Thus the main machinery overhaul can be done in a day or two. Since in peacetime the main engines are used more overhaul ashore and a new unit can be installed immediately. car. take the engine change unit out of the ship for hot parts, can be designed to be easily replaced. This means change unit which includes the combustion chambers and short life, say 5-10,000 hours and the other for a long life, A marine gas turbine can be built in two parts; one for a could work on it. repair of all the pumps and pipes, valves and controls associated and lack of accessibility restricted the number of men who because the machinery was compressed into a small space with the dispersed steam plant. This work took a long time retubing, for condenser cleaning and for examination and availability needs to be just as high. Hitherto, the steam hands or fleet maintenance bases for boiler cleaning and machinery has dictated a relatively long period in dockyard usage is much less than that of a merchant ship but our availability needs to be just as high. Hitherto, the steam machinery has dictated a relatively long period in dockyard production orders are too small. The diesel is too big for our shipping. We need therefore to have our ships seen to be at sea to deter any foreign power from interfering engine. Another advantage gas turbines give us is the ability to sail from harbour at very short notice and to increase speed from 5 to 30 knots in a minute or two. The machinery is very responsive to demands for power. This feature is attractive since ships in harbour are vulnerable and high accelerations are needed to attack the faster nuclear submarines at sea today. We could not have made this change, and it is a complete change, from steam and diesel to gas turbines, if we had not had 23 years of sea experience with gas turbines to build on. My predecessors in the Admiralty were farsighted engineers. As early as 1942 the use of the gas turbine in fast motor boats was foreseen. We have come a long way since then and now I wish to survey the steps we have taken and to show why we have decided to use the engines we have chosen.

The Navy's requirements for a warship propulsion system

The payload of a warship is its weapons system. The proportion of a warship's displacement devoted to the propulsion plant and fuel has to be kept to a minimum to allow the maximum payload. We therefore seek small, compact, light-weight and efficient machinery. We need efficient machinery so that the fuel stowage required can be a minimum for a given endurance. However, this is not a straightforward requirement since the maximum endurance is specified at a cruising speed which varies from 14-20 knots, depending upon the type of ship. The propulsion machinery should also be designed to operate with the minimum number of men on watch and to require than any other system, except those associated with domestic services, this means the peacetime availability of a fully efficient warship can be increased.

Another advantage gas turbines give us is the ability to sail from harbour at very short notice and to increase speed from 5 to 30 knots in a minute or two. This enhances the ability of our warships to attack the faster nuclear submarines at sea today. The Machinery is very responsive to demands for power. This feature is attractive since ships in harbour are vulnerable and high accelerations are needed to attack the faster nuclear submarines at sea today. We could not have made this change, and it is a complete change, from steam and diesel to gas turbines, if we had not had 23 years of sea experience with gas turbines to build on. My predecessors in the Admiralty were farsighted engineers. As early as 1942 the use of the gas turbine in fast motor boats was foreseen. We have come a long way since then and now I wish to survey the steps we have taken and to show why we have decided to use the engines we have chosen.

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Inhoud van dit nummer:

Main Propulsion Gas Turbines in the Royal Navy door Captain P. D. Tatton-Brown.
On the critical speed range of ships in restricted waters door J. P. Hoof.
De viscositeitsregelaar van AB Källé-Regulator.
Onderwatertechnologie.
Nieuwsberichten.
the minimum number of manhours for routine maintenance. One way to reduce the watchkeeping crew in Frigates and Destroyers is to control the speed of the ship directly from the bridge.

A warship must be able to resist nuclear, biological and chemical attack. For the propulsion plant this means that the air taken in for combustion shall not come in contact with any of the crew. Any radioactive particles, bugs or foul gases will be drawn deep into the engines and discharged into the ship through a closed system. The crew, in the meantime, will be in a gastight citadel breathing filtered air. Warships' machinery must also withstand high shock forces and be able to operate when partly submerged.

Operationally, we need quiet machinery which will not radiate noise into the sea to set off acoustic mines and not be easily detected by hostile submarines. We need machinery which will operate efficiently in the arctic and the tropics and always start up quickly. Finally, we have called for a high probability that it will run without failure continuously for several thousand hours.

We have found that all gas turbine propulsion systems are now capable of meeting our requirements better than any other type of propulsion system. They are compact, lightweight and reasonably efficient at their full power rating. The lightweight, compact machinery can be installed on shock resisting and noise isolating mountings. A gas turbine's combustion air and cooling air must be ducted to the engine in a closed system. These ducts are much larger than those required for other propulsion systems since a gas turbine is very sensitive to intake pressure losses and this is a considerable handicap to the warship designer. So we are getting from the gas turbine many desirable features but we have to accept two handicaps - a need to burn relatively expensive diesel fuel and a need for large intakes and exhaust trunks.

In our new Frigates we are installing 50,000 BHP (37.3 MW) in place of the 30,000 SHP steam plant fitted in the Leanders.

The Engine Room crew for this larger Frigate is reduced from two officers and 47 men to one officer and 28 men. The Engine Room crew for this larger Frigate is reduced propulsion plant and the change from random, unplannable defect rectification in a steam plant to the more predictable maintenance of a gas turbine. Although unexpected failures of the gas turbines will occur, the effect on the ship will be reduced since we fit two engines per shaft and full performance can be restored by replacing the failed unit. Another saving is the reduction in the number of underwater fittings. The number of holes in the ship's bottom and the size of the holes are reduced considerably. We have made the most of this aspect, thereby further saving ship's staff work.

To sum up, Naval requirements now are:
1. an ability to operate world-wide;
2. ability to stay at sea for extended periods;
3. ability to get underway quickly;
4. ease of operation by remote control;
5. good endurance at Fleet speeds;
6. good resistance to damage and to nuclear, biological and chemical warfare;
7. high power to weight ratio;
8. minimum shipboard maintenance for good availability and minimum crew.

Gas Turbines in Fast Patrol Boats

The first marine gas turbine went to sea in 1947. It was fitted in MGB 2009, a gunboat, a "Gatric" engine, an aircraft jet engine with a specially designed power turbine and gearbox. It had a 9-stage axial flow compressor with aluminium blades fitted on a forged aluminium drum and carried on two oil mist lubricated ball bearings. A 4-stage power turbine with molibdenum-vanadium blading drove the ship through a double-helical single reduction gearbox with a Sinclair self-synchronising clutch. The gas generator was started by a 24 volt DC motor which spun the compressor at 1,000 r.p.m., when fuel was admitted and the motor was kept driving up to 2,000 r.p.m. which was the self-sustaining speed. Idling speed was 3,000 r.p.m. From idling to full speed took 15 seconds.

The gas turbine and its gearbox and clutch were subjected to a few months shore testing in a test house. The air intakes to this test house were fitted with filters to prevent rapid deterioration of compressor performance by blade fouling because of the smoke laden atmosphere. The engine developed 2,550 b.h.p. at 1,087 r.p.m. with a compressor speed of 7,220 r.p.m. and a specific fuel consumption of 1.06 lb/b.h.p./hr. Diesel oil was used. The specific weight of the Gatric without its gearbox was 1.72 lb./h.p. compared with the 1,300 h.p. Packard engine which it replaced and which had a specific of 2.25 lb./h.p.

MGB 2009 sailed into the Solent for sea trials in August 1947. 50 years after the Turbinia and both vessels did about 34 knots. They were both the forerunners of a long line of turbine plants. (Figures 1 and 2.) The contract for the Gatric was placed in 1943 with Metropolitan Vickers.

During the trials with MGB 2009, many useful lessons were learnt. The importance of the intake design was first discovered, the effect of salt deposit on the compressor blading was measured, the low maintenance requirement in the first 50 hours at sea surprised the crew but the noise distressed them. Oil and dirt on the compressor blading became a problem and the technique of water washing to remove the salt was developed.
The next class of Fast Patrol Boats was designed around engines installed in Brave Borderer. This Class of Fast Patrol Boats were established for material investigations. One engine was run at the Naval Marine Wing, N.G.T.E., in which the engine has run for over 5,000 hours with regular Carboblast cleaning. The essential difference between the two installations was that the trials engine installation incorporated a large fine mesh filter in the lower pressure side of the lubricating system whereas the FPB's system did not. Unfortunately, lack of space prevents fitting of such a filter and therefore other measures have to be relied upon to prevent a recurrence of the failure. These are as follows:

- Substitution of a synthetic lubricating oil, Specification DERD2487 (RN designation OX38) for the less thermally stable mineral oil, Specification DERD2497/1 (RN designation OEP71) formerly used.
- Restricting the use of the compressor cleaning procedure and improving the servicing of the lubricating system.

In two years of operating experience since the incident, there have been no further troubles with lubricating oil. The Royal Navy has accepted a few months ago, the latest design of fast craft from Vospers Thyroncroft Ltd.

This is to be a Class of three fast training boats, each propelled by two Rolls Royce Proteus gas turbines for "sprint" and diesels for cruising. The Proteus has been developed to give 4,250 b.h.p. at the "sprint" rating at 15 °C. The engine installation is very similar to that adopted in the very successful Brave Class.

The Proteus has been used at the National Gas Turbine Establishment for material investigations. One engine was run with controlled airborne salt ingestion at the Naval Marine Wing. A Nimonic 90 first stage turbine stator segment was used to monitor corrosion and demonstrated the effect of salt contamination levels as shown below:

Fig. 3. HMS Bold Pioneer. This Fast Patrol Boat was fitted with two Metropolitan Vickers G2 Gas Turbines and ran into its trials in 1951.

Fig. 4. The G2. Two of these engines were installed in the Bold Pioneer. The photograph shows the method of mounting and the enclosure round the hot parts. 4,500 SHP.

Fig. 5. HMS Brave Borderer. This Class of Fast Patrol Boats were fitted with the British Siddeley Proteus Gas Turbine. Three gas turbines in each craft. The intakes are behind the bridge facing aft and the exhaust discharges through the transom.
Fig. 6. The Marine Proteus Gas Turbine. This is an unusual engine with the intake and exhaust at the same end of the engine. The compressor is shown under the combustion can, the last stage is centrifugal and this reverses the air flow.

At the end of the last of these runs, a pack aluminised Nimonic 105 first stage turbine blade failed and sulphidation was detected in the blades from this row. These blades had been subjected to both the test at 0.05 p.p.m. sodium chloride and the final one at 0.005 p.p.m. sodium chloride. It is noteworthy that the attack appeared to occur only during the last few hundred hours. This was considered to be due to local failure of the pack aluminised coating. Under operational conditions at sea in a Fast Patrol Boat, severe corrosion occurred in Proteus cast nozzle guide vanes in Nimonic 90 after 800 hours. A change in material to pack aluminised X40 produced dramatic reductions in the rate of corrosion. (Figure 7.)

From our extensive experience in fast small craft, we have learnt the troubles that can arise in gas turbines in ships. The troubles are principally concerned with the salt in the air and fuel. As a result of this salt, considerable development in materials has been undertaken and water and carbo-blast compressor cleaning has been used and this has led to other troubles.

In all these vessels, diesel oil was burnt in the gas turbine. No consideration was given to the use of heavier or cheaper fuel. All these engines were simple cycle sets with no components added in order to achieve better part load performance. We now come to two attempts to produce complex gas turbine propulsion plants.

