my task. Let me thank Mr. Sulzer warmly on behalf of my fellow guests and myself for the splendid way in which he proposed the toast, and ask my fellow guests to stand with me and drink a toast to the Institute coupled with your name, Sir.

The President, in reply, said:

Lord Mancroft's seventh husband of the film star may not have known how to make what he knew he had to do interesting, but Lord Mancroft himself most certainly did. Thank you very much, Lord Mancroft, for proposing the toast to the Institute and also for the way in which you did it.

I was reading the other day a splendid little book called *Bizarre Ships of the 19th Century*—by one of our members too—which I can most heartily recommend to the very few people in the room who will not already have come across it. It opened—almost—with the caustic comment of a 1920s marine engineer on an early Diesel installation in which "everything moved except the propeller".

The marine engineer of 1970 would echo that sentiment —not about the propeller not moving, but "everything". There may not be much precedent for your having to listen to the same President twice at these functions—he usually "moves" too—but I have been appalled, in my two years, to see just how fast matters have been moving for engineers since April 1968. Now we are not, of course, talking about engine development—though I am sure that more unmanned engine rooms are in building now than were contemplated. then. We are not even talking about horsepower, though I am sure what was considered a riskily high level of output in early 1968 is commonplace today—just as the 250 000-ton tankers are old hat. Why, they are even being delivered now, so it is no longer smart to have one on order. No, I am talking about the organizational developments within the profession —whether one is to be a chartered engineer or not, and whether it matters.

I am sorry. I ought not to have allowed myself to get so far without recognizing that, as usual, we will have a two, to one ratio of guests to members here, and our guests will be wondering how long I am going to go on about Institute business. To them I must say, "Forgive me"—there are one or two things which must be said, even on an occasion like this, because if they are not said now, an opportunity will be lost of telling some of our members, and some of our fellow Institutes, and the Council of Engineering Institutions, or C.E.I., what we feel about things.

Three points, then.

First, our constitution. The developments within C.E.I., which change almost daily, underline our conviction that it is essential that our Institute should provide satisfactorily for both the chartered engineer and the technician. If our constitution is approved by the Privy Council, both are to have rights in the running of our Institute, both will have titles. The technologist will be "fellow" and the technician "member", and if what we would like to see achieved by C.E.I. comes about they will also have the names of Chartered Engineer, Technician Engineer, or Engineer Technician, all with suitable designatory letters. This will only be possible if people within C.E.I. are prepared to reconsider decisions taken many years ago at a time of very different conditions prevailing from those of today. I ask those who are concerned to think again and thus try to provide a structure for the engineering profession which is simple, logical and clearly understandable to all. We have always recognized the part which both chartered and non-chartered engineers have to play in marine engineering and our responsibility as an Institute to both. Again we recognize the value to the marine engineer himself and to the profession of having a ladder of advancement. We shall continue to do all that we can to persuade C.E.I. to proceed along the shortest path to achieve these goals.

Second, if we are going to see a State Register of Engineers, in addition to registers maintained by the institutes and by C.E.I., we are going to have to work hard to demonstrate to our members, and even more to potential new members, that membership of our Institute in addition to

State registration is going to be worth the price—by seeing that the value of the updating of professional knowledge which we can provide really means something to them.

We must make sure that the quality of our learned society function, our papers, journals, our periodical conferences, our library facilities, is really what our members need and gives them an added dimension to their professional knowledge which they can feel will add to their capabilities, and improve their performances and prospects. If we do not they will not join us. It will be a definite challenge to us.

Finally, the best way we can demonstrate to members that membership is worth paying a subscription for is to persuade the employers that they are better off with engineers who are members of our Institute than those who are not members of an institute at all. I feel that this is a very exciting prospect, because it does mean that we must find out from employers what they would like us to do for our members to make them more valuable as employees than outsiders. And then we must try to provide what is required. This really would provide an incentive for marine engineers to join the Institute and give yet another opportunity for fulfilment to the Institute itself.

I am sorry that I have been perhaps too serious. But I would like to add one more serious note, to say how much I, as a non-engineer, have enjoyed my two years as President, including the immensely stimulating IMAS Conference and I am taking the opportunity of saying this here now, because, as I think I hinted in my remarks here last year, there is just a possibility that there are one or two more members present tonight than there will be at the A.G.M. when I shall be thanking you for having me, formally.

Do you remember how, in the Kipling "Just So" story, "How the elephant got his trunk", the result of the intense agony of the elephant's child when he had his nose pulled for a whole day by the crocodile from the great, green, greasy Limpopo River was the creation of the beautiful long and useful trunk with which he was able to pay back the unkindnesses of his relatives on his return through the jungle? When in April, you respond to my cry like that of the elephant's child, "Let go, you're hurting me", and set me free from my presidential duties, I have no doubt that what I have learnt from the Institute will be for me every bit as valuable as the trunk was to the first young elephant to have one. (*Applause*)

Marine Engineering and Shipbuilding Abstracts

No. 7, July 1970

The Situation After Ten Years of Marine Automation

In addition to the obvious cost savings resulting from reduced manning, it is claimed that automation affords further financial benefits in respect of lower fuel costs, lower maintenance bills and fewer machinery breakdowns. In these areas it is very difficult to quantify the savings which may result from automation. In respect of fuel costs however, it is clear that on a motor ship the close control of fuel viscosity, fuel valve coolant temperature and early detection of poor valve action will all make a contribution to fuel savings. These control facilities are essential on engines burning heavy fuel and it is in this respect that they contribute to a major cost saving. On a steam ship the savings can be much greater, as fuel rate is highly dependant on operating the plant at optimum design conditions.

Automation can contribute to reduced maintenance bills by ensuring that the machinery is run at its design conditions, by warning of deleterious conditions, and by producing accurate records essential to planned maintenance. Again, the savings attributable to automation are not easily quantified, and the only valid evidence would be the long-term comparison of similar ships, with and without automation, operated by crews of equal competence, and on the same or similar service routes.

The fitting of automatic control equipment imposes a further burden on the maintenance load, as the control equipment itself must be maintained. The situation is further complicated by the fact that machinery installations have advanced in complexity in parallel with the fitting of automatic controls. On some medium-speed-engined ships the increased cost of maintaining the engines has completely swamped any maintenance saving contributed by automatic controls, particularly where the engines are operating on heavy fuel.

Regarding the claim that automation reduces machinery casualties, it is clear that a highly reliable alarm system and control equipment to reduce speed or stop the engine in the event of serious malfunction, will substantially reduce the damage resulting from failures in critical services such as lubricating and cooling systems. On motor ships the most essential safeguards additional to normal temperature, pressure and level alarms are scavenge fire and crankcase oil mist detection, H.P. fuel line leakage, and turbocharger vibration monitoring equipment in the case of the largest engines. It will be many years before statistical evidence becomes available to assess the savings resulting from the provision of such safeguards, but most shipowners regard this expenditure rightly as an insurance premium.—*The Motor Ship, February, 1970, Marine Automation Section. Vol. 50, pp. 3-5.*

Thermally Coupled Turbines

There are now a dozen large and fast 80 000 shp twin-screw ships on order for the Europe to Far East container trade and eight of 120 000 shp for Sea-Land's operations. The schedule of the Far East ships is likely to involve non-stop runs at constant power for periods of 10 to 12 days, burning 500 tons of fuel per day, and bunkering at Panama.

An interesting proposal for powering vessels in this category has been made by the General Electric Co. of U.S.A. Where machinery has to run for long periods at constant speed there is a case for a single, cross-compound plant in which the H.P. and L.P. turbines each develop identical power at the design point and each drives one of the propellers. Such a plant then reduces to a single 80 000 shp twin-screw cross-compound turbine requiring only two turbine casings and two single-input locked-train double-reduction gears. Since no mechanical coupling is required between the turbines and only one input is involved the speed reduction could well be by compound epicyclic gears. Reheat clearly comes into its own in such an application and G.E. calculations have shown that with steam at 1500 lb/in^2 and 950°F, with reheat to 950°F, an sfc of less than 0·4 lb/shp h is possible.

It might be thought that such an arrangement would be very inflexible in operation but it is worth noting that a reduction in power to 80 per cent of the total design figure would result in a load transfer from the L.P. to the H.P. line corresponding to less than two per cent of shaft speed. By adjusting the reheat temperatures this could be less than four per cent of shaft speed difference down to 25 per cent power, a figure unlikely to be much used in the type of ship under discussion except perhaps in fog.

Provision would have to be made on each turbine rotor for astern running but it is felt that the time has come for a drastic review of the astern power requirements of the classification societies and statutory bodies. In ships of this type there is a limit to the astern power which can be used effectively and this is generally well below the actual amount specified. A concession in this direction should be obtained, particularly if some form of hydrodynamic brake, such as the twin-rudder Sperry system were adopted for taking way off the ship rapidly with, possibly, the provision of powerful bow thrusters. Such a ship would be able to keep out of trouble much more effectively than one with a high ratio of astern power.—*Marine Engineer and Naval Architect, February, 1970, Vol. 93, p. 75.*

Decca Equipment in Large Esso Tankers

The four 253 000 dwt tankers for Esso Petroleum Co. Ltd. which are being built in the U.K.—two each by Swan Hunter Shipbuilders and Harland and Wolff—will have a Decca ISIS-300 system to provide surveillance of 120 points on the main machinery, cargo pumps and electrical generating equipment. The unique local scanning facility of the Decca ISIS-300 system is of maximum advantage in very large vessels, since the transducer cabling terminates at six local 20-way scanners, so disposed in the engine-room, to reduce the length of these leads. The scanners can operate independently and so give the system a much higher integrity where one central scanner is used. The local scanners are connected by a single multi-core cable to the central processor in the machinery control room. The control panel of the central processor with its digital readout display is remotely located on the centralized control console, and selector switches enable the operator to observe and monitor all the machinery conditions which are sensed by the system. Ninety of the parameters are, in addition, checked for high and low alarm limit conditions and the results displayed on two main alarm display units. A typewriter produces routine log sheets at three or six-hourly intervals and also records all alarm conditions.

In the engine-room the system will process all the important temperatures and pressures in the boilers and closed feed system, turbine and gear bearing temperatures, and some specialized measurements such as shaft speed, ship speed, fuel rate and main steam flows. The bearings of all the cargo, stripping and ballast pumps will be measured and checked for high temperature. The transducers for the pump bearings are located in the pump room and it was necessary to ensure intrinsically-safe circuits for these transducers.—*Marine Engineer and Naval Architect, February, 1970, Vol. 93, p. 73.*

Fishing Vessels

The modern fishing ship is more complicated and, has become ton for ton, very much more expensive to build and equip than most other types of merchant vessels. During the past 20 years and at an accelerating pace during the decade of the 1960s it has gone through a near revolution in design and in operational use. The once almost standard single-deck wet fish trawler, restricted in range by the time she could preserve her fish in ice, has been transformed into the two-deck ocean-roaming refrigerated stern trawler.

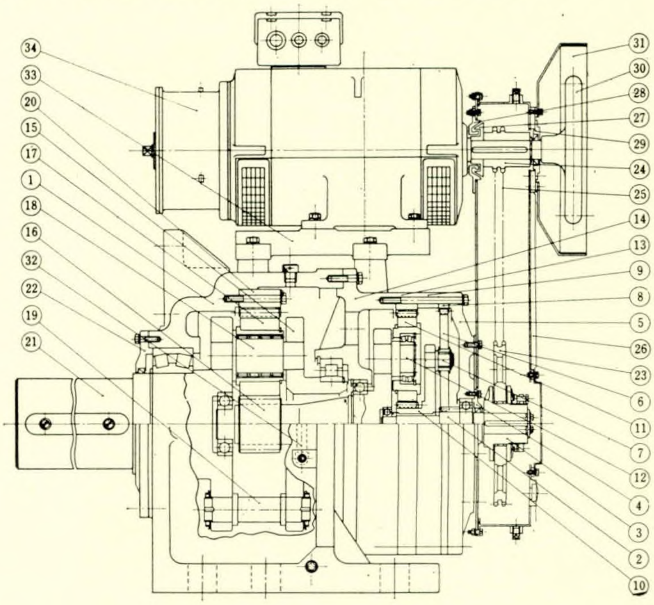
Through the introduction of the hydraulic power block able to handle synthetic fibre nets which could envelop a cathedral, the purse seiner has grown in some fleets from a boat around 60 to 80ft into a small ship of 140 ft and longer. Factory ships 600 ft length and larger are in operation or under construction. The American west coast pole fishing tuna clipper has been replaced by purse seiners of 1000 tons.

New materials such as glass fibre reinforced plastic and ferro-cement, are now being regarded as practical replacements for wood and steel for the smaller boats, and glass fibre may eventually be used for trawlers and purse seiners larger than 100 ft.—*The Motor Ship, February, 1970, World-Wide Fishing Section. Vol. 50, pp. 1, 2, 5, 6, 9.*

Turning Gear for Large Engines

Conventional turning gear using spur gearing in cascade is bulky, heavy and requires a powerful and expensive motor. Toyo Seimitsu Zohki Co. Ltd. claim that, by using planetary gears in series, their IMT-type turning gear for a 20 000 bhp Diesel engine is only one quarter the weight of a conventional gear of comparable capacity.

High transmission efficiency, about 87 per cent, is attained for a reduction ratio of about 1 to 1200 by virtue of the accurately cut and heat-treated teeth and the use of ball or roller bearings. Consequently the motor can be much smaller. All the rotating parts are enclosed in an oil bath and no lubrication is required except during periodical inspections.



1) Reduction gear housing (third stage)
2) High speed pinion
3) Torque-limiting clutch
5) End cover
6) 1st Planet gear
8) 1st Annulus gear
9) Spacer
10) 2nd Stage pinion
11) 2nd Planet gear
12) 2nd Planet spindle
13) 2nd Annulus gear
14) Housing (1st and 2nd stage) and intermediate bearing support
15) 3rd Stage planet carrier
16) 3rd Stage pinion
17) 3rd Stage planet gear
19) Spacer
20) 3rd Stage annulus gear
21) Output shaft
23, 24, 25) Chain transmission
26) Chain case
30) Handwheel

Longitudinal section through the three-stage epicyclic reduction train

The above figure shows that the motor is mounted over the reduction gear case and drives the primary sun wheel, by chain, through a torque-limiter. There are three planetary gear trains in series with progressively wider gear faces and heavier tooth profiles. The final stage planet wheel carrier is exceptionally robust. Ten models in the range cover output shaft torques from 100 kg/m (725 lb/ft) to 9000 kg/m (65 700 lb/ft). The largest, powered by a 15 kW motor, weighs only 2150 kg and is 1·64 m (5 ft 6 in) in overall length to the extreme end of the turning pinion shaft (this length includes the free length for accommodating the pinion in the disengaged position). The base plate is 1 m wide (3 ft 3 in) and the overall height is 1·35 m (4 ft 5 in). The gear can be fitted with a brake if required.—*Marine Engineer and Naval Architect, March 1970, Vol. 93, p. 132.*

Autopilots and Steering Problems of Ships

In the past, owing to deficiencies in mathematical representation and hydrodynamic techniques, autopilot control systems design has been intuitively and experimentally based.

The recent marriage of interests of hydrodynamicists and control engineers now provides an analytical basis. The previous generation of autopilots is unsuitable for the ever growing fleets of large tankers, many with poor steering characteristics. Important economical operation of giant ships is forcing the designer to define operational requirements and specify more closely achievable-system performance. This in turn involves acquisition of ship and hydrodynamic data, but important parts of such data are now generally available at the ship design stage. One solution is to incorporate in the autopilot sufficient parameter flexibility to meet the requirements of a range of ship forms. When data are available, it is possible to predict system performance and recommend parameter settings.—*Tomlinson, P., Shipping World and Shipbuilder, March 1970, Vol. 163, pp. 411–412.*

Improving the Manoeuvring Performance of Diesel-powered Vessels

Vessels with fixed-pitch propellers propelled directly by Diesel engines or through gear units usually show good manoeuvring performance which, however, deteriorates with increasing engine speed and higher degree of turbo-charging.

Prior to M.A.N.'s first highly turbo-charged VV 40/54 engines being commissioned to service in fixed-pitch propeller layouts, these problems were thoroughly examined and trials carried out with various braking systems on three vessels of the same design.

