MERCHANT SHIP APPLICATIONS OF MEDIUM SPEED GEARED DIESEL ENGINES AND ASSOCIATED AUXILIARY MACHINERY

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In a written contribution MR. S. M. ADAMSON, B.SC. (Associate Member) said that to his knowledge, this was the third paper to be read to the Institute in which the design and use of auxiliary machinery modules were discussed.

Whereas he fully agreed that there was considerable merit in designing an installation as far as practicable with modules, the use of modules for modules sake regardless of economic and general overall viability, must not be tolerated. Unfortunately the papers read to the Institute on this subject were all guilty of this to a greater or lesser extent. For instance, the Main Engine Services module described by the author came into this category.

The policy of designing large modules around large heat exchangers had many disadvantages not least of which was the unnecessarily large floor area taken up, and the elaborate arrangements necessary to support and lift the module. Furthermore, such a module by virtue of its size must be fitted into the ship very early in the installation stages, sometimes before the main engines, and this was not always convenient.

In Mr. Adamson's opinion the arrangement adopted in some of the SD 14 Liberty replacement ships was a good compromise arrangement. In such installations, large heat exchangers were fitted on to a bulkhead in the close vicinity of a module or modules containing the associated pumps and other equipment.

The author stated in the paper that an alternative Main Engine Services module had been designed to include tank type lubricating oil pumps. How was this achieved and what were the author's reasons for not using tank type pumps as standard, and what advantages accrued from fitting them on to a module rather than mounting directly on built-in double-bottom lubricating-oil drain tanks?

One of the decisions made early in the development of the B.S.R.A. standard ranges of modules, some examples of which were illustrated in the paper, was that only standard marine valves would be fitted on all modules.

In view of the quite liberal usage of valve chests on some of the modules described, should it be assumed that the use of standard valves would add complexity to the modules, and if so did the author consider that some sizes and designs of valve chests should be retained as standards?

MR. J. E. CHURCH, C.ENG. (Member) wrote that he found Mr. Buchanan's paper of very great interest as he had been advocating medium speed geared Diesel engines for over 20 years and had been patiently waiting for them to reach their present stage of development when they would be a very attractive proposition for many ships.

He was therefore generally in agreement with everything described in the paper except that in his view serious mistakes were being made in the development of this type of machinery in the matter of method of driving the essential auxiliaries.

Experience had shown that the weakest link in any propulsion system was the electrical link which provided motive power for driving essential auxiliaries. In the case of slow running machinery, in spite of many attempts, a satisfactory method of driving lubricating oil and jacket water and other circulating pumps from the main machinery had not been found and the best solution was inevitably to employ electrically driven pumps. As engine powers increased so did the horsepower for driving these essential auxiliaries, and Diesel generators had to increase in size from 100 kW in rapid stages to over 600 kW. The electrical installation had consequently become large and expensive and complicated but nevertheless still the weakest link in the chain and as the value of ship and cargo dependent upon a single engine had increased enormously so had the dangers which could follow an electrical black out.

In Mr. Church's view the advent of medium speed machinery had made it possible to overcome these disadvantages at one fell swoop, for it now became simple and practicable to arrange for engine attached auxiliary drives, yet misguided designers were using this to perpetuate electrical troubles by adding shaft driven alternators which did nothing to remove the weakness of the electric link, but in fact increased, for an engine failure then meant a *total* black-out which was quite unnecessary, and unacceptable.

The obvious solution was that each medium speed engine must drive its own essential auxiliaries direct. Fortunately practically every design included a built-in lubricating oil pump and some also a jacket water circulating pump and so far as Mr. Church was concerned no engine was worth considering without these, which must be properly designed and built-in by the engine designer, for to add them afterwards as an afterthought was not the same thing. A seawater circulating pump also should be engine-driven although this was less important for sudden failure of an electric one did not necessarily stop the main engine immediately as there would usually be time for the standby pump to be brought into use.

The same remarks applied to the lubricating oil pump for the gear case, and gearing manufacturers must design this with suitable drive from their gearing and build the pump into their gear case.

As a result the electric power dwindled back to 200 or 300 kW per machine and the propeller would continue to revolve right through any electrical failure, which would then not be a black-out, for emergency lighting would render all things safe whilst electric power was being restored. There need then be only one standby electrically driven pump for each service but this would seldom be used and the entire engine room installation would consist of fewer units. Only thus would reliability with simplicity really begin.

As far as modules were concerned Mr. Church had yet to find one which he considered acceptable and although the merits of these were undisputed they were of more benefit to the builder at time of installation than to the operators throughout the ship's life. His principal concern was that they usually impeded rather than improved accessibility for operation and maintenance and his requirements under this heading were that every pump and auxiliary must have complete access all round without obstruction and this generally was not possible with any module, some of which were in fact unnecessarily cramped. He therefore refused to accept for 20 years of a ship's life any such difficulties in the way of repairs and maintenance merely to save a little money and inconvenience to the builder at the time a ship was built.

Mr. I. Bennett (Graduate) wrote that whilst he agreed with the author on the advantages to be gained by much greater standardization of arrangements, systems and auxiliary machinery modules, the author's concept of the main machinery being supplied as a package by the main engine builder must be questioned. The co-ordination of the installation design was normally undertaken by the Engineering Division of a shipyard. The paper underestimated those aspects of a given contract dependent upon the specific requirements of that contract and the duty of the vessel involved. As a result the proposed scope of supply of an engine builder acting as a main machinery contractor was too wide. The main machinery contractors would not be able to operate effectively unless they had the same access to the owner's and the shipyard's hull, electrical, purchasing and contracts departments as had the shipyard's own engineering design departments.

Apart from standard ships such as "Replacement Liberty Ships" it would be unusual for all the data required by a main machinery contractor to be available at the time a main engine was ordered. Therefore contractural difficulties could be anticipated as the effects of the overall ship design emerged.

It was proposed that an initial investment in technical effort would make available a range of module designs which could be universally applied over a period thereby reducing costs. Unfortunately the components of the modules had not been standardized and the service requirements of engines were changed by developments in engine design and ratings. Classification and statutory requirements were also changing and therefore there was constant technical effort required in updating the standard ranges of module design. The difficulties were highlighted by the fact that it was not possible to design a standard module common to two makes of main engine.

The only economic solution to this problem was to develop typical configurations for each module but only prepare working drawings for actual contracts: however, it should be possible to use these working drawings for several different contracts.

The paper mentioned the improved opportunities for standardization arising from the use of modules but in conflict with this statement was the use of valve chests in several of the modules. These were non-standard items and were usually avoided.

When employing modules in installation design other basic design requirements should also be met, for example, the need to minimize piping. In Fig. 1 the various modules in the fuel system were dispersed all around the engine room. Could the author comment on the techniques employed during the design study to optimize pipe runs? The fuel oil transfer pump module provided a further example. In many vessels the heavy oil transfer pump had a severe suction problem which necessitated it being placed as close to the tank top as possible. However, the standard design would not be suitable in these cases because the motor of the Diesel oil pump would then be in the bilges.

The preferred choice between wax element and pneumatic control valves was left in doubt. Could the author give further details of the extent of the investigation into this matter and state whether the considerable service experience with wax element valves had been assessed? This type of valve should be satisfactory with c.p. propeller installations in which the engines were running constantly. This was a factor which could further improve the case for the c.p. propeller as detailed in Appendix II.

Other factors which could reduce the additional cost for the c.p. propeller would be the use of engine driven pumps and the adoption of large tanker practice of having only two main alternators, one engine driven and one independent set. A considerable saving in cost could be achieved by utilizing a wet sump lubrication system.

This was offered by most engine builders but was rarely taken up by owners and shipbuilders. Could the author comment on this fact?

When discussing auxiliary machinery modules it was advisable to emphasize the need for even greater attention to planning. The size of the larger modules approached that of the main engines and gear-box and therefore the assemblies must be available for shipping in the correct sequence. In these considerations the facilities of the building yard must also be taken into account because these might demand an early start on the assembly of steelwork on the berth. Thus the availability of modules and the steelwork erection programme must be carefully integrated.

Author's Reply

The author fully agreed with Mr. Adamson that the use of modules regardless of economic and other considerations was indefensible. Ideally, of course, a comparative costing exercise should be carried out on modular and non-modular principles. The practical difficulties of such a comparison were legion: variations in shipyard procedures made it difficult to establish a broadly applicable basis for evaluating the savings in installation costs from the use of modules, and the full economic advantages accruing from series production of standard modules could not be assessed until the principle had been utilized in a number of ships.

In the author's opinion, the Main Engines' Services module was more compatible with modern shipyard facilities than a multiplicity of individual components. The Main Engine Services module installation should be carried out before the main engine installation. In the author's experience medium speed main engines were installed relatively late in the building cycle after the bulk of outfit equipment. The fitting of large heat exchangers to bulkheads below the engine room flats after the main engines were installed would also be no easy task.

The amount of floor space occupied by vertically mounted heat exchangers was surely not very great, and the author pointed out that the Main Engine Services module was not designed around the heat exchangers as Mr. Adamson suggested. The design was a combination of the concepts of "Cooler module" *v.* "System module", as explained in the paper in some detail. The removal of the heat exchangers from the module to an adjacent bulkhead introduced further joints in the lubricating oil piping system which had to be made on board, thereby increasing the danger of the ingress of dirt into the l.o. system and increasing the amount of on-board erection work.

The same considerations applied to the tank type l.o. pumps. Free standing pumps, on the module, were adopted for the basic module design to enable the greatest proportion of the l.o.

system to be erected in the clean conditions of the workshop ashore, where the system was flushed, tested and sealed before despatch to the ship. In preparing an alternative module design for tank-type pumps, therefore, an attempt had been made to preserve the integrity of the system by incorporating the pumps into the module. The pumps were mounted on the top of the module and projected down through wells in the module base. The wells therefore, would be sealed extensions of the double bottom tanks. This configuration was not adopted as standard as it was not possible to provide a common tank type standby and either four tank type pumps or one free standing unit would be required to provide standby capacity. By mounting tank type pumps on the module not only was system cleanliness facilitated but the motors were consequently mounted well above the bilges with little extra seating cost.

The author was grateful to Mr. Church for his logical explanation of the advantages of engine driven auxiliaries and there was no doubt that these were the considerations responsible for the increasing popularity of this type of arrangement. The author also drew attention to Mr. Church's qualifying clause that the auxiliary drives "must be properly designed and built in by the engine designer," for this had often not been the case in the past. The failure of an engine-driven pump drive mechanism unfortunately often incapacitated not only the pump but also the engine, so that unless this part of the engine received adequate design attention, the sought-after increased reliability of the machinery would not be achieved.

If the sole advantage of the modular concept were to save a little money and inconvenience to the builder, then the author would agree that there would be little point in installing them. Some of the other advantages, which were explained in more detail elsewhere in the paper and its discussion are reiterated briefly here:

1) increased reliability during initial running due to improved cleanliness of pipe systems after erection, particularly lubricating oil;
2) improved performance and reliability of main propulsion machinery due to accurate matching of auxiliary systems to the main engines and each other—the performance of each auxiliary module was accurately known from type testing, and variations in reliability and performance due to variations in shipbuilders' practices for the sizing and arrangement of auxiliaries were avoided;
3) simplified identification and improved availability of spares for component parts of the module through standardization and a reduction in the number of type variations.

The accessibility for operation and maintenance for the modules described in the paper was studied with some care, and as described in the reply to Mr. Scrimgeour in the previous discussion, all the designs were submitted to a number of shipowners for their comments before finalization.

In replying to Mr. Bennett, the author pointed out that the concept of the main machinery being supplied as a package by the main engine builder was not an original one, nor was it one that had remained at the conceptual stage only. As previously mentioned, in the field of high powered steam turbine installations there were several manufacturers offering and installing standard steam turbine propulsion packages with considerable success, and, as could be seen by reference to the earlier discussion of the paper, the Diesel engine manufacturers concerned with the engines directly studied in the paper were both willing to offer the same arrangements for their own engines, and had to a limited extent already done so. The proposed scope of the engine builder's supply was no wider than that already covered by the steam turbine manufacturers' package, and while one must agree that the variations from ship to ship and owner to owner were numerous, there was no reason why the contract or extent of supply must in every case be identical. It was not necessary, nor even customary, for all the details required by the engine builder to be available at the time of the placing of the contract. In that the main machinery contractor was no more than a subcontractor, albeit a major one, there was no reason why he should have the broad access to the shipyard departments suggested.

The author agreed fully with the daunting extent of the work involved in setting up the range of standard module designs and the problems of updating the designs for alterations in engine ratings, and classification society requirements. The extent to which detailed standardization could be taken would of necessity be a function of experience.

The use of valve chests in the module designs was, it must be admitted, a compromise between standardization and compactness (a compromise which had been avoided in all those designs for which B.S.R.A. were responsible, both in the study described in the paper and in their work on modules for slow speed Diesel engines). In those modules designed by the author's own organization, valve chests were only resorted to where it was clearly established that the use of separate standard valves would necessitate an increase in the size of the whole module and where this increase was held to be unacceptable. It was fully agreed that these were undesirable items in the light of the quest for standardization of components.

With reference to the dispersal of fuel oil system modules these were:

a) oil fuel transfer module;
b) purifier module;
c) surcharge module.

Before positioning a) and b) the heavy oil fuel tanks were positioned symmetrically to port and starboard to provide uniform weight distribution about the ship centreline, thus either module could be situated anywhere between the two tanks without incurring extra heavy oil pipe length. Lubricating oil purification piping was of low cost relative to that on the Main Services module; the purifier module could not therefore be situated over the lubricating oil tanks on the centreline and was located under the flat which supported both Diesel oil settling and ready use tanks together with the heavy oil tank. The transfer pump module was fitted into the remaining space on the other side. The surcharge module was flat mounted as the saving in pipe length by floor mounting did not offset the disadvantage of an extra seat.

If the transfer pump module in a particular installation were required to be closer to the tank top, a vertical Diesel oil pump could simply be substituted for the horizontal unit shown on the module thus keeping the motor out of the bilges.

The choice between pneumatically controlled valves and wax element thermostatically controlled valves should be made on the following criteria:

i) was a high rangeability or turn down ratio required in the application?
ii) what transient and steady state temperature deviations from the set point were admissable?
iii) were any forms of cascade control required?

In selecting the pneumatic valves the choice of control functions between the sensing element and the final valve actuation could be varied considerably. Furthermore, wide variations in valve characteristics were available with these valves allowing the control characteristics to be accurately matched to the system. These factors meant that where cascade control was required pneumatics must be adopted and furthermore, if the engine builder limited the temperature deviations of cooling water and lubricating oil temperature, again they should be adopted.

The experience of wax element controlled valves was taken into account and undoubtedly these devices had many advantages in terms of simplicity and ease of initial setting up over the pneumatic elements. They did, however, have a significant hysterisis in their operational characteristics and also suffered from the fact that the temperature sensing and control functions must take place at the same point in the system—this latter factor was a very important one.

The duty which determined the rangeability of control valves in jacket cooling water and lubricating oil temperature control systems was a combination of the external cooling fluid temperature and the engine heat input. Low input temperature cooling fluids added a considerable amount to the rangeability requirements, so consequently this major factor depended on the routing of the ship and not on the propulsion plant.

It was true to say that the continuously running engine should have a lower range of heat outputs to accommodate. However, c.p. propellers were used much more frequently in manoeuvres and so could put higher demands on the cooling system's response.