In 1946, a contract had been placed with Rolls Royce for the development of a naval gas turbine to drive a 200-ton gunboat. This contract was an attempt to produce an efficient engine based on the latest aero practice. The engine was required to provide for economical low power cruising so a complex cycle consisting of LP axial compressor, intercooler, centrifugal compressor 1st stage, intercooler, centrifugal 2nd stage, heat exchanger, combustion chamber, three single-stage turbines and one two-stage power turbine was chosen. The life of this engine was to be 1,000 hours of which 300 were to be at full power. The prototype engine ran ashore in 1951 and developed 5,300 h.p. This was a compound cycle gas turbine chosen after a detailed comparison of different gas turbine designs. This detailed comparison led to the elimination of the use of reheat due to the difficulties of control but apart from this, it was a very complex cycle requiring a lot of preliminary and component development. The installation was remarkable for a marine gas turbine since its specific fuel consumption curve was flat between 40 per cent and 80 per cent power. (Figure 8.)

The lessons learnt from the RM60 were many; for the first time it became clear that ball and roller bearings cannot give the life and reliability required for naval gas turbines. The problems of noise arose and the sensitivity of the engine to variations in ambient air temperature was noted. The power fell off from 5,400 to 4,500 at 85 °F. The RM60 was indeed a bulky and expensive engine.

In 1946, a contract was placed with the English Electric.
Company for a gas turbine to replace one of the steam turbine plants in HMS Hotham, a Frigate with steam turbo-electric machinery. It was hoped to install this set, the EL60A, on one shaft of the Frigate thus getting a direct comparison with the steam set. A long life gas turbine was to be built with a simple cycle with heat exchange, the compressor turbine running in parallel with a similar smaller power turbine driving an alternator. (Figure 9.)

The set was ready for shore trials in 1951. By this time the RM60 was running ashore and the heavier EL60A design, after its shore trials was scrapped in 1952. The plant was heavy, 27 lb/h.p., it was not very flexible in operation and it showed few advantages over a steam plant. Clearly, heavy-weight gas turbines constructed like steam turbines were not going to be suitable for warships.

Both the RM60 and EL60A were to be developed to give good part load performance. Figure 10 showed how well this was achieved but only at the expense of high first cost and a loss of the simplicity and lightness which had been the great advantages of these engines.

The Boost Principle

In the original MGB 2009 and in the Bold Pioneer, the gas turbines were for use for high speeds only. The manoeuvring and cruising was all carried out on IC Engines. This recognised the weakness of the gas turbine, its poor efficiency at part loads. Endeavours to overcome this by using intercoolers, reheat and complex cycles had not proved successful.

The most suitable approach for Fast Patrol Boats was a diesel for low speeds and a gas turbine for high speeds. The most suitable approach for Frigates and larger ships seemed to be steam with gas turbine boost.

Extensive studies showed a 15,000 SHP steam plant and two 7,500 SHP gas turbines as the most suitable for the large County Class GMDs (Figure 11) and a 12,500 steam plant and one 7,500 gas turbine for the smaller Tribal Class Frigates. In this way, a single design of gas turbine would be suitable for two classes of ship. So in the mid fifties, the decision was made to go to COSAG (Combined Steam and Gas Turbine) drive on a single shaft. This was an excellent way of getting gas turbines to sea. The steam plant could provide the long life base load and manoeuvring power unit with the compact gas turbine providing the boost power.

The G6 gas turbine was actually the fifth designed by the company which in 1961 was known as AEI. For the first time, a major warship was to have gas turbines and they had to face new requirements for gas tightness, shock, ability to operate partly submerged in water and high reliability.
The design was very conservative and the weight per s.h.p. was increased compared with the G2. An extensive shore testing and proving of the engine was carried out at the manufacturer’s works with supporting experimental development at the National Gas Turbine Establishment. 1,400 hours of development running were carried out ashore before HMS Ashanti went to sea in 1961.

Now the gas turbine had to prove its reliability and maintainability in direct comparison with a steam plant in the same ship. In the last nine years, the Navy has grown to be quite confident in the ability of the G6 to meet all that has been asked of it. No longer is it purely a boost engine but it can be used to drive the ship alone and a reversing gear train has been installed so the ships can leave harbour on gas turbines alone. The G6 engines have done over 70,000 hours. Recently some ships have done 14,000 hours on 4-G6 engines in a year and COSAG ships are spending at least 20 per cent of their time on gas turbines alone. They are popular at sea because they are quick to light up and require fewer men on watch. The Fleet is now building up G6 engine hours at the rate of 20,000 per year.

The G6 gas turbine (Figure 12) has raised many interesting technical problems, not least has been the reclamation of worn shaft journals. Hitherto, the use of chromium plating had been prohibited for high speed shafts. In the G6, the shafts run with peripheral speeds of 200 feet per second. The only permitted salvage was the machining of the shaft to one or other of two standard undersizes and the provision of suitable undersized bearings. Although such an expedient is workable when dealing with mass production quantities of engines and spares, its application to the small numbers involved in this case threatened to create a logistic problem and obviously the only desirable and acceptable solution would be to have engines and spares with only one size of bearing and shaft.

After reviewing general experience of chromium plated shafts in other applications, the decision was taken to fit one in a seagoing G6 engine and assess its performance under operational conditions. After two years running, the shaft was removed for detailed metallurgical examination and passed as satisfactory.

Chromium plating has therefore been adopted as a standard salvage procedure and all shafts and bearings will be of the same nominal size and will have the same tolerances.

The Rolls-Royce Olympus Marine Gas Turbine — TM1A

This engine was originally developed in 1953 to propel the Vulcan V Bombers. In 1962, the industrial version of the engine was commissioned for the CEGB at Hams Hall, Birmingham. In 1964, the Admiralty ordered an Olympus gas turbine to be developed for marine propulsion, as a boost engine.

The initial changes from the aero engine included the following:

a. change of compressor blade material from aluminium to stainless steel;

b. new turbine blade material;

c. increased thrust bearing capacity;
d. modified combustion chambers to burn diesel fuel at sea level;
e. strengthening of the casings to resist the high shock loadings which may be caused by underwater explosions.

The test running on the Olympus TM1A began at Ansty, near Coventry, in August 1966. There were a succession of small defects which were cured by modifications until in May 1968 after 1,500 hours running, a first stage LP compressor blade failed. Within three weeks a similar failure occurred in the Olympus TM1A which had just gone to sea in *HMS Exmouth*. The subsequent investigation of this defect and its cure is described later in this paper.

During the development running of the TM1A, there were failures of the first stage turbine blades. The failures usually occurred in the top root serration or near this point. The failures were caused by the blade vibrating at its first flap resonance; this fatigued the material at a highly stressed point in the blade root. The dimensions of the blade root and disc were such that the force to hold the blade in the disc could be concentrated on the top serration. The blade manufacturing tolerance also permitted a rather large variation between blade shroud tip gaps. These two factors led to reduced damping of the first flap vibration. Superimposed on this problem was a rotor disc resonance at high speed. In addition, the blade was designed to give the optimum balance between gas bending and centrifugal stresses at high altitude corresponding to the aircraft cruise rating.

An extensive investigation was necessary to discover the causes of these failures and produce remedies. The design of the blade was changed: it was leaned in the disc so the optimum stress distribution was achieved at sea level, closer tolerancing of dimensions ensured shroud tip gaps are even and the blade serrations were chamfered to reduce stress concentrations. At the same time, the turbine disc was made thicker to remove the resonance from the running range.

By the end of 1968, the Olympus TM1A at Ansty had completed 1,000 hours in its fully modified state with a running cycle equivalent to 3,000 hours in a Frigate. During this running, salt water was sprayed into the air intakes to simulate a marine environment. The TM1A is still at sea in *HMS Exmouth* and has been cleared for installation in *HMS Bristol* and in Frigates being built for Malaysian, Iranian and Libyan navies.

*HMS Exmouth* (Figure 13) went to sea in the Spring of 1968. She was the first major warship to be propelled entirely by gas turbines. The 15,000 s.h.p. steam plant was removed and replaced by two Proteus cruising gas turbines and one Olympus "high speed or boost" gas turbine driving one shaft with a controllable pitch propeller. This installation was designed to try out the Olympus TM1A at sea once it had been decided to fit the TM1A in *HMS Bristol*. In the *Bristol*, the 15,000 h.p. Olympus is the boost engine which is fitted in conjunction with a 15,000 h.p. base load steam plant.

*HMS Exmouth* and her Olympus TM1A (Figure 14) have taught us a lot of lessons. Among others, we have learned the need for very careful attention to be paid to the design of intakes and exhausts and for the fuel to be clean. A conversion can never be as successful as an original design and much of the trouble in *Exmouth* was due to the attempt to fit a large gas turbine into a small hull designed for a steam plant.

The conversion has been a success; we have developed a confidence in the marinised aero gas turbine, the Fleet has learnt to live with a gas turbine ship and we are now avoiding in a class of ships the mistakes we made in one.

The first lesson was learnt very soon after *Exmouth* first went to sea; as already reported, a first stage LP compressor blade failed and this coupled with the same failure in the shore test engine initiated an extensive investigation. Examination showed them to be rapid fatigue failures in the first or second flapwise mode. Severe fretting in the blade roots confirmed this diagnosis. This was surprising since earlier strain gauge investigations had shown no large strains on these blades. This earlier testing was done, however, on an engine with an unrestricted intake. So one possible cause to be investigated was excitation of the blades by a flow distortion from the intakes used in the Ansty test bed and in the ship.

Water and air models were tested to explore the flow conditions in these intakes. These tests showed conclusively that the flow conditions into the intake were very poor. A standing vortex was entering the front of the engine; this is clearly shown in the photograph of the water model (Figure 15). Strain gauge tests confirmed the presence of high vibration amplitudes.

The solution adopted was to fit a cascaded bend to give positive guidance to the air negotiating the turn into the engine. The water model tests showed that the flow conditions into the engine were little affected by distortion of the flow into the bend when the cascaded bend was fitted. (Figure 16.)

We have now decided to fit a cascaded bend at the intakes to all our gas turbines since we do not have the space in a warship to provide an intake plenum chamber of the size required to give the flow an opportunity to settle.

The propulsion turbine installations are designed to allow ready removal of the Olympus gas generator or Proteus engines through their intakes. The cascaded bend has made this less easy. Nevertheless, this route for the removal and replacement of gas turbine change units has been selected for all our ships.

The intakes have knitted-mesh filter arrangements for salt exclusion and acoustic lining and splitters for intake noise reduction. Provision was made for engine cleaning by either water-washing or by Carboblaster. The fuel supply system received close attention. The main tanks were cleaned as thoroughly as possible to remove all traces of boiler oil and the new fuel transfer system fitted was provided with coarse and fine (10 micron) filters for removal of solid particles and with coalescent filters for removal of water. The system downstream of the coalescer was constructed using copper-nickel-iron tubing.

The control system for the propulsion engines inter-relates...
the engine fuel controls and the propeller pitch control such that on any of the possible modes of propulsion the ship can be operated throughout the available power range using a single control level. This includes operation in reverse pitch.