This revealed that the engine delivers a high braking moment if compression work can be used for braking, reducing at the same time expansion work as much as possible. In the system used so far, the camshaft is shifted into reversing position after the fuel to the engine has been cut off ('no fuel' position) and the main starting valve is opened as soon as engine speed has been decelerated sufficiently. In the meantime, the trailing propeller is rotated by the flow of water and rotates the engine crankshaft in ahead direction. Owing to the fact that the camshaft is already in the reverse direction, compressed air from the starting air bottle enters the cylinder during the compression stroke through the opened starting air valves and brakes the up-stroke of the piston. Once the pressure in the cylinder exceeds the starting-air pressure, part of the air passes back into the starting air pipe through the starting-air valves so that during the following expansion stroke only the air that has remained in the cylinder performs expansion work.

With the new system, to begin with only a braking valve is opened. Compressed air can now reach the cylinder starting-air valves through the starting-air pilot valve, forcing them open during the compression stroke. The master starting valve remains still closed. Since, as opposed to the previous system, the starting air pipe is unpressurized, considerably more air from the cylinders can leave through the starting air pipe *via* the venting valve during the compression stroke and, in particular, shortly before the piston reaches T.D.C. where the cylinder pressure is at its highest and where the piston makes hardly any movement. During this time the air pressure in the cylinder drops considerably and the following expansion stroke starts on a much reduced pressure level so that the remaining expansion work is only small. The main starting valve opens only after engine speed has dropped to zero, and the engine is started in the reverse sense of direction so that ship motion is rapidly decelerated owing to the fact that the propeller turns in the reverse direction.—*Holland Shipbuilding, January 1970, Vol. 18, pp. 37–38.*

Twin-screw Landing Craft

The twin-screw landing craft *Rabha* was built by the Arnhem Shipbuilding Cy. for the ruler of Qatar. After successful trials it has recently been delivered to its owners.

Rabha is especially conceived for the transport of trailers and other rolling equipment from and to Doha, the capital of Qatar, in the Persian Gulf. The ship has been designed by and built under survey of Dutch Dredge Consultants of Dordrecht.

For loading and unloading cargo directly onto and from the beach *Rabha* has a flat landing ramp which is controlled from the forecastle by two electric tackles.

The ship can supply fresh water, fuel oil and equipment to oil rigs and other craft working offshore. For this purpose she has two Samson posts with two 5-ton derricks and electrically driven cargo winches.

In addition to the two electric driven anchor winches on the foredeck there is an electric anchor-mooring winch on the aft deck with two warping barrels and a main drum suitable for 200 m of anchor cable with a diameter of 1 in.

The boat-deck has been provided with a motor launch with a Diesel engine of 16 hp and a crane to swing this launch outboard.

In the engine room have been installed two main Paxman engines. Each develops 337 hp at 1200 rev/min, with reduction gear-box. Reduction to the propellers is 3 : 1. Furthermore there are two auxiliary Ruston engines, each developing 52 hp at 1800 rev/min. They each drive an AC generator of 40 kVA and the necessary bilge, ballast, fuel, fire-fighting, freshwater and sanitary pumps. The accommodation aft is suitable for a crew of 10 men.—*Holland Shipbuilding, January 1970, Vol. 18, p. 38.*

Transportation of Smaller Vessels by Sea-going Ships

Two river tankers were transported from Zhdanov (on the Sea of Azov) to Ust-Kamchatsk (on the eastern coast of Siberia) by the ocean-going collier *Dubno,* in 1967. The U.S.S.R. register gives the particulars of this vessel as length o.a. 139·5 m, breadth o.a. 18 m, depth 10·3 m, deadweight 9750 tons.

Principal particulars of the tankers are:

Length, o.a.	50 m
Breadth	8·84 m
Depth	1·8 m
Lightweight	130 tons (approximately)

The two tankers were loaded and unloaded by *Dubno* without the use of cranes. *Dubno* was anchored fore and aft and fitted with a removable ramp arrangement, made up of individual ways, which extended over the ship's side at an angle of 16° to the horizontal. The bottoms of the tankers were each fitted with three welded sliding carriages. *Dubno* pumped ballast to its tanks along the side the ramp was fitted so as to bring the outer edge of the ramp to a depth greater than the draught of the tankers (angle of heel 12–14°). One of the tankers was then brought alongside the ramp so that its sliding carriages connected with the ways. Cables had already been connected from the lugs of the bow and stern sliding carriages to quadruple-purchase tackles *via* guiding rollers. The running ends of the tackles were taken on to a capstan. After being positioned, the tanker was partly drawn up on to the ramp, which at this time was at an angle of 28–30° to the horizontal, by the capstan. When only the bilge and part of the outboard side of the tanker remained in the water, *Dubno* was righted by pumping the ballast to the other side. After this the tanker was drawn right up on board *Dubno* and secured on deck. The ramp arrangement was then unshipped and erected on the other side, and the second tanker was taken aboard in the same way. The two tankers were then lashed together and the ramp arrangement unshipped for the voyage.

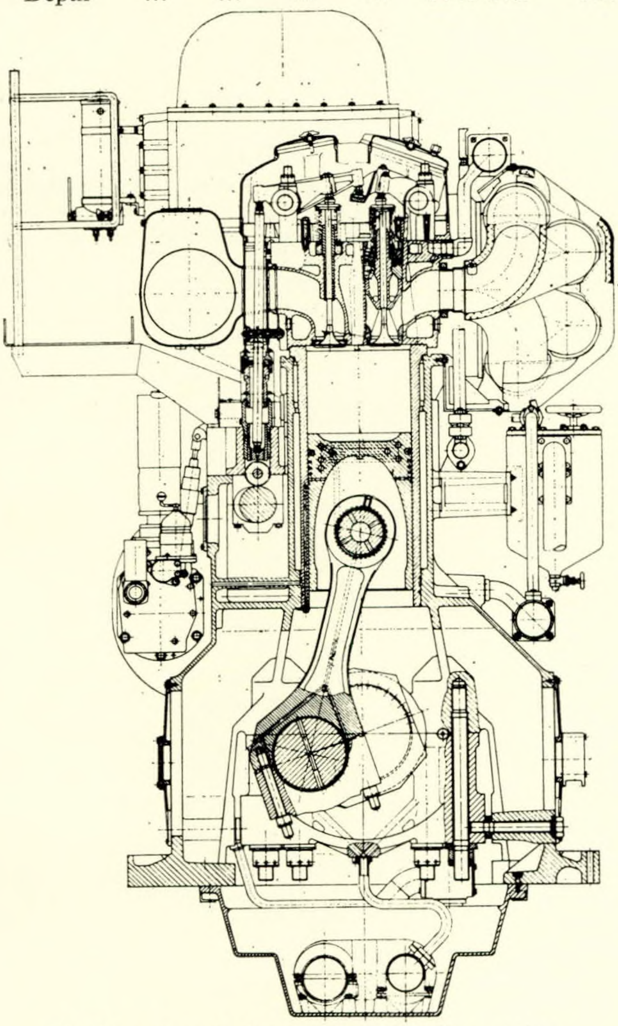
When their destination was reached, the two tankers were launched by reversing the loading operation.—*Tokarev, B., Morskoj Flot, 1969, No. 8, p. 13–14; Jnl Abstracts B.S.R.A., November 1969, Vol. 24, Abstract No. 28 399.*

Deutz Engine in Powerful Clyde Tug

Steel and Bennie Ltd., Glasgow tugmasters, have taken into service the new 2400 bhp *Warrior* for ship-handling duties. She is the most powerful tug operating on the Clyde, her power reflecting the requirements of the giant tankers and high freeboard container ships now entering the Firth. *Warrior* has been built by James Lamont and Co. Ltd. of Greenock, who were also responsible for the installation of the machinery.

Principal particulars are:

Length, b.p.	112 ft 0 in	34·13 m
Breadth	30 ft 0 in	9·14 m
Depth	14 ft 0 in	4·27 m



Section through the Deutz 540 in-line engine

On trials *Warrior* attained a speed of 13 knots and achieved a bollard pull of 38 tons. New in a Clyde ship-handling tug is the 30-ton 300 fathom hydraulic towing winch which can be operated from several positions, including the wheelhouse. The anchor windlass is also hydraulically powered.

Propulsion is by a Deutz uni-directional SBV6M540 engine developing 2400 bhp at 600 rev/min, driving a fixed pitch propeller working within a Kort nozzle rudder, through a Spiroflex coupling and Type GUW638 reverse and 3·5 to 1 reduction ratio gear.

The Deutz SBV6M540 engine has a cylinder bore of 370 mm ($14\frac{9}{16}$) piston stroke of 400 mm ($15\frac{3}{4}$ in) giving a cylinder displacement of 43 litres (2626 in^3). At the maximum rating of 2400 bhp and 600 rev/min the b.m.e.p. is 13·95 kg/cm^3 (198·4 lb/in^2) with a mean piston speed of 8 m/sec (1575 ft/min). This is a cast iron framed four-valve four-cycle engine with underslung crankshaft (new to the larger Deutz engines) oil-cooled pistons and individual fuel pumps. The crankshaft is carried in shell bearings in caps secured by four hydraulically-stressed studs, all on the transverse centreline.—*Marine Engineer and Naval Architect, March 1970, Vol. 93, pp. 122–123.*

Special Products Tanker with Three Pump Rooms

The 23 580 dwt tanker *Anco Swan*, built by Eriksbergs Mek Verkstads AB for I/S Saga Swan (Iver Bugge of Larvik) has the cargo space divided into 35 tanks with a total capacity of 30 407 m^3. The tanks are specially coated for the different cargoes and the tank washing plant consists of a 180 m^3/h pump, two steam heaters, and six portable rotary washing machines. The three main pump rooms, arranged between the longitudinal bulkheads, contain ten cargo pumps, each with a water rate of 250 m^3/h. Dimensionally the ship is similar to the BP, Moss and Athel Line products-carrier built by the yard and the associated Uddevallavarvet. A dry powder fire extinguishing plant has been installed for the tank deck, cargo tanks and bunker tanks. The propelling machinery comprises a seven-cylinder Eriksberg-B and W engine of 11 500 bhp. Engine room automation is according to the EO requirements of Det Norske Veritas, allowing the space to be unattended for periods up to 24 hours. The steam generating plant comprises three oil-fired boilers with a total capacity of 18 tons per hour, supplying steam to deck machinery, cargo pumps, bilge pumps and heating coils in cargo tanks and bunker tanks for heavy oil.—*Marine Engineer and Naval Architect, March 1970, Vol. 93, p. 136.*

Yugoslavian-built Car Bulk Carrier for Norwegian-U.K. HBS Group

The 25 550 dwt *Borgestad* serves as a good indication of the increasingly sophisticated class of ship being produced by Yugoslavian yards.

Borgestad is distinguished by being the first of a class of three car/bulk carriers for the HBS Group—a Norwegian/British consortium comprising Yngvar Hvistendahl, A/S Borgestad and Silver Line Ltd. These three ships each have a comprehensive system of centralized automated control for the main and auxiliary machinery and the controllable-pitch propeller, and each is equipped with the Blohm and Voss folding design of car decks.

Borgestad has been built by the Uljanik Shipyard, at Pula, which is also building the two sisterships, *Silvermain* and *Milena*. She is of all-welded construction and has a bulbous bow, and transom stern.

Principal particulars are:

Length, o.a.	500 ft 2 in
Breadth	85 ft 3 in
Draught (summer)	36 ft $4\frac{3}{8}$ in
Corresponding deadweight ...	25 550 tons
Cubic capacity	1 012 211 ft^3
Car capacity	about 1900
Service speed	15 knots

Powered by an Uljanik–B. and W. 6K74EF engine developing 11 600 bhp the vessel is also equipped with Yugoslavian-built auxiliary engines; in fact much of the engine room and deck equipment was constructed under licence in that country.

The main engine is reversible and drives a KaMeWa c.p. propeller, control of the engine speed and propeller pitch being arranged from the bridge and from a machinery control room. Automatic load control of the main engine is provided and overspeed protection is afforded by a Woodward governor.—*The Motor Ship, March 1970, Vol. 50, pp. 585–587.*

Specialized Ore and Container Carrier

One of the more interesting combination types of ship which will be commissioned this year is the 12 100 dwt ore/container ship *Darwin Trader* at present fitting out at the Newcastle, N.S.W., yard of the State Dockyard, Australia. The design of this vessel is very similar to that of an ore/bulk carrier except that those holds which would normally be empty when carrying ore will in fact be loaded with empty containers.

Thus, *Darwin Trader* is being constructed to the requirements of Lloyd's ✠100 A1 clasification with the notation "Strengthened for ore cargoes—alternative (or specified) holds may be empty".

Principal particulars are:

Length, overall	458 ft 0 in
Breadth, moulded	70 ft 0 in
Depth, moulded	47 ft 6 in
Loaded draught (containers) ...	23 ft 6 in
Loaded draught (ore)	30 ft 0 in
Max. deadweight (approx.) ...	12 100 tons
Ore capacity	10 720 tons
Main engines	2 × S.E.M.T.–Pielstick 16 PC2V
Total m.c.r.	13 120 bph
Service speed	15·5 knots

Since the vessel has a dual role the hull design does not incorporate the double hull associated with many cellular container ships. Instead, the double-bottom margin plates are carried up to the shell plating to form hopper sides at the bottom of the holds, extending from the forward end of No. 2 hold to the aft end of No. 7 hold. In addition, these hopper sides provide extra water ballast capacity for stability purposes when carrying loaded containers only and for good propeller immersion while in ballast.

A Flume system of passive tank stabilization is provided and consists of separate upper and lower tanks situated forward of midships. It is anticipated that with this system a reduction of roll in the order of 75 per cent will be attained in the sea conditions normally encountered on the service route. To aid manoeuvring in port, a Vickers-type bow thruster of about 7·5 ton thrust is fitted and the 5-ft diameter propeller is driven by a 700 hp electric motor.—*The Motor Ship, March 1970, Vol. 50, p. 598.*

Ship Manoeuvres in Harbours

The paper begins with a general outline of the basic elements of harbours and their design problems, which are interrelated with ship draught and maneouvring ability. Some rudimentary percentage capital-cost breakdowns are given for four widely-differing harbours.

Theoretical methods of analysing ship manoeuvres in harbours and approaches are impracticable owing to the many independent parameters affecting a ship's movement. The author considers a new "graphical method", devised at the Harbour Works Laboratory of the Technical University of Denmark in collaboration with A. Sørensen and other pilots, to be the most convenient in use. A description of this method is given, and illustrated by examples of berthing or departure manoeuvres at five harbours. There is also a table showing approximate values of the various forces to be considered when manoeuvring a loaded 120 000 dwt tanker.

Finally, the author suggests some ship improvements to ease the difficulties involved in berthing; these include bow thrusters of adequate power in relation to the ship's size, improvement of stopping ability (especially in the case of large turbine tankers, for which c.p. propellers are suggested), and improved mooring equipment for large ships—*Paper by Svendsen, I. A., presented at the Thirteenth Scandinavian Ship-Technical Conference, Trondheim, 1968; Jnl Abstracts B.S.R.A., December 1969, Vol. 24, Abstract No. 28 459.*

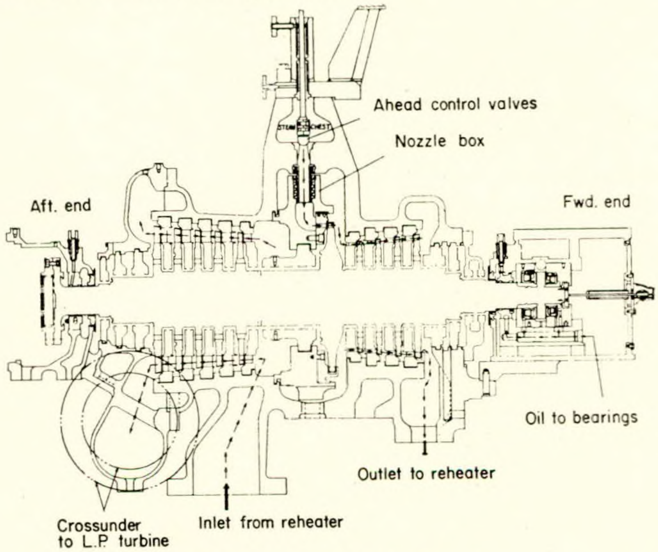
Marine Turbine Propulsion

The modern steam turbine for ship propulsion is an advanced and highly developed form of prime mover ideally suited and adapted to the marine environment. It can be readily modified and tailored to meet a variety of applications from the high speed container ships to arctic tankers.