The reasons for which engine driven pumps were not considered in the study were stated in the paper, as was the manner in which, to a large extent, this particular aspect of the study had been overtaken by events in the three years since its inception due to the marked increase in popularity of engine driven pumps. With engine driven pumps and especially when the main engines were flexibly mounted, wet sumps were the obvious choice. With motor driven pumps dry sumps draining to double bottom tanks offered a number of operational advantages particularly ease of access. Means to ensure that the oil would have an adequate residence period for de-aeration in the tank under extreme pitch and roll conditions could also be provided more easily.

While agreeing that the reduction in the number of alternators to two would reduce first cost, the resulting inflexibility of the generating system would, it is felt, be unacceptable to many owners.

The author expressed his gratitude to those who had contributed to this extended discussion of the paper.

Marine Engineering and Shipbuilding Abstracts

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Heavy-duty Gas Turbine and its Marine Application

Gas turbines come in many sizes, shapes, arrangements and cycles: simple-cycle machines have low cycle efficiencies and are of limited marine application. Single-shaft machines have narrow operating speed ranges and require special application consideration. The most suitable gas turbine is a two-shaft, heat-recovery-cycle unit.

A two-shaft machine is one in which the load shaft and compressor shaft are mechanically independent, being aerodynamically and thermodynamically linked through a stator-supported variable area nozzle. Such a machine is basically a variable-speed offering complete flexibility for marine use since it can be used with all standard transmission systems, including gearing with a c.p. propeller, reversing gear with fixed-pitch propeller or an electric transmission system. The variable area load turbine nozzle incorporated in the design of the two-shaft machine allows reduction of air flow and the maintenance of maximum firing temperature resulting in better part load efficiency than any other unit arrangement.

Horsepowers available are listed in the table showing units ratings along with specific fuel consumptions. Because gas turbine cycles are susceptible to inlet and exhaust duct drops which in turn are subject to design choices (such as additions of inlet restrictions, elbows, etc., and exhaust restrictions, boilers, heaters, etc.), no attempt was made to include these losses in the table. A good estimation of such loss can be made by allowing 2 per cent loss for each 4 in H_2O inlet drop, 1 per cent loss for each 4 in H_2O exhaust drop. As an example, a steam generator or water heater in the exhaust would result in 6 to 8 inches pressure drop and reduce output by $1\frac{1}{2}$ per cent to 2 per cent. Each 4 in drop in the inlet or exhaust would, additionally, adversely affect the fuel rate by 1 per cent. Transmission losses would also further reduce output. Intermediate ratings not listed in the table are available by derating the next largest machine and essentially an almost infinite number of ratings is available or by combination or units.

The kinship of the heavy duty marine gas turbine to the steam turbine is readily discernible featuring heavy horizontal flanges, sleeve bearings and robust construction to provide long operating life to overhaul. Low maintenance is further enhanced by specially incorporated design features.

UNIT RATINGS

Model	Output: Metric hp* (maximum continuous)	gr/hph
M 1002R	4680	213
M 3002R	11 650	206
M 5002RA	23 400	200
M 5002RB	28 300	201
M 7002R	57 750	193

* Rating: horsepower at 65°F (15°C) 14·7 lb/in^2 (abs.), no inlet or exhaust pressure drops. SFC based on 17 450 Btu/lb. (lower calorific value).

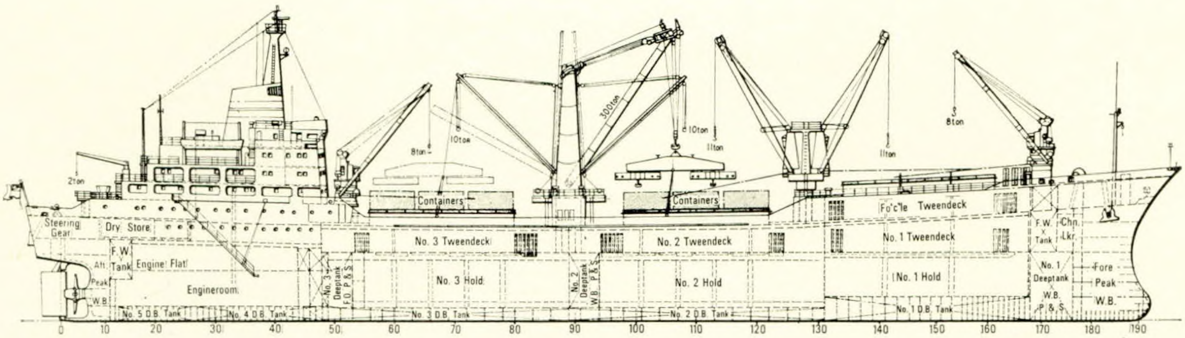
Each gas turbine is base-mounted with its accessories, wiring and piping in place. Such an arrangement allows unit testing with its own system so that each unit is positively proof-tested not only for performance of aerodynamic and thermodynamic parameters but for such additional items as electrical integrit, system performance, device functions and many more purposes.—*Cullen, P. J., The Motor Ship, November 1969, Vol. 50, pp. 366–370.*

Heavy Lift Ship

The British India Steam Navigation Company has recently taken delivery of the 10 000 grt *Amra*, the first of two vessels with 300 ton lift facilities to be built by the Swan Hunter Group at the Readhead yard.

In addition to the heavy lift equipment the ship is fitted with comprehensive break bulk cargo gear, and has three large single hatches which will facilitate a considerable degree of mechanized handling, which the owners hope will lead to a most flexible employment.

The vessel is built to Lloyds Register of Shipping Classification ✠ 100 A1, "Strengthened for Heavy Cargoes", and is fitted with a ram bow and a transom stern. A c.p. propeller is coupled to a Harland and Wolff B and W 6K74EF Diesel.



General arrangement of Amra

Principal particulars are:

Length, o.a.	500 ft 0 in
Length, b.p.	470 ft 0 in
Breadth moulded	70 ft 0 in
Depth to upper deck	40 ft 6 in
Service load draught	28 ft 9 in
Gross register	10 024 tons
Hold capacity (bale)	663 000 ft^3
Hold capacity (grain)	715 000 ft^3
Service speed	17 knots
Deadweight/draught ...	13 950 tons/31 ft 6 in

Telephone facilities are fitted at the control positions to control the ballasting operations necessary when handling these lifts. 1300 tons of ballast capacity is available for counter listing, and water can be transferred at a rate of 500 tons per hour. Suctions have been arranged so that this rate may be achieved with a 20 degree list. The loading rate of the Stülcken derrick may be doubled when working with a load of up to 50 per cent of the SWL by securing half the purchase to the foot of the derrick. A main traverse, fitted with moveable weights for load balancing, and two cross traverses also form part of the equipment.

Propulsion is by a 6-cylinder Harland and Wolff/B and W two-stroke single acting, crosshead Diesel. The normal service rating is 10 600 bhp (metric) at 120 rev/min. The engine is arranged to operate on heavy oil of up to 3500 Redwood No. 1 at 100°F. Two turbo blowers of Brown Boveri type VRT500 are mounted on the back of the engine and work in conjunction with the two air coolers.—*Shipbuilding and Shipping Record, 5th December 1969, Vol. 114, pp. 18–19; 21.*

High-speed Belgian Engines in Sea-going Tugs

The Cockerill TR-240 CO-type was introduced within the last few years and has been used in a number of interesting marine and land applications. Two of these engines were installed recently in two sea-going tugs, *Wielingen* and *Jacques Letzer,* which were built at the Beliard and Murdoch shipyard at Ostend for the Union de Remorquages et du Sauvetage (Towing and Salvage Group) of Antwerp.

Intended primarily for providing assistance to large tankers each tug is powered by an eight-cylinder Cockerill TR-240 CO-type engine rated at 2000 bhp at 1000 rev/min and is provided with a 200 ton/h fire pump, foam throwing equipment and a connexion box for submersible salvage pumps. The overall length of each vessel is 33·33 m with an overall beam of 9·22 m and a depth of 4·6 m. Maximum speed is 13 knots and the static tractive effort is 24 tons.

The Cockerill TR-240 CO-type engine is a supercharged four-stroke machine with cooling of the combustion air between the turbocharger and the inlet manifold; its mean effective pressure at normal service output is 16·18 kg/cm^2 with a piston speed of 10·16 m/sec. The engine has an underslung crankshaft which is a surface-hardened, one-piece forging supported in a one-piece cast-steel frame. The cylinder heads are of the four-valve type, the exhaust valves having renewable seats. Other features of the engine include aluminium alloy pistons with cast-in cooling coils and thin wall, tin-aluminium bearings.

To prevent undercooling of the charge air, the air cooler is circulated from the fresh water jacket cooling system. The engine works with a dry sump, the lubricating oil returning to a tank built into the hull of the vessel, from which it is drawn by an engine driven pump. A drive is also taken off the forward end of the engine for a 300 hp hydraulic pump controlling a hydraulic towing winch. Manufactured by Baensch the winch has a capacity of 650 m with a rated pull of 10 tons at a speed of 23 m/min and a permissible static load of 50 tons.

A Vulkan flexible coupling connects the engine to a Hindmarch-Messian reverse/reduction gear-box. This has two forward speeds (169 and 189 rev/min) and a reverse speed of 170 rev/min. Each ratio has its own built-in hydraulically controlled coupling, and to allow the propeller—a steerable Kort Nozzle unit—to be braked and held stationary, brakes are arranged on the secondary shafts of the reducing gear, while a power take-off driving a generator is provided from the primary shaft. Remote control of the gear-box and the engine is provided from the wheelhouse, monkey island and engine room, the machinery being arranged for unattended operation.—*The Motor Ship, November 1969, Vol. 50, p. 378.*

Underwater Lighting and New Light Sources

When light passes through water it is strongly attenuated compared to passage through air. As one proceeds downward from the surface of the sea the light available from the sun decreases in an exponential manner. In very clear water, for example, the light level at 100 metres depth has decreased to 1 per cent of that at the surface and the light has a pale green diffuse character; at 200 metres it has dropped to 0·01 per cent and the light appears dark green. Below 300 metres the sea is virtually in total darkness.

Even in the clearest water artificial lighting is required for visibility below 200 metres depth and is often required at shallower levels. Optical character of water is greatly dependent upon location and depth. Light attenuation is the result of a combination of absorption and scattering due to the spectral characteristics of the water itself, the materials dissolved in the water, the particulate matter suspended in the water and small organisms, such as plankton, living in the water. The scattering effects are practically independent of wavelength since the particle size is much larger than the wavelength of light. The absorption, however, is wavelength dependent being high in the red and low in the blue-green region.

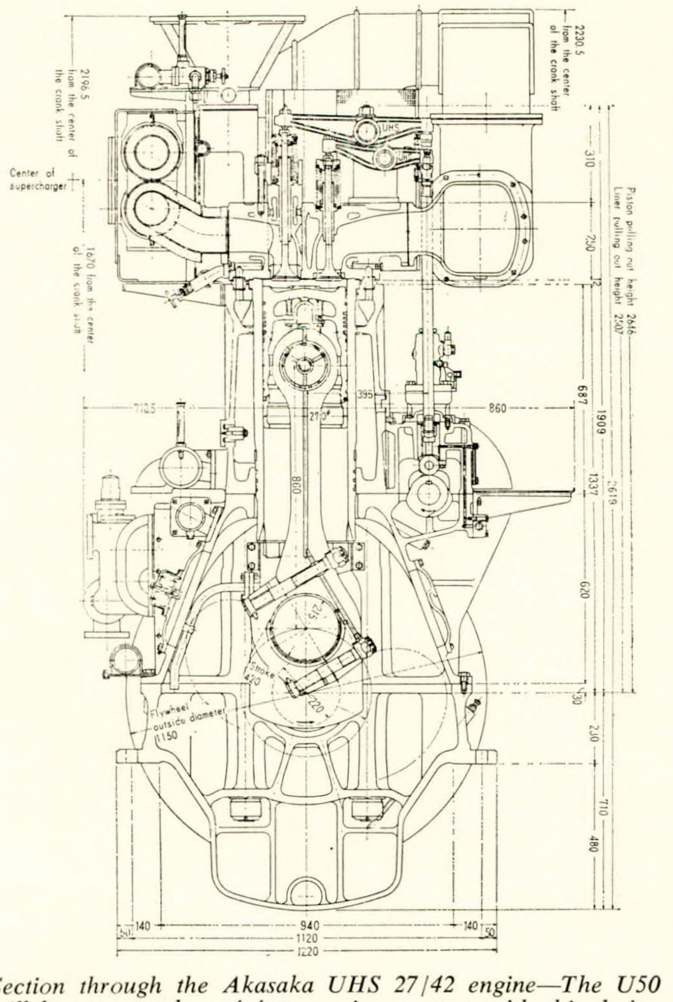
The majority of underwater lighting for visual observation and movies has utilized the quartz tungsten halogen incandescent lamp. It is a simple, compact source which requires no auxiliary equipment for its operation. It has a comparatively low luminous efficacy of approximately 20 lm/w but has a broad continuous spectrum, strongest in the red, which provides good colour rendition. The strong absorption of sea water in the red and the low luminous efficacy of this source, however, results in short viewing distance.

For many years attempts were made without significant success to improve the colour rendition and luminious efficacy of high intensity discharge lamps. Recently, technological advances were made which significantly improved both parameters.

From consideration of the spectral characteristics of sea water and the sensitivity of detectors, the thallium iodide-mercury discharge was chosen as a basis for introducing a significantly improved lamp for underwater lighting application. It has a luminous efficacy of 100 lm/w which is double that of the conventional mercury discharge and five times better than incandescent. In addition, it is a significantly better match to the peaks of sea water transmission, eye sensitivity and black and white television camera tube response.—*Larson, D. A.* and *Rixton, F. H., Under Sea Technology, September 1969, Vol. 10, pp. 38–39; 56–57.*

Japanese Diesel Engine Developments

The Akasaka Iron Works, whose factories are at Yaizu, an important fishing port in the south of Japan, have a new highly-developed four-valve four-stroke engine which, in direct-reversing form, has been giving good service in distant water fishing vessels. This is their model UHS 27/42 which, from six cylinders of 270 mm bore by 420 mm stroke,



Section through the Akasaka UHS 27/42 engine—The U50 will have a number of features in common with this design

develops 1000 bhp at 390 rev/min (corresponding to 16 kg/cm^2 bmep and 5·46 m/s mean piston speed. It is a very well proportioned cast iron-framed engine.

The company is now actively engaged in a much more powerful engine, the 6U50, which is being built with the support both of the Ministry and the Japan Ship Machinery Development Association. Many of the components have already been rig-tested and it is anticipated that the prototype will run early next year. As will be seen from the table below, while the specific cylinder output is high at 1000 bhp, the revolutions are relatively low at 380 rev/min.

Akasaka 6U50 engine:

Cylinder bore ...	500 mm	(19·68 in)
Piston stroke ...	620 mm	(24·41 in)
Piston area ...	1963 cm^2	(304 in^2)
Swept volume/cylinder	1221	(7402 in^3)
Cylinder output ...	1000 bhp	
Speed	380 rev/min	
Bmep	19·5 kg/cm^2	(277 lb/in^2)
Mean piston speed	7·8 m/sec	(1534 ft/sec)
Bhp/pa, cm^2 (in^2)	0·51	(3·29)
Bhp/sw vol, 1(in^3)	8·2	(0·135)
Specific fuel cons g(lb)/bhp-h	153	(0·336)

This too is a cast iron engine with four valves operated by push rods and stepped rockers, like the UHS 27/42. The six-cylinder prototype will be 7·83 m in overall length, 2·12 m in breadth and 3·54 m high above the shaft line with a weight of 75 tons.