In HMS Exmouth, machinery control is exercised either from the Machinery Control Room (MCR) or directly from the bridge. Engine start-up and selection is controlled from the MCR only.

The ship completed its first series of trials and the machinery was accepted for Naval service. She then headed north for cold weather trials, a major objective of these being to investigate intake icing. Very bad weather conditions were experienced, including those which result in accumulation of "black ice" on the ship’s upper works, but the intakes stayed free of ice except in conditions of heavy snow and low air temperature, when thin films of ice started to form on the outer wire-mesh guards of the intake. No icing of the knitted-mesh filters occurred at any time, but conditions of freezing fog, which would probably be the most likely to ice them up, were not encountered. Difficulties with the fuel system arose during operation in the extremely rough sea conditions, in that frequent blockage of the coalescer filter by dirt was experienced, particularly after refuelling at sea in these conditions. The trouble probably arose because of accumulation of dirt in the ship’s fuel tanks being stirred up by the ship’s motion, so that the fuel system was presented with heavy fuel contamination. A finer degree of particulate filtration upstream of the coalescer, using additional duplex filters, is being provided. Probably the tank cleaning was inadequate, as it is extremely difficult to clean out these tanks effectively, so dirty fuel may be less of a problem in new ships.

It has also become evident that under such sea conditions the intake filtration was not effective. The Olympus engine was removed for inspection after this series of trials, without the usual waterwashing. It was found to have fairly heavy salt deposits in the compressor, though was otherwise in quite a satisfactory state. It is probable that much of the salt ingress has been through leaks subsequently found around the edges of the filter pads, as the evidence is that such leaks can be extremely detrimental to filter performance. This is receiving attention but the other lesson which emerges is the need to locate intakes in well sheltered positions and to provide effective means of removing the bulk of the heavy spray before it reaches the pads.

An unexpected problem which has occurred with the Olympus has arisen from cleaning with Carboblast. Carboblast dust has caused clogging of air passages around the fuel burner tips, with the result that, in some instances, there has been extinction of one or more burners on rapid deceleration, with subsequent failure to relight and a resultant loss of power. Burner modifications are being introduced to avoid this blockage; meantime, the engine is being cleaned by waterwashing only. This problem had not arisen in the shore trial of the engine. In the shore trials schedule the Carboblasting was followed by a period of high-power running and this evidently cleared the passages.

In operation, the power plant has proved to be extremely versatile and simple to handle. Machinery maintenance has been far easier than for the steam installation, in part because of the improved accessibility and watchkeeping has also been relatively easy because of the monitoring equipment in the Machinery Control Room. The engineering staff has had no difficulty in making the change to the new propulsion system.

Fig. 15. Water analogy rig test or air intake in HMS Exmouth shows a stable vortex standing on the wall opposite the air intake.

Fig. 16. This sketch shows the modification fitted to the air intake in HMS Exmouth, a 90° cascaded bend.
Ship handling has been quite satisfactory and the Olympus can be and indeed is, normally used for manoeuvring in and out of harbour.

Temperatures in the machinery spaces are much lower than for a steam installation and the humidity is also much reduced. Noise levels in these spaces are fairly high and ear defenders are used. Upper deck noise levels have proved to be higher than desirable. Most of this noise is from the exhausts, which are unsilenced, so provision of exhaust silencers is being arranged.

It will be seen that a number of valuable lessons have been learned from this conversion, both in terms of confirmation that the general concept represents a suitable propulsion plant and also in showing aspects of the machinery and its installation which have required modification.

We have now achieved over 2,000 hours on the Olympus and have recently experienced a second failure of the 1st row LP compressor blading. This failure is due to pitting from the excessive salt that has been deposited on the engine. The failure has caused us to examine very carefully the methods we employ for removing salt from the air and the material of this blading. We have already altered the air intakes from those shown (Figure 13) which were fitted in August 1969. We are considering changing the 1st row LP compressor blades from stainless steel FV520 to a high nickel alloy INCO 718.

The Olympus/Tyne Main Propulsion Machinery Package

The next stage of our development programme has been to go from the combined steam and gas turbine plants to the all gas propulsion package. To do this, we could not go to a single gas turbine per shaft for several reasons. Consider three of the Navy's requirements from a propulsion system:

a. good endurance at Fleet speeds;
b. good resistance to damage;
c. ability to stay at sea for extended periods.

To get good endurance at Fleet speeds we need a propulsion plant that is efficient at 20 per cent of full power. This is one of the disadvantages of a gas turbine; its poor part load performance. To overcome this, a cruise gas turbine has been used which gives its maximum efficiency at about 4,000 h.p. This second gas turbine also provides the alternative propulsion unit in the event of action damage to or a random failure of the main engine. At present, we are not sure of the reliability of gas turbines so it is prudent to fit two per shaft. The main engine is the Olympus TM3B and the cruise engine chosen is the Tyne RM1A; both these engines are now under development for the Type 42 Destroyer (Figure 17) to be named Sheffield.

The Olympus TM3B has been developed as a narrower engine than the TM1A, more suitable for installation side by side in...
the narrow hulls of Destroyers and Frigates. This has been done by narrowing the exhaust volute and power turbine underframe. The gas generators and power turbines remain interchangeable.

**Comparison of Olympus TM1A and TM3B Engines**

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<th>TM1A</th>
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<tr>
<td>Length, overall</td>
<td>22.5</td>
<td>11.8</td>
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<tr>
<td>Height, from top of engine bearer to exhaust flange</td>
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<td>10.3</td>
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<tr>
<td>Width, overall</td>
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<td>8.0</td>
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<td>Weight, dry</td>
<td>25.5</td>
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The TM3B version of the Olympus engine is now under test at Ansty. (Figure 18). It has completed some 500 hours of various development running and is scheduled to complete 3,000 hours of testing by the end of 1971. At the TM3B rating the engine will develop over 27,000 b.h.p. at 15 °C with a specific fuel consumption better than 0.49 lb/h.p./hr. The higher power has been achieved by fitting cooled nozzle guide vanes and changing the first stage blade material to Nimonic 115. The TET is raised from 1150K to 1215K.

Shore tests are run on a naval operating cycle which involves starts, stops and frequent and rapid changes of power. The engines are also run with salt water injected into the air stream, 0.01 parts per million salt weight for weight. There is no substitute for endurance testing ashore. The Olympus TM3B will go to the Naval Marine Wing at the NGTE in 1972 for further endurance testing. Here, in addition to salt in the air, 0.6 p.p.m. sodium is added to the fuel and the uptakes and downtakes are identical to those installed in Sheffield. The aim is to get 10,000 hours running on the Olympus ashore to prove the engine’s reliability.

**The Rolls-Royce Tyne RM1A**

This engine, unlike the Olympus, has not been used as an industrial engine for power generation or pipe line pumping. On the other hand, it has achieved over 7 million cumulative hours flying in commercial airlines. The Tyne Mark 621 was tested at the Naval Marine Wing of the National Gas Turbine Establishment in 1967. This was an aero Tyne with materials changed to suit the marine environment. This test was run with air contaminated by 0.01 part per million by weight of salt in the form of natural sea water throughout. The trial consisted of 42 test cycles each of 24 hours duration and each cycle included 10 hours at full power. The trial was successful and the engine strip afterwards showed only slight corrosion of the h.p. turbine blades. The small air cooling holes in the h.p. turbine blades were not found to be blocked by salt as had been half expected. The combustion chambers had stood up well to the change to diesel fuel, the only failure was the burning of the flame tube flares.

This 1,000 hour run was followed by a period of compressor cleaning trials which showed that carboblast was perfectly safe but should be used sparingly. The best cleaning routine remains the use of fresh water in combination with small quantities of carboblast.

The Tyne 621 has since been installed in two hydrofoils built in the U.S.A. by the Grumman Aircraft Corporation.

As a result of the trials at the Naval Marine Wing, the Tyne was considered a suitable engine for warship propulsion. It was decided to develop the engine as the cruise engine for Destroyers and Frigates. There were three areas where deve-
Development was needed. The h.p. turbine blade material, the flame tube design and the LP compressor blade coatings. The salt water erosion of the LP compressor blade coatings could not be overcome by the use of new coating material so a change in the basic blade material to Titanium was decided upon. Titanium was chosen because its density is similar to that of aluminium so only minor changes are required to the compressor design. A change to cheaper but much heavier stainless steel blades would have required an expensive redesign and development of the compressor. Figure 19 shows the stages in the development of the Tyne from the turbo-prop engine through the first marine version to the Navy's free power turbine version the RM1A, which is here called by its earlier name the RB209.

The development of the engine can be traced from Figure 19 which illustrates one of the major changes in design. In the aero engine and the Mark 621, the power take off is on the left, the drive being taken from the LP compressor turbine. In the naval version, the RB209, the last two stages of the LP compressor turbine have been uncoupled from the first stage to make a free power turbine.

Figure 20 shows a drawing of the Tyne propulsion module and Figure 21 shows a cross-section through the module. On the left is the cascaded bend in the combustion air intake. This bend is portable and can be hoisted clear to allow the engine change unit to be removed and replaced through the combustion air downtake. The gas turbine change unit which weighs 1½ tons, is supported by cantilevers carried on the power turbine mounting frame. In this engine, unlike the Olympus, the gas turbine change unit includes the power turbine which runs at 14,500 r.p.m. at full power. On the right hand side of the illustrations, the 4-to-1 primary reduction gearbox can be seen. This reduces the Tyne output shaft speed to that required for the primary pinion in the main double reduction gearbox.

The Tyne engine in its naval version, the RM1A, ran very successfully in December 1969 and 220 hours of testing have shown the engine to be basically suitable for a warship cruise engine. Performance has been better than forecast; at a turbine entry temperature of 1150K, the bare engine developed 4,420 b.h.p. compared with the 4,100 b.h.p. forecast. The specific fuel consumption was 0.478 lb/b.h.p./hr. which was better than the 0.494 forecast. The first engine did 80 starts in the first 47 hours running and the second build has done 180 hours running and the minor troubles experienced with the first engine seem to have been eliminated. It is remarkably vibration free. We aim for 1,200 hours of development running and 4,000 hours of endurance testing before the first engine goes to sea in Amazon in 1972. The engine could be developed to give over 5,000 b.h.p. and a better tropical performance if certain modified cooling arrangements were fitted. This would permit an increase in turbine entry temperature but would take a few more years and cost a lot more money.

The need to undertake this development is now under discussion. To the marine engineer, it is a highly desirable development programme; it will keep the design team in being while the engine goes to sea, it will give more power and thus longer life at the critical Fleet cruising speeds and it will give better ship performance in the tropics. If the engine is capable of...
this development, then it is technically desirable to do it since it gives advantages. The question now is, "is it cost effective?".

The Module Concept
Throughout this paper, I have referred to the use of a Module. This appears to be a device to make the lightweight aero engine into a heavy marine engine. In some ways, this is true since we must provide shipbuilders with the sort of machinery with which they are familiar. The module, however, solves many problems and fulfils many functions.