The cross-compound turbine arrangement is universally used for ship propulsion for all but the lowest power levels. The figure shows the cross-section of a high pressure turbine with reheat. Steam at initial conditions of 1450 lb/in²g and 950°F is admitted through valves which open sequentially and which are located at the middle of the casing. The steam flows in a forward direction through the high pressure section to the outlet to the reheater. The steam returning from the reheater at 950°F enters again at mid casing and flows in the aft direction to the cross-under. Special features to be noted in this cross-section are:

1) separate nozzle box with centre-line support;
2) wheel and diaphragm construction;
3) solid one-piece rotor.

While reheat is the newest entry in steam plant designs and is uniquely suited to large, high utilization ships where fuel costs are unusually important, non-reheat alternates will continue to find wide application.



Reheat high-pressure turbine

The principle of external combustion and the relatively modest temperatures of the steam plant cycle permit the use of standard well-behaved materials in the steam path. Because the steam plant operates as a closed cycle plant, a high purity level of the working fluid can be maintained and corrosion pitting and foreign particle erosion can be essentially eliminated. Thus the contours and the surface finishes of the steam path should remain in good condition indefinitely, i.e. for the life of the equipment, thereby maintaining good thermal efficiency.

Water droplet erosion on last stage buckets has not been a problem to date. Bucket vibrations must be carefully controlled if long life is to be consistently achieved in the steam path elements; it has been one of the most difficult problems facing the designer of all kinds of turbo-machinery and in the marine turbine with its variable speed operation the problem takes on an added dimension.

The design of last stage buckets to prevent trouble due to "low-per-rev" vibration is an excellent example of the need to design to marine conditions and not to extract rules from the electric utility or industrial experience.

The land bucket operates at constant speed and is quite slender because it has been tuned to avoid the dangerous "low-per-rev" resonances. The marine bucket, which must be capable of operation at all speeds up to the maximum, is made stiff and strong so that vibration stresses at the unavoidable resonances will not be high enough to cause fatigue.

A totally different problem is presented in the design of first stage bucket. There the vibration excitation comes from the once-per-revolution loading and unloading of the buckets due to the partial arc and the buffeting action of high velocity, high density steam. It was not until fifteen to twenty years ago that a general understanding of this particular type of bucket vibration problem began to develop. It took a rather sophisticated mathematical analysis to properly assess the problem and with the advent of the modern digital computer, design calculations became practical. About a decade ago a new system for the design of buckets was put into operation which combined theoretical calculations with empirical data derived from extensive experience.—*Bowden, G. J., Schiff and Hafen, 1970, Vol. 22, No. 2, pp. 158–161.*

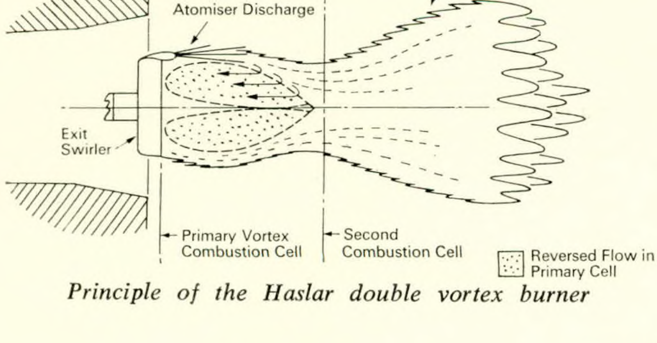
Novel High Density Burner

Engineers at the Admiralty Marine Engineering Establishment at Haslar, Gosport, who are responsible for developing high performance oil fuel burning equipment for naval boilers, have invented a method of introducing fuel to the combustion air stream in such a way as to achieve intense combustion which can be precisely controlled.

The conventional arrangement is to deliver combustion air to the furnace through a register with a centrally mounted flame-stabilizing swirler in the discharge; normally, the fuel is injected as a diverging spray, by a single-orifice atomizer head on the swirler axis, and passes through the recirculation zone induced in the air flow pattern. With such equipment, only a small proportion of the total fuel circulates and burns in this zone and the flame shape follows the development of a single cone.

With the present invention, the fuel is injected from individual heads, located around the recirculation zone and directed, convergingly, to circulate and burn in this flame-stabilizing recirculation zone. The additional energy added by the twin-fluid injection helps to alter significantly the flame development. In this mode of operation, the recirculation zone becomes a primary combustion cell with a well-defined boundary dividing it from the second cell, in which combustion is completed. Since relatively precise control can be established over the first cell, by co-operation of aerodynamic means and twin-fluid injection, the invention offers the potential of positive fine control of combustion in the critically important initial stages of flame development. A particular feature of performance of the equipment investigated at Haslar has been the order of intensity of combustion realized within the first cell, estimated to be as high as 10×10^6 Btu/ft^3/h.

One form of the invention is shown in the figure. This shows fuel atomizers mounted on the exit swirler shroud and firing axially, although the angle of discharge can be varied to optimize the balance between radial and tangential velocity.

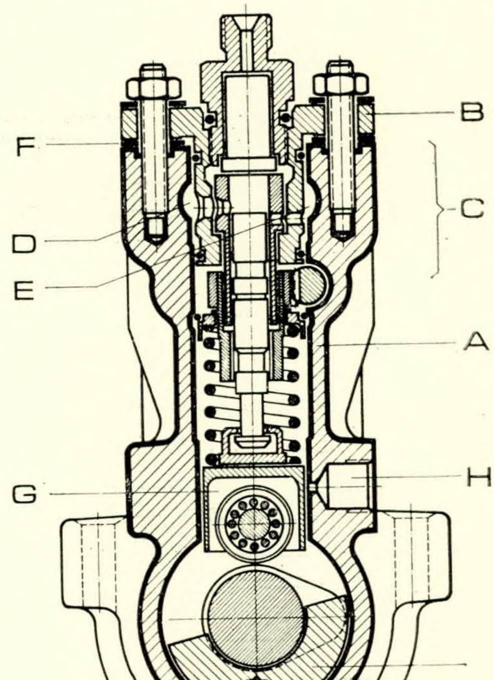


Principle of the Haslar double vortex burner

The invention, therefore, covers an air register for fuel burning equipment of the suspended flame type including means of introducing fuel to the combustion airstream positioned at or about the location of the exit swirler and displaced away from the longitudinal axis of the air register. Development work has been carried out by AMEE and some test results are available.—*Marine Engineer and Naval Architect, March 1970, Vol. 93, p. 118.*

Austrian Fuel Injection Pumps

Friedmann and Maier AG, of Hallein/Salzburg, have introduced a new graded series of pumps suitable for a wide range of engine powers. All have aluminium housings containing steel sub-assemblies which comprise the entire high-pressure elements with their delivery valves and pipe connexions. These elements, secured individually within the aluminium housing by two studs, have lapped metal-to-metal joints, resulting in minimum dead volume, but sealing between the high pressure and low pressure bodies is effected by means of O-rings. Camshafts, taper roller bearings and tappets are generously proportioned for long life and heavy loading, and a centre bearing is provided in the larger sizes.



A Light alloy body
B Stee cover and branch carrier
C High pressure assembly with delivery valve
D Inlet port
E Overflow passage
F Shims for effective stroke adjustment
G Needle roller tappet assembly
H Lubricating oil connexion from engine

Section through P-type pump showing how cover is extended downwards to support the barrel

Features of the design are barrels with separate suction and spill ports (bi-flux) and delivery valves in which the valve body, plunger and spring form a complete unit. The plungers have the usual helical control edges and are rotated by means of toothed quadrants and a rack. The elements can be individually calibrated from outside by rotating the steel insert and the effective stroke is adjusted by inserting shims below the flanges of the barrel carriers.

The cam lift of some of the types is much higher than is generally the case with pumps of comparable size, making it possible to achieve large delivery quantities or rapid delivery over short injection periods. Selection of suitable cam shapes, plunger diameters and types of delivery valve enable the injection pump to be optimized for the engine.

With the other types the fuel feed pump can be fitted at either end of the housing to suit drive from the left or right of the engine. All pumps can be lubricated from the engine system, in which case they need no attention.

The PO pump has been developed in co-operation with Professor Dr. Anton Pischinger of the Technical University, Graz. It is completely enclosed and, together with the governor housing and integral fuel feed pump, forms a compact unit. The three cylinder pump is 203 mm (8 in) in length, 92 mm (3⅝in) in width, 134 mm (5¼ in) in height and weighs 4·9 kg (12·1 lb). The pump is counterpart to the distributer-type models and is claimed to be less sensitive to inferior or contaminated fuel and simpler to service. Fuel leak-off is led into the pump casing, where it is used to lubricate the tappets and taper-roller bearings. Only normal fuel filtration is necessary.—*Gas and Oil Power, January/February 1970, Vol. 66, pp. 30–31.*

Bulk Carriers for the New 'Gearbulk' Consortium

Of the six bulk carriers on order for the new Gearbulk consortium, two have been built at the Saint-Nazaire yard of Chantiers de l'Atlantique. These vessels, *Alain L.D.* and *Robert L.D.*, have been delivered to S.A. Louis-Dreyfus and Co., Paris. The first of these, *La Pampa,* was launched on January 22 at the Harland and Wolff shipyard. This vessel will have a 24 800 dwt and two Munck deck cranes.

Principal particulars are:

Length, o.a.	498 ft 4¾ in	151·95 m
Length, b.p.	466 ft 8⅛ in	142·28 m
Breadth, moulded ...	74 ft 1½ in	22·60 m
Draught, maximum ...	31 ft 5⅝ in	9·59 m
Deadweight	19 110 tons (M)	
Displacement	25 045 tons (M)	
Gross tonnage	12 705·31	
Hold capacity	808 894 ft³	22 894 m³
Machinery output ...	10 940 shp at 120 rev/min	
Service speed	16 knots	

The propelling machinery in *Alain L-D* and *Robert L-D* consists of two SEMT-Pielstick type 12 PC 2 V 400 engines of 11 160 hp output (max) at 450 rev/min. These are four-stroke turbocharged vee-type 12-cylinder engines, driving a single shaft through a Renk 450/120 rev/min reduction gearing. The four-bladed ACB controllable-pitch propeller is of cn/al alloy and has a diameter of 5·80 m.

The propelling machinery can be controlled either from the bridge or from a special compartment in the engine room. From the wheelhouse the machinery is electrically controlled by a simple lever acting simultaneously on the engine speed and propeller pitch. Each engine can be stopped, started, engaged or disengaged from the shaft by means of a four-position changeover switch. From the control room the two engines and propeller pitch can be controlled separately by push buttons. Stopping, engaging or disengaging the engines from the shaft is by means of a three-position changeover switch. The remote control system has been designed by Chantiers de l'Atlantique with the co-operation of Soc. Télémecanique.—*Shipping World and Shipbuilder, March 1970, Vol. 163, pp. 427–428; 431–432.*

Dutch-built Vessel for German Owners

Motor vessel *Hede* is the third of a series of three open shelterdeck cargo vessels built by two Dutch shipyards for shipowners Dietrich Sander of Bremen, West Germany.

Hede has a flush main deck and 'tweendeck, a cruiser stem and stern. She is of generally modern appearance and a very practical type of coaster. Deadweight amounts to 1000 tons and the hold capacity is 64 000 ft³ in bales, 70 000 ft³ in grain or 36 ISO containers of 20 tons.

The ship was built to the rules and under inspection of Germanische Lloyd and classified ✠ 100 A 4 E2. It also complies with the regulations of the Federal German Association for Coastal Shipping.

Naturally a ship of this type has all machinery, crew accommodation and the bridge aft.

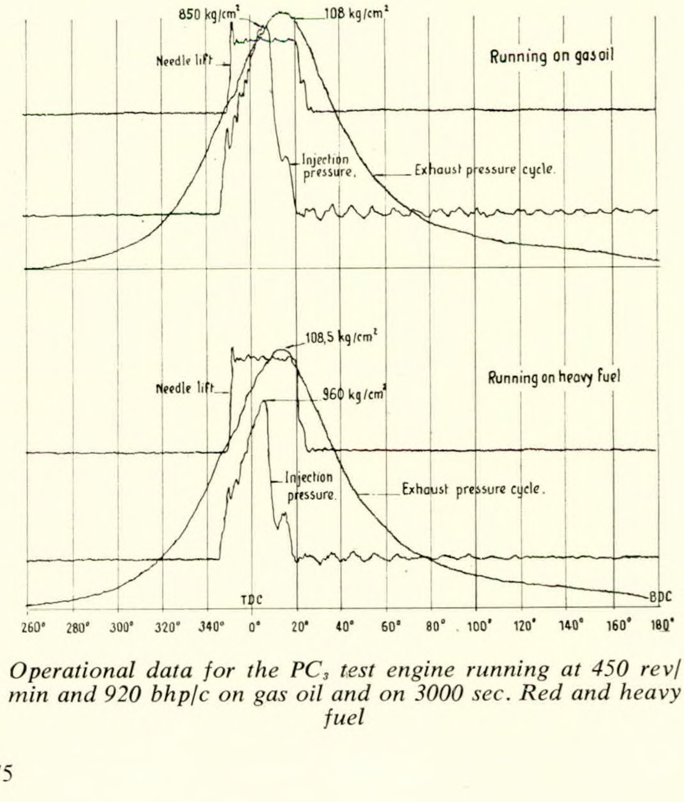
Principal particulars are:

Length, o.a.	57·16 m
Length, b.p.	51·62 m
Moulded breadth	10·00 m
Depth to shelterdeck	6·25 m
Depth to main deck	3·95 m
Draught (loaded)	3·91 m
Tons deadweight	1000
Tons g.r.t.	approximately 470
Cargo capacity	70 000 ft³
Propulsion engine	900 hp
Speed	11 knots

Power for propulsion is provided by an eight-cylinder, four-stroke MWM Diesel engine, type T6 D 440-8. It develops 900 hp at 750 rev/min and gives the vessel a speed of 11 knots. The steering gear is electro-hydraulic and operates a Hodi jet rudder. Hand steering in case of emergency remains possible.—*Holland Shipbuilding, January 1970, Vol. 18, pp. 27; 28; 30.*

Progress with the S.E.M.T. Pielstick PC3 Engine

The first PC3 engine was put into service on the Saint Denis testbed in October, 1967, and has so far completed 4500 running hours. The last 3000 hours of this period have been with the engine developing powers in the 900 to 950 bhp/cylinder range. Furthermore, the engine has been burning fuel with a viscosity of 3000 sec. Red. 1 at 100°F, for the latter 100 hours. Other properties appertaining to this heavy fuel are a sulphur content of 2·9 per cent, a Conradson index of 9·6 per cent, and a vanadium content of 80 ppm.



Operational data for the PC_3 test engine running at 450 rev/min and 920 bhp/c on gas oil and on 3000 sec. Red and heavy fuel

The number of running hours already obtained shows that any major problems of engine operational reliability are now entirely solved. The utilized powers are already in excess of the maximum continuous power provided for the initial rating and are close to the power intended for future uprating. Furthermore, a second engine—a 12PC3 unit—has been in service in Saint Nazaire for almost a year now and has confirmed the results obtained from the first test engine.

Endurance tests so far carried out have been of sufficient duration to allow conclusions to be made.

The heavy fuel combustion process in the PC3 cylinder is the same as the PC2 engine and the figure represents the pressure recorded in the cylinder at 920 bhp/cyl with the engine burning heavy fuel and Diesel oil. It will be noticed that the firing delays and the maximum combustion pressures are the same—the injection advances in the two tests, were similar. Furthermore, this power is developed at 450 rev/min, which means that with a mean effective pressure of 19·5 kg/cm^2 and a maximum combustion pressure of only 108 kg/cm^2, the mechanical loads are limited to reasonable values.