One IHI-BBC VTR500 turbocharger will deliver at 2·47 kg/cm^2 (abs). The crankshaft, by Kobe Steel, is of tempered chrome molybdenum steel. Composite steel/alloy oil-cooled pistons are used. It is intended to offer the engine for use on fuel of up to 1500 secs viscosity with limits of 3·5 per cent s at 100 ppm Va. A Vee-form version is planned for the future.—*Marine Engineer and Naval Architect, November 1969, Vol. 92, pp. 470–473.*

Cunard's Atlantic Causeway Enters ACL Group Service

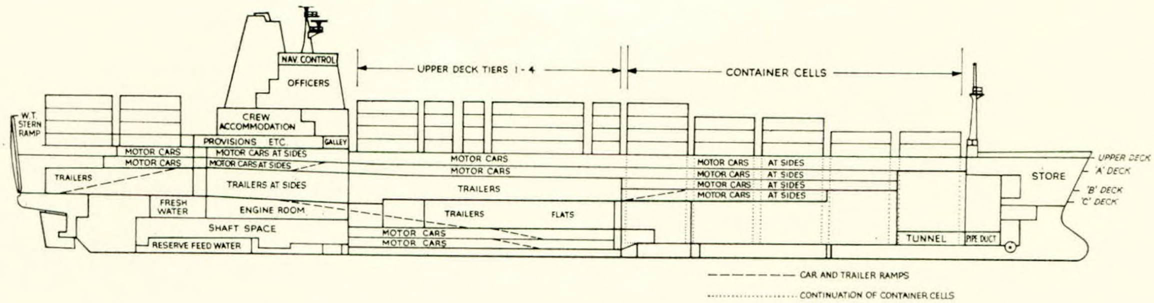
The first ship in the second series of Atlantic Container Line vessels to enter service is the Wallsend-built *Atlantic Causeway*, owned by the Cunard Steam-Ship Company Ltd.

ACL second generation ships principal particulars are:

Length, o.a.	212·17 m
Length, b.p.	196·00 m
Breadth, moulded	28·00 m
Depth to upper deck	19·36 m
Draught on summer freeboard ...	9·31 m
Corresponding deadweight ...	18 219 tons
Container capacity including trailer decks ...	750 × 20 ft (or equivalent)
Machinery	2 × 17 500 shp steam turbines
Service speed	24·25 knots

From an engineering aspect, perhaps the most significant feature of *Atlantic Causeway* is that the twin-screw propulsion machinery has been supplied by AEI (now the Turbine Generator Division of English Electric-AEI).

The sets installed in the Cunard and Holland America ACL ships are of AEI's H46M single-cylinder design of 17 500 shp. Cross-compound, two-cylinder machines were also considered at one time but the simplicity of the single-cylinder design and the reduced width, compared to two-cylinder plant, decided the choice in favour of the single-cylinder machines. The need to restrict the width of the uptake casing in the main vehicle deck was also instrumental in the choice of the single-cylinder turbines. The two roof-fired Foster Wheeler ESD III boilers which supply the turbines in the Cunard ships are arranged end-to-end on the fore-and-aft centre-line, an unusual arrangement but obviously necessary in the in-



Cargo stowage arrangements of Atlantic Causeway

terests of packing a 35 000 shp propulsion system into the fairly narrow aft section of the ships, while at the same time providing ample space for the ramps and for vehicle manoeuvring on the deck above.

The AEI turbines in all cases drive through reduction gearing of AEI design and manufacture of the double helical, dual-tandem articulated type. These gear units will normally deliver 17 500 shp each at service revolutions but the actual maximum ahead service power is 19 250 shp per shaft.

Control of the main machinery can be effected from the wheelhouse, a spacious machinery control room at boiler burner level in the engine-room or from standard local control stations.—*Shipbuilding and Shipping Record, 12th December 1969, Vol. 114, pp. 31; 33.*

Propulsion Systems for Dynamic Positioning

Dynamic positioning is the technique of holding a ship or platform in a fixed position at sea with the use of propulsive or thrust power. From the definition three basic requirements are obvious.

First, there must be a means of knowing where the ship or platform is in relation to the bottom or other known reference points and a way of measuring deviations from this fix.

Secondly, there must be a power plant and propulsive system which provides adequate thrust in the proper direction and time in order to maintain the position.

Thirdly, there must be an instrumentation and control system which relates position data to the required fix position and activates the power system to maintain this position.

Perhaps one of the most demanding applications for dynamic positioning is *Glomar Challenger*. This is a 10 500 ton, 402 ft, sophisticated drilling ship being used for the $12·5 million Deep Sea Drilling Project. The vessel is now on Leg Six in the Pacific Ocean on a world-wide ocean floor drilling programme at water depths up to 20 000 ft.

Global Marine, the ship's designer, selected tunnel mounted fixed propellers for dynamic positioning, because of anticipated higher reliability and protection provided by recessed installation.

Four transveres tunnel thrusters are used—two in the bow and two at the stern. The bow tunnels are relatively long, the longest being approximately 50 ft. The stern tunnels, installed in the centreline skeg are about 7 ft long.

The thrusters are Schottel model S300 four-bladed units with 35½ in pitch. They are 59 in in diameter and generate about 18 000 lbs. of thrust at 545 rev/min. The right angle gear in each hub has a reduction of 2·57:1 and is driven by a spline-connected vertically mounted GE Model CD-6513 DC traction motor which develops 850 hp at 1400 rev/min.

The application of fixed thrusters on *Glomar Challenger* means that more horsepower must be provided as compared with rotatable units which give thrust in the exact direction required.

The AC Electronics Defence Research Lab of GM was selected to supply the position control system. It operates with four hydrophones, bottom sonar beacon, and a model Sigma 2 Scientific Data System digital computer, The Honeywell developed position control system intended for MOHOLE was installed as a back-up.

The requirement for the position control system is to maintain a position within a circle with a radius of 3 per cent of drill depth. Thus for drilling to 18 00 ft the ship must be maintained in a position not greater than 540 ft away from the vertical line above the drill hole. According to Global Marine reports, the ship maintained position with beam winds up to 45 knots and beam currents of 2 knots when drilling in water depths between 11 000 and 18 000 ft.—*Under Sea Technology, September, 1969, Vol. 10, pp. 44–47.*

Series of Multi-Purpose German-built Ships

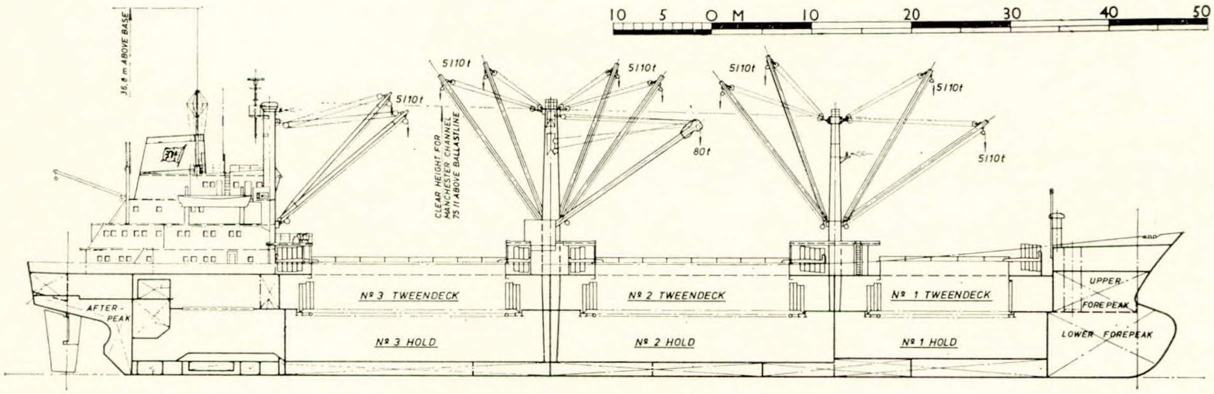
The first of a new series of versatile three-hold cargo vessels designed to carry bulk and general cargoes, grain, heavy lifts and containers has recently been completed by the LMG yard of Orenstein-Koppel AG, Lubeck, West Germany. This ship, *Carlo Porr*, 7460 dwt was delivered to Franz Hagen, one of an affiliation of six German shipowners. Self-propelled model experiments were carried out by the VWS Berlin model tank establishment at the designed full scantling draught to obtain the most favourable ship form, and performances checked at the reduced tonnage and ballast conditions. The lines plan was produced by Messrs. Maierform GmbH., who recommended the adaption on an SV-type bulbous bow. A transom stern, offering economy in building and maximum deck space aft, was also chosen.

A special requirement of the owners was that the vessels should be capable of trimming by the stern at any service draught by suitable arrangement of the ballast tanks. Stability calculations for bulk, grain and container cargoes were made in detail. It was found that additional grain bulkheads would need to be erected in No. 2 hatch only, to comply with the latest regulations for some types of heavy grain. When loaded exclusively with containers, these could be carried two-high on deck, but with some weight limitations.

Principal particulars of *Carlo Porr* are:

Length, o.a.	116·70 m
Length, b.p.	107·95 m
Breadth, moulded	17·20 m
Depth, to upper deck	9·90 m
Depth, to lower deck	6·70 m
Draught, full scantling	7·52 m
Corresponding deadweight ...	7460 tons
Corresponding speed	15·2 knots
Draught, to tonnage mark ...	6·53
Corresponding deadweight ...	5924 tons
Corresponding speed	15·7 knots

Main propulsion is by an Atlas-MaK eight-cylinder supercharged engine with four-stroke operation, developing 4000 bhp at 375 rev/min and running on blended fuel of 200 sec. Redwood 1 viscosity. Power is transmitted to the propeller shaft via a Vulkan E 2280 coupling and Tacke type H8N800 reduction gear with a ratio of 1:2·78, to give a propeller speed of 135 rev/min. The propeller thrust is taken by



General arrangement of Carlo Porr

a double horseshoe bearing fitted in the gear-box, and the shaft is doubly supported in the stern tube which is oil-lubricated and sealed by Simplex stern tube glands.

As model tests had not established a clear superiority for either a four- or five-bladed propeller with regard to vibration characteristics, a four-bladed propeller was chosen for its higher efficiency.—*The Motor Ship, November 1969, Vol. 50, pp. 354–356.*

Anaerobic Self-curing Plastic Gaskets

In addition to good sealing properties, gaskets must be capable of resisting attack by the fluid to be sealed and of coping with the sealing pressure and range of operating temperature involved. Also, an important factor is that they must retain bold tightness to ensure that high tensile and torsional loads are carried effectively.

From these considerations, it is clear that there are a number of design and functional problems associated with conventional compression gaskets made from soft materials, as well as such non-functional problems as their cost and ease of replacement in the field, and the need to stock many different shapes and sizes. It is therefore interesting to note that many of these problems appear to have been solved successfully by so-called "anaerobic" gaskets which can be produced in any shape by the direct application to the mating surfaces of flanges of a liquid sealing compound that converts to a tough, hard film of thermoset material in the absence of air between the surfaces.

The sealing compound, introduced under the designation of "Plastic Gasket" by the Locite Corporation, is a thick self-curing polyester resin containing 30 per cent of an inorganic filler and having a viscosity of from 15 000 to 25 000 centipoises, enabling it to fill clearances up to 0·2 in without substantial pressure loss. Also, the inorganic filler provides an effective low-pressure seal on flanges immediately after application of the compound. In fact tests have shown that such a seal is capable of withstanding pressures ranging from about 250 lb/in² for a 3 in flange to 5000 lb/in² for a 1 in flange and that, in this respect, anaerobic gaskets are, as soon as they are applied, at least equal to non-hardening pipe dopes. After full curing, these anaerobic gaskets can cope with substantially higher pressures, i.e., from 2000 lb/in² for a 3 in flange to 8000 lb/in² for a 1 in flange, and, here again, the tests have indicated that they are equal to conventional soft gaskets in sealing efficiency.—*Wittemann R. G. and Pearce M.B., Mechanical Engineering, August 1969, Vol. 91, pp. 26–29. Engineers' Digest, October 1969, Vol. 30, pp. 65–67.*

Effect of Strength and Thickness on Notch Ductility

For a number of years, standards and code-writing bodies have been attempting to specify fracture toughness in terms of the Charpy V-notch rather than the Charpy key-hole test, because various investigators have shown that V-notch test results correlate much better with service experience. However, the change to V-notch specifications has been deterred by uncertainty concerning the best criterion for establishing transition temperatures and the effect of strength and thickness on transition temperatures. Therefore, five steels covering a wide range of yield and tensile strengths (ABS-C: 39/63, A302-B: 56/88, HY-80: 81/99, A517-F: 121/134, and HY-130: 140/-148) were tested as quarter-, half-, single-, and double-width (QW, HW, SW, and DW) Charpy V-notch specimens in the longitudal and transverse directions and with through-thickness and surface notches. Transition temperatures were determined for various energy-absorption, lateral-expansion, and fracture-appearance criteria.

The results showed that energy-absorption criteria for determining transition temperature should increase with strength to ensure a constant notch ductility. Thus the best method for determining transition temperature was the direct measurement of lateral expansion. Of the lateral-expansion criteria evaluated, the 15 mil value agreed best with fracture-mechanics considerations.

The average increase in transition temperature was 60°F/from QW to HW specimens, was 26°F/from HW to SW specimens, and was 2°F from SW to DW specimens. This indication of maximum constraint for the SW specimen was not consistent with the effects produced when the standard V-notch was replaced with a fatigue crack. Consequently, the size of the Charpy test specimen that should be used for evaluating thick plates has not been established and requires additional study.

The effects of strength and thickness on transition temperature were much larger than the effects of testing direction, notch location, or notch acuity.

The results indicate that of the various criteria for evaluating the Charpy V-notch impact-test performance of structural steels, lateral expansion is the best criterion for compensating for the important effects of steel strength and plate thickness. Moreover, its validity is supported by fracture-mechanics concepts.—*Welding Journal, October, 1969, Vol. 48, pp. 441-s–453-s.*

Application of Toran Radiolocation System in Sea Trials

In the sea trials of the 213 000-dwt Shell tanker *Magdala*, the Toran radio-electric location system was used for tracing the ship's trajectories. This system, which operates on the usual hyperbolic principle, was originally developed for oil-prospecting, and has since been used in various hydrographic and allied applications. Its use for the *Magdala* trials was its first application in sea trials. The existing Decca chains, though excellent for navigational needs and for trials in general, were considered to be unsuitable for the accurate measurements needed in the *Magdala* trials, particularly those in the Baie de la Seine. Toran uses frequencies of the order of 1·5 to 2 MHz (the Decca-chain frequencies are about 100 kHz). The two Toran chains used in these trials were temporary ones set up for the purpose; to prevent certain ambiguities in positional readings (which can be avoided in permanent installations at the expense of some complication), a "phase-carrier" helicopter was used.

After explaining the principles of the Toran system, the author describes the equipment and procedures used, with special reference to the speed, turning, and stopping trials of *Magdala*. Numerical recordings of the times and hyperbolic co-ordinates were printed-out automatically every 10 or 20 seconds, as necessary. The ship's positions could be quickly plotted, by hand, to give a rapid approximate review of the results; results of greater accuracy could be made available quickly by fairly simple calculation. Complete processing of the data is best done by computer. The performance of the Toran system was excellent; in permitting a very exact kinematic analysis of the trials, it has fulfilled its essential task. The equipment is not difficult to set up.