The module is a standard enclosure for the gas turbine change unit and the power turbine. The aim is to provide in the module acoustic, thermal and defensive enclosures, local controls for machinery operation, space and access for maintenance operations, ventilation and fire protection equipment. The module is installed in the ship in two or three parts. The power turbine mounting frame is secured to the hull on the necessary constant position mounting units or on shock mounts and lined up to the gearbox. The acoustic enclosure is then supported on separate mounts and joined to the power turbine frame by a flexible joint. This enclosure is a relatively light structure and can be supported on simple mounts. Then the air intake enclosure and cascaded bend can be fitted and this allows the downtakes and uptakes to be installed. When the ship construction is well advanced and a few months before basin trials, the gas turbine change unit can be supplied and installed in the module. To do this, the cascaded bend in the inlet is removed and the change unit is lowered down the intake. In this way, the delicate and expensive change unit is not delivered before it is required; it is protected from the vibrations and hazards of ship construction and its installation can be used to demonstrate the provision of the engine change facilities required.

The standard module and change unit can be erected and tested in the maker’s factory under clean conditions. The shipbuilder’s installation task is simplified and the layout of the module is standard from ship to ship from Frigate to Destroyer of Cruiser. This simplifies our training task. The MOD specification for this module is the result of considerable thought and incorporates many compromises so that it can be widely used. Figures 18 and 20 show drawings of the modules and Figure 21 shows a photograph of a model of the Tyne module. The enclosure carries acoustic lagging and also provides the ship with protection against blast entering the intake, containment against chemical, biological and nuclear fall-out and it carries within it fire detectors and extinguishers. The module is used for carrying the CO₂ fire extinguisher bottles, the engine local controls and testing equipments, the primary reduction gearbox if required, the lubricating oil tank and terminal points for fuel, oil, air and electric supplies. The module is the concept which is used to achieve the aim of rapid engine changes in service. The Olympus module, complete with down and uptakes will go on test very soon and start what I hope will be a profitable development programme. The module is now being “maintenance evaluated” bij artificers in the Rolls Royce factory.

Maintenance Evaluations
Last year when the development of the Olympus and Tyne engines was proceeding at Rolls Royce, Ansty, a small team of three, Chief Petty Officer Artificers, was established at the firm to make recommendations for the support of the engines in service. Their task is to prepare the maintenance schedules and the job methods for routine servicing; to study the possible modes of failure and devise repair methods; to list the spare parts, tools, etc., required for each task; to criticise the maintainability of the equipment and suggest modifications and to check the technical publications. This has proved a very useful exercise. Twelve modifications in the design have been accepted and incorporated to improve maintenance. All parts which require servicing and maintenance are now readily
accessible and the working space for a man to do the work has been provided in designing the installation.

Conclusions

The Royal Navy, as the memory of World War II recedes, is getting smaller and smaller but this should not be an excuse for any reduction in quality. We still need first-class ships and to give our men the best equipment available but we cannot afford to do this over a wide variety of items, so we must standardise. For the main propulsion systems we must have a standard system for all our ships as far as possible. Standard spare units, standard instruction books and standard methods of operation. We cannot afford variety. We will offer the ship designer a very small and limited range of equipment.

For our Destroyers, the Type 42 Class, the first of Class is Sheffield and for our Frigates, the Type 21 Class, the first of Class is Amazon, we have a standard main propulsion machinery package consisting of the Olympus TM3B, the Tyne RM1A and an excellent simple double reduction David Brown gearbox. The whole package drives through shafting to a Stone's controllable pitch propeller and is controlled by a Hawker Siddeley Dynamics propulsion control system. The design is aimed at high ship availability and low complement.

The ship installation is such that only one gas turbine provides power to the propeller at a time. Each engine can be run up and tested before being connected to the main gearing through synchro-self-shifting clutches which are provided between each turbine and the gearing. The ship is designed to operate worldwide for four years with the minimum support between refits. It is this requirement which determined the use of COGOG rather than CODOG. The cruising gas turbine may require more fuel but it needs fewer men and can be quickly changed. The larger heavier diesel is not so easily accommodated in a warship and not so easily changed when at the end of its overhaul life. The ship designer needs to look at the total fuel and machinery weight and the life cycle costs and this is where the COGOG plant shows to advantage.

Such a lot of design effort, incorporating all the lessons learnt in Exmouth, has gone into our present standard propulsion system that we are reluctant to make a change. Nevertheless, technology does not stand still. The present plant has disadvantages in high fuel consumption, high first cost and requires a large area of intakes and exhaust trunks. There are three ways ahead possible. We can either replace our Olympus/Tyne combination with another pair of more efficient engines, we can use a pair of identical engines which will give us good part load performance and can both be run to give full power or we can develop a pair of main engines with a reasonably good part load performance which will propel a warship at the full speed required and can be used individually driving on one or both shafts for cruising.

The technical choice facing the Navy now is either to continue with the development of marine versions of the aircraft engines; in doing this we must accept the rather steep specific fuel consumption curve or to develop an industrial type of gas turbine with a complex cycle to give good part load performance.

The aircraft gas turbine allows us to take a fully developed aero engine which is in airline service and modify it for marine use. This is practicable today when engines designed for subsonic aircraft are being developed. If, however, the Concord development and multi-role combat aircraft development are brought to successful conclusions, future aircraft engines may be designed for supersonic operation. These engines will not be so easily modified for marine propulsion.

With the development of aero engines, usually greater efficiency is achieved by increasing the turbine entry temperature and this could exceed 1,350K in a marine version of the RB211. At present we have TETs of 1,150K at sea in Exmouth.

In 1972 we shall go up to ver 1,200K with the Olympus TM3B. We cannot be certain we shall obtain long lives at sea with h.p. turbines operating above 1,200K. There must be some doubt about operating over 1,300K.

The alternative is to achieve high part load efficiency by intercooling and heat exchanging. This means going away from the simple cycle to adopt the complex cycle comparable to
RM60 and EL60A previously described. The Royal Navy has not had very happy experience of these engines. The former only ran for a short period at sea and the latter was never installed in a ship. Nevertheless, the complex cycle gas turbine may have a future, as better materials are developed or perhaps it will be adapted to suit a reactor heat source. Of one thing I am sure; the Navy will not be able to afford to develop a complex cycle gas turbine for its own use. If any other industry is intending to do so we would be interested.

Since we are currently committed to developing marine versions of aero engines, we have surveyed the engines available today and have chosen two for detailed study; the Rolls Royce RB211 and the Rolls Royce/Snecma M45H. The RB211 could provide a marine engine to replace the Olympus TM3B giving the same power and the M45H could provide a marine engine giving about 50 per cent more power than the Tyne. Both these engines offer much improved specific fuel consumptions and use much less air than the ones they replace, they are therefore attractive to the ship designer.

Today, Frigates of the Leander Class have steam turbines developing 15,000 s.h.p. on each of two shafts, Frigates of the Exmouth Class have a similar set developing 15,000 s.h.p. on one shaft. In the past, 15,000 s.h.p. has been adopted as a standard propulsion power. The size of Frigates has tended to increase but this tendency is usually checked and reversed as the grip of economy applies its pressure. The requirement for a standard propulsion plant of 15,000 s.h.p. is foreseen and this could be reliably and economically achieved by a pair of 7,500 h.p. engines.

Since submarines go faster and cruising speeds are tending to rise, an alternative is to use a 25,000 h.p. main engine and a 7,500 h.p. cruise engine in much the same way as the Tyne/Olympus package will be used in the ships now building. This will give a high speed warship but one with an expensive propulsion system.

A further alternative is to use one 25,000 h.p. engine on each shaft, each engine capable of driving both shafts either by a turbo-electric drive or by a mechanical gearbox.

Whichever way is chosen, we seem to need a 25,000 h.p. engine and a 7,500 h.p. engine to replace the two in our present standard range. A preliminary study of the RB211 shows a possible way of marinating this engine (Figure 22). Here, the aero engine is modified like the Olympus to give a separate power turbine. This engine could be designed to replace the Olympus in its standard module quite quickly. The two stage power turbine shown will drive at the same speed as the Olympus. This design is very attractive since the reduction in specific fuel consumption and the reduction in the air required for full power, permit a reduction in the total machinery and fuel weight. Figure 23 shows a comparison between the Olympus/Tyne propulsion system endurance with the characteristic knee and a possible endurance with a RB211 installation propelling on one RB211 up to 25 knots. The fuel carried gives the same endurance on both systems at 17 knots— below this speed the Tyne is superior but above it the greater endurance is obtained on the RB 211.

The M45H aero engine is now under test at the Bristol Engine Division of Rolls-Royce. It is designed and manufactured bij Rolls-Royce and the French firm Snecma in co-operation. It will be used to power an airliner being designed and made in Germany. This engine is therefore a European venture an I believe we should see it as a European marine engine. Britain is now part of Europe again and the Navy like the rest of the nation must seek European co-operation.

Here is a technological field where we can offer Europe our help and thus gain an alliance. To delay would encourage the growth of a competitor. Also with France and Germany we are bound by an alliance, Western European Union; we need not give them any commercial advantage but we could all have better warships.

Our peacetime role is to deter aggression. In this we are supported by allies. To deter the enemy we must have reliable propulsion systems. The Royal Navy is technologically in advance of the rest of the world in its main ship propulsion system. It is a simple propulsion system since there is nothing more useless than a warship that will not move.

Acknowledgements

This paper is published with the permission of the Ministry of Defence but all the opinions expressed herein are my own. I wish to acknowledge the assistance in the preparation of this paper given by members of the Gas Turbine Section of the Ship Department of the Ministry of Defence, of the Naval Marine Wing at the National Gas Turbine Establishment, Messrs. Rolls-Royce Ltd, and Messrs. Y. ARD Ltd.