Injection pressures which have been recorded are a little higher with heavy fuel than with Diesel oil, but this can be explained by the difference of viscosity of the fuel at the injection pump inlet. The injection pressure at 920 bhp/cyl and 450 rev/min with Diesel oil, is in fact 850 kg/cm^2 compared with 950 kg/cm^2 when running with a fuel of 3000 sec. Red. 1 heated to 95°C and 900 kg/cm^2 when the same fuel is heated to 110°C.

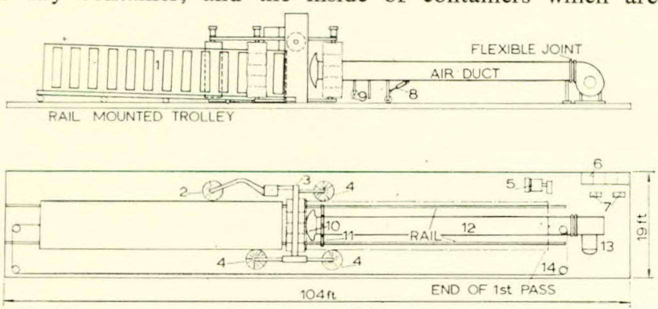
The condition of the injectors after 1000 hours in service is entirely satisfactory. During this period they have only been examined once, and were not dismantled since the atomization obtained at the test pump was normal. These results show that the interval separating injector inspection does not need to be reduced in spite of the higher mean effective pressures on the PC3 engine. On the contrary the interval may be increased as the tests continue.

This increase of inspection intervals is in fact the main conclusion drawn from the four-cylinder test engine as wear-down rates of components submitted to frictional action are approximately the same value, if not smaller, than those of the PC2 engines.—*Gallois, J., The Motor Ship, January 1970, Vol. 50, pp. 456–458.*

Container Cleaning Machine

The problem of cleaning containers, both inside and outside, is likely to increase in the future as an increasing amount of the world's merchandise is transported by this means. This problem becomes particularly acute when containers are required to carry successive cargoes of incompatible materials.

A machine has now been developed by J. S. and L. Birch (Engineering) Ltd. which is suitable for washing the outside of any container, and the inside of containers which are fully lined with glass-fibre. It is claimed that this machine, the principal features of which can be seen in the figure, can clean a 40-ft container inside and out in four minutes, and a 20-ft one in about three minutes.



1) Container
2) One ½ H.P. wrap-round brush
3) HP top brush
4) Three 1½ H.P. side brushes
5) 30 gal/min pump
6) Four chemical tanks
7) Two chemical dosing pumps
8) Support leg
9) Roller support
10) Air dryer nozzle
11) Cleaning nozzles on ring
12) Air duct
13) 20 H.P. fan for cold air
14) Container trolley capstan

Container cleaning machine

Once the container has been placed on the trolley by crane, it is drawn past five rotating brushes arranged as shown, so that the outside, including the ends, is scrubbed clean by the brushes after being washed by water sprays. The bottom is cleaned by high-pressure water jets. The inside is sprayed by means of a ring of high-pressure water jets, to which has been added measured quantities of an approved chemical cleaning agent.

As the container is withdrawn, the inside and outside are rinsed with fresh water and the inside is blown dry by cold air supplied by a 20 hp fan. This drying process is claimed to have an efficiency of 98 per cent.—*Shipping World and Shipbuilder, March 1970, Vol. 163, p. 419.*

Gas as an Engine Fuel

This paper discusses the properties required of a fuel gas, particularly in an engine, and indicates the advantages of natural gas compared with gases hitherto available. Special emphasis is laid on liquid natural gas which the authors believe will be of great importance in the future.

The development of the gas engine from early times is reviewed and the reasons for its decline in popularity are considered.

The rise in the dual-fuel engine in the sewage treatment industry and the modern gas engine in the U.S.A. are described and their relative merits compared. Brief reference is made to natural gas as a fuel for gas turbines and steam boilers also.

The gas and dual-fuel engines installed or on order by the North Thames Gas Board, of which the authors have personal knowledge, are described in some detail.

Experiments in the use of liquid natural gas as a fuel in the transport industry are outlined and the paper concludes with a review of some possible applications of liquid natural gas, including the use of the cold effect.—*Eke, P. W. A. and Walker, J. H., Trans.I.Mar.E., April 1970, Vol. 82, pp. 121–138.*

Oil Tanker Cargo Pumps

The last decade has witnessed many changes in oil tankers, the most remarkable being the three-fold increase in deadweight tonnage of the largest vessels afloat so that vessels of 326 000 dwt are in service, a 400 000 dwt tanker is under construction and 200 000+ dwt tankers are becoming almost commonplace. Another interesting field of development is in the product tankers where there has not only been a considerable increase in size but also a tendency towards more sophisticated cargo handling equipment enabling a wide variety of "parcels" to be carried at any one time.

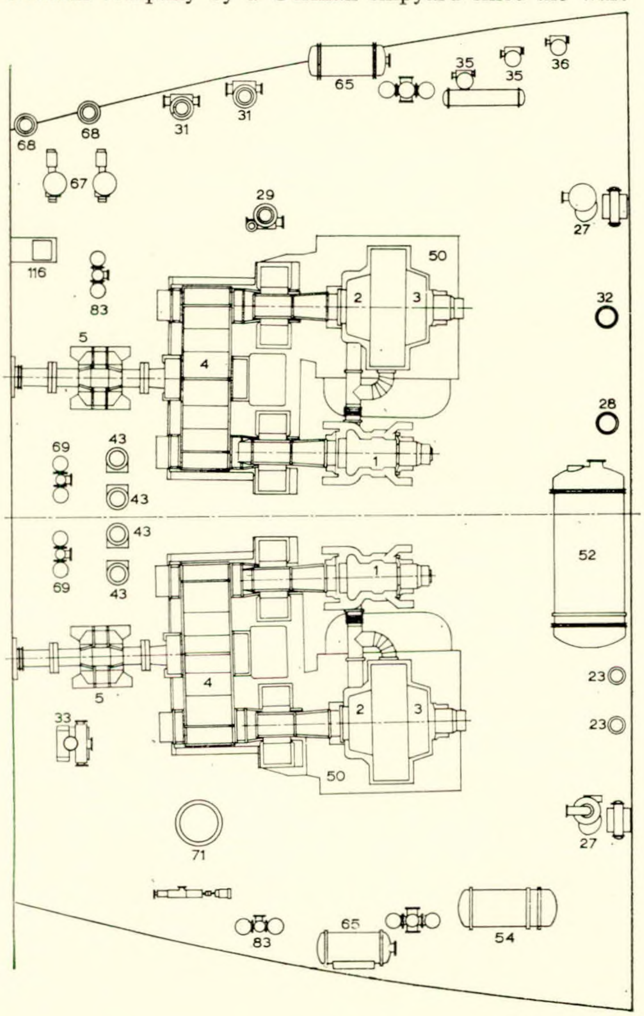
Like all other vessels, tankers, whether crude oil or refined products carriers, only earn revenue when at sea, and therefore time spent in port loading and discharging is not only costly so far as regards port dues, etc., but also through loss of earnings. Therefore it is in the interest of the shipowners to reduce the turn-round time to a minimum. Generally the loading rate is dependent on the shore facilities at the loading terminal but it must also be borne in mind that when berthing at a loading port a tanker is ballasted to a certain percentage of its deadweight tonnage. In many of the larger vessels currently in service some of this ballast is carried in permanent ballast tanks on board ship and can be discharged while cargo is being loaded. In other vessels the ballast is carried in certain of the cargo tanks and thus the loading sequence has to be programmed so that the cargo being loaded can be directed into the empty cargo tanks first, enabling the ballast holding tanks to be discharged with the ship's pumps.

When discharging, the vessel relies almost exclusively on its own pumps for transfer of its cargo ashore. Pumping rates have kept pace with and in some cases, even exceeded, the growth rate of tankers so that the large vessels of today turn round in the same, or even less, time as their much smaller counterparts of a decade or so ago.

Thus the economies of a tanker operation depend to a high degree on efficient cargo handling. A review of the cargo oil pumps and oil pumping systems currently available from some of the major manufacturers of these equipments is given. —*Shipbuilding International, April 1970, Vol. 12, pp. 16–23.*

AEG Turbines—"Hamburg"

Hamburg is the first large passenger liner to be built for a German company by a German shipyard since the war.



1) Hp turbines
2) Lp ahead turbines
3) Astern turbines
4) Gearing (de Schelde)
5) Thrust blocks
23) Auxiliary extraction pumps
27) Turbines for main circulating pumps
28) Auxiliary seawater circulating pumps
29) Fire and general service pump
31) General service pumps
32) Bilge and ballast pump (centrifugal)
33) Bilge and ballast pump (reciprocating)
35) Sanitary pumps
36) Swimming bath pump
43) Main lubricating oil pump (102 m^3/h)
50) Main condenser
52) Auxiliary condenser
54) Drain cooler
65) Main oil coolers
67) Lubricating oil separators
68) Lubricating oil heaters
69) Lubricating oil suction strainers
71) Oily bilge separator
83) Main lubricating oil pressure strainers
116) Separator cleaning bench

Arrangement of machinery at first platform in the turbine room; steam is supplied by three Foster Wheeler ESD III boilers at 62 kg/cm², 520°C

The choice of a turbine propulsion unit for a passenger liner presents many advantages, two of which are especially outstanding:

1) the perfectly smooth running of a turbine results in the absence of vibration and noise which could be transmitted into the accommodation, and hence contributes greatly to passenger comfort;
2) all propeller speeds between zero and full ahead or zero and full astern can be attained independent of any intermediate speed stages. Manoeuvrability and therefore the safety of the ship is thus greatly improved, especially in narrow waters. Also contributing to the safety of the ship is the fact that a twin-screw unit has been chosen. This enables rudder manoeuvres to be supplemented, if necessary, by the corresponding propeller manoeuvres;

Hamburg's propellers together absorb a maximum continuous output of 23 000 shp (metric) at 137 rev/min, corresponding to a speed of 23 knots.—*Michel, F., Marine Engineer and Naval Architect, March 1970, Vol. 93, pp. 129–131.*

Conversion of Cargo Vessel to Oil Drilling Rig

The self-propelled deep-water oil drilling ship *Typhoon*, which was jumboized and converted from the war-built refrigerated cargo ship *Karin* was commissioned recently.

Fully equipped with sophisticated oil drilling equipment and with a mooring system which will permit her to drill at sea in the floating condition, *Tyhphoon* will be used where water depths are too great for drill platforms supported by underwater mat foundations or by legs embedded in the ocean floor.

Typhoon is now 380 ft. long o.a., including a new 42 ft midbody built by the Bethlehem yard. Her width has been expanded from 50 to 70 ft by 10 ft wide sponsons running about 250 ft along both sides of her hull. The sponsons, built by the yard and welded to her sides, give her additional stability and also serve as part of the base for the drilling derrick which towers 170 ft above her decks.

The ship's electrical service is provided by three 500 kW d.c. generators driven by Enterprise Diesels. The drilling equipment is powered by three 500 kW a.c. generators driven by Caterpillar D-398 Diesels and six 645 kW and two 1500 kW d.c. generators, driven by three EMD-16-645 EI Diesels. The mooring system consists of four Model 300 Wilson double-drum mooring winches. Each drum is rigged with 3500 ft of 2¼ in wire rope and one 30 000 lb anchor.—*Shipbuilding and Shipping Record, 9th January 1970, Vol. 115, p. 29.*

Hydraulics on Board New Ships

Shipboard systems are among the earliest applications of controlled, fine resolution, high-power hydraulics. Typical of its traditional marine duties are steering, hoisting, and, on combat ships, ammunition handling and gun positioning.

Hydraulic power is used to position ships' rudders in response to automatic or manual control signals from the pilot house. One factor that led to the almost universal application of hydraulics was the change-over of ships' electrical power from direct current to alternating current.

One of the important problems with hydraulic systems in the past has been contamination. Control units, pump units, and actuators were delivered to a shipyard; the yard installed the equipment and supplied all interconnecting electrical cabling and hydraulic piping. Even with special procedures, a certain amount of dirt (such as weld scale and steel chips) was introduced into the system. The contamination quickly damaged variable-delivery (closed hydraulic loop) piston pumps. Manufacturers of large ship steering gear (Rapson slide ram group with separate variable-delivery pump units) also had this problem. It has been almost eliminated by changing to a "unitized package", which makes it unnecessary to expose the hydraulic system to contamination after in-plant assembly and test. The units are filled with hydraulic oil and sealed at the factory prior to shipment. Installation is confined to mechanical mounting of the preassembled package and electrical wiring.

Stepped "on-off" hydraulic systems—fixed-delivery vane pumps and solenoid operated valves—are not as vulnerable to contamination damage as closed systems. Furthermore, performance can approach that of the variable-delivery pump system by using a multiple output amplifier to control two or more fixed-delivery pumps in steps. This arrangement is comparable to a fine and coarse control where one pump output is used for small rudder movements and a second is added for larger rudder errors.

Hydraulic actuators for positioning the rudder may take one of several forms, i.e. double-ended cylinder, clevis-mounted single ended cylinders, opposed rams with links and tillers, rotary vane, or Rapson slide. The actuators, when combined with hydraulic pump units and associated circuits, provide the basis for many types of hydraulic steering gear ranging from ½ to 75 hp and larger. In conjunction with controls such as Gyropilot and electric stands, lever steerers, lever pilots, and the Sperry Autopilot 8T they form a total ship steering system.—*Breeden, R. H. and Thomas, W. G., Marine Engineering/Log. January 1970, Vol. 75, pp. 104–105.*

Heavy Duty Marine Gas Turbines

Ideal marine propulsion systems should have high reliability, low installed cost, ease of installation, residual fuel burning capability, low fuel consumption, simple unattended operation, system flexibility, ease of maintenance as well as low cost maintenance, which is not necessarily the same thing. Such propulsion system should be available in a wide range of powers.

The gas turbine offered for mechanical drive propulsion are all two-shaft machine, that is, the compressor drive turbine and load turbine are not mechanically connected, thus providing a wide range of operating speeds. Should electric drive be desired for the large icebreaking tankers now being planned for Alaskan North Slope service, a single shaft unit ideally suited for electric generation will be offered. The recent voyage of the tanker *Manhattan* confirmed the need for high torque at low speeds, and for 100 per cent reversing power, both characteristics of d.c. electric drive.

The heavy duty gas turbine is capable of burning a wide range of fuels. The dual fuel capability of the heavy duty gas turbine may be of interest to the Liquefied Gas Carrier. The engines are capable of burning any combination of liquid and gas fuels, continuously sensing the adjusting to proper proportions as boil-off is reduced.

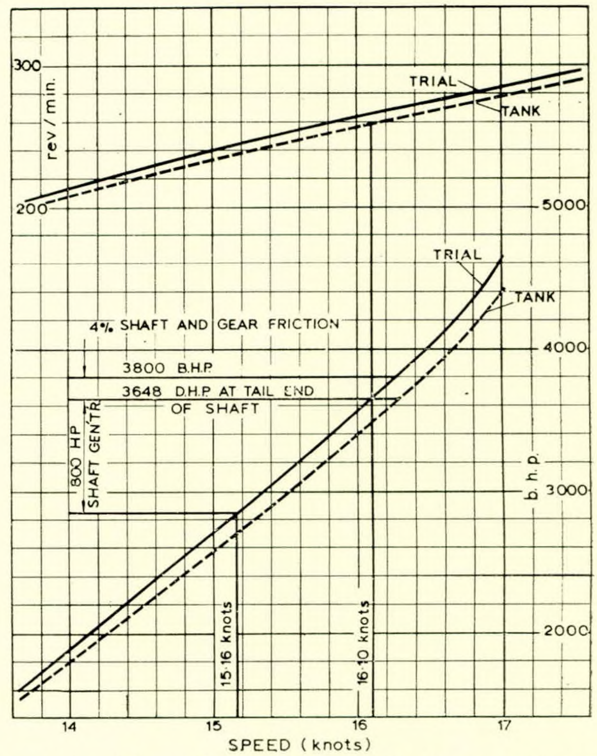
System flexibility was mentioned as one of the desirable attributes of a marine propulsion system. The gas turbine provides the basic component to what may be described as a total energy system that can be employed in a variety of ways, due principally to the large amount of waste heat available.