The author mentions that Toran chains are at present being installed on the French coasts, and expresses the hope that, in conjunction with equipment specially adapted for shipboard use, they will lead to the further development of the techniques used for *Magdala*.—*Nizery B., Asst. Tech. Marit. paper presented at 1969 Meeting; Journal of Abstracts of The British Ship Research Association, August 1969, Vol. 24, Abstract No. 27 770.*

Ore Oil Carrier Converted for Phosphoric Acid Transport

The largest cargo of phosphoric acid yet to arrive by ship at Rotterdam, was recently discharged at the tank installation of Pakhoed N.V. at Rotterdam. The cargo arrived in the Swedish motor vessel *Vassijuare* of 23 500 dwt. The vessel, which is owned by Grängesberg-Oxelösund, was specially converted for the carriage of phosphoric acid. Neoprene sheet rubber lining has been applied to the roof, sides and bottom of the tanks as well as to all nozzles and openings. The tanks have been provided with rakes for preventing deposition of solids. The rakes consist of a superstructure, driver with motor, shaft, blade and centre steady bearing.

All wetted parts were constructed of 316L stainless steel. A pump with a capacity of 500 metric tons per hour is available for barge loading. Two 200 metric tons per hour pumps are designed for the loading of road and rail tank-cars, for which weigh bridge facilities are available for FFM's use.

All valves at the tanks have been made of stainless steel gate valves. All other line valves are rubber lined Saunders diaphragm valves. The instrumentation of the new tank group consists of high and low alarms, continuous digit indication of level and temperature.

The motor vessel *FFM-Vasiqaure*—a former ore-oil carrier—has been specially converted by Götaverken for Fertilizantes Fosfatados Mexicanos (FFM) into a chemical and oil tanker.

The chemical and oil tanker *Vassijaure,* converted to Lloyd's ✠ 100 A1, has the following dimensions:

Length, o.a.	595 ft 6 in
Length, b.p.	560 ft 0 in
Breadth, moulded	74 ft 6 in
Depth, moulded	44 ft 3 in
Draft on summer freeboard ...	34 ft 2 in
Gross tonnage	15 505 rt
Net tonnage	8426 rt

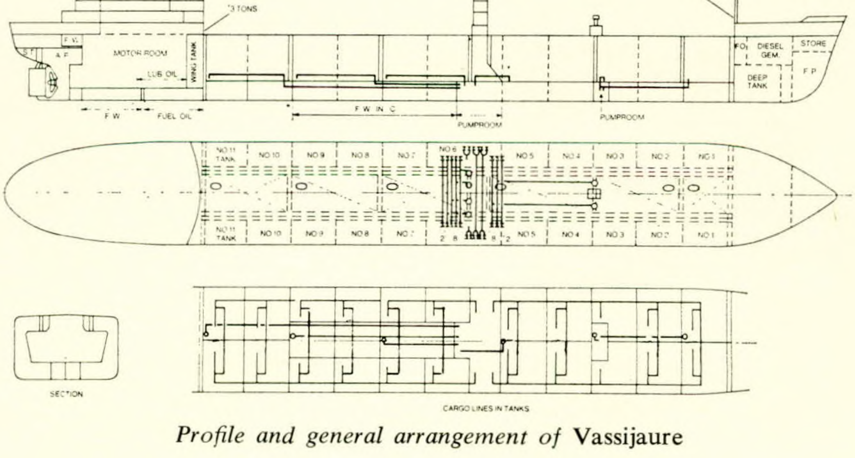
Capacities of stainless steel lined tanks (100 per cent): Tank No. 1, 828 m³; tank No. 2, 2716 m³; tank No. 3, 3172 m³; tank No. 4, 2710 m³; tank No. 5, 2716 m³; tank No. 6, 2552 m³. Total stainless steel tanks 14 695 m³.—*Holland Shipbuilding, October 1969, Vol. 18, p. 117.*

Regulating the Oil Feed in Cylinders of Marine Diesel Engines

The incorrect regulation of the feeding of lubricating oil to the cylinders of a Diesel engine (in both quantity and timing) can lead to a breakdown in the operation of the lubricant and the consequent occurrence of abrasive wear and scoring of the cylinders.

In 1966 the Odesso School of Marine Engineering carried out an experimental investigation into the lubricating-oil feeding process in the cylinders of large low-speed marine Diesel engines. The engine studied was a B & W 684VT2BF180 installed in the motor ship *Kosmonavt*. In B & W engines the feeding of oil to the cylinders is synchronized with the movement of the piston. The makers recommend that the lubricating oil should enter the cylinder at the moment the piston rings pass the oil-feed holes in the cylinder liner, and that, in order to ensure that this occurs at the correct time, the delivery stroke of the lubricator plunger should cease when the crankshaft, for a given cylinder is at a position 74°–78° after b.d.c. The investigation carried out showed that this recommendation was far from satisfactory.

It was found that using the makers' recommendations for regulating the oil feed caused the greater part of the oil to enter the cylinder before the piston rings reached the oil-feed holes. Consequently the oil fell on the cylindrical part of the piston above the uppermost piston ring, creating deposits of carbon which led to abrasive wear and scoring of the cylinder



Profile and general arrangement of Vassijaure liner. Examination of the pistons shows that the carbon deposits are sharply defined and occur in six positions (corresponding to the number of oil-feed holes in the cylinder), on the cylindrical part of the piston head above the uppermost piston ring.

It was established that altering the recommended regulations of the lubricator from 78° to 92° after b.d.c. considerably improved the distribution of the lubricating oil on the piston rings and working surfaces of the cylinder.

On the basis of this and similar investigations on other B & W engines, the Odessa Marine Engineering School has introduced its own recommendations for the regulation of the lubricating-oil feed to the cylinders of these engines.—*Miryushcenko, A. and Chukhrienko, S., April 1969, Morskoj Flot, No. 4 pp. 23–24; Journal of Abstracts of The British Ship Research Association, August 1969, Vol. 24, Abstract No. 27 842.*

Canadian Hydrofoil

The vessel designated *FHE/400* HMCS *Bras d'Or*, has a load displacement of 212 tons and at a hull-borne speed of 12 knots her endurance and seakeeping ability is comparable with those of a conventional destroyer-escort. Her foilborne speed exceeds 60 knots in calm water and in a state five sea is of the order of 50 knots. Her foilborne range is several hundred miles.

She is approximately 150 ft long and her hull is designed for minimum resistance and motions when hull-borne. Her narrow beam, 21·5 ft, is possible because of the stabilizing influence of the foil system at all speeds. He construction is all-welded aluminium.

The surface piercing foil system is of "canard" configuration with 90 per cent of her weight carried on the main foil of 66-ft. span. The hull clearance is about 10 ft at 50 knots and at rest the non-retractable foils draw 24 ft. The foil system is of maraging steel (18 per cent Ni) with an anti-corrosion coating of neoprene.

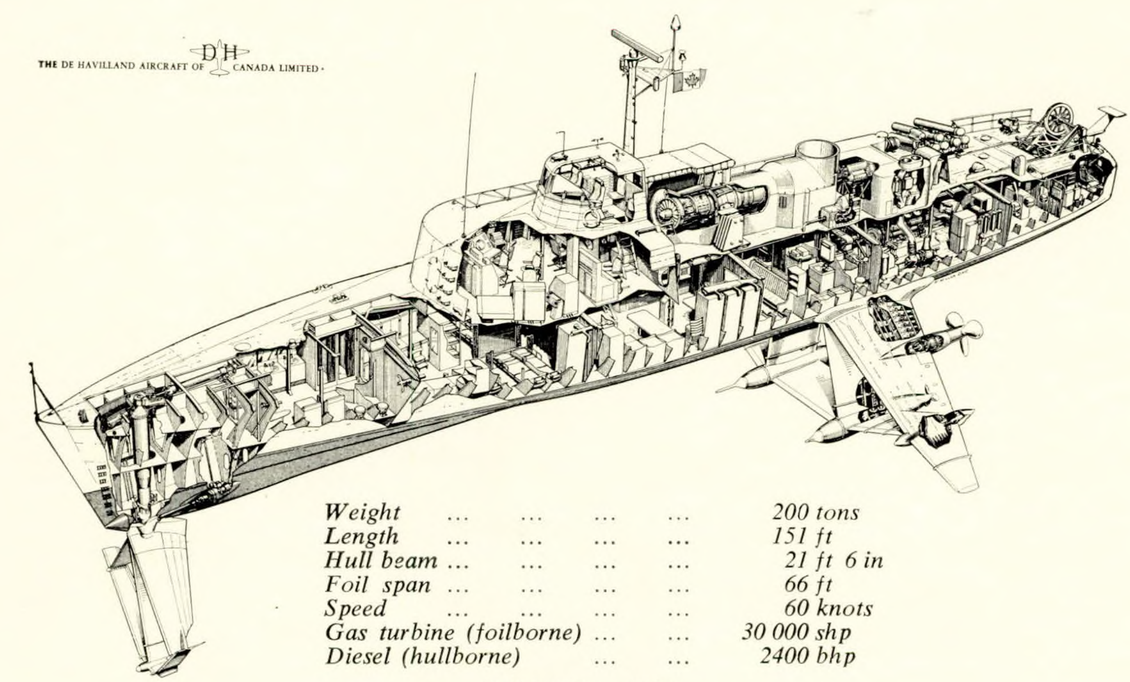
There are two separate propulsion systems. For the low-power, long-endurance, hullborne system a high-speed Diesel of 2000 bhp (maximum continuous) drives twin controllable-pitch propellers mounted on the take-off foils through a bevel-gear transmission link. Under foilborne operation these propellers are feathered.

For the high power required when foilborne an aircraft-type gas turbine of 22 000 shp (maximum continuous) is deck mounted to facilitate maintenance by complete unit replacement. Twin fixed-pitch supercavitating propellers absorb this power through another bevel-gear transmission system—they are mounted on pods at the base of the main foil struts.

When hullborne, auxiliary electrical and hydraulic power is provided by the Diesel. When foilborne the auxiliaries gear-box is driven by a 390 shp gas turbine separate from the high-power unit but arranged so that it can be coupled to the hullbone propellers in emergency. A small gas turbine of 190 shp is also available for emergency electrical and hydraulic power and supplies the starting air for the main gas turbine. All four engines burn JP-5 or high-distillate Diesel fuel.—*Rogers, T. H., Ship and Boat International, December 1969, Vol. 22, pp. 14–16.*

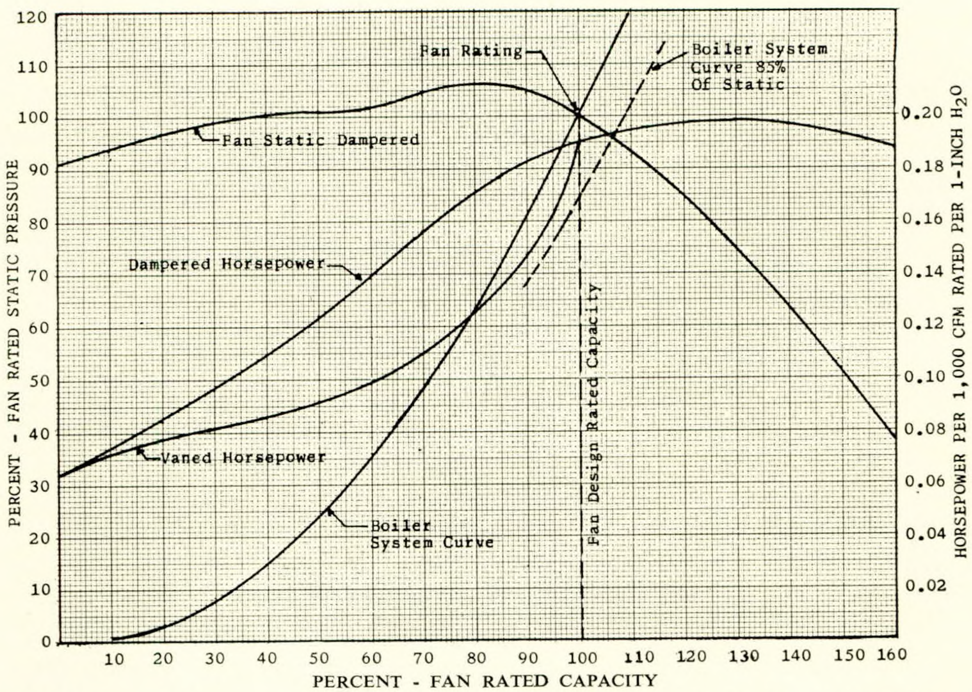
Flange Design in Sweden

The paper describes a new type of flange which is being developed in Sweden. It is more compact and lighter in weight than the current standards. The basic principles behind the design, and their application to the various components of the flange assembly, are explained. There is a discussion of the experimental work that was performed, together with other background information. The dimensions and working pressures that have been determined for a proposed flange series designed on these principles are also included. This proposal takes advantage of the newer steelmaking processes and the abilities of modern seals, such as O-rings, to make available an alternative series of pipe flanges to supplement those currently in general use. The principal features of this design are stiff, full-face, reduced-diameter flanges and slender, resilient bolts. These newer flanges may be used wherever lighter weight, more compact design, and improved performance are desirable.—*Webjorn, J., ASME paper No. 67-Pet-20 presented at 17–20 September 1967 meeting; Journal of Abstracts of The British Ship Research Association, August 1969, Vol. 24, Abstract No. 27 848.*



Weight	*200 tons*
Length	*151 ft*
Hull beam	*21 ft 6 in*
Foil span	*66 ft*
Speed	*60 knots*
Gas turbine (foilborne)	*30 000 shp*
Diesel (hullborne)	*2400 bhp*

HMCS Bras d'Or



Typical forced-draft fan performance curves—Based on a fan efficiency of 83 per cent, 100°F, 40 per cent RH and 29·92 in Hg

Boiler Forced Draft Systems

It is suspected that the boiler forced-draft systems in steam propelled vessels are considered, by many plant designers, to be rather uninspiring. They often do not seem to get the attention that their importance merits during the design phase. Yet failure of this system to meet requirements will impede plant performance at a very basic level.

The amount of air required for operation is a direct function of the amount of fuel burned. Fuel consumption is, in turn, based on heat balance calculations which usually state theoretical requirements in pounds per hour of a fuel oil having a higher heating value of 18 500 Btu/lb at 100°F (38°C) and constant pressure.

Each 100 Btu/lb variation in heating value of the fuel results in a 0·54 per cent change in air requirements. Since heating values can deviate from the standard by amounts approaching 700 Btu/lb, it follows that air requirements per pound of fuel can increase in the order of 4 per cent because of this factor. However, the correction for chemical composition of the fuel is expected to partially cancel out this increased air requirement.

In addition, the fans must also provide sufficient air for losses due to leakage in the ducting, boiler casing, soot blower penetrations and, when installed, rotary regenerative air heaters. As an average, it is suggested that the following figures be used for design purposes:

Double-cased boiler and ducting ...	2·5 per cent
Single-cased boiler and ducting ...	1·5 per cent
Regenerative rotary heaters ...	15·0 per cent

The pressure, or draft, required at the fan discharge is governed by the physical characteristics of the boiler draft system and, therefore, can be quite different from one installation to another of similar power. The units used are almost always inches of water and the fan is usually specified to deliver so many inches of static pressure at the discharge at rated cubic feet per minute.

The fan static pressure must be sufficient to overcome the following losses: differential between fan inlet and outside barometric pressure, air and gas side ducts, steam air heater or regenerative heater air side, boiler casing (windbox), fuel oil burner registers, boiler superheater(s) and main bank, economizer or regenerative heater gas side, and stack velocity less natural draft).

In order to assist the designer in arriving at a reasonable prediction of fan performance and driving power requirements where fan manufacturer's data may not be available, and to illustrate factors affecting fan selection the figure has been prepared, representing a typical fan performance curve for a centrifugal unit with backwardly inclined blades and inlet vane control.—*Giblon, R. P., Shauer, M. and Rolih, H., Maritime Reporter/Engineering News, 1st October 1969, Vol. 31, pp. 18; 23.*

Dynamics of the Electric Propulsion Installation of Icebreaker 'Kiev' While Operating in Ice

The regulation system for the propulsion installation aboard the icebreaker *Kiev* was designed on the basis of recommendations based on the results of tests carried out on the systems incorporated in her sister ships *Moskva* and *Leningrad*. As a result the systems aboard *Kiev* show some important differences.