The Navy is fortunate in having a happy co-operation with its main machinery contractors and only in co-operation so each understands the other's needs will we make progress.
References


APPENDIX

Propulsion Gas Turbines now in service or on order

<table>
<thead>
<tr>
<th>Engine</th>
<th>G6</th>
<th>Olympus</th>
<th>TM3B</th>
<th>Proteus 52M/550</th>
<th>Tyne RM1A</th>
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<tr>
<td>Maker</td>
<td>GEC Ltd</td>
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<td>Rolls-Royce Ltd</td>
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<tr>
<td>Max rating at 15 °C (59 °F)</td>
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<td></td>
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<td></td>
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<td>inlet air temp, hp</td>
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<td>24,000</td>
<td>27,200</td>
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<td>4,250</td>
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<tr>
<td>Long life (in excess of 2,000 hr in all cases), hp</td>
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<td>19,000</td>
<td>21,000</td>
<td>3,600</td>
<td>4,250</td>
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<tr>
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<td>Main propulsion boost</td>
<td>Main propulsion boost</td>
<td>Main propulsion boost</td>
<td>Two-axial propulsion boost</td>
<td>Main propulsion boost</td>
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<td>39</td>
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<td>None</td>
<td>4</td>
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<tr>
<td>Cycle (see Note 1)</td>
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<td>2SC/LP</td>
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<td>882</td>
<td>917</td>
<td>1,593</td>
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<tr>
<td>°F 1,460</td>
<td>1,619</td>
<td>1,683</td>
<td>1,190</td>
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<td>Kin 1,066</td>
<td>1,155</td>
<td>1,930</td>
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<td>Over-all pressure ratio</td>
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<td>228</td>
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<tr>
<td>Thermal efficiency at max power, %</td>
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<td>at 50 % power, %</td>
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<td>21.2</td>
<td>23.2</td>
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<td>One-axial with centrifugal final stages</td>
<td>Two-axial</td>
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<tr>
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<td>LP-5</td>
<td>LP-5</td>
<td>13</td>
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</tr>
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<td></td>
<td>HP-7</td>
<td>HP-7</td>
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<td>HP-9</td>
</tr>
<tr>
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<td>Three-axial</td>
<td>Three-axial</td>
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</tr>
<tr>
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<td>LP-1</td>
<td>LP-1</td>
<td>CT 2</td>
<td>LP-1</td>
</tr>
<tr>
<td></td>
<td>PT 2</td>
<td>HP-1</td>
<td>HP-1</td>
<td>PT 2</td>
<td>HP-1</td>
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<tr>
<td>rpm/1,000</td>
<td>CT-6.85</td>
<td>LP-6.2</td>
<td>LP-6.35</td>
<td>CT-11.75</td>
<td>LP-14.5</td>
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<td></td>
<td>PT-4.9</td>
<td>HP-7.81</td>
<td>HP-7.95</td>
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This article was presented before the North-East Coast Institution of Engineers and Shipbuilders in Newcastle upon Tyne on Monday 9th November 1970.
ON THE CRITICAL SPEED RANGE OF SHIPS IN RESTRICTED WATERWAYS*)

by J. P. HOOFT**) 

SUMMARY

A description is given of the flow phenomena which appear in a canal when a ship passes at a speed in the critical range. A brief discussion is included concerning the theoretical additional resistance of a ship, travelling at a critical speed due to the bore. This energy absorbing bore occurs throughout the critical range and travels ahead of the vessel. Model resistance tests were performed and indicate that the additional resistance due to the bore substantiate the theoretical findings. Self propulsion tests also demonstrate that a ship can achieve and sustain a speed which is in the critical range if merely provided with sufficient power.

1. Introduction

As a ship passes through a canal three distinct flow situations can occur. These situations are directly dependent on the vessel's speed and are termed sub-critical, critical and super-critical flow.

In some studies concerning the behaviour of ships in restricted waterways it was concluded that navigation of self propelled ships in the critical and super-critical speed range was impossible.

Other observers, however, noticed that ships with sufficient power could sail at speeds which were critical or even super-critical.

Due to this ambiguity, further analysis of the flow phenomena in a restricted waterway was undertaken. In addition to model tests were performed. This report includes the major findings of this investigation.

The phenomena which occur behind an obstacle in a two-dimensional flow in shallow water are described. Then the flow phenomena upstream and downstream of a ship navigating in a canal are discussed in detail. The additional resistance of the ship in the critical speed range has been calculated based on the foregoing considerations.

Model resistance and self-propulsion tests were performed to verify the results of the theoretical calculations.

2. Description of the two-dimensional flow

Behind an obstacle in a two-dimensional flow, a stationary train of gravity waves will be created in which the wave velocity c relative to the water equals the velocity v of the flow. The velocity U with which energy propagates with respect to the water is given by:

\[ U = \frac{1}{2} c \left( 1 + \frac{2m}{\sinh 2mh} \right) = n c \]  

(2-1)

for small amplitude waves m is the wave number and h the depth of the water.

The wave energy will then be propagated downstream with a velocity v-U relative to the obstacle.

\[ v - U = (1 - n) v \]  

(2-2)

The relation between the wave length \( \lambda \) and the flow velocity \( v \) is expressed by:

\[ v = c = \sqrt{\frac{g\lambda}{2\pi}} \tanh \frac{2\pi h}{\lambda} = \sqrt{\frac{g}{m}} \tanh mh. \]  

(2-3)

When the water depth becomes large relative to the wave length, then the wave energy travels downstream with a velocity \( \frac{1}{2} v \) relative to the obstacle. When the water depth becomes small relative to the wave length, then the velocity of the wave energy relative to the obstacle becomes zero. In this case the obstacle has no wave resistance because no energy for the generation of waves is needed.

This simplified description given e.g. by Lighthill (4) has to be corrected when the flow velocity is nearly critical, that is to say, when the Froude number \( v/\sqrt{gh} \) approaches unity.

In the sub-and super-critical speed range all the water that reaches the obstacle, can pass it. This means that the flow is of a steady-state nature. The flow phenomena around the obstacle can then be calculated by means of the steady-state Bernoulli equation in addition to the equation of continuity. As the flow passes the obstacle the water speed is increased and the pressure decreased compared with these quantities far ahead of the obstacle.

On the other hand there exists a speed range at which neither the continuity equation nor Bernoulli’s equation can be satisfied. The flow is no longer steady and therefore one has to use the equations for unsteady flow to obtain a description of the phenomena in this critical speed range.

It is found that water piles up in front of the obstacle. The raised water level forms a bore which travels upstream.

At the rear of the obstacle there is a deficiency of water since continuity is not maintained. Therefore a simple wave of expansion exists just aft of the obstacle.

In a bore, energy is dissipated by friction. In a simple wave no energy is lost. Therefore even in a two-dimensional case there exists a critical speed range instead of one critical speed, because there is a speed range covering \( \sqrt{gh} \) where the resistance of the obstacle is increased by the amount of the resistance due to the energy loss in the bore. Only in the super-critical speed range will the wave resistance become zero.

It should be noted that the range of critical speeds tends to zero when the width of the canal tends to infinity.

3. Flow phenomena upstream

We now concern ourselves with the case of a ship sailing with a constant velocity in a rectangular canal with constant cross-section over the whole length. To describe the flow beside and in front of the ship only the average values are used. Therefore the following assumptions are made:

a. The flow velocity is constant over the whole section of the canal.

b. The water level is constant over the width of the canal.

\[ V_1 \quad h_1 \quad V_3 \quad h_3 \]

Fig. 2. Ship navigating in a canal in the sub- or super-critical speed range.


**) Netherlands Ship Model Basin, Wageningen.
The relation between the speeds and water depths in the cross-section 1 and 2 is obtained from the method used by Kreitner (5). The equation of continuity is:

\[ \frac{v_o h_o w}{v_d (h_2 w - A)} = (3-1) \]

in which \( A \) is the underwater cross-section of the ship and \( w \) the width of the canal. The Bernoulli equation for steady-state flow is:

\[ \frac{1}{2g} v^2_0 + h_0 = \frac{1}{2g} v^2_2 + h_2 \]  

(3-2)

The speed range in which the equations (3-1) and (3-2) give no real solutions, is determined by the maximum quantity of water that can pass the ship:

\[ \frac{d}{d v_2} \left( \frac{h_2 w - A}{v_2} \right) = 0 \]  

(3-3)

At the beginning of the critical speed range the wave velocity at some cross-section alongside the ship becomes critical because the velocity of the water at this place equals \( \sqrt{gh} \), in which \( h \) is the water depth at that cross-section. At this moment the wave front is perpendicular to the direction of the ship and therefore the front of the raised water level will also be perpendicular to the longitudinal axis of the canal. Now the raised water level forms a bore which travels upstream. The mathematical description of a bore is given e.g. by Stoker (6).

![Fig. 3. A bore at rest](image)

Relative to the bore, the equation of continuity is:

\[ \frac{U_o h_o}{U_1 h_1} = 1 \]  

(3-4)

The momentum law is:

\[ U_o^2 h_o + \frac{1}{2} gh_o^2 = U_1^2 h_1 + \frac{1}{2} gh_1^2 \]  

(3-5)

Combining equations (3-4) and (3-5), one obtains:

\[ U_o = \sqrt{\frac{h_1 + h_o}{2 h_1}} \]  

(3-6)

In the case of a ship sailing with a speed in the critical range (see Figure 4), the bore velocity relative to the water is:

\[ v_a = \sqrt{gh_o} \sqrt{\frac{h_1 + h_o}{2 h_o}} = \sqrt{gh_1} \]  

(3-7)

The flow just in front of the ship (see cross-section 1 in Figure 4) will be stationary with respect to the ship. This means that the velocity is just large enough to let the whole mass quantity \( \rho v_1 h_1 w \) pass the ship. In the critical speed range, therefore, equation (3-8) holds:

\[ \frac{d}{d v_1} \left( v_1 h_1 w \right) = 0 \]  

(3-8)

![Fig. 4. Ship navigating in a canal in the critical speed range](image)

The relation between the speeds and the depths of the cross-sections (1) and (2) can be found with the same method as is used in the sub-critical speed range. The equation of continuity now is:

\[ \frac{v_1 h_1 w}{v_2 (h_2 w - A)} = (3-9) \]

The Bernoulli equation for stationary flows is:

\[ \frac{1}{2} v^2_1 + gh_1 = \frac{1}{2} v^2_2 + gh_2 \]  

(3-10)

One now has five equations ((3-4), (3-5), (3-9) and (3-10)) to find the solutions for the five unknown quantities, which are the water depths \( h_1 \) and \( h_2 \) and the velocities \( v_1 \) and \( v_2 \) of the flow and \( v_a \) of the bore.

In Figure 5 the velocities \( v_1 \) just in front of the ship and \( v_2 \) alongside the ship are given as a function of the Froude number \( \frac{v_0}{\sqrt{gh_o}} \).

![Fig. 5. Flow velocities in the critical speed range](image)

In Figure 6 the water depths are given corresponding with Figure 5.

The two examples in Figures 5 and 6 are those for which Helm (7) has given the resistance in the critical speed range.

4. Flow phenomena downstream

4.1. Sub- and super-critical speed range.

In this speed range the average flow phenomena behind the ship are determined by the frictional resistance of the ship. The frictional resistance in this case means the total resistance minus the wave resistance.

The wave resistance is found when the wave energy that travels downstream, is divided by the ship speed.

If only the average values for the velocities and water depths are taken, then one finds from Figure 2 with the aid of the law of momentum:
\[ (p v_o^2 h_o w + \frac{1}{2} \rho gh_o^2 w) = R_1 \]
\[ (p v_3^2 h_3 w + \frac{1}{2} \rho gh_3^2 w) = R_1 \]
in which \( R_1 \) is the frictional resistance of the ship.