The utilization of this capability will depend, of course, on the type of ship involved. A tanker, for instance, may use waste-heat generated steam for tank heating and cargo pumping. Steam may also be used with a turbine-generator set to provide electrical power, and also can be used for hotel heating.

Each shaft would have a 12 000 shp propulsion gas turbine which would also provide steam for electrical generation and heating services. In port, a series 1000 GT generator set developing 3000 kW and waste heat steam would be used. The result is an extremely economical and flexible system. In general, the wide range of powers available coupled with waste heat utilization permits the ship designer to tailor the plant for his specific needs without accepting costly or restrictive compromises.—*Hefner, W., Schiff und Hafen, 1970, Vol. 22, pp. 161–162.*

Cruise Vessel for Antarctic Voyages

A cruise vessel designed for taking small parties of travellers on voyages of exploration to some of the more remote parts of the world, including Antarctica, has been built for the Norwegian company Kommandittselskapet A/S Explorer and Co. by Uudenkaupungin Telakka Oy (Nystads Varv AB), Finland. This vessel, *Lindblad Explorer*, is the first vessel in the world to be built specifically for Antarctic cruising.



The speed trial results of Lindblad Explorer *—2480·75 grt*

Lindblad Explorer has been constructed under the supervision of Det norske Veritas for classification ✠ A1 Ice Class A, and to comply with the requirements of SOLAS 1960, the latest IMCO regulations and U.S. Coast Guard regulations for fire protection. The hull is ice-strengthened but not provided with an ice-breaker bow, and is transversely framed throughout. There is an ice knife at the stern to protect the rudder.

Principal particulars are:

Length, o.a.	239 ft ½ in	72·86 m
Length, b.p.	211 ft 9½ in	64·5 m
Breadth, extreme	46 ft 0 in	14·02 m
Depth to upper deck	23 ft 9 in	7·24 m
Draught	13 ft 1 in	4·0 m
Deadweight	541 tons	550 tons(M)
Gross tonnage	2480·75	7027·62 m³
Net tonnage	1142·18	3235·64 m³
Underdeck tonnage	1023·25	2898·72 m³
Tweendeck tonnage	672·13	1904·05 m³
Trial speed	16·1 knots	
Service speed	15·0 knots	

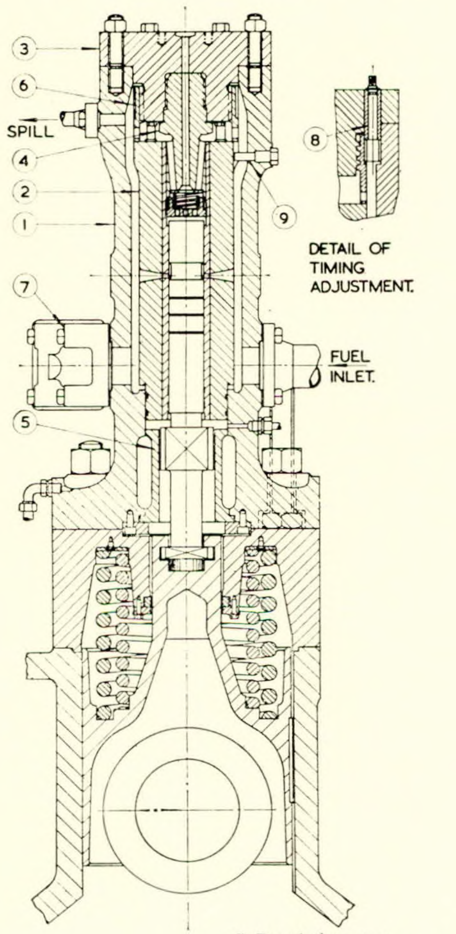
A system of roll stabilization tanks called Cont-Roll, to the design of Knud E. Hansen I/S, Denmark is fitted, and these U-shaped tanks are located just aft of frame 60 and extend across the beam of the ship. Despite its lower specific gravity, fuel oil is used in these tanks which form an emergency fuel reserve.

Propelling machinery comprises two turbocharged, eight-cylinder, four-stroke MaK Diesel engines type 8M 452. These have a cylinder diameter of 320 mm, stroke of 456 mm and output of 1800 hp at 450 rev/min, and are arranged for running on marine Diesel fuel. These engines drive the four-bladed Liaeen stainless steel controllable pitch propeller at a speed of 260 rev/min through Lohman and Stolterfoht reduction gearing and pneumatic clutches. Propeller diameter is 2850 mm and maximum pitch is 2565 mm.—*Shipping World and Shipbuilder, March 1970, Vol. 163, No. 3843, pp. 387–391.*

Progress with B and W Diesel Engines

The KEF-engines are produced with cylinder diameters of 420 mm, 620 mm, 740 mm and 840 mm. The basic structure follows the VT2BF-types which have been in service since 1959.

Three engines of the K74EF type delivered from the Copenhagen works have been in service for about 2000-3000 hours loaded at p_i = 9·5–11 kg/cm² and at 100–125 rev/min.



1) Housing
2) Barrel
3) Cover
4) Central guide and suction valve assembly
5) Regulating arm
6) Threaded distance ring
7) Shock absorber
8) Timing adjustment
9) Guide pin

The type of fuel-pump fitted to the super-large-bore

At inspections through scavenging ports the cylinders, pistons and piston rings have been found to be in good condition, and so have the crosshead bearings, crank bearings, and main bearings which have been inspected.

The KFF engine type has no far been built as super-large-bore engines only—with a cylinder diameter of 980 mm. Two 7-cylinder engines of this type are in service. The first engine has been running some 10 000 hours and the second for 5500 hours.

Both engines are loaded correspondingly to p_i = 11·5 kg/cm² and are running at 103–108 rev/min. The engines have, since their trials, been running solely on heavy oil, mostly bunkered in the Persian Gulf and with a viscosity of between 700 and 2000 sec. Red. I.

Immediately after the testbed running of these engines a careful inspection was undertaken of the running-in conditions of the main and crankshaft bearings, camshaft bearings and of crosshead bearings. This examination revealed extremely smooth and even bearing surfaces. Later inspection of the bearings in service has confirmed this first impression.

As regards the camshaft bearings it should be mentioned that the tightening of the bearing shells is now accomplished hydraulically in order to obtain an equal and safe tightening. This method was introduced on the first engine after a short time in service and after movement had been observed in the foremost camshaft bearing after 48 hours' service.

Original mechanical tightening of the bolts proved very difficult because of the limited working space available. The change to hydraulic tightening was undertaken while the ship was en route from Lisbon to Las Palmas. Such a repair was undertaken on one half of the engine at a time, while the remainder of the cylinders were working. Subsequent inspection of the bearings proved that this was effective and reliable.

The new type of fuel pump (shown in the diagram) is performing satisfactorily. Leakages at the tightening ring at the foot of the pumping piston (approx. 29 g/h/cyl.) occurred during the first service period, but since the tightening ring has been worked-in the leakage has, in fact, been reduced to vanishing point. In order to reduce the running-in period of about 1500 hours, we have replaced the tightening rings by a type having a thinner scraping edge.—*Moller, K., The Motor Ship, January 1970, Vol. 50, pp 431-433.*

Supply and Buoy Vessels

Marine Industries Limited of Sorel, Canada, has recently completed two supply and buoy vessels designed by Gilmore, German and Milne of Montreal for the Canadian Coastguard Service.

These two vessels, *Provo Wallis* and *Bartlett,* serve in the coastal water of the Maritimes as replacements for the CCG's *Brant* and *Seabacon* which have been withdrawn from service because of age.

The all-welded steel hull has an icebreaker bow, a cruiser stern, and is strengthened for operation in ice to Lloyd's Class II and overstrengthened by 25 per cent to meet the requirements of the Canadian Department of Transport. It has a raised foc's'le and a long poop extending aft from midships. The machinery is located aft and the accommodation is arranged on the upper, poop and boat decks. Subdivision of the hull from forward to aft is arranged as follows: forepeak, buoy winch compartment, cargo hold, control room, main engine room, auxiliary machinery compartment and aft peak. A Flume stabilizing system has also been incorporated.

Principal particulars:

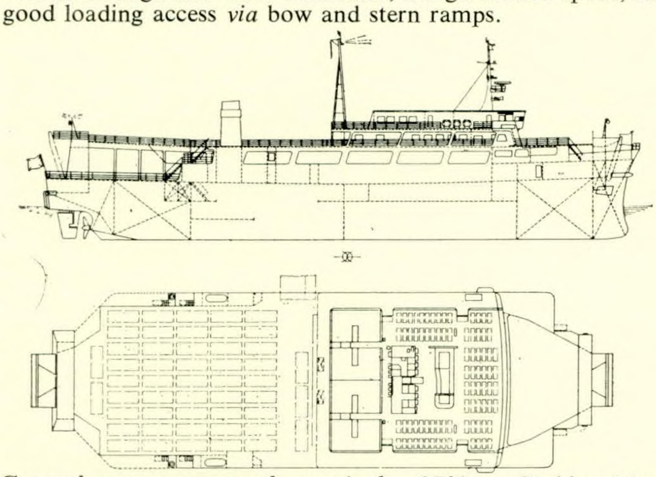
Length, o.a.	189 ft 3 in
Length, b.p.	169 ft 9 in
Breadth moulded	42 ft 6 in
Depth moulded to upper deck ...	16 ft 6 in
Draught	12 ft 0 in

Each vessel is powered by two Mirrlees National turbo-charged reversible six-cylinder KLSDM6 Diesel engines driving two Lips four-bladed, 6 ft 6 in (1·98 m) diameter controllable pitch propellers. Power from the engines is transmitted to the propellers *via* Vulcan-Sinclair fluid drive couplings with scoop control. The engines each have an output of 1050 bhp at the designed speed of 340 rev/min which corresponds to a ship speed of 11 knots. Control of the main engines can be effected from three places on the bridge, the engine room, control room and locally at the engines. Three Paxman Diesel generating sets are installed to provide the vessel's electrical requirements, each set comprising a 311 bhp

Diesel engine driving a 185 kW generator supplying current 460 V. A Ruston emergency generator is also fitted. Other machinery items include two Vapor-Clarkson modulatic boilers to meet steam requirements, two Reavell air compressors and Brown Brothers rotary vane steering gear.—*Shipbuilding International, April 1970, Vol. 12, pp. 24–25.*

World's Largest Catamaran Vessel

The 2700 gross ton *Rokko Maru*, claimed to be the world's largest catamaran ferry, is now carrying vehicles and passengers on the 112-mile route between Kobe City and Takamatsu City on Shikoku Island, Japan. The vessel is owned by the Kansai Steamship Co. and was built by the Shimizu yard of Nippon Kokan. Her twin-hull design offers a large amount of deck space; this gives *Rokko Maru* a remarkable vehicle capacity of 42 heavy trucks, 10 light trucks and 50 cars, in addition to 580 passengers. Other features of the ferry are outstanding manoeuvrability—achieved by a shallow draught and the distance between the two propeller shafts, greater stability than conventional vessels through less wave resistance, a high service speed, and good loading access *via* bow and stern ramps.



General arrangement plans of the 2700 gt Rokko Maru *showing the large vehicle capacity on two decks and the fore and aft ramps*

Principal particulars are:

Length, o.a.	274 ft 6 in
Length, b.p.	255 ft 5 in
Breadth, moulded (total) ...	82 ft 0 in
Breadth, moulded (single hull) ...	23 ft 0 in
Depth, moulded	26 ft 3 in
Draught, moulded	15 ft 4 in
Gross register	2700 tons
Service speed	18·8 knots

Propulsion is by four 1440 bhp Daihatsu 8DSM-26 engines, two in each hull. Each pair is geared to one shaft, and the total output from the four engines is 5760 bhp at 695 rev/min, giving a service speed of 18·8 knots.—*The Motor Ship, January 1970, Vol. 50, p. 479.*

Manoeuvring Large Tankers

The author describes the Coanda effect thus: when a fluid passes through a relatively fine slit and thereby enters another fluid, if one of the lips of the slit orifice is extended in such a way that it continually diverges from the outlet axis of the slit, then a region of reduced pressure is created at the wall formed by the extended lip and the fluid will follow this wall.

After providing an explanation for this phenonmenon, the author gives some brief information on studies being undertaken by the Chicago Bridge and Iron Company. An investigation is being made on the use of the Coanda effect to create a reduced pressure in the water at a chosen point along the length of a ship in order to manoeuvre it in harbour without having to use external aids such as tugs and berthing hawsers.

Both fixed and movable devices, placed below the waterline, have been studied. The necessary power for forcing the primary fluid (water or steam) through the slits can be obtained with water pumps or by using steam under pressure. If steam is used, it can be produced by injecting sea-water into the flame of a special burner (a Coanda "tuyere" or nozzle), thus avoiding saline deposits in boilers and piping. The extended lip should be designed so that the greatest possible mass of ambient fluid is entrained. In tests now being carried out, the application of the device to large tankers is being particularly studied, though smaller ships are not being neglected.

An alternative scheme may be to install the devices in the quay wall, where they would act as suckers and draw the ship to the wall and retain it there. The device, as installed in a tanker, could be used in association with single-point moorings.

The device can produce depressions up to 90 per cent absolute vacuum, and the suckers should be able to provide an average force of 5 tonnes m^2 of contact surface.—*Paper by Coanda, H., presented at the 1969 Meeting of the Ass. Techn. Marit.; Jnl Abstracts B.S.R.A., December 1969, Vol. 24, Abstract No. 28 458.*

Comparative Full-scale Studies of a Nozzle Propeller on a Giant Tanker

The Swedish Ship Research Association (S.S.F.) has begun the first full-scale investigation into the possibility of fitting nozzle propellers to large tankers. After extensive model tests in co-operation with the Swedish State Tank (S.S.P.A.), a full-size nozzle propeller is to be fitted to a tanker in order to carry out tests under service conditions. The tanker in question is the as yet unnamed sister ship to the 130 000-ton *Oceanus* (see Abstract No. 28 140, October 1969), at present under construction at Eriksbergs.

Sister ships have been chosen for this exercise so that a direct full-scale comparison of the performance of a nozzle propeller on ships of this size may be obtained. Model tests tend to indicate an increase in efficiency of 10–15 per cent, which would mean savings of about £20 000 per annum in a 200 000-ton tanker, although in this particular ship a 5–10 per cent increase is expected, giving a fuel-cost saving of £12 000 per annum. It is thought that the increased cost of a ducted propeller will be offset at the manufacturing stage by a reduction in machinery costs (on account of the improvement in propulsive efficiency). It is also possible that the nozzle will eventually replace the conventional rudder.

The propeller to be fitted is a controllable-pitch KaMeWa type, with a diameter of 8·3 m. The shroud is 3·4 m in length, and the assembly will weigh about 50 tons, and transmit 25 000 hp.—*Svensk Sjöfarts Tidning, September 1969, Vol. 65, pp. 20–21; Jnl Abstracts B.S.R.A., December 1969, Vol. 24, Abstract 28 456.*

Packaged Concept Applied to Automation for Propulsion Systems

Although the reliability of marine automation systems has to be at least equal to that of the propulsion system complex itself, this need, obvious though it may seem, cannot always be met at the present moment because of adherence to traditional practice by shipbuilders and shipowners alike.

The solution to this problem, according to at least one source, is to design the whole propulsion system, including auxiliary and control gear, as an entity and this work should preferably be carried out by the engine manufacturer. With this in mind a design study has been made by Vickers Ltd. in conjunction with Negretti and Zambra to produce a modular instrument system for sale with the M.A.N. medium-speed engines produced at the Vickers Engineering Works, at Barrow-in-Furness.

The Vickers/Negretti and Zambra approach is to install all the transducers on the engine at the construction stage and wire them to a common junction box. The indicators and alarms are fitted into specially designed cases or modules which can be mounted together in stacks without additional support. Alternatively, the cases can be inserted into racks, panels or consoles. Seven basic inserts are available for the cases, which between them will carry out all the functions required for an unmanned engine room, including the automatic starting up of standby equipment. Thus, apart from installing transducers on external equipment, the shipbuilder has only to cable from the engine to the modules with consequent notable saving in engineering costs.