The ship is provided with two basic regulation systems—a constant power system and a constant motor-speed system. The exciters of the main generators and the propulsion motors are the usual rotary type, each having three windings; however, only two (identical) windings are normally used. The two main generator-excitation windings, fed from regulators using magnetic amplifiers connected through a push-pull circuit, serve for the normal control of the propulsion installation. The third winding is switched in as a counter-compounding winding for emergency control of the installation when the regulator is out of action.

The systems were tested while the ship was operating in broken humped ice conditions in the Arctic, in order to evaluate the effectiveness of the design changes incorporated. While the propeller was operating in ice, considerable surges in current and power were still recorded, and drops in the speed of rotation of the Diesels were still great enough to lead to a possible increase in wear and fuel consumption. However, the observed variations in motor speed are considerably less than those in the sister ships, thereby stressing the importance of modifications and design changes based upon operational experience.—*Khaykin, A.* and *Syubaev, M., 1969, Morskoj Flot, No. 4, pp. 25–27; Journal of Abstracts of The British Ship Research Association, August 1969, Vol. 24, Abstract No. 27 849.*

Push Barge for Carrying Liquid Chemicals

The N.V. Scheepsbouwwerf en Lasbedrijf v/h J. V. Slob, Sliedrecht, has delivered the push-barge *Sieglind,* which has been specially designed and built for the carriage of liquid chemicals. The vessel was ordered from the yard by Messrs. Wijnhof & van Gulpen & Larsen N.V., Amsterdam.

The principal dimensions of the ship are as follows:

Length, b.p.	70·00 m
Breadth, moulded	9·45 m
Depth at side	3·20 m
Draught	3·20
Corresponding cargo capacity ...	1992 tons

The vessel has been given a double shell and double bottom. The cargo tanks have a total capacity of 1638 m^3 and are subdivided into three tanks for sulphuric acid with a capacity of 1372 tons and two tanks for caustic soda with a capacity of 1346 tons.

An engine room and pumproom have been arranged forward, within the latter a 10 in pump for the handling of sulphuric acid and a similar pump for the handling of the caustic soda. The pumps, which are of Bornemann manufacture, are driven by a DAF air-cooled Diesel engine type D.K.1160A, developing 165 hp at 1900 rev/min. The engine is electrically starting.

When the ship is proceeding from one point to another, the pumps will not be necessary, which means that the DAF Diesel engine is available to drive the Schottel retractable auxiliary steering propeller. This propeller is of the S 100 ZSV type, built by Schottel Nederland N.V. The Schottel propeller is of special significance because *Sieglind* forms part of a push-tow and is the foremost part of this tow. For this purpose the vessel is also provided with the necessary navigational aids.

For the heating of the caustic soda cargo, heating coils have been fitted in the tanks in which this cargo is carried. Furthermore a 'Kontatomat' with pump has been installed in the engine room.

Two Armstrong Siddeley auxiliary engines, type A.S.J.-2, of 20 hp at 1800 rev/min are also installed in the engine room. They drive the ballast pump and the windlass, and are suitable for general purposes. The quarters for the crew of the barge and the store room are arranged aft.—*Holland Shipbuilding, October 1969, Vol. 18, p. 110.*

Seagoing Barge for Carrying Dredge

Many contractors face the problem of how to transport dredges and auxiliary equipment to overseas project sites safely and quickly. When it is difficult or impossible to dismantle the dredge for transport on a seagoing ship, the contractor must consider the possibility of towing the dredge on its own bottom. This can only be done, however, if the dredger is sufficiently seaworthy, and will be issued with a certificate for the voyage.

Ulrich Harms Co. has developed a new line of large seagoing barges called *Mulus,* which can carry dedgers and big structural parts, such as cranes and bridge sections, and be towed directly to the work site.

The barges have the following dimensions: length 76 m, width 24 m, depth 4·8 m. Each unit has an uninterrupted deck surface of about 1800 m^2. The draft of the *Mulus* without cargo is only 0·6 m. Maximum loading capacity is approximately 5000 tons, and draft is only 1·30 m when the barge has a 1000 ton load.

The problem of how to get bulky cargo on and off the decks was solved by dividing the barge into fifteen separate compartments so that it can be flooded to a depth which will allow the floating cargo to be towed over the deck of the lowered barge. To raise the barge, compressed air is forced into the compartments, forcing the water out. Before the raising operation is begun, it is important to assure that the cargo is touching the deck at exactly the intended travel position. To accomplish this, the deck is equipped with guiding supports, which are also used to securely lash the cargo.

This operation appears to be simple in principle, but there is one major difficulty: as soon as the deck surface submerges below water-level, the hull loses its stability. To avoid this, the *Mulus* is flooded only one end, and to a depth commensurate with the depth of the floating cargo. Depending on the object, a water-depth of 8–12 m is required for the flooding process. Currents have to be taken into consideration especially in tidal waters where the currents change direction by 180 degrees. Furthermore it is necessary to have a pressure inside the empty tanks which is equivalent to the outside pressure of the corresponding water-depth (10 m = 1 atu). As soons as the deck-surface has been raised above water-level after floating on the objects the seaworthy lashing can be started. Discharging is carried out in the reverse manner.—*World Dredging and Marine Construction, November 1969, Vol. 5, pp. 18; 20.*

Twin Deck Solvents Tanker

A solvents tanker featuring twin decks has been built by Smith's Dock, a member of the Swan Hunter Group, for Sameiet Seafalcon, Norway. The ship is managed by Graff-Wang and Evjen of Oslo. Of interest regarding *Seafalcon's* construction is the arrangement of her main cargo tanks which are situated under the lower deck with trunks leading up to the weather deck. This arrangement has been adopted to comply with the owners' requirement to restrict the gross tonnage to under 2000.

Principal particulars are:

Length, o.a.	395 ft $8\frac{1}{4}$ in
Length, b.p.	328 ft 1 in
Breadth, moulded	50 ft 10 in
Depth, moulded to shelter deck ...	26 ft 7 in
Depth, moulded to main deck ...	19 ft $0\frac{3}{8}$ in
Draught, summer	18 ft $10\frac{3}{8}$ in
Deadweight on 18 ft—$10\frac{3}{8}$ in draught	4450 tons
Displacement on 18 ft—$10\frac{3}{8}$ ft draught	6275 tons
Gross tonnage	1999·71
Net tonnage	1322·74
Cargo tanks capacity (100 per cent)	186 229 ft^3
Service speed (approximately) ...	13·0 knots

Seafalcon is powered by a two-stroke, 16-cylinder Vee-type Diesel engine of Smit-Bolnes make. The output from this V316D type engine, which has a bore of 300 mm and stroke of 350 mm, is 3200 bhp at 300 rev/min m.c.r. This has a direct drive to the four-bladed stainless steel controllable pitch propeller supplied by Lips. The propeller diameter is 2600 mm with a hub diameter of 790 mm. Two electrically-driven pumps, one on standby, feed oil to the pitch change mechanism.

The propelling machinery can be pneumatically controlled from the bridge, by means of a single lever controlling both the engine speed and propeller pitch. In the engine room the propeller pitch is controlled by push buttons directly operating electric solenoid valves in the pitch control oil system, the engine speed being controlled by a lever fitted at the forward end of the engine.

Steam is generated in two Henschel vertical packaged fully automatic water tube boilers designed for a working pressure of 12 kg/cm². Each boiler provides a normal steam output of 1800 kg/h, with a maximum output of 2045 kg/h.

Hydraulic power for cargo pumping is provided by three hydraulic pressure oil sets. Each set comprises four pumps of Vickers make, driven through a flexible coupling, a common reduction gear-box and disc type friction clutch driven from the forward end of the Diesel alternator sets. Each pair of pumps is capable of supplying the necessary power to drive one main cargo pump giving a total of six cargo pumps if the three power units are in use. The hydraulic power units can be controlled locally or from a pump room situated at the poop front on the weather deck.

The cargo pumps are hydraulic motor-driven submersible deepwell pumps of the non-self priming, centrifugal type.

One cargo pump is fitted to each tank and the cargo piping arranged so that each tank can be completely isolated from every other if required, with independent filling and discharge connexions. A separate filling drop line is arranged at each tank to avoid the necessity for filling through the pumps. Four separate deck cross-over lines are provided, one for each pair of outboard cargo tanks, with line-blend valve isolation of each and a common connexion to one of the midship cross-over lines.—*Shipping World and Shipbuilder, October 1969, Vol. 162, pp. 1462–1465.*

Studies on Rudders with a Flap

An account is given of model tests, in a recirculating water-channel, on rudders of NACA 0020 section fitted with a trailing-edge flap. Three sizes of flap were used in the tests, the respective ratios (i.e. ratios of flap chord to total chord) being 0·5, 0·25 and 0·165. The models had a chord length of 200 mm and a maximum thickness of 40 mm; the aspect ratio was 1. The tests were made at a velocity of 0·69 metres/sec; the test section of the channel measured 1·2 by 0·8 m. Some tests were made in "going astern" conditions.

The test equipment is described, and the results are presented and discussed. It was found that:

i) the lift coefficient varied almost linearly with the flap angle;
ii) separation depended only on the rudder angle and was independent of flap angle;
iii) the changes in the position of the centre of pressure were a function of the ratio between the rudder angle and the flap angle;
iv) the torque on the rudder stock could be decreased by suitable selection of stock position;
v) the torque on the flap stock increased with flap area;
vi) the best performance was given by the rudder with a flap ratio of 0·25 and an angle ratio (i.e. flap-angle/rudder) of 2, and was twice as good as that of a conventional rudder.

—*Kato, H. and Motora, S., Journal Society Naval Architects, Japan, December 1968, Vol. 124, pp. 93–104; Journal of Abstracts of The British Ship Research Association, July, 1969, Vol. 24, Abstract No. 27 630.*

Powerful American Tugs for Netherlands Antilles

Gulfport Shipbuilding Corporation, Port Arthur, Tex., has recently delivered two harbour tugs to Lago Oil & Transport Co., Netherlands Antilles. They will be operated primarily at Aruba for the handling and docking of tankers. They also will serve as fireboats should the need occur. They are of 2800 hp each, single-screw design, with a length of 105 ft overall, a beam of 32 ft and depth of 17 ft.

A design output of 2800 shp at 900 rev/min is furnished by a single General Motors EMD turbocharged and after-cooled 16-645-E5 engine, transmitted through a Falk Model 4048 MRV reveres reduction gear with a ratio of 6·038 to 1·0 ahead and astern with air-controlled Falk airflex slip clutches. The entire machinery plant is controlled from the pilot house, including start-up and shut-down, and a monitoring and alarm system provides the necessary safety under observation of the pilot.

The 11 ft 3 in diameter three-bladed stainless-steel solid type propeller turns in a fixed modified type Kort nozzle at 149 rev/min. This specially designed combination of hull configuration, engine, propeller and Kort nozzle gives these tugs a tested bollard pull of 91 000 lb ahead or 79 000 lb astern.

An unusual feature on these boats is the three-rudder steering control system furnished by Vickers-Sperry. It consists of two independent steering gears with a common hydraulic supply. One steering gear system will operate the main steering rudder while the other will control the two flanking or backing rudders. This will give the tugs maximum control capability in both the ahead or astern manoeuvring positions.

The "Tugmonitor" automation system on these two tugs provides complete wheel-house control, without attendance in the engine room, of the main engines, generator sets, steering, pumps, foam, fire and bilge pumps. All components of these systems are monitored and any off-limit readings trigger both a visual and audible alarm system in the pilot house and engine room.

The deck equipment consists of two pairs of 16 in Bitts, one on the forward deck and one located just aft of the main house. Also on the forward deck is a 26 in hydraulic capstan with a 7500 lb line pull at 65 ft/min.—*Marine Engineering/Log, November 1969, Vol. 74, p. 88.*

Propeller Blade Trimming

At certain speeds the 1000 ton motor tanker *Pass of Glenogle* experienced vibration and in order to overcome this it was considered necessary to trim off the tips of the three-bladed 7 ft (2·13 m) dia bronze propeller. This task was carried out by three divers of the Underwater Maintenance Company, who are part of International Red Hand Marine Coatings, whilst the vessel was moored to a jetty alongside Husbands Shipyards at Southampton.

With the aid of underwater lighting and abrasive cutting discs the team began the difficult and stamina-demanding operation at 18.00 on a Saturday and completed the task by 07.00 the next morning. It entailed cutting three inches off the end of each of the 1¼ in (31·7 mm) thick blades, at the same time accurately following the curve of the blade tips. The cutting was carried out with the aid of templates made up by the Underwater Maintenance Company beforehand from the original propeller drawing.—*Shipbuilding International, December 1969, Vol. 12. pp. 26–27.*

New Vessel for Zeeland-Jutland Service

With the recent delivery of the 2450 gross ton *Morten Mols* from Aalborg Vaerft A/S, Denmark, the Mols Line now has its full complement of four roll-on/roll-off ferries on the increasingly popular Ebeltoft-Sjaellands Odde service which provides the shortest sea route between Zeeland and Jutland.

Morten Mols is a sistership to *Mikkel Mols*. Basically these two vessels are similar to the two earlier ferries *Maren Mols* and *Mette Mols* which inaugurated the service in 1966; but with increased passenger facilities and higher engine outputs. The particularly interesting machinery arrangement comprises four 14-cylinder B & W four-stroke non-reversible trunk-piston turbocharged engines of the 26 MTBF-40V design. These are each rated at 2520 bhp in continuous service at 600 rev/min, but the maximum continuous output is 2770 bhp at 620 rev/min. Each pair of engines is coupled through flexible couplings to a Renk double-reduction gearbox with a reduction ratio of 2·22:1, and thence to a Kamewa 4-bladed 2700 mm diameter c.p. propeller, giving the ship a service speed of 19·5 knots.

The four vessels, each with a capacity of 110/135 vehicles and 800 passengers, operate an hourly schedule in the summer season, thus allowing a turn-round time of only 30 minutes between the 90-minute sea passages. For this reason the drive-through roll-on/roll-off ferries, recently inspected by a member of our technical staff, are provided with highly efficient manoeuvring facilities for berthing at terminals specially constructed by the Mols Line; each vessel is provided with two spade rudders, a bow rudder and an 800 hp KaMeWa bow thruster. An additional navigation bridge is sited aft for berthing stern-in at Sj. Odde, and there are controls for both c.p. propellers and the bow thruster on each wing of both the forward and aft bridges.—*The Motor Ship, November 1969, Vol. 50, p. 348.*

Boiler Tube Expanding

The time involved in effecting a ship tube fitting job which would normally have to be carried out by hand was reduced by over 80 per cent by adaption of a pneumatically-operated tool.

The actual application was the fitting of 930 thimble tubes in the Nelvin exhaust gas boiler of M.V. *Glaneley* at the T. W. Greenwell Dry Docks of The Doxford and Sunderland Shipbuilding and Engineering Co. Ltd. The tubes were 17 in long by $3\frac{1}{2}$ in outside diameter, 8 gauge. As the working space in the boiler was only 15 in wide, normal tube expanding equipment would be difficult to use and, furthermore, expanding the tubes and belling by hand takes about 45 minutes for each tube. This process was reduced to about six minutes per tube using a Thor 3/RAW/40 nutsetter fitted with an adaptor hexagonal drive, to a $\frac{1}{2}$ in square drive and $\frac{7}{8}$ in (22 mm) square socket. The tool was used to carry out the initial expansion and each tube finished with six turns by hand to bell the tube end—a total saving of some 600 man hours.—*Shipbuilding International, December 1969, Vol. 12, pp. 28–29.*

Steam Turbine Tanker

The steam turbine tanker *Esso Cambria,* built by Verolme Verenigde Scheepswerven, Rotterdam, for the Esso Petroleum Company Ltd., London, is the first of three sisterships being built by Verolme for these owners.