In practice, however, the resistance \( R_1 \) is obtained from resistance tests. Therefore \( h_3 \) and \( v_3 \) are known from the equation (4-1) and the equation of continuity (4-2):
\[ v_o h_o = v_3 h_3 \]  
(4-2)

With the knowledge that the energy at the rear of the ship is less than in front of it, it is proved by Benjamin and Lighthill (8) that behind the ship, the average water depth is less and the average water speed is larger, than in front of the ship.

\[ \begin{align*}
  v_o &< v_3 \\
  h_o &> h_3
\end{align*} \]
(4-3)

**Fig. 7. A simple wave described by the method of characteristics**

On the "right" characteristics \( C_2 \), which originate in the region 1 where \( u = 0 \) and the depth is \( h_o \) the constant is \(-2 \sqrt{gh_o}\). Because a characteristic of this type passes through each point of region 2, we have here:
\[ \sqrt{gh} = \frac{1}{2} u + \sqrt{gh_o} \]
(4-6)

The "left" characteristics \( C_1 \), which are straight, (see, e.g., Stoker (6)) now are determined by:
\[ \frac{dx}{dt} = u + \sqrt{gh} = \frac{3}{2} u + \sqrt{gh_o} \]
(4-7)

Constantine now assumed that the acceleration of the ship at \( t = 0 \) is infinite. Then at each moment the velocity \(-U_3\) in the area 3 will be constant, so that the cross-section 4 moves with a velocity:
\[ \frac{x}{t} = \frac{3}{2} u_3 + \sqrt{gh_o} \]
(4-8)

At each moment equation (4-6) remains valid for the cross-section 4. Combining the equations (4-6) and (4-8) gives:
\[ \sqrt{gh_o} = \frac{1}{3} \left( \frac{x}{t} + 2 \sqrt{gh_o} \right) \]
(4-9)

Now it is possible to obtain a solution for the water depth \( h_3 \) and the velocity \( v_3 \) by combining the equations (4-8), (4-9) and the equation of continuity (4-10):
\[ v_3 h_3 = (v_o - u_3) h_o \]
(4-10)

Constantine made his assumption concerning the acceleration of the ship, because the actual acceleration is very difficult to describe for the general case. The acceleration depends on the propulsive power and the resistance of the ship. The objection against his assumption is that the flow phenomena behind the ship are defined by him without the influence of the friction resistance of the ship. Hence, with the calculated flow phenomena behind the ship one now can determine the ship resistance. Comparing this calculated ship resistance with the measured values will show that the agreement is poor.

When the acceleration is known, then also the form of the characteristics \( C_1 \) given by equation (4-7) is determined.

Analogous with the description at the rear of the ship for the sub-critical speed range it is easier to obtain the frictional resistance by tests. With the knowledge of the resistance it is possible to determine the average flow phenomena behind the ship.

The flow just in front of the ship and just at the rear of it is stationary with respect to the ship. The average values for the water depth and the flow velocity then follows from equation (4-11) which is identical to the equation (4-1) for the subcritical speed range.
\[ (p v_1^2 h_1 w + \frac{1}{2} \rho gh_1^2 w) = R_1 \]
\[ (p v_3^2 h_3 w + \frac{1}{2} \rho gh_3^2 w) = R_1 \]
(4-11)
in which \( R_1 \) is the friction resistance of the ship. Combining the equations (4-10) and (4-11) the unknown velocity \( v_3 \) and water depth \( h_3 \) can be solved.

5. The ship resistance in the critical speed range

From the foregoing it follows that in the critical speed range the ship has a resistance that can be divided in three parts:

a. The frictional resistance.
b. The wave resistance due to the occurrence of gravity waves.
c. The additional resistance due to the energy loss in the bore in front of the ship.

The resistances a) and b) in the critical speed range are of the same type as those in the sub- and super-critical speed ranges. The energy loss in the bore is:

\[
E = \rho g w U_0 \frac{(h_1 - h_0)^3}{4 h_1}
\]  

(5-1)

The extra resistance of the ship due to the bore then is given by the following equation and is represented in Figure 8.

\[
R_3 = \frac{E}{V_0}
\]  

(5-2)

The total resistance measured by Helm (7) is given in Figure 9.

At the end of the sub-critical speed range the ship resistance increases sharply due to the wave resistance, which for instance is explained by Inui (10).

At the beginning of the super-critical speed range the ship will have hardly any wave resistance. As is given in the introduction, the wave resistance for the two-dimensional case will be zero in the super-critical speed range.

In the description given in sections I to III, no discontinuities are found in the wave or frictional resistance. So the sharp fall in the total resistance at the beginning of the super-critical range can only be explained by the fact that here the bore suddenly disappears. The drop in resistance at the end of the critical speed range is of the same order of magnitude on the calculated resistance of the bore represented in Figure 8.

It therefore now is very plausible that formula (5-2) gives a good approximation for the extra resistance due to the bore in the whole critical speed range. The wave- and frictional resistance in the critical speed range then is obtained by subtracting the resistance due to the bore as calculated by formula (5—2) from the measured total resistance. This is done in Figure 10.

The behaviour of the wave and frictional resistance is in agreement with the above given concept of resistances.

If this concept is adapted one may suggest that the frictional resistance of the ship in the critical speed range can be determined from model tests by using the same extrapolation methods as are used in the sub-critical range, because the remaining resistance due to gravity waves and to the bore is only a function of the number of Froude.

Inspecting the resistance calculated by the method of Constantin (3) the following example is taken. When navigating with a speed of 16 km/h in a canal of a depth of 2 m and a width of 41.8 m, Helm has measured for the given ship a total resistance of 9.5 tons, while with the method of Constantin the calculated resistance is 21 tons.

Table 1

<table>
<thead>
<tr>
<th>Principal dimensions of the model</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculars</td>
<td>3.394 m</td>
</tr>
<tr>
<td>Breadth</td>
<td>1.084 m</td>
</tr>
<tr>
<td>Mean draft</td>
<td>0.126 m</td>
</tr>
<tr>
<td>Trim by the stern</td>
<td>0.265 m</td>
</tr>
<tr>
<td>Immersed volume</td>
<td>0.212 m³</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Principal dimensions of propellers</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter D</td>
<td>203.2 mm</td>
</tr>
<tr>
<td>Pitch uniform P</td>
<td>203.2 mm</td>
</tr>
<tr>
<td>Pitch ratio P/D</td>
<td>1.000</td>
</tr>
<tr>
<td>Developed-area ratio</td>
<td>0.703</td>
</tr>
<tr>
<td>Number of blades</td>
<td>4</td>
</tr>
</tbody>
</table>

Fig. 8. Ship resistance due to the energy loss in the bore

Fig. 9. Ship resistance given by Helm from resistance tests

Fig. 10. Ship resistance from Figure 9 reduced with the “bore resistance” from Figure 8

S. en W. — 37e jaargang no. 24 — 1970
Fig. 11. Small scale body plan of model

Nomenclature

\begin{align*}
A &= \text{cross-section of the ship under the water} \\
E &= \text{energy in the bore} \\
g &= \text{acceleration due to gravity} \\
h &= \text{water depth} \\
k &= \text{blockage factor} \\
m &= \frac{2\pi}{\lambda} \\
u &= \text{velocity} \\
v &= \text{velocity relative to the model} \\
w &= \text{width of the canal} \\
R &= \text{resistance} \\
P &= \text{mass density} \\
\lambda &= \text{wave length} \\
Q &= \text{torque in both propellers} \\
n &= \text{revs./min} \\
S &= \text{thrust of both propellers} \\
\eta &= \text{propulsive coefficient} \\
V_m &= \text{speed of model}
\end{align*}
6. Results of self-propulsion tests

Model tests were carried out in the critical range. During the propulsion tests the model was driven only by means of the two propellers. The principle dimensions of the model and the propellers are given in Table I, while a small scale body plan of the model is given in Figure 11.

The basin in which the tests have been carried out was 15.75 m wide and 225 m long.

If the power by which the propellers are driven is large enough, the ship model could achieve a speed in the critical range. In Figure 14 a photograph is given illustrating how the tests were carried out. During the test from which the photograph has been taken the speed of the model then only is a little smaller than the speed of the bore by which the very bore clearly could be demonstrated.

During the propulsion tests measurements were made of the ship’s squat and trim along with the torque, thrust and r.p.m. of the propellers. Also resistance tests were carried out with no appendages to the model. In Figure 12 and 13 the results of these tests have been plotted.

From Figure 12 it follows that the propeller efficiency at a speed in the critical range remains acceptable and even will increase for higher speeds.

From these tests it may be concluded that from a theoretical point of view there is no restriction to sail in a canal or river at a speed within the critical range. From a practical point of view, however, the ship resistance will increase largely at a speed at the lower limit of the critical range. At higher speeds the resistance will increase a little or sometimes even will decrease.

When the width of the canal increases then also the width of the bore increases. By this it will be clear that the amount of energy absorbed by the bore will become infinite when the width of the water is not restricted.

Consequently the ship can not sail at the critical speed \( \sqrt{gh} \) in a waterway with unrestricted width but many in fact surpass this speed by several means.

References

De vraag of men viscositeitsregeling of temperaaturregelning moet toepassen voor stookolie van scheepsmotoren is nog steeds een onderwerp van discussie gebleken. De keuze kan worden gemaakt tussen de drie volgende alternatieven.

1. Temperatuurregelning;
2. temperatuurregelning met registratie of indicatie van de viscositeit;
3. viscositeitsregeling met temperatuurindicatie.

De hierondergenoemde redenen duiden er op dat alternatief 3 de meest aantrekkelijke is.

1. Temperatuurregelning

Temperatuurregelning alleen is vandaag aan de dag niet meer efficiënt daar de moderne branders en dieselmotoren steeds gecompliceerder worden en deze stookolie een constante viscositeit moet hebben om bevredigend te kunnen werken. Indien men Bunker C-olie gebruikt met alleen een temperatuurregelning dan is het praktisch onmogelijk om een constante viscositeit te verkrijgen waar de Bunker C-olie in kwaliteit varieert. Het temperatuurverschil voor dezelfde viscositeit kan tot 30 °C (54 °F) uitlopen voor verschillende Bunker C-kwaliteiten.

Men kan zich indenken wat de juiste viscositeit of de juiste temperatuur voor verschillende kwaliteiten olie kan betekenen indien men weet dat een temperatuurvariatie van slechts 5 °C (9 °F) een lagere efficiëntie van 1-2 pct op uw bedrijf veroorzaakt. Temperatuurregelning alleen is daarom zowel technisch als economisch gezien een slechte oplossing en wordt derhalve alleen gekozen vanwege de lage prijs van de installatie.

2. Temperatuurregelning met registratie of indicatie van de viscositeit

Hier gebruikt men een temperatuurregelcircuits zoals in alternatief 1 en daarbij wordt een viscositeitsmeter gemonteerd alleen voor registratie en indicatie. De viscositeit wordt continu gemeten en de gewenste waarde wordt op de temperatuurregelning met de hand bijgeregeld, zodat een constante viscositeit wordt verkregen. Ook deze methode is niet geheel bevredigend vanwege het feit dat er continu toezicht moet zijn om met de hand te kunnen regelen. Het zwaartepunt ligt in het regelen van de viscositeit, alternatief 3, en dit is de meest gewenste regeling, mits men een betrouwbare transmitter heeft, waarvan de prijs eveneens niet exceptioneel hoog is.

3. Viscositeitsregeling met temperatuurindicatie

De huidige ontwikkeling gaat naar een viscositeitsregeling als zijnde de enigste manier voor een vol-automaatregeling voor de stookolie. Maar voor automatische viscositeitsregeling is een absolute betrouwbare viscositeitsindicator noodzakelijk, waaraan zeer zware eisen worden gesteld speciaal gezien de toepassing van deze transmitter aan boord van schepen.

Het meest gangbare methode is het meten van de drukval over een capillaire buis met een constante flow. Eurocontrol heeft sinds enkele jaren een nieuw type op de markt geïntroduceerd. Deze transmitter is bijzonder geschikt voor het automatisch regelen van de viscositeit. Hieronder geven wij u een nauwkeurige beschrijving van dit systeem, daar dit werkt volgens een geheel ander principe.