To illustrate the compass of the propulsion package offered by Vickers, a range of Vickers-M.A.N. medium-speed engines can be supplied, coupled to an epicyclic gear-box of Vickers design and manufacture together with a simplified fuel system, other auxiliary systems in module form, and the complete remote control and instrumentation systems. The main engines are in-line and Vee 40/54 types developing up to 10 000 bhp (metric) at 430 rev/min as well as the in-line and Vee 52/55 types; 18-cylinder units developing up to 18 000 shp (metric) at 45 rev/min.—*The Motor Ship, March 1970, Vol. 50, p. 603.*

Contra-Rotating Propellers for Large Merchant Ships

The effect of replacing a single screw by a set of contra-rotating propellers was studied in the light of the power needed by two characteristic ship types to maintain a given speed. Open-water tests were conducted with three pairs of contra-rotating propeller models, each of which had a different pitch; a linear relation was maintained between the pitch ratios of forward and after screws. Each of the models was then fitted to the hull of one of the forms examined and towed in a locked position.

The open-water results were used to plot the usual hydrodynamic characteristics in relation to the advance ratio and pitch ratio of the forward propellers. The efficiency/loading relation agrees well with the results of other workers. A comparison of the results with single screws shows that, for the same propeller diameter, the contra-rotating propellers have a higher efficiency (approximately 10 per cent) over practically the whole range of loadings examined. In addition to this, the optimum rev/min of the contra-rotating propellers is 30–35 per cent less than that of the single propeller. The contra-rotating propellers have little advantage over the single propeller at the same rev/min in open water. The optimal diameter of the contra-rotating system is 15–20 per cent less than that of a single screw.

Self-propulsion tests of a model of a *Sofia* class tanker (60 000-tons displacement) were run with each of the pairs of model propellers in order to evaluate hull/propeller interaction. The results of these tests were compared with the results of interaction tests for a 60 000-tons displacement tanker with a single screw whose diameter corresponded with the diameter of the forward propeller of the contra-rotating system. The comparison showed that, for the same screw diameter, the interaction between contra-rotating propellers and the hull is the same as between a single propeller and the hull. However, more detailed investigation is required into this problem.

A comparative evaluation is made of the propulsive characteristics of a *Sofia* class tanker, a tanker of 100 000-tons displacement, and a high-speed general-cargo ship, fitted with contra-rotating and single propellers. The installed power of the ships was 25 000 hp. The resistance and interaction coefficients used in this calculation were determined from the model tests in the case of the *Sofia* class tanker, and from systematic series data in the case of the other two ships. The results of this comparison are tabulated and show, *inter alia*, that, at the same diameter and the respective optimum rev/min values, contra-rotating propellers improve the propulsive characteristics of practically all merchant-ship types by approximately 10 per cent.—*Voevodskaya, E. N., Sudostroenie, No. 9, 1969, pp. 6–9; Jnl Abstracts B.S.R.A., December 1969, Vol. 24, Abstract No. 28 454.*

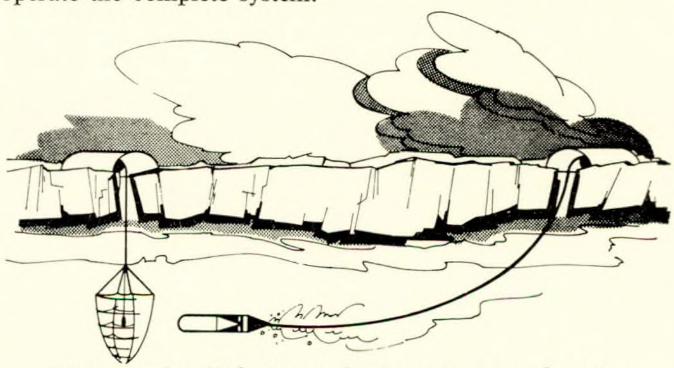
Unmanned Vehicle for Under-ice Seismic Exploration

An unmanned, torpedo-shaped submersible *Unitow* is being developed for seismic exploration under the ice by Marine Resources, Inc.

The vehicle, which is $1\frac{1}{2}$ ft wide and 12 ft long, can be submersed through a 24-in hole drilled in the ice pack. Its primary function is to tow a messenger cable from an opening about one mile away. The messenger cable is used to pull a hydrophone streamer cable between the two holes.

The relatively simple, battery-powered, 200 lb vehicle, which carries 7000 ft of messenger line, can cover about one mile in 10 minutes. The total operation, from initial acoustic checks of the vehicle to implantment of the hydrophone streamer, requires about one hour.

Unitow's run depth is preset and is maintained by a diaphragm and pendulum system, controlling the diving plane actuation motor. Only two technicians are required to operate the complete system.



Unmanned vehicle for under-ice seismic exploration

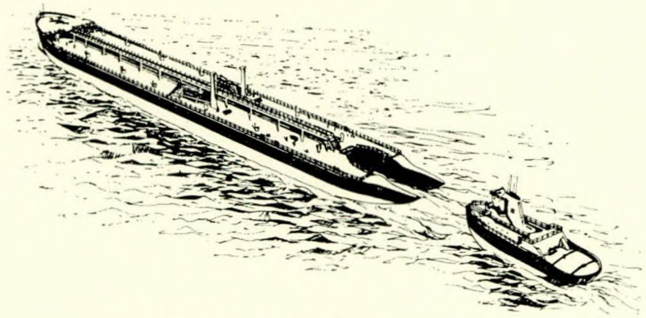
A snare assembly, lowered through the ice at the termination point of the run, "captures" the messenger cable paid out by the vehicle. A target transducer, suspended in the centre of the snare, provides the active acoustic homing signal for *Unitow*. A listening hydrophone, also suspended in the snare, indicates the passage of the vehicle through the snare and signals the shut-down of the propulsion system.

The cable reel on the vehicle locks automatically when the propulsion is shut down, preventing further cable pay out and allowing recovery of the positively buoyant *Unitow*. Gyro alignment accuracy of five degrees is sufficient, and the acoustic guidance system can guide *Unitow* home if the snare system is within 30 degrees of the axis of the vehicle.—*Undersea Technology, February 1970, Vol. 11, p. 28.*

Ocean-going Tanker Barge/Tug

A powerful pusher tug is being built in America to propel large ocean-going oil barges—a combination which could, of course, also be used instead of conventional dry cargo carriers or container ships. As the figure shows the tug fits into a "slot" at the stern of the specially-designed tanker barge to form a single unit having the same capacity and operational speed as a normal vessel of comparable overall dimensions. On arrival in port the tug can be uncoupled from the barge quite quickly and employed to propel another similar barge. The relatively expensive propulsion unit is therefore kept fully employed and operating costs should be lowered substantially. It is worth mentioning, incidentally, that the consultants for this interesting project are a London-based firm, Breit Engineering Incorporated.

The tug will be of 11 000 hp—the most powerful ever built in the U.S.—and 140 ft long by 46 ft beam and 26·5 ft draught.



Ocean-going tanker barge/tug

The tug will initially operate with a 532 ft-long oil tanker barge, which is being constructed by Alabama Dry Dock and Shipbuilding Company at Mobile, Alabama. The barge will be 87 ft wide and has been designed to carry some 280 000 barrels of petroleum products or about 38 000 tons. Similar tug/barge combinations are planned to carry bulk cargo containers and refrigerated cargo.

British medium-speed heavy-fuel-burning Diesel engines have been chosen to propel this powerful tug. Each of the twin screws will be driven by a 12-cylinder Vee-type Mirrlees KVMR12 Major engine having a maximum continuous service rating of 5564 bhp at 525 rev/min. This will give a shaft speed of 135 rev/min, each engine driving a KaMeWa controllable-pitch propeller through MWD (type R, size 12) reduction gear-box. A Holset flexible coupling will be fitted between engine flywheel and gear-box input shaft. Control consoles for the main engines will be situated in the engine room and also in the pilot house, the control position being chosen from the engine room.—*Shipbuilding International, April 1970, Vol. 12, p. 37.*

Fire-fighting Tug for the Port of Belfast

A berthing and fire-fighting tug for use in the port of Belfast has been delivered to her owners, John Cooper (Belfast) Ltd., by Appledore Shipbuilders Ltd. This vessel, *Coleraine,* is the first to be fitted with a Merryweather/Simon Snorkel, carrying a fire monitor.

Principal particulars are:

Length, o.a.	106 ft 3 in
Length, b.p.	95 ft 0 in
Breadth, moulded	28 ft 0 in
Depth, moulded	13 ft 6 in
Draught, aft	13 ft 6 in
Draught, forward	9 ft 6 in
Displacement, loaded	456 tons
Gross tonnage	211
Machinery output ...	2500 bhp at 500 rev/min
Free speed	$13\frac{3}{4}$ knots
Bollard pull	$32\frac{1}{2}$ tons
Fuel capacity	50 tons
Fresh water capacity	18 tons
Foam capacity	20 tons

Of principal interest is the tug's fire-fighting equipment, the main element of which is the hydraulically-powered elevating platform. During normal towing and berthing duties the Snorkel is folded and stowed at wheelhouse top level, thereby lowering the system's centre of gravity and reducing windage.

The apparatus consists of a pair of box section articulated booms built on a turntable. The operator's platform, which carries a water/foam fire monitor, is at the end of the upper boom and maintains a horizontal position irrespective of the position of the boom arms. Each boom can be raised or lowered independently and the turntable can rotate continuously in either direction. All platform movements are controlled by the operator *via* simple hand controls. Duplicate controls are fitted at the turntable position for remote operation if this should be required. Operation is entirely hydraulic, pressure being provided by a self-contained pump. A water curtain around the working platform protects the operator and there is a communications system between the platform and the turntable.

Propelling machinery for *Coleraine* consists of an eight-cylinder MWM Diesel engine which develops 2500 bhp at 500 rev/min. This has a bore of 360 mm, 450 mm stroke and is a four-stroke, turbocharged and intercooled engine fitted with a Brown Boveri turbocharger. Start air pressure is 425 lb/in^2. This engine drives the controllable-pitch propeller at 225 rev/min through a Lohmann and Stolterfoht gear-box and Pneumaflex clutch.

Bollard pull trials of *Coleraine* were carried out at Swansea using a Robertson 60 tons $\times$ 5 cwt hydrostatic weigher. A bollard pull of $32\frac{1}{2}$ tons was attained. The maximum speed attained during trials was $13\frac{3}{4}$ knots, and from this speed to being stopped in the water during the 'crash' stop manoeuvre took 22 seconds. To complete a 360 degrees circle under full power, turning to port or starboard, took 1 min 12 sec.—*Shipping World and Shipbuilding. April 1970, Vol. 163, pp. 545–547.*

After-body Vibration Reduced by Modifying Wake Distribution

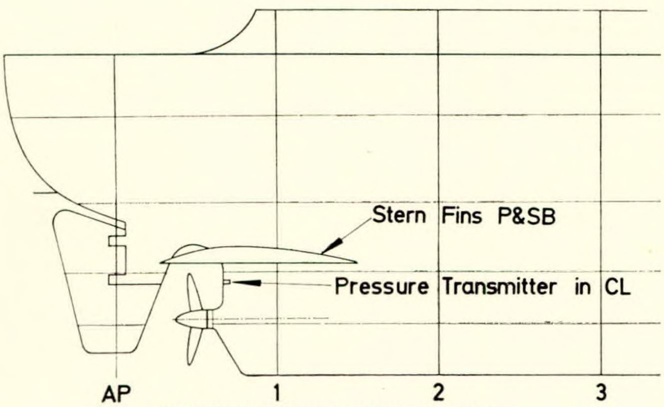
Two sisterships, *En Gedi* and *Avedat,* built in 1964 for the Zim Israel Navigation Company, Haifa, Israel, were intended for bulk cargo and for a carry capacity of about 33 000 dwt. At their construction date they were among the largest ships of this type and experience was, therefore, limited. Soon after the delivery serious afterbody vibration appeared. However, after extensive investigations at the Swedish State Shipbuilding Experimental Tank the vibration has been reduced to an acceptable level.

The severe vibration that appeared in the after body meant that the ships could not be operated at the continuous rated speed of the engine, 135 rev/min. When the rate of revolutions was increased above about 125 rev/min serious vibration appeared and this had a negative influence upon the working conditions of the crew and caused damage to the hull and equipment. For these reasons, the vessels had to be operated at reduced speed, which resulted in a lower profit.

Considerable efforts were made to avoid these serious conditions. Hull vibration was measured by Lloyd's Register of Shipping and by an American naval consultancy. Investigations were made in collaboration with the owners, the shipyard and the propeller contractor. As a result of these investigations it was recommended to change the four-bladed propeller to a six-bladed. The new six-bladed propeller did not, however, improve the situation.

To further consider the operating conditions, the shipowners consulted the Swedish State Shipbuilding Experimental Tank (S.S.P.A.). The resultive investigation started with resistance and propulsion tests for the actual hull form.

At the suggestions of the owners' naval architect and to improve the water flow through the propeller disc, the experimental tank designed flow accelerating fins to be placed on the stern above the propeller. Tests were made with different angles of attack for these fins and with different foil sections. These showed that the wake distribution could be favourably influenced by fins so that the high wake factors



Disposition of fins and pressure transmitter

After-body vibration reduced modifying wake distribution

in the upper part of the propeller aperture could be reduced to acceptable figures. Streamline tests also showed improvements in the flow. The most suitable fin design was then chosen from the tested alternatives. The stern fins were built and fitted to *En Gedi* in Holland during the spring of 1969. Subsequent trials showed that the expected improvements were reached. Lloyd's Register of Shipping measured the vibration and stated that for certain points the amplitude of vibration had been reduced to about 1/3 of earlier values. The society also stated that the measurements indicate, in general, that after fitting fins the vibration has now been reduced to an acceptable level.—*Shipbuilding and Shipping Record, 6th March 1970, Vol. 115, p. 31.*

Huge Dredging Crane

A huge floating crane wtih a 36-ton grab is being constructed at the Figee facility in Holland for a Bos and Kalis affiliate. Designed for heavy duty dragline work, the clamshell dredging crane will be capable of operating in open seas anywhere in the world.

Lifting capacity is based on 36 metric tons, which is the weight of the fully loaded clamshell bucket. The winches will have a total pull of 72 tons, which is enough to lift the bucket free from the suction of bottom mud.

The crane will have a maximum operating radius of 31 m and a minimum of 16·5 m. Its range of lift above sea level will be 28 m at all radii and below sea level 50 m at all radii. Hoisting and closing speeds for a 36-ton load will be about 65 m per minute.

The crane tower will be of welded tubular frame construction and the jib of welded steel plates and tubes.

The hoisting mechanism will consist of two independent two-rope winches operating a four-rope grab. The ropes will run directly from the winches *via* the top of the tower over the jib top to the bucket. The weight of the jib will be partly balanced by a counterweight running along guide beams on the back of the tower.

The pontoon on which the crane will be mounted will have an overall length of 50 m, width of 19·8 m, depth of 4 m, and draught of about 2 m.—*World Dredging and Marine Construction, February 1970, Vol. 6, p. 25.*

Moving Ships Sideways

A question that frequently arises when berthing and moving ships which are initially stationary is, how will they respond to the transverse direction to the action of external forces and moments applied to them? The forces and moments may be caused by the operation of tugs and lateral thrust units for example, or from a wind acting on the above water form. Combinations of these forces may also occur. There appears to be little published information dealing with this problem, and this is not surprising considering the complexity of an exact solution, however, it is an important problem aggravated these days by the immense size of ships, particularly tankers, that are being used. The derivation given is very simple and may be open to criticism for a number of reasons, further, it has not been correlated with measured values either from models or full size ships because of the lack of suitable information.—*English, J. W., Shipping World and Shipbuilder, April 1970, Vol. 163, pp. 545–547.*

Refrigerated Cargo Vessel for West Indies Run

A refrigerated cargo vessel built by Constructions Navales et Industrielles de la Méditerranée (CNIM) for Compagnie Générale Transatlantique has now been delivered. This vessel can carry containers, cars, rum and general cargo. On the outward journey from France the vessel will carry containers, cars, general cargo on pallets and dairy produce, returning from the West Indies loaded with bananas, pineapples, rum, and sometimes avocados.