The ship is a single-deck vessel without poop and forecastle. She is provided with two longitudinal bulkheads. The propulsion machinery, which is placed aft, consists of a Verolme-General Electric steam turbine installation. Principal particulars are:

Length, o.a.	347·80 m	1141 ft 1 in
Length, b.p.	392·20 m	1080 ft
Breadth, moulded ...	51·80 m	170 ft
Depth, moulded ...	25·60 m	84 ft
Draught, maximum ...	19·96 m	65 ft 6 in
Deadweight capacity	approximately 255 000 tons	
Cargo capacity ...	303 000 m^3	1 070 000 ft^3
Speed	approximately 16 knots	

Propulsion of Esso Cambria is by a Verolme General Electric MST-14 'package type' cross compound steam turbine installation developing approximately 32 000 shp. There is no reheat system. The hp turbine has eight stages with Curtiss wheel; the LP turbine, also has eight stages. Each of the two turbines delivers about half of the total power.

The principal design characteristics of the installation are as follows:

Steam temperature at HP turbine inlet 513°C.
Steam pressure at HP turbine inlet 64 ata.
Steam temperature at LP turbine inlet 247°C.
Steam pressure at LP turbine inlet 6·3 ata.
Condenser pressure 28·5 in Hg.
Output 31 550 shp.
Propeller speed 80 rev/min.

The speed of the HP turbine is 6652 rev/min, while that of the LP turbine is 3442 rev/min. These speeds are reduced to that of the propeller shaft through a General Electric articulated locked train gear.

Steam is provided by one Babcock & Wilcox radiation boiler without reheat. The boiler was built under licence by Verolme Machinefabriek and is provided with a Ljungström air-heater for the combustion gases. The boiler is roof fired and equipped with four Babcock oil burners with a wide range and an automatic soot-blasting installation. The principal characteristics of the boiler are as follows:

Capacity (superheat)	104 tons/h
Steam pressure after superheater outlet	64 ata
Steam temperature after superheater outlet	513°C
Feedwater temperature	222°C
Exhaust-gas temperature	327°C

The feedwater system of the main boiler consists of one LP feed heater, one deaerator and two HP feedwater heaters.

For take-home purposes the ship is fitted with an oil-fired auxiliary boiler of Verolme-IJsselmonde design and manufacture. Its capacity is 20 tons/hr at a steam pressure of approximately 17·5 ata and a steam temperature of 208°C.—*Holland Shipbuilding, November 1969, Vol. 18, pp. 36–37; 40.*

Comparison of Propulsion Plants with Two-Stroke Cross-Head Engines and with Geared Medium-Speed Engines

With the trend towards tankers and bulk carriers of increasing size and container vessels with increasing speeds, propulsion plants with unit outputs between 30 000 and 60 000 shp are now demanded from the manufacturers of such plants. Until recently, in merchant shipping the steam turbine was used exclusively for this output range. The fact that the steam turbine no longer monopolizes this range is due to direct-coupled super-large bore Diesels, and the progress in the development of high-output medium-speed propulsion machinery.

In the case of tankers, it is no problem to accommodate very large propellers and, consequently, the shipbuilders prefer shaft speeds between 80 and 100 rev/min. However, on account of the carrying capacity of such ships, the engine room must be as short as possible. A further factor of utmost importance in tankers is the selection of the drive system for the cargo oil pumps which, in the case of giant ships, may require a total of 16 000 hp or even more.

In a tanker propulsion plant incorporating two reversible main engines type V9V52/55, each having a maximum continous rating of 18 000 bhp at 430 rev/min the two main engines drive the propeller at a speed of 80 rev/min through a twin-input reduction gear. Flexible couplings are arranged between the engines and the reduction gear and there are clutches in the gear for manoeuvring. Propulsion plants of this size are suitable for tankers of between 175 000 and 200 000 dwt. In a tanker propulsion plant incorporating the M.A.N. super-large bore K6SZ105/180 engine, the two-stroke crosshead engine, directly coupled to the propeller shaft develops 24 000 bhp at 106 rev/min. Admittedly at 106 shaft revolutions the propeller for an 80 000 dwt tanker cannot be designed for optimum efficiency. Compared with geared propulsion plants, the shaft speed of which is about 80 rev/min in the case of tankers of this size, efficiency of a direct-coupled propeller should be about 6 per cent lower. However, the propulsion system incorporating the two-stroke crosshead engine has the advantage of its robustness and relatively simple arrangement. Moreover, the two-stroke crosshead engine can be accommodated in a very short engine room, which is of particular advantage in the case of tankers.—*Dipper, H., Shipbuilding and Shipping Record, 19th and 26th December 1969, Vol. 114, pp. 30; 35.*

Heavy Derrick

A high-speed multi-purpose cargo vessel to be provided with an extensive cargo handling system, the 14 000 dwt *Lübeck*, was launched at the yard of Lübecker Flender-Werke AG, for the German owner, Lübeck-Linie AG. This 148·5 m long vessel, powered by a K8Z 70/120E M.A.N. engine capable of developing 11 200 bhp at 140 rev/min, can attain a speed of 19 k.

With a total hold capacity of 22 400 m³ including about 570 m³ of refrigerated space and 1100 m³ of tank space, *Lübeck* is designed to carry general, bulk, refrigerated, containerized liquid and heavy cargoes. The cargo-handling system will comprize 18 light derricks and two heavy-lift tilting derricks of the builder's own design. These heavy derricks, with a deadweight capacity of 60 and 35 tons respectively, and each capable of serving two hatches, are of the System Flender type and will enable the vessel to handle her own container loads throughout.

The System Flender heavy derrick arrangement includes two king posts, the upper ends of which are rigidly interconnected, with a self-positioning pulley-block located on the underside of the cross-girder. The derrick head span block and the upper cargo block are constructionally combined in the multi-sheaved derrick head. Both the swivel pin of the self-positioning pulley-block and the gooseneck pin at the derrick heel are arranged in the same axial plane so that the derrick can be tilted between the hatch areas forward of, and abaft the king posts. This is done, with the aid of guys, by topping to an angle of 85° and using the trim of the vessel, or an auxiliary guy attached to the cargo hook, to dip the derrick head under the cross girder. Guys are attached on either side of the ship in positions athwart the king posts and in line with the hatch corners opposite the derrick. Before tilting, the guy-tackles are removed and the guy-pendants lengthened by using auxiliary wires.—*The Motor Ship, December 1969, Vol. 50, p. 418.*

Giant Ocean Going Tug

An order for a 14 000 bhp tug has been placed by the United Towing Co. Ltd. with Robb Caledon Shipbuilders Ltd., of Leith. This tug will be the most powerful ever built for British owners, the most powerful at present being the 5000 bhp United tug *Englishman*.

The new vessel is intended to have an outstanding performance and, with a bollard pull of over 100 tons, it will be well capable of handling 300 000-ton super tankers at a towing speed of 7 knots.

Worldwide operation is envisaged by the provision of extra large bunker capacity. The accommodation has been specifically designed for use in both tropical and arctic conditions.

The tug has been designed by Burness, Corlett and Partners Ltd. and incorporates their "Towmaster" nozzle and rudder features. The two powerful Diesel engines, through a twin-input, single-output gear-box, driving a controllable-pitch propeller, will ensure an anticipated free-running speed of 19 knots. When operating on only one engine the new tug will still have the capability of most ocean-going tugs now in service.

With an overall length of 263 ft (about 80 m) the lines of the vessel have been designed to provide a high degree of sea-keeping and dryness qualities. Deck machinery will include large winches capable of handling the 8 in (203 mm) wire rope which will be used for towing.—*Ship and Boat International, October 1969, Vol. 22, p. 43.*

41 700 dwt Bulk Carrier

The first of two medium-sized bulk carriers, the 41 700 dwt *Grecian Legend*, designed to carry a variety of cargoes including all types of grain, chemicals, ore and timber, was delivered from Scott Lithgow's Cartsburn yard in Greenock, to her owners Goulandris Bros., of London. This seven-hold vessel with machinery aft, is built to Lloyd's Classification requirements and strengthened for ore cargoes. The holds are of the self-trimming type with upper and lower wing tanks arranged for ballast, the upper wing tanks also being available for grain cargoes, when no feeders or additional bulkheads are required.

Principal particulars of *Grecian Legend* are as follows:

Length, o.a.	660 ft 0 in
Length, b.p.	620 ft 0 in
Breadth	92 ft 0 in
Depth	51 ft 3 in
Summer draught	38 ft 1 in
Corresponding deadweight ...	41 700 tons
Block coefficient	0·806
Gross register	22 998 tons
Net register	17 041 tons
Trial speed	16·8 knots
Complement	41

As the result of B.S.R.A. model tests the hull form has been fashioned with a modern cruiser stern and a new type of bulbous bow having a relatively flat curve at the forward perpendicular surface, which is expected to be most effective at about 80 per cent full engine power.

Propulsion is by a Scott-Sulzer 6RD90 turbocharged two-stroke engine with a maximum output of 14 150 bhp at 122 rev/min, and designed to burn heavy oil of viscosity up to 3500 sec. Red. 1 at 100°F.

The two turbo-blowers are of the Brown Boveri VTR 630 type, being fitted with ball and roller bearings and integral lubricating oil pumps. Lanchester balancer gear is installed at each end of the main engine to reduce the effect of hull vibration. The thrust shaft, intermediate and propeller shafts have been constructed with torsional strengths five to ten per cent above classification requirements and the four-bladed propeller, manufactured of Noveston alloy by Stone Manganese Marine Ltd., with a tip diameter of 19 ft 9 in, is secured by a low carbon steel Pilgrim nut.

Auxiliary power is provided by three 425 kW Laurence Scott alternators, driven by Ruston 6AP2Z pressure-charged and intercooled Diesel engines and delivering three-phase current at 60 cycles and 440 volts.—*The Motor Ship, December 1969, Vol. 50, pp. 402.*

New Approach to Maintenance of Large-Bore, Two-Stroke Engines

The developments which have taken place in the design of large-bore Diesel engines over the last few years are reviewed especially with regard to increased output and operational economy coupled with reliability. Although, during the same period, efforts have been made to improve steam-turbine propulsion plants (particularly in respect of fuel consumption), it is considered that the Diesel engine is still more economical for driving large tankers exceeding 200 000 dwt. A statistical survey (carried out by a Scandinavian shipowner) of a number of tankers exceeding 50 000 dwt has compared the "off-hire" periods of Diesel and turbine-driven tankers; the results appear to indicate that the Diesel also provides greater operational reliability.

The general problems related to Diesel-engine development and up-rating are discussed. Scavenging systems suitable for a high degree of supercharging are compared, and the reasons for adopting the simpler loop system in preference to a uniflow system are stated. Stationary and dynamic model tests were carried out to assess scavenging efficiency. Although the degree of turbocharging has risen from 25 per cent in 1956 to 115 per cent (corresponding to a mean effective pressure of 12·45 kg/cm², and an exhaust gas temperature before turbine of 435°C) in the latest super-large-bore KZ 105/180 M.A.N. engine there is still a good thermal safety margin.

To provide owners with engines of optimum propulsion efficiency to suit their individual needs, great attention has been paid to engine speed when determining the dimensions of cylinders, and, at the same time, power/weight ratios have been reduced by increasing the mean effective pressures. A table compares the most important design data of the largest KZ engine available in 1953 with the corresponding data of the present largest engine; the most significant points are the reduction of the specific weight from 60 kg/bhp in the earlier engine to 34 kg/bhp in the present engine, and a reduction in overall length.—*Paper by Scobel, H. H. and Richter, J. H., presented at the Diamond Jubilee International Meeting of the Society of Naval Architects and Marine Engineers, 18–22 June 1968; paper No. 18; Journal of Abstracts of the British Ship Research Association, August 1969, Vol. 24, Abstract No. 27 843.*

Use of Submersible in Helping to Salvage Tug from 670 ft Depth

When a 95-ton, 51 ft tug named *Emerald Straits* went down in a storm on 19th April while towing a scow in Howe Strait about 30 miles northeast of Vancouver, the Canadian Government decided to try to salvage the vessel intact and conduct a full-scale investigation. Three crewmen were lost in the accident, a fourth managed to swim safely to shore.

The Department of Transport awarded a contract in June to International Hydrodynamics Co. Ltd., a small Vancouver firm, to salvage *Emerald Straits* from 670 ft of water. If successful, the firm was to receive $110 000 but nothing if it failed.

It was no ordinary salvage job, however; it is believed to be the first attempt ever to bring up an object that large from 670 ft. Moreover, the tug had to be recovered in a horizontal and relatively undisturbed condition. The investigative job would be nearly impossible if the tug were badly damaged or equipment was shifted about during salvage.

Shortly after the accident, International Hydrodynamics had used its *Pisces I* submersible to locate and photograph the tug on the bottom. With the aid of the sole survivor, the tug was found lying in an upright position about 1000 yds from shore on a smooth but muddy bottom.

International Hydrodynamics purchased a 100 by 42 ft salvage scow for the job and outfitted it with two anchor winches, a logging winch, a new power generator and other standard salvage gear. The scow and a support barge for *Pisces I* were towed to the site on 27th June. A four-point moor was used to anchor the scow over the wreck site. *Pisces* support barge was moored on the leeward side of Anvil Islands, about two miles away.

Salvage work got underway on 28th June when *Pisces* again dived to the sunken bulk and spent nearly six hours moving around the tug, photographing it from every angle with still, motion picture and TV cameras. Since no drawings existed of *Emerald Straits*, the pictures were needed to plan the salvage.

After the photos of *Emerald Straits* were examined, a salvage plan was devized. *Pisces* would be used to cut two anchor chains on the tug's bow at the anchor windlass. The chains and anchors, it was thought would then slide clear of the hawse pipe, and toggle bars could be lowered from the surface and guided into the hawse pipe with the manipulator on *Pisces*. Once through, the toggle bars would be pulled tight against the side of the steel-hulled tug and serve as the lift points for the bow. A sling would then be passed from the bow to the after end of the ship, providing both fore and aft lines, for the required horizontal lift. The salvage team, on 13th July, with the sub in position to observe, began lifting the bow up with the $\frac{3}{4}$ in steel core load line attached to the toggle. The bow raised 10 ft out of the mud and the toggle held.

The next step was to fashion the sling and move it into position. The wire rope sling was large enough to be passed from the forward end of the tug to the stern quarter. Timber, which would splinter when actual lift began, was used as a spreader. With *Pisces* guiding the positioning by surface crews and clearing the wire rope from fenders and other obstructions as it passed toward the stern, the sling was positioned on 15th July.

All lifts were made in 10 ft increments. First the bow was raised 10 ft, then the stern was pulled up 10 ft, and so it went. By the morning of 22nd July, the tug was 170 ft off the bottom and operations ceased in order to revamp the deck gear. The next day, a second shift was added. By working two shifts around the clock, *Emerald Straits* was raised to within 76 ft of the surface by 10 a.m. on 24th July.