De viscositeit wordt gemeten door een torsiekraakt op een ronde schijf — fig. 2 — waarvoor een tweede schijf draait met een constante snelheid. Tussen de 2 schijven is een smalle opening welke instelbaar is. De ronddraaiende schijven worden uitgerust met radiaalsleuven. Door deze sleuven wordt continu nieuwe procesvloeistof tussen de schijven toegevoerd. Een grotere viscositeit veroorzaakt een hogere torsiekraakt op de stationaire schijf. Deze torsiekraak wordt door middel van een krachtbalansysteem in een pneumatisch standaarduitgangssinaal van 3-15 psi omgezet. De transmitter wordt direct in de olieleiding gemonteerd „in-line”. Geen dode tijd door shunting of een by-pass van de transmitter. De gemeten waarde is representatief voor de viscositeit van de hoofdstroom.

(Dit is niet het geval indien in een by-pass gemeten wordt.)

Max. druk 100 ato (1400 psi)
Max. temperatuur 180 °C (356 °F)

2. Door de radiaalsleuven wordt continu nieuwe procesvloeistof toegevoerd tussen de meetschijven — snelle response op variaties.

3. De transmitter is ongevoelig voor vaste delen in het medium waar er geen capillaire buizen of andere nauwe doorgangen aanwezig zijn welke verstoppingen kunnen veroorzaken.

4. Hoge meetnauwkeurigheid met standaard uitgangssignaal 3-15 psi welke recht evenredig is met de viscositeit.

Verder kan worden genoemd

De transducer werkt volgens het krachtbalansysteem en heeft bewezen absoluut ongevoelig te zijn voor trillingen. De calibratie kan makkelijk worden gecontroleerd zonder gereedschappen, met de hulp van een calibratiegewicht, dat met de transmitter wordt meegeleverd. Het meetbereik kan gemakkelijk worden veranderd.
Goedgekeurd voor schepen

Na een grondige test is de VISC 21P nu goedgekeurd voor schepen door Det norske Veritas en Lloyds Register of Shipping. Ongeveer 50 installaties, hoofdzakelijk aan boord van schepen (die nu ongeveer 3 jaar hebben gewerkt), en de testresultaten van eerdergenoemde instanties hebben bewezen dat de transmitter ook onder moeilijke omstandigheden welke zich aan boord kunnen voordoen, tot de volle tevredenheid werkt.

Installatievoorbeeld van de VISC 21P

De installatie is zeer eenvoudig en bestaat uit 3 hoofdonderdelen, die het monteren gemakkelijk en goedkoop maken: 1) viscositeitsmeter, 2) pneumatische regelaar met reducerstation en aanwijzingsoude; en 3) regelventiel met pneumatische aandrijving.

Belangrijk is dat de viscositeitsmeter voor het filter gemonteerd kan worden daar de VISC 21P ongevoelig is voor vaste delen. De na-ijzing is zeer kort en met een reactietijd van minder dan 1 seconde in de transmitter wordt een beveiligende regeling verkregen.

Indien de hoofdmachine gestart wordt, wordt het reducerstation op hand gezet en de temperatuur wordt met de hand in de gewenste tijd omhoog gebracht. Wanneer een temperatuur van 60°-70 °C (140°-158 °F) is bereikt, begint de viscositeitsmeter automatisch te werken en kan men omschakelen van dieselmotoren naar zware olie hetzij geleidelijk of direct. De temperatuur wordt dan omhoog gebracht totdat de gewenste viscositeit bereikt is.


Conclusie

Met een betrouwbare viscositeitsindicator kan een goede en volledige automatische viscositeitsregeling worden verkregen, ook aan boord van schepen. Het beschreven eenvoudige meetprincipe van de transmitter VISC 21P, de robuuste constructie en de goedkeuringsprocedure van Norske Veritas en Lloyds Register of Shipping waarborgen VISC 21P, de robuuste constructie en de goedkeuringen van deze eenvoudige meetprincipe van de transmitter.

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Het overige gedeelte van de voordrach­ten was van technische of wetenschappe­lijke aard. In het eerstgenoemde ver­band is het misschien interessant om in het kort de voordracht van R. Kaufman en A. J. Rothstein, beide van Deepsea Ventures, Inc., aan te halen. De titel was: „Recent developments in deep ocean mining“. In de voordracht werd afge­wogen welke eisen een efficiënte oceaan­mijnbouw stelt en hoever de huidige ken­nis thans strekt op de vijf meest kriti­sche gebieden van de oceaanmijnbouw: lokaliseren van mijnen, het aan het op­pervlak brengen van erts, het winnen van metalen en andere mineralen, het beantwoorden aan de op de markt ge­stelde eisen, en exclusieve mijnbouwrech­ten. De sprekers gaven een uiteenzet­ting van de ervaring van de firma met het exploratieschip Prospector en verwe­zen voor een volledig overzicht van de mijnbouwtechnologie naar de werkzaam­heden van anderen.

Verschillende plaatsen op de bodem van zowel de Atlantische als de Stille Oceaan zijn rijk aan mangaanerts, dat voorkomt in zgn. nodulen, met afmetingen varië­rend van 12 tot 200 mm. Concentraties van 9 tot 27 kg/m³ werden met behulp van televisiecamera’s waargenomen. De rijk­ste van zgn. nodulen zijn te vinden in de Stille Oceaan op een diepte van 5000 tot 9000 meter. Proeven in het centrale research laboratorium van Tenneco Chemicals (de moederonderneming van Deepsea Ventures) hebben uitgewezen dat m.b.v. hy­drometallurgische processen mangaan­nikkel, kobalt en koper uit de nodulen met zuiverheden van 98 pet en hoger kunnen worden gewonnen. De rijkste no­dulen vindplaatsen die men heeft aange­troffen, geven de volgende opbrengst: Mn: 25-30 pet, Cu: 1.0-1.6 pet; Ni: 1.0-1.8 pet; Co: 0.2-1.5 pet.

De aan boord van schepen kan worden ge­installde Tenslotte is in dit verband nog een woord over de expositie zeker op zijn plaats. De jaarlijkse MTS expositie heeft niet altijd een even groot aantal exposanten geteld. Dit jaar was de belangstelling van de industrie, met 114 exposanten, toch wel ruim te noemen. Of deze belang­stelling om te exponeren verband hield met de nabijheid van de federale over­heid (vorig jaar Miami) valt moeilijk uit te maken. Het gaf de argeloze bezoeker als indruk dat er duidelijke verwachtingen zijn in deze tak van industrie.

De op handen zijnde oprichting van een National Oceanic and Atmospheric Ad­ministration (NOAA) dat binnen het De­partement of Commerce zal opereren, is voor de industrie uiteraard van grote be­tekenis en zal mogelijk de oceanologie op een hoger prioriteitsplan brengen.

De scheepsbouwcommissie schrijft dit in een brief aan de beleidscommissie van Verolme. De verwachting is dat een dergelijk gesprek begin december zal kunnen plaatsvinden.

Voor deze bestudering zal een periode van minstens veertien dagen worden uitgetrok­ken, aldus de scheepsbouwcommissie in de brief, die een reactie is op een telegram van de beleidscommissie aan de commissie-Win­semius. In dat telegram werd ervoor gewaarschuwd dat de beleidscommissie zich tegen een even­tuale fusie tussen Verolme en Rijn-Schelde zal verzetten zolang zij de gevolgen van een dergelijk samengaan niet kan overzien. Ver­der wilde de beleidscommissie — voor de scheepsbouwcommissie enige besissing zou nemen — volledig worden ingelicht over de resultaten van de onderzoekingen door de commissie-Winsemius en over de eventuele maatregelen, die de arbeidsomstandigheden kunnen beïnvloeden.
De volgende heren zijn voor het Gewoon Lidmaatschap voorgedragen aan de Ballotage-Commissie:

J. J. VAN BEEK, Bedrijfsleider N.V. Scheepswaar "Voorwaarts", Kerkstraat 147, Hoogezaan.
Voorgesteld door A. J. Kraaijenbrink.

Voorgesteld door A. J. Kraaijenbrink.

Ir. J. VAN DAM, Nat.k. r., Wetenschappelijke Hoofдобtambenaar A Koninklijke Marine, Javastraat 61, Den Helder.
Voorgesteld door prof. ir. J. H. Krietemeijer.

T. VAN DOLDER, Surveyor to Lloyd's Register of Shipping, Corydastraat 21, Rotterdam-22.
Voorgesteld door J. G. F. Warris.

H. B. DRAPER, Technisch adviseur bij Radio-Holland N.V., Aukemastraat 1, Roden (Dr.).
Voorgesteld door A. J. Kraaijenbrink.

De president van de rechtbank, de heer J. Th. Janssen, wordt benoemd tot commissaris.

Achtschepen economisch

Voor het Marine Institute of Engineers heeft een voordoorvoerder van het Vickers-Concern verklaard dat aachtschepen binnen enkele jaren een economisch alternatief zijn. De totaalkosten voor zoveel „conventionele” container-vaartuigen volgens het rapport van de RDM rond de world trade bedragen ca. £ 65 miljoen. Een aachtsvaartuig dat Vickers voor ogen staat heeft een capaciteit van 1800 containers en is zeer snel. Zes van zulke eenheden kunnen hetzelfde presteren als zeven conventionele containerschepen en die kosten vandaag £ 76 miljoen.

Technische Hogeschool Delft

Geslaagd voor het doctoraal examen voor natuurkundig ingenieur:

J. H. Breeman, Den Haag (met if); W. K. J. Bremer, Rijsenburg; N. Drost, Delft; A. J. Emons, Rotterdam; E. Gerretsen, Delft; P. M. Gerson, Delft; S. Gomboert, Rotterdam; R. Haarsma, Rotterdam; H. van Halum, Haarlem; A. C. Haye, Rijsenburg (Z-H); C. A. van 't Hof, Delft (met if); L. A. M. Jansen, Dordrecht; M. van Kampen, Vlaardingen; J. C. M. de Keijser, Bodegraven; H. J. P. van Kleef, Vlaardingen; J. J. K. Loo, Beverwijk; A. A. A. Los, Groningen; W. C. Meese, Zeist; L. Nijs, Delft; A. J. Ottjes, Winschoten; A. M. Pagie, Delft; C. J. B. J. van der Pols, Koudkerk aan de Rijn (met if); E. van der Scharen, Voorburg; Z. G. Scargo, 's-Gravenhage; E. van der Schee, Vlaardingen; G. B. T. Tan, Delft; J. Vriend, Delft.

Geslaagd voor het doctoraal examen voor elektrotechnisch ingenieur:


Geslaagd voor het doktoraal examen voor atmosferisch ingenieur:


Dart Europe vertrekst

Binnen enkele weken zal de nieuwe mari- tieme dienst van de Dart Containerline Company Ltd. op de Noordafrikkane oostkant van het vasteland aan het werk gaan. De eerste containerschepen komen uit Schotland en varen naar Rotterdam.