Pointe des Colibris has been built for classification by Bureau Veritas ✠ 1. 3/3 L 1.1-A, CP ✠ RMC, RMC.V-AUT and has a Maierform bow, long forecastle and transom stern.

Principal particulars are:

Length, o.a.	152·55 m	500 ft 0 in
Length, b.p.	140·00 m	459 ft 0 in
Beam	21·00 m	68 ft 11 in
Depth to upper deck ...	12·65 m	41 ft 6 in
Draught	8·00 m	26 ft $2\frac{1}{4}$ in
Deadweight ...	8500 tons (M)	8360 tons
Displacement ...	15 124 tons (M)	14 890 tons
Gross tonnage		6777
Net tonnage		3331
Hold capacity	13 800 m³	487 000 ft³
Refrigerated cargo capacity	7760 m³	273 800 ft³
Rum tank capacity ...	2850 hl	62 600 galls
Machinery output ...	16 740 hp at 500 rev/min	
Service speed		21 knots
Speed on trial		22·34 knots

No. 1 hold has a weatherdeck hatch measuring 14·72 m by 5·2 m with a roller-blind Ermans cover. This hold is used for the carriage of cars and general cargo. At its after end, port and starboard at main deck level, are hydraulically-operated side-loading doors which are of CNIM make.

The propulsion machinery for *Pointe des Colibris* comprises two 18PC2V SEMT-Pielstick Diesel engines, built by Chantier de l'Atlantique, St. Nazaire. These have an output of 16 740 hp (M) at 500 rev/min and drive a single controllable-pitch propeller through a Renk gear-box. The propeller, which is of Ateliers et Chantiers de Bretagne (ACB) make, is four-bladed and of Alcunic material. It has a diameter of 6 m and rotates at a speed of 126 rev/min. This particular type of c.p. propeller has its servo gear located completely outside the hub. In the design used for this vessel the control oil is introduced into the system through the hollow part of the propeller shaft. This permits an increase in pressure of the actuating oil to about 75 kg/cm², which is more than double the pressure used when the oil enters the system at the servo motor. This higher pressure makes possible the use of a smaller and more efficient servo motor. If it is desired to increase pitch by a series of steps it is not necessary to stand by the control adjusting the pitch manually. The system can be switched to spread a pitch change over a period of 15 min once the combinator handle has been moved to the desired pitch setting. An AEI torsionmeter is fitted to the shaft.

During sea trials at a displacement of 9350 tons (M) and engine output of 12 300 hp (M) at a propeller speed of 125·6 rev/min and pitch of 20 degrees, a speed of 20·37 knots was attained. At this displacement and nominal engine output of 16 600 hp (M) with the propeller pitch set at 23 degrees and speed at 125·5 rev/min, a ship speed of 22·35 knots was attained.

Manoeuvring tests showed that the turning circle could be completed within 4 min 50 sec at an average radius of 0·22 nautical mile. A crash stop astern test from an ahead speed of 22 knots resulted in the ship being stopped in the water within 1 min 43 sec over a distance of 375 m.—*Shipping World and Shipbuilder, January 1970, Vol. 163, pp. 181, 182; 183; 185; 186.*

Study of Stress Concentration in the Region of the Hull Superstructure-end Connexions Using Large-scale Models

An experimental investigation, using two large-scale models, was carried out at the Leningrad Shipbuilding Institute in order to verify the accuracy of a formula for the approximate determination of the stress concentrations in a ship hull at the end connexions of the superstructure. This formula is given.

The principal particulars of the models are:

	Model 1	Model 2
Length	4800 mm	5500 mm
Length of superstructure (without fairing pieces)	1316 mm	3000 mm
Breadth of superstructure	280 mm	800 mm

The superstructure on Model 1 was located precisely amidships, and that on Model 2 was located in the general midship area. The superstructures on both models were connected to the sheer strake.

The other objectives of the investigation were to establish:

a) the size and character of normal stress distribution in the unattached ends of the fairing pieces, and in the deck in this region;

b) the effect of the length of the fairing pieces and the shape of their unattached edges on the distribution of normal and shear stresses in the superstructure ends;

c) the distribution of shear stresses in the joining area between the transverse and longitudinal parts of the structure;

d) the effect of an alteration in the length of the superstructure on the size of the stress concentration factor in the superstructure region of the hull, in order to verify theoretical prediction.

This experimental study also provided the opportunity to verify theoretical prediction of the greatest stress concentration in the ship hull in the region of the superstructure ends and expansion joints.

During the tests the length and shape of the fairing pieces on both models were varied.

The test results are presented in graphs and show, *inter alia*, that the length of the fairing pieces has the greatest effect on the size of the stress concentration at the ends of the superstructure, and that the shape of its unattached edge affects the position of the maximum stress concentration.

A comparison of the experimental and theoretical results (derived by using the approximate formula) shows reasonable agreement in determining stress concentration at the base of the transverse cut-outs of the expansion joints. This comparison also shows that the error in determining the coefficient of stress concentration, through the approximate formula, is not greater than 15 per cent.—*Sivers, N. L. and Rizhinashvili, G. M., Sudostroenie. 1969, No. 7, pp. 12–16; Jnl Abstracts B.S.R.A., November 1969, Vol. 24, Abstract No. 28 310.*

U.S. Bodies to Investigate Propeller Failures

An investigation estimated to cost $218 000 is to be mounted by the American Bureau of Shipping, the U.S. Maritime Administration, American President Lines and the States Steamship Company into recent propeller fractures in new U.S. ships.

Of the total amount, Mar.Ad. has appropriated $120 000 specifically earmarked for structural tests of the propeller of the cargo liner *Michigan* and the measurement of hull vibrations.

At least six propeller blades of the type to be tested have fractured after relatively short service periods. Rather than just correcting this problem by increasing propeller dimensions, it is desired to obtain data on the cause of the failures and to offer possible improvements in design criteria and methods. *Michigan* is to be made available by States Steamship for the study about mid-March. She is one of five sisterships of 14 149 dwt completed by Avondale Shipyards in 1968 and 1969, the others being *Colorado, Montana, Idaho* and *Wyoming*. By mid-summer of 1969 propeller trouble had started with the first three of these vessels, with the result that the vessels were involved in costly tie-ups at Hong Kong, Yokohama and Long Beach, California, until new propellers could be installed. The president of Avondale Shipyards, the builder of the vessels, denied any shipyard responsibility for the three vessels being immobilized for nearly two months while on maiden voyages. He said that the use of high-tensile steel in the ship's structure coupled with the five-bladed propeller design caused the propellers to fracture, and that the defect was not in the casting of the propellers which were made in the Avondale workshops.

Eight other ships which incurred propeller fractures were relatively new cargo liners of the Delta Steamship Lines, New Orleans. Of these, three were built by Avondale and had four-bladed propellers. The other five were constructed by Ingalls Shipbuilding Corporation and had five-bladed propellers. High-tensile steel was used extensively in the Ingalls-built Delta ships. Mr. Zach Carter, President of Avondale has explained that, in his opinion, the use of high-tensile steel made the hulls so light that considerable vibration is caused when the ship is driven at high speed by the 17 000 shp turbine propulsion plants which produce a service speed of 23 knots.

This vibration, he said, affects the propeller and causes fractures. The high-tensile steel used in these ships to make the vessel lighter and able to carry more cargo has been extremely difficult for the shipyard to handle.—*Shipbuilding and Shipping Record, 13th February 1970, Vol. 115, p. 8.*

Heavy Weather Repairs to Scotstoun by Hitachi

Heavy pounding and slamming while on passage off South Africa caused severe buckling in the forward double-bottom of the 11 183 gross ton cargo ship *Scotstoun* last year, and after temporary repairs had been carried out, the vessel proceeded to Japan to complete discharging her cargo before entering Hitachi Zosen's Kanagawa yard. Prior to the ship's arrival, underwater inspection was carried out at Moji and Yokohama, and prepaparations put in hand accordingly.

A two-day period in dry dock sufficed for a complete inspection of the damaged area which extended over most of the double-bottom in way of No. 2 hold. After six days moored alongside, the new block sections for the damaged area were ready and the vessel re-docked. The whole of the No. 2 d.b. tank section was removed, and 135 tons of new steelwork inserted. A total of six new block sections had been built by the yard complete with all internal pipework and fittings and with the tank internal surfaces painted, following the block erection techniques used in the shipbuilding yards of Hitachi.

While the repair was carried out the ship's sides were supported by side shores, and by temporary steel brackets and stiffeners within the hold. The repair which also included the renewal of the lower part of the hatch end centre pillar was completed in the planned time of 11 days.—*The Motor Ship, March 1970, Vol. 50, p. 590.*

Hull and Bulbous Bow Interaction

Resistance, sinkage, trim and wave pattern resistance were measured at two drafts in tests of a 0·80 block coefficient Series 60 model at the Glasgow University Experiment Tank. All the tests were repeated with each of two different protruding bulbs fitted to the model. Records were also made of the wave profiles produced by the naked hull, the hull and bulb combination and the bulb alone. Calculations were made to estimate the vertical force acting on each bulb.

The results confirm previous findings that the addition of a bulb affects viscous as well as wave resistance.

The usual assumption that hull and bulb wave profiles may be linearly superimposed to give the wave profile of the hull-bulb combination is found to be invalid for the model tested.—*Ferguson, A. M. and Dand, I. W., Paper submitted to the Royal Institution of Naval Architects; paper W2 (1970).*

Drilling on Seamounts

A revolutionary new rock-core drill enables scientists, for the first time, to obtain rock cores from outcrops on seamounts, the submerged mountains rising from the ocean floor. Previously rock samples could be collected only by dredging up loose boulders.

A new tool has been developed and built at the department of energy, mines and resources' Atlantic Oceanographic Laboratory, part of the Bedford Institute at Dartmouth, N.S.

Completely automatic, the drill is powered by the pressure of the ocean itself. A timing mechanism opens a valve allowing sea water to flow through a hydraulic power unit into a low pressure reservoir. Another mechanism regulates the drill pressure on the rock face.—*Canadian Research and Development. November, 1969, Vol. 2, p. 41.*

Propeller Loading Distributions

The present study reports the improvements made in the numerical procedure for evaluating propeller loading distributions which had been developed at Davidson Laboratory by adaptation of the unsteady-lifting-surface theory. A new approach, based on the fact that the assumed Birnbaum chordwise modes are not linearly independent, has achieved stability for the chordwise distribution, which had otherwise shown no sign of convergence with increasing number of modes. Other refinements of the numerical programme, including provision for arbitrary blade camber variation and overlapping of blade wakes, have improved the accuracy of both chordwise and spanwise loading distributions and brought the theoretical results closer to experiment.—*Tsakonas, S. and Jacobs, W. R., Journal of Ship Research, December, 1969, Vol. 13, pp. 237–257.*

A New Component of Viscous Resistance of Ships

The findings on a new component of viscous resistance are outlined. From the experimental study and some theoretical approaches it is pointed out that:

1) the new component of resistance is associated with the expenditure of energy in generating turbulence due to the breakdown of waves at the bow of ships and governed by Froude's law;
2) in particular at ballast condition of fuller ships the component occupies most of the so-called wave resistance derived by the usual three-dimensional extrapolation method such as Hughes' method. On the other hand the wave resistance derived from the measured wave patterns is quite small;
3) the component can be derived from the measured head loss which appears near the free surface and outside the usual frictional wake belt.

—Baba, E., Jnl. of the Society of Naval Architects of Japan, June 1969, Vol. 125, pp. 23–24.

Theoretical Investigations on Slamming of Cone-shaped Bodies

A general expression for the pressure distribution on a cone penetrating the water surface is derived from the similar procedure developed by Wagner for a wedge. The effect of trapped air between the falling body and the water is ignored for this investigations. The pressure distribution, velocity potentials, stream functions, and piled-up water phenomenon are summarized and compared with those for the wedge.—*Sheng-lun Chuang, Jnl. of Ship Research, December, 1969, Vol. 13, pp. 276–283.*

The Representation of Ship Hulls by Conformal Mapping Functions

A method is described by which an arbitrary hull surface may be approximated by an analytic function. The cross sections of the ship are represented by conformal mapping functions whose coefficients are polynomial functions of the longitudinal co-ordinate of the ship. Such a representation is intended to be used primarily for hydrodynamic calculations. However, the procedure used in generating the mapping function representation of the hull surface can, with some slight modifications, be used to freely form a hull surface. This could possibly be the basis of a mathematical hull design procedure.—*von Kerczek, C. and Tuck, E. O., Jnl of Ship Research, December, 1969, Vol. 13, pp. 284–298.*

Problems Arising from the Water Cooling of Engine Components

The local heat flow through the walls of components which form the boundary of the combustion chamber of internal combustion engines operating at high engine ratings exceed those found in most engineering equipment. Despite this the coolant-side face is normally in the "as cast" condition, sometimes with core sand adhering to it or embedded in the surface and a rust film. This paper describes the techniques employed in, and gives the results of, an extensive series of tests into the effect of such surfaces and additives on heat transfer and then goes on to describe the use of the resulting data to predict the operating temperatures of Diesel engine components in service.—*French, C. C., Paper presented at a meeting of The Institution of Mechanical Engineers, February 11, 1970; paper No. P29/70.*

Laminar Flow Gas Turbine Regenerators—The Influence of Manufacturing Tolerances

Several current designs for high effectiveness gas turbine regenerators involve low Reynolds number fully developed, laminar flow type surfaces. Such surfaces consist of cylindrical flow passages, of small hydraulic radius, in parallel. The cylinder geometry may, as examples, be triangular, as in some glass-ceramic surfaces, or rectangular, as in deepfold metal foil surfaces. This presentation demonstrates that manufacturing tolerances of several thousandths of an inch in passage dimension have a significant influence on the overall heat transfer and flow friction behaviour.—*London, A. L., Transactions of the A.S.M.E., Jnl of Engineering for Power, January, 1970, Vol. 92, pp. 46–56.*

Bending of Curved Circular Tubes

Based on the improved general solution for a thin, circular tube subjected to in-plane end moments, the effect of the radius ratio on the stress distribution, rigidity, and stress intensification factors is studied. The existing asymptotic solutions are re-examined and modified to reflect the effect of the radius ratio. The modified asymptotic formulas are compared with the existing experimental results.—*Cheng, D. H., and Thailer, H. J., Trans. A.S.M.E. Jnl of Engineering for Industry, February 1970, Vol. 92, pp. 62–66.*

Plastic Failure of Fibre-reinforced Materials

The paper is concerned with failure by plastic flow of the matrix of a fibre-reinforced composite sheet. For unidirectional reinforcement, a completely general state of plane stress in the median plane of the sheet is discussed, but for a two-ply laminate with an arbitrary angle between the two systems of reinforcing fibres only simple tension in any direction to the fibres is investigated.—*Prager, W., Transactions of the A.S.M.E., Jnl. of Applied Mechanics, September, 1969, Vol. 36, pp. 542–544.*

Mercury Slipring Assembly Designed for Winches

Meridian Laboratory Inc., has introduced a new mercury slipring assembly for application on oceanographic winches where high reliability and minimum electrical noise are required.

Engineered for shipboard use, the unit is made with a stainless steel housing, static o-ring seals, and a rotary lip seal to assure a totally sealed assembly. Stainless steel connectors are available with the unit.

The slipring is available with two to 40 terminals. It is designed to handle a wide range of voltage and current—from microvolts to over 600 volts and up to 35 amperes at speeds to 600 rev/min.

Rotational resistance change is less than 0·25 milliohm and contact resistance is 5 milliohms maximum.

The assembly's high performance capability permits television picture transmission without the need for a rotary coaxial joint.—*UnderSea Technology, November 1969, Vol. 10, p. 30.*

An Analysis of Cavitation Erosion

The paper is a review of results in the area of the mechanism of material erosion by cavitation. Experimental apparatus is described for investigation of impact erosion by a water jet and for studies involving cavitation erosion in a water tunnel. Qualitative and quantitative comparisons of the erosion resulting from each mechanism are made.