At this point, preventer chains were attached, the scow pulled up its anchor and headed toward Porto Beach, about 2 miles away, where the sandy bottom slopes gently. There the lift lines were passed from the scow to the derrick barge. At 9.30 a.m. on 24th July, *Emerald Straits* was lifted to the surface.—*Under Sea Technology, October 1969, Vol. 10, pp. 41–42.*

Submarine Rescue Vessel

The 10-ton, triple-sphere pressure hull of the U.S. Navy's deep submergence rescue vessel (DSRV) was recently installed in the vessel's torpedo-shaped outer hull at Lockheed Missiles and Space Company's Sunnyvale, California, facility. With the pressure hull now in place, the major portion of the rescue submarine construction has now been completed. The vessel will soon begin testing of all subsystems.

Made of high strength HY-140 steel, with ·738 in thick walls, the pressure hull will operate at depths to 5000 ft where the pressure is 2225 lb/in². The DSRV is the first deep submersible to use HY-140 as the pressure hull material.

The pressure hull, in the shape of three interconnecting spheres, rests inside a 50 ft outer hull made of glass reinforced plastic. It is held in place by rings of titanium.

The DSRV is designed to rescue the crews from submarines which lie disabled on the ocean floor. When it becomes operational in 1970, it will be able to put the rescue vessel on the site of a submarine disaster anywhere in the world within 24 hours.

By sealing its transfer skirt over the escape hatch of a downed submarine, the DSRV can take aboard up to 24 rescuees at a time. Inside the three $7\frac{1}{2}$ ft diameter pressure hull spheres, the DSRV crew and submarine rescuees will be protected from both cold and pressure.—*Marine Engineering/Log, November 1969, Vol. 74, p. 90.*

Zinc Dust Paints as a Prefabrication Primer

The one problem with zinc epoxy primer is the weldability of high tensile strength steel using a low hydrogen type of electrode. There is a tendency for blow holes to form in fillet welds and for hydrogen to diffuse into the deposited metal. This problem is important, so a new type of shop primer has been investigated during the last two years and this is soon to be tested in production.

A general comparison of the properties of four kinds of shop primers, a long life wash primer, an epoxy base zinc free primer, an inorganic zinc silicate primer and a zinc epoxy primer has been made. The grade of surface preparation to be used is decided mainly by the mechanism of adhesion between paint media and the steel surfaces as well as the degree to which the anti-corrosive properties of the steel surface depend on the type of surface preparation used.

The effects of shop primers on flame cutting, such as cutting speed, roughness of cut edge, slag formation, zinc fume etc., have been investigated and from the results of experiments it is clear that metal pigment affects flame cutting principally according to its melting point. Zinc, with a melting point of 420°C is much better than aluminium at 660°C. Heat absorbed during oxidation of aluminium has an important effect in slowing the cutting speed of steel coated with aluminium. On the other hand, slag is more easily removed when the primer contains aluminium because it does not stick directly to steel covered by an aluminium film. The effects of media on gas cutting are smaller than those of the pigment: zinc epoxy primer and inorganic zinc silicate are similar in this respect and, in fact, the four kinds of shop primer tests show a basically similar effect. When a small size nozzle is used to flame cut thin steel plate, a 10 per cent reduction in cutting speed is necessary compared with bare steel.

The following counter-measures are being used:

1) to maintain cutting speed on thin plate a larger nozzle is used;
2) clogging is prevented by inclining the tip towards the running direction and the clearance between the nozzle and steel is maintained at about 5 mm;
3) the film thickness of primer is maintained below 25 microns and in general, thickness is kept between 13 and 18 microns.

The following three points must be improved to cope with expected advances in construction techniques:

i) anti-corrosion properties
At present protection for 70 days from the start of fabrication until assembly is satisfactory because the regular painting is carried out before erection, but with the development of automatic painting machines for whole plates, regular painting will not be carried out until after erection is complete. In this case, the shop primer will need to protect the steel completely for at least six months.

ii) primer for electronic print marking methods
Although the use of numerically controlled gas cutting will substantially increase, there will be a useful range in which electronic print marking can be used. The shop primer should be able to be used for both these methods and zinc epoxy primer fills these requirements.

iii) effect on cutting and welding
In the near future, high speed and fully automatic gas cutting and welding machines or plasma cutters and line welders will be used on a large scale. The shop primers must be improved so that they do not interfere with cutting and welding when carried out at greatly increased speeds.

Sakami Shiro, Shipping World and Shipbuilder, January 1970, Vol. 163, pp. 161; 163; 165. (The original paper was presented at a symposium held by the Zinc Development Association, London.)

North West Passage

The N.W. Passage has long been a dream of navigators, and the discovery of oil on Alaska's north shore has revived interest in such a route for the conveyance of oil to the east coast of America and Europe. As reported widely in the press, the Humble Oil Company and its associates determined to test the feasibility of using large tankers in this trade, and acquired and altered the 115 000 dwt steam tanker *Manhattan* for this purpose, providing her with, among other modifications, an ice-breaking bow. She has recently completed a first experimental round-trip from New York successfully, and dealt with ice floes up to 15 ft thick as well as occasional ice ridges up to 40 ft. The Humble Oil Company's project manager stated on her return that, operationally and technically, the journey had "proved the passage to be operable". The analysis of the data will not be available until the Spring, and much thought will be necessary before any ships can be built specially for this trade. Among preliminary findings are that the 42 000 hp installed in the vessel is not nearly enough for dealing with heavy ice, and that the hull shapes will have to be very different from that of *Manhattan*.

So far as U.S. shipbuilding yards are concerned, the successful exploitation of this route by large tankers could mean a sizable new building programme, since such ships would be trading between two U.S. ports and accordingly would have to be built in U.S. yards. Of course, the opening of a N.W. Passage for general commercial use would not be confined to the Alaska-U.S. East Coast oil trade, but would attract shipowners operating between Europe and the Far East, since it would reduce the ocean passages by about 50 per cent compared with present routes.—*Todd, F. H., Shipbuilding International, January 1970, Vol. 12, pp. 16–17.*

Manoeuvrability Trials on Tanker Magdala: Effect of Depth

In this paper, further information is given on the manoeuvring trials of *Magdala*, and the results are presented and discussed. These trials, which were conducted in accordance with the recommendations made by the Tenth ITTC, allowed a systematic study of the effect of water depth on manoeuvrability. They were carried out at various depths, and included measurements of the rate of headchange, turning-circle tests, Kempf (zig-zag) tests, and Dieudonné (spiral) tests. The respective procedures are described in detail in appendices to the paper.

Some general conclusions are drawn from the results. In trials at a depth of 40 m, no effect of depth was detectable. At depths between about 35 and 30 m, the transfer distances and turning-circle diameters increased by about 3 per cent. The effect of depth on the rate of head change during a steady turn is very slight. In the Kempf test, in certain weather and other conditions the effect of water depth of secondary importance and is very small indeed. Even where the water-depth-draught ratio was only about 1·2, the ship remained manoeuvrable. The effect of water depth was found to be much less than is generally supposed; it is possible that a scale effect might exist.—*Nizery, B. and Page, J. P., Ass. Tech. Marit. paper presented at 1969 Meeting Journal of Abstracts of the British Ship Research Association, August 1969, Vol. 24, Abstract No. 27 768.*

Stopping Trials on Magdala: Effect of Depth

The Institut de Recherches de la Construction Navale has, for some years, been making a systematic full-scale study of the stopping of large tankers. Until the sea trials of *Magdala* these studies had not included trials in shallow water. Model tests to evaluate the effect of water depth on stopping have been misleading.

The present paper gives an account of the *Magdala* stopping trials. These trials, on full-load draught in various water-depths, consisted of tests in stopping from maximum speed and from lower speeds, and "J" tests in which the braking effect of the rubber is utilized. The results are presented in tables, curves, and diagrams, and are discussed in detail.

It is concluded that, as long as water-depth is not less than 1·3 times the draught, the stopping characteristics of a ship of this class do not differ from those in deep water. Water depths below this value appear to have a slightly favourable effect on stopping, but this effect is smaller than that of the external factors which may influence any stopping trials, and it should not be taken into account when manoeuvring in shallow waters. The simplicity of these conclusions does not detract from the value of the trials, since such results were expected and are reassuring for those concerned in the development of very large tankers. It is considered desirable that the study should be extended by taking every opportunity to carry out tests at the even smaller water-depths which may be encountered in the vicinity of ports (the smallest depth in the present stopping tests was 24·5m).—*Jourdain, M. and Page, J. P., Asst. Techn. Marit. paper presented at 1969 Meeting; Journal of Abstracts of The British Ship Research Association, August 1969, Vol. 24, Abstract No. 27 769.*

Roll-on/Roll-off Ferry Lengthened

One of the few ship-lengthening operations yet performed on a passenger ferry was completed recently when the 3547 gross ton roll-on/roll-off ferry *Queen of Esquimalt*, owned by British Columbia Ferries Ltd., was lengthened by the addition of an 84 ft mid-section at a cost of $2·2 million. The work was carried out by Burrard Dry Dock Co. Ltd., of North Vancouver, within the specified completion time of three months, after which the ferry resumed its previous service between Vancouver Island and the mainland.

Work commenced on the prefabricated mid-section—which was five decks deep—approximately one month before the ferry was released from service during the slack season. Preliminary work on the ferry before the cutting operation took place including stripping the vessel's existing galley and restaurant and sealing off pipes and electrical wires. The superstructure was then sliced through, using oxy-acetylene cutters, and the vessel dry-docked in a floating dock. The final cutting operation then took place; this was forward of a water-tight bulkhead, the object being to flood the dock after the cut, float off the aft section and leave the forward section in position in the dock.

The new length of the vessel is 426 ft which permits 192 cars to be carried.—*The Motor Ship, December 1969, Vol. 50, p. 414.*

Oil Tanker with Electronic Computer

Ishikawajima-Harima Heavy Industries Company, have received an order for a 138 000 dwt oil tanker from the Sanko Steamship Co. Ltd.

The tanker will be equipped with a Tosbac 3000 electronic computer which Toshiba (Tokyo Shibaura Electric Co. Ltd.) will manufacture, to make her the first full-fledged automatic ship. She is regarded in Japan as a test ship for the coming age of unmanned vessels.

The electronic computer will give an estimation of the ship's course (bearing on the ship's position being obtained from artificial satellites, combined with electronic equipments); it will provide information for relating to deck routine work, such as figuring out the best loading distribution and optimum control for cargo loading and discharging; the observation and diagnosis of trouble in the refrigerating machinery; for main engine routine work, such as data logging, the detection and correction of any trouble with the main engine; and the torque control of the main engine; a data logging programme will be provided; and it is understood that the computer will also be used to diagnose and give information on the treatment of simple illnesses of anyone on board.

As the computer control system will be experimental, the vessel will be equipped with conventional equipment. The tanker will have a crew of 32 but will be run by fewer than 15 men, it is hoped.—*Shipbuilding International, December 1969, Vol. 12, p. 19.*

Fatigue Strength of Large Single Pinned and Double Pinned Connexions Made From Alloy Steel FV520B

Fatigue tests have been conducted on single-pinned and double-pinned lugs made from alloy steel FV520B and loaded by means of 2 in diameter pins. The effects of pin fit for both clearance and interference and of various treatments were investigated. The treatments included side relieving the lug hole, bushing the bore, applying the Sulfinuz process to the lug or overstraining the lug bore. Two conditions of mean stress were investigated, namely, pulsating or repeated tension and 15 tons/in^2 mean tension.—*White, D. J., Paper submitted to The Institution of Mechanical Engineers for written discussion; Proceedings 1968–1969, Vol. 183, Part 1, Number 26.*

Investigation Into Flange Forces in Winch Drums

This paper describes a series of experimental tests which were carried out to investigate the forces acting on a winch drum during multi-layer rope winding. A new method of measuring flange forces is demonstrated and comparisons are made of test results for different rope constructions, rope tensions and spooling. To aid winch designers and operators, flange design curves are suggested which may be used to predict the failure limits of winch drums.—*Bellamy, N. W. and Phillips, B. D. A., Paper submitted to the Institution of Mechanical Engineers for written discussion; Proceedings 1968–1969, Vol. 183, Part 1, Number 27.*

Thermal Effects in Slider Bearings

For the particular operating and boundary conditions assumed, thermal distortion rather than variation of the lubricant properties is the cause of hydrodynamic lubrication of initially parallel, radially grooved, thrust bearings. Numerical solutions are presented to show the effect of relaxing these conditions. The beneficial effect of thermal distortion is proved to hold generally and the importance of proper mixing of the lubricant at the inlet is illustrated. Comparison of numerical solutions with existing experimental observations shows only a qualitative agreement.—*Hahn, E. J. and Kettleborough, C. F., Paper submitted to The Institution of Mechanical Engineers for written discussion; Proceedings 1968–1969, Vol. 183, Part 1, Number 31.*

Ore/Oil Carrier

Mitsubishi Heavy Industries recently delivered the 76 285 dwt ore/oil carrier *Volga Maru* to Sanko Kisen Kaisha. Main features of the vessel are minimized wave making resistance by the use of the Mitsubishi Bow; use of high performance cargo oil heating devices; oil bilge separator for engine room bilge and oily water separator for tank cleaning water, for sea pollution prevention; large exhaust gas economizer for fuel oil economy and for driving the turbo-generator during navigation.—*Mitsubishi Heavy Industries, October 1969, M. H. I. Review, No. 14, p. 29.*

Metallurgical Characteristics of Tough High-Strength Welds

A number of filler metals have been developed to provide tough weldments in the yield strength range of 80 000 to 120 00 lb/in^2. Little has been published about their metallurgical characteristics and why they have such a remarkable combination of strength and toughness. Two such alloys, both low-carbon Mn-Ni-Mo filler metals, were considered. These filler metals produce weld metals having nominal yield strengths of 90 000 and 110 000 lb/in^2. Since the properties of welds are affected by the initial solidification cooling rates and the heat from subsequent passes, studies were made of the initial weld deposit and the changes caused by the heat of subsequent weld passes.—*Lyttle, J. E., Dorschu, K. E. and Fragetta, W. A., Welding Journal, November 1969, Vol. 48, pp. 493s–498s.*

Effect of Welding Conditions on Cooling Rate and Hardness in the Heat-affected Zone

Relationships have been established between the arc welding parameters and the weld size and cooling rate for the continuous-electrode processes over a wide range of travel speeds. The cross-sectional area of the fused metal is related to the product of the effective heat and the cube root of the travel speed; the cooling rate is inversely proportional to the heat input, as predicted by the classical heat flow theory. The effect of travel speed and indentation load on the measured hardness in the heat-affected zone of low-alloy steel has also been investigated.—*Bradstreet, B. J., Welding Journal, November 1969, Vol. 48, pp. 499s–504s.*

Lateral Vibrations of a Rotating Shaft in a Viscous Fluid

A rigid rotating cylindrical shaft is vibrating along a diameter in a viscous fluid. Two different cases are investigated through the method of inner and outer expansions. The case of small amplitude vibrations is characterized by the diffusion of vorticity. The coupling of rotation with vibration introduces a small force, of both inviscid and viscous origins, perpendicular to the direction of oscillations. Rotation affects both the drag and normal force. The steady torque is increased due to the introduction of a steady secondary rotary flow.—*Chang-Yi Wang, Transactions of the ASME, Journal of Applied Mechanics, December 1969, Vol. 36, pp. 682–686.*

Failure Analysis of Lift Pad Studs for the Recovery of Objects from the Ocean

The report describes the application of failure analysis to a naval problem regarding the recovery of underwater objects. A lifting pad is attached to the object to be recovered by four studs which are explosively driven through undersized pad eyes into the submerged structure. Experimental trials by the U.S. Naval Ordnance Laboratory using a shock resistant tool for the studs resulted in breaking stresses of the order of 30 000 lb/in^2, far short of the tensile strength of 290 000 lb/in^2.—*Smith, H. L., Kies, J. A. and Romine, H. E., Naval Research Laboratory, Washington, D.C., August 1968; Report NRL-MR-1913, Marine Technology Society Journal, Sept./Oct. 1969, Vol. 3, p. 81.*