Tua wordt overgedragen

Het containerschip Tua, dat onlangs een proeftocht maakte, is door de werf Vuyk en Zn. overgedragen aan de opdrachtgevers Sea Container Chartering Ltd. te Londen. Het schip heeft een lengte over alles van 85,3 m en een breedte van 13,7 m. De brutolading is 7493. De breek- tonnage is 1578. De Tua is een schip van een serie van dertien waarvan er zeven bij Vuyk en zes bij de Zaanlandsche Scheepvaartmij gebouwd worden. Voor Vuyk zal de Tua een vloot van circa 90 schepen doen. De schepen zullen worden overgedragen aan de Engelse opdrachtgevers.

Tewaterlatingen

Op 11 november werd de sleeppopperziger Hurber River, die in opdracht van de Ballast-Nedam Groep N.V. behorende Anglo Dutch Dredging Company Ltd. werd gebouwd door de werf Gusto van IHC Holland, te Druten, voor de Nederlandse Dienst Water en Sea dit schip overgedragen aan de Britse opdrachtgevers. Lady Gillian Tomkins, echtgenote van de Britse ambassadeur in Nederland, Sir Ed ward Tomkins, verrichtte de doopplechtigheid. De tewaterlating die veel publiekelijke belangstelling trok, vond plaats om half vier. De Hurber River, die een lengte heeft van ruim 130 m over alles, wordt voortgestuwd door twee motoren van 8000 pk. Het schip, dat twee gedeelde laadruimten heeft, kan een maximale lading van 14,500 ton innemen. De oorspronkelijke bedoeling om het schip in januari zijn eerste proefvaart te laten maken, moest door omstandigheden vanwege de bellen nog vertragen worden. De eerste werkzaamheden van het schip staan nog niets vast. Aangenomen wordt echter dat het schip eerst een bezoek aan de Engelse opdrachtgevers zal brengen en zich dan naar Brazylië zal begeven voor de eerste baggerwerkzaamheden.

Bij de N.V. Scheepswerf Gebr. Coops te Hoogezand is medio nov. het motorkorvet Halcience te water gelaten, dat wordt gebouwd voor Engelse opdrachtgevers. De Halcience (bouwnummer 255) behoort tot het gladeltype en heeft een draagvermogen van circa 640 ton. De voornaamste afmetingen bedragen: lengte o.a. 47,75 m, breedte tussen 11,44 m, boegbreedte 8,70 m en een hoeltje van 3,46 m. In de machinekamer wordt een 8-cilinder kotseindmotor opgesteld van 320 pk, type TS8, 100 omw. per minuut, reductie 10 : 3. De bediening hiervan kan vanaf de brug geschieden door toepassing van een uitgebreide automatisering. De bouw het gesloten, dat onder toezicht van Lloyd's Register of Shipping en het Brits Ministry of Transport, vaarbeeld: home trade.
De Hertogin van Kent heeft deze dag de kiel van een schip van Swan Hunter in Wallsend ontving. Het schip draagt de naam "Amanda" en is het eerste van vier schepen die op de te water gelaten, dat wordt gebouwd in Haltlapa hydraulische lieren. De verdere uitrusting bestaat o.m. uit: stalen luiken op het bovendek, een holte tot tussendek 3,55 m en een diepgang van ca. 3,50 meter. De kiel van dit schip — dat een nog modernere uitrusting aan boord wordt gesteld, die door toepassing van een uitgebreide alarmerings- en signaleringsysteem van de brug af kan worden bediend. Dit laatste een technische uitrusting aan boord wordt zeer modern.

De bouw van het schip, dat speciaal wordt ingericht voor het vervoer van containers, gescheept onder toezicht van de Nederlandse rederij v/h H. H. Bodewes te Millingen a.d. Rijn (die ook de bouw van de dito schepen verzorgt). De schepen zijn tevens geschikt voor het vervoer van trailers en containers van 127 m lengte, 2400 m3 diepgang en 30 meter breedte, met een hijsvermogen van drie tot vier containerstappen. Zij zullen worden opgeleverd in de zomer van 1979.

De bouw is geheel en al door Nederlandse werf en de tekeningen werden aangeleverd door de Germanischer Lloyd en de seebouwtechnische uitrusting aan boord is ingericht voor het vervoer van trailers en containers van 127 m lengte, 2400 m3 diepgang en 30 meter breedte, met een hijsvermogen van drie tot vier containerstappen. Zij zullen worden opgeleverd in de zomer van 1979.

De bouw opdracht van dit schip werd in november van het vorige jaar verkregen via Conopish aan de Dutch Lloyd; de bouw is geheel en al door Nederlandse werf en de tekeningen werden aangeleverd door de Germanischer Lloyd en de seebouwtechnische uitrusting aan boord is ingericht voor het vervoer van trailers en containers van 127 m lengte, 2400 m3 diepgang en 30 meter breedte, met een hijsvermogen van drie tot vier containerstappen. Zij zullen worden opgeleverd in de zomer van 1979.


scheepsverf in Nagasaki, Japan. De lever- 
tijd is eind 1973, begin 1974. 
Chemit verhuisd in de laatste jaren 18 nieuwe 
tankers besteld, waarvan er acht alge- 
verd zijn. Twee tankers van 250.000 ton 
zullen gebouwd worden door Verolme. 
Voorts heeft Shell een order voor de 
tankers van 70.000 ton in aanbouw in 
Sparrow Point (V.S.). 
Shill International Marine heeft in Frank- 
rijk drie tankschepen voor het vervoer van 
olie gas besteld. Twee van deze sche-
pzen zullen gebouwd worden door Construc-
tions Navales et Industrielles de la Médi-
terranée in Sète, de derde door de werf 
Chantiers de L'Atlantique in St. Nazaire. De 
schepen zullen een laad capaciteit van 
75.000 kubieke meter, wat ongeveer 
degelijke grootte is als een tanker met een 
laadcapaciteit van 87.000 kronen.
Eender dit jaar bestelde Shell drie van de 
dezelfde schepen in Frankrijk. De zes 
schepen zouden de grootste vloeibaar-gas-
schepen ter wereld zijn. De schepen, die op de 
La Seyne gebouwd worden, zullen tank- 
hebben, die zijn gebaseerd op het nieuwe 
ontwerp van de Franse Gaz Transport. 

Vorksehipe 
Het m.s. Fes, groot 498 brt gebouwd in 1962, 
iederdom van G. Kuur te Groningen is ver-
kaard in de handen van de Shell Reynolds 
Group in New York, waar de schip wordt 
gebouwd in 1957 van S.A. Cockerill-Ougree 
in Vlissingen. De directie ontvangen van het 
ministerie van De-

Samenwerking sleepdiensten 
De ze Rotterdamse havensleepdiensten, 
waar de samenwerken in de Sleepvaartcentrale, 
hebben ver witte werklipe plannen om een nieuw 
maatschappij te vormen. Deze nieuwe orga-

necisatievorm zal naar de gezamenlijke 
sleepdiensten meedelen, door gezamenlijke 
bedrijfsvoering sterker en groter te maken dan 
het huidige systeem. Deze verandering zal, aldus de gezamen-

lleden zijn eveneens ingelicht. 

HAL wil grond winnen op zee 
De raden van bestuur van de Holland Ame-
rikalijn in Rotterdam en Bos Kals West-
minster Dredging Group N.V. te Sliedrecht, 
hebben onlangs meegedeeld dat de onder-
handelingen tussen beide maatschappen 
over een 50 pct. deelneming van de Holland 
Amerika Lijn in Westminster Gravels Ltd. 
in Southampton, een deel van de Bow 
Westminster Dredging Group in de nieuwe 
bedrijfsvoeringsmaatschappij zal worden 
on-

Milioenen opdracht voor Bronswerk, 
afdeling scheepsinstallaties 
De opdrachten voor de Koninklijke Maats-
chappij De Schelde te Vlissingen bestelde 
over een 50 pct. deelneming van de Holland 
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Geleide-Wapen-Fregatten voor de Konink-
lijke Marine heeft de afdeling Scheepsinstal-
latie van het Koninklijke politieke op-
dracht ontvangen van het ministerie van De-

fensie (Marine). 

Deze opdracht omvat het leveren en installe-
nen van complexe luchtbegalveelbedingen, meca-

nische-ventilatie- en proviantinstallatie. Te-

vens worden de installaties voor een speciale koelinstelling in schokvaste uitvoe-

ring voor het koelen van water dat nodig is voor het koudwatersysteem. 

Deze technische stukken worden geleverd door schroefcompressoren. 

Met deze opdracht, welke in de komende 
jaren zal worden uitgevoerd, heeft de afdel-
ing Scheepsinstallaties van Bronswerk een 
belangrijke aanvulling op de orderporte-

feuille.

Lloyd's Register class for more large L.N.G. ships with membrane tanks 
Another order for three large liquefied na-
tural gas carriers to Lloyd's Register class 
has been placed by Shell International Ma-
limited. Like the previous Shell order 
for three L.N.G. ships to the Society's class, 
reported in late April, the new vessels will 
have a capacity of 75,000 cu.m each and 
will utilise membrane cargo tanks 
instead of the conventional independent type of 
atank. At 232 m. length X 34.8 m. breadth 
X 20.7 m. depth, they will be about the 
same size as 87,000 d.w.t. tankers, making 
them — along with the previous three ships 
— the world's largest L.N.G. carriers 
with membrane tanks. Two ships will be built at 
Compagnie des Chantiers de l'Atlantique 
L. Seyne, and the third at 
Chantiers De l'Atlantique, St. Nazaire.

The ships that are building at La Seyne will 
have tanks based on a new design by Gaz 
Transport of France. Instead of adopting 
corrugated construction as in the previous 
Shell order, the tanks will be constructed 
from flat panels. The tank panels are to be 
folded inwards and edge welded so that the 
tanks can be kept under pressure 
without external tank supports. The 
structure can be accommodated by flexing of 
the edges about the folds, and automatic welding 
equipment is to be used wherever possible. Made of M63 steel having a high 
nickel content, the panels will have a 
thickness of approx. 950 mm X 400 mm X 0.5 mm thick. 

Each tank will, in fact, consist of two con-
tainers of identical construction, one inside 
the other to form the primary and secondary 
barriers, both being supported by insulation 
over the entire surface. The resulting two 
inner layers of insulation will consist of plywood 
boxes approximately 950 mm X 400 mm X 
200 mm thick, filled with silicone-treated 
expanded perlite powder. Joints will be 
formed with a low viscosity liquid. Polya-

milieu, which is made from a dry mix composed of 
expanded perlite powder and phenolic glue 
and filled in a 
building remote from the berth. Holes in 
the boxes will permit circulation of inert gas 
throughout the space to be 
vented. The gear for sealing and to fa-
militate monitoring the space for cargo 
leakage.

Apart from these three ships for Shell, 
Lloyd's Register class has been specified 
for many other large LPG or LNG carriers. Of 
the 46 ships built or on order over 
ten thousand tons gross under construction or in 
tour-build-
out the world on 30th September, 18 were 
for Lloyd's Register class including three 
LPG ships (capacity 52.000 c.m. each) and 
two LPG/LNG ships (capacity 35.000 c.m. 
each) to be built at La Seyne.

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