The author concludes that cavitation erosion research, especially material resistance evaluation work may be adequately carried out using impact erosion apparatus.—*Canavelis, R., Houille Blanche, 1968, Vol. 23, No. 2/3, pp. 189–196; Applied Mechanics Reviews, July 1969, Vol. 22, p. 749.*

Two New Turbo-Superchargers

The new turbo-superchargers (VT50 and T50) developed by Cummins Engine Co. Inc. (U.S.A.) incorporate various improvements, e.g.,

a) a bigger diameter of the turbine impellers, which reduces the speed and counters-pressure, thus ensuring better throughflow and heat dissipation;
b) to provide an effective oil seal the conventional seal has been replaced by a piston ring;
c) a built-in heat shield offers better protection against the heat of the exhaust section of the turbine;
d) the bearing housing has five fins improving heat transfer to the lubricating oil, thus reducing thermal deformations of the housing.

A new turbo-supercharger of Brown Boveri A.G. (VTR 900) was developed for big marine engines with cylinder bores upwards of 1000 mm. The new unit has an axial single stage turbine in a water-cooled housing with thick walls. The compressor is a radial single-stange compressor in a cast iron housing with sound isolation. The maximum compression ratio is 1 : 3 against 1 : 2·5 so far available, which allows bigger power levels.—*Versloot, J., Polytechnisch tijdschrift, 1970, Vol. 25, No. 8, p. 342–343.*

Noise and Vibration Research Applied to Diesel Engines

There is an increasing need to give attention to noise and vibration in many fields of engineering today. The multiplicity of problems arising is reflected in the increasing demand for the services of research organizations specializing in this work.

Although the reduction of noise and vibration forms a large part of this work, there is also considerable interest in the use of vibration signals as a means of obtaining information on the condition and performance of engines and other machinery.

In this article some of the techniques used in Diesel engine noise and vibration investigations are described, together with applications of current interest.—*Gray, G., Diesel Engineers and Users Association, January 1970. Publication No. 329.*

High Strain Torsion Fatigue of Solid and Tubular Specimens

A stress-strain history theory is developed for the entire life of solid cylinders subjected to reversals of torque under low-endurance fatigue conditions. Thin tubes and solid cylinders of a hot rolled mild steel are found to have identical stress range *versus* strain range relationships. However, fatigue behaviour is dependent on the stress field surrounding the crack initiation zone and hence the geometry of specimens has a most important effect in high strain fatigue processes. Cold drawn mild steel tubes that have an internal to outside diameter ratio in excess of 0·5 have reduced lives because of the absence of the constraining, less highly stressed core.—*Paper by Miller, K. J. and Chandler, D. C. presented at a meeting of The Institute of Mechanical Engineers, 14th January, 1970; paper No. P25/70.*

Midship Wave Bending Moments in a Model of the Mariner—Class Cargo Ship "California Bear" Running at Oblique Headings in Regular Waves

Vertical and lateral wave bending moments were measured at the midship section of a 1/96-scale model of the C4-S-1A Mariner-class cargo ship *California Bear*. The model was self propelled through a ship speed-range of 10 to 22 knots at seven headings to regular waves of lengths between 0·2 and 2·0 times the length between perpendiculars; moderate wave heights not exceeding 1/50 of the model length were used. Results are presented in charts of bending-moment-amplitude *versus* ship speed, with wave length as the parameter. Two ship loading conditions, representative of actual westbound and eastbound trans-Pacific voyages are covered.—*Numata, E. and Yonkers, W. F., Ship Structure Committee, November 1969, Report SSC-202.*

Effect of Residual Stresses on the Low Cycle Fatigue Life of Large Scale Weldments in High Strength Steel

A determination was made of the influence of various mechanical finishing procedures on residual stresses and the resulting effect on the low cycle fatigue life of tee-fillet welds in $1\frac{1}{2}$ in thick rolled steel plate with a yield strength of 80 000 lb/in^2. Included in this work were tee-fillet welds in the as-welded, ground, shot-peened, ground and shot-peened, and mechanically peened condition. Residual stresses were measured by a hole drilling technique developed at the Naval Applied Science Laboratory for application to linearly varying bi-axial stress fields. It was found that tensile residual stresses do not have a significant effect on fatigue life for the type of pulsating load used. Compressive residual stresses have been found to have a beneficial effect on fatigue life.—*Cordiano, H. V., Trans. A.S.M.E. Jnl. of Engineering for Industry, February 1970, Vol. 92, pp. 86–92.*

Experimental Studies of Orthogonally Stiffened Steel Plates

A series of ultimate load tests were performed on orthogonally stiffened welded steel plates to study the distribution of strains between ribs and plate and to compare actual load-deflexion behaviour with that predicted by a linear analysis. Results show that rib spacings will determine whether membrane or bending stresses predominate. When membrane stress is greater, the plate strains equal the rib strains. For a largely bending action, the rib strains are several times the plate strains. Linear analyses are seen to be overly conservative in predicting plate behaviour since they omit the important effect of membrane action.

Current design practice is based on linear analyses and thus is limited in its scope. The structural designer is made aware of the magnitude of deviation between actual and estimated behaviours.—*Kagan, H. A., Welding Journal (N.Y.), December 1969, Vol. 48, pp. 571-s-576-s.*

Under the Deep Oceans

A better understanding of world-wide geological processes is being made possible by the studies that are carried out by oceanographers. In particular, the deductions made from geophysical observations are now being checked by an extensive programme of drilling through the sea-bed of the deep oceans. The results to date confirm the hypothesis of sea floor spreading and continental drift, and at the same time increase knowledge of possible mineral wealth from below the sea-bed. Both the engineering employed in deep-sea drilling, and the new techniques that will be required for future exploitation of sea-bed minerals call for engineering inventiveness. The paper brings the scientific picture up to date in order that engineers may visualize what problems lie ahead in this sphere of activity.—*Paper by Gaskell, T. F., read at a meeting of The Institution of Engineers and Shipbuilders in Scotland, 16th December, 1969.*

Patent Specifications

Hydraulic Rudder-Actuating Gear

An object of this invention is to provide hydraulic rudder-actuating gear which occupies a relatively small space compared with conventional gear.

Referring to Figs 1, 2 and 3, the rudder-actuating gear comprises a single housing formed with two arcuate cylinders (1) and (2), the centrelines of which are respective arcs of a common circle. Pistons (5) and (6) are sidably mounted in the cylinders, and being pivotally interconnected by a common hinge (7), as shown in Fig. 2. The cylinders and the pistons have circular cross-sections.

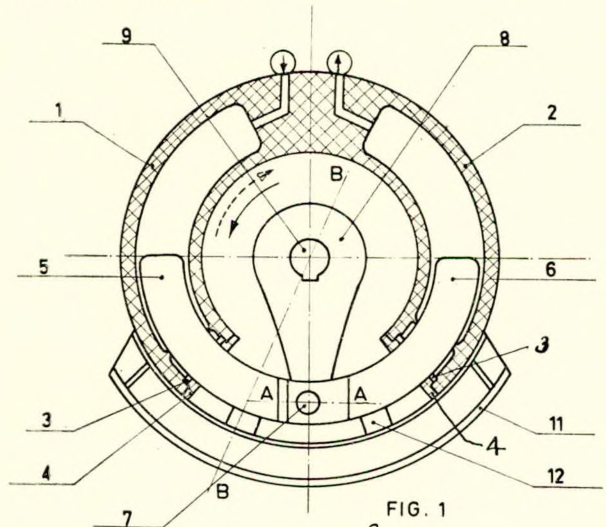


FIG. 1

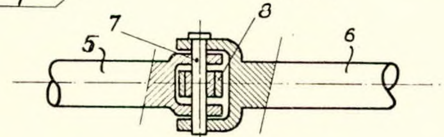


FIG. 2

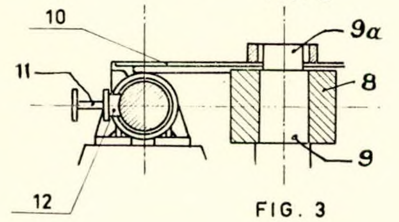


FIG. 3

The hinge pin (7) also pivotally connects each piston (5) and (6) to one end of a tiller arm (8) which is keyed at its other end to a rotatable about the axis of a rudder shaft (9).

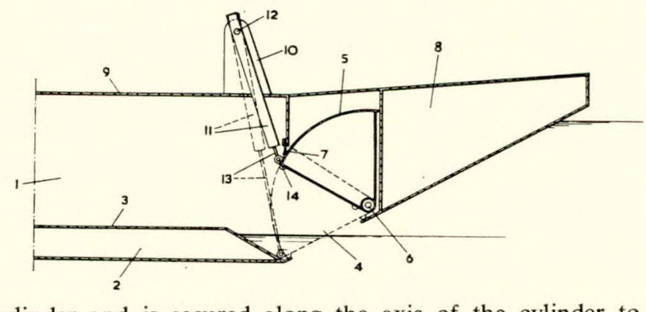
To lend rigidity to the assembly the cylinders (1) and (2) are interconnected externally by a fixed connecting plate (10) (Fig. 3) through which passes a cylindrical upper part (9a) of the rudder shaft (9) journalled so that the cylinders (1) and (2) are maintained concentric with the axis of rotation of the tiller arm (8). In operation the gear the pistons (5) and (6) slide over blocks 12, the latter transmitting outwardly directed components to the rigid housing.

In operation, hydraulic fluid under pressure is supplied to the cylinder (1) or (2) by means of a pump, and a fluid pressure differential is established between the cylinders by means of a suitable control valve to cause positive movement of the pistons (5) and (6) in unison in either circumferential direction selectively. In the case of Fig. 1, the pistons are moved in an anticlockwise sense causing corresponding rotation of the tiller arm (8), as indicated by the full line arcuate arrow.—*British Patent No. 1 174 028 issued to Ajamil, L. A. Complete specification published, 10th December 1969.*

Hopper Barges

This invention relates to a hopper barge for transporting material to be dumped in open water.

The figure shows, a barge comprising a cargo hold (1) with a double bottom (2), the upper wall of which is denoted by numeral (3). A passageway (4) in the stem communicates with the cargo hold and can be closed by means of a valve (5). The latter comprises a segment of a



cylinder and is secured along the axis of the cylinder to a hinge (6). A scraper member (7) is arranged to engage with the cylindrical surface of the valve to dislodge any debris as the valve (5) is opened.

The barge has an enclosed forepeak (8), separate from the cargo hold (1). The latter is covered by a deck (9).

A hydraulic jack (11) is provided to operate the valve (5). The jack is hinged at (12) in a stand (10) on deck (9). The piston rod (13) is hinged to valve (5) at (14).

The barge is used in the following manner. The valve (5) and a corresponding valve in the stern are closed and the cargo is pumped on board. At the place of discharge, the vessel stops and both the valves (5) in the stem and at the stern are simultaneously opened. Since the valve members are hollow, their buoyancy tends to assist this. The cargo then flows out and as a result the barge will rise so that the upper wall (3) of the tank (2) is above the water level. When the barge has been unloaded and is homeward bound again, the valves (5) are left open in order for the last remainders of the cargo to be scavenged from the hold by water washing over the upper wall (3). When the barge has entered the harbour and lies still, the upper wall (3) of the double bottom (2) will be dry and the valves (5) may be closed.—*British Patent No. 1 171 380 issued to Vurk, A. and Zonen's Scheepswerven N.V. Complete specification published, 19th November 1969.*

Dock for Ships

This invention relates to docks for ships, particularly floating docks. More specifically, it is concerned with the handling of very large ships such as supertankers and other vessels of 500 000 and upward tons deadweight. It is an object of the invention to provide docking facilities for the building, repair or periodic maintenance of the largest ships without the need to construct a permanent dry dock or a giant floating dock.

The invention provides a floating dock which has dock sea gates at at least one end while at least the other end has means for effecting a seal between the dock sides and bottom and the parallel mid body of the ship's hull. With this arrangement, the dock does not need to be long enough to accommodate the whole length of the ship but only to extend from beyond one end of the ship up to some point along the parallel mid body of the ship. Figs 1 to 7 illustrate the manner of use of the dock. Fig. 3 is a transverse section showing the dock positioned around the parallel mid section of a ship. The dock is sunk by means of its ballast tanks to a depth a little greater than the ship's draught. The ship is then floated into the dock and air bags (15) inflated to form a seal (14) between the dock side walls and bottom and the ship's hull, so that the working space in the dock can be pumped dry. Fig. 2 shows two seals (one such set for each end) creating a coffer dam (20). For working on the aft end of the ship, the end of the dock equipped with sea gates (12) (shown in Fig. 1 and Fig. 7) remains abaft the vessel as shown in Fig. 4. The gates are closed and the seals at the opposite ends of the dock are inflated so that a working space is created around the ship's stern between the sea gates and the inflated seals. The seals at the gated end of the dock are not used. Similarly, for working on the bow end of the ship (Fig. 7) the dock is situated with its gated end forward of the bow, the sea gates are closed and the seals at the opposite end of the dock are inflated against the ship's hull as before.

Fig.1

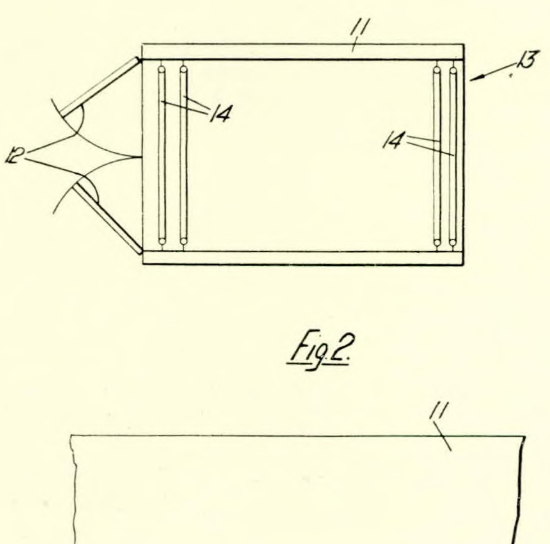


Fig.2

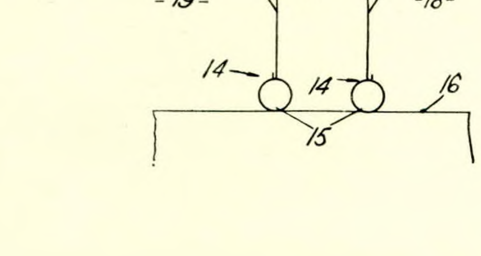


Fig.3

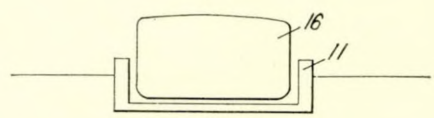


Fig.4

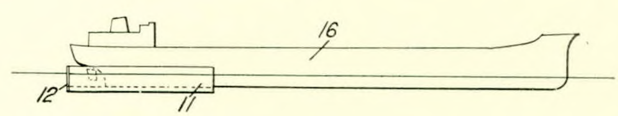


Fig.5

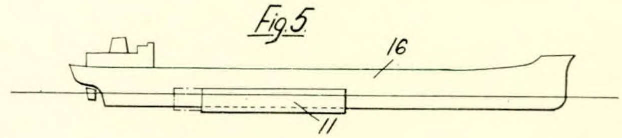


Fig.6.

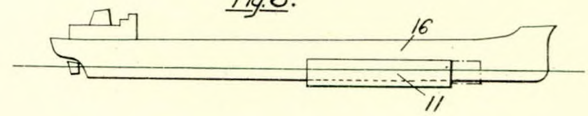
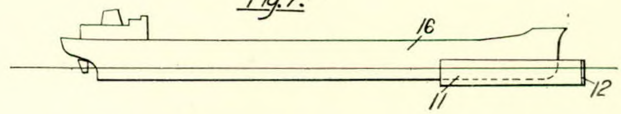


Fig.7.



For working on the parallel mid body of the ship, as illustrated in Figs 5 and 6, the sea gates are kept open and the working space is created by inflating the seals at both ends of the dock against the ship's hull. Figs 4 and 7 together show how any part of the length of a very large ship can be worked on using a dock which is not much more than a quarter the length of the ship.—*British Patent No. 1 169 295 issued to Appledore Shipbuilders Ltd. Complete specification published November 5th 1969.*