End Effect Bending Stresses in Cables

An analysis of bending stresses in flexible cables has been carried out. It has been found that stresses which arise due to a fixity at the boundaries or other points of discontinuity, decay in an exponential manner from such boundaries, similar to the edge effect solutions of shell theory. Such a phenomenon makes it possible to analyse a finite cable of sufficient length using solutions which are applicable only to infinite or very long cables.—*DeRuntz, J. A., Transactions of the ASME, Journal of Applied Mechanics, December 1969, Vol. 36, pp. 750–756.*

The Role of Non-destructive Testing Under the Sea

After noting the great technological advance in undersea capabilities which has taken place in the last five years, the author discusses present and potential uses of non-destructive testing on submersible vessels and other undersea projects. Problems of environment, pressures and other undersea hazards are considered.—*Hellier, C. J., Materials Evaluation, December 1969, Vol. 27, pp. 254–258.*

Opposed Piston Compression Machine for Preflame Reaction Studies

Mixtures of fuel and air ignite spontaneously if exposed to sufficiently high temperatures. Such ignition is not instantaneous and there is always a delay, which is of outstanding importance in cases in which spontaneous delay plays a part. This paper describes the development of an experimental technique which allows these studies to be extended to much higher pressures than those accessible in glass vessels, while retaining many of the advantages of laboratory vessel experiments.—*Affleck, W. S. and Thomas, A., Paper submitted to The Institution of Mechanical Engineers for written discussion, Proceedings 1968/69, Vol. 183, Part 1, Number 19.*

Bending-Bending Mode of a Rotating Tapered-Twisted Turbomachine Blade Including Rotatory Inertia and Shear Deformation

This paper presents the effects on natural frequency and mode shape of the inclusion of terms that are present in the general equations of motion to describe phenomena associated with rotary inertia and shear deformation. The coupling that exists between the flexural and torsional vibrations is not considered. Carnegie's formulation of the Lagrange Equations of motion is used and the set of field equations solved using Myklestad's adaptation of the Holzer method.—*Krupka, R. M. and Baumanis, A. M., Transactions of the ASME, Journal of Engineering for Industry, November 1969, Vol. 91, pp. 1017–1024.*

Investigation of Steady and Unsteady Flow Through a Napier Turboblower Turbine Under Conditions of Full and Partial Admission

This paper presents the results of tests on an axial-flow turbine and describes how they are obtained, in steady and unsteady flow. An analysis of turbine-test results obtained under the unsteady operating conditions is then given. It is shown that over a limited range of cyclic operation the mass flow and power output may be predicted by assuming that the turbine operates instantaneously as it would under steady-flow conditions and integrating over the engine cycle.—*Craig, H. R. M., Edwards, K. J., Horlock, J. H., Janota, M., Shaw, R. and Woods, W. A. Paper submitted to The Institution of Mechanical Engineers for written discussion; Proceedings 1968–1969, Vol. 183, Part 1, Number 30.*

Automatic Continuous Measurement of Sulphur Trioxide in Flue Gases

A new fully automatic instrument has been made for operation in environments remote from the control and read-out position. It consists of a) a sampling system and chemical analyser, specific for volatile hexavalent sulphur compounds yielding sulphate ions in aqueous solution, and b) power supply, control alarm, and registration facilities. The two are linked together by cables of any practical length. The principle of operation is the absorption of sulphur trioxide in an aqueous solution of isopropanol, reaction of the sulphate ions with solid barium chloranilate and the photometric determination of the acid chloranilate ion.—*Jackson, P. J., Langdon, W. E. and Reynolds, P. J., Journal of The Institute of Fuel, January 1970, Vol. 43, pp. 10–17.*

Tapered Spool Controller for Pressurized Oil Film Bearings

The first part of this paper is concerned with the description of the different methods commonly used in compensating pressurized oil film bearings. The load-oil film thickness relationship for different methods of compensaton and the power dissipated in bearings are given for the sake of comparison. The second part of this paper is designed to suggest another method of compensation, the theoretical analysis, the design equations of which have been presented.—*Morsi, S. A., Paper submitted to The Institution of Mechanical Engineers for written discussion; 1969, paper P21/70.*

Bag Sampler for Collecting Thirty Tons of Deep-Ocean Water

Water samples of almost any size can be obtained if the sample is pumped rather than lifted aboard a vessel. Although this may be done with a very long hose, it is quicker and in some cases easier to collect the water first in a low-strength, light-weight bag and to bring it near the surface for pumping. The bag—empty and collapsed—is lowered by ship's winch to the sampling depth, where a pressure sensitive release opens. Upward travel of a large, attached funnel causes the sampler to fill. The sampler then closes itself.—*Schink, D. R. and Anderson, M. C., Marine Technology Society Journal, Sept./Oct. 1969, Vol. 3, pp. 49–58.*

Gas Turbine Ship Callaghan's First Two Years of Operation

The authors have endeavoured to answer two basic questions: 1) how has this ship measured up, thus far, to her designers' expectations? 2) What lessons have been learned from the ship's operating experience which would benefit future designers, builders, and operators of similar ships? The paper focuses only on those aspects of *Callaghan* which are peculiar to her alone as a gas turbine propelled, roll-on/roll-off cargo ship.—*Zeien, C., Smith, H. F. and Hirst, F. W., Paper presented at the Annual Meeting of The Society of Naval Architects and Marine Engineers, 12–14 November 1969; Paper No. 10.*

Sonodiver Dive Pattern Analysis

Sonodiver, an unmanned deep submergence research probe, employs a velocity-regulated buoyancy control system which enables it to hover at a predetermined depth. A theoretical analysis of its motion was undertaken with the objective of rapidly achieving a condition suitable for obtaining acoustical data, i.e. a state of which Sonodiver's vertical velocity remains less than 0·1 ft/s, and the frequency of any up and down motion is kept below 0·5 cycles/min.—*Lambert, D. R., Marine Technology Society Journal, Sept./Oct. 1969, Vol. 3, pp. 17–29.*

Patent Specifications

Steering System for Marine Vessels

The vessel shown in Fig. 1 is fitted with a box keel (11) of the keel (12), with a forward extension (13) that unites in a smooth curve at (14), with a stem (15) depending from the bow (16), whereby a large area (17) is encompassed in which a bow rudder (18) is set. The rudder (18) is mounted on a rudder stock (19) extending vertically up into the bow (16).

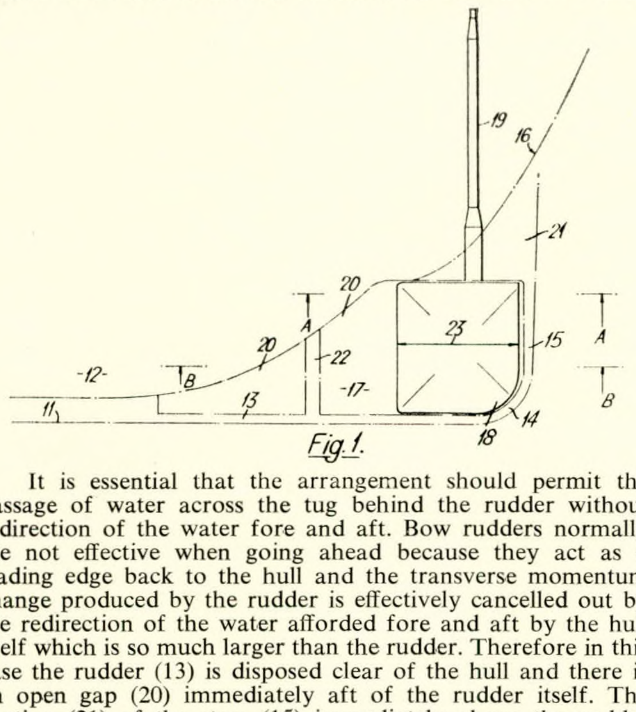


Fig. 1.

It is essential that the arrangement should permit the passage of water across the tug behind the rudder without redirection of the water fore and aft. Bow rudders normally are not effective when going ahead because they act as a leading edge back to the hull and the transverse momentum change produced by the rudder is effectively cancelled out by the redirection of the water afforded fore and aft by the hull itself which is so much larger than the rudder. Therefore in this case the rudder (13) is disposed clear of the hull and there is an open gap (20) immediately aft of the rudder itself. The portion (21) of the stem (15) immediately above the rudder is formed as a plate stem with no gap between it and the bow line, but aft of the rudder the area encompassed by the keel extension (13) and stem (15) is left open, with the exception of a vertical strengthening limb (22), to form the gap (20). When the rudder is put over, the water which has been directed across the vessel by the rudder is allowed to proceed completely clear of the bow of the tug and therefore the transverse momentum change producing a sideways force is allowed to develop undisturbed with substantially no influence from the hull proper. The length of the gap fore and aft should be at least of the order of the chord length (23) of the rudder itself.

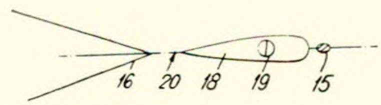


Fig. 2.

Plan view on the line A - A of Fig.1

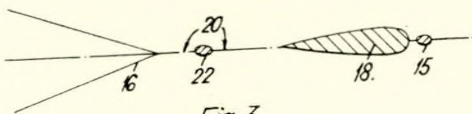


Fig. 3.

Plan view on the line B - B of Fig.1

Tip plates may be fitted to the rudder to prevent the spillage of water over the top and bottom. This is not an essential feature but is helpful in operation; however, with suitably shaped bows the water at the top of the rudder has a considerable downward component as it follows the shape of the hull and this vertical component may be made use of to prevent spillage and effectively becomes the hydrodynamic equivalent of a tip plate.

The rudder is balanced so that it has a normal degree of balance when going ahead and is overbalanced when going astern. As the speed of the tug is lower going astern than going ahead, it is clear that it is possible to balance the torque requirements on the steering gear so that they are approximately equal ahead and astern.

A trailing edge of fish tail configuration in plan cross-section may be used for the bow rudder under certain conditions to increase the lift coefficient although it is realized that this may appreciably increase steering gear torque when going ahead. However, in this particular configuration it will be understood that such increased torque can be counterbalanced by moving the rudder stock aft. This has the subsidiary advantage that the torque on the steering gear astern is reduced and hence the two effects are mutually beneficial. *—British Patent No. 1 170 297 issued to Hydroconics Ltd. Complete specification published 12th November 1969.*

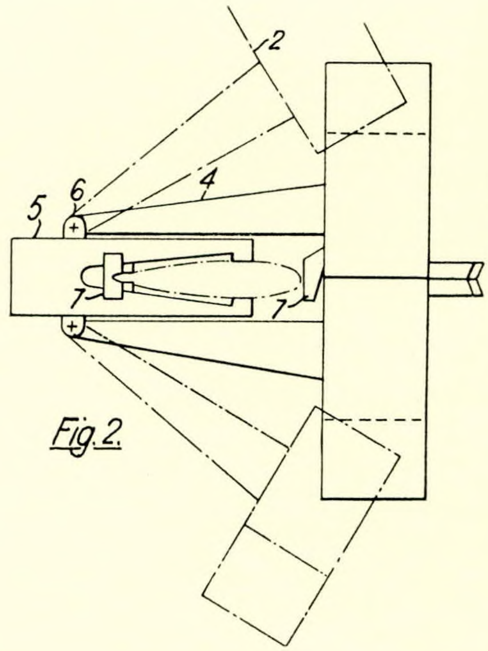


Fig. 2.

Device for Carrying out Engine Tests on Ships

This invention is based upon the problem of providing a device for carrying out standing engine tests on ships which is not substantially larger than the propeller diameter and in which the propeller is braked so that the engine drive of the ship can run under conditions corresponding to free travel.

According to the invention this is achieved by the fact that a divided capsule, the diameter of which is not substantially larger than the diameter of the propeller and not longer than the free space in the propeller well of the ship, is so detachably secured to the ship that the propeller can rotate freely within the capsule.

Fig. 1 shows the lateral elevation, Fig. 2 the plan view and Fig. 3 shows the front view of the device. The divided capsule consists of the front wall (1), the rear wall (2) and the cylindrically shaped periphery (3). The capsule is divided in the direction of the vertical diameter. A joint arm (4) which is articulatedly attached to the guide body (5) by the two joint bearings (6) is secured to each of the two capsule halves.

The guide body (5) is made forwardly forked so that the device is guided on the rudder of the ship. Adjustable or replaceable support elements (7) are provided on the capsule and on the guide body (5) which fix the position of the device in relation to the ship. Floodable and blowable float elements (8) are arranged on the capsule. The device is lashed to the ship by lines (9).

Water can flow into the interior of the capsule through an opening with slide valve (10). The air pipe (11) with shut-off elements renders possible the supply of air. Water or air-water mixture can be conducted away to the exterior through

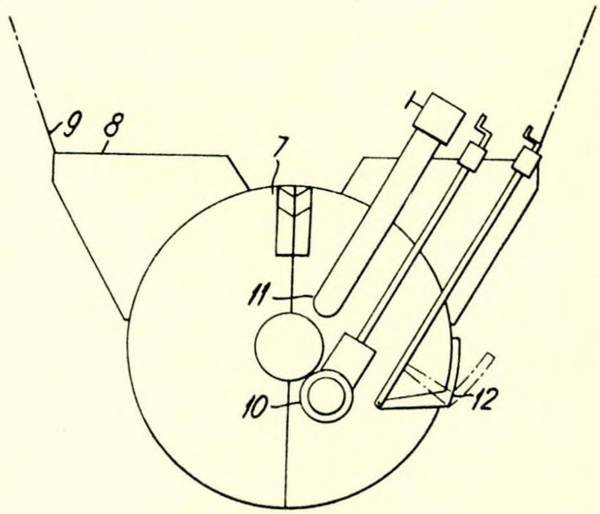


Fig. 3.

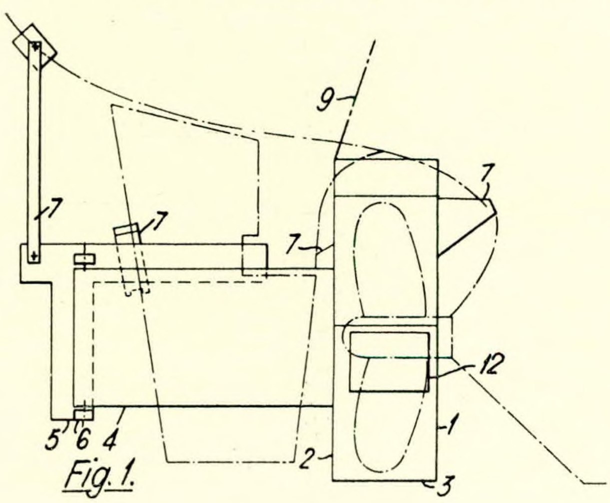


Fig. 1.

an opening with closure flap (12) in the periphery (3).

The fitting of the device is effected by floating out from the rear with pivoted-out capsule halves. When the intended position is reached the capsule halves are pivoted against one another and locked with one another. Lashing by means of lines (9) is then effected.

After the starting up of the drive engine of the ship, the power loading is regulated in that in the case of excessive braking the entering air mixes with the rotating water due to opening of the air pipe (11) and effects a reduction of the propeller torque. An increase of the brake effect can be achieved by opening of the slide valve (10) and the flap (12). In this case propeller and capsule act like a rotary pump. *—British Patent No. 1 173 262 issued to VEB Warnowwerft Warnemunde. Complete specification published 3rd December 1969.*

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